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Development of a Variable Roller Pump and Evaluation of its Power Saving Potential as a Charge Pump in Hydrostatic Drivetrains

VYSOKÉ UČENÍ TECHNICKÉ V BRNĚ
FAKULTA STROJNÍHO INŽENÝRSTVÍ
ÚSTAV MECHANIKY TĚLES,
MECHATRONIKY A BIOMECHANIKY

Ing. Peter Zavadinka

DEVELOPMENT OF A VARIABLE ROLLER PUMP AND EVALUATION OF ITS POWER SAVING POTENTIAL AS A CHARGE PUMP IN HYDROSTATIC DRIVETRAINS

NÁVRH REGULAČNÍHO HYDROGENERÁTORU S VÁLEČKY V DRÁŽKÁCH ROTORU A OHODNOCENÍ JEHO POTENCIÁLU ÚSPORY VÝKONU V HYDROSTATICKÝCH POHONECH

Zkrácená verze Ph.D. Thesis

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1 MOTIVATION

The actual mobile machinery market indicates a lot of global trends but one trend is stable, the trend of focusing on ownership costs minimization. It is caused by many factors, from which two are the most important. The first one is simply fuel cost. The second factor is the exhaust regulations driven by government regulations. The European EURO IIIb and US TIER 4 regulations are already valid [1], [2].

Many works focused on the main drives of mobile machines, can be found (for example [3], [4]) but auxiliary drives could also contribute to higher efficiency of mobile machines, for example fan drive, steering and charge systems are powered by the combustion engine too. The fixed charge pump delivers in some conditions higher flow than is required. An application of a variable charge pump (VCP) into hydrostatic transmissions, which can act according to the system needs, could save additional energy and therefore increase efficiency.

The variable charge pump in the hydrostatic transmission (HST) has to have displacement variability, compact design, durability, pressure compensator control availability and low cost. Based on this, variable gear pump, variable vane pump and variable roller pump were considered for the variable charge pump application. The roller pump and the vane pump are very similar. Manufacturing costs and the durability of the roller pump prefers the variable roller charge pump instead of the variable vane pump.

The power saving potential of variable charge pump (VCP) is depending on the ratio of pump displacement variability, the bigger the ratio, the higher the potential of power savings. Due to the high variability of the displacement of a variable roller charge pump, which is for example higher than the standard variability of a variable gear pump [5], it was chosen for efficiency analyses in this work. Different power savings, filling and cooling demands could be expected for different working cycles and various drivetrain structures (vehicles). The variable roller charge pump has to ensure power savings with an adequate filling and cooling function for various working conditions and vehicle types. Due to this, suitability and power saving potential of a variable roller charge pump in the hydrostatic drivetrain have to be evaluated. The power saving potential is not only depending on the displacement variability but also on the pump efficiency. Thus the full power saving potential of a variable roller pump could be achieved with an optimization of the port plates design because the efficiency of a roller pump is determined by port plate design similar to the valve plate design in axial piston pump.

The presented work should be focused on the energy saving potential of a variable roller charge pump used in mobile machines. Finally it means the potential for fuel savings and emissions. These savings should be achieved by the displacement variability of the charge pump. Previous implementation efforts of a variable charge pump were not successful because a lot of theoretical knowledge was not available and the power saving potential was not interesting for mobile machines producers in the past due to relatively low mobile machine ownership costs. The supposition is that with implementation of variable roller charge pump with maximal displacement 20 cm^3 into hydrostatic drivetrain could be saved about 3-5% of engine power when it is not necessary to ensure high charging flow. The durability of the pump, cooling demands decrease and possible drive simplification via system components changes in existing circuits are next possible benefits for mobile machinery.

1.1 GOALS

This work deals with a variable roller pump theoretical analysis, an optimization and its application into existing systems, especially into hydrostatic transmissions. Generally, the goals and the whole work can be divided into two parts:

The first part is more focused on the component analysis. The theory for roller pump is not generally published and this theory has to be derived in comprehensive from. This theory enables the simulation model building which allows, together with suitable optimization methods, efficiency optimization focused on the volumetric efficiency. The key elements which determine pump efficiency are the kidneys on the port plates. Due to this, whole optimization is realized through the port plate design changes. Proposed port plate designs have to be tested and measurements compared with simulations for the confirmation of the theory.

The second part of this work is focused on the system analysis based on a 1-D (one dimensional) drivetrain simulation. This simulation model has to include a thermal hydrostatic transmission sub-model as well. The power saving potential of the variable charge pump is analyzed for two different mobile working machines (vehicles). These analyses require a proposal of a variable charge pump implementation into hydrostatic transmission. The gained experiences are formed into sizing methodology for variable charge pumps. The work has to extend the actual state of the art in variable roller pump control and its application into hydrostatic transmission. Based on the previous lines and state of the art, the following goals can be defined:

- Derivation of theory (kinematics, dynamics, hydromechanics) of a variable roller pump (necessary for building a 1-D simulation model).
- Optimization of a variable roller pump with focus on the (volumetric) efficiency.
- Port plate design as one of the main variable roller pump parts responsible for efficiency value. This design has to use 1-D simulation model developed according to derived theory.
- Testing of designed port plates.
- Analysis of power saving potential for two selected mobile working machines.
- Proposal of a modification of a variable roller charge pump system in existing hydrostatic transmission systems.
- Creation of sizing methodology for variable roller charge pump in hydrostatic transmission.
- Extension of the state of the art in the field of the variable roller charge pump control and its use in hydrostatic systems.

2 STATE OF THE ART

The state of the art is divided into two parts. The first part deals with a roller pump and the second deals with variable charging systems.

2.1 ROLLER PUMP

With roller vane pump in the title or abstract and F04C as the European Classification, can be found 143 results in the worldwide patent databases. The effort to commercialize roller pumps in hydraulic area is known since 1928, when L. P. Barlow filed of the first patent applications GB306031A. Since that time roller pump has been started to be employed in a wide variety of applications for many years. See as well for example patents GB450595A, GB489955A, US2460018A, GB591143A, GB654808A, GB856687A, GB1171907A, US3938918A, JP59196986A and many others.

Literature search presented in this thesis shows some works dealing with roller pumps as well. Danardono and coauthors deal with fixed LPG roller vane pump in [6]. Kinematic and force analysis were done by Zhurba and Cleghorn in [7]. Authors deal with a fixed balanced (double acting pump) roller pump, which is used as a pressure source for transmission, accessory drive and other applications. Jiadi and coauthors analyzed a roller slot design for a fixed balanced (double acting pump) roller pump in [8]. The similar works from the same research group can be found in [9], [10]. Both articles deal with a roller pump for spraying applications. Actually no article about roller pump controls are available but articles which deal with a vane pump control could be used due to the identical principle of the displacement change. Very good papers were written by Karmel in [11], [12], [13].

Another literature sources are works written by Li in [14] and Dean in [15]. Energy savings of the variable vane pump used for engine lubricating are investigated by Meira and co-authors in [16]. The variable vane pump achieved average power saving about 56% in comparison with the gear pump. The state of the art in this work doesn’t focus primary on the alternative pump types such as variable vane pumps and variable gear pumps. Variable vane pumps are widely produced and implemented in many applications and theoretically they can be used for the charging system also. Work on the assumption, that roller pumps have higher durability and lower manufacturing costs prefer the application of a variable roller pump.

