Considerations for the Design and Control of Pulsatile Rotary Total Artificial Hearts

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Abstract
Heart disease remains the leading cause of death in the developed world. The scarcity of donor hearts and limited efficacy of drug therapy in patients with end-stage heart failure establish ventricular assist devices (VADs) and total artificial hearts (TAHs) for blood circulation support as a clinical necessity. Due to their small size and superior durability, rotary blood pumps (RBPs) have almost entirely replaced earlier positive displacement VADs, while the only available TAH remains the pneumatically actuated SynCardia TAH. The development of rotary TAHs is underway and promises to open up new long-term treatment options in the future; however, there is ongoing controversy with respect to the attenuated-pulse or nonpulsatile waveforms observed with RBP support, which is fuelled by the increasing occurrence of adverse events such as gastrointestinal bleeding in rotary VAD patients. While in particular rotary TAHs operated at constant speed provide pulseless (or attenuated-pulse), continuous flow perfusion, rapid impeller speed modulation to modulate the pump outflow may be a viable approach to artificially induce pulsatile arterial pressure waveforms. However, previous attempts failed to restore physiologic levels of pulsatility, which is attributed to challenges in the design and control of a device capable to generate similar pulse amplitudes and rates of change of pressure ($dP/dt$) as the native heart. Therefore, to work towards this objective, the primary aim of this PhD project was to investigate and identify favourable design characteristics and control strategies for rapid RBP speed modulation.

First, factors limiting device performance with speed modulation in praxis were derived theoretically and evaluated on the example of the BiVACOR TAH, where the motor drive was identified as the major source of power loss during rapid impeller acceleration. It was further indicated, that flat pressure head-flow (HQ) curves of the pump may improve the device performance. Secondly, a motor geometry analysis was performed using the finite element method (FEM) to evaluate axial flux motor designs with respect to their suitability for RBP speed modulation. The focus of the analysis was on geometries which may cater for favourable hydraulic geometries and with the objective to reduce axial attractive forces and rotor inertia and increasing the efficiency. A methodology to design a motor for desired characteristics was outlined and the required slot depth and permanent magnet thickness and
resulting rotor inertia to compensate for performance deterioration due to increased inner stator radii and gap lengths were determined.

Subsequently, speed control strategies were evaluated with the BiVACOR device in an in vitro study. The finding that the speed profile significantly influences the overall device efficiency and its pulsatile outflow led to the development of a model framework to numerically optimise rotary TAH speed profiles to maximise the $dP/dt$ and/or surplus haemodynamic energy ($SHE$) generated by the device. Based on the results, a potential control strategy for pulsatile rotary TAH was outlined, and the performance envelope in terms of maximum achievable pulsatility was explored on the example of the HeartMate II. It was found that pulsatility approaching physiologic levels with $dP/dt > 400$ mmHg/s can be generated with RBP; similar results were achieved in a preliminary in vivo study with the BiVACOR TAH applying optimised speed waveforms. However, rapid speed modulation may be implemented at the expense of a substantial increase in power consumption, thus device optimisation for the specific application may be required to implement a viable long-term pulsatile RBP operating mode.

The numerical framework was then finally applied to investigate the influence of hydraulic and motor characteristics on the ability of a device to generate pulsatility. Six different pumps corresponding to RBP ranging from centrifugal pumps to axial flow pumps were modelled and compared in the numerical model. The modelling approached was based on specific speed as a design variable determining the impeller type and steepness of the HQ characteristics of the pumps. It was found that the maximum pulsatility generated by pumps decreased with increasing specific speed (increasing steepness of the HQ curve), which substantiated the hypothesis that a flat HQ-curve is beneficial for rapid speed modulation. Lastly, the relative influences of motor model parameters and hydraulic efficiency on the pump were investigated, showing that the motor torque constant and hydraulic efficiency were the most influential characteristics with respect to the device performance, indicating that motor design changes to optimise maximum torque and efficiency may be worthwhile even when the sacrifice of a slightly increased rotor inertia must be made to facilitate those changes.
Statement of Originality

This work has not previously been submitted for a degree or diploma in any university. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made in the thesis itself.

Matthias Kleinheyer
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<th>Description</th>
<th>Unit</th>
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<tr>
<td>MCS</td>
<td>mechanical circulatory support</td>
<td>–</td>
</tr>
<tr>
<td>VAD</td>
<td>ventricular assist device</td>
<td>–</td>
</tr>
<tr>
<td>LVAD</td>
<td>left ventricular assist device</td>
<td>–</td>
</tr>
<tr>
<td>RVAD</td>
<td>right ventricular assist device</td>
<td>–</td>
</tr>
<tr>
<td>BiVAD</td>
<td>Bi-ventricular assist device</td>
<td>–</td>
</tr>
<tr>
<td>TAH</td>
<td>Total artificial heart</td>
<td>–</td>
</tr>
<tr>
<td>RBP</td>
<td>Rotary blood pump</td>
<td>–</td>
</tr>
<tr>
<td>FEM</td>
<td>Finite element method</td>
<td>–</td>
</tr>
<tr>
<td>(dP/dt)</td>
<td>rate of change of arterial pressure</td>
<td>(mmHg/s)</td>
</tr>
<tr>
<td>SHE</td>
<td>Surplus haemodynamic energy</td>
<td>(ergs/cm^3)</td>
</tr>
<tr>
<td>PP</td>
<td>Pulse pressure</td>
<td>(mmHg)</td>
</tr>
<tr>
<td>HF</td>
<td>Heart failure</td>
<td>–</td>
</tr>
<tr>
<td>LV</td>
<td>Left ventricle</td>
<td>–</td>
</tr>
<tr>
<td>IABP</td>
<td>Intra-aortic balloon pump</td>
<td>–</td>
</tr>
<tr>
<td>BTD</td>
<td>Bridge to decision</td>
<td>–</td>
</tr>
<tr>
<td>BTR</td>
<td>Bridge to recovery</td>
<td>–</td>
</tr>
<tr>
<td>BTT</td>
<td>Bridge to transplantation</td>
<td>–</td>
</tr>
<tr>
<td>DT</td>
<td>Destination therapy</td>
<td>–</td>
</tr>
<tr>
<td>RV</td>
<td>Right ventricle</td>
<td>–</td>
</tr>
<tr>
<td>TET</td>
<td>Transcutaneous energy transfer</td>
<td>–</td>
</tr>
<tr>
<td>VZP</td>
<td>Virtual zero power</td>
<td>–</td>
</tr>
<tr>
<td>MCL</td>
<td>Mock circulatory loop</td>
<td>–</td>
</tr>
<tr>
<td>SVR</td>
<td>Systemic vascular resistance</td>
<td>(dyn \cdot s/cm^5)</td>
</tr>
<tr>
<td>PVR</td>
<td>Pulmonary vascular resistance</td>
<td>(dyn \cdot s/cm^5)</td>
</tr>
<tr>
<td>WK</td>
<td>Windkessel</td>
<td>–</td>
</tr>
<tr>
<td>PI</td>
<td>Pulsatility index</td>
<td>–</td>
</tr>
<tr>
<td>(P_I_G)</td>
<td>Pulsatility index by Gosling</td>
<td>–</td>
</tr>
<tr>
<td>EEP</td>
<td>Energy equivalent pressure</td>
<td>(mmHg)</td>
</tr>
<tr>
<td>MAP</td>
<td>Mean arterial (aortic) pressure</td>
<td>(mmHg)</td>
</tr>
<tr>
<td>PPI</td>
<td>Pulse power index</td>
<td>(1/s^2)</td>
</tr>
<tr>
<td>AVO</td>
<td>Aortic valve opening</td>
<td>–</td>
</tr>
<tr>
<td>GI</td>
<td>Gastro-intestinal</td>
<td>–</td>
</tr>
<tr>
<td>AVM</td>
<td>Arteriovenous malformation</td>
<td>–</td>
</tr>
<tr>
<td>vWF</td>
<td>Von Willebrand factor</td>
<td>–</td>
</tr>
<tr>
<td>SMC</td>
<td>Soft magnetic composite</td>
<td>–</td>
</tr>
<tr>
<td>NeFeB</td>
<td>Neodymium-iron-boron</td>
<td>–</td>
</tr>
<tr>
<td>RFPM</td>
<td>Radial flux permanent magnet machine</td>
<td>–</td>
</tr>
<tr>
<td>AFPM</td>
<td>Axial flux permanent magnet machine</td>
<td>–</td>
</tr>
<tr>
<td>PM</td>
<td>Permanent magnet</td>
<td>–</td>
</tr>
<tr>
<td>MMF</td>
<td>Magneto-motive force</td>
<td>(A – turns)</td>
</tr>
<tr>
<td>(B_r)</td>
<td>Remanence flux density</td>
<td>(T)</td>
</tr>
<tr>
<td>(\mu_0)</td>
<td>Vacuum permeability</td>
<td>(Vs/Am)</td>
</tr>
<tr>
<td>(\mu_r)</td>
<td>Relative permeability</td>
<td>–</td>
</tr>
<tr>
<td>OCP</td>
<td>Optimal control problem</td>
<td>–</td>
</tr>
<tr>
<td>NLP</td>
<td>Nonlinear program</td>
<td>–</td>
</tr>
<tr>
<td>HQ</td>
<td>Pressure head-flow</td>
<td>–</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
</tr>
<tr>
<td>--------</td>
<td>--------------------------------------------------</td>
<td>-------------</td>
</tr>
<tr>
<td>AOC</td>
<td>Aortic compliance</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>PAC</td>
<td>Pulmonary arterial compliance</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>PVC</td>
<td>Pulmonary venous compliance</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>SVC</td>
<td>Systemic venous compliance</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>LA</td>
<td>Left atrium</td>
<td>–</td>
</tr>
<tr>
<td>RA</td>
<td>Right atrium</td>
<td>–</td>
</tr>
<tr>
<td>FTR</td>
<td>Force test rig</td>
<td>–</td>
</tr>
<tr>
<td>MTR</td>
<td>Motor test rig</td>
<td>–</td>
</tr>
<tr>
<td>$H_R$</td>
<td>Right pressure head</td>
<td>mmHg</td>
</tr>
<tr>
<td>$H_L$</td>
<td>Left pressure head</td>
<td>mmHg</td>
</tr>
<tr>
<td>$\eta_{mot}$</td>
<td>Motor efficiency</td>
<td>%</td>
</tr>
<tr>
<td>$T_{Load}$</td>
<td>Load torque</td>
<td>Nm</td>
</tr>
<tr>
<td>$P_{el}$</td>
<td>Electrical power consumption</td>
<td>W</td>
</tr>
<tr>
<td>$P_{mech}$</td>
<td>Mechanical power</td>
<td>W</td>
</tr>
<tr>
<td>$P_{loss}$</td>
<td>Power loss</td>
<td>W</td>
</tr>
<tr>
<td>$\eta_{hyd}$</td>
<td>Hydraulic efficiency</td>
<td>%</td>
</tr>
<tr>
<td>$P_{hyd}$</td>
<td>Hydraulic power</td>
<td>W</td>
</tr>
<tr>
<td>$T_{hyd}$</td>
<td>Hydraulic torque</td>
<td>Nm</td>
</tr>
<tr>
<td>CF</td>
<td>Continuous flow</td>
<td>–</td>
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<tr>
<td>PF</td>
<td>Pulsatile flow</td>
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</tr>
<tr>
<td>$F_x$</td>
<td>Axial force</td>
<td>N</td>
</tr>
<tr>
<td>$F_r$</td>
<td>Radial force</td>
<td>N</td>
</tr>
<tr>
<td>$T_{el}$</td>
<td>Electromagnetic torque</td>
<td>Nm</td>
</tr>
<tr>
<td>$n$</td>
<td>Impeller speed</td>
<td>rpm</td>
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<tr>
<td>BLDC</td>
<td>Brushless DC</td>
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</tr>
<tr>
<td>RM</td>
<td>Reference motor</td>
<td>–</td>
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<tr>
<td>BG</td>
<td>Baseline geometry</td>
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<tr>
<td>$N_{coil}$</td>
<td>Coil turn number</td>
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</tr>
<tr>
<td>$r_i$</td>
<td>Inner stator radius</td>
<td>mm</td>
</tr>
<tr>
<td>$r_o$</td>
<td>Outer stator radius</td>
<td>mm</td>
</tr>
<tr>
<td>$r_{i,core}$</td>
<td>Inner core radius</td>
<td>mm</td>
</tr>
<tr>
<td>$r_{o,core}$</td>
<td>Outer core radius</td>
<td>mm</td>
</tr>
<tr>
<td>$l_{gap}$</td>
<td>Gap length</td>
<td>mm</td>
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<tr>
<td>$d_s$</td>
<td>Slot depth</td>
<td>mm</td>
</tr>
<tr>
<td>$w_s$</td>
<td>Slot width</td>
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</tr>
<tr>
<td>$h_{pm}$</td>
<td>Permanent magnet thickness</td>
<td>mm</td>
</tr>
<tr>
<td>$a_{pm}$</td>
<td>Permanent magnet angle</td>
<td>deg</td>
</tr>
<tr>
<td>$h_{ys}$</td>
<td>Stator yoke thickness</td>
<td>mm</td>
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<tr>
<td>$h_{yr}$</td>
<td>Rotor yoke thickness</td>
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</tr>
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<td>$\Delta r_{slice,k}$</td>
<td>Thickness of k-th slice</td>
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</tr>
<tr>
<td>$r_{i,k}$</td>
<td>Inner radius of k-th slice</td>
<td>mm</td>
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<td>$r_{o,k}$</td>
<td>Outer radius of k-th slice</td>
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<tr>
<td>$r_{sim,k}$</td>
<td>Radius of simulated 2D-surface for k-th slice</td>
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<tr>
<td>$F_{\phi,k}$</td>
<td>Tangential force component of 2D-surface</td>
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<td>$F_{z,k}$</td>
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<tr>
<td>$F_{ax,k}$</td>
<td>Axial force contribution of k-th slice</td>
<td>N</td>
</tr>
<tr>
<td>$T_{el,k}$</td>
<td>Electromagnetic torque contribution of k-th slice</td>
<td>Nm</td>
</tr>
<tr>
<td>$f_{edge,B}$</td>
<td>Edge factor for correction of magnetic flux</td>
<td>–</td>
</tr>
<tr>
<td>$f_{edge,F}$</td>
<td>Edge factor for correction of axial force</td>
<td>–</td>
</tr>
</tbody>
</table>
\( f_{edge,T} \)  
- Edge factor for correction of torque

RCS  
- Radial cross section

\( B_z \)  
- Axial component of air gap flux density

\( P_{cu} \)  
- Copper losses

\( P_{core} \)  
- Core losses

\( P_{core} \)  
- Specific core losses

\( k_{ca} \)  
- Slot fill factor

\( p \)  
- Pole pair number

\( \rho_{Fe} \)  
- Iron density \( kg/cm^3 \)

\( \rho_{NdFeB} \)  
- Permanent magnet material density \( kg/cm^3 \)

FOC  
- Field oriented control

TC  
- Trapezoidal control

\( r_{rel} \)  
- Relative radius between inner and outer core radii

\( I_{mot} \)  
- Motor current

\( J_{rot} \)  
- Rotor inertia

\( \Theta_{el} \)  
- Rotor angle (electrical)

\( \omega \)  
- Radian frequency

LUT  
- Lookup-table

PI  
- Proportional-integral

\( \eta_{APat} \)  
- Power specific rate of change of pressure \( mmHg/s/W \)

\( \eta_{SHE} \)  
- Power specific surplus haemodynamic energy \( erg/cm^3/W \)

\( \eta_{PPI} \)  
- Power specific pulse power index \( 1/s^2/W \)

\( P_{Mean} \)  
- Mean motor power consumption \( W \)

CVS  
- Cardiovascular system

\( L_{cs} \)  
- Systemic characteristic inertance \( mmHg \cdot s^2/mL \)

\( R_{sa} \)  
- Systemic characteristic resistance \( mmHg \cdot s/mL \)

\( R_{ra} \)  
- Right atrial resistance \( mmHg \cdot s/mL \)

\( c_{ao} \)  
- Aortic compliance \( mL/mmHg \)

\( c_{sv} \)  
- Systemic venous compliance \( mL/mmHg \)

\( c_{ra} \)  
- Right atrial compliance \( mL/mmHg \)

\( L_{cp} \)  
- Pulmonary characteristic inertance \( mmHg \cdot s^2/mL \)

\( R_{cp} \)  
- Pulmonary characteristic resistance \( mmHg \cdot s/mL \)

\( R_{pa} \)  
- Pulmonary arterial resistance \( mmHg \cdot s/mL \)

\( R_{la} \)  
- Left atrial resistance \( mmHg \cdot s/mL \)

\( c_{pa} \)  
- Pulmonary arterial compliance \( mL/mmHg \)

\( c_{pv} \)  
- Pulmonary venous compliance \( mL/mmHg \)

\( c_{la} \)  
- Left atrial compliance \( mL/mmHg \)

\( Q_s \)  
- Systemic flow rate \( L/min \)

\( Q_{sm} \)  
- Mean systemic flow rate \( L/min \)

\( Q_{ics} \)  
- aortic inertia flow \( L/min \)

\( Q_{ra} \)  
- Right atrial flow rate \( L/min \)

\( P_{ao} \)  
- Aortic pressure \( mmHg \)

\( P_{cao} \)  
- aortic compliance pressure \( mmHg \)

\( P_{sv} \)  
- Central systemic venous pressure \( mmHg \)

\( P_{ra} \)  
- Right atrial pressure \( mmHg \)

\( Q_p \)  
- Pulmonary flow rate \( L/min \)

\( Q_{tcp} \)  
- pulmonary arterial inertance flow \( L/min \)

\( Q_{la} \)  
- Left atrial flow rate \( L/min \)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{pa}$</td>
<td>Pulmonary arterial pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{cpa}$</td>
<td>Pulmonary arterial compliance pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{pv}$</td>
<td>Pulmonary venous pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_a$</td>
<td>Left atrial pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{mc}$</td>
<td>Mean circulatory pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$L_{RBP}$</td>
<td>Fluid inerter in pump and cannulae</td>
<td>mmHg · s²/mL</td>
</tr>
<tr>
<td>$K_t$</td>
<td>Motor torque constant</td>
<td>Nm/A rms</td>
</tr>
<tr>
<td>$R_{mot}$</td>
<td>Motor phase resistance</td>
<td>Ω</td>
</tr>
<tr>
<td>$P_{in}$</td>
<td>Pump inlet pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{out}$</td>
<td>Pump outlet pressure</td>
<td>mmHg</td>
</tr>
<tr>
<td>$H$</td>
<td>Pump pressure head</td>
<td>mmHg</td>
</tr>
<tr>
<td>$Q_L$</td>
<td>Shunt (leakage) flow through BiVACOR TAH</td>
<td>L/min</td>
</tr>
<tr>
<td>$J(x(t))$</td>
<td>OCP objective function</td>
<td>–</td>
</tr>
<tr>
<td>$x(t)$</td>
<td>Vector of state variables</td>
<td>–</td>
</tr>
<tr>
<td>$u(t)$</td>
<td>Vector of control variables</td>
<td>–</td>
</tr>
<tr>
<td>$f(x(t), u(t), t)$</td>
<td>State functions</td>
<td>–</td>
</tr>
<tr>
<td>$g(x(t))$</td>
<td>Constraint function for the mean flow rate</td>
<td>–</td>
</tr>
<tr>
<td>$h(x(t), u(t))$</td>
<td>Constraint function for the maximum power consumption</td>
<td>–</td>
</tr>
<tr>
<td>$\psi(x(t))$</td>
<td>Constraint function for the mean circulatory pressure</td>
<td>–</td>
</tr>
<tr>
<td>$x_i(t)$</td>
<td>I-th state variable</td>
<td>–</td>
</tr>
<tr>
<td>$u_i(t)$</td>
<td>I-th control variable</td>
<td>–</td>
</tr>
<tr>
<td>$t_i$</td>
<td>Start (initial) time</td>
<td>s</td>
</tr>
<tr>
<td>$t_F$</td>
<td>End (final) time</td>
<td>s</td>
</tr>
<tr>
<td>$M$</td>
<td>Number of time segments</td>
<td>–</td>
</tr>
<tr>
<td>$N$</td>
<td>Number of collocation nodes</td>
<td>–</td>
</tr>
<tr>
<td>$\bar{x}$</td>
<td>NLP variable vector</td>
<td>–</td>
</tr>
<tr>
<td>$n_v$</td>
<td>Number of NLP variables</td>
<td>–</td>
</tr>
<tr>
<td>$n_x$</td>
<td>Number of state variables</td>
<td>–</td>
</tr>
<tr>
<td>$n_u$</td>
<td>Number of control variables</td>
<td>–</td>
</tr>
<tr>
<td>$\xi_{ik}$</td>
<td>Defect of i-th state variable at k-th collocation node</td>
<td>–</td>
</tr>
<tr>
<td>$J_{NLP}(\bar{x})$</td>
<td>NLP objective function</td>
<td>–</td>
</tr>
<tr>
<td>$w_{obj}$</td>
<td>Objective function weighting factor</td>
<td>%</td>
</tr>
<tr>
<td>$g_{NLP}(\bar{x})$</td>
<td>NLP constraint function for the mean flow rate</td>
<td>–</td>
</tr>
<tr>
<td>$h_{NLP}(\bar{x})$</td>
<td>NLP constraint function for the maximum power consumption</td>
<td>–</td>
</tr>
<tr>
<td>$\psi_{NLP}(x_0)$</td>
<td>NLP constraint function for the mean circulatory pressure</td>
<td>–</td>
</tr>
<tr>
<td>$\lambda_{f1}$</td>
<td>NLP objective function scaling factor</td>
<td>1/(ergs/cm³)</td>
</tr>
<tr>
<td>$\lambda_{f2}$</td>
<td>NLP objective function scaling factor</td>
<td>1/(mmHg/s)</td>
</tr>
<tr>
<td>$\alpha_{dp/dt}$</td>
<td>Soft maximum parameter for NLP objective function</td>
<td>–</td>
</tr>
<tr>
<td>$h_k$</td>
<td>Time step size</td>
<td>s</td>
</tr>
<tr>
<td>$S_\alpha$</td>
<td>Soft maximum function</td>
<td>–</td>
</tr>
<tr>
<td>PID</td>
<td>Proportional integral derivative</td>
<td>–</td>
</tr>
<tr>
<td>PR</td>
<td>Pulse rate</td>
<td>–</td>
</tr>
<tr>
<td>RMS</td>
<td>Root mean square</td>
<td>–</td>
</tr>
<tr>
<td>$p^*_e$</td>
<td>Maximum power consumption constraint</td>
<td>W</td>
</tr>
<tr>
<td>$Q^*_BEP$</td>
<td>Mean flow rate constraint</td>
<td>L/min</td>
</tr>
<tr>
<td>$n_q$</td>
<td>Specific speed</td>
<td>–</td>
</tr>
<tr>
<td>$Q_{BBEP}$</td>
<td>Flow rate at best efficiency</td>
<td>L/min</td>
</tr>
<tr>
<td>$H_{BEP}$</td>
<td>Pressure head at best efficiency</td>
<td>mmHg</td>
</tr>
<tr>
<td>$h_{BEP}$</td>
<td>Head at best efficiency in</td>
<td>m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Unit</td>
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<td>--------</td>
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</tr>
<tr>
<td>(\delta_s)</td>
<td>Specific diameter</td>
<td>—</td>
</tr>
<tr>
<td>(n_{qi})</td>
<td>Pump with specific speed (n_q = i)</td>
<td>—</td>
</tr>
<tr>
<td>(k_{s,T})</td>
<td>Motor torque constant sensitivity factor</td>
<td>%</td>
</tr>
<tr>
<td>(k_{s,J})</td>
<td>Rotor inertia sensitivity factor</td>
<td>%</td>
</tr>
<tr>
<td>(k_{s,R})</td>
<td>Motor resistance sensitivity factor</td>
<td>%</td>
</tr>
<tr>
<td>(k_{s,\eta})</td>
<td>Hydraulic efficiency sensitivity factor</td>
<td>%</td>
</tr>
<tr>
<td>(S_{SHE}(k_{s,i}))</td>
<td>Sensitivity of surplus haemodynamic energy to (k_{s,i})</td>
<td>(\text{ergs/cm}^3/%)</td>
</tr>
<tr>
<td>(S_{dP/dt}(k_{s,i}))</td>
<td>Sensitivity of rate of change of pressure to (k_{s,i})</td>
<td>(\text{mmHg/s/%})</td>
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**Contributions by others to the thesis**

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Luca Robertson: assisted in the initial evaluation of the implemented 2D-FEM axial flux motor model.

Professor Geoff Tansley and Dr Daniel Timms: proof read and provided critical feedback on the thesis in order to clarify aims, objectives and conclusions of this thesis.
Publications during Candidature

Peer-reviewed Journal Articles


Conference Abstracts


**Other**

Acknowledgement of Papers included in this Thesis

Included in this thesis is a published paper in Chapter 4 which is co-authored with other researchers. My contribution to the co-authored paper is outlined at the front of the relevant chapter. The bibliographic details for this paper including all authors, are:


(Where a paper(s) has been published or accepted for publication, you must also include a statement regarding the copyright status of the paper(s).

**Chapter 4**: the paper ‘Rapid Speed Modulation of a Rotary Total Artificial Heart Impeller’ has been published by the journal ‘Artificial Organs’ and a license to include the paper into my thesis was obtained on 20/08/2017 with the license number 4173340082785.

Appropriate acknowledgements of those who contributed to the research but did not qualify as authors are included in each paper.

(Signed) _________________________________ (Date) 21 August 2017
Matthias Kleinheyer

(Countersigned) ___________________________ (Date) 22-8-17
Supervisor: Professor Geoffrey Tansley
1 Introduction

Cardiovascular disease (CVD) remains the leading cause of death worldwide. In 2015, 17.7 million people died from CVDs, 7.4 million of which died due to coronary heart disease [1]. The main risk factors associated with cardiovascular disease are smoking, hypercholesterolaemia, diabetes and hypertension, which can mainly be ascribed to the modern lifestyle in the western industrial nations. Frequently, cardiovascular disease descends into heart failure requiring lifesaving intervention by way of surgery when medical management is no longer effective. The gold standard treatment is heart transplantation, however, the disparity between availability and need of donor organs is steadily increasing, resulting in a high mortality rate amongst these patients. Mechanical circulatory support (MCS) systems are often the only prospect for sustaining a patient whilst waiting for transplantation (bridge to transplantation), or as a permanent therapy option (destination therapy). Over the last decades, several approaches in development of implantable blood pumps have been made. These pumps can generally be separated into two subcategories:

- Ventricular Assist Devices (VADs) are blood pumps which are used to partially or completely support the function of the failing left (LVAD), right (RVAD), or both ventricles (BiVAD) respectively. LVADs pump blood from the left atrium or ventricle to the aorta in parallel to the heart. Similarly, RVADs cannulate the right ventricle or atrium to move blood into the pulmonary artery. VADs are normally implanted as bridge-to-transplant, destination therapy, or in milder cases of heart failure as bridge-to-recovery devices.

- In severe cases of left, right or bi-ventricular failure or fibrillation, a Total Artificial Heart (TAH) is implanted as a full replacement of the failing heart. TAHs typically comprise two pumps to support the systemic and pulmonary circulation. When a TAH is implanted, the native ventricles and heart valves are removed, and the device is connected to the atrial remnants and the aorta and pulmonary artery to autonomously assume the functionality of the native heart.

Ventricular assist devices of the first generation are characterized by their pulsatile nature, as these devices employ a positive displacement pumping mechanism driven by pneumatic or electric actuators. Flexible diaphragms are used to fill and empty an artificial pump chamber,
while mechanical valves allow blood flow in only one direction. However, there are critical issues associated with these pumps with respect to durability and cyclic fatigue of the flexible mechanical components. Furthermore, the devices are usually large in size, thus cannot be implanted in small women or children. To improve reliability, and to minimize the pump size, interest developed in leveraging the advantages of axial flow or centrifugal rotary blood pump (RBP) technology, utilising a rapidly spinning impeller to continuously propel blood. Due to the improved mechanical properties and smaller size of the resulting pumps, RBPs have almost entirely replaced positive displacement VADs [2]. In contrast, the development of rotary TAHs did not progress equivalently. The only commercially available device is the pneumatically operated SynCardia TAH (SynCardia Systems LLC, Tucson, AZ, USA), which relies on a paracorporeal pneumatic compressor and flexible diaphragms within the implanted pump units. To leverage the advantages of RBP technology, researchers strive to employ RBP for TAHs with currently developed devices [3–6], which, due to their potential durability, promise to provide new long-term therapy options in the future.

While the improved clinical outcomes with rotary VADs are evident [2], uncertainty exists about the chronic effects of RBP support on the mammalian physiology. Unlike earlier positive displacement pumps, RBPs do not inherently generate pulsatile outflow. When rotary VADs interact with the failing native heart, the outflow typically exhibits some level of attenuated pulsatility. However, adverse events such as gastrointestinal bleeding, arteriovenous malformations, aortic insufficiency, or pump thrombosis are increasingly observed with rotary VAD support [7], fuelling an ongoing debate about potential detrimental physiologic effects of RBP support and the necessity of the cardiovascular pulse.

The discussion is particularly significant with respect to rotary TAHs, which are implanted to replace the native ventricles and thus establish entirely pulseless blood circulation. Researchers have therefore attempted to artificially simulate the cardiovascular pulse by means of cyclic speed modulation of RBP impellers, where periodic alternations of low and high pump speeds are utilised to modulate the outflow in a pulsatile fashion [8]. The evaluated applications of such speed modulation modes included practical aspects such as improvement of the pump washout to avoid stasis and thrombus formation [9], improved control over ventricular unloading with rotary VADs [10,11], as well as attempts to fully restore physiologic pulsatility in a TAH setting [12,13], which may allow to maintain normal
baroreflex sensitivity and microcirculatory perfusion [14,15]. Conversely, concerns have been raised with respect to increased blood shear stress due to elevated operating speeds and acceleration profiles during rapid speed modulation, which may increase the risk of severe blood damage [16–18]. In addition, the increased motor power required to accelerate the impeller may reduce battery life and lead to blood or tissue damage by contact to potential hot spots due to excessive heat dissipation. However, to date, no report of the successful recreation of physiologic pressure and flow patterns with a rotary TAH is available in the literature. Ongoing controversy exists with respect to the inconsistent classification and quantification of the level of pulsatility generated in previous studies, failure to generate a physiologic pulse, and consequently the ability of currently available RBP designs to generate physiologic pulsatility [8,18–20]. A device design that allows to recreate a truly physiologic pulse would greatly enhance the significance of future research studies evaluating the biological effects of the presence or absence of the cardiovascular pulse [8]. However, challenges include the selection of an appropriate speed control profile and the development of motor and hydraulic systems exhibiting performance characteristics and efficiencies suitable to achieve this objective. Therefore, this project is focused on this engineering challenge and work towards the development of a suitable rotary blood pump design and control strategy, which allow mimicking of the characteristics of native pulsatile cardiac output. While the investigations presented here are focused on rotary TAHs, the presented findings and design guidelines may be similarly transferred to rotary VADs.

1.1 Research Aim

The aim of the thesis is to investigate and identify favourable design characteristics and control strategies for the motor and hydraulics of a RBP, which, when combined, may allow the future development of a RBP, which can autonomously and efficiently generate physiologic arterial pulsatility. The key objectives to achieve this aim include:

- Investigate axial flux motor geometries to improve motor performance and cater for desired hydraulic characteristics that improve device performance with rapid speed modulation.
- Evaluate the influence of different speed profile shapes on the generated haemodynamic pulsatility and required motor power consumption.
• Develop a numerical framework to investigate and optimize speed profiles for rapid speed modulation of different rotary blood pump (RBP) types in a total artificial heart (TAH) setting.
• Investigate the influence of RBP design characteristics on the ability of the device to generate pulsatile haemodynamics.

1.2 Thesis Structure and Contributions

This section provides a brief outline of the structure of the following chapters of the thesis, and highlights the contributions made to the knowledge in the research field of rotary blood pumps. In chapter 1, following this introduction, an in-depth review of the literature, comprising: aspects of the human cardiovascular physiology; the role of the arterial pulse; and the relevant engineering disciplines and tools applied in the following research chapters is presented.

Chapter 2 introduces the dynamic processes and loss mechanisms associated with rapid speed modulation of rotary blood pumps, to establish a fundamental understanding of the engineering challenges related to rapid RBP speed modulation. Subsequently, the initial experience with rapid speed modulation of an early version of the BiVACOR TAH is presented to identify limitations and areas of potential improvement. As such, chapter 2 builds the foundation for the research studies presented in the subsequent chapters.

In chapter 3, axial flux motor designs, which may improve device performance with rapid speed modulation were investigated with respect to motor efficiency, axial attractive forces, and rotor inertia. Based on the design of the BiVACOR TAH, the focus was on geometric constraints, which may serve to improve the hydraulic characteristics in favour of pulsatile haemodynamic outflow, specifically large air gaps and inner stator radii to allow for large impeller vanes and pump inflow diameters. A motor design methodology was derived, wherein the investigations were based on a two-dimensional finite element method (FEM) simulation model of the motor geometry, which was developed based on a previously reported quasi-three-dimensional simulation approach [21,22]. The model was extended to allow its application to motor geometries with large gaps and concomitant large magnetic flux leakage, and to calculate axial attractive forces and electromagnetic torque within an
acceptably small error margin compared to 3D-FEM simulations and prototype measurements.

In chapter 4 a detailed in-vitro study to investigate the effect of speed control profiles on the generated haemodynamic outflow and the required motor power consumption is reported, and trade-offs amongst speed profile characteristics with respect to different metrics of pulsatility were highlighted, leading to the conclusion that optimal speed profiles may exist to optimize pulsatility with respect to these metrics.

Based on these findings, a parameterized mathematical model and numerical framework were developed, which served to numerically optimize the speed profile under specified boundary conditions. The model and methodology, which are reported in chapter 0, were successfully validated for a simplified model of a single RBP supporting the systemic circulation, and for a model of the BiVACOR TAH interacting with both the systemic and pulmonary circulation. Further, examples are presented of optimized speed profiles to maximise the generated surplus haemodynamic energy (SHE) or the maximum rate of change of arterial pressure (dP/dt) for different scenarios; and the ability of the BiVACOR TAH to generate haemodynamic waveforms approaching physiologic levels of pulsatility with speed profiles obtained from the numerical simulation environment was demonstrated in a preliminary in vivo experiment. The findings may guide the future development of a control strategy for pulsatile RBP.

The developed model is finally applied to evaluate the effect of hydraulic and motor characteristics on the maximum pulsatility, which can be generated by a RBP. The investigation presented in chapter 6 comprises the evaluation of the hydraulic pressure sensitivity and efficiency, as well as motor characteristics such as the torque constant, stator resistance and rotor inertia. The relative influence of the different parameters was then compared to provide guidance for design decisions with respect to the hydraulic and motor design of a pulsatile RBP in the future.
1.3 Background and Literature Review

1.3.1 The Cardiovascular System

The cardiovascular system (Figure 1-1) is a sophisticated closed-loop system, which is divided into two subsystems, the pulmonary and systemic circulation. It contains approximately five to six litres of blood, which is pumped by the heart and circulates in an average time of one minute through the body. The circulatory system is responsible for exchange, distribution and transport of blood gases, nutrients, metabolites and hormones to musculature and organs.

Figure 1-1 – The human heart and cardiovascular system [23]. Licensed under CC BY 3.0 via Wikimedia Commons.
The pulmonary circuit is supplied by the right heart, which pumps blood through the bifurcated pulmonary artery to the left and right lungs. Here the blood is oxygenated, and subsequently passes through the pulmonary veins to reach the left heart, from where it is pumped into the ascending aorta and is distributed in the body. The systemic circulation comprises of a branched arterial system leading to the capillary bed in musculature and organs in the lower and upper body, where oxygen and nutrients are released, and a venous system, through which the blood is returned to the right heart.

1.3.1.1 Human Heart and Cardiac Cycle

The human heart is hollow-muscle and is the central organ of the circulatory system. In a healthy condition, the heart is approximately the size of its owner’s fist and pumps blood in a periodic rhythm with an approximate rate between 60 and 100 beats per minute in an adult, supplying the body with pulsatile blood flow. The heart consists of two pumps, the left and right heart, each divided in an atrial and a ventricular chamber, which are separated by an atrioventricular valve to avoid regurgitant blood flow. The outflow tracts of the ventricular chambers lead to the systemic (left) and pulmonary (right) circulation, which are separated from the ventricles by the two semilunar valves, the aortic and pulmonary valves respectively. The cardiac cycle is divided into systole, the ventricular contraction phase, and diastole, the ventricular relaxation phase (Figure 1-2). At the onset of systole, the ventricular muscle begins to contract, thus increasing intraventricular pressure and causing the atrioventricular valves to close.

The pressure increases isovolumetrically, until the ventricular pressure exceeds the arterial pressure and the semilunar valves open. At this stage, arterial and ventricular pressure are almost equal and the ongoing contraction forces the ventricular blood volumes to eject into the aorta and pulmonary artery respectively. Simultaneously, venous blood return from the pulmonary and systemic veins fills the left and right atrial chambers. Towards the end of the

Figure 1-2 – The cardiac cycle. Figure adapted from [24].
contraction phase the pressures begin to decrease until the aortic and pulmonary valves close. The ventricular pressures further decrease in an isovolumetric relaxation, until it falls below the respective atrial pressure and thus causes the atrioventricular valve to open and allows blood flow from the atria to the ventricles during diastole. The ventricular filling is assisted by minor contractions of the atria, which are known as the atrial systole.

The heart circulates blood an average flow rate of approximately $4 - 6 \, L/min$ at rest. However, several autoregulatory mechanism exist to adapt the cardiac output according to demand, which changes for example in response to the patient posture or level of activity. The change in blood flow is caused by an increase in beat rate and ejected volume per beat. Depending on changes of venous blood return from the body (preload), the myocardium autonomously adjusts its force of contraction during systole (contractility) to eject the increased end-diastolic ventricular volume, leading to an increase of the cardiac output. This mechanism is known as the Frank-Starling law of the heart. The change of ventricular output in response to changes in preload is illustrated in the cardiac output curves for the left and right ventricle in Figure 1-3A. The total cardiac output is further influenced by sympathetic and parasympathetic nervous stimulation, which lead to autonomous adjustments of vascular tone, heart rate and ventricular contractility. Changes of the ventricular output curves for different levels of sympathetic nerve activity are shown in Figure 1-3B.

![Figure 1-3](image.png)

Figure 1-3 – Ventricular output curves for (A) the left and right ventricle and (B) changes in sensitivity in response to changes in stimulation of the sympathetic nervous system or with a failing heart. Figures adapted from [24].
1.3.1.2 Heart Failure

Heart failure (HF) is a summarizing term for conditions in which the heart is unable to provide adequate blood flow, limiting the oxygen delivery to vital organs and muscular tissue [25]. While HF can be caused by a variety of diseases, including damaged heart valves, hypertension, infection, or diabetes, the most common risk factor to develop HF is ischaemic heart disease [24,26]. Ischaemic heart disease is a condition in which the coronary arteries are partially occluded, preventing adequate perfusion of the cardiac muscle and limiting its ability to contract. The most frequent cause of decreased coronary perfusion is the development of atherosclerotic plaque[24]. Symptoms of chronic heart failure include reduced exercise capacity, fatigue, and shortness of breath, while the condition typically gradually develops over time. In severe cases, local thrombus formation can occur, causing a full occlusion of a coronary artery and preventing blood flow to part of the heart. The consequent damage to the heart muscle – myocardial infarction – manifests itself through immediate development of acute heart failure resulting in loss of functionality in the affected tissue.

Most HF conditions affect the larger left ventricle (LV), while right heart failure often occurs secondary to LV failure. The loss of ventricular contractility causes a decrease in stroke volume and a compensatory rise in end-diastolic pressure and volume to activate the Frank-Starling mechanism [25]. Frequent concomitant conditions are ventricular dilation and pulmonary oedema, caused by blood backing up in the LV and left atrium.

1.3.2 Mechanical Circulatory Support

While early-stage and mild heart failure conditions are typically treated with drug therapy, severe cases of ventricular failure often require surgical intervention. The gold standard therapy for medically refractory heart failure is heart transplantation, however, there is a steadily increasing disparity between the stable number of available donor organs and increasing organ demand [27], which intensifies the need for alternative treatment options. In the USA an approximate of 2,200 donor hearts are available for as many as 100,000 patients with advanced-stage HF [28]. In Australia alone there are more than 100 people waiting for a heart transplant at any time [29]. A practical alternative to heart transplantation are mechanical circulatory support (MCS) devices. There are three major types of MCS
devices frequently implanted, namely intra-aortic balloon pumps (IABP), ventricular assist devices (VAD), and total artificial hearts (TAH).

1.3.2.1 Intra-Aortic Balloon Pumps

Intra-aortic balloon pumps (IABP) are devices operated in counterpulsation to the heart. A polyethylene balloon connected to a double-lumen catheter is inserted into the descending thoracic aorta, where it is inflated during ventricular diastole and deflated during systole [30]. The blood displacement during diastole results in a higher diastolic pressure and is associated with a potential increase in coronary flow, improving myocardial oxygen supply. At the same time, systolic pressure is lowered, decreasing LV load and thus myocardial oxygen demand. However, IABP operation requires a minimum of ventricular function, making them unsuitable for patients with severe ventricular failure [31].

1.3.2.2 Ventricular Assist Devices

Ventricular assist devices have become an increasingly important therapy option for advanced-stage heart failure [2]. These mechanical blood pumps are used to mechanically support the left (LVAD), right (RVAD), or both ventricles (BiVAD) of the failing heart. During VAD therapy, the native heart remains in place, while the pump assists and unloads the ventricle to restore sufficient cardiac output by pumping blood from the left atrium or ventricle to the aorta (LVAD) or from the right atrium or ventricle to the pulmonary artery (RVAD). Figure 1-4 illustrates the human heart supported by two HVAD pumps (HeartWare Inc., Framingham, MA, USA). The pump inlets are integrated into the devices and cannulate the left and right ventricle respectively. VADs can be implemented for different short or long-term therapy options, including bridge to decision (BTD), bridge to recovery (BTR), bridge to transplantation (BTT), or, for severely ill patients unsuitable or ineligible to receive a heart transplant, destination therapy (DT) [33]. In the years 2008 – 2014 the majority of patients registered in
the INTERMACS database underwent BTT strategy, closely followed by DT, however less than 1% of patients were implanted with the prospect of myocardial recovery [2].

The journey of mechanical heart pumps began in the early 1960s, where the first implant of a the Liotta TAH was performed by Denton A. Cooley [34]. Since these early days and specifically over the recent years, technology for VADs and TAHs has rapidly advanced, while different principles of functionality have been implemented. VADs and TAHs can be grouped by their inherent pumping mechanism, however a more detailed differentiation is given by grouping the devices into three generations. Early MCS pumps of the first generation are built as positive displacement-type pumps (Figure 1-5).

These devices use pneumatic or electrical drivers to operate air sacs, flexible diaphragms or pusher plates and displace the adjacent blood volume in a similar way as the native heart [33]. Mechanical heart valves are used to ensure unidirectional blood flow at the pump inlets and outlets. First generation devices can restore physiologic perfusion; however, they are plagued by several disadvantages. The required valves and flexible membranes undergo intense mechanical wear and ageing, limiting device lifetime and bearing the risk of providing thrombus formation sites in fatigue cracks [33,36]. Additionally, the large size required to incorporate a physiologic displacement volume often limits their applicability to large patients, leaving out smaller women and adolescents. Due to the large size, some of the devices are designed to be used extracorporeally, as for example the Thoratec/St. Jude PVAD (St. Jude Medical, Saint Paul, MN, USA) or the EXCOR (Berlin Heart GmbH, Berlin,
Germany) [33,37]. An example for a fully implantable first generation VAD is the HeartMate XVE (St. Jude Medical) [28].

![St. Jude HeartMate II](image1)

**A) St. Jude HeartMate II**

![HeartWare HVAD](image2)

**B) HeartWare HVAD**

Figure 1-6 – Cross sections of rotary blood pumps of the second and third device generation: (A) The St. Jude HeartMate II and (B) HeartWare HVAD, figures adapted from (A) [38] and (B) [39].

Newer MCS devices of the second and third generation aim to eliminate these limitations by leveraging the mechanical advantages of continuous flow rotary blood pump (RBP) technology. RBPs utilize a rapidly spinning impeller within a pump housing to continuously pump blood and thus eliminate the requirement for mechanical valves and allow for a significantly reduced building volume. Most state-of-the-art devices use axial-flow, mixed-flow, or centrifugal-flow pump technology [39,40].

The second and third device generation are mostly distinguished by the type of impeller suspension. Second generation RBPs, typically axial-flow pumps, use contact bearings such as shaft ball bearings or blood-immersed pivot or cone bearings made from ceramic or ruby (Figure 1-6 A) [41]. Shaft bearings have the inherent disadvantage of the requirement for
shaft seals, which isolate the bearing from the bloodstream, however they are subject to abrasive failure and clotting due to blood stagnation and trauma. If blood-immersed mechanical bearings are used, no seal is required, but the bearing itself exposes the blood to mechanical trauma [36]. Therefore, the latest devices of the third generation operate contactless utilizing hydrodynamic bearings or active/passive magnetic levitation to eliminate mechanical wear and enhance durability [28]. The suspension technologies can further be combined to a hybrid bearing, as in the design of the HeartWare HVAD centrifugal RBP, which is equipped with a passive magnetic radial bearing and an hydrodynamic axial thrust bearing (Figure 1-6 B) [39]. Most LVADs are connected to an inflow cannula in the left ventricular apex, while the outlet is connected to a woven Teflon, Polyester, or Dacron outflow graft, which is anastomosed to the ascending or descending aorta [42–45]. Depending on the device used and the implant strategy other possible cannulation sites are available; for example the left atrium for the inflow and the femoral artery for the outflow cannula [28]. A frequently encountered problem not limited to, but due to the smaller size of the right heart particularly prominent in RVAD support, is obstruction of the inflow cannula with myocardial tissue [46]. To alleviate this issue, the effective inflow cannula length can be shortened, or the right atrium can be cannulated instead of the right ventricle [3,37,47]. No rotary RVAD is currently clinically available, however, LVADs rea typically implanted to assist the right heart and operated at lower speed, or with a banded outflow to accommodate for the lower pressures required for the pulmonary circulation [48].

Rotary VADs are typically operated at almost constant speeds. Hence, due to their working principle they generate continuous outflow, which does not inherently exhibit pulsatility. However, the pump flow rate is closely related to the rotational speed of the impeller and differential pressure head between the inlet and outlet. The relationship between the three quantities is characterized by a device-specific HQ-characteristic (HQ-curve), which relates pressure head (H) and outflow (Q) at a constant speed (n) (Figure 1-7). The operating point on the curve is determined by the external load and pressure conditions in the circuitry of the pump. In the extreme load case, where the flow path is completely occluded, the flow rate reduces to zero and the pump generates the speed-dependent shutoff pressure head, which is the intercept of the HQ characteristic with the y-axis. Similarly, when the shutoff pressure head is enforced by the external circulatory pressures, the flow rate reduces to zero even
without occlusion of the flow path. Vice versa, when the pressure head is low, the flow rate increases.

When a RBP is connected to the native ventricle and the aorta, the pressure head is widely influenced by the ventricular contraction. The pulsation of the pump inlet pressure therefore generates a flow pattern with an attenuated pulse even when the pump is operated at a constant speed. The variation of the flow rate is dependent on the negative gradient (or pressure sensitivity) of the HQ-characteristic and the inertia of the fluid within the pump and cannulae. The speed of the device is typically set by a clinician and remains substantially constant to ensure adequate blood flow to the circulation.

![During the cardiac cycle, the operating Point moves in proximity of the constant speed HQ-curve](image)

Figure 1-7 – Example of measured pressure head – flow characteristics (HQ-curves) for various speeds of a centrifugal RBP. H, pressure head; Q, flow rate. The black arrows indicate the movement of the operating point of a VAD at constant speed \( n \) during the cardiac cycle.

However, the constant speed operation of LVADs bears the risk of over-pumping, leading to suction events and ventricular collapse at low or negative intraventricular pressures [49]. Automatic pump speed control to prevent suction remains challenging, as no implantable pressure sensor featuring long-term stability without the requirement for regular recalibration is clinically available [50]. However, algorithms to detect suction events and physiological control systems to autonomously adjust pump outflow based on the observation of
measurable device parameters have been subject to extensive research [51–53], while only few of the control methods have been implemented in commercial LVAD systems [54–56].

1.3.2.3 Total Artificial Hearts

Univentricular support with left ventricular assist devices has become a widely-used therapy for heart failure patients. However, LVAD support is insufficient for patients with severe biventricular failure. Right ventricular (RV) failure occurs in up to 40% of LVAD patients [57], who may then require subsequent RV support to maintain pressure balance between pulmonary and systemic circulations, to avoid complications such as anasarca, ascites, and renal and hepatic insufficiency due to right atrial hypertension [58]. While implantation of an additional RVAD is a possible treatment option, BiVAD support is associated with high mortality, clinical challenges with respect to fitting and speed control, and further requires the patient to carry two controllers and power sources [37,59,60].

Total artificial hearts (TAHs) provide an alternative treatment option for patients with severe biventricular failure, intractable ventricular arrhythmia, irreparable ventricular anatomical defects, or problematical anatomical VAD fitting [58,59]. TAHs comprise two pumping mechanisms to support systemic and pulmonary circulation, which can be provided by two independent pumps, or by a single device [3,9,61]. When a TAH is implanted, the native ventricles and heart valves are surgically removed and replaced with the device (or devices), which then autonomously provides blood flow to the pulmonary and systemic circulation. Typically, the atrial remnants are supported with Teflon felt strips and sutured to device-specific Dacron or latex membrane cuffs connecting to the left and right device inlets [58,62,63]. The pump outlets are connected to grafts, which are anastomosed to the aorta and pulmonary artery respectively.

Currently there are only two clinically approved TAH devices: the AbioMed AbioCor and the Syncardia TAH. The AbioCor is a positive-displacement pump comprising two polyurethane pump chambers adjacent to a common electrohydraulic drive [61,64]. A hydraulic fluid is pressurized by a miniature centrifugal pump in the central drive unit and switch by a two-way valve to alternatingly compress flexible sacs to displace the blood volume within the two rigid pump chambers. The device is powered through a wireless transcutaneous energy transfer (TET) system which inductively charges an implantable
battery. Pressure balance between the pulmonary and systemic circulation is regulated by a unique isolated hydraulic shunt circuit to reduce the filling volume of the right pump chamber with increasing left atrial pressures [58,65]. Therefore, the right pump output is reduced at excessive left atrial pressures enabling the device to autonomously avoid pulmonary congestion. The AbioCor was implanted in 14 humans between 2001 and 2004 with an average survival of five months [61]. However, the limitation to larger men due to the device size and significant clinical complications related to thromboembolic events leading to stroke caused the discontinuation of the device in 2007.

The Syncardia TAH (Figure 1-8) remains the only currently implanted TAH. The device originates from the early development of the Jarvik 7 TAH in 1982 and was implanted in more than 1,600 patients for durations up to more than three years, with a success rate of the bridge-to-transplant therapy between 48% and 79% [58,67]. It consists of two independent blood chambers, each incorporating an inflatable air-sac to pump blood through the unidirectional mechanical valves. The device is powered by an external air compressor through two percutaneous air hoses penetrating the skin. In contrast to the previously used pneumatic driver cart, which immobilized the patient, the newly developed portable Freedom driver was FDA approved for clinical use in 2014 and allows patients to be discharged from the hospital and undertake routine daily activities [68].
In normal operation the pneumatically activated diaphragm system always fully ejects the blood volume in the pump chambers, thus the device exhibits significant sensitivity to preload and essentially mimics the Frank-Starling mechanism [69, 70]. The outcomes with the Syncardia TAH have been promising; however, the device exhibits the typical drawbacks of positive displacement-type pumps, as discussed in section 1.3.2.2. The device is prone to wear and the percutaneous entry sites of the drive air hoses bear the risk of infection. Furthermore, the compressor contained in the pneumatic driver is noisy, and thus limits patient quality of life. Although the device is an integral part of MCS therapy option and it is approved for BTT and DT in patients at risk of imminent death [61, 71], the Syncardia TAH is currently unsuitable for long durations of ten years and beyond.

1.3.2.4 Future Prospects for Rotary Total Artificial Hearts

The clinical outcomes with ventricular assist devices based on RBP technology have been shown to be comparable or superior to those of positive displacement devices [72]. In the field of total artificial hearts this progress has not yet been made, however, the increased survival rates observed with RBP technology in LVAD and BiVAD patients (Figure 1-9) stands to reason, that rotary TAH may follow a similar trend in the future.

In BiVAD implants, the native ventricles remain and provide a diminished autoregulatory response due to the Frank-Starling mechanism of the failing ventricles. When the ventricles are removed and a rotary TAH is implanted, the outflow is truly nonpulsatile and the device relies entirely on its own ability to balance left-hand and right-hand flows in response to changes in vascular tone. The major engineering challenge in the development of rotary TAHs is thus the device requirement to

Figure 1-9 – Actuarial survival of patients registered in the INTERMACS database after primary device implant (June 2006 – December 2013). CF, continuous flow; PF, pulsatile flow. Figure first published in [73].
autonomously establish pressure balance between the pulmonary and systemic circulation, and adjust cardiac output in response to the demands of the body.

Due to the complex device requirements, no continuous flow TAH has found its way to clinical routine, however, several devices are currently under investigation. Frazier, Cohn et al. implanted dual rotary VADs as TAH in a series of experiments undertaken at the Texas Heart Institute (Houston, TX, USA), utilizing different types of RBPs, including the St. Jude Medical HeartMate II, the HeartWare HVAD, and the Jarvik Heart Jarvik 2000 [5,9,74,75]. The devices were operated at near-constant speed with regular adjustments for durations of up to 92 days, exhibiting stable haemodynamics. Following this series of experiments a case report of the first human implant of a continuous flow TAH composed of dual RBPs was published in 2012 [76]. The patient was suffering from end-stage heart failure and severe Amyloidosis, and was supported by the TAH for five weeks, before he succumbed to multiple organ failure due to his underlying disease. Successful balancing of the left and right circulation with the device combination in both the animal experiments and human device implantation through the inherent pressure sensitivity of the RBP was reported [76], however, the autonomous flow changes in response to moderate physical exercise of animals on a treadmill were as low as 2% – 4%. It should further be noted, that vascular resistances during animal experiments were pharmacologically regulated by administration of vasoactive agents and diuretics [9]. Drug administration to the human patient was not discussed.

The risks of imbalanced TAH operation include under-perfusion, venous congestion, pulmonary oedema, and atrial suction. Therefore, not only control of the absolute pump output, but also the relative performance of left and right pumps is required. While the inherent pressure sensitivity of RBPs can assist balancing, typical sensitivities of currently available RBP stay significantly behind those of displacement-type pumps and the native ventricles [77,78], indicating that the autonomous RBP response is insufficient to adequately balance all haemodynamic conditions occurring during routine daily activities, which may require greater flow adjustments.

Newer rotary TAH devices currently under development aim to incorporate additional inherent balancing mechanisms in the design. An example is the Cleveland Clinic CFTAH which is a single-device rotary TAH with two centrifugal impellers mounted to a single
common rotor [4,79,80], and which is hydrodynamically suspended in the radial direction. The axial position is passively restored by permanent magnet forces provided by the surrounding motor assembly. The axial displacement of the rotor is used to adjust the effective cross-sectional area of the left and right impeller volutes inversely to balance hydraulic forces originating from a pressure difference between the left and right circulation. For example, an increased pressure acting on the left impeller moves the rotor towards the right pump. Due to the altered position of the impeller within its volute, the efficiency of the left pump is increased, while the right pump efficiency is decreased. Consequently, the left and right pump flow rates are adjusted to restore physiological balance.

A similar device design is implemented in the BiVACOR TAH, which is under development by BiVACOR Inc. [3,81,82]. The device utilizes active magnetic levitation for contactless suspension of a single rotating impeller disk with centrifugal impellers on either side. The use of an axial-flux brushless-DC motor allows for a significantly reduced device size, while the pressure balance is maintained through passive alteration of the actively controlled axial impeller position. The rotor position is controlled utilizing a virtual zero-power (VZP) algorithm, allowing for minimal power consumption of the magnetic levitation system. A more detailed description of the BiVACOR device is given in section 1.3.5.

Not all challenges in the development of a universal rotary TAH have been addressed thus far; however, the experience reported by Frazier and Cohn shows the promising prospect of RBP technology in TAH designs. Furthermore, preclinical investigations of newly developed devices promise the imminent clinical availability of a rotary TAH. The transition to RBP devices has revolutionized the treatment of left ventricular failure, and it appears as if the same transition in the field of TAHs is only a matter of time.

1.3.3 Characterization and Modelling of the Cardiovascular System

The development of RBPs and corresponding control strategies require extensive testing of the device characteristics and employed algorithms. While in-vivo studies in animal models are valuable and necessary, the number of required animal studies should be minimized. Frequently used alternatives for preclinical testing are given by numerical simulations and hydraulic cardiovascular system simulators – so called mock circulatory loops (MCL) – which aim to replicate the haemodynamic waveforms of the human physiology and evaluate
the device and controller performance under similar conditions as they occur during interaction with the human heart and cardiovascular system.

The precise characterization and understanding of the cardiovascular system and haemodynamic waveforms generated by the native heart is vital to allow adequate modelling and replication of similar waveforms in in-silico and in-vitro studies. The following section outlines typical modelling techniques and their application in preclinical device testing.

1.3.3.1 Haemodynamic Waveforms

The haemodynamic pressure and flow waveforms generated by the native heart are widely affected by the condition of the arterial and venous system. When the heart contracts, the stroke volume ejected into the arterial system can be measured in terms of the volumetric flow rate \( Q \), which is typically specified in \( L/min \). The aortic flow waveform upon aortic valve opening is characterized by a sharp increase to a maximum flow rate of approximately 20 \( L/min \), which then decreases approximately linearly till the end of systole, resulting in a sawtooth-like shape of the flow waveform. A small backflow peak occurs in the instance before the aortic valve closes and prevents further reverse flow. Pressures in the cardiovascular system are typically denoted in millimetres of mercury (\( mmHg \)). The typical aortic pressure waveform exhibits a sharp upstroke followed by a slower rise to the peak pressure. When the aortic valve closes, the aortic pressure shows a small, sharp \( incisura \) – the dicrotic notch – followed by an exponential pressure decline while the ejected volume is distributed in the systemic circulation. A schematic illustration of human aortic flow and pressure waveforms is shown in Figure 1-10 [24].

Typically, systemic arterial pressure at rest varies between approximately 120 \( mmHg \) at the end of systole and 80 \( mmHg \) at the end of diastole. The properties of the viscoelastic vessel walls and the vascular tone determine the waveform and distribution of pressure within the arterial and venous circulation as well as the resulting flow rates in large and small vessels. The behavioural characteristics of the vascular system can be described with a set of characteristics, which are outlined in the following.
1.3.3.2 Characteristics of the Cardiovascular System

The vascular bed is a complex network of smaller and larger vessels distributed over the entire body. The arterial system, specifically the smaller arterioles and capillaries pose a hydraulic resistance, which must be overcome by the heart to sufficiently perfuse the body. Analogous to Ohm’s law, the resistance ($R$) of a rigid pipe segment can be calculated as the quotient of the pressure difference between the entry and exit sites ($\Delta P$) and the flow rate ($Q$) through the segment (Equation (1.1)).

$$R = \frac{\Delta P}{Q}$$  \hspace{1cm} (1.1)

While all perfused blood vessels contribute to vascular resistance, the systemic and pulmonary resistances are often summarized and denoted as Systemic ($SVR$) and Pulmonary Vascular Resistance ($PVR$). $SVR$ and $PVR$ are often calculated simplified with the mean values of aortic pressure ($P_{ao}$), right atrial pressure ($P_{ra}$), pulmonary arterial pressure ($P_{pa}$) and left atrial pressure ($P_{la}$), and systemic ($Q_s$) and pulmonary ($Q_p$) flow rates (Equations (1.2) and (1.3)). The factor 80 converts the resistance to the commonly used unit dyn·s/cm$^5$.

In a normal physiological condition of a human at rest, $SVR$ ($900 - 1400 \text{ dyn} \cdot \text{s/cm}^5$)\textsuperscript{1} is typically significantly higher than $PVR$ ($150 - 250 \text{ dyn} \cdot \text{s/cm}^5$). For this reason, mean $P_{ao}$ is approximately a factor 5 – 6 higher than mean $P_{pa}$ and the left ventricle is stronger and bigger than the right ventricle.

---

\textsuperscript{1}1 \text{ dyn} \cdot \text{s/cm}^5 = 10^5 \text{ Pa} \cdot \text{s/m}^3

---
Vascular resistance can only characterize the cardiovascular system to a limited extent, as it cannot account for effects such as the distensibility of the vessel walls. The vascular compliance ($C$) is a measure of the change in volume ($V$) of a vessel due to a change of the inner pressure resulting in stretch of the vessel walls (Equation (1.4)). The unit of vascular compliance is mL/mmHg. In some publications, the inverse term (the elastance $E$) is used instead of compliance [83].

$$C = \frac{dV}{dP} \approx \frac{\Delta V}{\Delta P} \quad (1.4)$$

The third commonly used characteristic to describe cardiovascular haemodynamics is the inertance ($L$). Inertance describes effects of the mass inertia of the blood and is a measure for the pressure difference required to accelerate the blood volume within a vessel. The relationship of inertance, pressure difference, and the time derivative of the flow rate is given in equation (1.5). The typically used unit for inertance is mmHg·s/mL.

$$\Delta P_L = L \frac{dQ_L}{dt} \quad (1.5)$$

While variations in the vascular resistances in response to changes in patient posture or exercise mostly affect the mean arterial pressure levels, dynamic waveforms are strongly depending on arterial compliance and inertance. As will be discussed in section 1.3.3.3, the influence of both characteristics is strongly dependent on the frequency of the excitation. Inertance is mostly effective in the low frequency components of the haemodynamic waveforms, while compliance affects dampening of higher frequency components. The vascular resistance is mostly located in the small arteries and arterioles, whereas Inertance represents the blood mass in the entire circulation and affects the rate of increase in flow rate during ventricular contraction. The greatest contribution to arterial compliance is found in the larger arteries, specifically the aorta [84]. However, with a contribution of up to 98% of
total vascular compliance, the venous compliance is significantly larger than arterial compliance [85]. Therefore, pressures and flow rates in the venous system exhibit significantly reduced pulsatility (Figure 1-11).

![Figure 1-11](image)

Figure 1-11 – Transition from pulsatile ventricular and arterial pressure waveforms to almost constant pressure in the large veins. Figure adapted from [86].

### 1.3.3.3 The Arterial Windkessel and Vascular Input Impedance

Based on the previously discussed characteristics, the first lumped-parameter model to describe the arterial system as ventricular load impedance – the two-element Windkessel model – was introduced by Otto Frank in 1899 [87]. Due to the similar hydraulic characteristics to the arterial system, the name of the model was derived from compressed air tanks, which were used in fire engines to equalize pressure fluctuations in the water supply. A mathematical analogy of the Windkessel model is given by an electrical equivalent circuit consisting of an electrical resistance and a parallel capacitance, representing the peripheral vascular resistance and arterial compliance respectively. In this model, electrical voltages correspond to pressure differences, while flow rates are represented by electrical currents. The analogies to electrical equivalent circuits typically used in modelling of the cardiovascular system are summarized in Table 1-1. While the two-element Windkessel model is a first approximation of the vascular characteristics, it fails to model arterial properties in frequency ranges above $4 \sim 5 \text{ Hz}$. In the native circulation, during semilunar valve opening, a high peak with a steep slope in the arterial flow rate is seen, therefore higher
frequencies in the Fourier spectrum are notably present and should be considered in a precise characterization of the waveforms. Furthermore, characteristics of the pressure waveform resulting from wave reflections in the branched vascular system cannot be represented by the two-element model.

Table 1-1 – Analogies used in computational models of cardiovascular haemodynamics represented by electrical equivalent circuits.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Electrical analogue</th>
<th>Model component</th>
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<tr>
<td>➤</td>
<td>Electrical current</td>
<td>Volumetric flow rate</td>
</tr>
<tr>
<td>➡</td>
<td>Voltage</td>
<td>Pressure difference</td>
</tr>
<tr>
<td><img src="R" alt="R" /></td>
<td>Electrical resistance</td>
<td>Vascular resistance</td>
</tr>
<tr>
<td><img src="C" alt="C" /></td>
<td>Capacitance</td>
<td>Compliance</td>
</tr>
<tr>
<td><img src="L" alt="L" /></td>
<td>Inductance</td>
<td>Inertance</td>
</tr>
<tr>
<td><img src="H" alt="H" /></td>
<td>Variable voltage source</td>
<td>Pump pressure head</td>
</tr>
<tr>
<td><img src="F" alt="F" /></td>
<td>Constant voltage source</td>
<td>Mean filling pressure</td>
</tr>
</tbody>
</table>

To account for a broader frequency spectrum, the Windkessel model was progressively enhanced and extended [84,88] with the characteristic impedance ($Z_c$) of the ascending aorta. Firstly, a resistance in series to the two-element Windkessel was introduced to represent the characteristic impedance of the ascending aorta (Figure 1-12, left) [88]. The model was later refined with an inertance element in parallel to the characteristic resistance (Figure 1-12, right).

**A) Three-element Windkessel**  

**B) Four-element Windkessel**

Figure 1-12 – Windkessel models with (A) three elements and (B) four elements. $R_c$, characteristic resistance; $R_p$, peripheral resistance; $C$, Compliance; $L$, inertance. Figures adapted from [84].

These additional components can be explained when the arterial characteristics are evaluated in the frequency-domain in terms of the vascular input impedance ($Z_{Art}$).
The input impedance is the frequency dependent relationship between the arterial flow and pressure waveforms. It is calculated as the quotient of the fast Fourier transforms (FFT) of the pressure and flow signals (Equation (1.6)) \cite{89,90}.

\[ Z_{\text{Art}} = \frac{\text{FFT}(P)}{\text{FFT}(Q)} \quad (1.6) \]

Figure 1-13 shows a comparison of the measured frequency dependent input impedance against the analytically derived input impedance of a fitted two-element, three-element, and four-element Windkessel model. The impedance modulus of the two-element model asymptotically declines to zero at high frequencies, while a residual high-frequency component remains in the measured impedance (indicated by the arrow in Figure 1-13). This characteristic was adapted through addition of the characteristic resistance in both the three- and four-element models. The additional parallel ineritance in the four-element model allows for a reduction of the impedance modulus in the lower frequency ranges (2 − 5 Hz), resulting in a more accurate representation of the arterial system by the model \cite{84}.

The parameters corresponding to measured data can be identified with numerical fitting in the frequency domain \cite{93,94}. The improved four-element Windkessel model is a lumped-parameter representation of the arterial system. It is unsuitable to predict wave reflections and pressure distributions in the branched arterial system including the capillary bed. However, the model allows for sufficiently accurate modelling of arterial haemodynamics as observed in the ascending aorta, thus it is a valuable tool to model the cardiovascular system.

Figure 1-13 – Phase and magnitude of the frequency-dependent vascular input impedance of the cardiovascular system compared to that of the three-element and four-element Windkessel (WK) models. Figure adapted from \cite{91,92}.
1.3.3.4 Extended Lumped-Parameter Models of the Cardiovascular System

The vascular Windkessel model is widely used in extended numerical models of the circulatory system. Several simulation models to investigate interactions between RBP, the vascular system and the human heart have been developed, most of which are based on extended electrical equivalent circuits. An example is a numerical simulation model developed by Vollkron et al., which includes the pulmonary and systemic vascular systems, ventricular and atrial contractility, a set of dynamic equations for a RBP, and haemodynamic effects of the cannulation [95]. RBP and cannulae were described by differential equations, while the circulatory system was divided into sub-branches and modelled with a lumped parameter electrical equivalent circuit. Heart contractions were modelled by means of compliance elements with a time-varying capacity, following a predefined function.

Lim et al. developed a similar model (Figure 1-14), which was parameter-optimized and focused on the modelling and close fitting of dynamic pressure and flow waveforms [83]. Baseline data has been derived from experimental in vivo studies in healthy animals. Model parameters have been optimized in a least squares sense in a range of operating points and thus allow a reasonably accurate prediction of the system response to a given input scenario. Similar to the model published by Vollkron, a time-varying elastance was used to simulate ventricular contractility.

![Figure 1-14 – Example of a cardiovascular system model. R, resistance; E, elastance; L, inertance; D, diode (valve); Q, flow rate; P, pressure. Figure first published in [83].](image-url)
Most numerical cardiovascular system models were developed to simulate the behaviour of the native heart within the circulatory system, or evaluate the interaction of ventricular contractions and mechanical circulatory support devices during the development of MCS devices and physiologic control strategies.

Both examples chosen here introduced lumped parameter models using electrical equivalent circuits to represent the arterial impedance. In both models the representations of the arterial system were slightly different than the four-element Windkessel model. Lim et al. implemented a series connection of a resistor and inductor instead of the characteristic impedance proposed by Stergiopulos et al.[84]; similarly, Vollkron placed a resistor in series with the arterial compliance instead of the resistor in parallel to the inertance element. While a similar time- and frequency-response is modelled despite the differences in component placement, it should be noted that the corresponding values needed to achieve similar system behaviour may differ.

A comparison of three different impedance models was performed by Segers, who evaluated the performance of the three-element Windkessel model (WK3) and four-element Windkessel models with parallel (WK4-p) and series (WK4-s) connection of characteristic resistance and inertance against a large cohort of healthy individuals [94]. In 80% of the individuals, all three models yielded very similar performance. The best goodness of fit was achieved with a WK4-s model; however, the corresponding predicted component values were physically impossible in 12% of all test subjects. In the remaining 20% of test subjects the WK3 and WK4-s models showed a poor quality of fit, whereas the WK4-p showed adequate behaviour throughout. The authors concluded, that no universal recommendation as to which model can deliver the most adequate frequency response could be made based on the evaluated data.

For the investigation of operation modes of a total artificial heart, the native ventricular functionality, atrial contractions, and the heart valves can be omitted, thus complexity is significantly reduced. Numerical lumped-parameter models for rotary TAH interaction have not been subject to extensive research, however, the approach can be considered similar to the modelling of the full cardiovascular system. An example of such a model has been presented by Nestler, who used a numerical model to evaluate the autonomous pressure
balancing through inherent pressure sensitivity of RBP operated at constant speed [96]. However, while the applied model was used to investigate continuous flow TAH operation, no device-derived pulsatility through rapid speed modulation was evaluated.

1.3.3.5 Cardiovascular Simulators – Mock Circulatory Loops

Mock circulatory loops (MCL) are mechanical hydraulic representations of the cardiovascular system, which have been extensively researched and developed by several research groups over the past decades [97–100]. Typically, the devices are based on similar lumped parameter models as the previously discussed numerical simulation models, whereas the lumped components of the vasculature are imitated with mechanical apparatus. Different techniques were employed to simulate vascular resistance and compliance elements. Pantalos et al. developed a MCL of the systemic and coronary circulation [97]. Open-cell foam, which was compressed with a sealed piston within a sealed chamber, was employed to create the desired amount of vascular resistance. Compliance elements were realised with spring loaded pistons on rolling diaphragms, whereas compression of the springs was varied to adjust the compliance to the required values for arterial and venous compliances [101]. Based on the four-element Windkessel model, the mechanical elements were adjusted to generate a similar vascular input impedance as the human arterial system. Ventricular contractions were simulated with a mock ventricle, comprising of a polyurethane sac inside a pressurized chamber, which was connected to an external pneumatic driver.

The MCL developed by Timms et al. comprised systemic and pulmonary circulations [98]. Electro-pneumatically controlled pinch valves were used to simulate vascular resistance. Pulmonary and systemic vascular compliance was imitated with sealed-to-atmosphere compressed-air Windkessel vessels, constructed from vertically mounted PVC pipes, which were sealed with pipe test plugs. A specified amount of air was entrapped within the compliance chambers above the fluid level to simulate vascular compliance. The vertical position of the test plugs was adjustable to vary the desired amount of compliance. Mechanical check-valves were incorporated to mimic the native heart valves. Ventricular function was simulated with clear PVC pipes similar to the compliance chambers, which were sealed with an end-cap and connected to an electronically controlled compressed air source to allow inflow of air during systole and venting during diastole, following a time-varying pressure. An example of a MCL is shown in Figure 1-15. Continuous refinements of
the described models have been made over time leading to adequate representations of the cardiovascular system. However, the mechanical nature of MCLs, specifically the utilized rigid materials and check valves, were observed to introduce some unphysiologic effects such as ringing in the pressure waveforms, which should be carefully considered, when the precise evaluation of haemodynamic waveform shapes is desired. Further, the typically employed control algorithms cannot fully recreate all haemodynamic and physiologic autoregulatory functions of the native cardiovascular system and myocardium. Therefore, despite good agreement of results obtained from MCL measurements and physiologic data, MCL investigations are typically an interim step in preclinical evaluations of physiologic control systems and cardiovascular devices, to reduce the required number of in-vivo studies.

![Mock Circulatory Loop](image1.png)  ![Schematic](image2.png)

**Figure 1-15** – (A) Mock Circulatory Loop of the systemic and pulmonary circulation [98,102] and (B) the corresponding schematic representation. LAC, left atrial compliance; LVC, Left ventricular compliance; AOC, aortic compliance; SVC, systemic venous compliance; RAC, right atrial compliance; RVC, right ventricular compliance; PAC, pulmonary arterial compliance; PVC, pulmonary venous compliance; $Q_c$, coronary flow meter; $Q_s$, systemic flow meter; $Q_p$, pulmonary flow meter. Images courtesy of BiVACOR Inc. (Houston, TX, USA).

### 1.3.4 The Role of the Cardiovascular Pulse

Despite the ongoing success of ventricular assist devices based on RBP technology, their introduction to the field of mechanical circulatory support opened an ongoing debate about the necessity and benefits of the cardiovascular pulse. The discussion is mainly fuelled by the increased observation rates of adverse events such as gastrointestinal bleeding, arteriovenous malformations, aortic insufficiency, or pump thrombosis, which are potentially
related to the diminished pulsatile or non-pulsatile flow generated by rotary VADs [7]. While rotary VAD support with diminished pulsatility is well established and proven to sufficiently support life for several years [103], attenuated-pulse perfusion may adversely affect the baroreflex leading to hypertension or increase the risk of impaired microcirculatory perfusion [14,15,104]. Therefore, the application of rapid impeller speed modulation to amplify or generate pulsatile flow in rotary VADs and TAHs is extensively debated regarding the potential advantages of pulsatile haemodynamics over attenuated-pulse or nonpulsatile perfusion. While those advantages include rather practical aspects such as pump washout effects reducing the risk of thrombus formation [20,105], multiple studies indicated that pulsatility may play an important role in the cardiovascular physiology. With no claim to completeness, this section will give a brief overview of past research concerning pulsatility with mechanical circulatory support (MCS). First, methods to quantify the cardiovascular pulse are discussed. Subsequently, the most commonly observed adverse events, which have been associated with attenuated pulsatility; the potentially beneficial aspects of pulsatile haemodynamics; and concerns raised about rapid RBP impeller speed modulation are presented.

1.3.4.1 Quantification of Pulsatile Haemodynamics

While numerous studies have been performed to evaluate the necessity of pulsatile perfusion in short- and long-term mechanical circulatory support, inconclusive and conflicting results facilitated an ongoing uncertainty with respect to the implications of attenuated-pulse or nonpulsatile haemodynamic flow. However, while the question of the purpose of speed modulated RBP operation divides the field of MCS research, there is wide consensus between its advocates and critics, that precise quantification of pulsatile haemodynamics is indispensable [18,20]. Differences in outcomes and conclusions amongst earlier studies may in part be caused by the inconsistent classification of a physiologic pulse [19]. The use of different metrics to compare haemodynamic waveforms results in inaccurate comparison and assessment, as the basic definition of a cardiovascular pulse is inconsistent. In particular, this applies to the haemodynamic output of rotary VADs, which was often labelled as “continuous flow”, although the waveforms exhibited some degree of pulsatility (referred to here as attenuated-pulse). This section discusses the most commonly used metrics to
assess the cardiovascular pulse. A summary of the corresponding formulae is given at the end (in Table 1-2).

The traditionally most commonly used classification of the cardiovascular pulse is the arterial pulse pressure \( PP \), which is defined as the difference between the maximum systolic \( P_{ao,\text{max}} \) and the minimum diastolic pressure \( P_{ao,\text{min}} \) [89]. It is a comparatively simple number to obtain, as only the measurement of the arterial pressure is required, which is often approximated through non-invasive measurement at the brachial artery. A similar metric evaluating the systemic flow rate is given by the pulsatility index \( PI \), which has been calculated as the difference between maximum flow rate \( Q_{s,\text{max}} \) and minimum flow rate \( Q_{s,\text{min}} \), divided by the mean flow rate \( Q_{s,\text{mean}} \) in the ascending aorta [106]. In the context of transcranial Doppler studies, \( PI \) was synonymously calculated with the blood velocity instead of the flow rate [107,108]. The pulsatility index was first introduced by Gosling et al. [109], whereas it was initially defined as the sum of energies in the Fourier harmonics of a flow velocity waveform \( PI_c \), and later superseded by the simplified formula [107,110].

However, the persistent use of \( PP \) and \( PI \) is controversial. In various publications, it has been increasingly emphasized that these simple metrics are insufficient to evaluate the effect of device-induced pulsatility. In particular, Ündar et al. [89,108,111–113] extensively stressed the need for the use of quantifications reflecting the energy content of the aortic pressure and flow waveforms, such as Energy Equivalent Pressure \( EEP \) and Surplus Haemodynamic Energy \( SHE \). \( EEP \) is calculated as the quotient of the total energy and the total volume delivered during the cardiac cycle (where energy is the product of flow rate and pressure). It was argued, that \( EEP \) allows quantification of the additional energy which translates to friction within the blood vessels [113], which may assist in stimulation of vasomotor activity [114]. As a measure of comparison between pulsatile and non-pulsatile flow, \( SHE \) is defined as the difference between \( EEP \) and mean arterial pressure \( MAP \). It serves to represent the additional energy delivered to the circulatory system in a pulsatile condition, when the same mean flow rate is generated as in a comparable continuous flow condition. Figure 1-16 shows a graphic illustration of the surplus haemodynamic energy as presented in [89]. The formula applied to calculate \( SHE \) typically includes a constant factor of 1,332, converting the unit to \( \text{erg/cm}^3 \) (= 0.1 \( J/m^3 \)).
While the unit of erg is outdated, it is frequently used in the context of SHE. However, it was argued, that the term “surplus haemodynamic energy” is misleading, as it is measured in a unit of pressure, rather than energy, hence Amacher et al. replaced it with the term “Surplus haemodynamic pressure” [11].

While the aforementioned quantities represent the approach to evaluate cardiovascular pulsatility in a single quantity, further efforts have been made to analyse differences in the shapes of induced pressure and flow waveforms. Specifically, with consideration of the stimulation of baroreceptors in the aortic arch and carotid sinuses (cf. section 1.3.4.2.1), it has been widely discussed that not only the amplitude of the pressure pulse, but also the rate of pressure change is an important characteristic in determining the cardiovascular response to pulsatile haemodynamics. Therefore, different studies suggested the use of the maximum slew rate of the aortic pressure during systole ($dP/dt$) to evaluate pulsatility [7,72,115–117].

Further analysis of the spectral flow components has been performed in a few studies [118,119], applying the so-called Pulse Power Index (PPI) as first proposed by Grossi et al. [120]. PPI was derived as a modification of the initial definition of the pulsatility index ($P_{I_G}$), in which the harmonic components of the flow are weighted with their respective frequency. It aims to reflect the relative sharpness of a given flow waveform with respect to its mean flow and frequency [121]. PPI was found to exponentially decrease with an increase in LVAD assist ratio (LVAD speed) at a fixed cardiac output condition and presented a significant correlation between these metrics and the pulsatility of the aortic pressure.
waveform. However, while the results of the in-vitro study show a good correlation, the in vivo results [118] have a high variance.

A summary of the introduced metrics is given in Table 1-2. Even though various metrics have been proposed, none of them has become a widely-accepted standard amongst clinicians and engineers involved in research and development around RBPs. Circulatory support with RBPs has been associated with a number of complications [7], some of which may be related to attenuated-pulse or nonpulsatile flow. There has been a wide range of potentially beneficial aspects of induced pulsatility during mechanical circulatory support indicated. Those benefits have been associated with different haemodynamic characteristics including the rate of change, amplitude, and energy content of pulsatile waveforms. Therefore, a broader evaluation of pulse waveforms with multiple quantifications appears advisable for the evaluation of effects of pulsatility on the circulatory system.

Table 1-2 – Frequently used measures for quantification of pulsatile haemodynamics. PP, pulse pressure; PI, pulsatility index; EEP, energy equivalent pressure; SHE, surplus haemodynamic energy; \( P_{ao} \), aortic pressure; \( Q_s \), systemic flow rate; \( Q_s,i \), \( i \)’th harmonic of systemic flow; \( \omega_i \), harmonic flow frequency.

<table>
<thead>
<tr>
<th>Metric</th>
<th>Symbol</th>
<th>Formula</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pulse Pressure</td>
<td>PP</td>
<td>( P_{ao,max} - P_{ao,min} )</td>
<td>[89,122,123]</td>
</tr>
<tr>
<td>Pulsatility Index by Gosling et al.</td>
<td>PI&lt;sub&gt;G&lt;/sub&gt;</td>
<td>( \sum_{i=0}^{n} Q_s^{2,i} )</td>
<td>[109,110,120]</td>
</tr>
<tr>
<td>Pulsatility Index</td>
<td>PI</td>
<td>( \frac{Q_{s,max} - Q_{s,min}}{Q_{s,mean}} )</td>
<td>[106,107,110]</td>
</tr>
<tr>
<td>Energy Equivalent Pressure</td>
<td>EEP</td>
<td>( \frac{\int P_{ao} \cdot Q_s , dt}{\int Q_s , dt} )</td>
<td>[113,124,125]</td>
</tr>
<tr>
<td>Surplus Haemodynamic Energy</td>
<td>SHE</td>
<td>( 1332 \cdot (EEP - \bar{P}_{ao}) )</td>
<td>[112,126,127]</td>
</tr>
<tr>
<td>Maximum pressure slope</td>
<td>( \frac{d}{dt} P_{ao,max} )</td>
<td>[104,115,123]</td>
<td></td>
</tr>
<tr>
<td>Pulse Power Index</td>
<td>PPI</td>
<td>( \sum_{i=0}^{n} Q_s^{2,i} \cdot \omega_i^2 )</td>
<td>[118,120]</td>
</tr>
<tr>
<td>Haemodynamic waveform shape</td>
<td></td>
<td>( P_{ao}(t); Q_s(t) )</td>
<td>– [12]</td>
</tr>
</tbody>
</table>

1.3.4.2 Physiologic Effects Associated with RBP Support and Reduced Pulsatility

While improved treatment outcomes with rotary blood pumps (RBPs) have been reported compared to their positive displacement predecessors, the long-term effects of attenuated or nonpulsatile flow remain unclear, as perfusion with attenuated or nonpulsatile flow patterns has been associated with increased observation rates of several adverse events. In this section, an overview of physiologic changes associated with the implantation of RBPs and the potential impact of reduced haemodynamic pulsatility is presented.
1.3.4.2.1 Pulsatility and the Baroreceptor Response

The baroreflex is an important mechanism for maintaining homeostasis of the arterial blood pressure by short-term adjustments of the vascular tone in response to variations in posture or exercise condition. Changes in pressure are sensed by these stretch-sensitive mechanoreceptors located in the viscoelastic walls of the aortic arch and the carotid sinuses [24,128]. In response to an increase in arterial blood pressure the arterial walls are stretched, resulting in the baroreceptor activation. Neural impulses are sent to the nucleus solitary tract in the medulla, which inhibits activity in the sympathetic nervous system, resulting in a release of neurohormones to regulate blood pressure, heart rate, and heart contractility [129]. While the exact underlying mechanisms are not fully understood, the basic functionality of the reflex has been explored and several studies showed that the response in firing rate exhibits effects of threshold activation, hysteresis and saturation [115,129–131].

A study evaluating baroreceptor activity in rabbits [130] found that the comparison of the response to pulsatile and non-pulsatile pressure depends on the relation of both the mean arterial pressure (MAP), and instantaneous systolic and diastolic pressures to a threshold pressure corresponding to the onset of receptor firing. A statistically increased receptor response to pulsatile pressures compared to constant pressures was reported for the receptors in both the aortic arch and the carotid sinuses [132]. Additionally, it was found that the response was related to the rate of change of pressure \( (dP/dt) \), where the peak of receptor activity with a pulsatile pressure coincided with the maximum \( dP/dt \) instead of the peak systolic pressure. In this context the baroreceptors are associated with being sensitive to stretch of the surrounding tissue rather than to static pressures [19]. These findings are consistent with those of Eckberg, who showed that the baroresponse in the carotid sinuses increases with \( dP/dt \) up to rates of approximately 300 mmHg/s until it reaches an asymptotic plateau in the range of the natural heart \((440 – 1180 \text{ mmHg/s})\) [115]. It was also shown that the baroresponse decreased with a steeper diastolic pressure drop rate. These reports suggest that receptor firing is maximal with a steep systolic increase and a slow diastolic decrease of pressure, as are seen in the pulsatile pressure waveforms produced by the natural heart.

A debate focusing on the role of the baroreflex in the long-term control of MAP continues. This is due to the uncertainty regarding the adjustment of baroreceptors to a chronically
changed pressure level, resulting in ineffectiveness of the reflex in the long-term regulation of MAP [133]. In contrast, studies found that pulsatility in arterial pressure can prevent this “resetting” of baroreceptors [134]. A comprehensive understanding of all interrelations concerning the baroreceptor reflex has not been established so far. However, multiple studies associated increased receptor firing with physiologic pulsatile perfusion, thus RBP support with attenuated-pulse or nonpulsatile perfusion may increase the vascular impedance [135]. A long-term lack of pulsatility may therefore influence vasotone of the arterial and/or venous system and may thus adversely affect venous blood return to the heart or lead to hypertension [14].

1.3.4.2.2 Pulsatility in Large Vessels

The healthy human circulatory system is characterised by the pulsatile nature of pressure and flow waveforms. When the native heart is failing, ventricular pressure reduces and blood flow as well as amplitude of the arterial pulse are reduced. While the implantation of rotary left ventricular assist devices (rotary LVADs) operated at a constant speed allows restoration of the mean flow to physiologic levels, their continuous flow nature does not inherently generate pulsatile outflow. Rotary LVADs continuously move blood from the left ventricle to the aorta, bypassing the aortic valve. Consequently, ventricular preload is reduced, which, due to the Frank-Starling mechanism, results in reduced myocardial contractility [20]. Therefore, depending on the operating speed and flow rate of the pump, the ventricular pressure may stay below the aortic pressure intermittently or through all pulse cycles – conditions in which the aortic valve will not open. The concomitant less frequent aortic valve opening (AVO) may result in fusion of the aortic valve leaflets and aortic insufficiency. Furthermore, a larger portion or all of the systemic blood flow passes through the pump, which results in further reduced pulsatility (Figure 1-17) and, potentially, in areas of flow stagnation in the ventricle and aortic root which may lead to thrombus formation [40,116,136] and potentially thrombus dislodgement upon single valve opening events, leading to stroke or pump stoppage.
Varying degrees of pulsatility before and after LVAD implantation. CF-LVAD, continuous flow left ventricular assist device; AVO, aortic valve opening. Lavare cycle refers to an intermittent speed reduction mode of the HeartWare HVAD. Figure first published in [20].

It has further been indicated, that prolonged continuous flow support may result in decreased stiffness and thinning of the aortic wall due to a decrease of smooth muscle cells and collagen [136–138]. Furthermore, enlargement of the aortic diameter was reported in this context, however, conflicting indications as to whether this is a cause of effect of aortic wall thinning were given.

Conversely, a more recent study found the aortic wall in LVAD patients to be substantially increased in thickness and stiffness compared to heart failure patients, leading to the conclusion that continuous flow support may accelerate vascular ageing [139]. While the exact mechanisms are not fully understood and contrary results were reported, multiple studies indicate an influence of continuous flow perfusion on the morphology of large arterial vessels, which may for example be related to a lack of vessel relaxation in diastole. While the outcomes may for example be further related to mean pressure and flow levels, type of pump, and level of pulsatility, further research is warranted by the findings of these studies.

1.3.4.2.3 Gastro-Intestinal Bleeding
Gastro-intestinal (GI) bleeding is one of the most common adverse events, which is increasingly observed in patients undergoing rotary ventricular assist device therapy
While there are multiple potential causes, the most common source of GI bleeding are arteriovenous malformations (AVMs) [141]. These friable, abnormal connections between the arterial and venous system bypass the capillary bed, pose an increased risk of bleeding. Potential causes for AVM development are an increased intraluminal pressure and decreased intestinal mucosal perfusion due to a reduction in pulse pressure [142,143]. Further investigations have been presented focusing on effects on the microscopic and cellular level. Variations in blood flow patterns result in changes in cyclic shear stress and strain imposed on the arterial walls. It was shown that pulsatile flow significantly augments the wall shear stress and is thus considered to have an important impact on the vascular endothelium [144,145]. This mechanotransduction system is involved in the regulation of apoptosis, angiogenesis, and blood pressure [72,89]. A change to non-pulsatile perfusion may lead to malformation of the arterial walls and occasionally to diseases such as arteriosclerosis or aneurysm, or a decrease in the release of nitric oxide (which acts vasodilating) [20,114,145,146]. Another demonstrated effect is the depletion of large von Willebrand factor (vWF) in LVAD patients [141,147]. High molecular weight vWF multimers play an important role in thrombus formation and haemostasis [148], and their absence has been associated with GI bleeding. However, a reduction of vWF multimers is likely primarily caused by excessive blood shear stress and subsequent vWF destruction within the RBP, rather than the diminished circulatory pulse itself [18,149].

**1.3.4.2.4 Blood Trauma**

In both positive-displacement and rotary blood pumps, blood passing through the device is subject to high mechanical stress in mechanical heart valves and thin blood passages at the impeller vane tips or hydrodynamic bearings. These shear forces may result in blood trauma such as haemolysis, platelet activation, lysis of von Willebrand factor multimers, and thrombosis [16,150]. Depending on the device type or design, the major factors influencing blood trauma are the maximum shear forces as well as the exposure time. It has been shown that shear-induced platelet activation and haemolysis increase with magnitude and exposure time to the shear stress [17,148]. The blood compatibility of RBPs is subject to the optimisation of the pump design [151], and may be influenced by factors such as the required rotational speed, blood gap size, flow velocity, and normal stresses in hydrodynamic bearings.
Consequently, concerns have specifically been raised that rapid RBP speed modulation may result in increased blood shear stress and concomitant haemolysis and vWF cleavage, while no detailed studies evaluating blood trauma due to rapid speed modulation modes are not available in literature, likely due to the complexity of the required experiments [18]. Consequently, future research is required to explore the haemocompatibility of rapid speed modulation. For example, it is imaginable that a substantial increase of blood damage at high flow rates during systole is to some degree accompanied by decreased blood trauma during diastole. Therefore, future studies with respect to blood compatibility under consideration of different pulse amplitudes and speed profile shapes are required. The significance of such studies would be enhanced with a RBP, which can deliver pulse waveforms similar to the native ventricle.

**1.3.4.2.5 Microcirculatory and End-Organ Perfusion**

Controversy has also been reported regarding the effect of pulsatile pressures on the microcirculatory level, as the pulse pressures in the capillary beds are small due to the dampening effect of arterial compliance. However, it has been shown that pulsatility does exist in the microcirculation [152], resulting in an increase in erythrocyte velocity and the number of perfused capillaries [72,108,117]. Furthermore, potential improvements in fluid exchange due to sinusoidal capillary pressure changes, resulting in a higher lymph formation and flow were reported [19,153]. As the major lymphatic trunks are wrapped around arteries, fluid movement is augmented by arterial pulsations, thus assisting in the prevention of oedema. Conversely, a lack of pulsatility may result in microcirculatory collapse and ensuing tissue ischaemia [19].

As a consequence, concerns with respect to sufficient end-organ perfusion with nonpulsatile perfusion have been raised [154,155]. Yoshioka et al. evaluated the end-organ function in 469 continuous-flow LVAD patients and found a gradual decline of renal function after an initial transient improvement post implantation [156]. Similar findings were presented in [157], however, as similar trends with pulsatile devices were observed, a direct correlation with continuous flow was suggested to be unlikely. Conversely, increased tissue metabolism and reduced peripheral vascular resistance in the microcirculation were stated as potential effects of improved end-organ perfusion with pulsatile blood flow [158]. While multiple studies found no impairment of cerebral, hepatic, and renal function over limited time frames
[159–161], potential long-term effects of reduced pulsatility have not been completely explored. Further, the clinical relevance of some presented findings was questioned due the use of healthy animals and inconsistent classification of the grade of pulsatility [40,162]. However, the currently available studies mostly indicate that end-organ perfusion may not be substantially impaired with nonpulsatile perfusion.

1.3.4.3 Rapid Speed Modulation of Rotary Blood Pumps

The previous section showed the potential benefits and controversy with respect to pulsatility with rotary blood pump (RBP) therapy. Consequently, many studies investigated the pulsatility generated by RBPs. Stanfield et al. showed that, in interaction with the native ventricle, centrifugal pumps operated at constant speed transferred significantly greater pulsatility compared to axial flow pumps [106]. Typically, centrifugal pumps operate at lower speeds and exhibit a flatter pressure head-flow characteristic (HQ-curve) compared to axial flow pumps, i.e. the flow output is more sensitive to changes in inlet and outlet pressures. Consequently, a larger variation of the flow rate in response to ventricular contractility is observed. However, many centrifugal RBP exhibit comparatively small dimensions of the blood flow path, which increases the characteristic resistance to blood flow through the pump, resulting in a larger pressure drop with increasing flow rates, thus in a steeper HQ-characteristic. A possible approach to enhance arterial pulsatility may therefore be to design a centrifugal pump with a maximally flat HQ-curve. However, this may further be accompanied by large negative flow peaks during diastole, where the pump pressure head is large, as can be estimated from the data presented in [106]. Furthermore, it is unlikely, even with such pump characteristics, that a physiologic pulse can be restored with constant speed, and this approach is not applicable to the case of a rotary total artificial heart (TAH) where ventricular contractility is completely absent. Therefore, a widely investigated approach is the recreation of pulsatility through rapid impeller speed modulation. In this section, an overview of previous studies with respect to speed modulation of VADs and TAHs is given.
1.3.4.3.1 Ventricular Assist Devices

Numerous studies investigated rapid speed modulation of ventricular assist devices in interaction with the native cardiac cycle. Investigations included asynchronous speed modulation, as well as approaches to synchronize VAD speed profiles to the native heart beat with the use of ECG signals [11,116,163]. While speed modulation may be used to enhance arterial pulsatility, it further has distinct effects on the native ventricle and the total cardiac output. This section briefly introduces some of the findings presented in previous studies, and introduces speed modulation modes implemented in commercially available VADs.

Pirbodaghi et al. applied saw tooth, square wave, sine wave and triangular waveforms in synchrony with the native heart beat in a Levitronix Centrimag VAD in vivo [164]. All applied profiles had the same mean speed and amplitude, while two different phase shifts to the native cardiac cycle were set to achieve copulsation (the high-speed pulse coincides with ventricular systole) and counterpulsation (the speed profile is delayed by half the length of the cardiac cycle compared to copulsation). Speed modulation significantly decreased ventricular load in terms of stroke work compared to constant speed operation. Counterpulsation was found to have a stronger effect than copulsation, whereas the speed profile shape did not have a significant influence. To allow a consistent quantitative evaluation of the relative timing between VAD speed profiles and the native heartbeat, Amacher et al. defined the phase shift of electrocardiogram-synchronised (ECG) VAD speed profiles with respect to the cardiac cycle in percent [11]. A phase shift of 0% is defined as the timing, at which the R-wave of the ECG coincides with the point of maximum speed of a sine wave profile, or the centre of the high-speed pulse of a square wave profile (Figure 1-18). A phase shift $\phi > 0$ then describes a delay of the speed profile by $\phi$ percent of the period length of the cardiac cycle. Applying this definition to Pirbodaghi’s study [164], copulsation equates to a phase shift of 25% and counterpulsation equates to a phase shift of 75%. Amacher performed a more extensive in-vitro study, in which the phase shift was varied from 0% to 100% in steps of 5% [11]. It was found, that ventricular unloading was optimal at $\phi = 80\%$ for a sine wave speed profile, which in line with previous findings. They further agreed with Pirbodaghi’s conclusion that the differences in the speed waveform do not lead to major differences in the evaluated haemodynamic criteria [11,164].
It was further shown, that maximum unloading of the native heart and maximum pulsatility in the aortic pressure and flow waveforms are competing objectives with respect to the phase shifts, as these maxima occurred at phase shifts of 80% and 30% respectively. The effect of changes in the speed profile waveform on haemodynamics were small compared to changes in phase shift, however, parameters such as pump power consumption and maximum $dP/dt$ were not evaluated, but may significantly vary with the chosen speed profile. In a subsequent study, further efforts were made to numerically find an optimal speed profile for VADs. A mathematical formulation of the conflicting objectives such as aortic valve opening, aortic pressure and flow pulsatility, and left ventricular unloading was implemented in terms of a weighed mathematical function, which was optimised for the different criteria [165].

The results found in literature promise the potentially beneficial effects of VAD speed modulation, while no established strategy to restore a truly physiologic pulse synchronously to the native heart has been applied clinically so far [165]. However, first commercial implementations of speed modulation modes for VADs are available or under development. An example is the so-called Lavare-cycle, which has been implemented for the HeartWare HVAD, and is currently available in the European market [55,166]. In this asynchronous speed modulation mode, the impeller speed is reduced for a three second period once per
minute. It aims to allow intermittent opening of the aortic valve and thus wash out the aortic root and avoid thrombus formation. Additional adjustment of the pulse timing and amplitude are available with the qPulse algorithm, which was implemented for the miniaturised HeartWare MVAD [167].

A rapid speed modulation mode with a higher frequency was included in the control of the HeartMate III centrifugal LVAD (St. Jude Medical), which allows setting of a rapid speed reduction, followed by a speed increase, to generate an artificial pulse with a physiologic beat rate, where the primary aim is washout of the pump [13,20].

### 1.3.4.3.2 Speed Modulation of Rotary Total Artificial Hearts

Due to the absence of the native heart, compared to VADs, rapid speed modulation of a rotary total artificial heart (TAH) is a seemingly simpler task, as the interaction with ventricular contractions need not be considered. However, as the device must provide the entire circulatory flow, the requirements for the motor and hydraulic performance are increased compared to that of a VAD. To restore truly physiologic pulsatile perfusion, peak flow rates greater than 20 L/min are required, which may pose a challenge to the device design.

Few studies have evaluated rapid speed modulation of rotary TAHs. An early approach to develop a pulsatile rotary total artificial heart is presented in [168], where a single centrifugal pump was used to perfuse both pulmonary and systemic circulation. The inlet and outlet were each connected to a three-way valve, which were synchronously controlled by a solenoid drive. The valves were used to alternatingly connect the RBP to the right and left circulation and thus provide pulsatile flow. Cardiac output was balanced by means of changes in the valve switching cycles. However, while the principle may be operational, the mechanism contradicts the advantages of contact-less third generation RBP and returns to the need for valves, which may increase blood damage and susceptibility to mechanical failure. A similar, more advanced approach is being developed by OregonHeart Inc. (Roseville, CA, USA) [169]. The device uses a single centrifugal impeller, which is shuttled between two outflows, which are connected to the aorta and pulmonary artery. It inherently delivers alternating pulsatile outflow without the need for mechanical valves. However, it is consequently not suitable to compare the effects of nonpulsatile and pulsatile perfusion on the mammalian physiology with a single device. The device was able to deliver a peak systemic flow rate of
17 L/min at pulse pressures between 40 and 60 mmHg in a mock circulatory loop, however, no further analysis of the generated pulsatility was presented. The device is currently under preclinical investigation.

A different study presented an in-vitro investigation of arbitrarily chosen speed profiles applied to both the left and right pump in a dual rotary VAD as TAH configuration [170]. The study investigated changes in the systemic haemodynamics in response to variations in speed profile amplitudes and beat rates, while constant mean speeds for both pumps were maintained. The left and right pump were synchronously pulsed in copulsation, and in counterpulsation modes and individually. It was found, that pulsatility, as assessed by SHE was proportional to systolic duration and inversely proportional to beat rate. Maximum pulsatility was achieved with square wave profiles with a duty cycle of 50% and low beat rates of 10 bpm, as this setting allowed equal and sufficient settling times for arterial pressures during systole and diastole. The influence of right pump speed modulation on systemic haemodynamics was found to be small. A physiologic EEP or SHE could not be achieved in this study. Furthermore, the study was later criticised due to the insufficiently modelled arterial Windkessel in the utilised mock circulation loop [18].

Fukamachi et al. varied pump speed of the CFTAH [79], a pump combining two impellers on a common rotor. They compared hydraulic pressure head-flow curves with and without speed modulation. It was found that the performance of the left pump was slightly decreased with speed modulation, while the change in right pump performance was dependent on the position of the moving impeller within the pump housing. It was concluded that increasing or decreasing the speed modulation amplitude can be used to alter hydraulic characteristics and improve flow balancing of the left and right impellers. The magnitude of the pulsatile outflow was not specified in detail. In a further study, simulation of a physiologic arterial pressure waveform with the CFTAH in a mock circulatory loop was attempted [78]. It was shown, that the waveform of the arterial pressure could be significantly altered with the choice of the speed profile shape. A custom profile to generate a physiologic pressure waveform was developed and compared to square wave and sine wave profiles (Figure 1-19). It was shown to require less power consumption at the same amplitude, however, it also resulted in a lower pulse pressure.
Further, it can be noticed, that the device acceleration was slow compared to the target speed signal (Figure 1-19 A, B). More detailed analysis of measures such as \( \text{SHE} \), \( \frac{dP}{dt} \), and the motor power consumption at comparable levels of pulsatility was not performed. However, the study indicated, that the choice of speed profile can significantly affect pump efficiency and output haemodynamics. It therefore appears promising to engage in further detailed studies to evaluate the effect of speed profile characteristics on haemodynamic pump output. The generation of physiologic waveforms which resemble those generated by the native heart has not been achieved to date. Therefore, in-depth study of pump characteristics influencing the device performance with rapid speed modulation may pave the way to generating physiologic pulsatility with rotary TAHs in the future, and allow subsequent studies considering the effects of the absence and presence of physiologic pulsatility on the human circulatory system with the same device.

Figure 1-19 – Example of rapid speed modulation profiles manually adjusted to obtain near-physiologic pressure waveforms. Panels show the target pump speed (A), actual pump speed (B), systemic flow rate (C), pulmonary flow rate (D), aortic pressure (E), and pulmonary arterial pressure (F). Figure first published in [12].
1.3.4.4 Ongoing Controversy

The above discussions indicate that restoring truly physiologic cardiovascular pulsatility may be beneficial to the circulatory system. However, it is important to consider possible side-effects and challenges, which may arise when rapid speed modulation of RBPs is considered. For example it was argued, that rapid speed modulation may lead to rotor suspension instabilities, haemolysis, or increased platelet activation and vWF cleavage [18], while the rapid acceleration and deceleration of the pump impeller may cause an excessive increase in device power consumption and potentially blood damage through heat generation. Furthermore, due to their size, typically used outflow grafts for LVADs may restrict the maximum flow rate, thus only allow insufficient flow pulsatility. Consequently, depending on the application and the RBP used, it may be counterproductive to modulate the pump speed. Furthermore, it was argued that clinical outcomes with continuous flow LVADs are similar or superior to pulsatile pumps and no significant changes in, e.g., end-organ perfusion were registered [160,171]. Conversely, similar study outcomes have been confronted with criticism of inadequately defined and measured pulsatile haemodynamics or disregarding the transferred pulsatility from the residual heart function in rotary LVAD patients [111,172,173].

The controversy around the cardiovascular pulse and the long-term effects of attenuated or non-pulsatile flow may only be settled, if long-term studies allowing a meaningful comparison of truly physiologic pulsatile and non-pulsatile perfusion can be designed. Such studies may require assessment of a device-related maximum level of pulsatility, which may allow to improve the circulatory autoregulatory response and/or microcirculatory perfusion, while maintaining acceptable levels of blood damage. In this context, a rotary TAH which allows the direct comparison of physiologic waveforms and pulseless circulation with the same device is likely to be a valuable research tool for future long-term studies. However, the design of such a device requires overcoming previously faced challenges with rapid RBP speed modulation. An example of a single-device rotary TAH, which may potentially be designed to fulfil the requirements in the future is the BiVACOR TAH, which is described in the following section.
1.3.5 A Rotary Total Artificial Heart – The BiVACOR® TAH

The BiVACOR® TAH is a single-device rotary total artificial heart [82,174]. The device uses a single spinning impeller, which comprises semi-open impeller vanes for two adjacent centrifugal pumps, each with a separate inlet and outlet to simultaneously support the pulmonary and systemic circulations. The outer diameter of the right sided impeller is substantially smaller than the left sided impeller to accommodate for the lower pressure requirements of the pulmonary circulation. The impeller hub is rotated with an axial flux permanent magnet brushless motor. The motor stator comprises twelve concentrated tooth-coils, and is located within the titanium pump housing, surrounding the left pump inlet. The stator coils are mounted to the teeth of the stator core, which is built from an iron compound soft magnetic composite material (SMC) to suppress eddy currents and reduce stator core losses. The motor rotor comprises a ring-shaped iron yoke and ten axially magnetised Neodymium-Iron-Boron (NeFeB) permanent magnets with alternating polarisation, which are mounted to the iron yoke. The utilized concentrated twelve-slot/ten-pole double layer winding is advantageous, as it results in a relatively large fundamental winding factor and it exhibits short end-winding turns in the radial direction, leaving a larger inner diameter [82,175–177]. The rotor assembly is located within the impeller hub, behind the left sided impeller vanes, which enforces a comparatively large magnetic gap. During operation, the stator is energized with alternating currents to generate a rotating magnetic field, which interacts with the rotor permanent magnet field to generate driving torque. The BiVACOR device is illustrated in Figure 1-20, showing an exploded view of the BiVACOR V1, illustrating the hydraulic, motor, and magnetic bearing systems (Figure 1-20A), and a rendered image of the BiVACOR V2 with atrial cuffs and outflow grafts (Figure 1-20B). The interaction of the rotor permanent magnets and the opposing stator iron core further causes an axial attractive force between the rotor and the stator. This force is counteracted by a hybrid passive and active magnetic bearing system, which facilitates contact-less magnetic levitation of the impeller within the pump housing. The bearing comprises three solenoids located around the right pump inlet, each manufactured from a horseshoe-shaped iron core and a mounted coil. The solenoids target a second magnetic assembly within the rotor hub, which faces the right sided magnetic bearing stator. It comprises two concentric axially magnetised NeFeB permanent magnet rings, which face the two legs of the magnetic bearing iron cores and are mounted to a ring-shaped magnetic yoke.
Figure 1-20 – The BiVACOR TAH. (A) Exploded view of the BiVACOR V1 [174] and (B) BiVACOR V2 with atrial cuffs (right) and outflow grafts (left); Image courtesy of BiVACOR Inc.

The permanent magnets in the magnetic bearing target provide a bias force of the order of the motor axial force, which acts in the opposite direction. This assembly results in an unstable force equilibrium in the centre of the magnetic gap. A movement towards the right side (magnetic bearing side) increases the attractive force of the magnetic bearing and reduces the attractive force of the motor, thus a net force is acting towards the right side; conversely, a movement towards the left side results in a net force towards the motor. The magnetic bearing solenoid coils are electronically controlled to actively levitate the rotor in the target axial hub position, and passively stabilize the rotor in radial direction. The solenoid currents can be altered to supply a positive bias force to attract the hub, or negative bias force to repel the hub. Position feedback to the control algorithm is provided by three sensors measuring the axial hub displacement.

Automated axial movement of the impeller hub utilising a virtual zero power controller [178] is used to alter the clearance between the impeller vane tips and the pump housing of the left and right pumps [96]. Due to the movement, the hydraulic efficiency of the pumps is modified in such a way, that balance of the flow rates supplied to the systemic and pulmonary circulation in response to changes in circulatory pressures and the resulting hydraulic forces acting on the impeller hub is achieved. For example, an increased pressure on the right circulation causes a hydraulic force acting towards the left side of the pump, which is detected.
through an increase in the magnetic bearing current required to maintain the axial position. In response, the control system automatically moves the rotor target position towards the right side, to regulate the bias current to zero, i.e. moves the impeller position to the new equilibrium point, constituted by the superposition of hydraulic axial force and permanent magnet forces from the magnetic bearing and motor. Consequently, the gap over the left blades is increased, and the gap over the right blades is decreased, resulting in a relative increase of the right flow rate, and a concomitant decrease of the right inlet pressure. Vice versa, an increase in the left inlet pressure results in a hub movement towards the left side, effectively maintaining a pressure and flow balance between the left and right circulation.

Within the magnetic gap between the motor stator and rotor assemblies, the left impeller vanes, a blood gap, and the titanium housing are located. Consequently, the ratio of the magnetic gap to outer motor radius is large compared to typical permanent magnet machines, resulting in challenges in design and modelling of the drive [179]. While the permanent magnets may be embedded in the stator vanes [55] or, if present, the impeller shrouds, this is not possible with the open-impeller design of the BiVACOR. Therefore, the gap length is considered large even in the context of rotary blood pumps.

During this PhD project, different iterations of the BiVACOR TAH were used in-vitro and in-vivo to evaluate the device’s performance in pulsatile operation, and identify challenges in RBP design under consideration of pulsatile operation.

1.3.6 Rotary Blood Pump Motor Drives
Design advancements of modern second and third generation RBP promise device operation for ten years and beyond. While state-of-the-art contactless rotor suspension systems allow for long-term wear free operation, the desired device lifetime implies specific requirements with respect to durability and reliability of the motor drive. While several types of electrical machines are in principle applicable to the typical torque and speed requirements, significantly simpler and more reliable operation is ensured with the use of a contactless motor assembly, which is applicable to switched reluctance machines (SRM), induction machines (IM), and brushless permanent magnet machines (BLPM). While the SRM and IM types are associated with a large torque ripple and a complex rotor assembly respectively [179], BLPM machines offer the advantages of a simple construction and high efficiency.
also due to the fact that the magnetic rotor field is provided by permanent magnets. Therefore, modern third generation RBPs are almost exclusively driven by BLPM motors. Two common types include the interior permanent magnet motor (IPM), where the permanent magnets are embedded within the rotor iron, and the surface magnet (SM) motor, where the permanent magnets are attached to the rotor surface. Typically, a three-phase winding is embedded in slots in the stator core, which is distributed around the circumference of the motor. The windings are supplied with three-phase alternating currents to generate a rotating magnetic field within the air gap, which interacts with the rotor permanent magnet field to generate electromagnetic torque to rotate the rotor. Consequently, no electrical connection to the rotor is required, which allows contactless embedding of the motor rotor within an RBP impeller, eliminating the need for a shaft [179]. Figure 1-21 shows the cross section of a typical brushless permanent magnet motor. The conductors of each stator winding \((U, V, W)\) are displaced by 120° around the stator circumference and generate a rotating magnetic field in the gap between rotor and stator, when supplied with symmetrical three-phase currents. The figure further shows the magnetic field lines corresponding to the rotor flux linkage \(\Psi_R\) generated by the permanent magnets. The interaction of stator and rotor magnetic fields in the air gap results in tangential forces and consequently torque, which sets the rotor in motion.

Figure 1-21 – Cross section of a typical brushless permanent magnet motor. \(\Psi_R\), rotor flux linkage; \(PM\), permanent magnet.
1.3.6.1 Blushless Permanent Magnet Motors

The two most common design variants of brushless permanent magnet motors are the radial flux (RFPM) and axial flux permanent magnet motor (AFPM). Both designs use the same working principle, while their geometry and design are vastly different. RFPMs exhibit a concentric structure of rotor and stator, wherein the rotor is most commonly located inside the stator. The permanent magnets in the cylindrical rotor provide a radial magnetic field which closes through the stator teeth and yoke, whereas the active conductors of the stator winding are embedded in the axial stator slots. In contrast, AFPM motors have a disk-shaped rotor comprising a ring-shaped rotor yoke and a number of typically arc-shaped axially magnetised permanent magnets[180]. The stator is located axially facing the rotor, whereas the stator slots progress radially. The active conductor length is therefore determined by the difference between the inner and outer radii of the stator core and rotor assembly. Figure 1-22 shows a comparison of a RFPM and AFPM machine of the same rating [181].

The RFPM machine has a larger axial length and overall volume, while the AFPM machine is shorter in axial length, while it has a bigger diameter. The large diameter to length ratio of AFPM motors allows a flat disc-shaped design with a high power and torque density [181]. Due to their geometric structure, RFPM motors are best suitable for axial flow RBP, while single or double sided AFPM machines are most commonly used in centrifugal blood pumps. Furthermore, the larger diameter allows the placement of a larger number of permanent magnet poles, thus making them a suitable choice for lower speed applications, which conforms with the lower speed requirements of centrifugal RBP compared to axial flow RBP. In the following section, an overview of design trends and challenges with AFPM motors for centrifugal blood pumps is given. 

Figure 1-22 – Permanent Magnet Machine Types. a) Radial Flux Machine; b) Axial Flux Machine [181].
1.3.6.2 Axial Flux Motors for Centrifugal Rotary Blood Pumps

One of the major disadvantages of the AFPM topology is the inherent axial force characteristic. Due to the interaction of the rotor permanent magnet field and the soft magnetic stator core material, strong axial attractive forces occur between the rotor and stator, which need to be carefully considered in the design of the impeller suspension. In RBP applications this is particularly important in the case of contactless hydrodynamic or electromagnetic bearing systems, where bearing instability and rotor touchdown may result in excessive blood damage. Further, geometric design limitations for RBP motors include the overall size and weight, the rotor inertia, and – in many cases – the minimum inner stator radius, where the pump inflow passes through the centre bore of the stator [55,182,183]. Therefore, in most applications a concentrated winding is chosen, as it reduces the wire length and radial dimensions of the end-winding turns [82,175,176,184], whereas suitable pole-slot combinations with respect to the torque and efficiency requirements may be chosen from available data tables in the literature [177,179].

Different methods to counteract or reduce the magnitude of axial attractive forces can be applied. Motors with a coreless stator are a design option for entirely eliminating axial attractive forces. These designs lack soft magnetic core material, to guide the flux through the phase coils, and thus only rely on magnetic stray fields interacting with current carrying conductors (Lorentz force theorem) [175,181]. Due to the absence of the stator iron, eddy current and hysteresis losses can only occur in the rotor yoke and permanent magnets, which may increase efficiency. Conversely, to improve the magnetic coupling and reduce leakage flux, the use of more PM material and/or small gap lengths are required to achieve the same performance, while the majority of losses are Ohmic losses in the stator winding [181]. In RBP applications the gap length between motor stator and rotor usually accommodates parts of the pump housing and fluid paths. Therefore, generally large gap lengths are required, amplifying the shortcomings of a coreless design; to improve efficiency, stators with magnetic core materials are widely used [175,183,185].
Alternatively, the axial force can be compensated with the use of a double-sided motor topology. Figure 1-23 shows an example of a single and a double-sided stator topology. In addition to the passive magnetic force balancing, the double-sided motor bears the advantages of improved efficiency, and potentially redundancy [55]. It is therefore well suited for the combination with a hydrodynamic axial bearing, as observed in the designs of the HVAD and VentrAssist (ex VentraCor, Sydney, Australia) centrifugal VADs [55,183]. In a double-stator design, the flux path of the permanent magnets is closed through both stators, hence crosses the air gap in both axial gaps. Therefore, no soft-magnetic rotor yoke is required, which allows for a thinner rotor design. However, depending on the motor design, the required permanent magnet thickness is determined by the air gap lengths and the stator design. In the case of both the HVAD and VentrAssist pumps, as well as the DuraHeart LVAD (Terumo Heart Inc., Ann Arbor, MI, USA), the stator is executed with a slotless design [55,181,183]. That is, the stator core does not exhibit distinct ferromagnetic teeth, and the coils are mounted on a ring-shaped stator yoke. Therefore, the slotless stator is a compromise between a coreless and a slotted stator which, to some extent, relies on magnetic coupling through leakage flux, while the flux path is guided by the stator yoke. Reasons to employ a slotless stator may include a reduction of weight, as well as reduction of the negative magnetic stiffness and therefore the suspension force load [183,186]. In contrast, similar to the coreless stator, large permanent magnets are required [181], which tends to increase the rotor weight and consequently inertia.