It becomes clear the holistic theory of roller pumps is not published in available literature yet. The base for the theory derivation is a book written by Ivantysyn and Ivantysynova [17]. Fitch and Hong describe the vane pump principle in [18] and the general hydraulic simulation approach in [19]. The book from Cundiff gives a base overview about vane pumps in [20]. The general one dimension hydraulic simulation approach can be found in books from Manring [21], Noskievic [22], Turza [23] and Nevrly with co-authors [24].

Search for actual applications of roller pump in the industry shows only two applications. The first application is an automotive industry, where this pump type is used for fuel pumping [25]. The second is a sprayer application (low pressure) [26], where roller pumps are used for chemicals pumping (*Fig. 2.1*).

Fig. 2.1: Fixed roller pumps produced by company Hypro [26].

2.2 VARIABLE CHARGING SYSTEM

Probably the most related patent to the variable charging system is the patent number US 8833069 B2. The invention deals with a variable charge pump connected with a variable flushing system. The flushing system is controlled electronically.

Till now no (for the purposes of present work) literature (article, book, and report) was found focused on the variable charge pump system development or evaluation of its power saving potential in hydrostatic transmission. The general approach to a system could be found in many books, which deal with hydrostatic transmissions, for example [17], [18], [20], [22], [24], [27], and [28], [29]. Next possible sources are doctoral theses as [3], [4] and articles such as [30]. In the [20] could be founded more information about charge pump sizing.

The previous state of the art investigation is focused on the charge pump displacement variability but it is necessary to mention that there is another possibility how to decrease charge pump power demands. It is clear, that the demand for the charge pressure level is depending on the HST working conditions. This alternative approach is based on the charge pressure regulation according to the HST demands. For example the charge pressure is reduced in idle mode with a separate electric actuated pressure relief valve. This is not a very sophisticated approach, but applied in some of today's mobile applications. There are certain limits of this method and power savings are lower than savings achieved with a variable charge pump. The disadvantage of the charge pressure regulation is the pump performance dependency on the charge pump pressure level. The most interesting combination and the most expensive could be a variable charge pump with electronically adjusted pressure control.

Based on the literature search it could be concluded that this doctoral thesis could fill a missing place in the actual State of the Art. Especially in the area of the roller pump dynamics, kinematics and system implementation. The main contribution of doctoral work to the actual State of Art will be complexity of the phenomena accounted for the variable roller charge pump investigation.

3 THEORETICAL BACKGROUND OF ROLLER PUMPS

Starting line for the roller pump research is the theory of vane pumps which is already very well published and confirmed by a lot of real applications.

The variable roller pump is one alternative way how to realize a variable pump for low pressure applications which is the main added value of this pump. The roller pump could be used as an auxiliary pump or as a charge pump. Roller pumps are not very common in hydraulic systems. Generally, the roller pump is a special modification of a vane pump where vanes are replaced by rollers. So, a lot of vane pump knowledge can be applied on roller pumps. The typical configuration of a roller pump with fixed geometric volume is shown in *Fig. 3.1*.

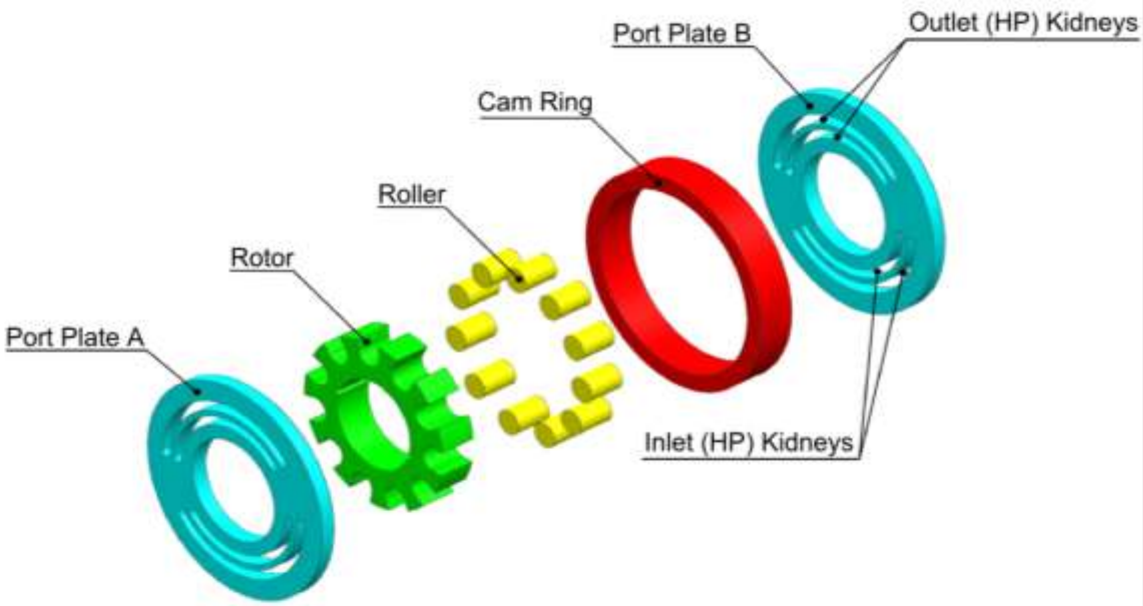


Fig. 3.1: Roller pump parts – exploded view.

The basic parts of roller pumps are rotor (carrier) with roller slots, rollers, cam ring (stator) and port plates. Inlet (low pressure - LP) and outlet (high pressure - HP) kidneys of the port plates can be split to outer and inner kidneys according to displacement chamber location. The outer displacement chambers are located between rotor and cam ring and are limited in the circumferential direction by the rollers. The inner displacement chambers are located between rotor and rollers in roller slots. The *Fig. 3.1* and *Fig. 3.2* help to explain the pump function.

The rotor is placed eccentrically; it means there is a distance between rotor center and cam ring center. This distance is called rotor eccentricity. The rollers can move radially in the slots and they are pressed against the cam ring by the centrifugal force. The rollers are loaded by the pressure in inner and outer chambers. For a correct pump function, it is necessary to ensure the sealing effect with the contact between rollers and cam ring. Due to the eccentricity of the rotor, the displacement of outer and inner chambers increase during half a rotor revolution, it means that the fluid is sucked and during second part of the revolution, the displacement decrease and the fluid is delivered to the pump outlet [17].

The kinematic equations for chamber displacements and fluid equations have to be derived according to the simulation method based on one-dimensional modeling with pressure build up equations. The next important step is the calculation of the cross section areas which are connecting the chambers to kidneys.

3.1 BASIC EQUATIONS

The equation given in [17] for calculation of a vane pump displacement can be used for the roller pump. The geometrical pump displacement is determined as:

$$V_g = 2 \cdot w_{rol} \cdot e_{rot} \cdot z_{rol} \cdot d_{cam} \cdot sin \frac{\pi}{z_{rol}} \tag{3.1}$$

where w_{rol} is the roller width, e_{rot} is rotor eccentricity, z_{rol} is number of rollers and d_{cam} is inner diameter of cam ring. A cut view and basic geometry of the roller pump are shown in *Fig. 3.2*.The effective output flow rate Q_e for constant pump Δp can be written as:

$$Q_e = n_{rot} \cdot V_g - (Q_{si} + Q_{se} + Q_{sk} + Q_{sf}) \tag{3.2}$$

where n_{rot} is rotor speed, Q_{si} are internal flow losses, Q_{se} are external flow losses, Q_{sk} are compression flow losses and Q_{sf} are filling flow losses. The filling losses Q_{sf} will be neglected because this work starts with lower speeds where a chamber filling was assumed as a relatively sufficient.