Further efforts to optimise the design of a single-sided axial flux motor for the ReinVAD left ventricular assist device (ReinVAD GmbH, Aachen, Germany) were presented by Pohlmann [179,182]. The study aimed to optimise the drive design with respect to efficiency, while maintaining a maximum axial attractive force. The discussed motor comprises a solid slotted ferromagnetic core, where a strong influence of eddy current losses on the drive efficiency was found, whereas the stator core material was chosen to minimize these losses. Different core materials were evaluated, while the potential use of soft magnetic composite (SMC)
materials to reduce eddy current losses were mentioned as a possible addition; however, while such materials may be particularly advantageous, they were not evaluated. The design optimization further included a reduction of the inner stator diameter and the pump inlet, which may result in a larger pressure drop at high flow rates and added potential for cavitation in the impeller eye, and thus may be disadvantageous for the application in a pulsatile RBP. In the context of the discussed RBP motors, the drive of the BiVACOR TAH exhibits the unique requirements of a comparatively large physical gap length. Further, the incorporation of large flow path diameters may benefit the pump performance at high flow rates, as may be required for rapid speed modulation. Therefore, in the context of RBP design for operation with rapid speed modulation, it may be favourable to identify and evaluate geometrical trade-offs which allow a large inner stator radius, a large gap length, and low rotor inertia, while increasing the efficiency and reducing or maintaining the axial attractive force.

1.3.6.3 Analytical Calculation of Force and Torque

1.3.6.3.1 Air Gap Flux Density

To derive analytical expressions for the axial force and torque in an axial flux brushless permanent magnet motor, the magnetic field in the air gap is considered. A vastly simplified calculation of the air gap flux density, can be derived as follows (the derivation was adapted from Pohlmann [179], who presented similar considerations). Figure 1-24 shows an exemplary excerpt of the magnetic flux. The considered flux path is closed through two permanent magnets, two stator teeth, the rotor and stator yokes, and crosses the air gap twice.

Ampere’s law states that the line integral of the magnetic field strength along any closed loop path equals the electric current enclosed by the loop, which, when a coil with the turn number $N_c$ and the current $i$ is enclosed, equals the magneto-motive force of the coil ($MMF_{coil}$):
Neglecting the leakage flux and with the simplification of a constant cross-sectional area of the flux path (i.e. assuming a homogeneous air gap flux density), for the case illustrated in Figure 1-24, Ampere’s law is simplified to:

\[
\int H \cdot dl = N_c \cdot i = MMF_{coil}.
\]  

(1.7)

where \( l \) considers the length of the flux path in the considered material, and the subscripts denote the portions within the iron stator core and rotor yoke \((fe)\), the permanent magnets \((pm)\), and air gap \((gap)\) respectively.

Assuming a linear demagnetization curve for the permanent magnets, the relationship between magnetic field strength \( H_{pm} \) and flux density \( B_{pm} \) is described by the remanence flux density \((B_r)\), the vacuum permeability \((\mu_0)\), and the relative permeability of the permanent magnet material \((\mu_{r,pm})\):

\[
H_{pm} = \frac{B_{pm} - B_r}{\mu_0 \mu_{r,pm}}
\]  

(1.9)

In the iron core and the air gap, the material equation simplifies to:

\[
H = \frac{B}{\mu_0 \mu_r},
\]  

(1.10)

thus equation (1.8) becomes

\[
\frac{l_{fe} B_{fe}}{\mu_0 \mu_{r,fe}} + \frac{l_{pm}(B_{pm} - B_r)}{\mu_0 \mu_{r,pm}} + \frac{l_{gap} B_{gap}}{\mu_0 \mu_{r,gap}} = i \cdot N_c.
\]  

(1.11)

Considering the significantly larger relative permeability of iron compared to that of the permanent magnet material and air \((\mu_{r,fe} \gg 1; \mu_{r,pm} \approx 1; \mu_{r,gap} = 1)\), the first term on the left side of the above equation can be neglected. Further, due to the assumptions with respect to the cross-sectional area of the flux path, and the neglection of any leakage flux, the
magnetic flux density in the air gap and the permanent magnets are identical and can be calculated as:

\[ B_{\text{gap}} = \frac{i \cdot N_c \cdot \mu_0 \mu_r \cdot \mu_r \cdot l_{\text{pm}} + B_r \cdot l_{\text{pm}}}{l_{\text{pm}} + \mu_r \cdot l_{\text{gap}}} \]  
\[ (1.12) \]

### 1.3.6.3.2 Force and Torque

From a macroscopic point of view, the generated force and torque can be derived from the magnetic field energy stored in the air gap. When all occurring motor losses are considered to be external to the motor, it follows from the law of conservation of energy, that any added electrical energy \((dW_{\text{el}})\) must be converted to magnetic field energy \((dW_{\text{mag}})\) and output mechanical energy \((dW_{\text{mech}})\) [177]:

\[ dW_{\text{el}} = dW_{\text{mag}} + dW_{\text{mech}}. \]  
\[ (1.13) \]

From above equation, the axial attractive force \((F_z)\) and electromagnetic torque \((T_{\text{el}})\) can be derived as [177,179] the derivative of the magnetic field energy stored in the air gap volume \(V_{\text{gap}}\) with respect to the gap length and the angular rotor position \(\Theta\):

\[ F_z = \frac{dW_{\text{el}}}{dl_{\text{gap}}} \]  
\[ (1.14) \]

\[ T_{\text{el}} = \frac{dW_{\text{el}}}{d\Theta} \]  
\[ (1.15) \]

with the stored magnetic field energy

\[ W_{\text{mag}} = \iiint_V \frac{B_{\text{gap}}^2(r, \phi, z)}{2\mu_0} r \, dr \, d\phi \, dz. \]  
\[ (1.16) \]

While a general approximation approach to calculate the air gap flux density was shown in the previous section, in praxis the assumptions made are invalid due to the presence of leakage flux and saturation effects in the ferromagnetic core materials. Therefore, analytical approaches based on Maxwell’s equations to calculate the magnetic fields with reasonable accuracy rapidly gain in complexity [187,188]. Specifically with a comparatively large gap length, as is typical in rotary blood pumps, the influence of leakage flux is large. Therefore,
electromagnetic field calculations with high accuracy are typically performed with numerical simulation methods, as outlined in the following section.

1.3.6.4 Numerical Modelling of AFPM Machines

Various techniques are used in the modelling of electrical machines. When the macroscopic machine parameters are known, the basic operating characteristics of a synchronous machine can be modelled with a simple electrical equivalent circuit. The necessary parameters are usually determined from measurements of the machine under operating conditions. However, to predict machine behaviour for a given geometry, detailed knowledge of the magnetic field distribution, which is described by Maxwell’s equations, is required. Due to the nature of the mathematical problem, the complexity of machine geometries, and the nonlinear characteristics of the materials used, the equations describing the field distributions are inherently complex. While simplifying assumptions allow the formulation of analytical equations, the size of the problem typically requires numerical methods to find a solution of the mathematical problem description.

An advanced simulation technique to find a numerical approximation of the electromagnetic field solution is given by the finite element method (FEM), which is described in detail in literature [189,190].

For this technique, the evaluated geometry is divided into a discrete set of typically triangular (or tetrahedral in 3-dimensional problems) geometry elements, which are defined by a geometrical grid (mesh) and a corresponding set of grid nodes. The magnetic field solution within the resulting finite elements is described as a linear or piecewise polynomial interpolation between the nodes [190,191]. A global solution of the magnetic field distribution is found by minimising an energy functional over the evaluated space, while electromagnetic forces are computed applying methods such as the eggshell method and Maxwell’s stress tensor [189,190,192]. FEM simulation models are often combined with analytical models, to predict the motor losses and efficiency (as will be introduced in chapter 3).

The accuracy of the field solution can be improved by increasing the number of elements; however, this increases the computational cost and can quickly result in high expenditure of time. Therefore, as an alternative to computation-intensive 3D-simulation models, field
distributions for radial flux machines are often calculated in a cross-sectional 2D plane and projected to the full machine length, which allows for considerably reduced computational expense.

While a sufficiently accurate approximation of many electromagnetic field problems can be obtained this way, the technique is not directly applicable to any geometry. In contrast to RFPM machines, the main flux paths in AFPM machines is not confined to a two-dimensional plane, thus 3D simulations or quasi-3D approximation methods are required to obtain an accurate field solution. Figure 1-25 shows an example of a two-dimensional approximation of the three-dimensional field problem (quasi-3D approximation) [21]. A 2D computation plane along the circumference of the motor is selected and transformed to a two-dimensional field problem. However, due to the curvature of the motor, the field distribution is different at different radii.

Therefore, the use of multiple computation planes allows the combination of the results obtained for multiple machine slices to improve accuracy [21,181,193].

Further, the leakage flux near the inner and outer motor radii reduces the air gap flux density (edge effect), thus – depending on the geometry – may have substantial influence on the generated force and torque. While Kurronen et al. [193] found the edge-effect to not cause a large error in their computational model, this may not be true for all geometries. Specifically, in motor geometries with large air gaps as common in RBPs, the edge effect may become significant. Therefore, further efforts have been made to include consideration of the edge effect.

Figure 1-25 – Transformation of the AFPM machine geometry to a 2D plane [21].
Sung derived a function to scale the field results depending on the radius of the obtained 2D solution [194]. The function is based on the Gauss distribution and scales the air gap flux density with values between zero and one. The function values were derived from the field solution obtained with a 3D-FEM simulation model. A similar approach was presented in [22], whereas the scaling function was derived analytically.

Numerical motor models are widely used to study the influence of geometry parameter variations on performance characteristics and optimise a given geometry for a specific application. Evaluated parameters may, for example, include the size and shape of permanent magnets and stator slots, gap length, radial and axial dimensions, as well as the winding topology and number of poles [21,184,195]. Parameter variations are often performed manually to study the isolated influence of specific parameters [184,196–198], while a more complex approach is the design optimisation using optimisation algorithms [195,199,200]. Mahmoudi et al. combined the genetic algorithm (GA) and FEM to design a 1 kW double sided one-stator-two-rotor (TORUS) motor. Pohlmann et al. used the differential evolution algorithm (DA) to optimize the relationship of torque and copper loss of a single-sided axial flux motor drive for a rotary ventricular assist device [195]. The algorithm was constrained within geometrical size constraints of the motor and a maximum axial force of 9 N to be compensated by a spiral-groove bearing. The geometry was optimised without consideration of frequency-dependent eddy current losses in the stator core, to reduce the required computational time. The resulting optimised geometry was later superseded with a manually adjusted geometry exhibiting a higher overall efficiency [179]. Consequently, while automated iteration strategies like the DA algorithm may lead to an optimal solution, they should be combined with a manual approach to investigate the effects of parameter changes and verify the optimized solution.

1.3.6.5 Motor Control Schemes

Depending on the intended application, multiple control strategies exist for brushless permanent magnet drives. In the case of a pulsatile rotary blood pump motor, precise and dynamic control and drive efficiency are major requirements. Control algorithms, which are able to meet these requirements, typically depend on instantaneous knowledge of the rotational rotor angle, to supply the stator phases with the correctly commutated currents to
maximize the efficiency and dynamic response of the motor. In praxis, the rotor angle can be measured using one or multiple sensors, based on optical measurements, magnetic reluctance changes, or the hall effect. However, due to the increased cost and complexity and reduced reliability related to the integration of suitable sensors in a RBP design, sensor-less position estimation methods are the most popular choice in RBP drives [182,201]. The two most widely used control schemes are briefly described and compared in the following.

1.3.6.5.1 Field Oriented Control of Permanent Magnet Synchronous Motors
The philosophy of the field oriented control (FOC) algorithm aims to perform dynamic real-time torque and speed control, while operating with the maximum torque per ampere (MTPA) stator current [202]. In the RBP typical case of a surface-mount permanent magnet machine, the maximum torque is generated when the magnetic rotor and stator fields are offset by an angle of $\Theta_{el} = 90^\circ e_l$. The aim of the FOC is therefore to generate a stator field, which is oriented perpendicular to the field generated by the rotor permanent magnets and thus allow minimisation of the required stator current to generate a demand torque and consequently minimise the occurring Ohmic losses in the stator winding. However, as the magnetic field components generated by the three phase motor currents are superimposed and thus coupled, the direct control of the magnetic field through the three phase stator currents proves to be difficult. Therefore, mathematical decoupling of the current components is performed by the Clarke transformation, which transforms the three-phase currents $(I_u, I_v, I_w)$ into a virtual equivalent two-phase system $(I_\alpha, I_\beta)$, which allows representation of the stator currents by two $90^\circ$ phase shifted alternating currents

$$\begin{bmatrix}
I_\alpha \\
I_\beta
\end{bmatrix} = \frac{2}{3} \begin{bmatrix}
1 & -\frac{1}{2} & -\frac{1}{2} \\
0 & \sqrt{3}/2 & -\sqrt{3}/2
\end{bmatrix} \begin{bmatrix}
I_u \\
I_v \\
I_w
\end{bmatrix},$$

with

$$I_u + I_v + I_w = 0$$

follows:

$$\begin{bmatrix}
I_\alpha \\
I_\beta
\end{bmatrix} = \begin{bmatrix}
1 & 0 \\
1/\sqrt{3} & 2/\sqrt{3}
\end{bmatrix} \begin{bmatrix}
I_u \\
I_v
\end{bmatrix}.$$

Due to the interconnection of the three phase windings, their sum is zero, thus the current vector can be described with two quantities without loss of information.
Subsequently, a second transformation is applied to transfer the two-phase quantities to a rotating $dq$-reference frame, which is defined with the time-dependent angle $\Theta(t)$ between the rotating $dq$- and the stationary $\alpha\beta$-coordinate systems as per the equation:

\[
\begin{bmatrix}
I_d \\
I_q
\end{bmatrix} = \begin{bmatrix}
\cos \Theta(t) & \sin \Theta(t) \\
-\sin \Theta(t) & \cos \Theta(t)
\end{bmatrix},
\]

(1.18)

The rotating reference frame is aligned with the rotor field vector, therefore, in the case of a permanent magnet synchronous machine, $\Theta(t)$ is equivalent to the (electrical) displacement angle between the rotor and stator. The coordinate frames and their relation to the rotor ($\Psi_R$) and stator ($\Psi_S$) flux linkage vectors are illustrated in Figure 1-26. As the $dq$-reference frame is rotating with the stator frequency, the transformed currents $I_d$ and $I_q$ corresponding to a symmetrical, sinusoidal three-phase system are DC quantities, which are independent of frequency. Due to the alignment of the $d$-axis with the rotor flux vector $\Psi_R$, the quadrature current $I_q$ is directly proportional to the generated drive torque, therefore the torque can directly be controlled through adjustments of $I_q$. Conversely, the current $I_d$ (direct current) does not generate torque, thus it is typically regulated to zero, to reduce copper losses. Subsequently, the Park and Clarke transformations are reversed to calculate the required phase voltages in the $uvw$-reference frame.

![Diagram](image)

Figure 1-26 – Illustration of the rotating $dq$-reference frame in relation to the three phase windings (left) and the corresponding vector diagram illustrating the location of the rotor ($\Psi_R$) and stator ($\Psi_S$) flux vectors in relation to the 120° phase shifted three-phase winding system, the stator oriented orthogonal two-phase $\alpha\beta$-reference frame, and the rotating field oriented $dq$-coordinate system.
To perform the Park and inverse Park transformations, instantaneous knowledge of the rotor angle is necessary. To avoid the necessity for positions sensors, the angle can be estimated utilizing sliding-mode or flux observer techniques [203–206]. These techniques allow estimation of the induced back-EMF voltage or flux linkage components based on mathematical models of the electrical stator circuit, and subsequently derive the position of the flux. As the back-EMF voltage reduces to zero at motor standstill, both methods require rotation of the rotor, before the angle $\theta(t)$ can robustly be estimated. Therefore, the motor start-up is typically performed in an open-loop mode, where a slowly accelerating rotating stator field is generated with a comparatively high current. Due to the magnetic coupling provided by the rotor permanent magnets, the rotor stays in alignment with the stator field. Once the rotor is at a sufficiently high speed to allow position estimation, the field-oriented control mode is activated and the motor speed is controlled through adjustments of $I_q$, utilizing a cascaded speed controller.

### 1.3.6.5.2 Trapezoidal BLDC control

An alternative, widely used method to control brushless permanent magnet motors is the trapezoidal brushless DC control scheme. For this algorithm, only the two of the three motor phases providing the highest torque are energised at each point in time. Consequently, one of the energised phases carries a positive current, which is returned as negative current through the second energised phase.

This allows to directly control the DC supply current (hence the name brushless DC motor), which is alternatingly commutated between the three motor phases, resulting in block shaped phase current waveforms. Typically, this control algorithm is applied for motors, which exhibit a rather trapezoidal shape of the

![Figure 1-27 – Phase current commutation of a brushless DC motor with trapezoidal control, showing the phase back-EMF waveforms $(E_u, E_v, E_w)$ and the corresponding phase currents $(I_u, I_v, I_w)$.](image-url)
induced back-EMF voltages \( (E_u, E_v, E_w) \), which is a function of the stator winding and permanent magnet configuration and shape [207,208]. The characteristic phase current commutation scheme and the corresponding back-EMF waveforms are shown in Figure 1-27. The commutation of the phase current occurs at six discrete commutation points, corresponding to rotor angles of \( \Theta(t) = 60, 120, 180, 240, 300, \) and \( 360^\circ \). The third, unenergised phase allows direct measurement of the corresponding back-EMF voltage, thus the zero-crossing of the waveform can be detected. In combination with the knowledge of the rotor speed, which is derived from the supply frequency, the zero-crossing detection allows to estimate the commutation angles, at which the phase sequence is switched to the subsequent state, and a different set of windings is energised [209]. While the trapezoidal control algorithm allows robust speed control and reduced complexity compared to FOC, it has limited applicability for motors exhibiting a sinusoidal back-EMF waveform shape, as it results in a torque ripple of approximately 15\% and reduces the torque and efficiency of the drive [207,210].

### 1.3.7 Speed Profile Optimisation

Few efforts have been made to numerically optimise speed profiles for RBP. However, it has been shown, that ventricular unloading, aortic valve opening, and arterial pulsatility are competing objectives with respect to the phase shift between ventricular contractions and applied VAD speed profiles [11,164]. Based on this finding, Amacher et al. analysed these objectives in a numerical study [165]. A model of the VAD supported systemic circulation was developed, and a physiologically motivated objective function was introduced. The objective function was constructed from the weighed sum of the flow through the aortic valve and the ventricular stroke work. As such, through adjustment of a weighing factor, it served as a performance index for ventricular unloading, aortic valve opening, or a compromise between the two. A nonlinear program was then implemented, to find the VAD speed profile, which optimised the performance index applying methods of trajectory optimisation. The study illustrated, that both performance metrics (ventricular unloading and aortic valve flow) were substantially improved with the optimised speed profiles compared to constant speed operation and sinusoidal speed modulation.

A similar study to optimise rapid speed modulation profiles for rotary TAH has not been performed to date, however, the results of Amacher’s study suggest, that a similar
methodology may be applied to optimise potential performance trade-offs such as arterial pulsatility and motor power consumption. Consequently, such a study may allow evaluation of the capability of RBP to generate arterial pulsatility, and identify the ensuing increase of the power requirement, as well as potential device limitations, which prevent RBP from generating truly physiologic pulsatility.

1.3.7.1 Trajectory Optimisation

Trajectory optimisation is a summarising term for a multitude of methodologies for optimising the control input to a system with respect to a mathematical performance measure and a set of constraints. The system dynamics may be described by a system of nonlinear differential equations

\[
\dot{x} = f(x(t), u(t), t), \quad t_l \leq t \leq t_F
\]  

(1.19)

where \(x(t)\) is the \(n_x\)-dimensional state vector and \(u(t)\) the \(n_u\)-dimensional control input vector. The system may be subject to constraints such as initial and/or final conditions and simple bounds on the state and control variables, as well as dynamic path constraints, which are typically defined as integral functions of the state variable trajectories in the considered time interval.

With the introduction of a mathematical performance index – the objective function – an optimal control problem (OCP) can be defined [211], which aims to find the control input \(u(t)\), which minimises the objective function

\[
\min_x \quad J(x(t), t),
\]  

(1.20)

subject to the state equations

\[
\dot{x} = f(x(t), u(t), t)
\]  

(1.21)

and a set of boundary conditions

\[
\psi(x(t_F), u(t_F), t_F).
\]  

(1.22)

To solve the OCP, it is typically converted to a nonlinear program (NLP), where the state and control variables are discretised and represented by piecewise polynomials, which are iteratively adjusted to find a solution to the state equations, which satisfies the mathematical
conditions of optimality. As such, essentially all practical numerical methods to solve the optimal control method based on newton's method in several variables [211,212] to find a solution satisfying the necessary condition

\[ g(x) = \nabla_x J = \begin{bmatrix} \frac{\partial J}{\partial x_1} \\ \frac{\partial J}{\partial x_2} \\ \vdots \\ \frac{\partial J}{\partial x_n} \end{bmatrix} = \mathbf{0}. \] (1.23)

A variety of methods is available, which can be loosely categorised as indirect (optimise, then discretise) and direct (discretise, then optimise) methods [213]. Examples are shooting methods, which are based on the forward-simulation and iterative adjustment of the control trajectory, and transcription methods, for which the entire time trajectory of the system is discretised, transformed to a set of constraint equations, and optimised simultaneously [213]. For an in-depth overview of different available optimisation methods, the reader is referred to the literature [211,212,214,215]. However, the direct transcription method, which, due to its robustness [212], was applied in this study, is introduced in more detail in chapter 0.

### 1.3.8 Summary

The previous section provides an in-depth review of the relevant literature for the following research studies. An overview of the human heart and cardiovascular system was given, and the state-of-the-art of mechanical circulatory support systems was introduced. With respect to the research topic, the role and benefits of the cardiovascular pulse, as well as adverse effects potentially related to a lack of pulsatility were discussed, and previous research concerning rapid RBP speed modulation was reviewed. Finally, relevant engineering disciplines and methods with respect to the modelling and design of axial flux motors for centrifugal RBPs, numerical trajectory optimisation, and modelling of the cardiovascular system were reviewed. In the following chapter, theoretical implications of the dynamic processes during pulsatile RBP operation are presented, and the performance envelope of a RBP operated with rapid speed modulation is explored on the example of the BiVACOR TAH.
2 Performance Limitations of RBP Operated with Rapid Speed Modulation

Rotary blood pump (RBP) speed modulation to produce variations in outflow has been the topic of numerous research studies in the past decades. However, most of these studies were evaluating rotary left ventricular assist devices (LVADs) and their interaction with the failing native heart, while the number of experiments with rotary total artificial hearts (TAHs) is significantly lower. Amongst studies investigating pulsatile rotary TAH operation, the successful recreation of physiologic pressure and flow patterns has not been reported. During the native pulse cycle, the aortic haemodynamics undergo rapid changes over wide ranges of flow rate and pressure. Rapid modulation of RBP impeller speed as an approach to recreate these dynamic conditions appears promising, however, it is associated with increased capacity requirements of the device speed, torque, power, and suspension force.

This chapter introduces the concepts and theory involved with rapid speed modulation of a rotary blood pump impeller to produce physiological pressure and flow patterns, while using the initial experience with the BiVACOR TAH as an example. As such, in the context of the thesis, this chapter builds the foundation for the research studies presented in the subsequent chapters. First, a collection of theoretical considerations and expectations with respect to RBP design requirements for pulsatile outflow operation is given, and potential design challenges are discussed. Subsequently, the discussed expectations are developed in an in-vitro scoping study, where the performance of the BiVACOR V2 TAH (the latest iteration of the device at the time of commencement of candidature) is evaluated. The limitations observed for the evaluated device are presented, challenges are identified, and potential approaches to improve device performance using hypotheses derived from literature and theory are discussed to build the foundation for the research studies presented in the subsequent chapters.

2.1 Aim

The aim of this chapter is to provide an improved understanding of the dynamic mechanisms influencing and limiting performance and efficiency of RBP operated with speed modulation, with a focus on centrifugal pumps. Design factors limiting the pump performance are
presented, and approaches to improve device design with respect to haemodynamic output through pulsatile speed modulation are discussed. Specifically, the objectives are as follows:

- Establish a theoretical understanding of influential factors on dynamic device performance.
- Evaluate the pulsatile performance of an early device iteration of the BiVACOR TAH.
- Investigate mechanisms of additional power loss due to rotor acceleration.
- Identify present device limitations, influential design parameters, challenges, and approaches to improve pulsatile performance.

2.2 Theoretical Implications of Dynamic RBP Speed Modulation

2.2.1 Native Pressure and Flow Gradients

To establish an initial understanding of RBP requirements for the recreation of a physiologic pulse, the waveforms and time derivatives of the native aortic haemodynamics are introduced here.

![Waveforms](image)

Figure 2-1 – (A) Aortic pressure $P_{ao}$ and (B) systemic flow rate $Q_s$ waveforms generated by the native heart, and the corresponding time derivatives (C) $dP_{ao}/dt$ and (D) $dQ_s/dt$. Waveforms adapted from [216]. Dashed black lines in (A, B) indicate a linearization of the maximal slopes $dP/dt$ and $dQ/dt$ respectively.
Figure 2-1 shows the human aortic pressure ($P_{ao}$) and systemic flow rate ($Q_s$) waveforms in a patient undergoing surgery, reported by Patel et al. [216]. The waveforms were reconstructed utilizing the pressure and flow harmonics provided in the publication text. The waveforms show that both aortic pressure and flow increase almost linearly during systole, exhibiting maximum derivatives of $dP_{ao}/dt_{max} = 571.7 \text{ mmHg/s}$ and $dQ_s/dt_{max} = 283.6 \text{ L/min/s}$ respectively, as indicated by the dashed black lines in Figure 2-1A and B. These waveforms were utilized to illustrate an estimation of influential design parameters and requirements for the design of a RBP in the following example.

2.2.2 Dynamic Pump Operation

The performance and efficiency of a rotary pump depend on several characteristics of the pump hydraulics, driving motor, impeller suspension, and the interaction amongst these systems. Power loss due to inefficiencies in all systems, as well as in the control electronics are unavoidable independent of the operating mode. However, an increase of losses is expected when the pump is operated with rapid speed modulation, which dynamically changes the operating point of the system. Since a rotary pump’s motor and hydraulic systems are characterised as having a best efficiency at one operating point, additional power loss is expected when accelerating and decelerating the RBP impeller compared to constant speed operation. The following section gives an introductory overview of the dynamic operation of hydraulic and motor systems during rapid impeller speed modulation.

2.2.2.1 Hydraulic Characteristics

The operating point of the hydraulic system is given by the interrelation of pump pressure head, flow rate, efficiency, and motor load torque. When a RBP is operated at constant speed in absence of ventricular contraction, the steady state operating point is determined by the pump speed and the hydraulic load, i.e. the vascular resistance. Figure 2-2 shows an example of the pressure head- flow (HQ) characteristics of an early prototype of the BiVACOR device (BiVACOR V1). The blue loop represents the pressure and flow corresponding to the waveforms of the natural heart shown in Figure 2-1. A constant left atrial pressure of $P_{la} = 8 \text{ mmHg}$ was assumed and subtracted from the aortic pressure waveform to represent the pressure head. The two highlighted HQ-curves at $n_1 = 1475 \text{ rpm}$ and $n_2 = 3400 \text{ rpm}$ show the maximum and minimum operating speed curves, which intersect with the native heart loop.
The black line in Figure 2-2A represents the hydraulic load line corresponding to the mean pressure head and flow rate. The hydraulic load was calculated from the mean values of pressure head and flow during the given native cardiac cycle (red $x$-marker). In the following,
a step speed change of the RBP is considered. When the pump speed is changed, the HQ-operating point follows a dynamic trajectory, which is mostly determined by the characteristics of the arterial Windkessel, the fluid inertia within the pump and corresponding cannulae, and the speed and acceleration profile of the pump motor. Further, due to the increased flow rate, the atrial pressure decreases. Therefore, the pressure head \( H_p = P_{ao} - P_{la} \) increases, and consequently, due to the negative gradient of the HQ-characteristic, the flow rate decreases. When the RBP then reaches the new steady-state speed, the operating point transitions towards the intersection of the HQ-characteristic corresponding to the new pump speed, and the hydraulic load characteristic. Due to the capacitive (compliant) character of the aorta and atrium, the changes of the pressure head are slow and lagging in phase when compared to the flow waveform.

When rapid speed modulation is considered to generate desired outflow waveforms, it is obvious, that, in order to generate a trajectory similar to the exemplar native heart trajectory with a RBP, rapid acceleration of the impeller is required, which results in a rapid increase in flow, followed by a rapid increase in arterial pressure. Further, the required speed profile resulting in a flow waveform comparable to the physiologic pulse is unknown, hence a dedicated consideration of the impeller speed control is necessary (see chapters 0 and 6). A simplified estimate of the required acceleration can hereby be calculated from the linearization of the systolic flow rate gradient (see Figure 2-1B):

Assuming a linear increase of the flow rate from 0 \( L/min \) to 20 \( L/min \), the time interval for the acceleration can be approximated as

\[
\Delta t = \frac{\Delta Q}{dQ/dt_{max}} = \frac{20 \ L/min}{283.6 \ L/min/s} = 70.5 \ ms
\]  

(2.1)

With the given minimum \( (n_1) \) and maximum \( (n_2) \) speeds, which intersect with the native heart pressure-flow loop in the HQ-plane, the order of magnitude of the required acceleration can be estimated as

\[
\frac{dn}{dt} \approx \frac{\Delta n}{\Delta t} = \frac{3400 \ rpm - 1475 \ rpm}{70.5 \ ms} = 27,294 \ rpm/s.
\]  

(2.2)

During the speed modulation cycle, a wide range of hydraulic efficiencies ranging from \( \eta_{hyd} = 0 \) to the best efficiency of the pump can be expected (Figure 2-2B), which is directly
related to the hydraulic power output and the hydraulic load torque $T_{hyd}$ (Figure 2-2C). The graphs show, that the torque substantially increases with both speed and flow rate and the steady state characteristics indicate, that the instantaneous hydraulic torque can be substantially larger than the steady-state torque (red x-marker in Figure 2-2C) at the mean flow rate. Consequently, a high drive torque capacity is required, whereas the increased load may result in excessive additional power loss within the motor (section 2.2.2.2); the motor needs to be upsized to accommodate this and designed around a different maximum efficiency point cf. steady state design.

The considerations presented here provide a first estimate of the required acceleration and load torque for these pump characteristics. However, due to the fluid inertance within the pump cavity and cannulae, the actual pump flow exhibits a phase lag to the rotor speed, which may result in a dampened flow response and consequently reduce the maximum $dQ/dt$ and $dP/dt$. Further, the decreasing atrial pressure results in higher required pressure head and, consequently, pump speed. A more detailed 1D-modelling approach of the RBP flow dynamics in interaction with the cardiovascular system is discussed in chapter 0.

2.2.2.2 Motor Characteristics

Contemporary RBPs are typically equipped with permanent magnet (PM) brushless motors. The mechanical equation describing the motor speed dynamics is given in equation (2.3).

$$\frac{dn}{dt} = \frac{60}{2\pi J_{rot}} (T_{el} - T_{Load}).$$

(2.3)

It can be observed, that the rotor acceleration $\left(\frac{dn}{dt}\right)$ is directly related to the difference between the generated electromagnetic torque $T_{el}$ and the load torque $T_{Load}$, as well as to the moment of inertia of the rotor ($J_{rot}$). In the case of a RBP motor, the load torque is approximately equal to the hydraulic torque $T_{hyd}$, when friction losses due to mechanical contact bearings or viscous losses from hydrodynamics bearings are neglected. Therefore, when the speed of a RBP is increased, the acceleration capacity decreases during the acceleration process, as the hydraulic load torque increases.

In combination with the assumptions for the load torque and acceleration in the previous section, equation (2.3) can be applied to calculate a rough estimate of the required electromagnetic torque. For the exemplary calculation, a rotor inertia of $J_{rot} = 21.93 \cdot$
$10^{-6} \text{ kg} \cdot \text{m}^2$ and a mean hydraulic load torque of $T_{\text{hyd}} = 0.06 \text{ Nm}$ are assumed (based on the BiVACOR V1 device, cf. Figure 2-2C), therefore the required torque can be calculated according to equation (2.4).

$$T_{el} = \frac{d}{dt} \left( \frac{2\pi n}{60} \right) \cdot J_{rot} + T_{\text{hyd}}$$

$$= \frac{2\pi \cdot 27,294}{60} \cdot s^{-2} \cdot 21.93 \cdot 10^{-6} \text{ kg} + 0.06 \text{ Nm}$$

$$= 0.122 \text{ Nm}$$

Although significant simplifications were applied, the estimation illustrates, that the required motor torque for physiologic pulsatile operation may significantly exceed the torque requirement corresponding to an equivalent continuous flow, in this case by more than $0.1 \text{ Nm}$ from approximately $0.02 \text{ Nm}$. The electrical power consumption is determined by the motor efficiency, which describes the relationship between the electrical power intake ($P_{el}$) and the mechanical power output ($P_{\text{mech,mot}}$) (equation (2.5)), whereas the electrical power is the sum of mechanical power and the losses in the motor ($P_{\text{loss,mot}}$).

$$\eta_{\text{mot}} = \frac{P_{\text{mech,mot}}}{P_{el}} = \frac{P_{\text{mech,mot}}}{P_{\text{mech,mot}} + P_{\text{loss,mot}}}$$

The motor losses are comprised of losses in the copper windings, as well as frequency dependent hysteresis and eddy current losses in the ferromagnetic core materials and permanent magnets. An example of an efficiency map of an axial flux brushless RBP motor is shown in Figure 2-3. The motor corresponding to the data shown in the figure was designed for a constant speed operating point of $n = 2500 \text{ rpm}$ at a load torque of $T_{\text{Load}} = 0.012 \text{ Nm}$. The map shows, that the efficiency around the design point is high with values around 70% to 80%. However, when the torque is increased, the efficiency rapidly drops, as the copper losses quadratically increase with the torque requirement [217]. This is specifically true for lower operating speeds, at which the maximum torque for acceleration is required during the pulse cycle. Consequently, considering the previous estimates of the relative increase of the hydraulic load torque, it is easy to predict that the instantaneous motor efficiency may significantly reduce during a dynamic speed modulation cycle. Therefore, the relative influence of the loss mechanisms in brushless PM motors is substantially dependent.
on the motor design. Specifically, the eddy current and hysteresis losses are dependent on the utilized core materials, and the magnitude and harmonic content of the time-varying flux density in the core and PM.

The copper losses depend on the motor current and the coil resistance, which varies with conductor material, wire cross sectional area, wire length per turn, and number of turns. While copper losses are not directly related to the flux density or frequency, the magnetic coupling between the stator and rotor determines the required stator current at each operating point, and thus influences copper loss. Specifically, a characteristic of the magnetic design, which is commonly shared between many different RBPs, is a comparatively large magnetic gap, which results from the requirement to incorporate a hermetically sealed housing, fluid paths, impeller vanes, and/or a hydrodynamic suspension within the gap [217,218]. Consequently, many RBP drive designs may not exhibit sufficient torque to generate a physiologic pulse with rapid speed modulation, as they are designed for continuous flow operation, which tends to result in increased copper losses due to a higher current requirement or a large winding turn number; or require large permanent magnets to provide sufficient magnetic coupling between the rotor and stator. While the latter may increase the torque constant, it also increases the rotor weight and consequently inertia, and thus negatively affect rotor acceleration and increase the required rotational energy at higher impeller speeds. Further, in case of an axial flux motor, it may increase the axial attractive force between rotor and stator imposing an increased load on the rotor bearings.

Figure 2-3 – Example of an efficiency map of an axial flux brushless RBP motor for various speed and torque operating points. Figure ©2011 IEEE [217].
A detailed discussion of design and control considerations for axial flux permanent magnet motors, and implications with respect to pulsatile RBP speed modulation is given in chapter 3.

2.2.2.3 Suspension Force

Further consideration is required with respect to force capacity of the rotor suspension system. Like the hydraulic torque, the magnitude and direction of hydraulic forces acting on a RBP impeller depend on the pump operating point. Therefore, speed modulated operation may result in significant perturbation forces loading the bearings suspending the rotor. Specifically, in the case of hydrodynamic or magnetic bearings, this may potentially lead to bearing failure and intermittent rotor touchdown, causing wear, excessive blood shear stress and consequently blood cell damage.

In centrifugal pumps, static axial thrust is caused by the difference between the high discharge pressure acting on the impeller underside, and the distributed inlet to outlet pressure acting on the impeller front \([102,219]\), thus changes of the thrust can be expected during a pulse cycle. Additionally, changes of the magnetic field in the air gap of the motor may add a dynamic component to the static force magnetic characteristic. In many radial flux motors, due to motor symmetries this may lead to rotor deformation and vibration; however, electromagnetic forces acting on the rotor of axial flux drives may directly act on the rotor suspension.

Axial hydraulic forces depend on the pump geometry and fluid viscous and inertial effects, and are often difficult to predict analytically, while electromagnetic forces in axial flux drives additionally depend on the applied control strategy. Therefore, an experimental evaluation of the dynamic forces is beneficial to ensure sufficient bearing capacity and stable rotor suspension.

2.2.3 Design Implications

The simplified analysis of the dynamic hydraulic requirements in section 2.2.2 showed, that, with the given hydraulic characteristics, substantial pump speed changes would be required to generate a flow swing similar to that of the native heart. That is, as the hydraulic characteristics exhibit a comparatively low pressure sensitivity (i.e. the HQ-curve shape is steep). Consequently, it may be hypothesized, that a more pressure sensitive (flatter) pump
characteristic is advantageous to reach high flow rates of up to 20 $L/min$ and above, as a lower relative speed change is required to cover the entire flow range during the pulse cycle.

Figure 2-4 – Example of the pump characteristics for two centrifugal pumps (CP) with different pressure sensitivities at two different speeds.

Figure 2-4 qualitatively shows the HQ-characteristics of two centrifugal pumps at two different pump speeds. The pump with the higher pressure sensitivity (CP2) covers a wider flow range with the same relative increase of the pump speed. The curves for both pumps correspond to the same speed ratio $\frac{n_{CP2.2}}{n_{CP1.1}} = \frac{n_{CP2.2}}{n_{CP1.1}}$.

Figure 2-5 – Effect of changes in (A) the inlet diameter (ID) and (B) impeller vane height (b12) on the pressure head-flow characteristic of a centrifugal pump [220].

The pressure sensitivity of centrifugal pumps depends on various design parameters and is subject to different fluid-dynamic effects within the pump, volute, and inlet and outlet conduits. An in-depth discussion of geometrical pump designs is outside the scope of this
work. However, it can be observed, that some geometrical pump characteristics resulting in an increased pressure sensitivity may be in direct contrast with favourable characteristics of an axial flux drive as described in the next few paragraphs. Figure 2-5 as an example shows changes in the pressure head-flow characteristics of a centrifugal pump, when the inlet diameter and the axial blade height (Figure 2-5B) are varied [220]. Figure 2-5A shows an increase of the pressure sensitivity with increasing inlet diameter, caused by a reduction of the pump shutoff pressure at low pressures due to additional recirculation loss at the inlet, and a reduction of friction loss at high flows. Both characteristics may be beneficial for pulsatile operation, as the reduced shutoff pressure head increases the required minimum speed to decrease the pump flow rate to $0 \, L/min$, while the reduced friction loss reduces the maximum speed. Further, a high inlet resistance caused by a small inlet diameter may reduce the pressure at the impeller eye and increase axial thrust particularly at high flows and increase the likelihood of cavitation, which may be mitigated with the choice of an increased inlet size.

It may further be hypothesized, that large flow paths through the pump are generally beneficial to reduce the characteristic resistance and thus pressure drop, leading to improved dynamics specifically at high flow rates. Figure 2-5B shows a substantial increase of the pressure head and the pressure sensitivity with the axial impeller vane height, which may allow reduction of the pump speed at both minimum and maximum flows during a pulse cycle.

Adjustments of both the vane geometry and the inlet diameter may be beneficial, however, considering a centrifugal pump design, where an axial flux motor is located around the pump inlet, these changes may significantly restrict the motor geometry. The use of thin blades enables a large cross-sectional area of the flow path within the pump. However, specifically when the impeller blades are too thin to incorporate sufficient permanent magnet material (in contrast to [185]), the motor permanent magnets are located axially below the impeller vanes, thus an increase of the vane height corresponds to an increase of the magnetic gap. Conversely, an increase of the inlet diameter restricts the minimum inner stator diameter. Both restrictions may have detrimental effects on motor performance or require adjustments of the stator and/or rotor geometry to maintain motor capacity and efficiency. Equally, changes in the rotor geometry may affect the rotor inertia and consequently acceleration. As
the implications of these design changes are not fully known, a parameter study to evaluate the performance, axial force characteristic, and feasibility of various axial flux motor geometries was performed, and is described in detail in chapter 3. Further, the effect of changes in the hydraulic characteristics were evaluated in chapter 6.

2.2.4 Haemocompatibility

The consideration and evaluation of haemocompatibility (e.g. red blood cell destruction) due to excessive shear forces during the acceleration and high-speed phases of an implemented speed modulation cycle is a key requirement for the development of a feasible pulsatile rotary blood pump. However, due to the various influences corresponding to the hydraulic pump design, and the complexity of their investigation, an evaluation of the haemocompatibility of pulsatile speed modulation modes is outside the scope of this research work. However, it is considered a mandatory next step in future work building up on this thesis.

2.2.5 Summary

The previous sections provide an overview of the expected implications of rapid RBP speed modulation. A rough estimate of design requirements for the development of a RBP providing truly physiologic pulsatile haemodynamics was discussed. Subsequently, to substantiate the presented considerations, speed modulation performance was investigated on the example of the latest BiVACOR TAH prototype at the time of candidature commencement (BiVACOR V2), as described in the following.

2.3 Scoping Experiments

In a scoping exercise to identify characteristics limiting the device performance in pulsatile operation, speed modulation algorithms were applied to the BiVACOR V2 device in vitro and in vivo. Hydraulic and motor losses were estimated in continuous flow and pulsatile operation, and hydraulic axial forces acting on the pump impeller were evaluated. The ‘operating envelope’ extracted from these scoping experiments informed subsequent research which is detailed in the following chapters.
2.3.1 Experimental Methods

2.3.1.1 Mock Circulatory Loop

In vitro evaluations were performed utilizing a previously developed mock circulatory loop (MCL) [98], which was modified for TAH evaluation [99] (Figure 2-6). Systemic and pulmonary circuits were built of rigid, clear polyvinyl chloride (PVC) pipe elements. Atrial and venous compliances were represented by vertical pipes open to the atmosphere; the aortic and pulmonary arterial compliances were each represented by a lumped, compressed air compliance chamber. The inlets of the BiVACOR TAH were connected in close proximity to the atrial and arterial compliance chambers. The device comprises two centrifugal impellers mounted to a common rotor hub to supply blood flow to the systemic and pulmonary circulations. Therefore, both impellers operate at the same rotational speed, while the rotor hub is magnetically levitated within the pump cavity. To accommodate for the difference in the required pulmonary and systemic arterial pressures, the left-sided impeller has a substantially larger diameter than the right-sided impeller. The radial clearance between the spinning impeller and the outer pump casing allow for a shunt flow between the left and right pump. Since the pressures on the left side are generally larger, a residual flow from the left atrium to the pulmonary artery through the device is observed during normal operation.

![Figure 2-6 – Schematic setup of the total artificial heart mock circulatory loop.](image)

AOC, aortic compliance; LA, left atrium; PAC, pulmonary arterial compliance; PVC, pulmonary venous compliance; PVR, pulmonary vascular resistance; \( Q_p \), pulmonary flow meter; \( Q_s \), systemic flow meter; RA, right atrium; SVC, systemic venous compliance; SVR, systemic vascular resistance.
Pressure measurements at each of the TAH inlets and outlets were performed with fluid-filled pressure transducers (PX181B-015C5V, Omega Engineering, Inc., Stamford, CT, USA). Flow rates were measured with two clamp-on ultrasonic flow probes (TS410-20PXL, Transonic Systems, Ithaca, NY, USA). Pulmonary vascular resistance (PVR) and systemic vascular resistance (SVR) were adjusted using pneumatic pinch valves (VMP032, AKO Armaturen & Separationstechnik GmbH, Trebur-Astheim, Germany), which were supplied with compressed air from voltage-controlled pneumatic regulators (ITV2030, SMC Corporation, Tokyo, Japan). After the desired resistance values were set, the pressure supplied to the pinch valves remained constant throughout the pulsing experiment. All measurements were performed at room temperature with a 40%wt glycerol-water solution to mimic the viscosity of blood at 37°C. The MCL was controlled with a rapid control prototypic system (DS1103, dSPACE, Paderborn, Germany), while the recorded data were postprocessed with MATLAB (MathWorks, Natick, MA, USA). Prior to the experiments, the MCL was filled with the working fluid and deaired, while the aortic (AOC) and pulmonary arterial (PAC) compliance chambers were open to atmosphere and the pressure sensors were zeroed. Subsequently, AOC and PAC were closed to atmosphere, and the fluid was added to the MCL until a pressure level of 12 mmHg was reached.

2.3.1.2 Hydraulic Torque and Force Evaluation

To allow estimation of the axial bearing load and hydraulic losses and efficiency during pulsatile operation, hydraulic torque and axial force were evaluated in a previously developed hydraulic force test rig (FTR), which was an updated version of the test rig presented in [221] and [102], applying the identical working principle. The test rig assembly incorporated a shaft driven version of the BiVACOR V2 impeller and the corresponding pump housing, whereas the shaft was rotated with an off-the-shelf 55W brushless DC motor (DT4260-24-055-04, Telco Motion, Houston, TX, USA). The 3D-printed plastic pump housing was mounted to a micrometer stage allowing three-dimensional adjustment of the housing position and consequently the relative position of the impeller within the housing. Hydraulic forces acting on the impeller were transferred to a rigid frame, which was mounted to a six-axis force/torque transducer (Mini40-E, ATI Industrial Automation, Apex, NC, USA), separating the pump housing from the assembly of motor, shaft, bearings and impeller. With this arrangement, the torque around the axis of impeller rotation is transferred through the
magnetic field of the driving motor, hence the measurement reflects the electromagnetic torque generated by the motor. Consequently, while the electromagnetic torque equals the hydraulic load torque in steady state operation, the measurement includes the accelerating torque during impeller speed changes, thus is not suitable for direct measurement of the hydraulic load during pulsatile operation. For all measurements, the impeller was positioned in the axial and radial centre of the pump housing.

The FTR pump was connected to the TAH-MCL to evaluate hydraulic forces under various steady state and dynamic operating conditions. The MCL was filled and the force/torque transducer was zeroed while the motor was at standstill. Initially, the FTR motor was operated with constant speeds to evaluate the steady state hydraulic load torque in various operating points. The pressure head-flow operating points of the left and right pump were adjusted by an automated test procedure, which varied pulmonary and systemic vascular resistances within moderate and extreme physiologic ranges [96]. Subsequently, the test rig was operated with speed modulation profiles with various amplitudes. The objective of these tests was to evaluate the order of magnitude of axial and radial hydraulic forces resulting from pulsatile flows in that version of the device. The MCL was set to a baseline condition corresponding to a human at rest (SVR = 1450 dyn · s/cm$^5$; PVR = 100 dyn · s/cm$^5$) [96]. Square waveforms with a duty cycle of 33% and a beat rate of 60 bpm were applied to the device and superimposed with an (almost constant) mean speed offset. The amplitude of the square waveforms (difference between maximum and minimum speed) was slowly increased from $\Delta n = 0 \text{ rpm}$ to $\Delta n = 1200 \text{ rpm}$ over a time interval of 250 s. A PI controller was used to maintain a mean flow rate of $Q_s = 5 \text{ L/min}$ by continuously adjusting the speed offset, which was added to the waveforms.

2.3.1.3 Motor Evaluation

The performance and efficiency of the axial flux motor driving the device was separately evaluated in a mechanical motor test rig (MTR; Figure 2-7). The test rig comprises a similar shaft-driven rotor suspension assembly as the force test rig. The shaft was connected to a rotor shell containing the rotor yoke and permanent magnets of the axial flux motor. On the far end, the rotor was coupled to an electronically controlled hysteresis brake (H11, Placid Industries, Lake Placid, NY, USA), which is mechanically connected to the shaft suspension and a six-axis force/torque transducer (Mini40-E, ATI Industrial Automation, Apex, NC,
USA) and provided the desired load torque. Similar to the FTR, the force transducer represented the only point of suspension of the rotor assembly. The motor stator was mounted to an adjustable suspension, allowing adjustment of the stator position in axial and radial directions. Data acquisition and load torque control was performed with a rapid control prototypic system (DS1103, dSPACE, Paderborn, Germany) in real-time.

![Figure 2-7 – Motor test bench used for the evaluation of the axial flux motor drive](image)

The gap between the stator and rotor permanent magnet surfaces was set to a relatively large 3 mm, which corresponded to the gap in the evaluated pump prototype, when the rotor is magnetically suspended in the axial centre position and 2.0mm impeller blades are included in the gap. The motor was operated with a motor controller employing a sensor-less field oriented control algorithm, implemented on a microcontroller-based off-the-shelf motor control development kit (DRV8312-C2-KIT, Text Instruments, Dallas, USA). The required knowledge of the rotor angle for the correct application of the stator currents was obtained from a sliding-mode back-EMF observer. The current in the hysteresis brake was supplied with a PWM amplifier (4122Z, Copley Controls, Canton, MA, USA), and feedback controlled utilizing the torque measurement and a PI-controller. The instantaneous motor power consumption \( P_{el} \) was computed from the phase voltage and current measurements obtained by the motor controller and transmitted to the data acquisition system via controller area network (CAN) bus connection. The force/torque transducer was zeroed under absence of the motor stator. The power consumption was evaluated to calculate the motor efficiency.
\((\eta_{\text{mot}})\) in various steady state speed/torque \((n, T_{\text{Load}})\) operating points, according to equation (2.6)².

\[
\eta_{\text{mot}} = \frac{2\pi n \cdot T_{\text{Load}}}{60 \cdot P_{\text{el}}}. \tag{2.6}
\]

The speed was adjusted between \(n = 500 \text{ rpm}\) and \(n = 3000 \text{ rpm}\) in steps of 500 \text{ rpm}, while for each speed, load torques between \(T_{\text{Load}} = 0.01 \text{ Nm}\) and \(T_{\text{Load}} = 0.07 \text{ Nm}\) in steps of 0.01 \text{ Nm} were generated by the hysteresis brake. A load torque of zero could not be evaluated, due to the static frictional torque caused by the shaft bearings.

### 2.3.1.4 Loss Estimation

The motor efficiency and hydraulic load torque data obtained from the FTR and MTR experiments were combined to separately estimate the losses occurring in the hydraulic and motor system. A double-quadratic function of the motor efficiency as function of the speed and power consumption was fitted to the obtained steady state data points to obtain the real-time efficiency of the motor, assuming quasi-steady operation. This assumption is valid, as electrical time constants of the motor are substantially lower than the mechanical and hydraulic time constants. The mechanical motor output power was calculated as the product of the electrical power consumption and the efficiency. The motor loss was calculated from the motor power consumption and estimated motor efficiency \(\eta_{\text{mot,est}}\) (equation (2.7)).

\[
P_{\text{loss,mot}} = P_{\text{el}} - P_{\text{mech,mot}} \approx P_{\text{el}} \cdot \left(1 - \eta_{\text{mot,est}}\right). \tag{2.7}
\]

The hydraulic pump output power was calculated from the measured pressure head and flow data according to equation (2.8), utilizing the systemic \((Q_s)\) and pulmonary flow rates \((Q_p)\), and the pressure heads of the left \((H_L = P_{ao} - P_{la})\) and right \((H_R = P_{pa} - P_{ra})\) pumps, from measurements of aortic \((P_{ao})\), left atrial \((P_{la})\), pulmonary arterial \((P_{pa})\), and right atrial \((P_{ra})\) pressure. The influence of the hub leakage between the left and right pumps on hydraulic power was not considered at this point.

\[
P_{\text{hyd}} = H_L \cdot Q_s + H_R \cdot Q_p. \tag{2.8}
\]

² Note that the factor 60 in the denominator represents the unit conversion of the speed \(n\) from \(\text{min}^{-1}\) to \(\text{s}^{-1}\).
To calculate the hydraulic loss, the mechanical load power of the pump is required ($P_{mech,hyd}$; equation (2.9)). Due to the additional power required to accelerate the rotor ($P_{acc}$), this power component is unequal to the mechanical motor power. Therefore, it was calculated utilizing the hydraulic load torque, which was estimated from a curve fit function ($T_{hyd,2100}(Q_s)$) through single-speed ($n = 2100 \text{ rpm}$) torque data as function of the systemic flow rate, obtained from the hydraulic FTR experiments. For changes of the pump speed, the curve fit function was scaled according to the pump similarity laws (equation (2.10)) [222].

$$P_{mech,hyd} \approx \frac{2\pi n}{60} \cdot T_{hyd,est}.$$  \hspace{1cm} (2.9)

$$T_{hyd,est} = \left(\frac{n}{2100 \text{ rpm}}\right)^2 \cdot T_{hyd,2100}\left(Q_s \cdot \frac{n}{2100 \text{ rpm}}\right).$$  \hspace{1cm} (2.10)

The hydraulic loss ($P_{loss,hyd}$) and efficiency ($\eta_{hyd}$) were estimated as:

$$P_{loss,hyd} = P_{mech,hyd} - P_{hyd}$$  \hspace{1cm} (2.11)

$$\eta_{hyd} = \frac{P_{hyd}}{P_{mech,hyd}} = \frac{P_{hyd}}{P_{hyd} + P_{loss,hyd}}.$$  \hspace{1cm} (2.12)

whereas the total efficiency of the pump is the product of motor and hydraulic efficiencies

$$\eta_{total} = \eta_{mot} \cdot \eta_{hyd}.$$  \hspace{1cm} (2.13)

then power balance is

$$P_{el} = P_{hyd} + P_{loss,hyd} + P_{loss,mot} + P_{rot},$$  \hspace{1cm} (2.14)

where the average power for acceleration is zero ($\bar{P}_{acc} = 0$) for steady state or periodic waveforms.

### 2.3.1.5 Initial Evaluation of Pulsatile Operation of the Levitated Prototype

Initial experiments with pulsatile operation of the device were performed with a magnetically levitated pump prototype in vitro and in vivo.
2.3.1.5.1 *In Vitro*

The magnetically levitated BiVACOR V2 prototype was connected to the previously described MCL. After filling the MCL, the device was operated at a constant speed to generate continuous flow ($CF$) and with a square wave profile (33% duty cycle), resulting in pulsatile flow ($PF$). The pulse speed amplitude was set to the maximum possible speed, at which stable magnetic levitation without radial touchdown could be maintained, while the mean speed was manually adjusted to reach a mean systemic flow rate of $Q_s = 5 \text{ L/min}$ in both $CF$ and $PF$ conditions. Haemodynamic and pump data were recorded with the dSPACE data acquisition system. Pulse pressure ($PP$), surplus haemodynamic energy ($SHE$), maximum $dP/dt$, and hydraulic and motor losses were calculated and evaluated during postprocessing.

2.3.1.5.2 *In Vivo*

In addition to the preliminary in vitro testing, square wave profiles (33% duty cycle) were applied for short periods during a chronic in-vivo study conducted with the BiVACOR V2, and haemodynamics and pump data were compared to the equivalent continuous flow case. The device was implanted in a male Corriente crossbred calf (71.4 kg). Ethics approval was granted by the IACUC (Institutional Animal Care and Use Committee) of the Texas Heart Institute (Houston, Tx, USA) under protocol number 2013-16 (Animal Welfare Assurance number A3505-01). As no staff or students of Griffith University were directly involved in the handling of the animal, ethics approval by the Griffith University Animal Ethics Committee was not required.

Prior to surgery, the animal was anesthetised, intubated, and automatically ventilated. A left lateral thoracotomy was performed at the fifth rib, and the ventricles were removed via cardiectomy above the atrioventricular groove and above the aortic and pulmonary valves, leaving the left and right atrial appendages intact. Atrial cuffs were trimmed and anastomosed to the atrial orifices. Integrity of the suture lines was tested with a leak tester and saline solution. Left and right outflow grafts were end-to-end anastomosed to the ascending aorta and pulmonary artery. Perivascular flow probes (24PAU, Transonic Systems Inc., Ithaca, NY, USA) were placed around the left and right outflow grafts, to monitor systemic and pulmonary flow rates throughout the study. The rotary TAH was placed in the chest and attached to the atrial cuffs and outflow grafts. Pressures in the left and right atria, as well as the ascending aorta and pulmonary artery were measured with
fluid filled pressure transducers (TruWave, Edwards Lifesciences, Irvine, CA, USA). Left and right atrial pressures, and aortic pressure were additionally monitored with pressure catheters (Mikro-Tip®, Millar Inc., Houston, TX, USA). Data recording was performed at a sampling frequency of 200Hz using a PowerLab data acquisition system (ADInstruments, Colorado Springs, CO, USA).

After successful implantation, the rotary TAH was turned on at slow speed. The animal was weaned from bypass, the chest was closed, and the animal was transferred to the ICU to wake up. The study was terminated after 30 days, while pulse speed waveforms were evaluated on postoperative day 8. Similar to the preliminary in vitro study, the speed amplitude was increased within the limitations of stable magnetic bearing operation. A maximum speed pulse amplitude of $n_{amplitude} = 650$ rpm was applied.

2.3.2 Results

2.3.2.1 In Vitro Haemodynamics

An example of pressure, flow, speed, and power waveforms generated in the MCL are shown in Figure 2-8. The motor speed waveform (A) shows that the instantaneous speed reached the systolic target speed, however, overshoot and ringing were observed (which were, outside of this research work, reduced through improvements of the motor controller in later experiments).
Figure 2-8 – Example of pulsatile waveforms with a square wave function generated with the BiVACOR TAH in a mock circulatory loop. The panels show (A) impeller speed and speed target, (B) systemic flow rate, (C) aortic pressure, (D) motor power consumption, (E) left atrial pressure, and (F) aortic $dP/dt$.