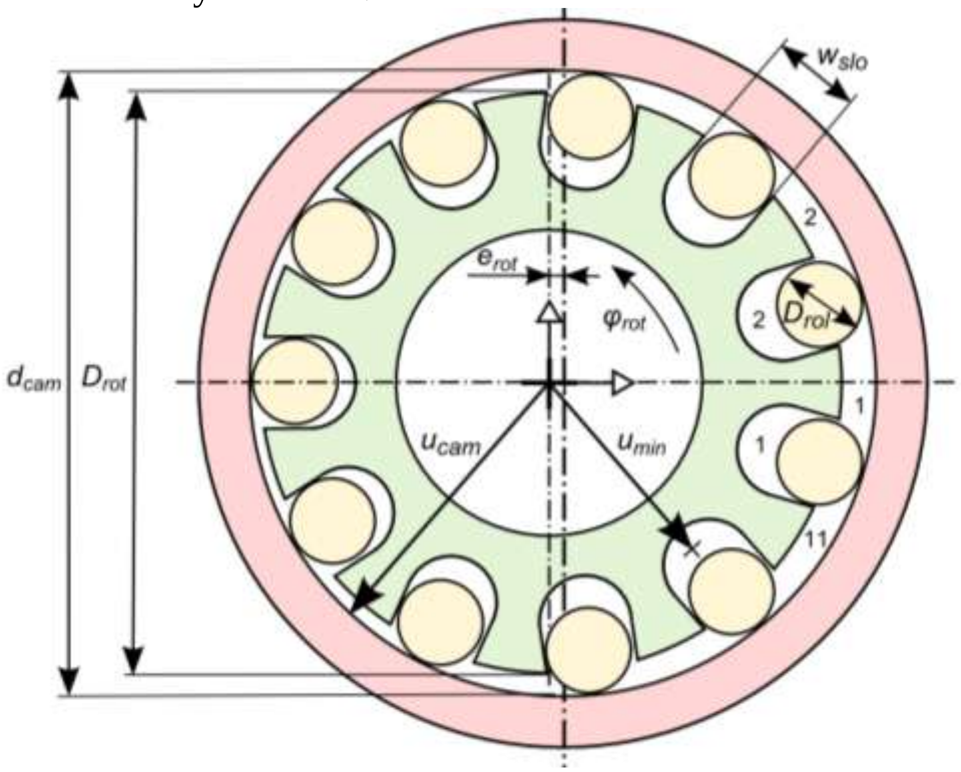


Fig. 3.2: Roller pump geometry – cut view.

External losses Q_{se} calculation is not included in the simulation model due to the assumption that external leakages are not significantly affected by port plate design. The internal leakages are shown in *Fig. 3.3*.

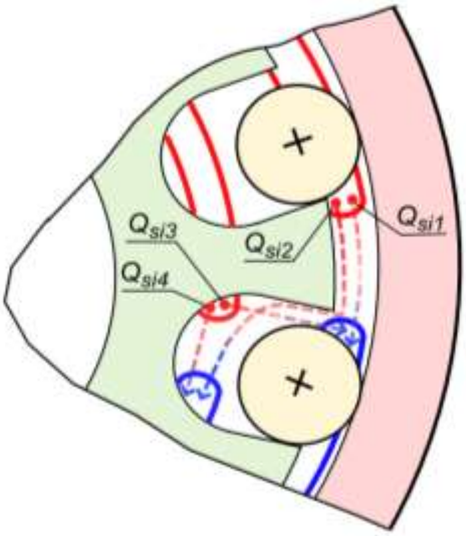


Fig. 3.3: Internal leakage in a roller pump – cross ports.

For calculation of the total amount of internal cross port leakage (CPL), the following equation is used:

$$Q_{si} = \sum_{i=1}^{z_{rol}} (Q_{si1} + Q_{si2} + Q_{si3} + Q_{si4}) \tag{3.3}$$

Partial internal leakages (CPL) Q_{sii} (Q_{si2}, Q_{si3}) can be calculated with using the orifice equation and pressure build up equations [18].

3.2 ROLLER EQUATIONS

The determination of roller forces is depending on the coordinate system which was chosen for final force determination.

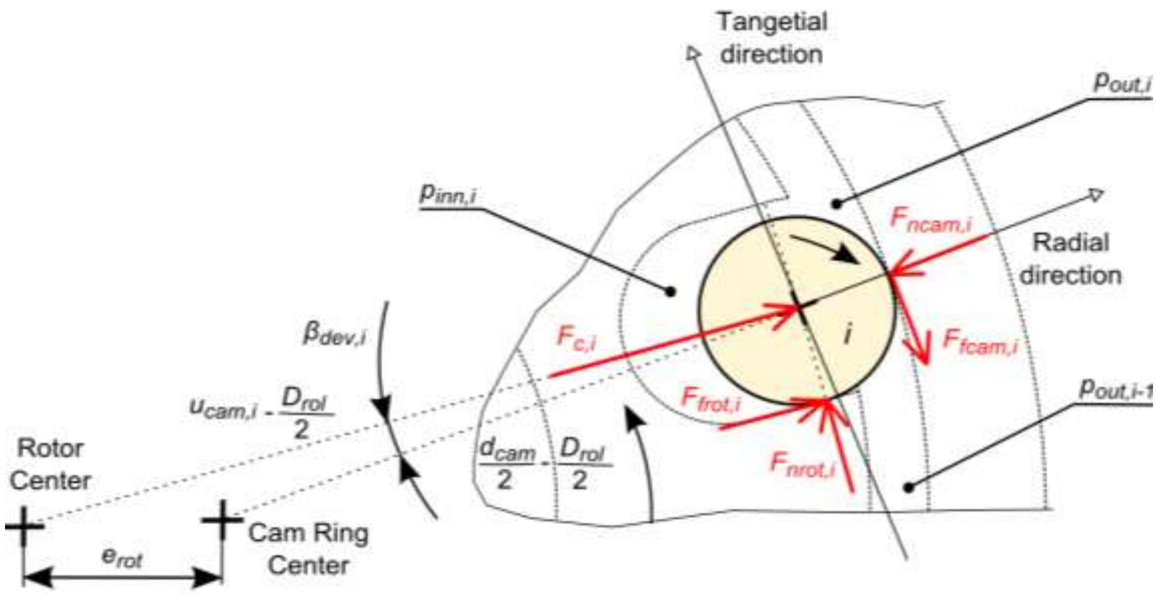


Fig. 3.4: Roller force analysis.