Hydraulic torque and motor efficiency measurements were used to estimate the respective mechanical power components (more detailed evaluations of motor and hydraulic characteristics can be found in chapters 3 and 6). Figure 2-9 illustrates the different power components during pulsatile operation. The graphs show the measured electrical motor power consumption and hydraulic output power, and the estimated mechanical motor power and hydraulic load power. The red and blue shaded areas between the curves correspond to the energy loss in the motor ($E_{\text{loss,mot}}$) and the hydraulics ($E_{\text{loss,hyd}}$). Areas shaded in green ($E_{\text{rot}}$) represent the change of rotational energy of the rotor, which increases when the rotor is accelerated ($P_{\text{mech,mot}} > P_{\text{mech,hyd}}$), and decreases when the rotor is decelerated ($P_{\text{mech,mot}} < P_{\text{mech,hyd}}$). The graphs show a steep increase in the electrical motor power consumption during rotor acceleration, whereas the mechanical motor power is significantly lower, i.e. the motor power loss is large. The highest hydraulic power loss corresponds to the highest systolic flow rate.
Figure 2-9 – Estimated and measured power components corresponding to the pulse waveforms obtained in the MCL. The graph shows the electrical motor power consumption ($P_{el}$), mechanical motor power ($P_{mech,mot}$), mechanical impeller load power ($P_{mech,hyd}$), and hydraulic output ($P_{hyd}$). Shaded areas represent the energy loss in the motor ($E_{loss,mot}$) and hydraulics ($E_{loss,hyd}$), and the rotational energy of the rotor ($E_{rot}$).

Figure 2-10 shows the corresponding mean values of all power components. The estimation error $e_{est}$ was calculated as the difference between the motor power consumption and the sum of the measured hydraulic power and the estimated loss components (cf. equation (2.14)). The results show a substantial increase of the motor power consumption from continuous to pulsatile operation (59.6% increase). Although the mean systemic flow rate is the same for both cases ($Q_s = 5 \text{ L/min}$), $P_{hyd}$ increased by 22%. Both hydraulic and motor power loss increased when the pump was operated with $PF$, while the percentage increase of motor losses (120.5%) was substantially larger than the percentage increase of hydraulic losses (34.5%). The errors of the loss estimation were 5.03% and 2.54% of the motor power consumption for $CF$ and $PF$ respectively, whereas the total power loss was underestimated in both cases.
The instantaneous efficiency of the motor, hydraulics, and total system for three pulse cycles are shown in Figure 2-11. It can be observed, that the motor efficiency drastically dropped at the start of acceleration (low flow end of the loop corresponding to $\eta_{\text{mot}}$). While the maximum motor efficiency during deceleration was almost 80%, it decreased to a minimum of 30.5%. The hydraulic efficiency shows a similar curve as expected from the steady state characteristics, showing a maximum of 39.2% during rotor deceleration, which decreased further at low flow rates. The total device efficiency was below 30% throughout the pulse cycle.

Figure 2-10 – Comparison of mean power components for continuous (CF) and pulsatile (PF) flow in (A) a stacked bar graph, and (B) tabular data. $P_{\text{el}}$, motor power consumption; $P_{\text{hyd}}$, hydraulic power output; $P_{\text{loss,hyd}}$, hydraulic loss; $P_{\text{loss,mot}}$, motor power loss; $e_{\text{est}}$, estimation error; % ↑, percentage increase from continuous to pulsatile flow.

<table>
<thead>
<tr>
<th></th>
<th>CF</th>
<th>PF</th>
<th>% ↑</th>
</tr>
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<tbody>
<tr>
<td>$P_{\text{el}}$ [W]</td>
<td>6.16</td>
<td>9.83</td>
<td>59.6</td>
</tr>
<tr>
<td>$P_{\text{loss,mot}}$ [W]</td>
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<tr>
<td>$P_{\text{loss,hyd}}$ [W]</td>
<td>2.61</td>
<td>3.51</td>
<td>34.5</td>
</tr>
<tr>
<td>$P_{\text{hyd}}$ [W]</td>
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<td>1.33</td>
<td>22.0</td>
</tr>
<tr>
<td>$e_{\text{est}}$ [W]</td>
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<td>0.25</td>
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<tr>
<td>$e_{\text{est}}/P_{\text{el}}$ [%]</td>
<td>5.03</td>
<td>2.54</td>
<td>–</td>
</tr>
</tbody>
</table>
2.3.2.2 Hydraulic Forces

The hydraulic forces evaluated in the FTR substantially increased with pulse amplitude. At the maximum pulse amplitude ($n_{ampl} = 1,200 \text{ rpm}$), the FTR pump generated a flow rate pulse of $32.3 \text{ L/min}$ (between $-7.2 \text{ L/min}$ and $25.1 \text{ L/min}$) and a pulse pressure ($PP$) of $45.8 \text{ mmHg}$. The corresponding waveforms are shown in Figure 2-12. The axial force reached a maximum value of $12.3 \text{ N}$ towards the left pump inlet, whereas the radial force had a maximum of $4.17 \text{ N}$. Forces in both axial and radial directions showed pulsating patterns correlating to the speed and flow rate changes, which were similar in shape at all amplitudes. The trends of maximum hydraulic axial and radial forces, maximum systolic flow rate, and $PP$ with increasing speed pulse amplitude are shown in Figure 2-13. In the equivalent CF condition, the axial and radial forces were substantially lower, with values of $2.44 \pm 0.19 \text{ N}$ and $0.65 \pm 0.22 \text{ N}$ respectively (y-intercepts in Figure 2-13A).
Figure 2-12 – Pulse waveforms generated with the FTR pump in vitro. Panels show (A) the motor speed and speed target, (B) systemic flow rate, (C) aortic pressure, (D) axial hydraulic force, and (E) radial hydraulic force.

The axial force was between a factor 3 and 4 higher than the radial force for all applied pulses, and increased approximately linear with the speed amplitude. A PP of 40 mmHg, which is typical for the native cardiovascular pulse, was generated with a speed amplitude of approximately 1000 rpm.

Figure 2-13 – Trends of (A) maximum axial ($F_{z,\text{max}}$) and radial ($F_{r,\text{max}}$) force and (B) maximum systolic flow rate ($Q_{s,\text{max}}$) and pulse pressure (PP) during the FTR pulsing experiment.
The corresponding axial and radial forces at that $PP$ were 10.28 $N$ and 3.45 $N$ respectively, while the systemic flow rate varied between 22.08 $L/min$ and $-5.85 L/min$.

### 2.3.2.3 In Vivo

Figure 2-14 show an example of pump data and haemodynamics generated with the BiVACOR TAH in vivo, while a summary of haemodynamic and pump parameters during $CF$ and $PF$ operation is given in Table 2-1. The maximum amplitude was lower compared to the in vitro results ($n_{ampl} = 650 \text{ rpm}$). Due to the higher flow demand of the bovine circulatory system, the systemic mean flow rate was $Q_s = 9.55 L/min$ at a mean arterial pressure ($MAP$) of 96.8 $mmHg$. Although the speed profile did not change, flow rate and pressure patterns varied over time with breathing and movement of the animal (Figure 2-14B, C).

---

**Figure 2-14** – Example of pulsatile waveforms with a square wave function generated with the BiVACOR TAH in vivo. The panels show (A) impeller speed and speed target, (B) systemic flow rate, (C) aortic pressure, (D) motor power consumption, (E) left atrial pressure, and (F) aortic $dP/dt$. 

90
Compared to the native heart waveforms, maximum arterial $dP/dt$ was low, reaching a maximum of 267.2 mmHg/s, while the average $dP/dt$ was 199.4 mmHg/s. Similarly, $SHE$ was low, showing an average of 3996.7 erg/cm$^3$ (native heart > 20,000 ergs/cm$^3$ [135]). The average pulse pressure was $PP = 27.84$ mmHg, while the systemic flow rate showed a high swing with an average difference between minimum and maximum flow rate of $\Delta Q = 16.93$ L/min, intermittently reaching values up to 20 L/min and 0 L/min during systole and diastole respectively.

Figure 2-15 shows the magnetic bearing actuator currents over three pulse periods. The currents show large spikes due to the increased force requirement in pulsatile operation. Further, the spikes for the three actuators exhibit different peak values, which indicates the presence of radial and/or tilt forces. To avoid partial saturation of the magnetic bearing stator legs, which may result in nonlinear bearing performance and instabilities, and due to impending atrial suction, the speed amplitude was not increased beyond 650 rpm.

<table>
<thead>
<tr>
<th></th>
<th>CF</th>
<th>PF</th>
</tr>
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<tbody>
<tr>
<td>$n_{ampl}$ [rpm]</td>
<td>–</td>
<td>650</td>
</tr>
<tr>
<td>$\overline{P}_{el}$ [W]</td>
<td>10.58</td>
<td>12.63</td>
</tr>
<tr>
<td>$Q_s$ [L/min]</td>
<td>9.56</td>
<td>9.55</td>
</tr>
<tr>
<td>$MAP$ [mmHg]</td>
<td>97.26</td>
<td>96.86</td>
</tr>
<tr>
<td>$\overline{P}_{ia}$ [mmHg]</td>
<td>13.23</td>
<td>15.22</td>
</tr>
<tr>
<td>$PP$ [mmHg]</td>
<td>–</td>
<td>27.84</td>
</tr>
<tr>
<td>$dP/dt$ [mmHg/s]</td>
<td>–</td>
<td>199.4</td>
</tr>
<tr>
<td>$SHE$ [erg/cm$^3$]</td>
<td>–</td>
<td>3996.7</td>
</tr>
</tbody>
</table>

Table 2-1 – Summary of haemodynamic and pump data comparing continuous flow (CF) and pulsatile flow (PF) operation. Pulse pressure ($PP$), $dP/dt$, and surplus haemodynamic energy ($SHE$) were averaged over 10 pulse cycles.

Figure 2-15 – Currents in the three magnetic bearing actuators (B1, B2, and B3) over three square wave speed modulation cycles in vivo.
2.4 Discussion
This chapter provides a broad introduction to pulsatile speed modulation of rotary blood pumps using the example of the BiVACOR total artificial heart (TAH). Influential parameters determining the speed and torque requirements of the drive were derived, and a first estimate of requirements for a pulsatile RBP was made. Subsequently, a preliminary evaluation of pulsatile square waveforms with the BiVACOR V2 TAH was performed, and factors limiting the device performance were observed.

The power losses generated in the hydraulics and motor of a levitated BiVACOR prototype were differentiated through estimation of the shaft power, and the hydraulic torque in pulsatile and continuous flow operation. Although an estimation error was observed in both modes, the results identified the pump motor drive as the primary cause of power loss in pulsatile operation. Estimation error is mostly attributed to the axial movement of the impeller, which was neglected in the estimation. However, due to the axial force acting towards the left inlet and the virtual zero power control of the levitation system, the impeller moves towards the right inlet (away from the motor). Therefore, the magnetic gap length of the motor increases and the movement may thus influence the motor capacity and efficiency, as well as fluid leakage characteristics between the left and right pumps. While the hydraulic loss was larger than the motor loss in constant speed operation, the additional power loss caused by pressure head and flow rate variations is comparatively small. Conversely, although pulse waveforms with a truly physiologic character were not achieved, motor losses substantially increased. The preliminary results indicate, that the motor efficiency drops specifically at low speeds and high torque, which is caused by significantly increased copper losses due to the high current requirement during acceleration. The results substantiated the hypothesis of the need for increased drive requirements to allow efficient pump operation with pulsatile speed waveforms. Consequently, investigation of methods to improve the drive efficiency and torque capacity appears promising. During the MCL experiments, ringing in the pressure waveforms was observed, which is in part caused by the imprecise motor target speed tracking, and by the rigid piping used for the MCL construction, which may cause undamped reflection and interference of pressure waves throughout the loop and affect the measurement of $dP/dt$ and/or $SHE$. Therefore, adaption of the MCL was considered to generate a more physiological pressure and flow rate response to pump speed changes.
The measurement results for dynamic axial forces, as well as the observed large bearing currents during the in vivo evaluation showed the large additional strain imposed to axial and radial bearings, when the impeller speed is rapidly changed. A more than four-fold increase of both axial and radial hydraulic forces was observed between constant speed operation and a speed amplitude of 1200 rpm. In both the in vitro and in vivo evaluations of the magnetically levitated prototype, the maximum speed amplitude was limited by the suspension capacity, which was directly related to the hydraulic forces. Additionally, in the in vivo experiment, intermittently or continuously low atrial pressures bear the risk of atrial suction, which may further limit the applicable speed amplitude and lead to additional bearing stress and adverse effects on the vasculature. In the case of an electromagnetic bearing, the increased stress may further lead to additional power loss in the bearing actuators. While the evaluation of rotor suspension systems for pulsatile RBP is not a major focus of this thesis, it is important to consider the force load imposed on the suspension during rapid speed modulation, which includes hydraulic as well as electromagnetic forces. Consequently, the corresponding pump motor drive should be designed with consideration of static and dynamic forces on the rotor, which may pose additional stress to the suspension system.

The first evaluations presented here show, that RBP such as the BiVACOR device may have the potential to generate near-physiologic waveforms. However, while PPs of 40 mmHg and beyond were generated, physiologic values of $dP/dt$ or SHE were not achieved. This may be related to limiting parameters such as the torque capacity, rotor inertia, and hydraulic pressure head-flow characteristic, which may require improvement to allow physiologic pulsatile operation. Additionally, suitable control techniques and speed profiles (such as sine wave, square wave, or other), which allow to adequately replicate native haemodynamics, while maintaining an acceptable increase of the electrical power consumption, are currently unknown.

2.5 Hypothesis and Thesis Overview

The initial evaluation of the BiVACOR V2 TAH prototype highlighted the major device characteristics influencing the acceleration, hydraulic performance, and power consumption of a rotary TAH operated with pulsatile impeller speed modulation. The results showed, that the device prototype was unable to generate physiologic waveforms, however, the analysis
suggested, that specific design changes may lead to substantially improved performance and reduced power requirements. Therefore, these preliminary findings built the foundation for the investigations performed in the following main chapters of the thesis.

The main hypothesis for this work is that the ability of a RBP to generate pulsatile outflow can be substantially improved by means of design and control. Specifically, it is hypothesised, that a flat pressure head-flow characteristic is beneficial for the device performance with rapid speed modulation, and that the shape of the applied speed profile has substantial influence on the generated pulsatility.

As the motor was identified for the primary cause of losses, particular attention was paid to the design of axial flux motors for centrifugal RBPs to cater for hydraulic designs leading to flat pressure head-flow characteristics while maintaining or improving the motor performance. Figure 2-16 shows a structural overview of the performed studies and their relation to the identified design limitations and challenges. In chapter 3, the suspension force load caused by electromagnetic forces in a centrifugal pump driven by an axial flux permanent magnet motor were considered with respect to the motor design and control strategy. Further, a finite element method (FEM) analysis of various motor geometry parameters and their effect on the axial forces, motor torque and efficiency, rotor inertia, and geometric design constraints was performed to determine guidelines for RBP axial flux motor design with the constraints for to pulsatile RBP operation, including large air gaps and motor inner diameter. The FEM model model used was verified with measurements of an axial flux motor prototype in the described MTR (section 2.3.1.3). In chapter 4, a detailed in vitro analysis of the effects of speed profile shape and amplitude with respect to performance metrics such as $dP/dt$, surplus haemodynamic energy ($SHE$), and the pulse power index ($PPI$) is presented. Chapters 0 & 6 comprise the numerical optimization of speed modulation waveforms, and the investigation of the relative influence of device characteristics on the maximum achievable pulsatile output. The results presented in this thesis may therefore assist in device design decisions, to improve performance when rapid impeller speed modulation is considered. While the following research chapters are intended to be more generally applicable to rotary blood pump development, the presented considerations always need to be set in context with a given device design and the corresponding design constraints with respect to size, cost, and most importantly, biocompatibility and patient safety.
Figure 2-16 – Overview of the identified major influential factors considered in this PhD project, and their consideration within the studies in the following chapters.
3 Design Considerations for a Rotary Blood Pump Drive System

The previous chapter highlighted the motor as the primary source for power loss, when the BiVACOR total artificial heart (TAH) was operated with rapid speed modulation. Due to the similar design constraints, such as maximum attractive forces or a large gap length, this finding is likely similarly applicable to other RBP types. While the generation of losses in motor and hydraulics are dependent on the specific design of a rotary blood pump (RBP), a similar distribution of power loss between the systems is plausible for other RBPs. The large difference in the motor losses between continuous flow (CF) and pulsatile flow (PF) operation in the evaluation presented was mostly attributed to the relatively large axial magnetic gap between the motor stator and rotor (the physical gap length equals $l_{gap} = 3 \text{ mm}$, which amounts to approximately 22% of the total motor height). This increases the magnetic flux leakage and thus causes a low magnetic coupling between the stator and rotor. Therefore, the current requirement and consequently the copper losses are increased, which is in contrast to larger industry machines. These losses then become particularly evident at high required torques, as required for the acceleration during rapid speed modulation.

It was further indicated that an increase of the left pump blade height and the inlet diameter may benefit the hydraulic performance of the pump. In the evaluated prototype, these measures would require a further increase of the magnetic gap, and an increase of the inner diameter of the motor, which further reduces the magnetic coupling and the active conductor length in the slots.

In a previous study, Pohlmann et al. investigated axial flux motor geometry parameters with respect to their effect on efficiency and axial attractive force, and found that a reduction of the inlet diameter of the ReinVAD left ventricular assist device (LVAD) and concomitant reduction of the inner motor diameter allowed for improvement of the drive characteristics [182]. However, this approach is in direct contrast to the here-proposed design goal of a large inner stator diameter, which may allow incorporation of a large pump inlet size leading to a flat pressure head-flow (HQ) characteristic of the impeller. Further, Pohlmann evaluated different solid iron-based stator core materials, which resulted in a large influence of eddy current loss in the core material. It was acknowledged, that the use of a soft magnetic composite (SMC) material may allow for larger efficiencies, however, no evaluation of SMC stator cores was performed. While these materials suppress eddy currents and thus may allow
for a significant reduction of core losses, they exhibit a lower permeability than iron. However, specifically at relatively large magnetic gaps, the reluctance of the air gap is significantly larger than that of the stator core, thus the influence of the reduced permeability of SMC is comparatively low, and its use appears promising for this given application.

Therefore, in this study, a parameter investigation of motor geometries allowing for large blade heights and large inner diameters on the example of a 12-slot/10-pole brushless DC (BLDC) configuration with a slotted SMC stator core based on the pump geometry of the BiVACOR TAH was performed within a numerical framework using the finite element method (FEM). The analysis was focused on the drive torque, efficiency, rotor inertia, and axial attractive forces corresponding to each motor geometry, to evaluate drive designs which may improve motor performance and/or reduce the axial suspension load in pulsatile RBP operation.

3.1 Aims

The primary aim of this chapter was to numerically investigate geometry parameter combinations of three-phase axial flux permanent magnet motors with a 12-slot/10-pole configuration, which may benefit a centrifugal pump design for improved performance in pulsatile operation, specifically when a large inner diameter and/or a large magnetic gap are chosen to improve hydraulic performance. The secondary aim was to investigate strategies to decrease the axial force load of the rotor suspension system. Specifically, the study aims were to:

- Derive a numerical FEM-model to allow a quick evaluation of various motor geometries based on a SMC stator core material with respect to motor torque, efficiency, rotor inertia, and axial attractive force.
- Identify performance trends corresponding to motor geometry parameters and combinations of these leading to favourable design characteristics for speed modulation of a rotary blood pump drive to produce pulsatile outflow.
- Evaluate the feasibility of axial flux motor designs with relatively large gaps and inner diameters for the use in a rotary blood pump capable of pulsatile and non-pulsatile outflow.
3.2 Methods

3.2.1 FEM Model development

In order to evaluate different motor geometries with reduced computational time, a numerical framework based on a quasi-three-dimensional (3D) FEM slice model of an axial flux motor was developed. When such a model is employed, it is important to consider the influence of flux fringing near the inner and outer diameters, which increases with increasing magnetic gap lengths. Therefore, to develop and verify the 2D-simulation based slice model, a full 3D model was implemented as an interim step. The 3D model was validated against measured data obtained for an example axial flux motor with the previously described motor test rig (MTR). Subsequently, magnetic flux, torque, and force data obtained with the 2D model were validated against the full 3D model.

3.2.1.1 3D Model

For verification of the 2D-FEM framework, a magnetostatic 3D-FEM model was implemented in ANSYS Maxwell (ANSYS Inc., Canonsburg, PA, USA). The model was based on and validated against an outdated BiVACOR axial flux motor prototype used in BiVACOR V1 and included in this study as a reference motor (RM), and was parameterised to allow simple adjustment of the geometry. The RM is a 12-slot 10-pole permanent magnet motor with a concentrated tooth-coil winding. The twelve coils are inserted in the slots of a ferromagnetic stator core manufactured from a SMC material. The rotor consists of ten Neodymium-Iron-Boron (NdFeB) axially magnetised permanent magnet plates with alternating polarisation mounted to a ring-shaped iron yoke (99.8% purity). For the validation of the RM, axial force, torque, and efficiency were evaluated over wide ranges of air gaps, load torques, and operating speeds with the MTR (cf. chapter 2). Due to the inverse symmetry of the machine type, only one half of the motor was modelled, and antiperiodic boundary conditions were imposed to the cut faces, to account for the flux distribution in the second half of the motor. Consequently, the calculated results for axial force and torque were doubled. Figure 3-1 shows the implemented model half motor and a definition of the geometric parameters. In the figure, the stator is shown at the bottom, the rotor is shown at the top, and the rotational (axial) axis is the $z$-axis. The coils (transparent) were implemented as solid copper blocks with a single winding turn around the stator teeth, whereas the stator excitation was implemented in the form of a stranded current excitation through the cross
section of each coil, which equalled the magnetomotive force \(MMF_{coil}\) generated by all turns \(N_{coil}\) of the respective coil excited with the current \(I\).

\[
MMF_{coil} = I \cdot N_{coil}
\]  

(3.1)

Geometry changes were facilitated through scaling of the magneto-motive force with the change of the effective cross-sectional area due to changes in the slot geometry, thus a constant slot current density independent of the slot geometry was maintained for a constant current input. The modelled rotor half consists of five permanent magnet (PM) plates and half of the ring-shaped rotor yoke. A summary of the parameters and the corresponding values of the initial model implementation is further given in Table 3-1. The inner \(r_{i,core}\) and outer \(r_{o,core}\) core radii apply to the stator core, rotor yoke, and permanent magnets. The inner stator radius \(r_i\) denotes the total inner radius

![3D model of half the axial flux permanent magnet reference motor (RM).](image)

Figure 3-1 – 3D model of half the axial flux permanent magnet reference motor (RM). \(l_{gap}\), gap length; \(d_s\), slot depth; \(w_s\), slot width; \(h_{pm}\), permanent magnet thickness; \(\alpha_{pm}\), permanent magnet angle; \(h_{y,s}\), stator yoke; \(h_{y,r}\), rotor yoke; \(r_{o,core}\), outer core radius; \(r_{i,core}\), inner core radius. The inner and outer core radii apply to the stator core, rotor yoke, and permanent magnets.

The 3D-FEM model was used to calculate the steady-state magnetic field, torque constant, and axial attractive force solutions for the RM, to serve as an interim step in the validation of
the developed 2D-simulation based quasi-3D model, which is introduced in the following section.

Table 3-1 – Summary of the 3D-FEM model parameters defining the geometry of the axial flux reference motor (RM). The given parameter values were used for the model validation. PM, permanent magnets.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( r_{o,\text{core}} )</td>
<td>Outer core radius (rotor, stator, and PM)</td>
<td>24.5</td>
<td>mm</td>
</tr>
<tr>
<td>( r_{i,\text{core}} )</td>
<td>Inner core radius (rotor, stator, and PM)</td>
<td>14</td>
<td>mm</td>
</tr>
<tr>
<td>( r_{i} )</td>
<td>Inner stator radius (including end-winding turns)</td>
<td>10.8</td>
<td>mm</td>
</tr>
<tr>
<td>( l_{\text{gap}} )</td>
<td>Physical gap length (stator surface to PM surface)</td>
<td>3</td>
<td>mm</td>
</tr>
<tr>
<td>( d_{s} )</td>
<td>Slot depth (equivalent to tooth height)</td>
<td>6.2</td>
<td>mm</td>
</tr>
<tr>
<td>( w_{s} )</td>
<td>Slot width</td>
<td>6.4</td>
<td>mm</td>
</tr>
<tr>
<td>( h_{\text{pm}} )</td>
<td>Permanent magnet thickness</td>
<td>1.4</td>
<td>mm</td>
</tr>
<tr>
<td>( \alpha_{\text{pm}} )</td>
<td>Permanent magnet angle</td>
<td>35</td>
<td>deg</td>
</tr>
<tr>
<td>( h_{y,s} )</td>
<td>Stator yoke thickness</td>
<td>1.5</td>
<td>mm</td>
</tr>
<tr>
<td>( h_{y,r} )</td>
<td>Rotor yoke thickness</td>
<td>1.6</td>
<td>mm</td>
</tr>
<tr>
<td>( B_{r,\text{PM}} )</td>
<td>Permanent magnet remanence induction</td>
<td>1.26</td>
<td>T</td>
</tr>
<tr>
<td>( N_{\text{coil}} )</td>
<td>Coil turn number</td>
<td>110</td>
<td>–</td>
</tr>
</tbody>
</table>

3.2.1.2 2D Field Approximation

To allow accelerated evaluation of various motor geometries, a planar model approximation of the three-dimensional geometry was implemented. The model comprised a combination of multiple 2D-FEM simulations to approximate the field distribution and resulting electromagnetic torque and axial attraction force. The simulation framework was based on the freeware software package Finite Element Method Magnetics (FEMM) [223], which was interfaced and programmed with MATLAB (The MathWorks Inc., Natick, MA, USA).

The 2D model approximation was based on the multi-slice quasi-3D method [21,22,194,224]. For this purpose, the machine is divided at multiple ring-shaped cut-faces around stator circumference. For each of the resulting motor slices, a separate simulation of the magnetic field at the mean radius of the respective slice was performed utilizing a 2D-FEM model. Effectively, the curved surface around the stator circumference is unwrapped to a planar 2D surface. An example of the field solutions in the curved surface within the 3D model and the corresponding 2D-approximation is shown in Figure 3-2.
Figure 3-2 – Illustration of the quasi-3D approach to approximate the magnetic field in the motor components and magnetic gap. The panels show examples of (A) the magnetic flux density solutions for a single curved surface at the mean stator radius in the 3D-FEM model, (B) the unwrapped planar surface in the 2D-simulation framework, and (C) multiple curved surfaces at different radii to derive a combined solution, illustrated in the 3D model.
The model implementation allowed the simulation of different numbers of slices \( n_{slice} \), whereas the thickness of the \( k \)-th slice \( (\Delta r_{slice,k}, k = 1, ..., n_{slice}) \) was identical for all slices, and is calculated from the number of slices and the inner and outer core radii \((r_{i,core}, r_{o,core})\) as

\[
\Delta r_{slice,k} = r_{o,k} - r_{i,k} = \frac{r_{o,core} - r_{i,core}}{n_{slice}},
\]

where:

\( r_{i,k} \), inner radius of the \( k \)-th slice

\( r_{o,k} \), outer radius of the \( k \)-th slice.

The radius of the \( k \)-th simulated 2D-surface \( (r_{sim,k}) \) was defined at the mean radius of the respective slice:

\[
r_{sim,k} = \frac{r_{o,k} + r_{i,k}}{2} = r_{i,core} + \left(k - \frac{1}{2}\right) \cdot \Delta r_{slice}.
\]

For each simulated slice, the tangential and axial force components \( F_{\phi,k} \) and \( F_{z,k} \) were calculated with FEMM, using the weighted Maxwell stress tensor approach [223]. The axial force \( (F_{ax,k}) \) and electromagnetic torque \( (T_{el,k}) \) contributions of each slice were calculated according to equations (3.4) and (3.5).

\[
F_{ax,k} = \Delta r_{slice,k} \cdot F_{z,k}.
\]

\[
T_{el,k} = \Delta r_{slice,k} \cdot r_{sim,k} \cdot F_{\phi,k}.
\]

While the combination of multiple simulation planes allowed for approximation of the stator curvature and the corresponding change of the magnetic field with increasing radius, the effect of flux density reduction due to flux leakage at the inner and outer radii (edge effect) cannot be considered with a simple multi-slice approach. However, as motor geometries with large magnetic gaps were specifically considered in the parametric analysis (section 3.2.3), the influence of the leakage flux is significant, and cannot be neglected. Therefore, an approach similar to the approach presented by de la Barrière et al. [22] was chosen to include consideration of the edge effect.
As the air gap flux density near the inner and outer radii is reduced, the torque and axial force contributions of the respective machine slices calculated with the 2D simulation model were expected to be overestimated. Therefore, a radius-dependent correction function for the magnetic flux in the air gap \( f_{\text{edge}, B}(r, z) \) based on an approximation of the radial flux distribution was defined, and subsequently used to derive appropriate correction functions for the force and torque \( f_{\text{edge}, F}(r); f_{\text{edge}, T}(r) \) contribution of each motor slice were derived (cf. section 3.2.1.3).

Figure 3.3A shows an excerpt of the non-uniform magnetic flux density distribution calculated with the 3D-FEM model in a radial cross section (RCS) plane intersecting with the centre of a stator tooth. The field dispersion in the air volumes outside of the magnetic core region \( r > r_{o,\text{core}} \) and \( r < r_{i,\text{core}} \) results in a decreased flux density in the magnetic gap, and consequently a reduction of the generated torque and axial force. To allow consideration of the flux leakage, a corresponding 2D-FEM model of the RCS in a two-dimensional \( r-z \)-plane was implemented (Figure 3.3B). As the three-dimensional return path of the magnetic flux cannot be included in the two-dimensional model, periodic boundary conditions were applied to the upper end of the rotor yoke and the bottom end of the stator yoke, resulting in a direct return of the flux from the rotor yoke to the stator yoke.

The RCS-model allows calculation of the relative reduction of the axial magnetic flux density component due to the edge effect \( f_{\text{edge}, B}(r, z) \) as a function of the radius. For this purpose, the values of \( B_z \) in the air gap region of the RCS-model were divided by the magnetic flux...
density under the assumption of a radially uniform field distribution in the air gap 
\(B_{z, RCS, unif.}\):

\[
f_{edge,B}(r, z) = \frac{B_{z, RCS}(r, z)}{B_{z, RCS, unif.}}.
\] (3.6)

Hereby, the value of \(B_{z, RCS, unif.}\) was calculated through a second computation cycle of the 
RCS model, for which the relative permeability of the air regions outside of the magnetic 
core region \(r < r_i; r > r_o\) was set to an infinitesimally small value \(\mu_{r, unif} \ll \mu_{r, air}\), 
enforcing a uniform distribution of the flux in the gap between rotor and stator. It should be 
noted that, specifically, when the gap length is large, the values of \(B_z\) at the same radius may 
significantly differ.

3.2.1.3 Combination of the 2D-Simulations

In order to derive the scaling functions \(f_{edge}(r)\), the analytical expression for the force 
calculation with the Maxwell stress tensor in cylindrical coordinates (equation (3.7)) \[192\] is 
considered.

\[
\vec{F} = \oint_S T \cdot \vec{n} \, ds,
\] (3.7)

where \(S\) is a surface enclosing the considered body, \(\vec{n}\) is a unit normal vector to the surface 
\(S\), and

\[
T = \frac{1}{\mu_0} \begin{bmatrix}
\frac{B_r^2 - B_\phi^2 - B_z^2}{2} & B_r B_\phi & B_r B_z \\
B_\phi B_r & \frac{B_\phi^2 - B_r^2 - B_z^2}{2} & B_\phi B_z \\
B_z B_r & B_z B_\phi & \frac{B_z^2 - B_r^2 - B_\phi^2}{2}
\end{bmatrix}
\] (3.8)

is the Maxwell stress tensor.

For the derivation of the axial force scaling factor \(f_{edge,F}\) it is advantageous to consider an 
enclosing surface \(S\), which is in infinitesimally small distance from the stator core surface, 
whereas the influence of the stator winding is neglected. Therefore, the integration over \(S\)
according to equation (3.7) can be considered approximately along the stator core surface. The resulting integration surfaces are illustrated on the example of one slot pitch in Figure 3-4.

For the radially outward and inward facing surfaces \((S_1, S_5)\), the normal vector \(\vec{n}\) is equal to the positive and negative radial unit vector \(\vec{e}_r\) and \(-\vec{e}_r\) respectively. Therefore, the axial force contribution of each surface simplifies to the integral of \(T_{31} = B_z B_r\). However, due to the large permeability of the core material and the magnetic flux density enters the surface approximately perpendicular to the surface \((B_\phi, B_z \approx 0)\), and the integral is approximately zero. Similarly, the axial force contribution of the surfaces facing in tangential directions \((S_2, S_6)\) is approximately zero. Consequently, the axial force arises from the axially facing surfaces on the top of the teeth, and the top and bottom of the stator yoke \((S_4, S_3, S_7)\). As shown before, the non-normal components of the flux density vanish, hence it can be derived from the Maxwell stress tensor, that the axial force is approximately proportional to the square of the axial B-field component penetrating the stator core surface:

\[
F_{ax} \sim B_{z, core}^2.
\] (3.9)

Consequently, the correction function for the axial force was derived from equation (3.6) at the stator surface \((z = 0)\) and defined as:

\[
f_{edge,B}(r) = f_{edge,B}(r, 0).
\] (3.10)

While the influence of different air gap flux distributions over the stator teeth and slots was neglected, the approximation resulted in good agreement between force calculation results of the implemented 3D-FEM and 2D approximation models (see section 3.2.2).
Due to the torque contribution of the magnetic field within the stator slots, which is acting on the tangentially facing surfaces of the teeth \((S_2, S_6)\), a similarly simple derivation of the motor torque correction function \(f_{\text{edge}, r}(r)\) based on the RCS-model cannot be made. Instead, the correction function was empirically derived. The best agreement between 3D-FEM and 2D-approximation results were found for a linear relationship to the field reduction \(f_{\text{edge}, B}(r)\) averaged over the air gap length, which was extracted at ten equidistant axial positions within the gap:

\[
f_{\text{edge}, r}(r) = \frac{1}{10} \sum_{i=1}^{10} \left( f_{\text{edge}, B} \left( r, \frac{(i - 0.5) \cdot l_{\text{gap}}}{10} \right) \right)
\]  

(3.11)

Finally, the correction functions were applied to combine the force and torque contribution of each simulated circumferential motor slice. The model convergence for different numbers of model slices \(n_{\text{slice}}\) was evaluated for two methods of applying the edge factors to the partial solutions. Firstly, the values of the edge factors at the centre radius of each slice were used to scale the force and torque results accordingly (method 1, equation (3.12)). Secondly, the average values of \(f_{\text{edge}, F}\) and \(f_{\text{edge}, T}\) between the inner and outer radii of each slice \((r \in [r_{i,k}, r_{o,k}])\) were calculated and applied as scaling factors (method 2, equation (3.13)).

To determine the model convergence, the calculated torque and force were compared for various numbers of slices up to \(n_{\text{slice}} = 100\). Due to the simulated 12-slot/10-pole motor type and the large evaluated gap lengths, the cogging torque and force ripple are comparatively low [225], hence they were neglected in the evaluation of axial force and efficiency.

\[
F_{ax} = \sum_{k=1}^{n_{\text{slice}}} F_{ax,k} \cdot f_{\text{edge}, F}(r = r_{\text{sim},k})
\]  

, method 1  

(3.12)

\[
T_{el} = \sum_{k=1}^{n_{\text{slice}}} T_{el,k} \cdot f_{\text{edge}, T}(r = r_{\text{sim},k})
\]

\[
F_{ax,\text{avg}} = \sum_{k=1}^{n_{\text{slice}}} F_{ax,k} \cdot \overline{f_{\text{edge}, F}}(r \in [r_{i,k}, r_{o,k}])
\]  

, method 2  

(3.13)

\[
T_{el,\text{avg}} = \sum_{k=1}^{n_{\text{slice}}} T_{el,k} \cdot \overline{f_{\text{edge}, T}}(r \in [r_{i,k}, r_{o,k}])
\]
3.2.1.4 Loss Models

Different models were used to analytically calculate the motor efficiency. The loss mechanism typically considered in brushless permanent magnet machines are:

- Copper losses in the stator windings \( P_{cu} \), due to current flowing through the coil resistance.
- Hysteresis losses \( P_{hyst} \) in the stator and rotor core components due to the changing direction of magnetisation in the ferromagnetic material.
- Eddy current losses \( P_{ec} \) caused by circular currents in the conductive core materials and permanent magnets, which are induced by fast changing magnetic fields.
- Friction loss in the rotor bearings \( P_{fr} \).

In the case of a magnetically levitated RBP, bearing friction losses do not apply, while windage losses are considered part of the required hydraulic torque. Furthermore, these losses are low in the case of a hydrodynamic bearing or fluid immersed pivot bearing, hence they were not considered here.

The copper losses in the stator resistance follow from Ohm’s law and can be calculated as the product of the square of the applied current and the resistance. However, this requires detailed knowledge of the coil wire size turn number, and winding connection. In order to not require an assumption of the winding selection, the copper losses can be calculated utilising the applied MMF and stator geometry parameters, and the slot fill factor \( k_{cu} \), which was derived from the RM, and assumed to be constant for all simulated machines. The stator MMF was applied according to the assumption of a field-oriented control scheme with ideal commutation (see section 3.2.4). For the resulting three-phase sinusoidal stator excitation, the copper loss is calculated as:

\[
P_{cu} = 3 \cdot 4 \cdot \frac{M_{MF}^2_{coil}}{A_{slot} \cdot k_{cu}} \cdot l_{mean} \cdot \rho_{cu} \cdot \frac{1}{1 + \alpha_{cu}(T - T_{20})}, \tag{3.14}
\]

where the factors 3 and 4 account for the three-phase system and four coils per phase in the considered machine type. The mean coil turn length (Figure 3-5) and cross-sectional slot area \( A_{slot} \) follow directly from the geometry parameters, and \( \rho_{cu} \) is the specific resistance of copper. The last term in the equation considers the change of the resistance with operating temperature, using the temperature coefficient for copper \( \alpha_{cu} = 3.93 \%/^\circ K \) and the reference
ambient temperature \( T_{20} = 20°C \). Due to the similar boundary conditions of the application, the assumption of an operating temperature \( T \) of 45°C was adapted from [182]. The slot fill factor of the RM was determined to \( k_{cu, RM} = 66\% \); however, a more recent prototype reached a fill factor of approximately \( k_{cu} = 70\% \), which was adapted for the simulations in the following geometry parameter variation. Further improvements to maximize the fill factor may be possible with an appropriate choice of the winding wire, which may further influence a final slot geometry design depending on available wire types.

When a solid conductive stator core material is chosen, eddy current losses are typically the dominant cause of core losses. A precise evaluation of hysteresis and eddy current losses in the ferromagnetic core materials can be made with transient FEM simulations of the motor at various rotor angles. However, while such a microscopic analysis can deliver accurate solutions, the required simulations are time-intensive and require high computational power. In this case of a soft magnetic composite (SMC) material in the stator core, eddy current loss is significantly reduced. Further, compared to the copper losses and due to the considered large magnetic gaps, the alternating field content within the rotor yoke and permanent magnets is low, hence the rotor losses are small and were not considered in the model (cf. [179]).

In the application considered here, the relative performance decrease of motor geometries with comparably large inner diameters and/or air gaps was evaluated, whereby a reasonably accurate estimation of the core losses is sufficient. The combined core losses caused by hysteresis and eddy currents were expressed according to the widely used Steinmetz equation [226,227], which was fitted to specific core loss data of the SMC material to yield an approximation of the specific core loss in \( W/\text{kg} \) as a function of the amplitude of the according to equation (3.15).

\[
p_{\text{core}} = 0.0993 \cdot \hat{B}^{1.958} \cdot f_{el} + 4.743 \cdot 10^{-5} \cdot \hat{B}^2 \cdot f_{el}^2 \tag{3.15}
\]
This equation allows the separate calculation of the core loss contribution in several parts of the motor utilizing the simulated local flux densities in the stator teeth, tooth tips, the yoke section under the teeth and slots, and the corresponding core material volumes. The points at which the magnetic flux density was extracted for the core loss calculation are illustrated in Figure 3-6.

The core loss components due to magnetic flux density components in \( \phi \) and \( z \)-directions were separately evaluated and added. The total core loss \( P_{\text{core}} \) was calculated as the sum of the contributions of the harmonic components of the flux density at the evaluated points in each motor slice, and the corresponding core volume and material density (equation (3.16) [226,228].

\[
P_{\text{core}} = \sum_{i=1}^{n} P_{\text{core}}(B_{\phi,i}, f_i) + P_{\text{core}}(B_{z,i}, f_i).
\]

While this approach allows reduction of the computation time, the core loss calculation only gives a rough estimate. However, due to the relatively large magnetic gap lengths and to the SMC stator core material, the core losses are expected to be relatively small compared to the copper losses for most evaluated geometries.

The simulation results and loss models calculated from the quasi-3D simulation framework were validated against force, torque, and efficiency measurements of the RM (section 3.2.2). The evaluation approach applied here allowed a quick, initial evaluation of the simulated motor geometries, while a more detailed analysis with transient 3D-FEM simulations for a small parameter range was reserved for obtaining precise predictions of the motor efficiency required for a design optimisation.
3.2.2 Model Validation

3.2.2.1 Convergence of the 2D Model

The model convergence for different numbers of model slices $n_{slice}$ was evaluated for two methods of applying the edge factors $f_{edge}$ as per equations (3.12) and (3.13). For illustration of the two methods, the relative radius between inner and outer core radii is introduced:

$$r_{rel} = \frac{r - r_{l,core}}{r_{o,core} - r_{l,core}}.$$  \hspace{1cm} (3.17)

Figure 3-7 shows the edge factor as a function of $r_{rel}$ between 0% and 100% ($r \in [r_{l,core}, r_{o,core}]$) for a number of slices of $n_{slice}=5$, and the edge factors calculated to the two scaling methods introduced in section 3.2.1.3. The graphs show that due to the steep nonlinear decrease of $f_{edge,B}$ near the edges ($r_{rel} = 0\%, 100\%$), the calculated average values for each slice are lower than the function values evaluated at the center of the slice $\left(f_{edge,B}(r_{sim,k})\right)$, that is, the scaling factors derived from method 2 are lower than those derived from method 1.

Figure 3-7 – Illustration of the edge factor for $n_{slice} = 5$. The graphs show the $f_{edge,B}$ as function of the relative radius $r_{rel}$, the evaluated values at the center radii of the simulated slices $f_{edge,B}(r_{sim,k})$ (method 1), and the averaged edge factors over the width of each slice $r \in [r_{l,k}, r_{o,k}]$ (method 2).

The convergence of the two scaling methods as a function of the number of slices is shown in Figure 3-8. The graphs show that both methods converge towards the same solutions for large $n_{slice}$. At $n_{slice} = 100$, the difference between the two methods were $\Delta T_{100} = 0.015 \text{ mNm}$ and $\Delta F_{100} = 0.028 \text{ N}$ respectively. However, the results for method 2 converged
at significantly lower numbers of slices. Compared to the solution with 100 slices, for one slice the errors of both axial force and electromagnetic torque were below 1%. For $n_{\text{slice}} = 5$, an error of $e_F = 0.245\%$ for the axial force and $e_T = 0.04\%$ for the torque were calculated. Consequently, for the following analysis, all calculations were scaled with method 2 (with the averaged edge factors according to equation (3.13)). As the required simulation time linearly increased with the number of slices, 5 slices were chosen to allow for the simulation of a large number of motor geometries.

3.2.2.2 Reference Motor Evaluation

To validate the air gap field distribution, the results of the 2D slice model and the 3D model were compared. Figure 3-9 shows an exemplary comparison of the axial flux density components in the magnetic gap at relative radii $r_{\text{rel}} = 10\%, 50\%$, and $90\%$. The graphs show the flux density computed with the 3D-FEM model, the unscaled results of the 2D-FEM model, and the 2D results scaled with the averaged edge factors for $n_{\text{slice}} = 5$.
The results for $r_{rel} = 50\%$ (Figure 3-9 B) show that the results between the 2D and 3D models are in excellent agreement (root mean square deviation $e_{rms} = 0.0058$ T). As $r_{rel} = 50\%$ corresponds to the average core radius, the edge effect has comparably low influence, and the scaled solution differs only marginally from the raw solution $f_{edge,B}(r_{rel} = 50\%)$. The plots show the axial component of the Flux density calculated with the 3D model, the raw solution calculated with the 2D model, and the 2D solution scaled with $f_{edge,B}$. 

Figure 3-9 – Comparison of the flux density distribution in the axial center of the magnetic gap, evaluated at relative radii $r_{rel}$ of (A) 10%, (B) 50%, and (C) 90%. The plots show the axial component of the Flux density calculated with the 3D model, the raw solution calculated with the 2D model, and the 2D solution scaled with $f_{edge,B}$. 

The results for $r_{rel} = 50\%$ (Figure 3-9 B) show that the results between the 2D and 3D models are in excellent agreement (root mean square deviation $e_{rms} = 0.0058$ T). As $r_{rel} = 50\%$ corresponds to the average core radius, the edge effect has comparably low influence, and the scaled solution differs only marginally from the raw solution $f_{edge,B}(r_{rel} = 50\%)$. 

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Figure 3-9 A and C show a large difference in the magnitude of the field solutions calculated with the 2D and 3D model, whereas the solution scaled with the edge factors \( \overline{f_{\text{edge},B}}(r_{\text{rel}} = 10\%, 90\%) = 0.83 \) show good agreement (RMS-difference \( e_{\text{rms}} < 0.01 \text{T} \)) with the 3D-FEM simulation results.

Figure 3-10 shows a comparison between the measurements obtained with the prototype in the motor test rig (MTR) and the simulation results for the 3D and 2D FEM models. The axial force decreased exponentially with increasing gap lengths, showing good agreement between measurement and simulation results. The forces simulated with the 3D model were within a 2% error of the measurements, while the 2D model slightly overestimated the forces at larger gaps. With increasing gap lengths, the percentage error between the 2D and 3D models increased up to a maximum of 6.07% at \( l_{\text{gap}} = 4.8 \text{mm} \), whereas the maximum absolute difference was 0.39N. At the smallest evaluated gap length \( l_{\text{gap}} = 2.8 \text{mm} \) the relative error between the models was below 0.45%. The deviation of the forces obtained with the 2D model, especially at large gap lengths, is attributed to the approximation of the edge effect in the 2D model environment.

For the torque simulations, MMF values corresponding to the product of the winding turn number and motor currents of \( I_{\text{mot}} = 0, 1, 2, \) and 3 A were applied to the motor models. The
simulated torque increased linearly with the applied motor current. The error between the calculations of the 2D and 3D models was 0.26%, whereas the 2D model slightly underestimated the torque.

During the measurements, a fixed load torque was applied to the motor and the current requirement at each operating point was evaluated. Compared to the 3D simulations, the measured current requirement was up to 10% higher at the lowest torque (which is a difference of $\Delta I_{\text{mot}} = 0.08A$), but was generally in good agreement with the simulation results and within an error margin of 2% for $T_{\text{Load}} \geq 40\, mNm$.

A comparison of the measured and simulated motor efficiency is shown in Figure 3-11. The simulated efficiency data was obtained from the 2D-FEM model and the introduced loss models. The graphs show the efficiency and power consumption as functions of the motor speed for load torques of $T_{\text{Load}} = 30, 40, \text{ and } 50\, mNm$. The trends between the measurements and simulations were in good agreement, with a maximum difference of the measured and simulated efficiency of 2%. The model underestimated the power consumption at $T_{\text{Load}} = 50\, mNm$ by up to 0.58 $W$, while it only marginally differed at load torques of 30 and 40 $mNm$.

Figure 3-11 – Comparison of (A) motor efficiency and (B) the power consumption at different speeds between the prototype measurements and the 2D-FEM and loss models at load torques of $T_{\text{Load}} = 30, 40, \text{ and } 50\, mNm$. 

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Altogether, the simulation results showed good agreement with the prototype measurements. Discrepancies between the results may be caused by inaccuracies in the modelling approach and the operation of the motor prototype during the measurements. The motor was controlled with a sensor-less field-oriented control algorithm, which may be subject to estimation error of the rotor angle and thus a slight reduction of the efficiency, especially at low motor currents (cf. Figure 3-10B). Additionally, an increase of the winding temperature at larger load torques may have caused an increase of the stator winding resistance of the evaluated motor prototype during the experiments (in particular at the largest load torque).

Furthermore, the modelling approach inherently introduces inaccuracies. This specifically applies to the core loss model based on the Steinmetz-equation, which was introduced to obtain a rough estimate of the occurring core losses. The losses were not computed for each node of the finite element mesh, but only at defined characteristic points within the core instead. However, due to the large air gaps and the SMC core material considered here, the core losses and the harmonic content in the stator core flux density distributions were low. Considering the required computation time for the large number of evaluated motor geometries, the loss model provided a sufficient approximation for the investigation of performance trends with respect to design benefits for rapid speed modulation.

Nevertheless, the results of the subsequent study must be viewed within the presented error margin. Specifically, the comparatively coarse calculation of the core losses allows for a broad comparison of motor geometries as presented here, but cannot be utilized for a precise prediction of motor efficiencies. Consequently, the 2D model may not be suitable for fine adjustments or design optimizations of a given motor design. However, the observed results are adequate for the comparison of multiple geometry variants to find an initial design for a rotary blood pump application, under evaluation of wide parameter ranges and with low time expenditure. While further fine design iterations of a given geometry using 3D-FEM and/or transient simulations may be required to precisely predict the drive efficiency, it was concluded that in the context of the fundamental study proposed in the following, the performance predictions obtained from the 2D-FEM model showed sufficient accuracy.
3.2.3 Motor Geometry Evaluation

The main objective of the motor geometry evaluation was to investigate the feasibility of motor geometries with large air gap lengths and inner diameters, relative to the outer diameter. Therefore, a geometry parameter variation was performed. Based on the evaluated reference motor (RM), the outer core radius of the evaluated motor geometries was kept constant at a value of \( r_{o,core} = 24.5 \text{ mm} \), assuming an unchanged outer rotor diameter.

For the evaluation of the inner and outer motor radii, it is important to notice, that the dimensions are not only determined by the core size, but also by the end winding turns of the stator coils (Figure 3-12). In the present case of a concentrated double-layer winding, two adjacent coils share one slot and each coil spans one slot pitch, whereas the coil width is half of the slot width. Consequently, the end winding turns exceeds the core dimensions by the half slot width, which affects the total stator inner \( (r_i) \) and outer radius \( (r_o) \), which follow from the core and slot dimensions:

\[
\begin{align*}
r_i &= r_{i,core} - 0.5 \cdot w_s \\
r_o &= r_{o,core} + 0.5 \cdot w_s.
\end{align*}
\]

Further, it should be noted, that the maximum slot width is dependent on the inner core radius, given by the slot width for which neighbouring slots intersect at the inner radius (see Figure 3-12). The effects of the varied geometry parameters were evaluated under consideration of typical design goals, which variously included to reduce the rotor inertia, maximize the motor efficiency, and/or reduce or maintain the maximum axial attractive force. The efficiency was evaluated at a typical operating point with a load torque of \( T_{Load} = 30 \text{ mNm} \) and a speed of \( n = 2100 \text{ rpm} \). The rotor inertia was analytically calculated with the geometry parameters, the mass densities of the utilized iron and neodymium-iron-boron (NdFeB) permanent magnet materials \( (\rho_{Fe}, \rho_{NdFeB}) \), and the number of rotor poles \( (2p) \) as per equation (3.20):
The variation of the geometry parameters was started from an initial baseline geometry (BG), whereas the individual effect of each parameter was evaluated within a specified range. A summary of the evaluated parameters, the corresponding baseline value, and the evaluated range are summarized in Table 3-2. For all simulations, a constant permanent magnet grade with a remanence flux density of \( B_r,PM = 1.26 \ T \) was assumed; however, it should be noted that for an industrial motor design the highest permanent magnet grade (with a higher remanence flux density), which can be implemented without the risk of demagnetisation due to temperature, may be chosen to minimise the required permanent magnet volume to provide the desired air gap flux density.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Baseline</th>
<th>Minimum</th>
<th>Maximum</th>
<th>Step size</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( l_{\text{gap}} )</td>
<td>3</td>
<td>2</td>
<td>5</td>
<td>0.5</td>
<td>mm</td>
</tr>
<tr>
<td>( r_{\text{core}} )</td>
<td>14</td>
<td>14</td>
<td>20</td>
<td>1</td>
<td>mm</td>
</tr>
<tr>
<td>( w_s )</td>
<td>6.5</td>
<td>3</td>
<td>9.5</td>
<td>0.5</td>
<td>mm</td>
</tr>
<tr>
<td>( r_l )</td>
<td>10.75</td>
<td>10.75</td>
<td>18.5</td>
<td>–</td>
<td>mm</td>
</tr>
<tr>
<td>( d_s )</td>
<td>6</td>
<td>3</td>
<td>12</td>
<td>1.5</td>
<td>mm</td>
</tr>
<tr>
<td>( h_{\text{pm}} )</td>
<td>1.2</td>
<td>1</td>
<td>2.6</td>
<td>0.2</td>
<td>mm</td>
</tr>
<tr>
<td>( \alpha_{\text{pm}} )</td>
<td>35</td>
<td>15</td>
<td>35</td>
<td>5</td>
<td>deg</td>
</tr>
<tr>
<td>( h_{y,r} )</td>
<td>1.5</td>
<td>1</td>
<td>2.5</td>
<td>0.25</td>
<td>mm</td>
</tr>
<tr>
<td>( h_{y,s} )</td>
<td>1.5</td>
<td>1</td>
<td>2.5</td>
<td>0.25</td>
<td>mm</td>
</tr>
</tbody>
</table>

In a subsequent investigation to improve motor characteristics, combinations of parameter changes were derived from the initial findings, and the corresponding geometries were simulated. Firstly, an optimal ratio of the permanent magnet (PM) angle and thickness was evaluated. Therefore, a parameter variation of \( \alpha_{\text{pm}} \) for a constant magnet volume was performed, i.e. \( h_{\text{pm}} \) was adjusted for each value of \( \alpha_{\text{pm}} \) according to equation (3.21):

\[
J_{\text{rot}} = \left( r_{\text{core}}^4 - r_{\text{c,core}}^4 \right) \cdot \left( \frac{2p}{4} \cdot \frac{2\pi}{360\,\text{deg}} \cdot h_{\text{pm}} \cdot \rho_{\text{NdFeB}} + \frac{\pi}{2} \cdot h_{y,r} \cdot \rho_{\text{Fe}} \right)
\]

(3.20)

\[
h_{\text{pm}} = \frac{35\,\text{deg}}{\alpha_{\text{pm}}} \cdot h_{\text{pm,35}}
\]

\[
h_{\text{pm,35}} = 1.2, 1.5, 1.8, \ldots 2.7\,\text{mm}
\]

\[
\alpha_{\text{pm}} = 15, 20, 25, 30, 35\,\text{deg}.
\]

(3.21)
Secondly, a suitable PM angle $\alpha_{pm}$ to maximize the motor efficiency was chosen, and a combined parameter variation of the PM thickness $h_{pm}$ and the gap length $l_{gap}$ was performed, to find the required PM thickness, for which the efficiency of the BG can be maintained, when the gap length is increased.

Subsequently, the ratio of inner core radius $r_{i,\text{core}}$ and slot width $w_s$ leading to the highest efficiency for a specified total inner radius $r_i$ (equation (3.18)) was investigated for values of $r_i = 12, 13, 14, 15, \text{ and } 16 \text{ mm}$. Lastly, for each of the chosen combinations of $r_{i,\text{core}}$ and $w_s$, a required set of parameters to restore the axial force and efficiency of the BG was found.

### 3.2.4 Motor Control
At the end of this chapter, a brief discourse with respect to the control of the RBP motor is presented. The two most widely used control algorithms for brushless DC (BLDC) are the field-oriented control (FOC) and trapezoidal control (TC). Throughout the FEM motor evaluation described above, it was presumed that the motor is operated with a FOC algorithm with ideal commutation, while the cogging torque was neglected. Consequently, the calculated efficiency and force characteristics represent a best-case scenario, as the motor is assumed to constantly operate with its optimal efficiency with respect to the defined operating point. Conversely, with the commonly used TC algorithm, the stator currents are commutated with a step size of 60° electrical, thus the commutation angle varies within a margin of $\pm 30^\circ$ from its optimum. Therefore, the generated electromagnetic torque and consequently efficiency may be reduced, while undesirable stator field components generating a positive or negative axial force may be introduced. The influence on torque and force of the two control algorithms were compared simulating the rotor rotation of one electrical period for the reference motor (RM), and the implications with respect to rapid speed modulation are briefly discussed.

### 3.3 Results
In the following section the results of the parameter study performed with the 2D-FEM framework are presented. First, isolated changes of all parameters were evaluated. Subsequently, the effect of large gap lengths and inner stator radii, which may allow for improvement of the hydraulic characteristics, were investigated and a design strategy to maintain motor characteristics despite changes of these parameters is presented.
Lastly, two motor control strategies are briefly compared under consideration of rapid impeller speed modulation.

### 3.3.1 Isolated Parameter Changes

Using the validated 2D model to allow rapid exploration of the design envelope, a study was set up to evaluate the effects of geometry changes. Each parameter was varied in isolation, while all other parameters remained constants. The initial results are summarized in Figure 3-13. The panels show the effects of the air gap length \( l_{\text{gap}} \), slot width \( w_s \), slot depth \( d_s \), inner core radius \( r_{\text{core}} \), PM thickness \( h_{\text{pm}} \), and PM angle \( \alpha_{\text{pm}} \) on motor efficiency \( \eta_{\text{mot}} \), axial attractive force \( F_z \), and rotor inertia \( J_{\text{rot}} \), copper losses \( P_{\text{cu}} \), and stator core losses \( P_{\text{core}} \).

The solution corresponding to the baseline geometry (BG) is circled in all panels. An efficiency of \( \eta_{\text{mot}} = 75.04\% \), axial force of \( F_z = 16.91 \, N \), and rotor inertia of \( J_{\text{rot}} = 10.39 \cdot 10^{-6} \, kg \cdot m^2 \) were found. All varied parameters affected \( \eta_{\text{mot}} \), whereas an increase of \( l_{\text{gap}} \) had the most significant effect, resulting in a near linear reduction of almost 25\%, over an increase of 2\,mm. The corresponding axial force decreased exponentially. Due to the slot and inner diameter geometry, an increase of \( w_s \) from the BG was not possible. A decrease of \( w_s \) from 6.5\,mm to 3\,mm resulted in a decrease of \( \eta_{\text{mot}} \) of approximately 14\%, whereas the axial force increased linearly from 16.91\,N to 23.97\,N due to tan increase in pole face area. The slot depth showed a strong effect on \( \eta_{\text{mot}} \) with an asymptotic increase, and a maximum efficiency of 84.38\%, when the slot depth was doubled from \( d_s = 6 \, mm \) to \( d_s = 12 \, mm \). A decrease of \( d_s \) to 3\,mm had a stronger effect, reducing \( \eta_{\text{mot}} \) to 56.31\%. The axial force marginally reduced with increasing slot depth, with a maximum of \( F_z = 17.69 \, N \) at \( d_s = 3 \, mm \), and a minimum of \( F_z = 16.49 \, N \) at \( d_s = 12 \, mm \). The increase of the inner core radius resulted in a strong reduction of efficiency and axial force to a minimum of \( \eta_{\text{mot}} = 49.43\% \) and \( F_z = 6.99 \, N \) at \( r_{\text{core}} = 20 \, mm \). An increase of the permanent magnet (PM) thickness from \( h_{\text{pm}} = 1.2 \, mm \) to \( h_{\text{pm}} = 2.6 \, mm \) linearly increased \( F_z \) to a maximum of 33.2\,N and asymptotically increased \( \eta_{\text{mot}} \) to a maximum of 81.72\%. The only two parameters affecting the rotor geometry and thus influencing the rotor inertia were \( r_{\text{core}} \) and \( h_{\text{pm}} \). The largest simulated PM thickness resulted in an increase of \( J_{\text{rot}} \) to a maximum of \( 1.56 \cdot 10^{-6} \, kg \cdot m^2 \), which is an increase of 49.18\% compared to the BG. An increase of
the inner core radius had an opposite effect, decreasing the rotor inertia by 37.7% to $J_{rot} = 6.47 \cdot 10^{-6} \text{ kg} \cdot \text{m}^2$. A decreasing PM angle $\alpha_{pm}$ resulted in a decrease of efficiency, axial force, and rotor inertia.

Figure 3-13 – Results for the isolated parameter variation. Panels show the effects of the (A) gap length $l_{gap}$, (B) slot width $w_s$, (C) slot depth $d_s$, (D) inner core radius $r_{i,core}$, (E) permanent magnet thickness $h_{pm}$, and (F) permanent magnet angle $\alpha_{pm}$ on motor efficiency ($\eta_{mot}$), axial attractive force ($F_z$), rotor inertia ($J_{rot}$), copper losses ($P_{cu}$), and stator core losses ($P_{core}$). Circled solutions indicate the baseline geometry.
The panels in the two bottom rows show the copper losses and core losses, for all geometries. Generally, the core losses were substantially lower than the copper losses (approximately 11.5% of $P_{cu}$ for the BG). Values of up to $P_{core} = 0.43 \, W$ were found when the gap length was decreased or the permanent magnet thickness was increased. Both geometry adjustments increase the magnetic coupling between stator and rotor, and thus were accompanied by a substantial increase of the axial force. For these geometries, the copper losses were inversely affected, as the current requirement to generate the desired torque was reduced. Similarly, a change of the slot depth increases the cross-sectional area of the coils (and thus the winding turn number),

While the above presented results show the effect of isolated parameter changes, it is difficult to understand the effect of combinations of parameter variations. For example, an increase of the inner core radius allows for a larger pump inlet diameter, however, it resulted in a costly decrease of the motor efficiency, especially above 17 mm. An alternative approach to analyse the data is shown in Figure 3-14. The plotted curves represent the same solutions, as presented above, whereas the motor efficiency is plotted against the axial attractive force. The solutions of each parameter variation resulted in a characteristic trajectory within the $F_z - \eta_{mot} -$plane, which suggested, that the effect of combined parameter changes may correspond to a combination of their respective trajectories. An example of such a combination, for a change of the inner core radius $r_{i,\text{core}}$ and the slot depth $d_s$ is indicated by the red arrows in Figure 3-14.
3.3.2 Permanent Magnet Shape and Gap Length

It was noticeable, that the trajectories for \( l_{gap} \), \( h_{pm} \), and \( \alpha_{pm} \) were approximately in line with each other, i.e. the relative change of the motor efficiency and axial attractive force is similar. Therefore, a combined evaluation of these parameters is reasonable. Firstly, the effects of the PM angle and height were evaluated. The results are shown in Figure 3-15.

![A) Efficiency](image)

**Figure 3-15** – Effect of the permanent magnet angle \( \alpha_{pm} \) on (A) motor efficiency \( \eta_{mot} \) and (B) axial attractive force \( F_z \), for different constant permanent magnet volumes. Values of \( h_{pm,35} \) represent the magnet height at \( \alpha_{pm} = 35 \) deg, while \( h_{pm} \) was adjusted for each angle according to equation (3.21). The red ellipses indicate the approximate range of \( \alpha_{pm} \), for which the motor efficiency is highest.

As indicated by the red ellipses, the results showed a maximum of both \( F_z \) and \( \eta_{mot} \) for \( \alpha_{pm} \) between 25 deg and 30 deg, which slightly moved towards larger angles with increasing volume (increasing \( h_{pm,35} \)). That is, at these angles the ratio of permanent magnet thickness and angle provided the largest air gap flux density (magnetic coupling between rotor and stator). To simplify the following analysis, a value of \( \alpha_{pm} = 27.5 \) deg was chosen for all following simulations, as it increases the motor efficiency for a specified PM volume (and consequently rotor inertia).

Figure 3-16 shows the results of the combined parameter variation of \( l_{gap} \) and \( h_{pm} \) for \( \alpha_{pm} = 27.5 \) deg. Figure 3-16 A shows the plot of \( \eta_{mot} \) at \( T_{Load} = 30 \) mNm against \( F_z \).
for the previous results at \( \alpha_{pm} = 35 \text{ deg} \), all trajectories for the evaluated air gaps overlapped on a common curve, which further intersected the solution corresponding to the baseline geometry. The same efficiency data are plotted against the PM thickness in Figure 3-16 B. The black dashed line corresponds to the efficiency of the BG. Therefore, the intersection of each parameter variation curve and the black line represents the required PM thickness to restore the characteristics of the BG, when the gap length is increased.

It is apparent, that the efficiency for gap lengths \( l_{gap} \geq 2 \text{ mm} \) increased asymptotically with increasing permanent magnet thickness. While an increase of \( h_{pm} \) had a large effect on \( \eta_{mot} \) at PM thicknesses between 1 mm and 1.5 mm, a further increase at value above 3.5 mm showed a small influence. At a gap length of \( l_{gap} = 5 \text{ mm} \) the efficiency did not reach \( \eta_{mot} \) of the BG, even at the largest evaluated PM thickness \( h_{pm} = 5 \text{ mm} \). At a gap length of \( l_{gap} = 4.5 \text{ mm} \), \( \eta_{mot} \) reached approximately the baseline efficiency with \( h_{pm} = 5 \text{ mm} \), but was marginally lower \((-0.2\%)\). It can further be observed that at a gap length of \( l_{gap} = 1.5 \text{ mm} \) the efficiency slightly decreased with increasing permanent magnet thickness for \( h_{pm} > 1.75 \text{ mm} \), while the axial force continued to increase (Figure 3-16 B). This effect is attributed to the increasing core loss at small gap lengths. The copper and core loss components for all geometries are plotted in Figure 3-17. The graphs show that the copper
loss at 30 $mNm$ rapidly decreased with increasing PM thickness and decreasing gap length, as both geometry changes resulted in an increased magnetic coupling between the rotor and stator and thus reduced the current requirement to generate a specified torque. Conversely, the core loss increased with increasing magnet thickness and decreasing gap length (as expected with stronger magnetic coupling and higher core flux density). Consequently, at large gaps, the core loss was small and negligible compared to the copper loss. Conversely, at a combination of a small gap length and large PM thickness the core loss was of similar magnitude or exceeded the copper loss. However, due to the large increase of the axial attractive forces, these geometries are not suitable for the given application.

![Figure 3-17](image)

Figure 3-17 – (A) Copper and (B) core loss components for all geometries evaluated in the gap length and permanent magnet thickness variation.

The required permanent magnet thickness to yield the same efficiency as the baseline geometry at different air gaps was extracted from the data shown in Figure 3-16 B (intersection of each curve with the dashed line), and plotted against the gap length in Figure 3-18 A. The corresponding values of the rotor inertia are plotted in Figure 3-18 B. The solutions correspond to a PM angle of $\alpha_{pm} = 27.5 \, deg$, while the red markers show the PM thickness and rotor inertia of the BG with $\alpha_{pm} = 35 \, deg$.

At a gap length of 3 $mm$ the efficiency and axial force of the motor with $\alpha_{pm} = 27.5 \, deg$ and $h_{pm} = 1.35 \, mm$ were equivalent to the BG. However, the motor with the adjusted permanent magnets required only 88.4% of the PM mass compared to the BG. The
corresponding inertia was reduced to $J_{rot} = 9.88 \times 10^{-6} \, kg \cdot m^2$, which is a reduction by 4.94%. With increasing gap lengths, the required PM height steeply increased up to $\approx 5 \, mm$ at a gap length of $4.5 \, mm$, which was a factor 3.7 higher than the corresponding $h_{pm}$ at $l_{gap} = 3 \, mm$.

The corresponding rotor inertia was more than doubled. A reduction of the gap lengths allowed a nearly linear reduction in magnet height whilst maintaining axial force and efficiency. At a gap of $l_{gap} = 1.5 \, mm$, the resulting inertia was reduced by 22.1%. The plotted values are summarized in Table 3-3. It should be noted, that the permanent magnet grade was not varied, while magnets with a higher remanence flux density may increase the air gap flux density and thus allow to reduce the required permanent magnet thickness.

Table 3-3 – Required PM thickness ($h_{pm}$) and resulting rotor inertia ($J_{rot}$) to restore axial force and efficiency of the baseline geometry ($\eta_{mot,BG}$) after changing the gap length ($l_{gap}$).

<table>
<thead>
<tr>
<th>$l_{gap}$ [mm]</th>
<th>$h_{pm}$ [mm]</th>
<th>$J_{rot}$ [10^{-6} , kg \cdot m^2]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5</td>
<td>0.6</td>
<td>7.70</td>
</tr>
<tr>
<td>2.0</td>
<td>0.8</td>
<td>8.25</td>
</tr>
<tr>
<td>2.5</td>
<td>1.05</td>
<td>8.95</td>
</tr>
<tr>
<td>3.0</td>
<td>(PM adjusted)</td>
<td>1.35</td>
</tr>
<tr>
<td>3.0</td>
<td>(baseline geometry)</td>
<td>1.2</td>
</tr>
<tr>
<td>3.5</td>
<td>1.8</td>
<td>11.16</td>
</tr>
<tr>
<td>4.0</td>
<td>2.6</td>
<td>13.47</td>
</tr>
<tr>
<td>4.5</td>
<td>$\approx 5.0$</td>
<td>$\approx 20.46$</td>
</tr>
<tr>
<td>5.0</td>
<td>($\eta_{mot,BG}$ not reached)</td>
<td>(&gt; 5.0)</td>
</tr>
</tbody>
</table>
3.3.3 Inner Stator Radius

Figure 3-19 shows the effects of changes of the (total) inner stator radius \( r_i \) at the initial gap length of \( l_{\text{gap}} = 3 \, \text{mm} \); the magnet geometry was changed according to above findings \( (\alpha_{\text{pm}} = 27.5 \, \text{deg}; h_{\text{pm}} = 1.35 \, \text{mm}) \). As both the inner core radius \( (r_{i,\text{core}}) \) and slot width \( (w_s) \) affect the total inner radius, a combined variation of the two parameters was performed. Figure 3-19A shows the results for all valid combinations, illustrating the change of the axial attractive force \( (F_z) \) and the motor efficiency \( (\eta_{\text{mot}}) \). With increasing \( r_{i,\text{core}} \) both the efficiency and attractive force reduced.

\[
\begin{align*}
\text{A)} & \quad \text{Efficiency vs. axial force} \\
\text{B)} & \quad \text{Efficiency vs. } r_i \\
\text{C)} & \quad \text{Axial force vs. } r_i \\
\text{D)} & \quad \text{Rotor inertia vs. } r_i
\end{align*}
\]

Figure 3-19 – Effects of the variation of inner core radius \( (r_{i,\text{core}}) \) and slot width \( (w_s) \). Panels show the relationships of (A) axial force \( (F_z) \) and motor efficiency \( (\eta_{\text{mot}}) \), (B) inner stator radius \( (r_i) \) and \( \eta_{\text{mot}} \), (C) \( r_i \) and \( F_z \), and (D) \( r_i \) and the rotor moment of inertia \( (J_{\text{rot}}) \). Red circles correspond to the baseline geometry.
An increase of the slot width at constant \( r_{i,\text{core}} \) typically resulted in an increase of \( \eta_{\text{mot}} \), while \( F_z \) decreased, however, for \( r_{i,\text{core}} \geq 18 \text{ mm} \) the efficiency showed a maximum at large \( w_s \) between 7 and 8 mm. A further increase of \( w_s \) resulted in a decrease of the efficiency, while the axial force continued to decrease. The maximum efficiency (\( \eta_{\text{mot}} = 75.04\% \)) was reached with the largest simulated \( r_{i,\text{core}} \) at the maximum slot width of \( w_s = 6.5 \text{ mm} \). In contrast, \( \eta_{\text{mot}} \) reduced to values as low as 30.32\% at \( r_{i,\text{core}} = 20 \text{ mm} \) and the lowest simulated slot width of \( w_s = 3 \text{ mm} \). Results for the same simulations are shown in Figure 3-19 B, whereas the efficiency is plotted as function of the total inner stator radius (\( r_i \)). It can be observed, that the curves for constant inner core radii intersect and overlap at several points. For example, an inner radius of \( r_i = 15 \text{ mm} \) can be designed with inner core radii \( r_{i,\text{core}} \) of 17, 18, and 19 mm, and corresponding slot widths of 4.6, and 8 mm respectively. For this example, the highest efficiency (\( \eta_{\text{mot}} = 63.26\% \)) was found for the combination of \( r_{i,\text{core}} = 18 \text{ mm} \) and \( w_s = 6 \text{ mm} \). The corresponding axial force (\( F_z = 11.68 \text{ N} \)) and rotor inertia (\( J_{\text{rot}} = 8.25 \cdot 10^{-6} \text{ kg} \cdot \text{m}^2 \)) were in the mid-range between the corresponding geometries with inner core radii of 17 (\( F_z = 16.37 \text{ N} ; J_{\text{rot}} = 8.94 \cdot 10^{-6} \text{ kg} \cdot \text{m}^2 \)) and 20 mm (\( F_z = 7.53 \text{ N} ; J_{\text{rot}} = 7.43 \cdot 10^{-6} \text{ kg} \cdot \text{m}^2 \)). Consequently, when only combinations of \( r_{i,\text{core}} \) and \( w_s \), which result in a specified inner radius, are considered, an optimal combination with respect to the maximum efficiency can be found.

![Graph showing motor efficiency \( \eta_{\text{mot}} \) for the synchronized variation of \( r_{i,\text{core}} \) and \( w_s \), for a constant inner stator radius of \( r_i = 15 \text{ mm} \).](image)
A corresponding example is shown in Figure 3-20, which shows $\eta_{mot}$ as a function of $r_{l,core}$, which was varied between 16.5 mm and 19.5 mm in steps of 0.5 mm. For each geometry, the slot width was adjusted, such that the inner stator radius is $r_i = 15 mm$. The resulting curve showed a parabolic shape, whereas the maximum efficiency ($\eta_{mot} = 63.26\%$) was found at $r_{l,core} = 18 mm$ and $w_s = 6 mm$. The efficiency decreased towards both higher and lower values of $r_{l,core}$, reaching a minimum within the evaluated range of $\eta_{mot} = 53.47\%$ at $r_{l,core} = 16.5 mm$ ($w_s = 3 mm$).

### 3.3.4 Combined Parameter Adjustments

Consequently, when a specified inner stator radius $r_i$ is given, it is expedient to choose the combination of $r_{l,core}$ and $w_s$ which yields maximum efficiency. However, due to the increase of $r_i$, the efficiency and axial force of this geometry were reduced. Therefore, changes of the other geometry parameters which allowed improvement of the characteristics were investigated.

![Graph showing $\eta_{mot}$ vs. $F_z$](image)

Figure 3-21 – Example of geometry adjustments leading to approximately the same axial force ($F_z$) and motor efficiency ($\eta_{mot}$) as the baseline geometry (red circle), while increasing the inner stator radius $r_i$ from 10.75 mm to 15 mm (with $r_{l,core} = 18 mm$; $w_s = 6 mm$). An increase of the inner core radius ($r_{l,core}$) and decrease of the slot width ($w_s$) resulted in reduced efficiency. The baseline efficiency was restored through an increase of the permanent magnet thickness ($h_{pm}$) and slot depth ($d_s$).
The previous results of the isolated parameter variation (cf. Figure 3-14) suggested, that the efficiency and force can be restored through adjustments of the PM height \(h_{pm}\) and slot depth \(d_s\). Therefore, these parameters were iteratively adjusted to yield a motor geometry exhibiting similar characteristics as the BG. The trajectories in the \(F_z - \eta_{mot}\) - plane corresponding to the geometry adjustments made in the example of \(r_i = 15\ mm\) are shown in Figure 3-21. The figure shows the results for a parameter variation, starting at the baseline geometry (red circle) with the equivalently adjusted PM geometry \(\alpha_{pm} = 27.5\ deg; h_{pm} = 1.35\ mm\). Firstly, the inner core radius and slot width were adjusted to \(r_{i,core} = 18\ mm\) and \(w_s = 6\ mm\), resulting in an inner stator radius of \(r_i = 15\ mm\), increased from 10.75\ mm. To restore the BG force and efficiency, the PM thickness was increased to \(h_{pm} = 2\ mm\), and the slot depth was increased to \(d_s = 7.6\ mm\). The deviation of the motor efficiency \((\Delta\eta_{mot} = 0.09\%)\) and axial force \((\Delta F_z = 0.05\ N)\) between the start and end points of the combined trajectory were negligibly small.

The same methodology was further applied to inner radii of \(r_i = 12, 13, 14,\) and \(16\ mm\). For each targeted inner radius, and an optimal ratio of \(r_{i,core}\) and \(w_s\) to maximize the efficiency was found. Subsequently, the required values of the slot width and permanent magnet thickness to restore \(F_z\) and \(\eta_{mot}\) of the baseline geometry were respectively derived. The required parameters for all evaluated \(r_i\) are shown in Figure 3-22. The slot width at which the highest efficiency was yielded tended to decrease with increasing desired \(r_i\). However, as a minimum step size of \(0.5\ mm\) was evaluated, a value of \(w_s = 6\ mm\) was chosen for inner radii of \(r_i = 13, 14,\) and \(15\ mm\). The slot widths corresponding to \(r_i = 12\ mm\) and \(16\ mm\) were chosen as \(w_s = 6.5\ mm\) and \(5.5\ mm\) respectively. Consequently, the inner core radius increased almost linearly with the targeted inner stator radius. The required permanent magnet size and slot depth increased steadily, while a stronger increase at large \(r_i\) was observed. Despite a maximum increase of \(h_{pm}\) by 66\% (at \(r_i = 16\ mm\)), the calculated rotor inertias for all evaluated inner radii were below the inertia of both the BG, and BG with adjusted PM shape. The minimum inertia \((J_{rot} = 9.25 \cdot 10^{-6} kg \cdot m^2)\) was found at \(r_i = 14\ mm\). However, it should be noted, that although the inertia was reduced, the relative change of \(J_{rot}\) compared to the required adjustments in response to a change of the gap length (cf. section 3.3.2) was small.
Figure 3-22 – Parameters for adjusted geometries for target inner stator radii of $r_i = 12, 13, 14, 15,$ and $16 \ mm$. The ratio of inner core radius $r_{i,\text{core}}$ and slot width $w_s$ was adjusted to yield the highest efficiency ($\eta_{\text{mot}}$) for all cases, then the slot depth $d_s$ and PM thickness $h_{pm}$ were increased to restore the efficiency and axial force ($F_z$) of the baseline geometry. Red markers show parameters corresponding to the baseline geometry with (cross) and without (circle) PM adjustment to $\alpha_{pm} = 27.5 \ deg$, $h_{pm} = 1.35 \ mm$.

Panels show the (A) required slot depth, (B) PM thickness, (C) rotor inertia, and (D) inner core radius.

The required slot depth increased up to a maximum of $d_s = 9 \ mm$, which is a 50% increase compared to the baseline geometry. A summary of the geometry parameters and rotor inertias of the BG and all adjusted motor geometries is given in Table 3-4.
Table 3-4 – Required PM thickness \((h_{pm})\) and slot depth \((d_s)\) to restore axial force and efficiency of the baseline geometry \((BG)\) after changing the inner stator radius \((r_i)\) through changes of the inner core radius \((r_i,\text{core})\) and the slot width \((w_s)\).

<table>
<thead>
<tr>
<th>(r_i) [mm]</th>
<th>(r_i,\text{core}) [mm]</th>
<th>(w_s) [mm]</th>
<th>(d_s) [mm]</th>
<th>(h_{pm}) [mm]</th>
<th>(J_{rot}) ([10^{-6} \text{ kg} \cdot \text{m}^2])</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.75 ((BG))</td>
<td>14.0</td>
<td>6.5</td>
<td>6</td>
<td>1.2</td>
<td>9.88</td>
</tr>
<tr>
<td>10.75 ((BG \text{ adj.}))</td>
<td>14.0</td>
<td>6.5</td>
<td>6</td>
<td>1.35</td>
<td>10.39</td>
</tr>
<tr>
<td>12.0</td>
<td>15.25</td>
<td>6.5</td>
<td>6.15</td>
<td>1.45</td>
<td>9.68</td>
</tr>
<tr>
<td>13.0</td>
<td>16.0</td>
<td>6</td>
<td>6.85</td>
<td>1.48</td>
<td>9.38</td>
</tr>
<tr>
<td>14.0</td>
<td>17.0</td>
<td>6</td>
<td>7.2</td>
<td>1.65</td>
<td>9.25</td>
</tr>
<tr>
<td>15.0</td>
<td>18.0</td>
<td>6</td>
<td>7.6</td>
<td>2.0</td>
<td>9.33</td>
</tr>
<tr>
<td>16.0</td>
<td>18.75</td>
<td>5.5</td>
<td>8.9</td>
<td>2.35</td>
<td>9.40</td>
</tr>
</tbody>
</table>

3.3.5 Yoke Thickness

In the previously presented evaluation, the influence of the stator \((h_{y,s})\) and rotor yoke thicknesses \((h_{y,r})\) was neglected. However, prior to the analysis, the influence of both parameters was evaluated. Both yoke thicknesses were varied between values of 0.5 mm and 2 mm. The corresponding effect on the motor efficiency is shown in Figure 3-23. The results showed a strong efficiency reduction of almost 10% when \(h_{r,y}\) was decreased to 0.5 mm.

![Figure 3-23](image)

Figure 3-23 – Effect of (A) the rotor yoke thickness \(h_{y,r}\) and (B) stator yoke thickness \(h_{y,s}\) on motor efficiency. The red circles indicate the baseline geometry.

However, when the rotor yoke thickness was increased, the gain in efficiency was marginal. Similarly, an increase of the stator yoke thickness did not substantially affect \(\eta_{mot}\). While a decrease of \(h_{y,s}\) resulted in an efficiency reduction, the minimum value at \(h_{y,s} = 0.5 \text{ mm}\) was 70.05%. 

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Due to the low influence of both parameters, they were widely neglected in the geometry evaluation. However, specifically when a large permanent magnet thickness is chosen, the motor performance may be deteriorated due to rotor yoke saturation, if $h_{y,r}$ is not adjusted appropriately. The results presented in section 3.3.2 showed that at a gap length of $l_{gap} = 5 \text{ mm}$, the efficiency of the BG could not be restored despite a large increase of the permanent magnet thickness $h_{pm} = 5 \text{ mm}$. Therefore, the effect of increasing the rotor yoke thickness was additionally evaluated for this specific case. The results for values of $h_{y,r}$ up to $3 \text{ mm}$ are shown in Figure 3-24.

![Figure 3-24](image)

Figure 3-24 – Influence of the rotor yoke thickness $h_{y,r}$ on motor efficiency at a permanent magnet thickness of $h_{pm} = 5 \text{ mm}$ and gap length of $l_{gap} = 5 \text{ mm}$. All other geometry parameters are unchanged from the baseline geometry.

The curve shows, that a slight increase of $\eta_{mot}$ was yielded with an increase of $h_{y,r}$ up to $2.25 \text{ mm}$. However, despite the increased yoke thickness, a maximum value of $\eta_{mot} = 71.75\%$ was found, thus due to the large air gap reluctance the efficiency could not be restored to that of the baseline geometry ($\eta_{mot,BG} = 73.53\%$). The results indicate that yoke saturation had only a marginal effect on the drive performance in the discussed case.

While the results showed that the influence of partial local saturation (as observed in the rotor yoke) was small the decrease of both rotor and stator yoke thickness highlighted the potential performance deterioration, when a too small yoke thickness is chosen. Consequently, considering the objective to reduce the rotor inertia, the obvious approach is to minimize $h_{y,r}$ to the point at which saturation effects begin to become prominent. Similarly, the stator yoke
thickness would be minimized to reduce the stator height. However, the results justified the limited attention paid to both geometry parameters.

3.3.6 Control Algorithm Comparison

Figure 3-25 shows a comparison of the torque and force with trapezoidal commutation and sinusoidal commutation (field-oriented control) computed with the 2D-FEM model for the reference motor.

![Figure 3-25](image)

Figure 3-25 – Comparison of simulation results for trapezoidal and sinusoidal commutation (field-oriented control) showing (A) torque and (B) axial force of the reference motor.

The torque corresponding to the FOC commutation showed a negligible ripple, while the torque corresponding to trapezoidal commutation (TC) shows a large decrease of the torque near the commutation points ($\Theta_{el} = 30, 90, 150 \ldots deg$), which occur with a frequency of six times the electrical frequency. The minimum torque (28.23 mNm) was approximately 15% below the maximum value, which was approximately equal to the average torque with FOC commutation. The axial force with TC showed a large sawtooth-like ripple of approximately 2.5 N at the same frequency. Conversely, the axial force with FOC commutation was almost constant at the mean value of the force with TC.

Figure 3-26 shows a comparison of the torques and forces with TC, simulated with the 3D and 2D FEM models. The results for motor currents of 1A and 4A are shown. The simulated torques were similar between the 3D and 2D models, whereas the 2D-FEM results were within an error of less than 3% lower than the 3D-FEM results. For both models, the relative
torque ripple was approximately the same for both simulated currents. The computed axial forces show a similar pattern and approximately the same average force for all simulation. The amplitude of the force ripple increased linearly with the motor current. While the force error between the 2D and 3D models was low at the average force, the amplitude of the force ripple was underestimated by the 2D model by approximately 30%.

![Graph showing comparison of simulation results for torque and axial force](image)

Figure 3-26 – Comparison of simulation results for (A) torque and (B) axial force of the reference motor with trapezoidal commutation obtained from the 3D and 2D FEM models, evaluated with motor currents of $I_{mot} = 1A, 4A$.

### 3.4 Discussion

The presented motor parameter variation aimed to investigate design trade-offs allowing an increase in the inner stator radius and the magnetic gap length to cater for geometry requirements of the pump which may allow improvement of the hydraulic characteristics and efficiency, specifically for rapid speed modulation. Furthermore, the insights gained from this study allow consideration of the presented results in relation to different design intentions corresponding to the limitations and geometrical constraints of a given device design. In the case of the BiVACOR TAH, maintenance of the axial force characteristic is important to ensure adequate adjustment of the motor and magnetic bearing interaction, which may similarly apply to other devices with an axial magnetic suspension. Conversely, in the case of a hydrodynamic bearing, it may be particularly valuable to reduce the axial force to avoid rotor touchdown. This specifically applies to rapid speed modulation, where variation in hydraulic forces is expected. However, geometrical constraints to the axial stator and rotor heights or the total outer diameter may apply, thus limit the applicable range of parameters.
such as the slot depth, permanent magnet thickness, or slot width, which, due to the end-winding turns, affects the outer diameter.

The isolated parameter variation study illustrated the effects these geometry changes on the axial force, rotor inertia, and efficiency of the motor. With respect to improvements of the efficiency, an increase of the slot width $d_s$ appeared to be the most valuable variation, as it allowed for a substantial increase of the efficiency, while the axial force was slightly reduced and the inertia remained unaffected. An increase of $d_s$ may therefore further allow to maintain a high efficiency, when the objective is to reduce the axial force and/or rotor inertia. For this objective it appeared beneficial to maximise the slot width $w_s$, as it allowed for improvement of the efficiency and reduction of the force in most geometries.

The subsequent analysis showed how multiple geometry parameter changes can be combined to reach a specified axial force and efficiency on the example of the objective to increase the inner stator radius and the axial gap length, whilst maintaining the characteristics of the baseline geometry. Similarly, depending on the design objectives, the methodology can be applied to reach different values of the axial force and efficiency.

The analysis further showed, that an increase of $l_{gap}$ can, to some extent, be offset with an increase of the magnet thickness, without substantially changing the axial force and efficiency. However, the concomitant substantial increase of the rotor inertia suggested, that a large increase of the PM thickness may be detrimental to the performance with rapid speed modulation. While the design of a motor with a large gap length may be feasible, geometries with $l_{gap} > 4 \text{ mm}$ appeared to be more suitable for devices, in which a reduction of the axial force is beneficial to the suspension performance, and the loss of efficiency can be compensated with an increase of the slot width or inner core radius. However, further adjustments of the shape of the PM cross section may allow reduction of the rotor inertia.

The analysis of the combined variation of $r_{i,\text{core}}$ and $w_s$ showed, that the ratio of the two parameters can be optimized with respect to a required inner stator radius. In the presented study, the parameter combination yielding the highest efficiency was chosen, as the subsequent aim for the evaluated geometries was to restore the characteristics of the baseline geometry. However, depending on the design goals, a possible different approach may be to
maximize the ratio of efficiency and axial force, as it may allow to reduce $F_z$ and increase $\eta_{mot}$. In this case, it may be more expedient to maximize the slot width, however, the concomitant increase of the outer radius must be considered.

The evaluation of stator geometries with inner stator radii between $r_i = 12$ and 16 mm showed, that $r_i$ can be increased without substantially adversely affecting the rotor inertia, which was attributed to the reduction of the rotor yoke and permanent magnet volume as their inner radius also increased. However, the sacrifice of an increased stator and/or rotor height may be required to maintain the motor characteristics. Such a geometry adjustment may therefore be valuable, if a larger inner radius allows for a substantial improvement of the hydraulic characteristics.

It may further be possible to find an optimal ratio of $r_{i,core}$ and $w_s$ to minimize the rotor inertia, however, the results should be viewed in light of the gap length variation, which had a significantly larger effect on $J_{rot}$. It is further important to notice that the rotor inertias calculated here only comprise the rotor assembly of the rotor. However, in a RBP, where the rotor assembly is embedded in the impeller hub, additional mass and inertia due to for example the outer impeller shell must be considered. In the case of the BiVACOR TAH, this further comprises the magnetic bearing rotor assembly, hence the motor rotor only accounts for approximately half of the total inertia.

### 3.4.1 Motor Control

The simulation of axial force and torque for the trapezoidal (TC) and field-oriented control (FOC) methods showed that TC has substantial disadvantages with respect to the large force and torque ripple, which may reduce efficiency and introduce impeller vibration. However, it should be noted, that both methods were evaluated under the assumption of ideal computation. In the case of a FOC algorithm with position estimation through an observer, the estimated rotor angle may deviate from the real rotor position, thus cause performance degradation and increased force. Hence, the performance with FOC may vary with the quality of the obtained position estimation.

The comparison of force and torque with TC obtained from the 2D and 3D-FEM models highlighted an important limitation of the 2D model, as an error in the force ripple amplitude
of 30% was found. This error is attributed to the application of the edge factor $f_{edge,F}$ in the force calculation of the 2D model. In section 3.2.1.3, where the edge factor was derived, it was stated that the influence of the motor coils was neglected. This assumption provides reasonable results with ideal commutation with FOC, where ideally no interaction between the stator and rotor fields leading to a net axial force is introduced. However, in TC, where the commutation angle deviates up to 30° from its optimum, this assumption becomes invalid. While the axial force interaction of rotor permanent magnet field and stator iron core is related to the square of the air gap flux density, this is not true for the interaction of rotor and stator fields. Consequently, the modelling approach presented here is not suitable to correctly evaluate the corresponding force components with TC. However, as an approach to improve the model in future work, it may be possible to derive an adjusted edge factor from separate simulations with and without stator and/or rotor excitation, thus obtain the electromagnetic force within a reduced error margin.

3.5 Limitations and Future work

The limitations of the presented study include inaccuracies due to the modelling approach, as well as the limitation to a specific type of motor, whereas the principles of the performed analysis translate to other motor types and sizes and may therefore similarly be applied to a VAD motor. Firstly, the utilized 2D-FEM model showed slight deviations of the calculated force and torque. While the error increased with an increased air gap length, the comparison of simulation and measurement results for the reference motor showed an acceptable error margin for the simulated force, torque, and efficiency. Further, the core loss calculation based on the Steinmetz equation, as well as the approximation of axial force and torque with the application of the presented edge factors may result in a different error margin for different parameter combinations. Consequently, the modelling approach is not suitable for geometry optimisations and fine parameter adjustment to yield a motor exhibiting specified, desired characteristics. However, for the broad evaluation and identification of performance trends presented here, the model accuracy was concluded to be sufficient.

It should be noted, that the evaluation was only performed for a specified motor type (12-slot, 10-pole), whereas a soft magnetic composite (SMC) was the only evaluated core material. While the methodology is similarly applicable, future work may therefore include
the evaluation of motors with different slot-pole combinations, as well as different core materials.

However, it should be noted that the evaluation of core losses with the Steinmetz-equation is not directly applicable to solid ferromagnetic core materials, which may exhibit substantially higher eddy-current losses. A possible extension of the model in the future may therefore be to incorporate transient simulations to allow a more accurate approximation of eddy current losses in such materials.

Lastly, the geometry evaluation was performed for a single operating point and a single outer stator core radius. For an axial flux RBP drive, the core radius as well as the motor operating range are typically defined by the impeller dimensions and the hydraulic pump characteristics. Consequently, the methodology may be adapted to different pump types through variation of the outer motor diameter and the required speed and torque.

3.6 Conclusion

The presented analysis showed that the adjustment of motor design parameters to cater for desired hydraulic characteristics can yield favourable results with respect to the motor efficiency, axial force, and/or rotor inertia. The results further illustrated the effect of increasing gap lengths and inner stator radii, and how the corresponding performance degradation can be compensated through adjustment of stator and rotor parameters. The presented methodology allows a quick estimation of the effects of geometry parameter changes on the motor efficiency, axial force, and rotor inertia. To gain a full understanding how these characteristics affect the pump performance when operated with rapid speed modulation, the relative influence of the hydraulic characteristics, motor efficiency, and rotor inertia will be investigated in the following chapters. This understanding may further guide design decisions for axial flux RBP motors, when rapid speed modulation is considered.
STATEMENT OF CONTRIBUTION TO CO-AUTHORED PUBLISHED PAPER

This chapter includes a co-authored paper. The bibliographic details of the co-authored paper, including all authors, are:


My contribution to the paper involved:

- Designed and performed in-vitro experiments (80%)
- Postprocessing and graphic illustration of data (100%)
- Analysis of results (90%)
- Wrote and edited the paper (80%)

(Signed) ____________________________ (Date) 21 August 2017
Matthias Kleinheyer

(Countersigned) ____________________________ (Date) 22-8-17
Supervisor: Professor Geoffrey Tansley
4 Evaluation of the Influence of Speed Profiles on Pulsatility for a Rotary TAH

In the previous chapters, challenges related to rotary blood pump (RBP) speed modulation and design considerations for electromagnetic drive and impeller suspension systems were discussed. While the presented considerations are applicable regardless of the applied speed modulation mode, the effect of the speed waveform on the motor power consumption and haemodynamic output is unclear. Amongst previous studies, which evaluated the modulation of rotary total artificial heart (TAH) impeller speed, the chosen profiles were typically simple waveforms such as sine or square waves [4,170,229], and the generated waveforms were substantially dissimilar from native physiology. An attempt to alter the speed waveform in a way, that a physiologic arterial pressure waveform is simulated, was reported by Shiose et al. [12]. Although their results could not fully replicate native pulsatility, the differences in the haemodynamics corresponding to different speed profiles were evident in the pulse pressure, and pressure and flow waveforms. Consequently, the speed profile chosen may significantly influence the shape of haemodynamic waveforms, the maximum pulsatility generated, and the required motor power consumption.

In the work presented here, the implications of the speed profile choice were investigated in an in vitro study utilising the BiVACOR TAH (BiVACOR V2). The haemodynamic outflow of six speed modulation profiles was characterised with four different metrics quantifying pulsatility in the systemic arterial pressure and flow waveforms. The results were compared with respect to motor power consumption (important for battery drain), to evaluate the potential long-term feasibility of impeller speed modulation with respect to the devices battery lifetime. The in-vitro evaluation results presented in this chapter have previously been published in a peer-reviewed journal [8].

4.1 Aim

The aim of this chapter was to evaluate the influence of different speed profile shapes on the generated haemodynamic pulsatility and required motor power consumption. The objectives related to this aim were:

- Evaluate the effect of speed profiles on different metrics of pulsatility and identify the associated characteristics of the speed profiles related to these metrics.
• Identify the effect of different speed profiles and their amplitude on the motor power consumption.

4.2 Methods
Different metrics to quantify pulsatile haemodynamics, which are commonly used in literature, were previously discussed. In the evaluation presented here, the pulse pressure ($PP$), surplus haemodynamic energy ($SHE$), the arterial $dP/dt$, and the pulse power index ($PPI$) were applied to evaluate the differences amongst implemented speed profiles. To remind the reader of the metrics used and the corresponding formulae, a brief summary is given below. Subsequently, the utilised mock circulatory loop (MCL) is described and the experimental protocol is outlined.

4.2.1 Quantification of pulsatility
The arterial pulse pressure ($PP$, equation (4.1)) is the most commonly used metric to compare pulsatile haemodynamics, and therefore was evaluated for all applied pulse waveforms. It is defined as the difference between the maximum systolic pressure ($P_{ao,max}$) and the minimum diastolic pressure ($P_{ao,min}$) in the aorta [89]:

$$PP = P_{ao,max} - P_{ao,min}.$$  \hspace{1cm} (4.1)

Surplus haemodynamic energy ($SHE$, equation (4.2)) has been suggested to better reflect the energy content of the aortic pressure and flow waveforms. The fraction in the equation is known as energy equivalent pressure ($EEP$), which represents the total haemodynamic energy per volume passing through the cross section of the aorta [113]. The difference between $EEP$ and the mean arterial pressure ($MAP$) serves as a measure of the additional haemodynamic energy per volume ejected in each pulse cycle compared to the continuous flow case. The factor $1,332$ converts the unit from $mmHg$ to $ergs/cm^3$ ($= 0.1 J/m^3$ in SI units). Although $erg$ is an unconventional and outdated unit, it is commonly used in the context of $SHE$, hence it was used here to maintain comparability with other studies. It should further be noted, that despite its name, $SHE$ is measured in a unit of pressure.

$$SHE = 1,332 \left( \frac{\int Q_s \cdot P_{ao} dt}{\int Q_s dt} - MAP \right) = 1,332 \cdot (EEP - MAP),$$  \hspace{1cm} (4.2)
Due to its potential effect on the baroreflex [115,130] and thus the autoregulatory response, the rate of change of the aortic pressure \( (dP/dt, \text{equation (4.3)}) \) is an increasingly used metric for pulsatility [104,116,117,230]. It was therefore evaluated in this study:

\[
dP/dt = \left( \frac{d}{dt} P_{ao} \right)_{max}.
\]  

(4.3)

Lastly, the pulse power index (PPI) as first proposed by Grossi et al. in [120] was evaluated. The PPI represents the relative power of a pulsatile waveform with respect to non-pulsatile equivalent flow [118] and is a relative measure of the alternating component of the flow rate. It is calculated from the square of the harmonic arterial flow waveform components, which are weighed with their respective frequency, and set in relation to the square of the mean flow rate (equation (4.4)):

\[
PPI = \sum_{i=0}^{n} \frac{Q_{5,i}^2 \cdot \omega_i^2}{Q_{5,0}^2},
\]  

(4.4)

where:

- \( Q_{5,i} \), amplitude of the \( i^{th} \) flow harmonic
- \( \omega_i \), frequency of the \( i^{th} \) flow harmonic
- \( Q_{5,0} \), mean flow rate.
- \( n \), number of included harmonic components.

### 4.2.2 In vitro setup

Six different speed waveforms were used to induce an arterial pulse into the BiVACOR TAH. Pump performance was evaluated in vitro with regard to pulsatile haemodynamic values and electrical power consumption, comparing four simple and two more-complex custom waveforms over a wide range of pulse amplitudes. The speed profile comparison was
performed with the (then) latest iteration of the BiVACOR (BiVACOR V2) in a previously
developed mock circulatory loop [99], which was modified for TAH evaluation (Figure 4-1).
Systemic and pulmonary circuits were built of rigid, clear polyvinyl chloride (PVC) pipe
elements. Atrial and venous compliances were represented by vertical pipes open to the
atmosphere; the pulmonary arterial compliance was represented by a lumped, compressed air
compliance chamber. In contrast to the previously described in vitro setup (chapter 2), the
systemic arterial section of the MCL was simulated with 300 mm length of 35 mm (1\text{\frac{3}{8}}\text{inch})
silicone rubber tubing to reduce the effect of high-frequency pressure wave reflections, which
were dampened by the viscoelastic walls of the tubing. The tubing’s distensibility was
modified by constrictive bands, which were placed around the tubing to partially limit radial
expansion and to tune aortic compliance to a physiologic value (1.2 ml/mmHg) at a mean
pressure of 90 mmHg.

Figure 4-1 – Schematic setup of the total artificial heart MCL. AOC, aortic compliance;
LA, left atrium; PAC, pulmonary arterial compliance; PVC, pulmonary venous
compliance; PVR, pulmonary vascular resistance; $Q_P$, pulmonary flow meter; $Q_S$,
 systemic flow meter; RA, right atrium; SVC, systemic venous compliance; SVR,
systemic vascular resistance.

Pressure measurements were performed with fluid-filled pressure transducers (PX600F,
Edwards Lifesciences Corp., Irvine, CA, USA) at each of the pump inlets and outlets. Flow
rates were measured with two clamp-on ultrasonic flow probes (BioProTT \text{\frac{3}{4}}-inch, em-tec
GmbH, Finning, Germany). Pulmonary vascular resistance (PVR) and systemic vascular
resistance (SVR) were adjusted by using pneumatic pinch valves (VMP032, AKO Armaturen
& Separationstechnik GmbH, Trebur-Astheim, Germany) that were supplied with compressed air from voltage-controlled pneumatic regulators (ITV2030, SMC Corporation, Tokyo, Japan). All measurements were performed at room temperature with a 40%wt glycerol-water solution to mimic the viscosity of blood at 37°C. Data postprocessing was performed with MATLAB (MathWorks, Natick, MA, USA).

4.2.3 Experimental protocol

The mock circulatory loop was set to a baseline condition typical of a resting human at constant TAH speed. Vascular resistances were set to \( SVR = 1450 \text{ dyn} \cdot s/cm^5 \) and \( PVR = 100 \text{ dyn} \cdot s/cm^5 \). The pump speed was adjusted to achieve a mean systemic flow rate \( (Q_s) \) of 5 \( L/min \), resulting in left \( (H_L = P_{ao} - P_{la}) \) and right pump pressure heads \( (H_R = P_{pa} - P_{ra}) \) of 80 mmHg and 17 mmHg, respectively. Table 4-1 shows the baseline settings and haemodynamic data.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Systemic vascular resistance</td>
<td>SVR</td>
<td>1450 \text{ dyn} \cdot s/cm^5</td>
</tr>
<tr>
<td>Pulmonary vascular resistance</td>
<td>PVR</td>
<td>100 \text{ dyn} \cdot s/cm^5</td>
</tr>
<tr>
<td>TAH impeller speed</td>
<td>( n )</td>
<td>1950 rpm</td>
</tr>
<tr>
<td>Motor power consumption</td>
<td>( P_{el} )</td>
<td>6.14 W</td>
</tr>
<tr>
<td>Left atrial pressure</td>
<td>( P_{la} )</td>
<td>11 mmHg</td>
</tr>
<tr>
<td>Aortic pressure</td>
<td>( P_{ao} )</td>
<td>91 mmHg</td>
</tr>
<tr>
<td>Right atrial pressure</td>
<td>( P_{ra} )</td>
<td>6 mmHg</td>
</tr>
<tr>
<td>Pulmonary arterial pressure</td>
<td>( P_{pa} )</td>
<td>23 mmHg</td>
</tr>
<tr>
<td>Left pump pressure head</td>
<td>( \Delta P_L )</td>
<td>80 mmHg</td>
</tr>
<tr>
<td>Right pump pressure head</td>
<td>( \Delta P_R )</td>
<td>17 mmHg</td>
</tr>
<tr>
<td>Systemic flow rate</td>
<td>( Q_s )</td>
<td>5 ( L/min )</td>
</tr>
<tr>
<td>Pulmonary flow rate</td>
<td>( Q_p )</td>
<td>7.5 ( L/min )</td>
</tr>
</tbody>
</table>

Subsequently, different periodic pulsatile speed profiles, which had been used previously by other groups [11,12,164], were applied to the pump motor. These profiles included a sine wave, square waveforms with duty cycles of 20% and 50%, a backward-facing sawtooth function, and two empirically derived lookup-table (LUT) based speed profiles. A Proportional-Integral (PI) control loop, which was executed once per pulse cycle, was used to offset the mean pump speed and thus readjust the mean systemic flow rate to 5 \( L/min \)
(controller constants were: \( K_p = 5 \text{ rpm L/min} \); \( K_i = 50 \text{ rpm L/min s} \)). All speed profiles were normalised to a range of [0,1], and then multiplied with an amplitude factor to generate a specified peak-to-peak speed amplitude. The beat rate was kept constant at 60 bpm, while the amplitude factor was slowly increased from 0 to 1000 rpm. A block diagram of the implemented pulse control system is shown in Figure 4-2.

![Block diagram of the implemented pulse control system](image)

**Figure 4-2** – Implemented pulse control system. The normalised pulse profile shapes were multiplied with the output of a ramp generator. The mean speed was adjusted with a variable offset to maintain a mean flow rate of \( Q_s = 5 \text{ L/min} \) through PI-control.

Haemodynamic and pump characteristics were recorded, and pulsatility was quantified by calculating \( PP, dP/dt, PPI, SHE \), and the mean motor power consumption \( (P_{Mean}) \). Because the speed profiles varied in shape, each profile generated a different \( PP \) at a given speed amplitude, hence the comparison of haemodynamics at the same speed amplitude is misleading. Therefore, the haemodynamic characteristics of each speed profile were, compared in two ways. First, \( dP/dt, PPI, SHE \), and \( P_{Mean} \) were evaluated as a function of \( PP \) to compare the shape and energy content of the pressure and flow waveforms at the same pulse amplitude. Second, \( dP/dt, PPI \), and \( SHE \) were normalised to \( P_{Mean} \) and evaluated as a function of \( PP \) (equations (4.5) – (4.7)). The normalised quantities represented the power specific pulsatility and served as a measure of how efficiently \( dP/dt, SHE \), and \( PPI \) could be generated with each speed profile.

\[
\eta_{dP/dt} = \frac{dP/dt}{P_{Mean}} \tag{4.5}
\]

\[
\eta_{SHE} = \frac{SHE}{P_{Mean}} \tag{4.6}
\]
4.2.4 Pressure head-flow loop analysis

Besides the comparison of measures intended to express pulsatility in a single value, the dynamic characteristics of the pressure head as a function of the left pump flow (HQ-loops) were compared for their shape and size. Observations based on the haemodynamic characteristics of the recorded loops were then used to modify a lookup-table-generated (LUT) speed profile and corresponding HQ loop in an attempt to generate more physiologically representative haemodynamic waveforms and to reduce power consumption. Starting with a square wave profile, the LUT was iteratively adjusted while the changes in the HQ loop shape were observed in real time. The results for two empirically generated LUT shapes (LUT1, LUT2) were compared to the aforementioned profiles. Speed profile LUT1 was created by iteratively adjusting the rate of speed change and the height and duration of the systolic speed peak to reduce peaks in the instantaneous power consumption and, consequently, $P_{\text{Mean}}$. The profile shape in diastole was adjusted to avoid reverse flow and achieve a transition of the operating point along the y-axis – that is, at a flow rate of approximately zero. Adjustments of the relative systolic duration and maximum speed amplitude also affected $dP/dt$, $SHE$, and $PPI$. The profile LUT2 is a variant of LUT1 with a relatively shorter systolic peak and higher amplitude. It was attempted to increase the generated $dP/dt$ without substantially increasing the power requirement, as the acceleration at the onset of systole is reduced compared to the step speed change in a square wave function.

4.3 Results

Figure 4-3 shows the PP generated by all evaluated speed profiles as a function of the speed amplitude, which showed strong variation between the profiles. For example, a 20% square wave profile with a peak-to-peak amplitude of 500 rpm generated a PP of 25 mmHg, whereas a sine wave profile with the same amplitude generated a PP of 40 mmHg. Consequently, the profiles were compared with respect to equal generated PP and the associated power consumption, rather than with respect to the speed amplitude. The highest PP which was reached with all speed profiles was approximately 37 mmHg.
Figure 4-3 – Pulse pressures generated by the evaluated speed profiles as a function of the speed amplitude.

Figure 4-4 compares the HQ loops at $PP$s of 0, 10, 25, and 35 $mmHg$. The plotted dashed loops correspond to the native heart waveforms by Patel [216] (where the pressure head was approximated by $H \approx P_{ao} - 12 \ mmHg$). HQ-Loops corresponding to the sine wave profile (Figure 4-4A) exhibited an oval shape centred on the stationary operating point at a flow rate of 5 $L/min$ and an $MAP$ of approximately 90 $mmHg$. The square wave loops (Figure 4-4B, C) showed a small increase in pressure, followed by a high peak in the flow rate following the pulse onset (i.e. the point of lowest $H_L$, indicated by red $x$-markers in Figure 4-4). The operating point transition was approximately perpendicular to the steady state HQ-curves. During the constant speed phases (edges at lowest and highest abscissa values), the operating point shifted parallel to the static HQ characteristics, as the aortic pressure changed slowly compared to the step change in flow rate. The loop shape of the sawtooth function (Figure 4-4D) resembles the characteristics of the square profiles at the instance of the speed step, whereas it resembles the oval shape of the sine wave loops during the ramped speed decrease. Figure 4-4E and F show the HQ-loops for LUT1 and LUT2. Both profiles showed a diastolic flow rate transition close to the y-axis at 35 $mmHg \ PP$. In contrast, the loops of sine, square, and sawtooth profiles extended toward negative flow rates; that is, there was reverse flow through the pump at low speed. For all profiles, the enclosed loop area was approximately proportional to $SHE$. 

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The loops for the square 20% and LUT2 profiles showed some resemblance with the native heart loops (in particular the high maximum flow rate); however, the overall appearance of the pump-generated loops was substantially different from that of the native heart loop.

Figure 4-4 – HQ loops of the BiVACOR TAH for different profiles with speed amplitudes generating pulse pressures of 0, 10, 25, and 35 mmHg. Loops are cycled in an anti-clockwise direction; red x-markers indicate the onset of systole. Applied speed profiles were (A) sine wave, square wave at duty cycles of (B) 20% and (C) 50%, (D) sawtooth, and lookup-table generated profiles (E) LUT1 and (F) LUT2. Black lines are steady state HQ-curves at 1500 – 2700 rpm. Dashed loops correspond to the native heart waveforms by Patel [216] (cf. section 2.2.1).

Figure 4-5 shows the waveforms of motor speed ($n$), aortic pressure ($P_{Ao}$), systemic flow rate ($Q_S$), and motor power consumption ($P_{el}$) at a PP of 35 mmHg for each evaluated profile, as well as the aortic pressure and systemic flow waveforms of the native heart [216]. Instantaneous values of $P_{el}$ of 60 W or more (which is a factor of 10 higher than $P_{Mean}$ in constant speed operation) were observed during the speed step increase with the square and sawtooth profiles. Maximum $P_{el}$ was slightly lower for LUT2 (45 W) and was significantly
reduced for LUT1 (25.7 W). The comparison with the pressure and flow waveforms of the native heart showed substantial differences. Square 20% and LUT2 profiles showed the highest $dP/dt$ (284.7 mmHg/s and 304.27 mmHg/s respectively), while both values were substantially lower than that generated by the native heart (571.7 mmHg/s). However, the $PP$ of the native heart was also higher (52.33 mmHg). The pressure slope at the onset of systole was superimposed with some higher frequency ringing, which may be attributed to the characteristics of the MCL and pump hydraulics. With a short, high systolic peak, and a diastolic flow rate of approximately zero, the flow waveform for LUT2 most closely resembled the native heart waveform.

![Flow waveform comparison](image)

**Figure 4-5** – Pulsatile Haemodynamics: Aortic pressure, systemic flow, and instantaneous power consumption waveforms for one pulse cycle at a pulse pressure of 35 mmHg. Speed waveforms shown at the top of figure are schematic representations only. $n$, pump speed; $P_{ao}$, aortic pressure; $P_{el}$, motor power consumption; $Q_{s}$, systemic flow rate. Dashed lines in the panels for $P_{ao}$ and $Q_{s}$ show the native heart waveforms by Patel [216].
Figure 4-6 shows the trends of $P_{\text{Mean}}$, $dP/dt$, $SHE$, and $PPI$ as a function of $PP$. After these values were computed for each pulse cycle, a regression function was calculated and plotted for each dataset speed profile and quantity. Figure 4-6A shows an exponential increase of power consumption with pulse pressure. The sine wave profile was associated with the lowest $P_{\text{Mean}}$ over the whole $PP$ range, whereas the 20% square wave function was associated with the highest $P_{\text{Mean}}$: 12.53 $W$ at 40 $mmHg$ $PP$, which corresponds to an increase of more than 100% compared to that observed at constant speed. In contrast, the 20% square wave generated a substantially higher $dP/dt (> 300 \text{ mmHg/s})$, which approached the values generated by the native heart ($440 – 1180 \text{ mmHg/s}$ [19]). The $dP/dt$ generated by the 20% square wave was only exceeded by LUT2 for $PP > 30 \text{ mmHg}$ with a maximum of 390 $\text{ mmHg/s}$. Profile LUT1 showed a $dP/dt$ and $PPI$ in the midrange between the 20% square wave and sine wave profiles, while the power consumption stayed below 8.6 $W$ for $PP$ up to 40 $\text{ mmHg}$ (20% square wave: 12.53$W$; sine wave: 7.71$W$). The percentage increase of $P_{\text{Mean}}$ compared to the constant speed case varied between 15.4% (sine wave) and 57.8% (20% square wave) at 35 $\text{ mmHg}$ $PP$. These values substantially increased at a higher $PP$ (40 $\text{ mmHg}$), at the cost of between 23.2% (sine wave) and 100.4% (20% square wave) additional power consumed.

The trend of $dP/dt$ (Figure 4-6B) with increasing $PP$ was near-linear for all tested profiles except the two LUT profiles, which showed a stronger increase of $dP/dt$ towards higher $PP$; in contrast, $P_{\text{Mean}}$, $SHE$, and $PPI$ (Figure 4-6A, C, D) increased exponentially with $PP$. $SHE$ as a function of $PP$ was similar for all profiles over the whole range, while a maximum $SHE$ of 10,978 $ergs/cm^3$ (native heart $> 20,000 \text{ ergs/cm^3}$ [135]) was generated by a 50% square wave profile at a pulse pressure of 52 $\text{ mmHg}$. The $PPI$ reached values up to 12.5 $s^{-1}$ at 40 $\text{ mmHg}$ $PP$ and increased almost proportional to $P_{\text{Mean}}$ for all profiles.
Figure 4-7 shows the trends for the power specific pulsatility parameters $\eta_{dpdt}$, $\eta_{SHE}$, and $\eta_{PPI}$ as a function of $PP$. The curves for $\eta_{dpdt}$ show a near-linear increase before a maximum is reached, and the curves decrease at higher $PP$s because of the disproportional increase in power consumption. The 20% square wave generated the highest $\eta_{dpdt}$ values for $PP$ up to 25 mmHg. LUT2 shows superior results for $\eta_{dpdt}$ for $PP > 25$ mmHg with a maximum of 34.2 mmHg/s/W, indicating more efficient pump operation. The curves for $\eta_{SHE}$ and $\eta_{PPI}$ show a sigmoidal shape in the evaluated $PP$ range. The 20% square wave and LUT2 profiles exhibited the steepest increase of $\eta_{PPI}$ with $PP$; the highest values for $\eta_{SHE}$ over the whole
$PP$ range correspond to the sine wave. Table 2 shows a summary of the results for constant speed and all pulsatile profiles at $PP$s of 10, 25, and 35 mmHg.

Figure 4-7 – Power specific pulsatility parameters (A) $\eta_{dP/dt}$, (B) $\eta_{SHE}$, and (C) $\eta_{PPI}$ as a function of $PP$ for the evaluated speed profiles.
Table 4-2 – Haemodynamic data of all evaluated speed profiles at PP of 10, 25, and 35 mmHg. † P\textsubscript{Mean,CS} is the percentage increase in mean power consumption compared to the constant speed case. For each pulse pressure, the shaded values indicate the best values (highest pulsatility/lowest power consumption) respectively.