The radial direction x_{Rrol} (axis) for coordinating system was chosen on the connection line from roller center to the cam ring center with positive orientation heading out from the rotor center in *Fig. 3.4*. The tangential direction x_{Trol} is perpendicular to the radial direction and positive axis orientation is corresponding to the rotor rotation. The roller rotation acceleration around own roller centerline is defined by the variable $\ddot{\varphi}_{rol}$. The roller mass is m_{rol} and the roller inertia is I_{rol}. *Fig. 3.4* shows: normal roller forces F_{ncam} and F_{nrot}, friction roller forces F_{fcam} and F_{frot} and centrifugal roller force F_c. According to *Fig. 3.4* and previous text, the following equations of motion (radial equation: (3.4), tangential equation (3.5) and rotation equation (3.6) are written:

$$\begin{aligned} m_{rol} \cdot \ddot{x}_{Rrol,i} = & \\ & = \left[F_{c,i} + F_{frot,i} + \left(p_{inn,i} - \frac{p_{out,i} + p_{out,i-1}}{2}\right) \cdot w_{rol} \cdot D_{rol}\right] \cdot \\ & \cdot cos(\beta_{dev,i}) + \left[F_{nrot,i} - \left(\frac{p_{out,i} - p_{out,i-1}}{2}\right) \cdot w_{rol} \cdot D_{rol}\right] \cdot \\ & \cdot sin(\beta_{dev,i}) - F_{ncam,i} \end{aligned} \tag{3.4}$$

$$\begin{aligned} m_{rol} \cdot \ddot{x}_{Trol,i} = & \\ & = \left[-F_{c,i} - F_{frot,i} - \left(p_{inn,i} - \frac{p_{out,i} + p_{out,i-1}}{2}\right) \cdot w_{rol} \cdot \right. \\ & \left. \cdot D_{rol}\right] \cdot sin(\beta_{dev,i}) - F_{fcam,i} - F_{cor,i} + \\ & + \left[F_{nrot_i} - \left(\frac{p_{out,i} - p_{out,i-1}}{2}\right) \cdot w_{rol} \cdot D_{rol}\right] \cdot cos(\beta_{dev,i}) \end{aligned} \tag{3.5}$$

$$I_{rol} \cdot \ddot{\varphi}_{rol,i} = (F_{frot,i} - F_{fcam,i}) \cdot \frac{D_{rol}}{2} \tag{3.6}$$

The spring force – the force acting against the compression spring - can be calculated as a sum of all forces acting on the cam ring [31].

4 ROLLER PUMP CONTROL

All pump control principles used for variable roller pumps are based on the displacement change. Generally, the purpose of the displacement control is to control pump performance or/and decrease pump power demands when it is not necessary. Investigated controls are shown in *Fig. 4.1*.

The function of the simple PC control can be described with following sentences. The pump displacement is proportionally depending on the rotor-cam ring eccentricity, which changes with the cam ring motion around the pivot. At low outlet (discharge) pressure (HP) the pump displacement is in maximum (maximum rotor eccentricity) due to higher spring force (spring preload) and the lower outlet pressure force acting on the cam ring. The resultant force pushes on the cam ring and holds the cam ring at the maximal rotor eccentricity (maximal displacement). A growth of the outlet pressure increases the outlet pressure force on the cam ring, which acts against the spring force which becomes smaller than the outlet pressure force. The resultant cam ring force changes direction and moves (rotates) the cam ring to the lower eccentricity (displacement). It results into a decrease of the outlet flow which reduces the pump outlet pressure.

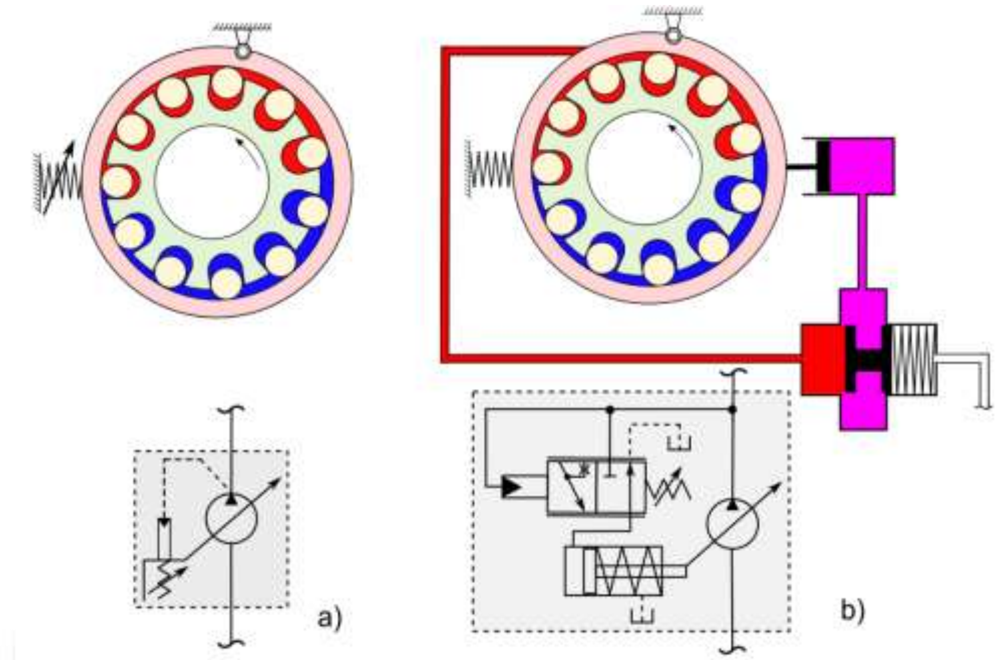


Fig. 4.1: Proposed control variants for variable roller pump.
a) Variable roller pump with simple pressure compensator control.
b) Variable roller pump with spool pressure compensator control.

The bode graph for the linearized simple PC VCP (variable charge pump) is shown in *Fig. 4.2* [32]. Another example of the performed measurements can be seen in *Fig. 4.3*. All measurements and analysis confirm suitability of selected controls.

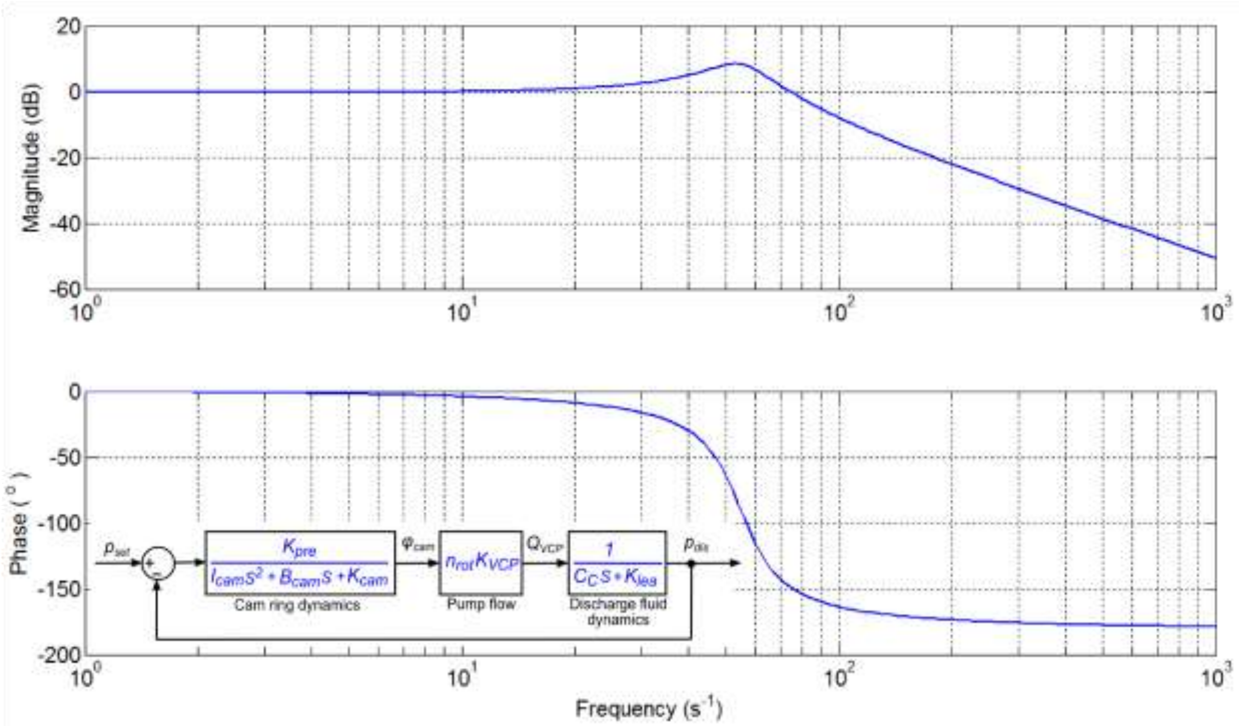


Fig. 4.2: Bode diagram of the linearized VCP pump with the simple PC control.

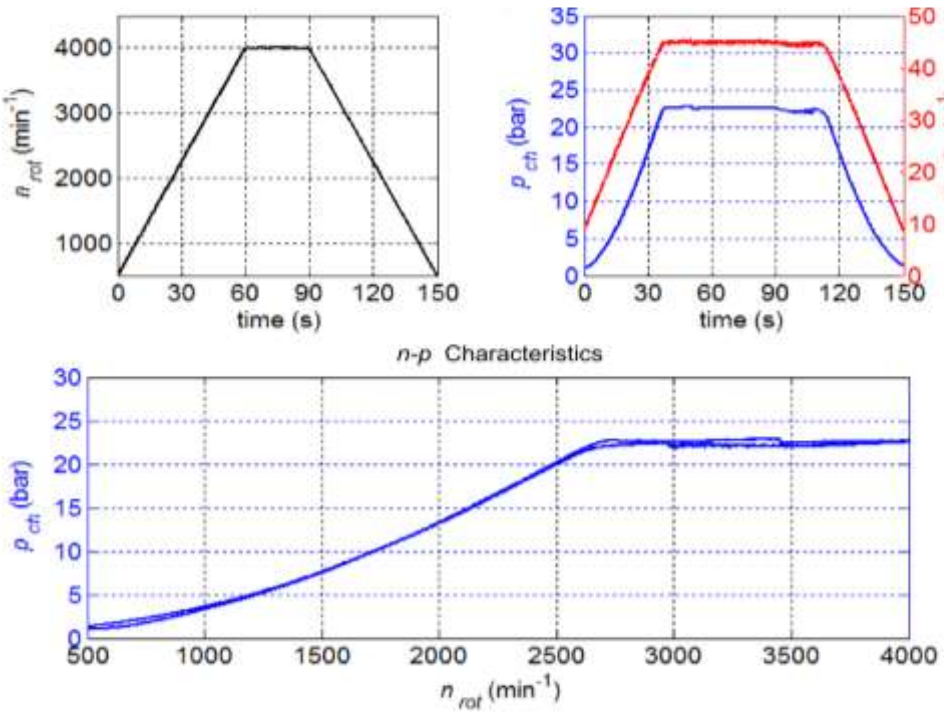


Fig. 4.3: Measured speed, charge pressure, pump flow and n-p characteristic (Speed ramp measurements, spool PC control).

5 PORT PLATE OPTIMIZATION

The port plate optimization is focused primary on the volumetric efficiency which contributes to also the overall efficiency. Especially the kidney design (notches design *Fig. 5.1*) of the port plates has a huge impact on the pump volumetric efficiency and pressure profile in the chambers, which is related with component wear and noise.

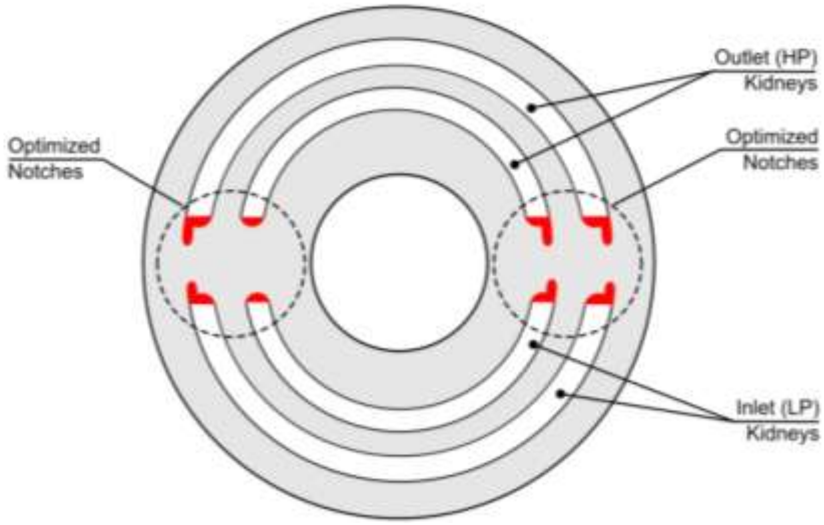


Fig. 5.1: Port plate with optimized kidneys - notches.

From the previous section it is clear that the volumetric efficiency optimization was realized with port plates (with kidneys design).The optimization in this work is based on the 1-D simulation model realized in Simulink (pump hydraulics and mechanics) and the theory-principle from work [33].

The simulation based optimization produced two port-plate designs. The first design was focused on the increase of the volumetric efficiency and the second design was focused on the chamber pressure profile smoothing. The starting point for optimization was the port plate design A which was designed based on the intuitive approach and testing of many variants of designed hardware. The port plate design B (*Fig. 5.2*) and C were designed by the developed simulation/optimization tool which significantly shortens port plate design time; especially in comparison with first design A.