<table>
<thead>
<tr>
<th>PP (mmHg)</th>
<th>Profile</th>
<th>P\textsubscript{Mean}</th>
<th>P\textsubscript{Mean,CS} (W)</th>
<th>dP/dt (mmHg/s)</th>
<th>SHE (erg/cm³)</th>
<th>PPI</th>
<th>η\textsubscript{AP,dt}</th>
<th>η\textsubscript{SHE}</th>
<th>η\textsubscript{PPI} (1/s²/W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Const. speed</td>
<td>6.14</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Sine</td>
<td>6.32</td>
<td>2.93</td>
<td>96.57</td>
<td>475.7</td>
<td>0.41</td>
<td>15.28</td>
<td>75.2</td>
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<td>7.98</td>
<td>112.43</td>
<td>323.9</td>
<td>0.97</td>
<td>16.95</td>
<td>48.8</td>
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<tr>
<td></td>
<td>Square 50%</td>
<td>6.55</td>
<td>6.68</td>
<td>98.72</td>
<td>432.5</td>
<td>0.66</td>
<td>15.07</td>
<td>66.0</td>
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<tr>
<td></td>
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<td>6.51</td>
<td>108.59</td>
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<td>1.20</td>
<td>16.59</td>
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<td></td>
<td>LUT1</td>
<td>6.34</td>
<td>3.26</td>
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<td>14.62</td>
<td>60.8</td>
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<tr>
<td></td>
<td>LUT2</td>
<td>6.33</td>
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<td>13.59</td>
<td>50.9</td>
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<td>16.61</td>
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<td>20.45</td>
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<td>0.255</td>
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<tr>
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<td>20.03</td>
<td>171.84</td>
<td>1239.6</td>
<td>3.30</td>
<td>23.32</td>
<td>168.2</td>
<td>0.447</td>
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<tr>
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<td>26.21</td>
<td>214.5</td>
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<tr>
<td>35</td>
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<td>3770.7</td>
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<td>8.86</td>
<td>28.84</td>
<td>325.3</td>
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<td>8.07</td>
<td>33.90</td>
<td>355.4</td>
<td>0.899</td>
</tr>
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</table>
4.4 Discussion

The haemodynamic results showed significant differences in the trends of \(\frac{dP}{dt}, SHE, PPI,\) and \(P_{\text{Mean}}\). As might be expected, the highest \(\frac{dP}{dt}\) values were generated by profiles with comparatively short durations of systolic high-speed operation. Comparing the 20% and 50% square waves in Figure 4-4, it is apparent that the loops corresponding to the lower duty cycle are oriented toward higher flow rates, resulting in higher \(\frac{dP}{dt}\). This trend resulted from the greater speed amplitude required to generate the same PP with a lower duty cycle. Consequently, the corresponding \(PPi\) as a measure of the alternating flow component was also increased. A higher \(\frac{dP}{dt}\) was generally accompanied by greater motor power consumption, because acceleration to high speed levels during systole required substantial power to generate the required torque at the beginning of the speed step, resulting in a high peak in instantaneous power consumption. The large speed difference resulted in a high peak flow output and, subsequently, a faster increase in aortic pressure. This increased the hydraulic load torque and decreased motor efficiency because of the copper losses in the stator winding, which quadratically increase as a function of motor current. Furthermore, the time to accelerate the fluid to higher velocities was prolonged, requiring additional power. The trend of reducing \(\eta_{dpdt}\) with higher pulse pressure shown in Figure 4-7A indicate the rapid decrease in hydraulic and motor efficiency that occurs at high speed amplitudes.

The lowest relative difference between speed profiles was observed in the generated \(SHE\) levels, which had a maximum variation of 28% at 40 \(mmHg\) PP (sine compared to sawtooth profile). Profiles with relatively longer systolic durations tended to generate higher \(SHE\). In contrast \(PPI\) and \(\frac{dP}{dt}\) levels were higher for profiles with relatively shorter systolic durations. The results indicated a possible trade-off between haemodynamic energy levels in terms of \(SHE\) and flow variability as represented by \(PPI\) and \(\frac{dP}{dt}\). This trade-off appeared to be related to the relative systolic duration when a specific PP is considered.

Khalil et al. reported similar results in in-vitro measurements of \(SHE\) and PP for two LVADs in a TAH configuration [170]. Their investigations included square-wave profiles with duty cycles of 25%, 50%, and 75%. Measurements were taken at various beat rates and two different speed amplitudes. Their results showed that PP and \(SHE\) generally decreased with higher beat rates. This was related to the settling time of pressures and flows, which could
not attain their minimum and maximum values before the next speed pulse cycle started. Changes in the relative systolic duration showed a maximum SHE at 50% systolic duration, while it was similar for durations of 25% and 75%. The results for 25% and 50% systole were comparable to the findings presented, however, no profile with a systolic duration of > 50% was evaluated. It is important to note that the results in that study were compared at the same speed amplitude rather than the same PP. When results for both high and low speed amplitudes are considered, it can be observed that a 50% square wave profile with an amplitude of 2000 rpm generated comparable PP to the 25% square wave at an amplitude of 4000 rpm, while SHE was similar for both profiles. At a beat rate of 60 bpm SHE appeared to be increased for the 50% systolic duration, which is in accordance with the findings presented here.

Shiose et al. applied square wave, sine wave, and custom speed modulation waveforms to a single-device continuous flow TAH in vitro and evaluated PP and power consumption [12]. A custom waveform was optimised to generate a physiologic arterial pressure waveform shape. A relative increase in power consumption of 16.2% at a pulse pressure of 28 mmHg was reported, which is comparable to the findings of this study. The custom speed profile showed a decrease of both PP and power consumption at the higher evaluated speed amplitudes compared to sine and square waves. While other quantities were not evaluated, a variation of arterial compliance and SVR was performed. While changes in arterial compliance substantially affected PP, MAP appeared to remain constant. Changes in SVR affected both PP and MAP. The results for both experiments show the importance of a detailed characterization of arterial tree parameters, which have substantial influence on pulsatility.

To formulate an objective for favourable pulsatile haemodynamics in terms of waveform shape, amplitude and beat rate, a deeper understanding is required of the influence of haemodynamic characteristics related to the measures of pulsatility utilised here. For example, it is conceivable that increased SHE levels may result in better perfusion of the microvascular bed; moreover, greater variability of the systemic flow rate may lead to increased stimulation of the baroreflex and the vascular endothelium, thus improving the autoregulatory response. Furthermore, the speed waveform shape can be chosen to improve the motor power requirements. For example, the sawtooth function generated inferior $dP/dt$,
SHE, and PPI at lower values for $\eta_{dPdt}$, $\eta_{SHE}$, $\eta_{PPI}$ compared to a 20% square wave. In contrast, the LUT profiles showed the potential of shape variations to improve both power consumption and haemodynamic values to some extent. Numeric simulations may therefore prove beneficial in the process of optimizing the speed trajectory.

For further research, it should be noted that alterations in the hydraulic pump characteristics, pulse rate, and parameters of the arterial Windkessel, especially the arterial compliance, have significant influence on haemodynamics; thus, any potentially optimal waveform may depend on patient-specific parameters. The results also showed that the highest measured $dP/dt$ approached physiologic levels, while a flow waveform with similar appearance as the native heart waveform could be generated with the LUT2 profile. However, the pulsatility produced by all profiles generally stayed below the pulsatility generated by the native heart, which was mostly attributed to the limitations of the pump hydraulics, motor, and suspension capacity. Furthermore, the study was limited by the in vitro setup used. The measurement of pressures in the mock circulatory loop was limited by the use of fluid-filled pressure transducers, which mechanically filtered the signals (natural frequency $f_0 = 40 \text{ Hz}$) and may have limited the maximum measurable rate of pressure change. Furthermore, the simulated arterial Windkessel cannot fully reflect the characteristics of the human arterial system, which was evident in the comparison between the waveforms and HQ-loops generated by the TAH and the native heart. Considering the results of this study, a similar investigation should be repeated in vivo. Future studies should include the evaluation of performance with different hydraulic designs (such as axial/centrifugal flow pumps), which may provide further insight into design and control criteria favouring the efficient and effective re-creation of an artificial pulse with characteristics resembling those of the native pulse. A device fulfilling such criteria would greatly enhance the relevance of future in vivo studies evaluating pulsatile perfusion modes with regard to physiological benefits, biocompatibility, thrombogenicity, device washout, and power consumption.

While this study focused on the general ability of a RBP to generate near-physiologic pulsatile waveforms with different speed profiles, several other factors need to be considered for a clinical application of pulsatile RBP speed modulation. Most importantly the potential effects of added shear stress on blood damage and the coagulation cascade need to be evaluated. The power loss and thus heat generated in the pump should be limited to avoid
blood or tissue damage by contact to potential hot spots as well as improve battery-lifetime. Furthermore, it is most important to maintain balanced flow conditions between the systemic and pulmonary circulations. In the clinical setting, a limited fluid volume is available in the atrial remnants and inflow cuffs of the rotary TAH, which can result in a high risk of atrial collapse, especially at high pulse amplitudes or due to mismatched vascular resistances. Furthermore, a potentially increased risk of damage to the remodelled arterial tree due to excessively high flow rates and pressure gradients should be evaluated and weighed against the risk of vascular malformations due to perfusion with diminished pulsatility [137]. These factors warrant the requirement of a wide range of pre-clinical studies and careful application of pulsatile operation modes, where generating the maximum achievable pulsatile output may not be the favourable operation strategy, rather an optimum level of speed modulation may be found. Research on physiological control systems to apply pulse waveforms within reasonable patient-specific operational limits, may prove valuable in this context.

4.5 Conclusion
In this chapter, six different impeller speed modulation profiles and their associated waveforms as produced by the BiVACOR V2 were compared. These six waveforms were evaluated in terms of maximum $dP/dt$, $SHE$, $PPI$, and power consumption as a function of the generated $PP$. Compared at the same $PP$ the $SHE$ levels appeared to decrease for profiles with relatively shorter systolic durations, while flow variability (in terms of $PPI$ and resulting $dP/dt$) showed an opposite trend and a stronger dependency on the speed waveform. High flow variations were generally accompanied by a substantial increase in power consumption. The results manifested the necessity to precisely quantify pulsatile haemodynamics with more than one single measurement and highlighted the possibility to improve both haemodynamic characteristics and power consumption with the choice of an optimised speed waveform. Based on this finding, a numerical study to optimise the speed profile with respect to the trade-off between the generated maximum $dP/dt$ and $SHE$, and the motor power consumption was performed (see chapter 0).

The findings of this study further indicated the requirement for high flow and acceleration capacities of the pump’s motor and hydraulic design to generate physiologic pulsatility. Future research should aim to evaluate the beneficial effects of maximum $dP/dt$, $PPI$, and/or $SHE$ on baroreceptor functionality, microcirculatory perfusion, and blood compatibility in
terms of haemolysis and the risk of acquired von Willebrand disease with different speed profiles. Knowledge of the relations between each of these parameters and corresponding benefits to the cardiovascular system would allow the formulation of requirements for an optimised speed profile from a physiologic perspective.
5 Optimal Speed Control of Pulsatile Rotary Total Artificial Hearts

The previous chapters have shown that the task of recreating the native cardiovascular pulse with rotary blood pump (RBP) speed modulation is nontrivial. A solid understanding of device characteristics and parameters of the cardiovascular system (CVS) influencing haemodynamics is vital in developing a pump which is capable of generating physiologic waveforms in a pulsatile mode of operation. In vitro experiments in mock circulatory loops (MCL) offer a valuable platform to evaluate device performance, however, it is typically difficult to precisely replicate the flow and pressure response of the native arterial tree with respect to measures such as SHE and \( \frac{dP}{dt} \). Furthermore, the evaluation of multiple pump prototypes is time-consuming and the manufacturing can be cost-intensive.

Alternative, commonly used approaches to evaluate pump operation modes are numerical simulations. Numerous computer models of RBP and the CVS have been developed and published over the recent decades. The complexity of applied models varies from a simple representation of the arterial tree with a two-element Windkessel model [231] to detailed computational fluid dynamics (CFD) models adapted from geometrical data obtained using medical imaging techniques such as magnetic resonance imaging (MRI) or computed tomography (CT) [232]. While models of higher complexity have, for example, been applied to perform a more detailed analysis of flow patterns with respect to wall shear stress or stagnant flow areas [233–235], various lumped parameter CVS models have been shown to predict haemodynamic waveforms with sufficient fidelity.

Computer simulations of the cardiovascular system further allow the application of numerical techniques to optimize the control of RBP for a specified objective, as is outlined in the following chapter.

5.1 Aim

In order to enhance the relevance of future clinical studies evaluating the biological effects of the presence or absence of a true physiologic pulse, development of a device design and control strategy, which allow close replication of physiologic haemodynamics, is vital. Therefore, the aim of this chapter is to develop a numerical framework to investigate and optimize speed control strategies for pulsatile operation of different rotary blood pump (RBP) types in a total artificial heart (TAH) setting. For this purpose, a methodology was developed
to obtain speed profiles, which are individually numerically optimized for maximum
haemodynamic pulsatility in different operating conditions. In the following, a generalizable
nonlinear state space model of the dynamic interaction between rotary blood pumps and the
CVS in absence of the native ventricles is derived. Subsequently, the model equations are
included in the formulation of a constrained optimal control problem (OCP), with the aim to
minimize an objective function based on previously discussed measures for haemodynamic
pulsatility, subject to constraints such as an average flow output and maximum motor power
consumption limit. The numerical methods used to discretise the OCP and solve the
corresponding nonlinear program (NLP) are presented, and NLP solutions for various
scenarios are applied to evaluate different RBP speed modulation modes with respect to their
ability to generate physiologic haemodynamics, using figures of merit such as the generated
maximum arterial $dP/dt$ and surplus haemodynamic energy ($SHE$). In summary, the
objectives to achieve the aim of this chapter are as follows:

- Derive a mathematical model for an accurate representation of the dynamic
interaction between the cardiovascular system and a rotary TAH.
- Develop a nonlinear program (NLP) to find optimized speed profiles for a specified
objective function and within certain boundary conditions, and evaluate the optimized
solutions with respect to their suitability for implementation in a control strategy for
pulsatile rotary TAH.
- Validate and evaluate the numerical framework in vitro and in vivo.

5.2 Methods – Numerical Modelling and Optimization

The following section outlines the development of a numerical model environment. First, the
dynamic equations describing numerical models of the pulmonary and systemic circulatory
systems and a generic rotary blood pump are introduced. A numerical framework for
optimizing speed trajectories based on these models is developed. Initially, to reduce the
problem size and complexity, only a single RBP connected to the systemic circulation is
considered. This ‘reduced model’ is validated here, to be further used in the subsequent
chapter (chapter 6).

Lastly, the models of the systemic and pulmonary circulation are combined to a dual
circulation model, wherein two RBP models form a rotary TAH to represent either dual pump
or single device TAHs, such as the BiVACOR TAH. The dual circulation model is validated with the BiVACOR device and evaluated in a preliminary in vivo study.

5.2.1 Generalizable Dynamic Model of the TAH-CVS Interaction

5.2.1.1 Cardiovascular System

The cardiovascular system model implemented here was derived from combined aspects of different lumped-parameter models previously discussed in literature [83,236]. An identical structure comprising an arterial impedance model, venous circulation, and atria was used for the systemic and pulmonary circulations (Figure 5-1).

![Figure 5-1 – Electrical circuit analogue of the vascular model used for both the systemic and pulmonary circulation. Symbols in parentheses correspond to the pulmonary circulation.](image)

However, for the specific application to study the interaction with a TAH, the model complexity was significantly reduced. The time-varying compliance or elastance elements typically used to simulate ventricular and atrial contraction, as well as the one-way heart valves were omitted (as they play no role in a TAH application).

Lim et al. [83] further modelled the venous circulation by representing the systemic venous circulation and the vena cava with two separate lumped elastance components, which were separated by an additional systemic venous resistance. In the model described here, similar
to the approach described in [236], the venous system was modelled with a single compliance element.

Table 5-1 – Model parameters and symbols for the definition of systemic and pulmonary circulation. The variable type in the bottom part of the table distinguishes state variables (SV) and dependent variables (DV), and defines the model inputs and outputs.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
<th>Source</th>
<th>Init. Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_{cs}$</td>
<td>Systemic characteristic inertance</td>
<td><em>In vivo est.</em></td>
<td>0.027</td>
<td>mmHg · s²/mL</td>
</tr>
<tr>
<td>$R_{cs}$</td>
<td>Systemic characteristic resistance</td>
<td><em>In vivo est.</em></td>
<td>0.034</td>
<td>mmHg · s/mL</td>
</tr>
<tr>
<td>$R_{sa}$</td>
<td>Systemic arterial resistance</td>
<td>[96] (adapted)</td>
<td>1.068</td>
<td>mmHg · s/mL</td>
</tr>
<tr>
<td>$R_{ra}$</td>
<td>Right atrial resistance</td>
<td>[83]</td>
<td>0.012</td>
<td>mmHg · s/mL</td>
</tr>
<tr>
<td>$C_{ao}$</td>
<td>Aortic compliance</td>
<td><em>In vivo est.</em></td>
<td>1.51</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>$C_{sv}$</td>
<td>Systemic venous compliance</td>
<td>[236]</td>
<td>85.0</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>$C_{ra}$</td>
<td>Right atrial compliance</td>
<td>[96]</td>
<td>2.0</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>$L_{cp}$</td>
<td>Pulmonary characteristic inertance</td>
<td>[237]</td>
<td>0.0059</td>
<td>mmHg · s²/mL</td>
</tr>
<tr>
<td>$R_{cp}$</td>
<td>Pulmonary characteristic resistance</td>
<td>[237]</td>
<td>0.086</td>
<td>mmHg · s/mL</td>
</tr>
<tr>
<td>$R_{pa}$</td>
<td>Pulmonary arterial resistance</td>
<td>[96] (adapted)</td>
<td>0.055</td>
<td>mmHg · s/mL</td>
</tr>
<tr>
<td>$R_{la}$</td>
<td>Left atrial resistance</td>
<td><em>In vivo est.</em></td>
<td>0.0575</td>
<td>mmHg · s/mL</td>
</tr>
<tr>
<td>$C_{pa}$</td>
<td>Pulmonary arterial compliance</td>
<td>[237]</td>
<td>2.35</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>$C_{pv}$</td>
<td>Pulmonary venous compliance</td>
<td>[236]</td>
<td>10</td>
<td>mL/mmHg</td>
</tr>
<tr>
<td>$C_{la}$</td>
<td>Left atrial compliance</td>
<td><em>In vivo est.</em></td>
<td>1.577</td>
<td>mL/mmHg</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Variable</th>
<th>Description</th>
<th>Type</th>
<th>Init. Cond.</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_s$</td>
<td>Systemic flow rate</td>
<td>Input</td>
<td>0</td>
<td>L/min</td>
</tr>
<tr>
<td>$Q_{ics}$</td>
<td>Slack variable (aortic inertance flow)</td>
<td>SV</td>
<td>0</td>
<td>L/min</td>
</tr>
<tr>
<td>$Q_{ra}$</td>
<td>Right atrial flow rate</td>
<td>Input</td>
<td>0</td>
<td>L/min</td>
</tr>
<tr>
<td>$P_{ao}$</td>
<td>Aortic pressure</td>
<td>DV / Output</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{cao}$</td>
<td>Slack variable (aortic compliance pr.)</td>
<td>SV</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{sv}$</td>
<td>Central systemic venous pressure</td>
<td>SV</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{ra}$</td>
<td>Right atrial pressure</td>
<td>SV / Output</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$Q_p$</td>
<td>Pulmonary flow rate</td>
<td>Input</td>
<td>0</td>
<td>L/min</td>
</tr>
<tr>
<td>$Q_{icp}$</td>
<td>Slack variable (pulmonary art. inert. flow)</td>
<td>SV</td>
<td>0</td>
<td>L/min</td>
</tr>
<tr>
<td>$Q_{ia}$</td>
<td>Left atrial flow rate</td>
<td>Input</td>
<td>0</td>
<td>L/min</td>
</tr>
<tr>
<td>$P_{pa}$</td>
<td>Pulmonary arterial pressure</td>
<td>DV / Output</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{cpa}$</td>
<td>Slack var. (pulmonary art. compl. pr.)</td>
<td>SV</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{pv}$</td>
<td>Pulmonary venous pressure</td>
<td>SV</td>
<td>12</td>
<td>mmHg</td>
</tr>
<tr>
<td>$P_{la}$</td>
<td>Left atrial pressure</td>
<td>SV / Output</td>
<td>12</td>
<td>mmHg</td>
</tr>
</tbody>
</table>

When the component values for compliance and resistance elements are adjusted accordingly, the difference in the impedance of the venous system is negligible, thus does not significantly affect atrial and arterial pressure and flow waveforms. Further, the described model aims to evaluate the performance of a pump interacting with the circulation, while the exact pressure distribution within the circulation is only of minor interest. Therefore, this reduced variant was chosen to reduce model complexity and computation time for the subsequently described speed profile optimization algorithm (see section 5.2.3). The
pulmonary and systemic arterial systems were each represented by a four-element Windkessel model, with a parallel structure of characteristic arterial inertance and resistance [84]. The venous system and atria were accordingly modelled each with a compliance component, separated by an atrial inflow resistance to model venous return. A summary of parameters and symbols representing the systemic and pulmonary circulation in the initial model is given in Table 5-1. Parameter values were obtained from literature and estimated from previous in vivo aortic flow rate and pressure measurements with standard techniques [93].

The mathematical description of the systemic and pulmonary circulation was formulated as state-space model. Each of the energy storages, i.e. compliance (C) and inertance (L) components, represents an inner state of the system and is described by a differential equation defining the relationship between pressure difference (ΔP) between the component terminals and the flow rate (Q) (equations (5.1) and (5.2)).

\[
\dot{Q}_L = \frac{1}{L} \cdot \Delta P_L \quad (5.1)
\]

\[
\Delta P_c = \frac{1}{C} \cdot Q_c \quad (5.2)
\]

The system inputs were defined as the arterial and atrial flow rates entering and exiting the circulation respectively. Correspondingly, the system outputs are the resulting arterial and atrial pressures. Each circulation is then fully defined by a set of initial conditions (see Table 5-1), four differential equations defining each state and its relation to the other states, and an algebraic equation describing the arterial pressure as a function of the state variables respectively. The set of equations is summarized in the example of the systemic circulation in (5.3) – (5.7). Analogous, the equations for the pulmonary circulation are obtained by substituting each symbol with its equivalent as indicated by symbols in parentheses in Figure 5-1.

\[
P_{ao} = P_{cao} + R_{cs} \cdot (Q_S - Q_{lcs}) \quad (5.3)
\]

\[
\dot{Q}_{lcs} = \frac{R_{cs}}{L_{cs}} \cdot (Q_S - Q_{lcs}) \quad (5.4)
\]

\[
\dot{P}_{cao} = \frac{1}{C_{ao}} \cdot \left( Q_S - \frac{1}{R_{sa}} \cdot (P_{cao} - P_{sv}) \right) \quad (5.5)
\]
The model is handled as a state-space system, where the eight state equations are given by equations (5.4) – (5.7) and their equivalents for the pulmonary circulation. The inputs of the subsystem are defined as the atrial and arterial flow rates \( Q_{la}, Q_s, Q_{ra}, \) and \( Q_p \), while the output vector comprises of the corresponding pressures \( P_{la}, P_{ao}, P_{ra}, \) and \( P_{pa} \). The full state space equations and matrices are given in Appendix A.

5.2.1.2 Rotary Blood Pump Dynamics

For the simulation of the interaction of a TAH with the CVS, a generalizable model of a rotary blood pump was implemented. The model comprises the hydraulic pump characteristics, the mechanical motor load, as well as a simplified motor model to evaluate the drive power consumption. All equations describing the RBP were implemented based on a minimal set of parameters to allow straightforward replacement of the modelled pump and compare hydraulic and motor characteristics accordingly. A schematic representation of the RBP model is shown in Figure 5-2; a summary of the corresponding symbol definitions is given in Table 5-2.
Table 5-2 – Symbol definitions for the implemented RBP model.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Type</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( n )</td>
<td>Impeller speed</td>
<td>State variable</td>
<td>( \text{rpm} )</td>
</tr>
<tr>
<td>( Q )</td>
<td>Pump flow rate</td>
<td>State variable / Output</td>
<td>( \text{L/min} )</td>
</tr>
<tr>
<td>( J_{\text{rot}} )</td>
<td>Rotor inertia</td>
<td>Parameter</td>
<td>( \text{kg} \cdot \text{m}^2 )</td>
</tr>
<tr>
<td>( K_t )</td>
<td>Motor torque constant</td>
<td>Parameter</td>
<td>( \text{Nm/A} )</td>
</tr>
<tr>
<td>( L )</td>
<td>Fluid inertia</td>
<td>Parameter</td>
<td>( \text{mmHg} \cdot \text{s}^2/\text{mL} )</td>
</tr>
<tr>
<td>( R_{\text{mot}} )</td>
<td>Motor phase resistance</td>
<td>Parameter</td>
<td>( \Omega )</td>
</tr>
<tr>
<td>( I_{\text{mot}} )</td>
<td>Motor current</td>
<td>Input</td>
<td>( \text{A} )</td>
</tr>
<tr>
<td>( P_{\text{in}} )</td>
<td>Inlet pressure</td>
<td>Input</td>
<td>( \text{mmHg} )</td>
</tr>
<tr>
<td>( P_{\text{out}} )</td>
<td>Outlet pressure</td>
<td>Input</td>
<td>( \text{mmHg} )</td>
</tr>
<tr>
<td>( H(n, Q) )</td>
<td>Steady-state pressure head</td>
<td>Polynomial function</td>
<td>( \text{mmHg} )</td>
</tr>
<tr>
<td>( T(n, Q) )</td>
<td>Steady-state hydraulic load torque</td>
<td>Polynomial function</td>
<td>( \text{N} \cdot \text{m} )</td>
</tr>
</tbody>
</table>

The RBP is described by two differential state equations representing the flow rate through the pump and the rotational speed of the impeller (equations (5.8) and (5.9)). A third, algebraic equation describes the electrical motor power consumption \( (P_{\text{el}}) \) as a function of the motor current, based on a simplified motor model which neglects the frequency-dependent iron and eddy-current losses (equation (5.10)). The RBP motor was modelled as a vector-controlled three-phase brushless permanent magnet motor, where the motor current \( I_{\text{mot}} \) is the RMS-value of the three-phase sinusoidal currents.

\[
Q = \frac{1}{L} (H(n, Q) - (P_{\text{out}} - P_{\text{in}})) \quad (5.8)
\]

\[
\dot{n} = \frac{60}{2\pi \cdot J_{\text{rot}}} (I_{\text{mot}} \cdot K_t - T(n, Q)) \quad (5.9)
\]

\[
P_{\text{el}} = P_{\text{cu}} + P_{\text{mech}} = 3 \cdot (I_{\text{mot}})^2 \cdot R_{\text{mot}} + 2\pi K_t \cdot I_{\text{mot}} \cdot \frac{n}{60} \quad (5.10)
\]

The load-dependent dynamic motor speed equation is a function of the rotor inertia, motor torque constant, and the applied motor current. The pressures at the pump inlet and outlet, as well as the motor current are treated as model inputs. The flow rate through the pump is dynamically calculated and returned as the model output. Here, the inerance component represents the fluid inertia in the pump cavity and inlet and outlet cannulae based on dynamic measurements, while the nonlinear functions \( H(n, Q) \) and \( T(n, Q) \) return the steady state values for the pressure head and hydraulic load torque at a certain impeller speed and pump flow rate. To reduce the required model parameters, both functions were implemented based on the similarity laws for rotary pumps, which allow prediction of the hydraulic performance.
of geometrically similar pumps with a different size and speed from a known operating point. Under the assumption, that the hydraulic efficiency remains unchanged for the predicted operating point, the following proportionalities to the impeller speed and impeller diameter \( D \) can be derived from non-dimensional analysis [102,222,238]:

\[
Q \propto nD^3 \quad (5.11)
\]

\[
H \propto n^2D^2 \quad (5.12)
\]

\[
T \propto n^2D^5 \quad (5.13)
\]

Based on the proportionality to speed, the pressure head-flow (HQ) characteristic and torque-flow characteristic at a single speed were obtained, normalized, and implemented as a polynomial of arbitrary order. Consequently, the speed and flow rate dependent steady-state pressure head \( H(n, Q) \) is represented by the \( l \)-th order polynomial with coefficients \( k_{h,0} \ldots k_{h,l} \) and calculated per equation (5.14). Analogously, the hydraulic torque is represented by \( m \)-th order polynomial with coefficients \( k_{t,0} \ldots k_{t,m} \) and calculated according to equation (5.15).

\[
H(n, Q) = n^2 \sum_{i=0}^{l} k_{h,i} \cdot \left(\frac{Q}{n}\right)^i \quad (5.14)
\]

\[
T(n, Q) = n^2 \sum_{j=0}^{m} k_{t,j} \cdot \left(\frac{Q}{n}\right)^j \quad (5.15)
\]

### 5.2.2 Interaction of the Cardiovascular System and the Total Artificial Heart

#### 5.2.2.1 Reduced Model of the Systemic Circulation

The implementation of the cardiovascular system and RBP models allows their straightforward combination. While the CVS model describes the change in atrial and arterial pressures to the flow rates entering and exiting the circulation, the RBP model calculates the pump flow rate as a function of the inlet and outlet pressures, and the motor current.

As will be presented in chapter 6, the numerical framework developed here is further applied to evaluate the influence of different pump design criteria on device performance in pulsatile operation. To simplify this analysis, it was performed on the example of a single RBP. Here, the influence of the interaction between pulmonary and systemic circulation is of minor
interest, therefore the simulation of only one circulatory loop is sufficient. Therefore, a reduced model of the systemic circulation was implemented. The reduced model was based on the assumption, that pulsatility in the systemic and pulmonary venous systems is small and the corresponding pulmonary and systemic venous pressures are similar. Thus, the influence of pulsatility in venous pressure during the pulse cycle on the pump performance is negligible. Consequently, only the left pump and the surrounding vascular model elements were in included in the reduced model. The corresponding electrical circuit analogue is shown in Figure 5-3.

The reduced model comprises the left RBP, left atrium, aortic Windkessel, and systemic venous compliance. The corresponding elements of the right side were omitted and replaced by a direct connection between the systemic venous circulation, and the left atrium. In the resulting system, the outputs of the RBP model are used as inputs to the CVS model, and vice versa. Consequently, the models are interdependent, while the only input quantity remains the motor current of the pump. The use of the reduced model is advantageous for the given application, as it reduces complexity and computational time. Furthermore, as the model was validated in a mock circulatory loop, it allowed the use of a smaller hydraulic loop with reduced complexity.

5.2.2.2 Dual Circulation Model with the BiVACOR TAH

When two separate RBPs are used as a TAH, the pumps can simply be implemented with two instances of the RBP model, each of which is operated independently. In this configuration, continuity dictates that to each pump applies that the inlet flow rate equals the outlet flow rate. That is, the right atrial flow rate equals the pulmonary flow rate, and the left
atrial flow rate equals the systemic flow rate respectively. However, some specific single-device TAH models, such as the SmartHeart® TAH and the BiVACOR® TAH, combine both left and right impellers on a single rotor, thus only a single equation for the speed of the common rotor is required. Additionally, due to the specific hydraulic designs, both devices exhibit an internal flow path through the device, which allows fluid leakage (shunt flow) between the left and right pump cavities. As will be discussed in more detail in section 5.3.2, the shunt flow ($Q_L$) through the BiVACOR device was modelled with a multivariate regression model obtained from dynamic pump data (equation (5.16)).

$$Q_L = c_{QL1} \cdot Q_S + c_{QL2} \cdot Q_P + c_{QL3} \cdot n_L + c_{QL4} \cdot (P_{ao} - P_{pa}) + c_{QL5} \cdot (P_{la} - P_{ra}).$$  \hspace{1cm} (5.16)

For simplification of the model handling in the implementation of the NLP, it was assumed, that the leakage occurs exclusively between the left and right atria, and therefore the equations for the pulmonary and systemic flow rates remain unaffected. The left and right atrial flow rates are calculated according to equations (5.17) and (5.18). In the work presented here, the dual circulation TAH model was exclusively used with the model of the BiVACOR TAH, while a comparison of single-impeller pump characteristics was performed in a reduced model (see chapter 6). However, although outside the scope of this thesis, the model implementation can equivalently be used for the dual RBP TAH case, where the coefficients for calculation of the shunt flow are set to zero.

$$Q_{ra} = Q_P - Q_L \hspace{1cm} (5.17)$$

$$Q_{la} = Q_S + Q_L. \hspace{1cm} (5.18)$$

The combined model comprised two separate RBP models for the left and right pump, the pulmonary circulation, the systemic circulation, and the shunt flow path is shown in Figure 5-4.
Numerical Speed Profile Optimization

The previous section describes the development of mathematical models to simulate the interaction of a RBP with the systemic circulation and a rotary total artificial heart with the systemic and pulmonary circulations. The state-space definition of the derived models allows the straightforward application of numerical optimal control methods to obtain a speed profile, which optimizes the trade-off between motor power consumption and haemodynamic pulsatility generated by the RBP under specified boundary conditions. In the following, the development of a nonlinear program (NLP), which discretises the state-space model and solves the underlying optimal control problem (OCP), is described. The solutions obtained from the NLP represent the optimal speed control regime with respect to a specified set of model parameters, boundary conditions, and the specified objective function (section 5.2.3.3). This potentially allows derivation of an energy efficient speed modulation control strategy for rotary TAHs to allow a meaningful evaluation of the physiologic effects of pulsatility in future research studies. The work described in the following was inspired by a previous study [165], in which rotary ventricular assist device (VAD) speed profiles were numerically optimized with respect to the trade-off between ventricular unloading and the volumetric flow rate through the aortic valve. A similar methodology was applied here to evaluate the trade-off between $SHE$ and $dP/dt$ generated by a rotary TAH.

The OCP and NLP presented here were formulated on the example of the dual circulation model, including the systemic and pulmonary circulations, and a rotary TAH with a shunt.
flow path. While the reduced model of the systemic circulation (section 5.2.2.1) requires fewer equations, it can be formulated analogously from the electrical circuit analogue.

5.2.3.1 Formulation of the Optimal Control Problem

The objective to find a speed trajectory optimizing the pulsatile output of the TAH can be formulated as an optimal control problem (OCP) for the time interval $[t_I, t_F]$ of length $T$. Formally, the OCP defined here is to determine the trajectory of the control vector $u(t)$ and its corresponding state vector $x(t)$ that optimize the arbitrary scalar performance index (the objective function) $J(x(t), t)$, subject to the dynamic system state equations, upper and lower bounds on the state variables, initial and terminal conditions, and nonlinear path constraints. The state vector for the dual circulation model comprises of the $n_x = 12$ TAH and CVS model state variables as defined in Table 5-1 and Table 5-2 (equation (5.19))\(^3\).

$$x(t) = [x_1(t), ..., x_{12}(t)]^T = [n_L, Q_S, Q_{ics}, P_{cav}, P_{sv}, P_{ra}, n_R, Q_P, Q_{tcp}, P_{cpa}, P_{psv}, P_{ta}]^T$$

(5.19)

The control vector $u(t)$ comprises of the $n_u = 2$ control inputs, which in the generalized model are the pump motor currents of the left and right pump (equation (5.20)).

$$u(t) = [u_1(t), u_2(t)]^T = [I_{mot,L}, I_{mot,R}]^T$$

(5.20)

The BiVACOR TAH, in which left and right impellers are coupled on a common, single motor driven rotor, was modelled by equalizing the right pump speed with the left pump speed ($x_{7,BVC}(t) = x_{1,BVC}(t)$), and setting the right pump motor current input to zero ($u_{2,BVC}(t) = 0$). The hydraulic torque was approximated by a single polynomial as a function of the systemic flow rate and the impeller speed. The equations defining the OCP for the generic model are given by equations (5.21) – (5.30).

---

\(^3\) Scalar quantities are denoted by lowercase letters ($x_i$); vector quantities are denoted by **bold** lowercase letters ($x$)
The objective function (equation (5.21)) was motivated by the haemodynamic finding presented in chapter 4, that a trade-off may exist for RBP speed profiles with respect to maximizing the surplus haemodynamic energy ($SHE$) and maximum rate of change of aortic pressure ($dP/dt$). A detailed description of the constructed objective function and its NLP implementation is given in section 5.2.3.3.

Equation (5.22) implements the dynamic model equations, given by the differential equations describing the TAH and CVS. The path constraint equations (5.23) and (5.24) enforce the specified mean systemic flow rate $\bar{Q}_s^*$ and the mean motor power consumption limit $\bar{P}_{el}^*$ of the left pump – or a single-device TAH. The corresponding functions $g(x(t))$ and $h(x(t),u(t))$ are defined in equations (5.31) and (5.32).

$$g(x(t)) = \left( \frac{1}{t_F - t_I} \int_{t_I}^{t_F} x_2(t) dt - \bar{Q}_s^* \right)$$

(5.31)

$$h(x(t),u(t)) = \left( \frac{1}{t_F - t_I} \int_{t_I}^{t_F} \left[ 3 \cdot R_{\text{mot}} + \frac{2\pi K_t}{60} \cdot u_1(t)x_1(t) \right] dt - \bar{P}_{el}^* \right)$$

(5.32)

Equation (5.25) enforces the mean circulatory pressure ($P_{mc}$) as an initial condition at the time $t_I$ based on the fluid volumes corresponding to the compliance elements in the circulatory system model. The pressure in each element changes in response to the volumetric
flow rate into or out of the element, thus the fluid volume can be calculated as the product of the compliance value and pressure. The total circulatory volume is assumed to remain constant during a single pulse cycle, as does the sum of the volumes in all compliance elements, thus $P_{mc}$ is enforced by imposing the volume condition to the pressure states of the compliance elements (equation (5.33)).

$$
\psi(x(t)) = C_{ao} \cdot x_4(t) + C_{sv} \cdot x_5(t) + C_{ra} \cdot x_6(t) \\
+ C_{pa} \cdot x_{10}(t) + C_{pv} \cdot x_{11}(t) + C_{la} \cdot x_{12}(t) \\
-(C_{ao} + C_{sv} + C_{ra} + C_{pa} + C_{pv} + C_{la}) \cdot P_{mc}
$$

(5.33)

The remaining OCP equations equate the state and control vectors at the start and end of the simulated time interval to generate periodic waveforms (equations (5.26) and (5.27)) and impose the upper and lower bounds on the state, control, and time variables (equations (5.28) – (5.30)).

### 5.2.3.2 Nonlinear Programming

To solve the OCP, the introduced equations were discretised and converted to a nonlinear program (NLP) using the direct transcription method (equations (5.36) – (5.44))\(^4\). The time interval $[t_I, t_F]$ was discretised into $M$ time segments of length $h_k = t_{k+1} - t_k$ between $N = M + 1$ collocation nodes. The time dependent state vector $x(t)$ was transformed to the collocated NLP variable vector $\bar{x}$, comprising the state and control variables at all $N$ collocation nodes with a total of $n_v = N \cdot (n_x + n_u)$ variables (equation (5.34)).

$$
\bar{x} = \begin{bmatrix} x_0^T, x_1^T, ..., x_{N-1}^T, u_0^T, u_1^T, ..., u_{N-1}^T \end{bmatrix}^T
$$

(5.34)

An implicit Runge-Kutta method of the first order – the backwards-Euler method – was applied to solve the state equations according to equation (5.22), as it has been previously shown to yield preferable results in the context of simulations of the CVS-RBP interaction [165]. In the NLP, the system dynamics transformed to a set of $n_x \cdot (N - 1)$ nonlinear defect constraints of the form

\(^4\) The following notation was adopted:
- $x(t_k) \equiv x_k$
- $x_i(t_k) \equiv x_{i,k}$
\[ \zeta_{i,k} = x_{i,k} - (x_{i,k-1} + h_{k-1} \cdot f_i(x_k, u_k)) = 0, \quad \begin{cases} i = 1, \ldots, 12 \\ k = 1, \ldots, N - 1 \end{cases} \quad (5.35) \]

which serve to enforce satisfaction of the \( i \)-th state equation at the \( k \)-th collocation node respectively (equation (5.37)). All equations are discretised and the time integrals were approximated corresponding to the backwards-Euler integration scheme.

\[
\min_{\bar{x}} \quad J_{NLP}(\bar{x}) \quad (5.36)
\]

\[
\text{s.t.} \quad \zeta_k = \left[ [\zeta_{1,1}, \ldots, \zeta_{12,1}], \ldots, [\zeta_{1,N-1}, \ldots, \zeta_{12,N-1}] \right]^T = 0 \quad (5.37)
\]

\[ g_{NLP}(\bar{x}) = 0 \quad (5.38) \]

\[ h_{NLP}(\bar{x}) \leq 0 \quad (5.39) \]

\[ \psi_{NLP}(x_0) = 0 \quad (5.40) \]

\[ x_0 - x_{N-1} = 0 \quad (5.41) \]

\[ u_0 - u_{N-1} = 0 \quad (5.42) \]

\[ \bar{x}_i \leq \bar{x}_k \leq \bar{x}_u \quad (5.43) \]

\[ t_i = t_0 < t_1 < t_2 < \ldots < t_{N-1} = t_F. \quad (5.44) \]

The discretised path constraint functions \( g_{NLP}(\bar{x}) \) and \( h_{NLP}(\bar{x}) \) are given in equations (5.45) and (5.46), the discrete initial condition \( \psi_{NLP}(x_0) \) is equivalent to the continuous-time case.

\[
g_{NLP}(\bar{x}) = \frac{1}{t_{N-1} - t_0} \cdot \left( \sum_{k=1}^{N-1} x_{2,k} \cdot h_{k-1} \right) - \bar{Q}_s^* \quad (5.45)
\]

\[
h_{NLP}(\bar{x}) = \frac{1}{t_{N-1} - t_0} \cdot \left( \sum_{k=1}^{N-1} \left( 3 \cdot u_{1,k}^2 \cdot R_{\text{mot}} + \frac{2\pi K_t}{60} \cdot u_{1,k} x_{1,k} \right) \cdot h_{k-1} \right) - \bar{P}_e^* \quad (5.46)
\]

The NLP is solved by minimizing the discretised objective function \( J_{NLP}(\bar{x}) \) with Newton’s method and satisfying the \( n_c = ((N - 1) + 1) \cdot n_x + n_u + 3 \) dynamic constraints defined by (5.37) – (5.42). This requires calculation of the derivatives of the constraint and objective functions with respect to the NLP variables, thus all implemented functions are required to be smooth. The entire time trajectory of the state variables is hereby calculated simultaneously, while the dynamic state equations (5.37) are typically only satisfied after the final iteration. Although the NLP obtained through direct transcription results in a large number of variables, the problem structure is sparse. While the objective function and the nonlinear constraint functions \( g_{NLP}(\bar{x}) \) and \( h_{NLP}(\bar{x}) \) are calculated as a function of the entire time trajectory, the remaining constraint equations only depend on the variables corresponding to one node in the case of the constraints enforcing periodicity (equations
(5.41) – (5.42)), or two adjacent nodes for the defect constraints (equation (5.37)). Consequently, the majority of first derivatives equate to zero.

The NLP was solved with SNOPT (Stanford Business Software Inc., Palo Alto, CA, USA), a software package based on a sequential quadratic programming algorithm specialized for solving large-scale sparse nonlinear optimization problems. To accelerate the program execution, the solver was provided with a combined vector of the objective function and all dynamic constraint functions, together with the corresponding Jacobian matrix, which contains the partial first derivatives with respect to the vector of NLP variables \( \bar{x} \). Further, a rudimental initial guess assuming constant values for the state and control variables in the expected order of magnitude was passed to the solver. The solution was iteratively refined by sequentially solving the NLP for \( N/4 \), \( N/2 \), and \( N \) collocation nodes, and updating the initial guess with a linear interpolation of the solution of the previous iteration respectively. The number of collocation nodes \( N \) was determined by the length of the simulated time interval and the chosen time step size (see 5.3.1.1). Before passing the NLP to SNOPT the state and control variables were scaled with respect to defined baseline values for pressures, flows, motor speeds, and currents, thus all state variables handled by the NLP solver were of the same order of magnitude.

5.2.3.3 Objective Function

The implemented objective function was used to evaluate the existence and nature of a trade-off between the \( SHE \) and \( dP/dt \) in the systemic circulation, as previously suggested in chapter 4. It was shown, that the shape of the speed profile affects these two quantities in different ways; hence it is a valid assumption that the results of optimizations for the two objectives yields different results. Therefore, the objective function of the OCP was implemented as the sum of two function components, whereas the relative influence of each component was adjustable with a weighting factor \( w_{obj} \in [0,1] \) (equation (5.47)).

\[
J(x(t)) = -(J_1(x(t)) \cdot (1 - w_{obj}) + J_2(x(t)) \cdot w_{obj})
\]

(5.47)

The function is negated, as the objective function is minimized during the optimization process, while the objective is to maximize \( SHE \) and \( dP/dt \) in the aorta respectively. The two quantities were represented by the function components \( J_1(x(t), t) \) and \( J_2(x(t), t) \), which were defined as follow:
where:

\[
P(x(t)) = \left( x_4(t) + R_{cs}(x_2(t) - x_3(t)) \right).
\]

The transcription and discretization of equation (5.48) is straightforward, as the integral functions can be replaced by the corresponding approximation as per the backward-Euler integration scheme. However, equation (5.49) cannot easily be transcribed as the maximum function is not differentiable, thus it is fundamentally inconsistent with the underlying optimization theory. The soft maximum-function — a smooth approximation of the maximum function — was therefore used to transcribe the equation.

5.2.3.3.1 Soft Maximum Approximation

The soft maximum-function is a differentiable function and approximates maximum function \( f_{\text{max}}(c_1, ..., c_n) = \max\{c_1, ..., c_n\} \) of a set of \( n \) numbers \( c_1, ..., c_n \in \mathbb{R} \). The definition is given in equation (5.50).

\[
S_\alpha(c_1, ..., c_n) = \frac{1}{\alpha} \ln \left( \sum_{i=1}^{n} \exp(\alpha \cdot c_i) \right)
\]  

\( S_\alpha \) has the following properties:

1. \( \lim_{\alpha \to +\infty} S_\alpha(c_1, ..., c_n) = \max(c_1, ..., c_n) \)
2. \( \lim_{\alpha \to -\infty} S_\alpha(c_1, ..., c_n) = \min(c_1, ..., c_n) \).

The accuracy of the approximation is significantly dependent on the order of magnitude of the numbers \( c_1, ..., c_n \), the choice of the arbitrary parameter \( \alpha \), and the prominence of the maximum. However, for sufficiently large values of \( \alpha \), the soft maximum can accurately approximate the maximum of a set of numbers. This correlation suggests the choice of a very large \( \alpha \). However, the order of magnitude of the terms \( \exp(\alpha \cdot c_i) \) rapidly increases with
growing $\alpha$. Therefore, the parameter needs to be chosen carefully, to avoid exceeding the representable numerical range of the processing machine.

### 5.2.3.3.2 Discretised Objective Function

The continuous objective function of the OCP was transcribed to a discretised approximation in the NLP, based on the soft maximum function. The discretised objective function reads

$$J_{\text{NLP}}(\bar{\mathbf{x}}) = -(\lambda_{j_1} \cdot J_{1,\text{NLP}}(\bar{\mathbf{x}}) \cdot (1 - w_{\text{obj}}) + \lambda_{j_2} \cdot J_{2,\text{NLP}}(\bar{\mathbf{x}}) \cdot w_{\text{obj}}).$$  \hspace{1cm} (5.51)

The two function components are thereby defined as

$$J_{1,\text{NLP}}(\bar{\mathbf{x}}) = 1,332 \cdot (E(\bar{\mathbf{x}}) - M(\bar{\mathbf{x}}))$$  \hspace{1cm} (5.52)

$$J_{2,\text{NLP}}(\bar{\mathbf{x}}) = \frac{1}{\alpha_{dP/dt}} \ln \left( \sum_{k=1}^{N-1} \exp \left( \alpha_{dP/dt} \cdot (P_k - P_{k-1}) \cdot \frac{1}{h_{k-1}} \right) \right),$$  \hspace{1cm} (5.53)

where

$$E(\bar{\mathbf{x}}) = \sum_{k=1}^{N-1} x_{2,k} \cdot \left( x_{4,k} + R_{cs} (x_{2,k} - x_{3,k}) \right)$$

$$M(\bar{\mathbf{x}}) = \frac{1}{(t_{N-1} - t_0)} \sum_{k=1}^{N-1} h_{k-1} \cdot \left( x_{4,k} + R_{cs} (x_{2,k} - x_{3,k}) \right)$$

and

$$P_k = \left( x_{4,k} + R_{cs} (x_{2,k} - x_{3,k}) \right).$$

The factors $\lambda_{j_1}$ and $\lambda_{j_2}$ were introduced to scale the ranges of $J_{1,\text{NLP}}(\bar{\mathbf{x}})$ and $J_{2,\text{NLP}}(\bar{\mathbf{x}})$ to a similar order of magnitude. The chosen model-independent parameter values used for the objective function were found empirically and are given in Table 5-3. All estimated values of $dP/dt$ were within a tolerance of $-1 \text{ mmHg/s}$ when compared to the actual maximum values.

<table>
<thead>
<tr>
<th>Table 5-3 – Model-independent parameters of the objective function.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>$\alpha_{dP/dt}$</td>
</tr>
<tr>
<td>$\lambda_{j_1}$</td>
</tr>
<tr>
<td>$\lambda_{j_2}$</td>
</tr>
</tbody>
</table>
5.2.3.4 Jacobian Matrix

For each iteration step of the optimization process according to Newton’s method, the partial first derivatives of the objective \( J_{\text{NLP}}(\vec{x}) \) and constraint functions \( c(\vec{x}) \) (equation (5.54)) with respect to the discretised vector of state and control variables \( \vec{x} \) is required.

\[
c(\vec{x}) = [\xi_k^T, g_{\text{NLP}}, h_{\text{NLP}}, \psi_{\text{NLP}}, [x_0 - x_{N-1}]^T, [u_0 - u_{N-1}]^T]^T
\]  
(5.54)

While SNOPT offers the built-in functionality to estimate the first derivatives by finite differences, this process is time-consuming. The execution of the NLP was significantly accelerated by providing the iteratively calculated \((n_c + 1) \times n_v\) Jacobian matrix to the solver. The structure of the Jacobian is given in equation (5.55).

\[
\mathbf{G} = \begin{bmatrix}
\frac{\partial J_{\text{NLP}}}{\partial x_{1,0}} & \ldots & \frac{\partial J_{\text{NLP}}}{\partial x_{12,N-1}} \\
\frac{\partial c_1}{\partial x_{1,0}} & \ldots & \frac{\partial c_{n_c}}{\partial x_{12,N-1}} \\
\vdots & \ddots & \vdots \\
\frac{\partial c_{n_c}}{\partial x_{1,0}} & \ldots & \frac{\partial c_{n_c}}{\partial x_{12,N-1}} \\
\frac{\partial J_{\text{NLP}}}{\partial u_{1,0}} & \ldots & \frac{\partial J_{\text{NLP}}}{\partial u_{2,N-1}} \\
\frac{\partial c_1}{\partial u_{1,0}} & \ldots & \frac{\partial c_{n_c}}{\partial u_{2,N-1}} \\
\vdots & \ddots & \vdots \\
\frac{\partial c_{n_c}}{\partial u_{1,0}} & \ldots & \frac{\partial c_{n_c}}{\partial u_{2,N-1}} 
\end{bmatrix}
\]  
(5.55)

The size of the matrix and thus computing time increase with the number of collocation nodes. However, only a fraction of the matrix components is non-zero, thus appropriate performance of the solver was reached. Due to the complexity and large number of the derivatives, they are not explicitly given here, however, they can be derived from the NLP equations with standard methods of differential calculus.
5.3 Methods – Model Validation and Evaluation

The following section describes the methods applied to validate and evaluate the developed numerical framework. First, the model of the reduced circulatory model is validated, and a description of different scenarios, for which optimized solutions of the nonlinear program (NLP) were evaluated, is given.

Subsequently, the dual circulation model with the BiVACOR TAH is validated, and a preliminary evaluation of NLP solutions with the device in vivo is described.

5.3.1 Reduced Circulatory System Model

5.3.1.1 Runtime and Convergence

Figure 5-5 shows the solver convergence and runtime. Panel (A) shows the RMS-difference between successive obtained speed profiles with increasing node count (decreasing time step size) in revolutions per minute. The evaluated step sizes were $h_k = \{100, 50, 20, 10, 5, 3.3, 2.5\}$ ms, which correlates to the numbers of time intervals $M = \{10, 20, 50, 100, 150, 200, 300, 400\}$. Panel (B) shows the corresponding runtime of the solver to obtain the NLP solution.

The waveforms corresponding to the solutions with $M = \{10, 20, 200, 400\}$ are shown in Figure 5-6. Panel (A) shows solutions for $w_{obj} = 0$%; panel (B) shows solutions for $w_{obj} = 100$%. All solutions correspond to a power consumption limit of $\tilde{P}_{el}^* = 14 \, W$. 
For $M \geq 150$ the RMS-difference of the obtained speed waveforms was below 10 rpm, which corresponds to 0.087% of the RMS-value at $M = 400$ ($n_{rms} = 11,470$ rpm). The runtime of the solver substantially increased for $M > 200$ (Figure 5-5B). It can be observed, that the speed, pressure, and flow waveforms for $M = 200$ and $M = 400$ overlap for both evaluated values of $w_{obj}$ (Figure 5-6), and show only marginal differences ($\Delta n_{rms} = 5.9$ rpm). Therefore, a step size of $h_k = 5 \text{ ms}$ ($M = 200$) was chosen for the following simulations.

![Graphs of speed, pressure, and flow waveforms for $M = 10$, $M = 20$, $M = 200$, and $M = 400$.](image)

Figure 5-6 – NLP solutions for time step sizes of $h_k = \{100, 50, 5, 25\} \text{ ms}$, for objective function weighting factors of (A) $w_{obj} = 0\%$ and (B) $w_{obj} = 100\%$.

5.3.1.2 In Vitro Validation

The reduced model of the systemic circulation was validated in a minimalistic mock circulatory loop (MCL), consisting of a fluid reservoir, a HeartMate II axial flow rotary blood pump (St. Jude Medical, St. Paul, MI, USA), a manual resistance valve to simulate vascular resistance, and a 35 mm diameter soft silicone rubber tubing section of 300 mm length (cf. section 4.2.2) to mimic the aortic Windkessel effect. Selected optimal control scenarios were simulated and reproduced in the MCL. For this purpose, arterial impedance parameters of the MCL were identified with standard techniques [93] and employed in the NLP models. While
the model input of the NLP was the motor current, for the validation in the MCL the resulting speed trajectories were used as input to the speed controller of the RBPs, while the current requirement and the resulting haemodynamic waveforms were compared to the simulated results.

The experiments were performed with a 40% (by weight) glycerol-water solution at room temperature, which exhibits a similar viscosity and density to blood at 37°C. Pressures in the fluid reservoir and at the beginning of the compliant tubing, were measured utilizing fluid filled pressure transducers (PX181B-015C5V, Omega Engineering, Inc., Stamford, CT, USA), while the flow rate at the pump outlet was measured with clamp on ultrasonic flow meters (TS410-20PXL, Transonic Systems, Ithaca, NY, USA). The HeartMate II RBP was controlled with a custom motor controller, employing a sensorless field oriented control algorithm implemented on an off-the-shelf motor control development kit (DRV8312-C2-KIT, Text Instruments, Dallas, USA).

A schematic illustration of the experimental MCL of the systemic circulation is shown in Figure 5-7. The pump parameters such as the pressure head-flow and torque-flow characteristics were based on steady-state and dynamic in vitro measurements in the same setup, and dimension and weight measurements of a HeartMate II rotor. A centrifugal pump connected to generate flow in the opposite direction (counter-pump) to the HeartMate II device was connected to a three-way valve in the venous section downstream of the resistance valve.

![3-way valve](image)

Figure 5-7 – Mock circulatory loop for the evaluation of the reduced model NLP. $R_{sa}$, systemic arterial resistance; $AOC$, aortic compliance; $Q_s$, systemic flow rate; $P_{ao}$, aortic pressure; $P_{la}$, left atrial pressure; RBP, axial flow rotary blood pump.
Depending on the position of the three-way valve, the outlet of the counter pump was connected to the circuit and the speed was manually set to enforce negative flow rates through the HeartMate II, and thus allowed extension of the recorded pressure head and torque data to negative flow rate ranges at different operating speeds of the HeartMate II. Haemodynamic data were acquired with a dSPACE rapid control prototyping system (DS1104, dSPACE GmbH, Paderborn Germany) and postprocessed with MATLAB (The MathWorks Inc., Natick, MA, USA).

Figure 5-8 – Comparison of simulated waveforms and MCL measurements with a HeartMate II axial flow RBP. $n_{RBP}$, pump speed; $Q_s$, systemic flow rate; $P_{ao}$, aortic pressure; $I_{mot}$, motor current. Solid lines represent measurements, dashed lines represent simulations results.
Exemplary waveform results for the validation of the reduced circulatory model with a HeartMate II pump are shown in Figure 5-8. The graphs show the waveforms for pump speed ($n_{RBP}$), systemic flow rate ($Q_s$), aortic pressure ($P_{ao}$), and motor current ($I_{mot}$).

The simulations were constrained with five different levels of maximum motor power consumption ($P_{el}^* = 8, 9.5, 11, 12.5, 14 \, W$), resulting in a gradual increase of the pulse amplitude. For all shown scenarios, the minimum flow rate boundary in the NLP simulation was set to zero, thus negative flow did not occur. Each simulation was further constrained to generate a mean systemic flow rate of $Q_s^* = 5 \, L/min$ and was optimized with an objective function weighting factor of $w_{obj} = 0$, i.e. the waveforms were optimized to maximize $SHE$.

The speed waveforms for all cases exhibited a similar characteristic shape, showing a systolic pulse, which is rounded at the top, followed by a steep speed reduction, a small dip, and a ramped speed decrease during diastole. The flow waveforms exhibit a rounded pulse in systole, followed by a flow rate of zero in diastole. With an increasing power consumption limit the flow pulse becomes more and more sharp, with increasing maximum flow rate and decreasing pulse duration. The resulting pressure pulse is triangular shaped at the lowest power consumption, whereas the pressure slope in systole increases with an increasing power consumption limit. Consequently, the maximum of the arterial pressure waveform moves toward systole, occurring earlier within the cardiac cycle. The speed tracking of the custom motor controller was adequate and all measured waveforms coincided closely with the simulation results, while the maximum required motor current was underestimated by the simulation at larger flow amplitudes (Figure 5-8, bottom).