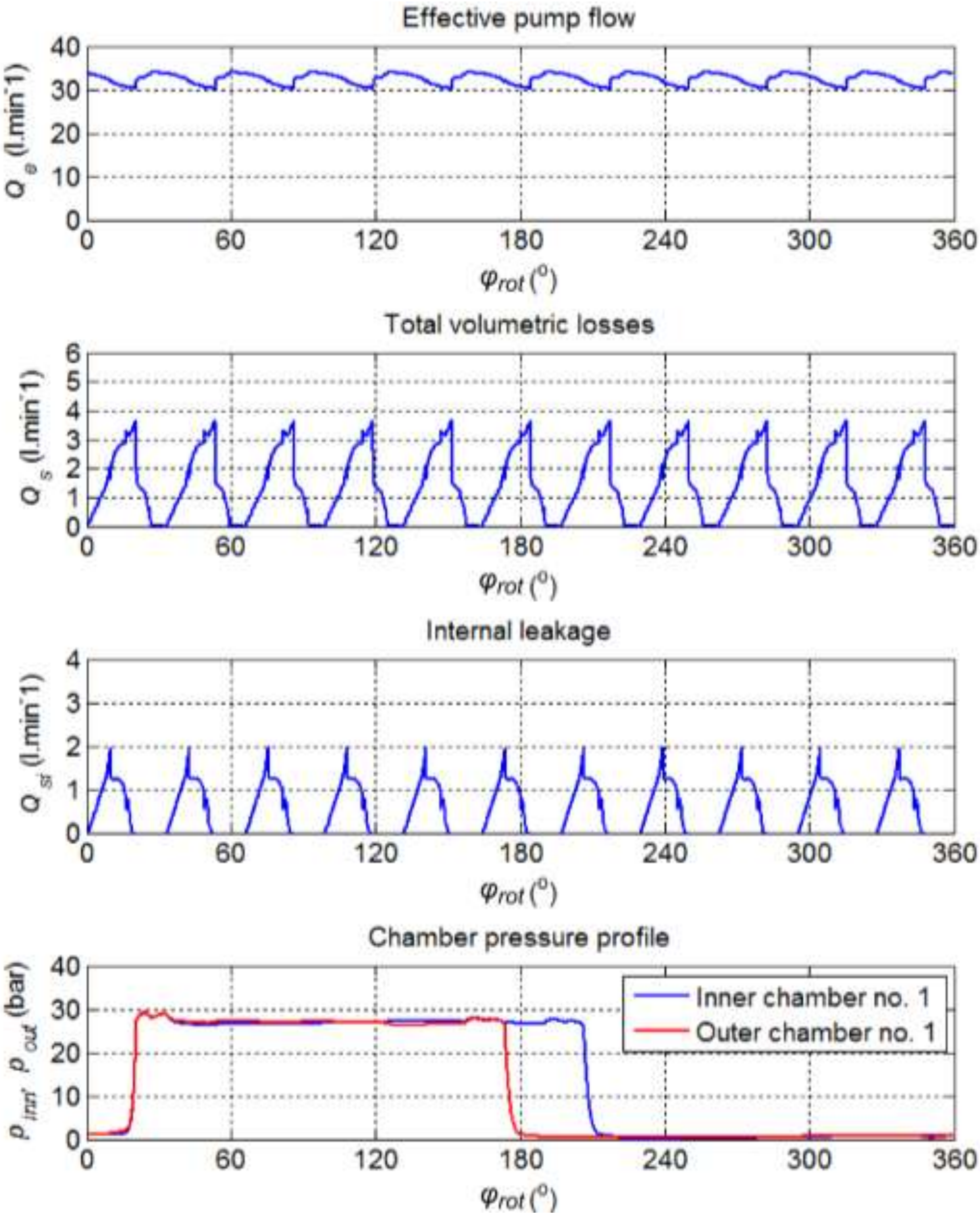


Fig. 5.2: Simulation results of optimized port plate B.

Design B was optimized for a higher volumetric efficiency – lower cross port leakage, design C was optimized for a smooth chamber pressure profile. This pressure profile smoothing was achieved by a slight increase of the cross port leakage.

6 VERIFICATION METHOD

Simulation and optimization results are validated in two areas. The first area is the validation of the volumetric efficiency. This was realized by comparison between simulation results and measurements of proposed designs. The second validation area is focused on the spring force (cam ring forces) verification.

All measurements were performed at the performance test stand, which is equipped with all necessary equipment and all sensors have a suitable accuracy and resolution. The purpose of the following hydraulic circuit (*Fig. 6.1*) is to evaluate (calculate from measured values) volumetric, hydro-mechanical and overall efficiency of the variable charge pump system.

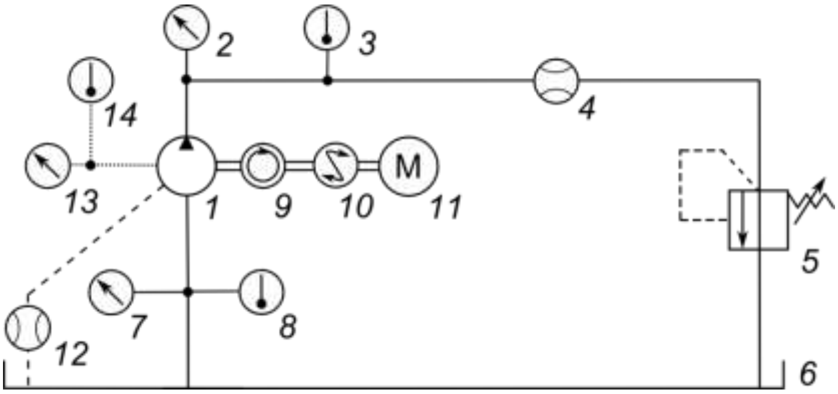


Fig. 6.1: Hydraulic circuit for simulation measurements (1-VCP).

The new two port plate designs B and C were produced according to the optimization results in the previous section. The pump prototype hardware with the best manufacturing tolerances was selected for measurements purposes. During efficiency measurements, only port plates were changed on the test hardware, minimizing other factors which can influence measurements results.

The best possible way how to measure spring force is to implement a hydraulic cylinder and a piston on the cam ring (*Fig. 6.2*). Theoretically, at steady state conditions, the pressure in the hydraulic cylinder corresponds to the mean spring force.

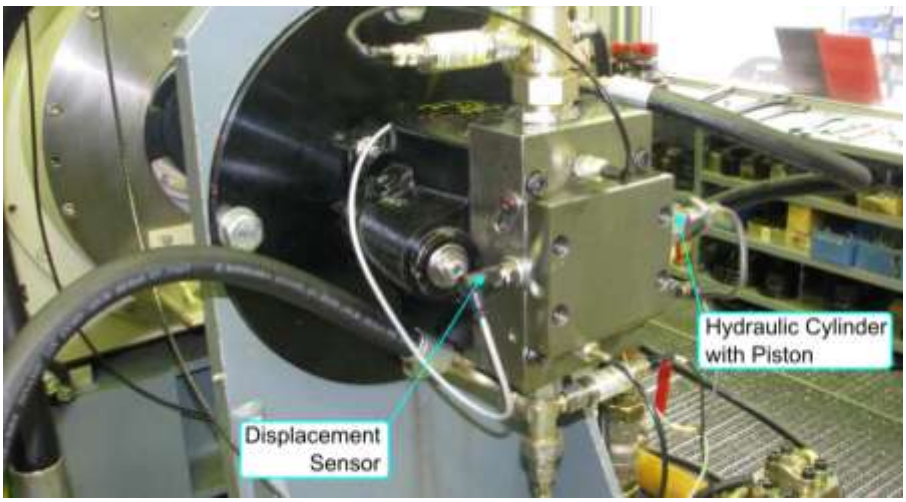


Fig. 6.2: The variable charge pump during spring force measurements.

6.1 VERIFICATIONS OF SIMULATIONS RESULTS

The measurements show that the effective flow of the pump new design increases or decreased according to optimization demands (as shown in *Tab. 6.1*) with acceptable wear marks. In the end, the measured volumetric efficiency of the pump shows the same tendency as the simulation results. Simulated flows show lower numbers than measured primary due to the external leakage neglection. Port plate B measurements show a higher effective flow for a higher delta pressure than for a lower delta pressure. Probably it was caused by the different port plate pressure balance, which makes the port plate sealing effect better at higher delta pressures.

Tab. 6.1: Comparison of simulated and measured effective flows.

Port plate ver.	Speed	System pressure	Simulation		Measurement	Deviation
			Effective flow	Volumetric losses	Effective flow	
	n_{rot}	Δp	Q_e	Q_s	Q_e	
	(min^{-1})	(bar)	(l·min^{-1})	(l·min^{-1})	(l·min^{-1})	(%)
A	1500	15	24.10	1.39	25.12	-4.1
A	1500	30	23.90	1.59	24.33	-1.8
A	2800	15	45.56	2.01	47.67	-4.4
A	2800	30	45.30	2.27	47.38	-4.4
B	1500	15	24.31	1.18	25.61	-5.1
B	1500	30	24.09	1.40	25.81	-6.7
B	2800	15	45.93	1.64	48.17	-4.7
B	2800	30	45.67	1.90	49.10	-7.0
C	1500	15	23.45	2.04	23.74	-1.2
C	1500	30	22.91	2.58	22.26	2.9
C	2800	15	44.90	2.67	45.61	-1.6
C	2800	30	44.40	3.17	45.01	-1.4

The spring force comparison confirms a good agreement between simulations and measurements.