The corresponding measurement and simulation results for $SHE$, $dP/dt$, mean systemic flow rate ($\bar{Q}_s$), and motor power consumption are shown in Figure 5-9. Measured values for $SHE$ and mean flow rate were close to the simulation results, with an error between 6% and 17.5% (highest percentage error at the lowest amplitude) for measured power motor consumptions ranging between $8.1 \, W$ and $14.8 \, W$. The measured power consumption was slightly higher than the corresponding simulated values, showing a maximum deviation of 6.4%. The measured $dP/dt$ values at low amplitudes were slightly higher than the simulated values, with a difference of up to $27 \, mmHg/s$, which may in part be related to measurement noise on the pressure signal. However, the trends were similar and the error reduced with increasing speed amplitude. A similarly good agreement of simulation and measurement results was
achieved with further scenarios, including variation of the systemic arterial resistance, objective function weighting factor, and scenarios including negative instantaneous flow rates, which were not all shown here.

Figure 5-9 – Comparison of simulation and measurement results for the HeartMate II in the reduced model – collected data for (A) $dP/dt$, (B) SHE, (C) mean systemic flow rate $\bar{Q}_s$ and (D) electrical motor power consumption $P_{el}$.

The reduced model adequately predicted the hydraulic and motor dynamics of the Heartmate II in vitro. The basic components of the numerical model and the MCL representing the vascular impedance of the pulmonary and systemic circulation were previously validated [83,98,236,239], suggesting that the representation of the mammalian circulation with appropriately chosen parameters of the NLP model is similarly adequate. The model correctly predicted the trends in SHE, $dP/dt$, and motor power consumption according to the chosen objective function and NLP constraints. The differences between the measurement and simulation results can be attributed to measurement noise and model inaccuracies. Furthermore, the measured motor power consumption was slightly above the simulated values, which may be caused by imperfect commutation of the motor currents in the experiment, and the neglection of core losses in the motor model.

5.3.1.3 Evaluation of the NLP Solutions for Different Scenarios

The previous discussion of in-vitro evaluated speed profiles in chapter 4 showed that sine, square, and sawtooth speed waveforms tend to generate negative instantaneous flow rates, when the pulse amplitude is increased. As the resulting waveforms were unlike native heart waveforms, NLP solutions constrained with and without a minimum instantaneous systemic
flow boundary of zero \( (Q_s \geq 0; Q_s \text{ unconstrained}) \) were compared and verified against sine-wave and square-wave (20% duty cycle) speed profiles. The waveforms for all profiles were obtained by forward-simulation in a MATLAB/Simulink implementation of the reduced systemic circulation model and the HeartMate II. As the required control input waveforms to generate square wave and sinusoidal speed profiles under consideration of the maximum motor acceleration were unknown, a PID compensator controlling the motor speed to follow the respective sine or square target waveform was implemented and subjected to the same current limits as were applied in the NLP. The controller gains were tuned and achieved fast tracking of the target waveforms without significant overshoot or ringing. Hereby, the mean pump speeds with sine and square profiles were automatically adjusted to set the specified target mean flow rate \( (\bar{Q}_s^*) \), while the amplitude of the speed profile was adjusted to set the target mean motor power consumption \( (\bar{P}_{el}^*) \) for different operating points. Subsequently, NLP solutions for different parameters of the constraint and objective functions were obtained to evaluate trends with respect to the speed waveform shape and the generated pulsatility. Parameters, which were varied and discussed, included the objective function weighting factor, the maximum motor power consumption, the pulse rate \( (PR) \), and the mean systemic flow rate. The parameters of the cardiovascular system model were set according to the values given in Table 5-1 (section 5.2.1.1), where parameters of the arterial Windkessel and atria were derived from previous in vivo measurements in a 75 kg calf. All model parameters were held constant, except for the systemic arterial resistance \( (R_{sa}) \), which was adjusted for the variation of the mean systemic flow rate. Applicable to all solutions, \( R_{sa} \) was set according to equation (5.56), to yield a mean pump pressure head of 90 mmHg.

\[
R_{sa} = \frac{80}{7.5 \cdot 10^{-4}} \cdot \frac{90 \text{mmHg}}{\bar{Q}_s^*} - R_{la}
\]  

(5.56)

Table 5-4 – Varied parameter ranges for which NLP solutions were obtained and evaluated. \( Q_{s,\text{min}} \), minimum systemic flow rate; \( \bar{P}_{el}^* \), mean motor power consumption; \( \bar{Q}_s^* \), mean systemic flow rate; \( w_{obj} \), objective function weighting factor; \( PR \), Pulse Rate.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Values</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \bar{P}_{el}^* )</td>
<td>6,8,10, [...] ,24,26</td>
<td>W</td>
</tr>
<tr>
<td>( w_{obj} )</td>
<td>0,25,50,75,100</td>
<td>%</td>
</tr>
<tr>
<td>( Q_{s,\text{min}} )</td>
<td>(-\text{Inf}, 0)</td>
<td>L/min</td>
</tr>
<tr>
<td>( \bar{Q}_s^* )</td>
<td>3,5,7</td>
<td>L/min</td>
</tr>
<tr>
<td>( PR )</td>
<td>30,60,90</td>
<td>bpm</td>
</tr>
</tbody>
</table>
5.3.2 Dual Circulation Model and BiVACOR TAH

In the context of this work, the dual circulation and BiVACOR TAH models were exclusively applied for a preliminary in vivo evaluation of the simulated waveforms. The model validation and animal experiment methods are outlined below.

5.3.2.1 In-vitro Validation

Similar to the reduced circulatory system model, the dual circulation model was validated in vitro, utilizing a prototype of the BiVACOR TAH. The MCL setup comprised the systemic and pulmonary circulations. The setup was identical to the setup described in the previous chapter (section 4.2.2). The pressure head-flow characteristics and fluid inertia parameters were obtained from steady state and dynamic measurements in the same MCL setup with various combinations of systemic (SVR) and pulmonary vascular resistance (PVR). The hydraulic torque characteristics were obtained from measurements in the previously described hydraulic force test rig (section 2.3.1.2). The shunt flow through the device was modelled with a regression polynomial as function of left and right flow rates, impeller speed, and the differences between arterial \((P_{ao} - P_{pa})\) and atrial \((P_{la} - P_{ra})\) pressures (equation (5.16)). The corresponding data were obtained from pulsatile operating conditions with sinusoidal and rectangular speed waveforms at various magnitudes and SVR/PVR combinations, collected in a MCL and fitted into a multiple linear regression model. The MCL control and data acquisition was performed with a rapid control prototypic system (DS1103, dSPACE, Paderborn, Germany), while the recorded data were postprocessed with MATLAB.

A comparison of measurement and simulation results showing the motor speed and current, and systemic arterial pressures and flow rate waveforms, is given in Figure 5-10. The figure shows five different scenarios, for different values of the objective function weighting factor \((w_{obj} = 0\%, 25\%, 50\%, 75\%, 100\%)\). All other parameters were kept constant, including the limit for the motor power consumption, which was set to \(P_{el}^* = 8W\). The waveforms for the \(w_{obj} = 0\%\) case were similar to the results shown for the HeartMate II validation in the previous section (cf. Figure 5-8E), while the speed waveforms showed characteristics resembling the empirically, manually obtained speed profiles LUT1 and LUT2 (chapter 4). With increasing weighting factor, the systolic portion of the optimized speed waveforms transformed in shape. Profiles corresponding to a higher weighting factor exhibited a sharp
speed peak at the onset of systole, which increased in size with increasing $w_{obj}$. Accordingly, the rounded systolic speed plateau seen for $w_{obj} = 0\%$ was widened and reduced in height, compensating for the increase in power consumption during the early systolic speed peak.

Figure 5-10 – Comparison of simulation and measurement results for the dual circulation model performed with the BiVACOR TAH – waveforms of systemic haemodynamics and motor speed and current. $n_{RBP}$, pump speed; $Q_s$, systemic flow rate; $P_{ao}$, aortic pressure; $I_{mot}$, motor current; $w_{obj}$, objective function weighting factor. Solid lines represent measurements, dashed lines represent simulations results.

All measured systemic flow rate waveforms were in good agreement with the corresponding simulation. The shapes of the measured aortic pressure waveforms were similar to the simulated waveforms, but were superimposed with a harmonic ringing, causing a dip of the pressure after the initial rapid increase. The rate of change of pressure significantly increased
with the weighting factor, leading to a sharp upstroke at the pulse onset over approximately two third of the pulse pressure, followed by a slower increase to the maximum systolic pressure in the case of $w_{obj} = 100\%$. The corresponding pulmonary arterial pressure and flow waveforms, as well as the shunt flow through the device, are shown in Figure 5-11. The measured pulmonary flow rate waveforms were close to the simulation results during systole, however, showed a steeper decline during diastole. The corresponding pulmonary arterial pressures were in the same order of magnitude as the simulated results, however, the shape of the waveforms was substantially dissimilar. The shunt flow waveforms of measurements and simulations were similar, while the systolic speed peak in the cases with a weighing factor of 75\% and 100\% caused a downward peak in the shunt flow rate, which was underestimated by the simulation.

Figure 5-11 – Comparison of simulation and measurement results for the dual circulation model performed with the BiVACOR TAH – Waveforms of pulmonary haemodynamics and shunt flow through the device. $Q_p$, pulmonary flow rate; $P_{pa}$, pulmonary arterial pressure; $Q_L$, shunt flow rate; $w_{obj}$, objective function weighting factor. Solid lines represent measurements, dashed lines represent simulations results.
The mismatch of the pressure waveforms is attributed to the hydraulic characteristics of the utilized mock circulatory loop (MCL) and model inaccuracies. Specifically, the typically used rigid piping elements allow for the almost undamped progression of pressure waves through the hydraulic circuit. An often seen characteristic of MCLs including a model of the native heart is a high frequency ringing in the arterial pressure waveforms, which is typically filtered digitally [97,102,127]. The ringing is caused by water hammer surges caused by the rapid closure of mechanical check valves imitating the function of the native heart valves. Similarly, rapid pump speed changes can cause pressure waves with high frequency content of higher amplitude, which consequently add to disturbance and an unphysiologic response of the arterial pressure waveforms, which are not captured by the comparatively low-order model of the mammalian arterial system.

As described in chapter 4, the earlier used compressed air compartment representing the systemic arterial compliance was replaced with compliant silicon rubber tubing, which allowed for dampening of high frequency pressure waves in the aortic section of the MCL. However, as the pulmonary circulation is of subordinate interest for the presented work, the corresponding section of the MCL remained unchanged and a compressed air chamber was used to represent pulmonary arterial compliance in the experiments. The ringing and mismatch of the measured pulmonary arterial pressure waveforms is therefore mostly attributed to harmonic pressure wave reflections in the compliance chamber and the rigid piping. Further contributing factors may be inaccuracies in the polynomial and fit models of the pressure head-flow characteristics and the shunt flow through the BiVACOR device. Consequently, due to the simpler hydraulic setup and the absence of a compressed air compliance chamber, similar effects were not observed in the validation of the reduced model.

Despite the differences between measured and simulated haemodynamic waveforms in the pulmonary circulation, the measured results for SHE and \( dP/dt \) in the systemic circulation, mean systemic flow rate, and motor power consumption were in good agreement with the simulations (Figure 5-12). As anticipated, with increasing \( w_{obj} \) from 0% to 100% SHE decreased from 10,720 \( erg/cm^3 \) to 7,753 \( erg/cm^3 \), while \( dP/dt \) increased from 149.3 \( mmHg/s \) to 400.2 \( mmHg/s \).
Figure 5-12 – Comparison of simulation and measurement results for the dual circulation model performed with the BiVACOR TAH – collected data for (A) \( \frac{dP}{dt} \), (B) \( SH \), (C) mean systemic flow rate \( Q_s \), and (D) electrical motor power consumption \( P_{el} \). \( w_{obj} \), objective function weighting factor.

The measured motor power consumption increased slightly with the weighting factor, exceeding the simulated power consumption limit by up to 1W at \( w_{obj} = 100\% \). The systemic mean flow rate was slightly lower than the target 5 L/min, reaching a minimum of 4.53 L/min at a weighting factor of 25\%. The data shown in Figure 5-12 are further summarized in Table 5-5.

<table>
<thead>
<tr>
<th>( w_{obj} )</th>
<th>( S/M )</th>
<th>( \frac{dP}{dt} ) (mmHg/s)</th>
<th>( SHE ) (erg/cm³)</th>
<th>( Q_s ) (L/min)</th>
<th>( P_{el} ) (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0%</td>
<td>( S )</td>
<td>149.3</td>
<td>10,532</td>
<td>5.00</td>
<td>8.00</td>
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<tr>
<td></td>
<td>( M )</td>
<td>142.3</td>
<td>10,589</td>
<td>4.65</td>
<td>7.98</td>
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<tr>
<td>25%</td>
<td>( S )</td>
<td>189.3</td>
<td>10,469</td>
<td>5.00</td>
<td>8.00</td>
</tr>
<tr>
<td></td>
<td>( M )</td>
<td>181.5</td>
<td>10,782</td>
<td>4.53</td>
<td>7.94</td>
</tr>
<tr>
<td>50%</td>
<td>( S )</td>
<td>245.7</td>
<td>10,162</td>
<td>5.00</td>
<td>8.00</td>
</tr>
<tr>
<td></td>
<td>( M )</td>
<td>232.1</td>
<td>10,347</td>
<td>4.61</td>
<td>8.22</td>
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<tr>
<td>75%</td>
<td>( S )</td>
<td>320.9</td>
<td>9,279</td>
<td>5.00</td>
<td>8.00</td>
</tr>
<tr>
<td></td>
<td>( M )</td>
<td>310.7</td>
<td>9,379</td>
<td>4.63</td>
<td>8.49</td>
</tr>
<tr>
<td>100%</td>
<td>( S )</td>
<td>400.2</td>
<td>7,749</td>
<td>5.00</td>
<td>8.00</td>
</tr>
<tr>
<td></td>
<td>( M )</td>
<td>405.7</td>
<td>8,253</td>
<td>4.66</td>
<td>9.23</td>
</tr>
</tbody>
</table>
5.3.2.2 Preliminary In-vivo Evaluation

For the in vivo experiment, a set of optimized speed profiles was obtained from the dual circulatory model-based NLP prior to a scheduled animal experiment, and subsequently applied to the device. The preliminary study was performed early in the development phase of the NLP. Therefore, at the time, a thorough investigation of Windkessel parameters from previous animal studies had not been performed, and the objective function had not been fully implemented, thus an optimization to maximize $dP/dt$ was not performed. Consequently, all speed profiles were optimized to maximize $SHE \left( w_{obj} = 0\% \right)$, and the applied cardiovascular system parameter set was based on Windkessel parameters fitted to human aortic pressure and flow waveforms, which was adapted from literature [84]. Therefore, a deviation of measurement and simulation data was expected prior to the experiments. However, the study was performed in order to gain insight into the performance of the BiVACOR device with speed waveforms obtained from NLP results, and to determine their suitability for implementation in a speed control algorithm.

Five NLP solutions with power consumption limits of $\bar{P}_{el}^* = [8.5 \, W, 9 \, W, 10 \, W, 11 \, W, 12 \, W]$, a mean systemic flow rate constraint of $\bar{Q}_s^* = 5 \, L/min$, and a pulse rate of $PR = 60 \, bpm$ were generated, and the corresponding simulated speed waveforms were stored in a three-dimensional lookup-table (LUT) with two input parameters, namely the time relative to the pulse cycle, and the pulse amplitude. The pulse amplitudes for each profile were calculated as the difference between minimum and maximum speed $n_{amp} = n_{max} - n_{min}$. The lookup table was then used to generate speed profiles for continuous target amplitude inputs, whereas profiles for amplitudes between the NLP solutions were obtained by linear interpolation between the corresponding target speed values of the two nearest NLP solutions in each simulation time step. Similar to the previous in-vitro studies, the LUT storing the speed waveforms was implemented in MATLAB/Simulink and processed on a dSPACE rapid control prototyping platform (dSPACE MicroAutoBox, dSPACE GmbH, Paderborn Germany). During the experiment, the speed amplitude input to the LUT was slowly and manually increased from the minimum to the maximum speed amplitude over a time interval of $\Delta t = 150 \, s$.

The BiVACOR device was implanted in a female 80.5 kg calf in an acute animal study via left thoracotomy. After admission of general anaesthesia, the animal was intubated and
remained ventilated and anesthetized throughout the study. Atrial and arterial pressures, as well as systemic and pulmonary flow rate, were monitored and recorded analogously to the procedure described in section 2.3.1.5.2. The haemodynamic data obtained during the animal experiment was then evaluated and compared to the simulation data.

5.3.2.2.1 Ethics Approval

The study was conducted under main study protocol #2015-12, approved by the Institutional Animal Care and Use Committee (IACUC) of the Texas Heart Institute (Houston, Texas, USA). As no staff or students of Griffith University were directly involved in the handling of the animal, ethics approval by the Griffith University Animal Ethics Committee was not required.
5.4 Results and Discussion

In the following section, the solutions obtained from the nonlinear program (NLP) are presented. First, the reduced circulatory system model is considered. The convergence of the solutions and the solver runtime are presented, and the model is validated against in vitro measurements. Then, NLP solutions for different scenarios are evaluated.

Subsequently, the BiVACOR TAH in interaction with the systemic and pulmonary circulation is considered. Similar to the reduced model, the NLP solutions of the dual circulatory system model are validated against in vitro measurements, and a preliminary in vivo evaluation of speed profiles obtained from the NLP is presented.

5.4.1 Reduced Circulatory System Model

5.4.1.1 Evaluation of the NLP Solutions

Following the in-vitro model validation, the NLP solutions for various scenarios were evaluated with models of the reduced systemic circulation and the HeartMate II. Firstly, the results were compared against results obtained from sine and square waveforms. Figure 5-13 shows the corresponding results for $SHE$ and $dP/dt$ levels for power consumption limits $(\bar{P}_e)$ ranging between 8 W and 16 W.

The graphs show the results for optimized waveforms with three different objective function weighting factors ($w_{obj} = 0\%, 50\%, 100\%$), while NLP results with unconstrained systemic flow rate (left / A, B), and a minimum systemic flow rate of $Q_{s,min} = 0 L/min$ (right / C, D) were evaluated. The NLP solutions achieved both superior $SHE$ and $dP/dt$ levels compared to sine and square wave speed profiles, when the NLP solution was optimized for the specific metric. Optimization for maximum $SHE$ ($w_{obj} = 0\%$) yielded higher values of $SHE$ than all other profiles for all evaluated target power consumptions (A), while $dP/dt$ was low (B). Solutions optimized with a weighting factor of $w_{obj} = 100\%$ generated slightly lower $SHE$ values than the sine wave profiles for all power consumption limits, however, $dP/dt$ was significantly higher than the $dP/dt$ found for all other profiles. A compromise is shown with the NLP solution with $w_{obj} = 50\%$, which showed comparable SHE to the sine profile, and similar $dP/dt$ as the $w_{obj} = 100\%$ profiles, thus demonstrated the superiority of the optimized profiles over sine and square profiles.
Figure 5-13 – Comparison of the NLP results to sine and square wave profiles, showing graphs for $SHE$ and $dP/dt$ (A, B) without and (C, D) with a minimum systemic flow rate boundary of $Q_{s,\text{min}} = 0 \text{ L/min}$.

Figure 5-13 C and D show results for NLP solutions with a constrained minimum flow rate compared to the same results for the sine and square profiles. The constrained profiles showed slightly lower, but similar results for the maximum $dP/dt$, while $SHE$ was lower than the results for the unconstrained NLP solutions and the sine wave profiles, but above $SHE$ generated by the square profile. A summary of all acquired values for $dP/dt$ and $SHE$ is given in Table 5-6.
Figure 5-14 show the waveforms for two exemplary scenarios with and without a lower flow rate boundary, optimized with $w_{obj} = 0\%$ and identical power consumption. The waveforms for speed, aortic pressure, and systemic flow rate for the unconstrained case show approximately equal durations of systolic and diastolic portions of the pulse cycle.
Consequently, the maximum of the aortic pressure waveform is centered within the pulse cycle. Compared to the constrained scenario, the maximum speed and flow rate are reduced, while the pulse pressure is increased, when negative flow rates are allowed.

The given results showed, that specifically $SHE$ was significantly increased, when negative pump flow rates were tolerated (as also occurred with the evaluated sine wave profiles). Further, it was observed, that the negative diastolic flow rates caused higher atrial pressures at the onset of systole, which consequently increased the minimum atrial pressure during systole, hence it may be advantageous to avoid atrial suction in a clinical setting. However, despite these advantages, for all following simulations, the minimum flow rate boundary was set to zero for the following reasons:

1. While solutions allowing negative flow rates exhibited higher values of $SHE$ and $dP/dt$, the optimised waveforms for pressure and flow rate were unphysiologic. The aim of the study was to investigate RBP speed control schemes to allow recreation of near-physiologic haemodynamic waveforms replicating shape and characteristics of native haemodynamics, whereas no substantial reverse flow occurs during the natural cardiac cycle.
2. The numerical framework is subsequently applied to evaluate the effect of pump design characteristics (chapter 6). The pressure head-flow characteristics modelled for this purpose were derived from typical curves corresponding to different RBP types, as presented in literature [222]. The given graphs do not include curve progression in negative flow ranges, hence extrapolation would be required to simulate speed profiles resulting in negative flow rates.
It is acknowledged, that both generated $SHE$ and $dP/dt$ may be increased with waveforms allowing negative flow rates. However, for the evaluations performed in this work, these solutions are of minor interest, as the gained knowledge is similarly applicable to waveforms with temporarily negative flow rate waveforms.

5.4.1.1.1 Trends with Changes in Constraint and Objective Function Parameters
The haemodynamic waveforms and levels of $SHE$ and $dP/dt$ corresponding to the NLP results were significantly dependent on the applied constraints and objective, including the tolerated motor power consumption, the objective function weighting factor (see Figure 5-13), the required flow output, and the applied pulse rate ($PR$). The results presented in this section give an overview of typical changes of the waveforms and pulsatility corresponding to the influence of these parameters, as simulated with the reduced cardiovascular system model and the HeartMate II.

Figure 5-15 – Effect of the power consumption limit: changes in the optimised (A) speed, (B) aortic pressure, and (C) flow waveforms at $w_{obj} = 50\%$, $\bar{Q}_s = 5 \text{L/min}$, and $PR = 60 \text{bpm}$.

Figure 5-16 – Effect of the weighting factor: changes in the optimised (A) speed, (B) aortic pressure, and (C) flow waveforms at $P_{el} = 16W$, $\bar{Q}_s = 5\text{L/min}$, and $PR = 60 \text{bpm}$.

Figure 5-15 and Figure 5-16 show typical waveform changes for the power consumption limit and the weighting factor. An increase in power consumption typically resulted in a
reduced relative systolic duration and in an increase of the pulse amplitude and pulse pressure, independent of the weighting factor (Figure 5-15). An increase of $w_{obj}$ introduced a sharp peak at the onset of systole in the speed waveform, which progressed into a triangular shape of the speed waveform during systole at $w_{obj} = 100\%$. The corresponding curves of pulse pressure ($PP$), $SHE$ and $dP/dt$ as function of the power consumption for different values of $w_{obj}$ are shown in Figure 5-17.

![Figure 5-17 – Pulsatility metrics ((A) $PP$, (B) $SHE$, and (C) $dP/dt$) corresponding to different weighting factors as a function of power consumption ($\bar{Q}* = 5L/min, PR = 60 bpm$).](image)

Curves for $PP$ and $SHE$ were similar for all weighting factors, while both metrics decreased slightly with increasing $w_{obj}$. The maximum difference between solutions for $w_{obj} = 0\%$ and $w_{obj} = 100\%$ of 6.6 mmHg and 1,517 erg/cm$^3$ respectively was observed. Conversely, $dP/dt$ changed significantly, exhibiting a maximum difference of 382.3 mmHg/s. The maximum values of $SHE$ and $dP/dt$ were 12,250 erg/cm$^3$ and 459.6 mmhg/s respectively.

The effect of the weighting factor on $dP/dt$ and $SHE$ becomes clearer when the two metrics are plotted against each other in a performance map, as shown in Figure 5-18. All plotted data points represent optimised pulse patterns with a mean flow output of 5 $L/min$ at a pulse rate of 60 $bpm$. The maximum speed and motor current in the NLP were limited to 19,500 $rpm$ and 4.5 $A$ respectively, which corresponded to the limitations of the custom motor controller used for the in vitro model validation. Dashed lines represent solutions for
the same power consumption limits, while solid lines connect solutions with a constant value of \( w_{obj} \).

The instantaneous values for speed and torque reached their maximum limits for all evaluated profiles with \( w_{obj} \geq 50\% \) and \( \bar{P}_{el} \geq 8W \) during the pulse cycle, hence were limiting factors of the maximum \( dP/dt \). Conversely, waveforms optimised for \( SHE \) (\( w_{obj} = 0\% \)) did not reach either limit for \( \bar{P}_{el}^{*} \leq 18W \). For values of \( \bar{P}_{el}^{*} > 18W \), the NLP solutions for \( w_{obj} = 0\% \) and \( 100\% \) converged towards quasi identical solutions. Ultimately, a further increase of \( SHE \) and \( dP/dt \) is impossible when the limits of speed and torque are exploited, hence the maximum power consumption does not further increase, regardless of the targeted upper limit \( \bar{P}_{el}^{*} \). Similarly, at low powers, all solutions converged towards the continuous flow case.

The dashed, coloured outer limits in Figure 5-18 represent solutions, for which the maximum speed limit was omitted. Although, there is no hard speed limit, the solutions converge similarly, as the maximum speed is limited due to the increasing hydraulic torque at high
flow rates. The maximum achieved pulsatility was found with a $dP/dt$ of 518.3 mmHg/s and $SHE$ of 16,098 erg/cm³, while the maximum speed was 23,725 rpm.

The results show a rapid increase of $dP/dt$ with $w_{obj}$, while the corresponding decrease of $SHE$ was comparatively small. Hence, the results indicated, that the relative power requirement to generate a larger $dP/dt$ on overall power consumption was small compared to $SHE$. Consequently, the relative change of $SHE$ with variations of $w_{obj}$, specifically at higher $P_{el}^*$, is small for the HeartMate II. However, it should further be noted, that the change of $dP/dt$ and $SHE$ along the lines of constant power is not only depending on the weighting factor $w_{obj}$, but also on the relative scaling factors $\lambda_{j1}$ and $\lambda_{j2}$ (section 5.2.3.3.2) and the pump characteristics. While a variation of the scaling factors will therefore influence the distribution of solutions with $w_{obj} \in (0,1)$ within the performance map, equivalent solutions can be obtained when $w_{obj}$ is adjusted accordingly. However, the choice of $\lambda_{j1}$ and $\lambda_{j2}$ inevitably results in ambiguity of the weighting factor for values, which are between 0% and 100%. Consequently, as the predominant interest of the study was to investigate the maximum achievable levels of $SHE$ and $dP/dt$ with different pump characteristics, the following analysis was confined to solutions optimised with $w_{obj} = 0\%$ and $w_{obj} = 100\%$. Therefore, unless stated otherwise, all given values for $SHE$ correspond to optimised solutions with $w_{obj} = 0\%$, while values of $dP/dt$ correspond to the optimisation with $w_{obj} = 100\%$.

5.4.1.1.2 Variation of Pulse Rate and Mean Flow Rate
Comparable results were observed, when the simulation time interval, and consequently the pulse rate ($PR$) were varied. Figure 5-19 shows exemplary waveforms corresponding to a power consumption of $P_{el}^* = 16W$. The typical shapes of speed and flow waveforms had a similar amplitude for all evaluated $PR$. Due to the longer systolic duration at lower $PR$, the pressure waveforms showed a significantly higher $SHE$ and $PP$, which approximately increased proportional to the inverse of $PR$ (Figure 5-20).
The observations were in agreement with previous findings presented in literature [170]. While $SHE$ reached approached physiologic values ($SHE > 20,000 \text{ erg/cm}^3$) for the lowest pulse rate, the corresponding $PP$ reached an unphysiologic value of almost $80 \text{ mmHg}$. While further, an increase of $dP/dt$ was observed, the relative change with $PR$ was comparatively small. For all solutions corresponding to the results for $dP/dt$, the rotor acceleration was limited due to the use of the maximum motor current and consequently torque, hence a significant increase of $dP/dt$ was not possible. The maximum observed difference between $dP/dt$ for the lowest and highest pulse rate was $32.5 \text{ mmHg/s}$.

The variation of the mean systemic flow rate yielded similar waveform results, while mainly the motor power consumption was affected. According to expectation, an increase of the systemic flow rate corresponded to an increase in power consumption.
Figure 5-21 – Effect of the mean systemic flow rate: changes in the optimised (A) speed, (B) aortic pressure, and (C) flow waveforms at \( w_{obj} = 0\% \). Values for \( P_{el}^* \) were 10\( W \), 16\( W \), and 24\( W \) for mean flow rates of 3 L/min, 5 L/min, and 7 L/min.

Figure 5-22 – Maximum values of pulsatility metrics ((A) \( PP \), (B) \( SHE \), and (C) \( dP/dt \)) corresponding to different systemic mean flow rates \( \tilde{Q}_s^* \) as function of power consumption (\( PR = 60 \) bpm).

Figure 5-21 shows exemplary waveforms for three different mean flow rates, whereas the power consumptions for the illustrated profiles were chosen to reflect waveforms with a similar speed amplitude and \( SHE \). With increasing flow rate, the relative systolic duration increased, resulting in a higher \( PP \) with increasing mean flow rate, whereas \( dP/dt \) decreased. The corresponding graphs of \( PP \), \( SHE \), and \( dP/dt \) as functions of the power consumption are shown in Figure 5-22. \( SHE \) and \( dP/dt \) had similar curve shapes for all mean flow rates, while the curves were moved towards higher power consumptions with increasing flow output. \( dP/dt \) values were within \( \pm 15 \) mmHg/s from the solutions for \( \tilde{Q}_s^* = 5 \) L/min. The maximum \( SHE \) was similar for all flow rates. Maximum \( PP \) increased with flow output, up to a maximum of 51.9 mmHg.

5.4.1.1.3 Atrial Pressures and Suction

An important potential limitation to the applicable pulse amplitude in a patient is the minimum instantaneous atrial pressure. In the NLP solutions presented in Figure 5-15 - Figure 5-18, the atrial pressures were not limited to a minimum pressure boundary within the
expected range of occurring pressures, and a simulation of atrial suction was not included in
the cardiovascular system model. However, pulse profiles with higher amplitudes generated
negative instantaneous pressures during the pulse cycle. The instantaneous left atrial pressure
waveforms \( P_{LA} \) for all weighting factors at a power consumption of 16 W, and the
minimum left atrial pressures for all power consumptions and weighting factors are shown
in Figure 5-23. The minimum atrial pressure steadily decreased with increasing power
consumption for all weighting factors. The minimum observed pressure at the maximum
power consumption for the speed limited cases was \( P_{LA,\text{min}} = -5.96 \, mmHg \).

Profiles with lower values of \( w_{obj} \) corresponded to slightly lower minimum \( P_{LA} \), while the
maximum observed difference within the profiles for the same power consumption was
1.88 \( mmHg \). In reality, the observed pressures may cause atrial suction and consequently
choking of the systemic flow rate. Furthermore, the atrial output impedance is significantly
depending on the cuffs used to connect the total artificial heart (TAH) to the native
circulation, as well as the condition of the vascular inflow conduit and the atrial remnants.
Therefore, specifically when a TAH is implanted and the atria are partially removed, the
condition of the inflow conduit may significantly vary between patients. Further, the
magnitude of the decrease of the atrial pressure during pulsatile operation is depending on
the volume state of the patient. Therefore, a detailed analysis of atrial pressures with the NLP
was not intended, hence it was not implemented as a limiting factor to the pulsatility
generated by the pump.

![Atrial Pressure Waveforms and Minimum Atrial Pressure](image)

**Figure 5-23** – Atrial Pressures corresponding to the NLP solutions with \( PR = 60 \, bpm \)
and \( \bar{Q}_s = 5 \, L/min \) – (A) waveforms corresponding to a power consumption of \( P_{el} = 16 \, W \) and (B) minimum atrial pressures for all solutions as function of the power consumption.
5.4.1.2 Summary and Discussion

The evaluation of the NLP solutions showed trends of haemodynamic pulsatility with changes of the optimization objective and constraints. On the example of the HeartMate II, it was shown how optimized solutions for the objective to generate a high surplus haemodynamic energy, or a high $dP/dt$ respectively, are related. Further, the analysis showed, how maximum motor speed and torque limits influence the maximum pulsatility. While the excerpt of solutions shown here is specific to the simulated pump and cardiovascular system parameters, similar results were observed, when these parameters were changed.

When the here shown or similar optimization results are applied in a clinical setting, a deviation from the waveforms and calculated pulsatility is inevitable. Nevertheless, it is imaginable, that a control strategy for a pulsatile rotary total artificial heart can be derived from optimized waveform solutions. Although out of the scope of this research work, such a control strategy may be suitable to combine advantages of speed profiles such as sine and square wave profiles and/or improve the motor power consumption of the device. For solutions with increasing amplitude, the appearance of the generated pressure and flow waveforms was – with limitations – similar to native physiologic waveforms, exhibiting a high systolic flow peak approaching maximum flow rates of 20 $L/min$, and a corresponding steep increase in pressure, followed by an exponential decline during diastole. The given results suggest, that the choice of speed profiles corresponding to a high value of the objective function weighting factor ($w_{obj} \geq 50\%$) is reasonable, as these waveforms generated a high $dP/dt$, which outweighed a comparatively small loss in $SHE$. However, this may vary depending characteristics of the cardiovascular system and specifically the pump, and should therefore be evaluated with respect to a specific device.

Further, it appeared worthwhile to couple adjustments of the pulse rate and mean flow rate. While higher pulsatility was achieved with increasing flow output, $PP$ significantly increased accordingly. Consequently, an increase of $PR$ may be used to reduce $PP$, thus avoid excessively low or high pressures during the cardiac cycle, while potentially delivering higher pulsatile haemodynamic power and generating pressure and flow waveforms more similar to those observed in native physiology.
The highest observed values of $dP/dt$ were 459.8 mmHg/s and 518.3 mmHg/s for the speed and torque limited cases respectively, and thus approached the $dp/dt$ generated by the native heart ($440 – 1180$ mmHg/s) [115], while the corresponding maximum values of SHE (16,098 erg/cm$^3$) stayed below the reported values for the native circulation (> 20,000 erg/cm$^3$ [135]). However, as indicated before, SHE may be increased by decreasing the pulse rate, or allowing negative flow rates for the optimization. Furthermore, impending or intermediate atrial suction may limit the possible increase in the pulse amplitude or require measures to increase the atrial compliance to allow higher amplitudes. It is important to notice that the corresponding power consumptions ($\bar{P}_{el} > 24$ W) increased more than fourfold compared to the continuous flow (CF) case ($\bar{P}_{CF} = 5.88$ W). Therefore, the corresponding battery drain and heat generation within the device are likely unacceptable for the permanent implementation in a clinical setting. Consequently, improvement of the pump and motor efficiencies and characteristics is required to allow feasible generation of waveforms closely imitating the characteristics of native physiology.

5.5 Dual Circulation Model with the BiVACOR TAH

5.5.1 Preliminary In-vivo Evaluation of Optimized Speed Profiles

Figure 5-24 shows the optimized speed waveforms, which were implemented in a three-dimensional lookup table (LUT) prior to the animal study. Table 5-7 shows a summary of the corresponding mean speed, speed amplitude, and power data for each profile. Corresponding haemodynamic and pulsatility data is presented in Figure 5-25. As anticipated, the data collected during the animal experiment showed significant differences to the simulated data. It was mainly observed, that the measured systemic vascular resistance (SVR) was lower in the animal experiment (1085 – 1255 dyn·s/cm$^5$), when compared to the simulation results (1450 dyn·s/cm$^5$, Figure 5-25F), resulting in a lower mean arterial pressure (Figure 5-25C) and elevated flow rates (indicated by the data point colour), which ranged from 5.24 L/min to 6.11 L/min.

Consequently, the resulting hydraulic power and the motor power consumption (Figure 5-25E) were lower in the experimental data. It can further be seen, that the pulse pressure (PP) and surplus haemodynamic energy (SHE) were overestimated by the simulation (Figure 5-25A, D), while the measured $dP/dt$ values were higher in the in-vivo data (Figure
Atrial pressures stayed between 2 mmHg and 22 mmHg throughout the experiment.

Table 5-7 – Simulated Speed and Power data for the implemented lookup-table profiles.

<table>
<thead>
<tr>
<th>$P_{el}^*$ (W)</th>
<th>$n_{mean}$ (rpm)</th>
<th>$n_{amp}$ (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8.5</td>
<td>2140.4</td>
<td>188</td>
</tr>
<tr>
<td>9</td>
<td>2150.4</td>
<td>457</td>
</tr>
<tr>
<td>10</td>
<td>2164.7</td>
<td>781</td>
</tr>
<tr>
<td>11</td>
<td>2173.3</td>
<td>1005</td>
</tr>
<tr>
<td>12</td>
<td>2179.3</td>
<td>1208</td>
</tr>
</tbody>
</table>

Figure 5-24 – Optimized speed waveforms for the BiVACOR TAH obtained from NLP solutions for application in the animal study.

Figure 5-25 – Haemodynamic and pulsatility data obtained from the NLP solutions and during the animal study, showing (A) $PP$, (B) $dP/dt$, (C) mean aortic pressure ($MAP$), (D) $SHE$, (E) motor power consumption ($P_{el}$), and (F) systemic vascular resistance ($SVR$). $Q_{s.m}$, mean systemic flow rate. Applied speed profiles were waveforms optimized with the NLP prior to the experiment. Each data points from the in-vivo experiment corresponds to one pulse cycle, while the colour indicates the mean systemic flow rate.
Exemplary haemodynamic and pump waveforms obtained with NLP speed waveform with the highest amplitude \( (n_{amp} = 1208 \text{ rpm}) \) are shown in Figure 5-26. The aortic pressure varied between 72 \( \text{mmHg} \) and 128 \( \text{mmHg} \), while the maximum systemic flow rate during systole was 17.1 \( \text{L/min} \) at a mean flow rate of 5.58 \( \text{L/min} \). The flow rate showed a high peak during systole and decreased to values below zero in diastole. Contrary to the simulation results, the diastolic flow rate did level at zero, but at values between 2.1 \( \text{L/min} \) and 3.5 \( \text{L/min} \). The instantaneous motor power consumption peaked at 38.1 \( \text{W} \), whereas the mean value was 10.2 \( \text{W} \), which was an increase of approximately 3.7 \( \text{W} \), or 57\% compared to the lowest applied speed amplitude. The \( dP/dt \) of the aortic pressure reached values up to 470 \( \text{mmHg/s} \).

Figure 5-26 – Haemodynamic, speed, and power consumption waveforms obtained with the BiVACOR TAH at a speed amplitude of \( n_{amp} = 1208 \text{ rpm} \) in-vivo. Panels show (A) rotational impeller speed, (B) systemic flow rate, (C) aortic pressure, (D) motor power consumption, (E) left atrial pressure, and (F) aortic \( dP/dt \).

### 5.5.2 Summary and Discussion

The preliminary in vivo results illustrated, that the NLP solutions are substantially dependent on the cardiovascular system parameters. The data obtained during the in-vivo study showed similar trends to the simulated waveforms generated prior to the experiment, however, specifically the values of \( SHE \) showed a large deviation between measurement and simulation data, with significantly lower values observed in the in-vivo experiment. Hereby,
the differences are mostly attributed to incorrect parameters of the cardiovascular system model. These parameters were assumed prior to the experiment, and therefore not matched to the specific animal model. While the difference to the simulated waveforms was substantial, $SHE$ reached values up to $14510 \ erg/cm^3$, which was approaching physiologic values. Further, with maximum values $> 450 \ mmHg/s$, the observed aortic $dP/dt$ was in physiologic ranges, while the applied speed profiles did not cause atrial suction. The corresponding increase in the motor power consumption may require improvement, but stayed within reasonable levels for the evaluated speed profiles. Therefore, despite the deviation from the simulations, the results were promising with respect to the objective to recreate a physiologic pulse with a rotary total artificial heart. However, further studies are required to determine if adjustments of the speed profiles with patient-specific data are beneficial to maximize the RBP-generated pulsatility.

5.6 Limitations and Future Work

The limitations of the presented numerical framework included the applied simplifications and assumptions for the implemented cardiovascular system and pump models. Therefore, the extension to a higher order cardiovascular system model, or to include additional models to include power loss due to eddy-currents and iron hysteresis in the pump motor may improve the accuracy of the solutions. However, the NLP validation with the HeartMate II and BiVACOR pumps showed, that reasonable agreement between simulations and measurements could be achieved, whereas an extension of the model may substantially increase the computational effort. Further development and investigation of the MCL representation of the pulmonary arterial tree may allow for model improvements and further investigation of pulsatility in the pulmonary circulation, which was widely omitted in this study.

Lastly, an extended evaluation of NLP results simulated with aortic Windkessel data from different subjects is required to evaluate the requirement to adapt the speed profiles based on patient-specific data, when a control strategy based on numerically optimized speed profiles is implemented. Specifically, the variation of the speed profile shape in response to cardiovascular system model parameters should be evaluated in the future, to investigate the potential decrease of pulsatility or increase of motor power consumption with changes in CVS parameters, when the speed profile is not adjusted accordingly. For this purpose, further
numerical, in-vitro, and in-vivo investigations may assist in the development of a control strategy suitable for the long-term application in a pulsatile rotary total artificial heart. This may further require the development of advanced adaptive control and estimation algorithms to allow early detection and prevention of atrial suction. Future studies may include a separate evaluation of speed profiles optimized for maximum $dP/dt$ and maximum SHE, to determine the effects of the corresponding pulsatile haemodynamics on the mammalian physiology, and evaluate the shear stress and corresponding blood damage generated by RBP operated with speed profiles optimized for different objectives, and for different pulse magnitudes and rates.

5.7 Conclusion

The aim of this chapter was to develop a numerical framework for the evaluation of speed modulation modes for pulsatile rotary blood pumps. A nonlinear program to numerically optimize RBP speed profiles with respect to different haemodynamic objectives was developed and validated with in-vitro measurements. The dependency of the optimized solutions on specific constraint and objective functions was evaluated, and a control strategy for RBP based on the obtained solutions was outlined. The model was subsequently used for an evaluation of the effects of pump design characteristics on the device performance and maximum generated pulsatility, which is described in the following chapter.
6 Influence of RBP Design Characteristics on Induced Pulsatility

Various design characteristics influence the dynamic behaviour of a rotary blood pump (RBP). While it is obvious that improvements of the hydraulic and motor efficiency allow reduction of the power requirement, design trade-offs exist, where more than one parameter influencing the pump characteristics is affected. As shown in chapter 3, this may specifically apply to the motor drive. For example, a change of the permanent magnet height influences both the motor torque and the rotor inertia, which oppositely influence acceleration of the drive. Similarly, increasing the impeller vane height may improve the hydraulic characteristics, but in contrast require some sacrifice in motor performance due to a larger gap length.

In the previous chapter a methodology to derive an optimized control strategy for pulsatile RBPs was developed. The implemented numerical framework allows evaluation of the maximum producible pulsatility of a given RBP within the limits of the corresponding motor drive. Therefore, the effect of changes of the pump and drive design on the maximum $dP/dt$ and $SHE$ under optimal control can be investigated without the requirement to choose a generic speed profile such as a square wave or sine wave function.

In the following, the influence of different parameters of the previously developed nonlinear program (NLP) and the underlying reduced model of the systemic circulation are investigated. It was indicated earlier, that the steepness of the pressure head-flow characteristic (HQ-curve) of the pump may significantly influence the performance in pulsatile operation. While the HQ-characteristic depends on various geometrical parameters, it is further related to the pump type, size, and typical operating speed. Therefore, in this chapter a generalized RBP model based on typical pump characteristics for different pump types is derived, and the characteristics corresponding to six different modelled pumps exhibiting increasingly steep HQ-curves are compared. The results are discussed to gain a basic understanding of the influence of the HQ characteristic. Subsequently, the sensitivities of the generated $SHE$ and $dP/dt$ to changes of the motor torque constant, rotor inertia, hydraulic efficiency, and motor resistance (and consequently motor efficiency) are evaluated for two of the six pumps modelled. The results are then set in context and possible design preferences for pulsatile RBP are discussed.
6.1 Aim

The aim of this chapter is to investigate the influence of RBP design characteristics on the ability of the device to generate pulsatile haemodynamics. The objectives to achieve this aim are:

- Derive a generalized RBP model for different pump types such as axial flow, mixed flow, and centrifugal pumps and the corresponding motor drive.
- Evaluate the effect of the HQ-curve steepness on metrics of pulsatility.
- Investigate the relative effect of drive parameters on the same metrics.
- Set the results in context to discuss preferable RBP characteristics with respect to device design trade-offs.

6.2 Background

6.2.1 Specific Speed and Hydraulic Characteristics

The steepness of the pressure head-flow characteristics and hydraulic efficiency of RBPs (pressure sensitivity) are influenced by manifold geometrical design parameters of the pump housing, inflow and outflow conduits, and impeller. These parameters include the number of impeller vanes, the height and angle of the vanes at the inlet and outlet, or the shape of housing. Consequently, pumps with vastly different HQ-curves can be designed for a specific pump type and size. Design changes further influence the required speed needed to operate at a specified flow rate and pressure head. This specifically applies to the type of impeller and the corresponding flow direction. Compared to centrifugal pumps of similar dimensions, axial flow pumps operate at substantially higher rotational speeds, while centrifugal pumps typically exhibit a flatter HQ-curve \[219,222\]. However, with consideration of mixed flow pumps, the borders between the categories become blurred.

An important variable to characterise and distinguish different types of impellers is the specific speed at a pump’s best efficiency point (BEP) \(^5\), which is given by equation (6.1):

\[
N_q = n \cdot \sqrt{\frac{Q_{BEP}}{h_{BEP} \cdot \pi}},
\]

where:

\(^5\) Different notations of the specific speed are used, which are based on metric or imperial units for head and flow rate, or a truly dimensionless representation. In this work, the notation based on metric quantities was adopted from [222].
Although the value of \( n_q \) is not dimensionless, pressure head, flow rate, and impeller speed are typically normalised and the unit is omitted. A truly dimensionless representation \( \left( \omega_s = \frac{n_q}{52.9} \right) \) [222] can be calculated by substituting the impeller speed with the angular velocity, and substituting the pump head with the product of head and gravitational acceleration \( (g \cdot h_{BEP}) \). The specific speed is a type number to characterize the proportions of the pump impeller, setting the operating speed of a pump in relation to the corresponding flow rate and pressure head. The specific speed is independent of the geometric size of the machine, or its actual operating speed, therefore it is directly related to the pump type. Figure 6-1 shows typical impeller types corresponding to different ranges of specific speed.

Lower specific speeds correspond to centrifugal pumps \( (n_q \approx 7 - 30) \), whereas higher specific speeds \( (\text{typically } n_q > 170) \) correspond to axial flow pumps. The intermediate specific speed range comprises mixed flow/diagonal pumps.

High specific speed pumps typically exhibit a significantly steeper HQ-characteristic compared to their low speed equivalents corresponding to the same best efficiency point. An example of typical characteristics corresponding to various specific speeds, is shown in Figure 6-2. Similar characteristics are presented in pump design textbooks [219,222]. The panels show typical pressure head-flow curves and the corresponding efficiency and the required mechanical input power normalised to their respective values at BEP. It can be
observed, that the shutoff-head (head at zero flow) substantially increases with specific speed (denoted as $N_s$ in the figure). While the head-flow characteristics are not continued to the intercept with the flow axis, the head at a flow of 130% BEP decreases with increasing specific speed, and the trend of the increasing curve steepness continues up to this point. Consequently, it can be reasonably presumed, that the flow rate at which the head-flow characteristic crosses the x-axis decreases with increasing $n_q$.

It can further be observed, that the mechanical power input decreases with increasing flow for high specific speed pumps, while it increases with flow for low specific speed pumps. The resulting efficiency curve becomes narrower for larger $n_q$, as the relative efficiency at flow rates above and below the BEP is slightly lower for high specific speed pumps.

While the observed trends in principle hold true for many RBPs, the borders between axial flow, mixed flow, and centrifugal pumps are blurred. Therefore, an axial flow pump with a specific speed $n_q<100$ is conceivable. Similarly, the HQ-gradient can be significantly changed by adjustments of the geometrical impeller parameters, thus a centrifugal pump with a steeper HQ gradient than an axial flow pump for the same application and with the same best efficiency point can exist.

Figure 6-2 – Normalised pressure head-flow, efficiency, and mechanical power input curves for four different specific speeds. $N_s$, specific speed. Figure adapted from [241].

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Specifically for RBP s this notion is substantiated by a study by Smith et al. [242], who reviewed a total of 35 different RBP designs of different types and found a high variability in the performance and pressure sensitivities of the various evaluated designs.

6.2.2 Rotor Diameter

In the context of the previously stated assumption, that a flatter HQ-curve may be beneficial for rapid speed modulation, the given characteristics suggest, that centrifugal pumps may be more suitable for the application in a pulsatile RBP. However, in this context it is important to consider the rotor inertia associated with different pump types. It can be observed in Figure 6-1, that the vane diameter decreases with increasing $n_q$. This relationship between the specific speed and the impeller diameter can be quantified by the specific diameter, which is calculated according to equation (6.2) [243]:

$$\delta_s = D \cdot \frac{(h_{BEP})^{0.25}}{\sqrt{Q_{BEP}}}. \quad (6.2)$$

where:

- $D = \text{impeller diameter in } m$
- $Q_{BEP} = \text{flow rate at BEP in } m^3/s$
- $h_{BEP} = \text{pump head at BEP in } m$.

Similar to the specific speed, the specific diameter is a type number independent of the pump size, and indicates the typical impeller diameter corresponding to the impeller type. While specific speed and specific diameter are not directly interdependent, their relation is typically illustrated in a Cordier-diagram. Figure 6-3 shows an example of such a diagram, plotting the specific speed against specific diameter (truly dimensionless representations) for several RBP designs on a double-logarithmic scale [242]. The plotted specific speeds range between $\omega_s = 0.24$ and $\omega_s = 1.69$ ($n_q = 12.7$ and $n_q = 89.4$).
The figure shows the typical decrease of the specific diameter with increasing specific speed. This trend is in agreement with findings from industry, which relate the two characteristics in a Cordier-line for “well-designed” pumps, which was derived as a regression function through collected data from industrial pumps [243–245]. Due to the smaller rotor diameter, compared to centrifugal pumps, the fast running axial flow pumps require a relatively lower mechanical torque at the same output power and exhibit a lower rotor inertia, and thus typically allow faster acceleration. In contrast to the associated pressure head-flow curves, these characteristics indicate that high specific speed pumps may be favourable for rapid speed modulation.

While both the hydraulic and motor performance and the rotor inertia are strongly dependent on specific pump designs, it is unclear which characteristics are most influential. Hence, it may be beneficial to attempt to compare pumps of different types under similar conditions. Therefore, a model to allow such a comparison is proposed and evaluated in the following.
6.3 Methods

6.3.1 Generalized RBP Model of pumps with various specific speeds

Several different pumps are currently clinically available or under development, ranging from axial flow to centrifugal pumps, which exhibit different pressure sensitivities. When a RBP is developed under consideration of rapid speed modulation, the influence of the pump type should be considered. Due to the higher pressure-sensitivity, pumps exhibiting a flatter HQ-curve have been shown to transfer greater pulsatility from the native ventricle, when operated as a ventricular assist device at constant speed. However, which characteristics are favourable to autonomously generate pulsatility remains unclear, as theoretically any type of pump can generate similar waveforms, when the speed profile is adjusted accordingly. In this section, an attempt is made to formulate a generalized model of a RBP and corresponding motor, to investigate the effect of the pressure head-flow (HQ) curve steepness on rapid speed modulation performance. Six different example pump types are modelled based on typical HQ-curves corresponding to different specific speeds, and their performance is compared. For each pump type, the rotor inertia and the parameters of the motor were adjusted to yield comparable baseline conditions, with the aim of isolating the effect of the steepness of HQ-curves associated with the different specific speeds.

6.3.1.1 Hydraulic Characteristics

The hydraulic performance, in terms the pressure head-flow (HQ) and hydraulic efficiency curves, is determined by many parameters of the impeller and housing geometries. Consequently, a comparison of different pump types with respect to their efficacy during pulsatile operation is difficult, when no exact characteristics are given. While the steepness of the HQ-curve for a certain specific speed may substantially vary, a general trend of increasing steepness with increasing specific speed can be observed, while the shutoff-head increases and the maximum flow at zero head decreases.

To yield a realistic change of the HQ-curve gradients with specific speed, a possible approach is to model the hydraulic characteristics following the example of typical performance characteristics for pumps of different specific speeds as presented in literature [219,222,241]. However, as operating in a range with flow rates above the BEP is impractical for industry applications, the presented characteristics are typically only given up to 120 – 130% of the
BEP flow rate, thus their adoption requires extrapolation. Consequently, such data cannot be understood as an exact underlying model, but only as an indication of reasonable characteristics associated with different values of $n_q$. Nevertheless, various authors consonantly present characteristic HQ-curves with increasing steepness corresponding to increasing specific speeds (e.g. Figure 6-2, Figure 6-4 A). A RBP model approximation based on this recognition may therefore allow investigation, if a clear trend of device performance with rapid speed modulation, which corresponds to the hydraulic characteristics, is identifiable. The following approach was derived under this premise, hence the modelled characteristics should be understood as one possible pump type corresponding to each specific speed, whereas the increasing steepness of the HQ characteristics was paramount.

![Diagram](image_url)

Figure 6-4 – Normalized Performance characteristics for pumps of various specific speeds. (A) Exemplary pump characteristics as published in [222], reprinted with permission of Springer Nature. (B) Polynomial approximations of six pump curves shown in (A), which were scaled to a best efficiency point with $Q_{BEP} = 5 \text{ L/min}$; $H_{BEP} = 90 \text{ mmHg}$; $\eta_{hyd,BEP} = 30\%$ and extrapolated to $H = 0 \text{ mmHg}$.

A different terminology was used in this work and in the textbook by Gülich [222]; however, the following expressions are equivalent:

$$\frac{H}{H_{opt}} \equiv \frac{H}{H_{BEP}} ; \quad \frac{\eta}{\eta_{opt}} \equiv \frac{\eta_{hyd}}{\eta_{hyd,BEP}} ; \quad q^* \equiv \frac{Q}{Q_{BEP}}$$

6A different terminology was used in this work and in the textbook by Gülich [222]; however, the following expressions are equivalent:
The derived pressure head-flow and efficiency characteristics modelled following the example of typical performance characteristics for pumps with specific speeds between \( n_q = 20 \) and \( n_q = 250 \), as presented by Gülich (Fig. 4.11 A) [222]. Polynomial approximations of the hydraulic pressure head-flow and efficiency characteristics were fitted to extracted data from the exemplary HQ and efficiency data from the textbook. The normalized data were scaled to a common BEP, which was defined at a typical RBP operating point with a flow rate of \( Q_{BEP} = 5 \text{ L/min} \), a pressure head of \( H_{BEP} = 90 \text{ mmHg} \), and a maximum efficiency of \( \eta_{hyd,BEP} = 30\% \). To limit complexity of the model, the dip in the HQ-curve corresponding to \( n_q = 250 \) was not considered, as it would require a high order polynomial representation of the curve. The modelled hydraulic HQ and efficiency curves are shown in Figure 6-4 B. The operating speed at BEP, and the corresponding maximum flow rate \( (Q_{max}) \) and shutoff pressure head \( (H_{shutoff}) \) for all pumps are listed in Table 6-1.

<table>
<thead>
<tr>
<th>( n_q )</th>
<th>Identifier</th>
<th>( n_{BEP} \text{ [rpm]} )</th>
<th>( H_{shutoff} \text{ [mmHg]} )</th>
<th>( Q_{max} \text{ [L/min]} )</th>
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</thead>
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<td>( n_{q40} )</td>
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</tr>
<tr>
<td>80</td>
<td>( n_{q80} )</td>
<td>9,757</td>
<td>157.3</td>
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<tr>
<td>100</td>
<td>( n_{q100} )</td>
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<td>164.7</td>
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</tr>
<tr>
<td>250</td>
<td>( n_{q250} )</td>
<td>30,490</td>
<td>314.5</td>
<td>6.24</td>
</tr>
</tbody>
</table>

From the hydraulic power output and the corresponding efficiency in each operating point, a load torque function was derived from the power balance:

\[
T_{hyd} = \frac{60 \cdot P_{mech}}{2\pi \cdot n} = \frac{60 \cdot P_{hyd}}{2\pi \cdot n} \cdot \frac{1}{\eta_{hyd}} = \frac{60 \cdot Q \cdot H}{2\pi \cdot n} \cdot \frac{1}{\eta_{hyd}}
\]  

(6.3)

The mechanical power and load torque curves corresponding to the hydraulic characteristics shown above are plotted in Figure 6-5. A summary of the values and polynomials defining the characteristics of the modelled pumps is given in Appendix B. Lastly, as the fluid volume in the pump and cannulae can be assumed to be similar for all different modelled pump types, the fluid inertia \( (L_{RBP} = 0.31 \text{ mmHg} \cdot s^2/mL) \) was kept constant for all modelled pumps.
Figure 6-5 – Mechanical load characteristics of all modelled pumps \( (n_q = 20 - 250) \), showing (A) mechanical power and (B) hydraulic torque at the speed corresponding to the common BEP.