7 MODELLING OF HYDROSTATIC DRIVETRAIN FOR POWER SAVING EVALUATION

Previous lines confirmed that the variable roller pump can be used in HST. The next lines deal with a variable roller pump application as a charge pump and its power saving evaluation.

7.1 VARIABLE ROLLER CHARGE PUMP IN HST

Application of a VCP (variable charge pump) instead of a FCP (fixed charge pump) into standard-conventional charging system (shown in *Fig. 7.1a*) brings design changes into conventional HST according to *Fig. 7.1*.

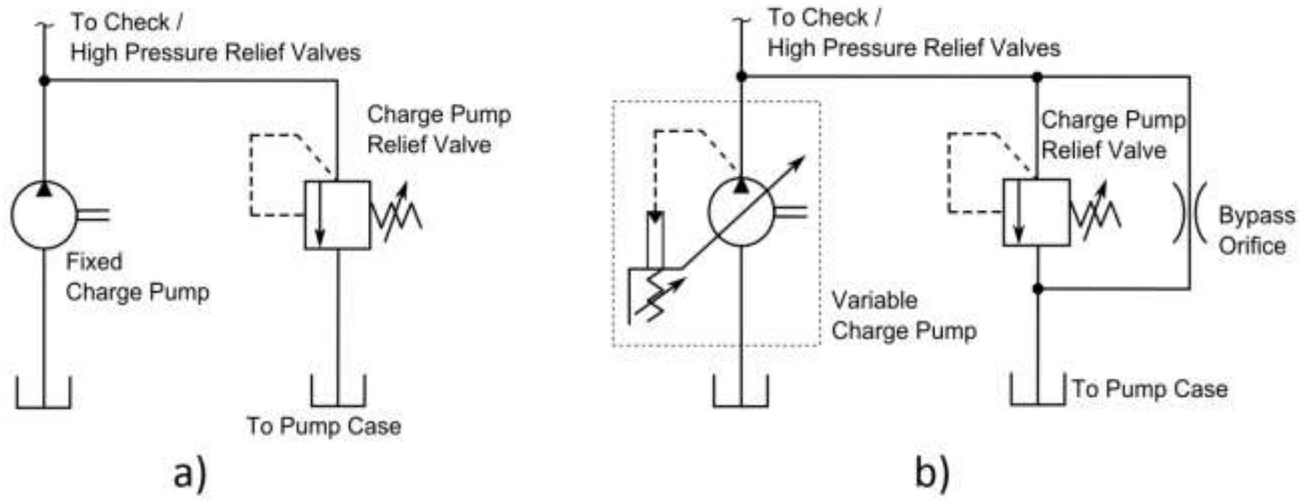


Fig. 7.1: HST fixed and variable charging systems.
a) Standard FCP system (Variant A).
b) VCP system with bypass orifice (Variant B).

Another possibility is a variable loop (motor) and pump flushing system shown in *Fig. 7.2*.

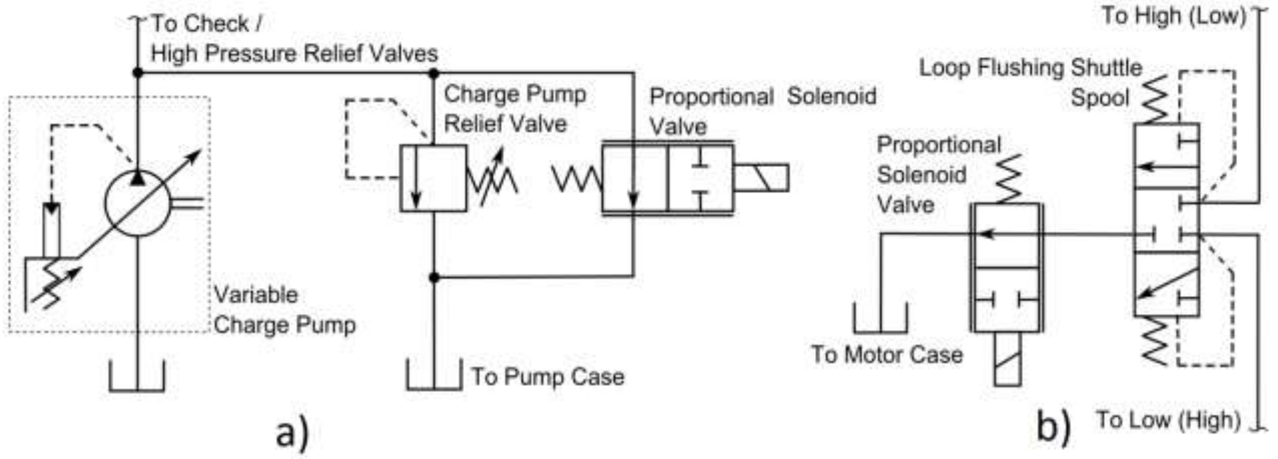


Fig. 7.2: HST variable charging and loop flushing systems.
a) VCP variable flushing system (Variant C).
b) Loop (motor) variable flushing valve (Variant C).

7.2 SIMULATION MODEL OF THE DRIVETRAINS

One possible way how to calculate VCP power savings is a 1-D drivetrain simulation. The modeling of the drivetrain is separated in two parts. The first part is the hydro-mechanical drivetrain modeling. The output from the first part is coupled with the output from the second part of the modelling which is the thermal model of the hydrostatic transmission.

7.2.1 Hydro-Mechanical Drivetrain Model

The hydro-mechanical drivetrain model was developed based on previous papers. The whole model is based on the one dimensional principle. The purpose of this model was a prediction of the drivetrain efficiency but simple dynamics was also included.

- **Engine**

The engine model is based on the work developed in [3], [4] and [34]. This model equipped with an engine speed regulator considers the engine dynamics based on measurements. The model includes a map of the engine torque and fuel consumption. The base equation for the engine shaft speed is:

$$\left(I_{eng} + I_{pum}\right) \cdot \dot{\omega}_{pum} = T_{eng} - T_{pum} - T_{dam} \tag{7.1}$$

where I_{eng} and I_{pum} represent rotational inertias of the engine and pump. The pump rotational speed ω_{pum} is the same as the engine shaft speed. The engine produces torque T_{eng}. The HST pump loads engine with torque T_{pum} (this torque includes also the torque from charge pump) and damping torque T_{dam}.

- **HST and Charge pump**

The HST efficiency model together with the charge pump efficiency model is based on Polymod. The Polymod approach, developed by Ivantysynova and Mikeska in [35], is based on a mathematical approximation of steady state measurements. Losses in the pump, motor and charge pump model predicts the efficiency of the HST. Losses in the hydraulic hoses between pump and motor are included too. The model considers the dynamics of the (HST) pump servo system, pump control, motor control, motor loop flushing system and charging system The dynamic part of the HST model is based on one-dimensional/ control volume modelling which consists of orifice equations, pressure build up equations and equations of motion, which can be found in [17], [18] and [24]. The model parameters were adjusted according to the measurements to achieve sufficient agreement between measurements and simulations. The VCP simple PC control model is based on the developed linearized control system model.