6.3.1.2 Motor Characteristics

Corresponding to each of the above derived theoretical hydraulic characteristics, an adjusted motor model was defined. Although the best efficiency point for all modelled pumps is identical, the corresponding motor characteristics significantly differ. While the converted mechanical power is in the same or order of magnitude, the relative magnitude of impeller speed and hydraulic torque are entirely different for pumps with a low or a high specific speed respectively. Consequently, the requirements for the maximum torque and speed of the driving motor depend on the specific speed of the pump. Due to geometrical advantages, different motor structures may be used for the different pump types. For example, the high specific speed axial flow pumps are typically equipped with a radial flux motor, while many low specific speed centrifugal pumps are driven by an axial flux motor. Consequently, typical rated torque and speed values vary for the same electrical input characteristics. To eliminate the influence of these differences, a generalized set of assumptions for the motor characteristics was formulated. Based on these assumptions, and the hydraulic characteristics for each pump, the torque constant and speed limits for a motor model corresponding to each pump were derived. The following basic assumptions and definitions for the motor model were made:
1. The stator winding resistance for each motor is equal ($R_{\text{mot}} = 1 \text{ ohm}$).
2. The efficiency of each motor is equal when the pump is operating at the hydraulic best efficiency point ($\eta_{\text{mot}, \text{BEP}} = \text{const}$).
3. The maximum mechanical output power of each motor (when operating with maximum speed and maximum torque) is equal ($P_{\text{mech,max}} = \text{const}$).
4. The rotor inertia of each motor is of such value, that the time interval to accelerate between two specified hydraulic operating points at a constant load torque is equal for each pump (see section 6.3.1.2.3).
5. The value of the rotor inertia of the pump with $n_q = 20$ ($J_{\text{rot},20}$) was assigned (see section 6.3.1.2.3).

The quantities defining the motor characteristics were individually calculated from the above assumptions and the previously derived hydraulic model. The derived motor model is discussed in the following.

6.3.1.2.1 Torque Constant and Motor Efficiency

The torque requirement of the motor is dependent on the hydraulic load of the pump. It can be observed in Figure 6-5 B, that the hydraulic load at the best efficiency point decreases with increasing specific speed. As the mechanical power at BEP is identical, the corresponding load torque decreases with $T_{\text{BEP}} \sim \frac{1}{n_q}$. Consequently, the torque and speed range of the motor need to be adapted to the pump requirements. For the given model, the torque was derived from the hydraulic load torque and the 1\textsuperscript{st} and 2\textsuperscript{nd} assumptions. As the motor model includes Ohmic losses only, the efficiency is calculated according to equation (6.4). The efficiency characteristic with changes in motor speed and current is shown in Figure 6-6.

$$\eta_{\text{mot}} = \frac{P_{\text{mech}}}{P_{\text{mech}} + P_{\text{cu}}} = \frac{2\pi \cdot T_{\text{mot}} \cdot \frac{n}{60}}{2\pi \cdot T_{\text{mot}} \cdot \frac{n}{60} + 3 \cdot I_{\text{mot}}^2 \cdot R_{\text{mot}}} \quad (6.4)$$
Figure 6-6 – Motor efficiency at three different load torques as a function of the speed normalized to the speed in the BEP.

Utilizing the hydraulic speed and torque at the best efficiency point, and the 1st and 2nd assumptions, equation (6.4) can be rearranged to substitute the motor current in the torque equation (6.5)

\[ T_{mot} = I_{mot} \cdot K_t \]  

(6.5)

to yield the torque constant\(^7\):

\[
K_{t,i} = \frac{T_{BEP}}{I_{BEP}} = \frac{T_{BEP}}{\sqrt{\frac{2 \pi \cdot T_{BEP} \cdot n_{BEP,i} \cdot (1 - \eta_{BEP})}{3 \cdot R_{mot} \cdot \eta_{BEP}}}}, \quad i = 20, 40, 60, 80, 100, 250
\]

(6.6)

As the stator resistance is identical for all modelled motors, so is the motor current and consequently the electrical motor power consumption at BEP. The chosen winding resistance for the generalized model \(R_{mot} = 1 \, ohm\) is within a reasonable range in the context of typical brushless DC drives for rotary blood pumps.

\(^7\) \(K_{t,i}\) refers to the torque constant of the motor corresponding to the pump with specific speed \(n_q = i\).
6.3.1.2.2  Speed and Torque Limits

In order to make a reasonable comparison of the defined pump models, appropriate limits for the speed and torque need to be defined. Typically, besides limitations originating from the rotor bearings or excessive heat generation, speed and torque are limited by the maximum supply voltage and current. In the model described here, the torque was limited by the corresponding limit of motor current. However, as the model of the electrical motor circuit was simplified, the inductive voltage drop and back-EMF voltage were not considered, and a definition of the maximum speed through a voltage limit was not possible; instead, the maximum speed was derived from a common maximum mechanical output power (3rd assumption). As the specific speed increases, the torque requirement and consequently the maximum electromagnetic torque (generated with the maximum current $I_{\text{max}} = 4.5 \, \text{A}$) given by

$$T_{\text{max}} = I_{\text{max}} \cdot K_t$$

(6.7)

decreases. As the maximum mechanical output power is constant, the maximum speed increases accordingly inversely proportional:

$$n_{\text{max}} = \frac{60 \cdot P_{\text{mech.max}}}{2\pi \cdot T_{\text{max}}}.$$  

(6.8)

An illustration of the calculated speed and torque boundaries adjoining the common maximum mechanical power limit at the point of maximum speed and maximum torque is shown in Figure 6-7. It can be shown that the ratio $n_{\text{max}}/n_{\text{BEP}}$ and therefore the speed range relative to the BEP speed are equal for all pumps. Following from the previously made assumptions it can be calculated that a maximum mechanical motor power of $P_{\text{mech}} = 45 \, \text{W}$ is equivalent to the 2.07 times the respective speed at BEP for all pumps ($n_{\text{max}} = 2.07 \cdot n_{\text{BEP}}$).
Fast rotor acceleration is important to maximize the $dP/dt$ generated by the pump. However, the ability of the motor to accelerate is not only influenced by the load torque, but also directly related to the moment of inertia of the rotor ($J_{rot}$). The inertia depends on the rotor diameter and the rotor mass, which typically includes the impeller housing and vanes, and an internal magnetic structure with permanent magnets and a ferromagnetic yoke to close the magnetic motor flux. While pumps with a high specific speed (axial flow pumps) have a comparably small specific diameter, a larger speed range must be covered during rotor acceleration, thus a simple comparison of the rotor acceleration ($dn/dt$) is inadequate. Therefore, to allow a fair comparison of the modelled pumps in the described model, the inertia was defined in such a way, that each of the modelled motors accelerates between two pump specific operating points within the same time interval $\Delta t_{acc}$. Hereby, it was assumed, that each motor is loaded with the corresponding hydraulic torque at best efficiency ($T_{BEP}$) and accelerates at full motor current ($T_{mot} = T_{max}$). The acceleration of the motor is given by the dynamic speed equation, in which, when the linear acceleration between a lower speed
operating point $n_{ls}$ and a higher speed operating point $n_{hs}$ is considered (Figure 6-8), the differential can be replaced by the speed difference:

$$\frac{n_{hs} - n_{ls}}{\Delta t_{acc}} = \frac{60}{2\pi \cdot J_{rot}} (T_{\text{max}} - T_{\text{BEP}})$$  \hspace{1cm} (6.9)

The relationship between the rotor inertias of the pumps for all specific speeds is defined, when $\Delta t_{acc}$ is defined. However, to not require an assumption for $\Delta t_{acc}$, the pump with specific speed $n_q = 20 \ (n_{q20})^8$ was assigned a reasonable inertia value in the context of RBPs. For this purpose, the values of specific speed and rotor inertia for two state-of-the-art rotary ventricular assist devices were estimated: The HeartMate II (St. Jude Medical, St. Paul, MI, USA) and HVAD (HeartWare, Framingham, MA, USA). The respective specific speed was calculated from hydraulic measurements in a mock circulatory loop, while the rotor inertias were estimated based on the measured physical rotor dimensions and weights according to the inertia formula for a solid cylinder. The estimated values are given in Table 6-2.

<table>
<thead>
<tr>
<th>VAD Type</th>
<th>Specific Speed</th>
<th>Rotor Inertia / $10^{-6} \text{ kg} \cdot \text{m}^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>HeartMate II</td>
<td>92.3</td>
<td>0.25</td>
</tr>
<tr>
<td>HVAD</td>
<td>20.65</td>
<td>5.5</td>
</tr>
</tbody>
</table>

Based on the very similar estimated specific speed for the HeartWare HVAD, the pump rotor inertia of the pump with $n_q = 20$ was equally defined to be $J_{rot,20} = 5.5 \text{ kg} \cdot \text{m}^2$. Rearrangement of equation (6.9) and equating the formula with the corresponding formula for the $n_q = 20$ pump results in:

$$J_{rot,i} = \frac{T_{\text{max},i} - T_{\text{BEP},i}}{n_{hs,i} - n_{ls,i}} \cdot \frac{n_{hs,20} - n_{ls,20}}{T_{\text{max},20} - T_{\text{BEP},20}} \cdot J_{rot,20} \ , \ i = 40, 60, 80, 100, 250$$ \hspace{1cm} (6.10)

The low and high speed operating points for each pump motor were individually calculated as the respective speeds at which the corresponding hydraulic system generated pressure.

---

^8 Pumps corresponding to a specific speed of $i$ are denoted with $n_{qi}$ in the following
heads of $H_1 = 80 \text{ mmHg}$ and $H_2 = 120 \text{ mmHg}$ against the hydraulic resistance, at which the pumps operate at BEP ($R_{hyd} = \frac{H_{BEP}}{Q_{BEP}}$).

The calculated rotor inertias for the simulated pumps and estimated values for the Thoratec/St. Jude Medical HeartMate II and HeartWare HVAD pumps are shown in Figure 6-9. Given the typical double-logarithmic decline of specific impeller diameter with specific speed [242] (cf. Figure 6-3), the inertia values found here are within a valid range for pumps in the considered operating range. Furthermore, the estimated specific speed and rotor inertia of the HeartMate II and HVAD fit seamlessly into the curve resulting from the applied assumption. The calculated inertia values corresponding to Figure 6-9 are summarized in Table 6-3.

### Table 6-3 – Values of the rotor inertia for all simulated pumps.

<table>
<thead>
<tr>
<th>$n_q$</th>
<th>$J_{rot} / 10^{-6} \text{ kg} \cdot \text{m}^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>5.5</td>
</tr>
<tr>
<td>40</td>
<td>1.32</td>
</tr>
<tr>
<td>60</td>
<td>0.55</td>
</tr>
<tr>
<td>80</td>
<td>0.296</td>
</tr>
<tr>
<td>100</td>
<td>0.189</td>
</tr>
<tr>
<td>250</td>
<td>0.026</td>
</tr>
</tbody>
</table>

6.3.1.3 Model Summary

The intention of developing the generalized RBP model was to investigate the effect of hydraulic characteristics with respect to the pump specific speed on the ability of an RBP to generate pulsatile haemodynamic waveforms. The model was constructed to allow a
reasonable comparison of the pump performance under similar basic conditions, thus aims to isolate the effect of specific speed and the corresponding HQ-curve steepness, while other influential parameters were held constant within a reasonable framework. The hydraulic characteristics for different specific speeds were estimated and constructed from typical pump data, which were extracted from literature. However, characteristics of real pumps corresponding to the investigated specific speeds may be substantially different, as they are influenced by various geometrical parameters. Further, several assumptions were applied for the corresponding model of the pump motor. For these reasons, the model consequently does not allow a generalized conclusion that any RBP from a specific category are more suitable for rapid speed modulation than others. However, it is expected, that trends with respect to increase or decrease of performance with specific speed become apparent, when simulated results for the modelled theoretical pumps are compared.

Table 6-4 – Parameter definition for the generalized pump model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q_{BEP}$</td>
<td>5</td>
<td>L/min</td>
</tr>
<tr>
<td>$H_{BEP}$</td>
<td>90</td>
<td>mmHg</td>
</tr>
<tr>
<td>$\eta_{BEP}$</td>
<td>30</td>
<td>%</td>
</tr>
<tr>
<td>$L_{RBP}$</td>
<td>0.31</td>
<td>mmHg · s²/mL</td>
</tr>
<tr>
<td>$R_{mot}$</td>
<td>1</td>
<td>Ω</td>
</tr>
<tr>
<td>$\eta_{mot}$</td>
<td>70</td>
<td>%</td>
</tr>
<tr>
<td>$P_{mech,max}$</td>
<td>45</td>
<td>W</td>
</tr>
<tr>
<td>$I_{rot,20}$</td>
<td>$5.5 \cdot 10^{-6}$</td>
<td>kg · m²</td>
</tr>
<tr>
<td>$I_{mot,limit}$</td>
<td>±4.5</td>
<td>A</td>
</tr>
</tbody>
</table>

6.3.2 Influence of RBP Characteristics on the Performance with Rapid Speed Modulation

6.3.2.1 Comparison of the Modelled Pumps

Based on the generalized RBP model introduced in section 6.3.1, the relative influences of hydraulic and motor characteristics on the device performance in pulsatile operation were investigated. First, six different modelled pumps, corresponding to the specific speeds $n_q = 20, 40, 60, 80, 100$ were evaluated for pulse amplitudes ranging from zero (continuous flow) to their respective maximum output, as limited by the maximum motor torque and speed. For all simulations, the pulse rate was set to $PR = 60 \text{ bpm}$, while the mean systemic flow rate
was $\tilde{Q}_s = 5 \text{ L/min}$. The parameters of the cardiovascular system model remained unchanged from the parameters introduced in the previous chapter. Consequently, in the continuous flow mode, all simulated pumps operated at the defined best efficiency point ($H_{BEP} = 90 \text{ mmHg}; Q_{BEP} = 5 \text{ L/min}$) and hence required the same motor power ($P_{el,CF} = 4.76 \text{ W}$). The corresponding NLP was solved for objective function weighting factors of $w_{obj} = 0\%$, 100\%, and the corresponding maximum $SHE$ and $dP/dt$ were evaluated. The target power consumption ($\tilde{P}_{el}^*$) was set to values of 5.5 W, 6 W, and 7.5 W, and was subsequently increased in steps of 2.5 W, until the speed and torque limits were exploited. Maximum generated pulsatility was then compared between the modelled pumps, and trends for the generated pulsatility with respect to specific speed were identified and discussed.

**6.3.2.2 Sensitivity to Parameter Changes**

Due to the applied assumptions for the modelled pumps, a generalized conclusion with respect to the suitability of pump types for rapid speed modulation is only possible to a limited extent. For a specific RBP design, parameters such as the hydraulic and motor efficiencies and the rotor inertia may be substantially different from the here assumed values. To understand how changes in these design characteristics affect the pump performance, the relative sensitivities of the pump performance with respect to changes of motor parameters and hydraulic efficiency were evaluated. For this purpose, a set of sensitivity factors $k_{s,i}$ was introduced, which was used for an isolated, stepwise adjustment of parameters under investigation. With their corresponding sensitivity factor, the evaluated parameters were redefined as per equation (6.11). The initially used parameters and the corresponding solutions were denoted with the subscript 0 (zero), and were replaced by the corresponding adjusted parameters:

\[
\begin{align*}
K_{t,adj} &= k_{s,T} \cdot K_{t,0} & \text{Motor torque constant} \\
J_{rot,adj} &= k_{s,J} \cdot J_{rot,0} & \text{Rotor inertia} \\
R_{mot,adj} &= k_{s,R} \cdot R_{mot,0} & \text{Motor resistance} \\
\eta_{hyd,adj} &= k_{s,\eta} \cdot \eta_{hyd,0} & \text{Hydraulic efficiency.}
\end{align*}
\]

Changes in the hydraulic efficiency were implemented by the corresponding change of the hydraulic load torque, which is proportional to the reciprocal of the sensitivity factor ($T_L \sim k_{s,\eta}$). For each parameter variation, the absolute values of $SHE(k_{s,i})$ and $dP/dt(k_{s,i})$, and the deviation from the initial value ($k_{s,i} = 100\%$) were evaluated:
The corresponding sensitivities of $SHE$ and $\frac{dP}{dt}$ to changes of the tested parameters were defined as the partial derivative of each quantity with respect to $k_{s,i}$:

$$S_{SHE}(k_{s,i}) = S_{SHE,i} = \left| \frac{\partial}{\partial k_{s,i}} SHE(k_{s,i}) \right| \approx \left| \frac{\Delta SHE}{\Delta k_{s,i}} \right|$$

$$S_{dP/dt}(k_{s,i}) = S_{dP/dt,i} = \left| \frac{\partial}{\partial k_{s,i}} \frac{dP}{dt}(k_{s,i}) \right| \approx \left| \frac{\Delta dP/dt}{\Delta k_{s,i}} \right|.$$  

(6.13)

The sensitivities therefore represented the change of $SHE$ or $dP/dt$ corresponding to a percentage change of the tested parameter and thus allowed a comparison of the relative influence of all parameters. Each sensitivity parameter was separately, systematically adjusted for values of

$$k_{s,i} = [75\%, 85\%, 95\%, 100\%, 105\%, 115\%, 125\%].$$

All sensitivity-values for were calculated in the mid-points between two evaluated values of $k_{s,i}$ and in the respective initial solution $S(k_{s,i} = 100\%)$ utilizing the solutions for $k_{s,i} = 95\%$ and $k_{s,i} = 105\%$. The units are accordingly

$$[S_{SHE,i}] = [\text{erg/cm}^3 / \%]$$

$$[S_{dP/dt,i}] = [\text{mmHg/s / \%}]$$  

(6.14)

The parameter variation was performed on the example of the generalized RBP mode, with the pump models corresponding to specific speeds of $n_q = 20$ and $n_q = 100$. All scenarios were optimized with at two power consumption limits ($\bar{P}_{el} = 7.5\, W$ and $\bar{P}_{el} = 12.5\, W$), whereas all discussed values of $SHE$ correspond to solutions optimized with $w_{obj} = 100\%$, and values of $dP/dt$ correspond to solutions optimized with $w_{obj} = 0\%$. All other model parameters remained constant with respect to the previous pump model comparison.
Therefore, solutions with $k_{s,l} = 100\%$ are identical to the respective solutions obtained in the previous analysis.

### 6.4 Results and Discussion

#### 6.4.1 Comparison of the Simulated Pump Models

Figure 6-10 shows the maximum values of SHE and $dP/dt$ corresponding to NLP solutions for all modelled pumps with specific speeds between $n_q = 20$ and $n_q = 250$ ($n_{q20}$, $n_{q250}$) as a function of the power consumption. The results show a consistent trend of both increasing $dP/dt$ and SHE with decreasing specific speed for all evaluated pumps at all power consumption limits. The highest $dP/dt$ was generated by $n_{q20}$, with a maximum value of 402 mmHg/s (Figure 6-10A). Conversely, the lowest $dP/dt$ levels were generated by the pump with the highest specific speed ($n_{q250}$, which was limited to a maximum $dP/dt$ of 321.4 mmHg/s. Similarly, the lowest and highest maximum SHE were generated by the same two pumps.

![Diagram](image)

**Figure 6-10 – Maximum values of pulsatility metrics ((A) $dP/dt$ and (B) SHE) corresponding to the modelled pumps with different specific speeds as function of power consumption ($\bar{Q}_s = 5$ L/min, $PR = 60$ bpm).**

SHE generated by $n_{q20}$ was substantially larger, with a highest maximum SHE of 11,696 erg/cm$^3$, which was 72% higher than the maximum SHE generated by
The maximum values for the intermediate specific speed pumps progressively decreased with increasing \(n_q\). It should further be noticed, that the power consumption, at which the maximum pulse amplitude was generated, increased with decreasing specific speed. Similar to the previously evaluated NLP solutions (chapter 0) the SHE as function of the power consumption asymptotically approached the maximum value, when the speed amplitude was limited by the maximum pump speed. Accordingly, Figure 6-10B shows, that the speed limitation affected pumps with higher specific speeds at lower power consumptions. Figure 6-11 shows an example of the normalized, optimized speed profiles of all modelled pumps at a mean power consumption of \(\bar{P}_e^* = 12.5\, W\). All profiles were scaled to the respective maximum speed. The speed amplitudes substantially increased with increasing specific speed. While \(n_{q_{100}}\) and \(n_{q_{250}}\) reached the speed limit, the required maximum speed of \(n_{q_{20}}\) is only \(0.81 \cdot n_{\text{max}}\). Consequently, the speed-reserve is higher, allowing the pump to further increase the power consumption and corresponding pulsatile output. Similarly, the minimum diastolic speed decreased with increasing \(n_q\).

Figure 6-11 – Speed profiles normalised to their respective maximum speed for all modelled pumps at a mean power consumption of \(\bar{P}_e^* = 12.5\, W\).

Figure 6-12 shows the normalized maximum values of speeds and torques for all pumps and power consumptions. Similarly, the results show, that pumps with increasing specific speeds reached the speed limits at lower power consumptions when optimized for both \(w_{\text{obj}} = 0\%\).
(A) and \( w_{obj} = 100\% \) (B). The corresponding maximum torques required to maximize \( SHE \) (Figure 6-12A, bottom) increased approximately linear with increasing power consumption, until the maximum speed limits were exploited. With further increasing power consumption, the maximum torques increased more steeply, until the torque limit was exploited as well. Conversely, when the speed profiles were optimized for \( dP/dt \), maximum torques were exploited at a low power consumption of \( \bar{P}_{el} = 6 \ W \) for all pumps. Again, the corresponding maximum speeds were exploited at lower power consumption limits with increasing specific speed.

Figure 6-12 – Normalized maximum speeds and torques during the pulse cycle for each pump at the evaluated power consumption, when optimized for (A) \( SHE \) \( (w_{obj} = 0\%) \) and (B) \( dP/dt \) \( (w_{obj} = 100\%) \).

The results show, that pumps with increasing specific speeds were not only limited by the maximum speed, but also by the maximum motor torque capacity. For example, at a power consumption limit of \( \bar{P}_{el} = 7.5 \ W \), all pumps exploited the maximum available torque during the pulse cycle (Figure 6-12B, bottom), but none of the pumps reached the maximum speed (Figure 6-12B, top). However, the corresponding maximum \( dP/dt \) values decreased with increasing specific speed (Figure 6-10B).
The difference of the speed and torque requirements for the different specific speed pumps is caused by the gradient of the pressure head-flow (HQ) characteristics of the pumps, which are progressively steeper with increasing specific speed. While all pumps, independent of the HQ-characteristics, can theoretically follow the same operating point trajectory, a significantly lower speed change relative to the continuous flow speed \(n_{BEP}\) was required, with a flat HQ-characteristic. That is, as relative flow rate changes from the minimum to the maximum flow rate occurred substantially faster than the corresponding changes in the pressure head. Consequently, the simulated pumps intermittently operated at a comparatively high flow rate and low pressure head, which is in favour of a pump with a flat HQ-characteristic.

Figure 6-13 illustrates exemplary HQ-curves for pumps with \(n_q = 20, 100\). The solid pressure head-flow loops show the trajectories of the pump operating points for the optimized pulse waveforms at \(\bar{P}_{el}^* = 10\ W\). The dashed and dash-dotted lines represent the steady state HQ-characteristics at \(n = n_{BEP}\) and \(n = 1.5 \cdot n_{BEP}\). BEP, best efficiency point.

Figure 6-13 illustrates exemplary HQ-curves for pumps with a low and a high specific speed \((n_{q20}, n_{q100})\). The solid pressure head-flow loops show the trajectories of the pump operating points for the optimized pulse waveforms at \(\bar{P}_{el}^* = 10\ W\). The dashed and dash-dotted lines represent the steady state HQ-characteristics at the BEP speeds \(n = n_{BEP}\) and the 1.5-fold of the BEP speeds \(n = 1.5 \cdot n_{BEP}\). It is obvious, that the maximum steady-state flow rates at comparable relative speeds in the operating range \((H_{RBP} = 70 – 110\ mmHg)\) are
substantially higher for \( n_{q20} \) compared to \( n_{q100} \). Consequently, the relative speed increase during systole is significantly higher for pump \( n_{q100} \), while the maximum flow rate in the optimized dynamic trajectory is lower.

### 6.4.2 Investigation of the Sensitivity to Motor Characteristics and Efficiency

The effects of changes in the torque constant for \( n_{q20} \) and \( n_{q100} \) at power consumption limits of 7.5 \( W \) and 12.5 \( W \) are shown in Figure 6-14.

#### Sensitivity to changes in the motor torque constant

![Figure 6-14](image)

Figure 6-14 – Sensitivity to changes in the motor torque constant: changes of (A) \( SHE \) and (D) \( dP/dt \) for \( n_{q20} \) and \( n_{q100} \) at power consumption limits of 7.5 \( W \) and 12.5 \( W \), (B,E) difference to the initial solution, and (C,F) the calculated sensitivities.

Variation of the sensitivity factor \( k_{S,T} \) was positively related to both \( SHE \) and \( dP/dt \), i.e. an increase of \( k_{S,T} \) and consequently the torque constant resulted in an increase of both metrics (Figure 6-14 A,D). The sensitivity of \( SHE \) was slightly higher for \( n_{q20} \) compared to \( n_{q100} \), while the sensitivity was higher at the 12.5 \( W \) limit compared to the 7.5 \( W \) limit for both pumps. Over the change of \( k_{S,T} \) between 75\% and 125\%, \( S_{SHE,T} \) decreased asymptotically for all simulated scenarios. The sensitivity-decrease for \( n_{q100} \) at 12.5 \( W \) was steeper for \( k_{S,T} > 100\% \) compared to the other scenarios, as the solutions were increasingly influenced by the maximum pump speed limit with growing \( k_{S,T} \). \( SHE \)-Sensitivities for all scenarios were between 49.07 \( \text{erg/cm}^3/\% \) and 66.82 \( \text{erg/cm}^3/\% \) in the initial solution (\( k_{S,T} = 100\% \)). The sensitivity of \( dP/dt \) was higher for \( n_{q20} \), whereas the sensitivities for the two different power
consumption limits for the same pump only differed slightly at values of $k_{s,T} < 90\%$ for both $n_{q20}$ and $n_{q100}$, ranging between $3.04 \text{ mmHg/s} / \%$ and $4.17 \text{ mmHg/s} / \%$.

The trends for changes in the rotor inertia sensitivity parameter $k_{s,J}$ are shown in Figure 6-15.

Contrary to the torque constant, increasing $k_{s,J}$ negatively affected both $SHE$ and $dP/dt$, resulting in a decrease of both quantities. The absolute change of the generated $dP/dt$ varied between approximately $+60 \text{ mmHg/s}$ and $-45 \text{ mmHg/s}$ for values of $k_{s,J} = 75\%$ and $k_{s,J} = 125\%$ respectively, and was almost identical for both pumps at both power consumption limits (Figure 6-15 E). Due to the definition of the sensitivity as an absolute value (section 6.3.2.2), the calculated values were all positive, and were between $1.59 \text{ mmHg/s} / \%$ and $2.79 \text{ mmHg/s} / \%$. The absolute values of $SHE$ were almost independent of $k_{s,J}$, with a maximum sensitivity of $4.48 \text{ erg/cm}^3 / \%$, which was significantly lower than the sensitivity to the torque constant (Figure 6-14 C; Figure 6-15 C). It can further be observed, that a decrease of the rotor inertia of approximately 25\% is required for $n_{q100}$ to generate a comparable $dP/dt$ to $n_{q20}$ in the initial solution, while the corresponding increase in $SHE$ is marginal.

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Graphs corresponding to changes in motor phase resistance \((k_{s,R})\) and hydraulic efficiency \((k_{s,\eta})\) are shown in Figure 6-16 and Figure 6-17 respectively. Naturally, increasing the phase resistance had a negative effect on SHE, decreasing the maximum by up to \(767.2\ erg/cm^3\), when the phase resistance was increased by 25\% \((k_{s,R} = 125\%)\). Similarly, a 25\% decrease of the resistance resulted in an increase of up to \(920.8\ erg/cm^3\). Both pumps showed similar sensitivity of SHE, which decreased with increasing \(k_{s,R}\) and was slightly higher at 12.5 W power consumption. Further, the sensitivity was generally slightly higher for \(n_{q20}\), compared to \(n_{q100}\). At low values of \(k_{s,R}\), sensitivity decreased for \(n_{q100}\) at \(\overline{P}_{el}^* = 12.5\ W\), as the pulse profile was limited by the maximum motor speed (Figure 6-16 C). None of the evaluated scenarios showed a significant influence of the phase resistance on \(dP/dt\), as the optimized values marginally changed, and all sensitivity values were below \(0.06\ \frac{mmHg}{s}/\%\) (Figure 6-16 D-F).

![Sensitivity to changes in the motor phase resistance](image)

Figure 6-16 – Sensitivity to changes in the motor phase resistance: changes of (A) SHE and (D) \(dP/dt\) for \(n_{q20}\) and \(n_{q100}\) at power consumption limits of 7.5 W and 12.5 W, (B,E) difference to the initial solution, and (C,F) the calculated sensitivities.

The hydraulic efficiency strongly affected both SHE and \(dP/dt\) (Figure 6-17), resulting in an absolute change of SHE of up to \(3152\ erg/cm^3\). \(S_{SHE,\eta}\) was comparable for all scenarios, while it was slightly lower for \(n_{q100}\) compared to \(n_{q20}\) for both power consumption limits. At higher values of \(k_{s,\eta}\), the pulse profiles of \(n_{q100}\) were again limited by the maximum speed.
at 12.5 W, resulting in a decreasing sensitivity of SHE. A decrease of \( \eta_{\text{hyd}} \) by 25% \((k_{s,\eta} = 75\%)\) resulted in a strong decrease of the \( dP/dt \) generated by both pumps at a power consumption of 7.5 W, as the decrease in hydraulic efficiency resulted in an increase of the continuous flow motor power requirement to \( \bar{P}_{el,CF} = 6.35 \) W. For all other evaluated solutions, \( S_{dP/dt} \) was comparable for both power consumption limits, while slightly higher for \( n_{q20} \).

Sensitivity to changes in the hydraulic efficiency

![Diagram showing sensitivity to changes in hydraulic efficiency](image)

Figure 6-17 – Sensitivity to changes in the hydraulic efficiency: changes of (A) SHE and (D) \( dP/dt \) for \( n_{q20} \) and \( n_{q100} \) at power consumption limits of 7.5 W and 12.5 W, (B,E) difference to the initial solution, and (C,F) the calculated sensitivities.

A comparison of the sensitivities of SHE and \( dP/dt \) to all evaluated parameters in the initial solutions \( S(k_{s,I} = 100\%) \) is shown in Figure 6-18. The graph shows, that hydraulic efficiency \((k_{s,\eta})\) had the highest influence on SHE \((89.21 - 108.65 \text{ erg/cm}^3/\%\)) for all evaluated scenarios, followed by the motor torque constant \((53.43 - 66.82 \text{ erg/cm}^3/\%\)). The rotor inertia \((k_{s,J})\) had the least influence on SHE. Conversely, it strongly influenced \( dP/dt \), which was more sensitive to only the torque constant \((k_{s,T})\). The motor resistance \((k_{s,R})\) had a relatively low influence on SHE, while the maximum \( dP/dt \) was almost not affected by changes of the parameter at all. The numerical values corresponding to Figure 6-18 are given in Table 6-5.
Figure 6-18 – Summary of the sensitivities of (A) SHE and (B) dP/dt to changes in the sensitivity parameters altering the motor torque constant \( (k_{s,T}) \), rotor inertia \( (k_{s,J}) \), motor phase resistance \( (k_{s,R}) \) and hydraulic efficiency \( (k_{s,\eta}) \). The evaluated pumps were \( n_{q20} \) and \( n_{q100} \) at power consumptions limits of \( \bar{P}^*_{el} = 7.5 \) W and \( \bar{P}^*_{el} = 12.5 \) W. All data shown are sensitivities in the initial solution \( (k_{s,I} = 100\%) \).

Table 6-5 – Sensitivities of SHE and dP/dt to changes in the sensitivity parameters altering the motor torque constant \( (k_{s,T}) \), rotor inertia \( (k_{s,J}) \), motor phase resistance \( (k_{s,R}) \) and hydraulic efficiency \( (k_{s,\eta}) \). Simulated values for \( n_{q20} \) and \( n_{q100} \) at power consumptions of \( \bar{P}^*_{el} = 7.5 \) W and \( \bar{P}^*_{el} = 12.5 \) W. All data shown are sensitivities in the initial solution \( (k_{s,I} = 100\%) \).

<table>
<thead>
<tr>
<th>Sensitivity</th>
<th>Parameter</th>
<th>( n_q = 20 )</th>
<th>( n_q = 100 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( S_{SHE} )</td>
<td>( k_{s,T} )</td>
<td>59.13</td>
<td>66.82</td>
</tr>
<tr>
<td></td>
<td>( k_{s,J} )</td>
<td>0.94</td>
<td>2.85</td>
</tr>
<tr>
<td></td>
<td>( k_{s,R} )</td>
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<td>33.38</td>
</tr>
<tr>
<td></td>
<td>( k_{s,\eta} )</td>
<td>108.65</td>
<td>105.32</td>
</tr>
<tr>
<td>( S_{dP/dt} )</td>
<td>( k_{s,T} )</td>
<td>3.487</td>
<td>3.441</td>
</tr>
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<td></td>
<td>( k_{s,J} )</td>
<td>1.904</td>
<td>1.857</td>
</tr>
<tr>
<td></td>
<td>( k_{s,R} )</td>
<td>0.051</td>
<td>0.014</td>
</tr>
<tr>
<td></td>
<td>( k_{s,\eta} )</td>
<td>1.698</td>
<td>1.605</td>
</tr>
</tbody>
</table>
6.5 Summary and Discussion

The comparison of pump characteristics corresponding to various specific speeds showed a general deterioration of the maximum generated pulsatility in terms of both SHE and \( \frac{dP}{dt} \) with increasing specific speed. As was shown in Figure 6-13, an identical relative speed increase results in a higher flow rate for \( n_{q20} \), as the corresponding HQ-curve is flatter. Therefore, the lower pulsatility of higher specific speed pumps shown in the simulation results, was ultimately not caused by the higher operating speeds, but by the characteristically steeper HQ-curves, which resulted in the requirement of a larger speed amplitude for the same instantaneous flow output. Consequently, pumps with lower specific speed were able to generate higher instantaneous flow rates, thus could deliver the same hydraulic energy in a shorter time interval, reach higher pressure and flow rate time gradients \( (dP/dt, dQ/dt) \), which resulted in increased pulsatility.

The results presented here are based on a generalized RBP model. For the implementation of the model, a range of assumptions and simplifications were necessary, which aimed to equalize the basic conditions of all modelled pumps and allow a generalized comparison of pumps exhibiting different HQ-curve steepness. Consequently, the findings must be viewed in the context of these assumptions. The results substantiated the notion, that flat HQ-characteristics are favourable for pulsatile operation of rotary total artificial heart pumps. However, while reasonable assumptions for the motor characteristics, and rotor inertia were made, a generalization to all types of RBP is inadmissible. In reality, the chosen model parameters are significantly influenced by manifold design choices. As was shown, the pulsatile pump performance can significantly vary with parameters such as the maximum hydraulic efficiency, rotor inertia, and motor torque constant and resistance, and maximum speed. For example, it was shown that a 25% lower rotor inertia of \( n_{q100} \) can led to a similar maximum \( dP/dt \) as that of \( n_{q20} \). Further, frequency dependent motor losses, such as eddy current and iron hysteresis losses were neglected, but – depending on the motor design – may have significant influence on the motor current requirement and torque generation. Hence, depending on the design of the hydraulics, and the location of the best efficiency point, a high specific speed RBP may exhibit a flatter HQ-curve, higher motor torque or efficiency, hydraulic efficiency, or combination thereof, resulting in higher pulsing performance compared to a low specific speed RBP. Nevertheless, it can be concluded, that compared to
axial flow pumps the generation of similar or higher pulsatility with centrifugal pumps is possible, despite their associated substantially larger rotor inertia.

The results indicated that, for similar pump designs or different iterations of the same pump, design choices leading to a flatter HQ-curve can benefit the performance with rapid speed modulation. Further, a hydraulic best efficiency point towards a higher flow rate may increase the pump efficiency at the maximum systolic flow rate, and therefore decrease power consumption and/or improve pulsatility. It should be noted that, compared to the pump characteristics evaluated here, significantly flatter HQ-curves are technically possible.

However, the potential advantages of such design decisions need to be carefully weighed against the corresponding effects on other determining parameters of the hydraulic and motor system, and the rotor inertia. To gain a better understanding of the effects of changes of those parameters, the relative influence of selected parameters was evaluated. The results may be applied to estimate changes of the pump performance in response to design changes, and help with design choices, which typically influence more than one of the tested parameters. For example, an increase in the rotor permanent magnet size or grade can increase the torque constant, however, will simultaneously increase the rotor inertia, whereas the performance improvement due to higher torque may outweigh the performance loss due to the higher inertia, if the relative increase of the inertia is comparatively low. Similarly, changes in the motor winding turn number may affect the torque constant and phase resistance, which in turn affects the maximum motor speed.

The results of the sensitivity investigation allow general insight of the relative effects of the varied parameters. The most influential parameters were the motor torque constant and the hydraulic efficiency. Hydraulic efficiency affects both the power loss in the hydraulic system and the motor. A lower efficiency imposes a greater load to the motor, hence affects maximum rotor acceleration and increases the motor current requirement and consequently copper losses. Hence, the required power consumption to generate similar SHE levels is increased, while the maximum $dP/dt$ is reduced. Similarly, the torque constant influences both motor efficiency and acceleration. Conversely, the motor resistance does not significantly affect the acceleration and consequently $dP/dt$, which is further consistent with
the finding, that \( dP/dt \) for all evaluated pumps reached its maximum at a low power consumption limit and changed insignificantly to a further increase of the limit.

However, it is important to note that the sensitivity may change depending on the maximum speed and torque limits imposed to the pump drive. It is therefore important to understand the limiting factors of a specific pump and drive system, as the limitation of maximum speed may for example diminish a possible gain due to a higher torque capacity, and vice versa. In this context, the results for the generalized RBP model showed a close dependency of the maximum \( dP/dt \) on the maximum torque, while \( SHE \) was more significantly influenced by the maximum speed limit. However, depending on specific design limitations, this observation may not hold true for all pumps. Consequently, when a RBP is designed for pulsatile operation, the torque and efficiency of the hydraulic and motor systems should be carefully evaluated over a wide operating range, to determine an appropriate ratio of maximum motor torque and speed and choose an appropriate winding accordingly. However, naturally, such choices must further be influenced by considerations concerning for example the bearing capacity and blood-compatibility of a RBP.

6.6 Limitations and Future Work

The limitations of this study included the assumptions made with respect to hydraulic characteristics, rotor inertia and motor models for the evaluation of modelled pumps corresponding to different specific speeds. It was shown that, despite the increasing rotor inertia, pumps with lower specific speeds generated higher pulsatility. A broad comparison of models corresponding to measured characteristics of various real devices may therefore substantiate the analysis presented here. However, the main objective of the comparison of the modelled pumps was to evaluate the effect of the pressure head-flow gradients corresponding to the pumps. Here, it was shown, that pumps exhibiting steep HQ-characteristics are more prone to exploit the maximum motor torque and speed, compared to pumps with flat HQ-curves. Specifically, when a change in the hydraulic characteristics of a specific impeller is concerned, a flatter HQ-curve may therefore have significant advantages. It should be noted that, depending on a specific device and motor design, the factors limiting the maximum pulsatility and the corresponding sensitivities to changes of motor and hydraulic parameters may differ from the presented findings. However, the results obtained for the high and low specific speed pumps \((n_{q20}; n_{q100})\) indicated, that the relative influence
is similar for different pump types, and the methodology can analogously be applied to any type of RBP to evaluate the device-specific limitations.

6.7 Conclusion

This chapter aimed to investigate RBP design and control characteristics with respect to their influence on the ability to generate cardiovascular pulsatility with rapid impeller speed modulation. The analysis was based on the previously developed and validated nonlinear optimization framework. The results indicated that a flat HQ characteristic is beneficial for rapid speed modulation, while the various other parameters such as the motor torque, rotor inertia and hydraulic efficiency can have substantial influence of the device performance. The presented results may assist in the weighing of design choices for RBP devices under development, when pulsatile device operation is intended, to ultimately lead to a device design capable of generating truly physiologic pulsatility. The presented work further provided a basis for future in vitro and in vivo evaluations to determine the blood compatibility and physiologic effects corresponding to the evaluated pulse profiles.
7 Conclusions

The primary aim of the thesis was to investigate and identify favourable design characteristics and control strategies for the motor and hydraulics of a RBP, which, when combined, may allow the future development of a RBP, which can autonomously and efficiently generate physiologic arterial pulsatility. As outlined in the introduction, the main objectives to achieve this aim were:

1. Investigate axial flux motor geometries to improve motor performance and cater for desired hydraulic characteristics that improve device performance with rapid speed modulation.
2. Evaluate the influence of different speed profile shapes on the generated haemodynamic pulsatility and required motor power consumption.
3. Develop a numerical framework to investigate and optimize speed profiles for rapid speed modulation of different rotary blood pump (RBP) types in a total artificial heart (TAH) setting.
4. Investigate the influence of RBP design characteristics on the ability of the device to generate pulsatile haemodynamics.

Furthermore, after design limitations and challenges with respect to the development of pulsatile RBP were established, the main hypothesis for this research was formulated (section 2.5). This hypothesis was that the ability of a RBP to generate pulsatile outflow can be substantially improved by means of design and control; specifically, that a flat pressure head-flow characteristic is beneficial for the device performance with rapid speed modulation, and that the shape of the applied speed profile has substantial influence on the generated pulsatility.

The first objective of the thesis was stated with consideration of this hypothesis, in particular with respect to the potential axial flux motor design requirements of large axial gaps and large inner radii, which may allow to incorporate an improved hydraulic design exhibiting a flat HQ-characteristic and increased efficiency. On the example of the device geometry of the BiVACOR TAH, a design methodology for axial flux motor drives to cater for these requirements was presented, where the effects of different geometry parameters on the drive efficiency, axial attractive forces, and rotor inertia were evaluated and combined to reach a
specified design objective. It was found, that a performance reduction due to increased axial gap length can, to some extent, be compensated with an increase of the permanent magnet thickness. However, the concomitant increase of the rotor weight and inertia adversely affects the maximum rotor acceleration and may thus prove to be detrimental to the design. Suggested alternative measures to improve efficiency with a reduced required permanent magnet volume included to increase the axial slot depth and optimise the ratio of permanent magnet angle and thickness. Conversely, it was found that the effects of increasing the inner radius could be offset without substantially affecting the rotor inertia, whereas the sacrifice of increased axial lengths of the rotor and stator were made. The findings suggested that both geometry adjustments can be reasonably incorporated in an axial flux motor design, if the requirement is justified by the performance increase of the hydraulic system. The presented methodology may further be applied to improve the motor efficiency or reduce the axial attractive force or rotor inertia of a given design.

With respect to the second objective, an in vitro study comparing the pulsatile output with six different speed profiles applied to the BiVACOR TAH was performed. The results showed that the motor power requirements, as well as the generated $dP/dt$, $SHE$, and $PPI$ substantially differed between the evaluated profiles. It was found that, at the same pulse pressure, profiles exhibiting a longer relative systolic duration tended to generate higher $SHE$, while $PPI$ and $dP/dt$ levels were higher for profiles with relatively shorter systolic speed pulses with a higher amplitude. The results strongly suggested, that optimal speed profiles may exist, which maximise the generated pulsatility with minimal motor power consumption. As such, the stated hypothesis that the shape of the applied speed profile has substantial influence on the generated pulsatility was confirmed. The results further suggested, that the objectives to increase $SHE$ and $dP/dt$ may be competing with respect to an optimal speed profile shape.

Building on the findings of the in vitro evaluation, a numerical framework was developed, which served to numerically optimise the speed trajectory with respect to the maximum $dP/dt$ and/or surplus haemodynamic energy ($SHE$) for a specified mean pump outflow and maximum motor power consumption limit; it thus met the criteria for the third objective of this research work. The in vitro model validation and a preliminary in vivo experiment with the BiVACOR TAH demonstrated the efficacy of the numerical approach. The evaluation of
the introduced weighted objective function showed on the example of the HeartMate II device, that the optimisation results for a hybrid objective to maximise $dP/dt$ and $SHE$ may be best suited to be developed into a control strategy of a pulsatile rotary TAH in the future. The numerical framework was further applied to determine the operating envelope of a rotary blood pump with respect to the maximum speed amplitude and the maximum generated haemodynamic pulsatility as imposed by the limitations of the electronic circuitry of the motor drive. The implemented nonlinear program was therefore established as a valuable research tool to evaluate hydraulic and motor performance characteristics with respect to their suitability for the application in a pulsatile rotary blood pump.

Consequently, this tool was applied to meet the fourth objective in a subsequent comparison of modelled pump characteristics. For this purpose, a generalised model of RBPs with six different specific speeds, which exhibited different gradients of the pressure head-flow characteristic (HQ-curve), was developed, and the modelled pumps were compared with respect to their performance with rapid speed modulation. The results strongly suggested, that pumps with HQ-curves with high pressure sensitivity (flat) show superior performance to those with a steep HQ-curve. It was found that both the maximum generated $dP/dt$ and $SHE$ monotonously increased with specific speed (and thus an increasingly flat HQ-curve), as the required relative speed change to generate the same pulsatile outflow is smaller with a flat curve. As such, the study results confirmed the main hypothesis of this thesis.

However, while this finding suggests, that RBPs with low specific speeds (centrifugal pumps) are better suited for rapid speed modulation despite their larger associated rotor inertia, the results must be considered in conjunction with the respective motor characteristics specific to the devices, when different pump types are compared. Nevertheless, the findings suggested that improvement of the pressure sensitivity of given pump may be particularly beneficial to the performance of a pulsatile RBP, when the motor system and rotor inertia are not affected by the corresponding design changes.

Furthermore, as discussed in chapter 3, adjustments to the motor geometry may allow to facilitate these design changes while improving the drive efficiency and/or rotor inertia. To consider their relative influence, the sensitivity of the maximum pulsatility to changes of these characteristics was evaluated for two of the modelled pumps. It was shown, that the motor torque constant and hydraulic efficiency were most influential with respect to the
generated maximum $dP/dt$ and $SHE$ respectively. The results may thus be used in conjunction with the findings of the motor geometry evaluation to guide design decisions to improve pulsatile output of a given RBP. For example, pump performance may be improved with an increase of the motor torque and hydraulic efficiency, even if the design change results in an increase of the rotor inertia. While a dedicated analysis specific to the considered RBP characteristics may be required to confirm the effect of design changes, the findings presented in this thesis provide a synopsis of how the device performance may be improved by design and control and may thus guide the development of a RBP, which can autonomously deliver a physiologic pulse in the future. In conclusion, the main research aim of this thesis – to investigate and identify favourable design characteristics and control strategies for the motor and hydraulics of a pulsatile RBP – was met by the presented studies.

### 7.1 Limitations and Future Work

The presented studies were to some extent limited by the assumptions and simplifications applied throughout the numerical modelling approach. However, the performed validation of the developed models showed good agreement with measurement results, hence despite the simplifications, the models were able to provide sufficient results to gain insight into the effects of parameter changes in both the FEM-motor model and the numerical optimal control environment. However, further refinement of both models may allow improvement of the accuracy of the results. Specifically, the developed quasi-3D FEM model may be improved in the future to allow a more accurate calculation of stator core losses, in particular when different types of core materials are considered. Further, the accuracy of the applied edge factors may be improved to include the effect of the interaction between the stator and rotor fields in the generation of axial attractive forces.

The comparison of pumps with different specific speeds was limited by the assumption made for the generalised model. While the objective of the modelling approach was to compare the modelled pumps under equal boundary conditions with respect to the motor characteristics and hydraulic efficiency. Consequently, this modelling approach does not consider potential differences in the achievable pressure sensitivity and hydraulic or motor efficiencies with specific types of RBP, such as centrifugal, mixed flow, or axial flow devices. Therefore, despite the typically flatter HQ-characteristics associated with centrifugal RBP, the findings of this study do not admit the conclusion, that these pumps are generally better suited for the
application in a pulsatile RBP. However, similar studies comparing the performance characteristics of specific devices may be performed with the developed numerical framework in the future. Additionally, the optimal control simulation environment may be improved with more accurate models of the motor, pump hydraulics, and cardiovascular system. Furthermore, an investigation of the effect of changes in the parameters describing the arterial and atrial impedance should be performed to evaluate how the obtained optimised speed profiles are affected by the state of the cardiovascular system. Findings of such an extended study may further be used to develop a physiological control strategy for the long-term application in a pulsatile rotary total artificial heart controller.

The most important area of future work arising from this thesis is the consideration of biocompatibility of the proposed rapid speed modulation modes. While the in vivo results showed that a near-physiological pulse was generated with the BiVACOR TAH, the effect of rapid speed modulation on blood damage was not considered in this study. Consequently, before the implementation of such operating modes are considered in long time RBP support, thorough investigations of the effect on vWF cleavage, haemolysis, platelet activation and white blood cell count must be performed. Furthermore, while a rotary TAH designed to replicate native arterial pulsatility may be designed, the benefits of the generated waveforms on the mammalian physiology are not clear. However, such a device would greatly benefit the research field, as it would allow detailed long-term studies to evaluate biocompatibility with truly pulsatile and nonpulsatile perfusion with the same device. Further, rapid speed modulation modes generating attenuated pulsatility may be evaluated to potentially find a level of pulsatility, which ensures maintenance of adequate levels of blood compatibility, while the induced pulse may enhance baroreceptor activity, improve microcirculatory perfusion, or reduce the incidence of arteriovenous malformations and/or gastrointestinal bleeding. As such, new opportunities for a wide area of future research arise from this work, when the findings are applied to the development of a pulsatile RBP and such a device progresses to preclinical trials.
8 Appendix A

This section summarises the state-space representation (equations (8.1) – (8.2)) of the implemented model of the systemic and pulmonary circulation (see section 5.2.1.1) showing a block diagram (Figure 8-1) and the corresponding system matrices ((8.3) – (8.9)).

\[
\dot{x}(t) = A \cdot x(t) + B \cdot u(t) \\
y(t) = C \cdot x(t) + D \cdot u(t)
\]

Figure 8-1 – Block diagram of the state-space model. \(u(t)\), input vector; \(x(t)\), state vector; \(y(t)\), output vector; \(A, B, C, D\), system matrices.

\[
u(t) = \begin{bmatrix} Q_{ao}, Q_{ra}, Q_{pa}, Q_{la} \end{bmatrix}^T
\]

\[
x(t) = \begin{bmatrix} Q_{ics}, P_{cao}, P_{sv}, P_{ra}, Q_{tcp}, P_{cpa}, P_{pv}, P_{la} \end{bmatrix}^T
\]

\[
y(t) = \begin{bmatrix} P_{ao}, P_{ra}, P_{pa}, P_{la} \end{bmatrix}^T
\]
\[
A = \begin{pmatrix}
\frac{-R_{cs}}{L_{cs}} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & -\frac{1}{C_{ao}R_{sa}} & \frac{1}{C_{ao}R_{sa}} & 0 & 0 & 0 & 0 & 0 \\
0 & \frac{1}{C_{sa}R_{sa}} & -\frac{1}{C_{sp}R_{sa}} & \frac{1}{C_{sp}R_{ra}} & 0 & 0 & 0 & 0 \\
0 & 0 & \frac{1}{C_{ra}R_{ra}} & -\frac{1}{C_{ra}R_{ra}} & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & -\frac{R_{cp}}{L_{cp}} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & -\frac{1}{C_{pa}R_{pa}} & \frac{1}{C_{pa}R_{pa}} & 0 \\
0 & 0 & 0 & 0 & 0 & -\frac{1}{C_{pa}R_{pa}} & \frac{1}{C_{pa}R_{pa}} & -\frac{1}{C_{la}R_{la}} \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & -\frac{1}{C_{la}R_{la}} \\
\end{pmatrix}
\] (8.6)

\[
B = \begin{pmatrix}
\frac{R_{cs}}{L_{cs}} & 0 & 0 & 0 \\
\frac{1}{C_{ao}} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & -\frac{1}{C_{ra}} & 0 & 0 \\
0 & 0 & \frac{R_{cp}}{L_{cp}} & 0 \\
0 & 0 & \frac{1}{C_{pa}} & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & 0 & -\frac{1}{C_{la}} \\
\end{pmatrix}
\] (8.7)

\[
C = \begin{pmatrix}
-\frac{R_{cs}}{L_{cs}} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & -\frac{R_{cp}}{L_{cp}} & 1 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\
\end{pmatrix}
\] (8.8)

\[
D = \begin{pmatrix}
\frac{R_{cs}}{L_{cs}} & 0 & 0 & 0 \\
0 & 0 & 0 & 0 \\
0 & 0 & R_{cp} & 0 \\
0 & 0 & 0 & 0 \\
\end{pmatrix}
\] (8.9)
9 Appendix B

This section summarises polynomial fits for the pressure head-flow and torque-flow characteristics of the modelled pumps \( n_{q20} - n_{q250} \) (see section 6.3.1). The pressure head and load torque as functions of the speed and flow rate are calculated according to equations (9.1) and (9.2). The polynomial coefficients are summarised in Table 9-1 and Table 9-2. Table 9-3 summarises the corresponding motor model parameters.

\[
H(n, Q) = n^2 \cdot \left( A_H \cdot \left( \frac{Q}{n} \right)^3 + B_H \cdot \left( \frac{Q}{n} \right)^2 + C_H \cdot \left( \frac{Q}{n} \right) + D_H \right) \\
T(n, Q) = n^2 \cdot \left( A_T \cdot \left( \frac{Q}{n} \right)^4 + B_T \cdot \left( \frac{Q}{n} \right)^3 + C_T \cdot \left( \frac{Q}{n} \right)^2 + D_T \cdot \left( \frac{Q}{n} \right) + E_T \right)
\]  

(9.1)  
(9.2)

Table 9-1 – Polynomial coefficients for pressure head-flow characteristics of the modelled pumps.

<table>
<thead>
<tr>
<th>( n_q )</th>
<th>( A_H )</th>
<th>( B_H )</th>
<th>( C_H )</th>
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<tbody>
<tr>
<td>20</td>
<td>(-1.6586 \cdot 10^2)</td>
<td>(-0.4539)</td>
<td>(7.0564 \cdot 10^{-5})</td>
<td>(1.8318 \cdot 10^{-5})</td>
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<tr>
<td>40</td>
<td>(-5.0826 \cdot 10^2)</td>
<td>(0.0819)</td>
<td>(-1.1723 \cdot 10^{-3})</td>
<td>(5.4443 \cdot 10^{-6})</td>
</tr>
<tr>
<td>60</td>
<td>(-1.3836 \cdot 10^3)</td>
<td>(0.3868)</td>
<td>(-1.1518 \cdot 10^{-3})</td>
<td>(2.7286 \cdot 10^{-6})</td>
</tr>
<tr>
<td>80</td>
<td>(-4.0663 \cdot 10^3)</td>
<td>(1.9802)</td>
<td>(-1.3259 \cdot 10^{-3})</td>
<td>(1.6521 \cdot 10^{-6})</td>
</tr>
<tr>
<td>100</td>
<td>(-9.0186 \cdot 10^3)</td>
<td>(4.4884)</td>
<td>(-1.5534 \cdot 10^{-3})</td>
<td>(1.1075 \cdot 10^{-6})</td>
</tr>
<tr>
<td>250</td>
<td>(-8.8646 \cdot 10^4)</td>
<td>(28.2542)</td>
<td>(-3.7221 \cdot 10^{-3})</td>
<td>(3.38295 \cdot 10^{-7})</td>
</tr>
</tbody>
</table>

Table 9-2 – Polynomial coefficients for torque-flow characteristics of the modelled pumps.

<table>
<thead>
<tr>
<th>( n_q )</th>
<th>( A_T )</th>
<th>( B_T )</th>
<th>( C_T )</th>
<th>( D_T )</th>
<th>( E_T )</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>0</td>
<td>(-0.03333)</td>
<td>(1.39889 \cdot 10^{-4})</td>
<td>(4.00549 \cdot 10^{-7})</td>
<td>(1.07137 \cdot 10^{-9})</td>
</tr>
<tr>
<td>40</td>
<td>0</td>
<td>(-0.02707)</td>
<td>(4.54701 \cdot 10^{-5})</td>
<td>(8.69386 \cdot 10^{-8})</td>
<td>(1.6639 \cdot 10^{-10})</td>
</tr>
<tr>
<td>60</td>
<td>0</td>
<td>(-0.02436)</td>
<td>(-7.81746 \cdot 10^{-6})</td>
<td>(5.24223 \cdot 10^{-8})</td>
<td>(5.6828 \cdot 10^{-11})</td>
</tr>
<tr>
<td>80</td>
<td>0</td>
<td>(-0.05737)</td>
<td>(7.59278 \cdot 10^{-6})</td>
<td>(2.52123 \cdot 10^{-8})</td>
<td>(2.7074 \cdot 10^{-11})</td>
</tr>
<tr>
<td>100</td>
<td>0</td>
<td>(-0.12601)</td>
<td>(4.51713 \cdot 10^{-5})</td>
<td>(7.70521 \cdot 10^{-9})</td>
<td>(1.5477 \cdot 10^{-11})</td>
</tr>
<tr>
<td>250</td>
<td>(1.544 \cdot 10^3)</td>
<td>(-1.41898)</td>
<td>(3.39152 \cdot 10^{-4})</td>
<td>(-3.5537 \cdot 10^{-8})</td>
<td>(2.9399 \cdot 10^{-12})</td>
</tr>
</tbody>
</table>

Table 9-3 – Motor characteristics of the modelled pumps.

<table>
<thead>
<tr>
<th>( n_q )</th>
<th>( n_{BEP} \text{ [rpm]} )</th>
<th>( K_t \text{ [Nm/A}_{\text{rms}}] }</th>
<th>( J_{rot} \text{ [10}^{-6}\text{ kg} \cdot \text{m}^2] }</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>2,439</td>
<td>0.018910</td>
<td>5.5</td>
</tr>
<tr>
<td>40</td>
<td>4,878</td>
<td>0.009455</td>
<td>1.32</td>
</tr>
<tr>
<td>60</td>
<td>7,318</td>
<td>0.006303</td>
<td>0.55</td>
</tr>
<tr>
<td>80</td>
<td>9,757</td>
<td>0.004728</td>
<td>0.296</td>
</tr>
<tr>
<td>100</td>
<td>12,196</td>
<td>0.003782</td>
<td>0.189</td>
</tr>
<tr>
<td>250</td>
<td>30,490</td>
<td>0.001513</td>
<td>0.026</td>
</tr>
</tbody>
</table>


10 Bibliography


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