- **Mechanical Transmission and vehicle**

The model of the vehicle and the mechanical transmission was developed according to equations mentioned in [3], [4], [34]. The model uses a simplified one dimensional vehicle model according to the equation (7.2).

$$\begin{aligned} m_{veh} \cdot \dot{v}_{veh} = \\ = \frac{T_{mot} \cdot i_{mec} \cdot \eta_{mec}}{r_{whe}} - m_{veh} \cdot g \cdot \\ \cdot \sin(\vartheta) - m_{veh} \cdot g \cdot f_{rol} \cos(\vartheta) \cdot \mathrm{sign}\,(v_{veh}) \end{aligned} \tag{7.2}$$

The vehicle mass is represented by m_{veh}, vehicle longitude speed by v_{veh}, motor torque by T_{mot}, mechanical transmission ratio by i_{mec}, mechanical transmission efficiency by η_{mec} (or by inverted value in the case of the braking mode), wheel radius by r_{whe}, slope angle by ϑ, gravitational constant by g and rolling resistance coefficient is corresponding to the variable f_{rol}.

7.2.2 Hydro-Mechanical Drivetrain Model

The thermal model coupled with a hydro-mechanical one dimensional model is used for a determination of the cooling / loop flushing requirements in the HST. Due to simulation speed requirements the thermal approach is also based on the 1-D approach.

The older works such as [22], [36], [37], [38] and newer from Maha fluid power research center [39], [40] were used for the thermal model development in this work. The general equation (7.3) of control volume energy E can be written [40]:

$$\frac{\mathrm{d}E}{\mathrm{d}t} = \sum \dot{m}_{in} \cdot h_{in} - \sum \dot{m}_{out} \cdot h_{out} + \dot{q} - \dot{W} \tag{7.3}$$

where the input oil mass flow is $\dot{m}_{in}$ with the enthalpy h_{in} and the output oil mass flow from the control volume is $\dot{m}_{out}$ with the input enthalpy h_{out}. The heat rejection rate from the control volume is $\dot{q}$ and $\dot{W}$ is the work rate.

The main advantage of the control volume approach in hydraulic and thermal modelling is that this approach enables to connect the hydraulic model with the thermal model. The thermal model output is the oil temperature and the inputs are: ambient temperature, power loss and flows in and out of the control volume which are assumed as equal. The simulation model assumed that all power losses (P_{loss}) in the circuit are transferred into heat of the hydraulic oil. Each hydraulic component in the thermal part of the simulation model (hose, pump, motor and tank) is a connection of two node equations (fluid mass equation and component mass equation).

The forced convection heat transfer occurs inside of the components and it is a connection between fluid mass node and component mass node. The conductive and radiative heat transfer equations connect each component mass node to the surrounding (with ambient temperature). In the end, this connection gives the fluid mass passive conductive and radiative cooling ($\dot{q}$) through the component mass node. Base on the *Fig. 7.3*, the general thermal model was setup. The model considers only basic components in the hydrostatic transmission.

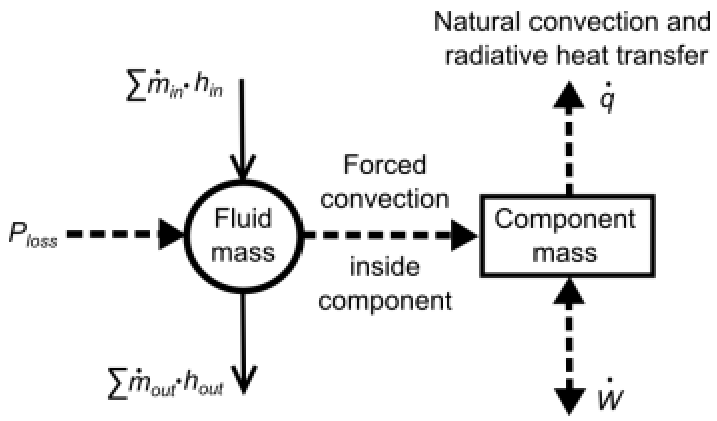


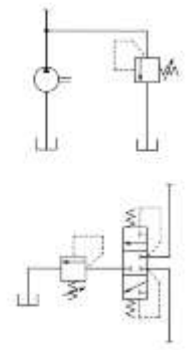
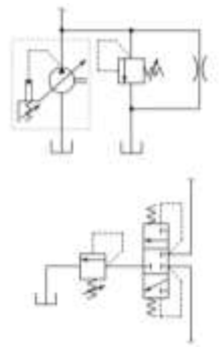
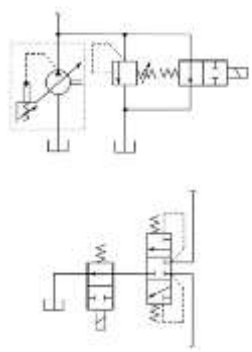
Fig. 7.3: Thermal modeling approach for hydraulic components [40].

8 POWER SAVING POTENTIAL

The power saving output is one of the most important outputs of this work. The resultant power saving potential can be easily recalculated to the energy, fuel, emission or cost savings if the relevant data are available. The next important outputs from the simulation are information about the operating parameters like pressures, temperatures, speeds etc. The evaluation of the VCP power saving potential is based on the simulations for typical operating conditions of the selected vehicles [30]. The different vehicle parameters or the different operating conditions can result into different power savings [41].

The one example of the simulation results for combine harvester (weight 20 ton, max engine power 300 kW) is shown in *Tab. 8.1*.

Tab. 8.1: Combine harvester harvesting simulations.

<table>
<tr><td></td><td colspan="2">A: FCP 40cm³
(standard system)</td><td></td><td rowspan="5"></td></tr>
<tr><td>Charge pres.</td><td>27.5 bar</td><td>Pump case temp.</td><td>62.6 ℃</td></tr>
<tr><td>Displacement</td><td>100%</td><td>Motor case temp.</td><td>77.3 ℃</td></tr>
<tr><td rowspan="2">Power in</td><td rowspan="2">7.35 kW</td><td>Pump case flow</td><td>54.0 l·min⁻¹</td></tr>
<tr><td>Motor case flow</td><td>50.9.1 l·min⁻¹</td></tr>
<tr><td></td><td colspan="2">B: VCP 40cm³
with bypass orifice
(3 mm bypass orifice,)</td><td></td><td rowspan="5"></td></tr>
<tr><td>Charge pres.</td><td>27.6 bar</td><td>Pump case temp.</td><td>66.6 ℃</td></tr>
<tr><td>Displacement</td><td>73 %</td><td>Motor case temp.</td><td>76.7 ℃</td></tr>
<tr><td rowspan="2">Power in</td><td rowspan="2">5.65 kW</td><td>Pump case flow</td><td>28.3 l·min⁻¹</td></tr>
<tr><td>Motor case flow</td><td>47.8 l·min⁻¹</td></tr>
<tr><td></td><td colspan="2">C: VCP 40cm³
with variable pump and loop flushing system, (sett. 80 ℃)</td><td></td><td rowspan="5"></td></tr>
<tr><td>Charge pres.</td><td>27.6 bar</td><td>Pump case temp.</td><td>76.5 ℃</td></tr>
<tr><td>Displacement</td><td>56 %</td><td>Motor case temp.</td><td>79.4 ℃</td></tr>
<tr><td rowspan="2">Power in</td><td rowspan="2">4.58 kW</td><td>Pump case flow</td><td>13.4 l·min⁻¹</td></tr>
<tr><td>Motor case flow</td><td>45.2 l·min⁻¹</td></tr>
</table>

The example of the telehandler (weight 9 ton, max engine power 75 kW) simulation results for one loading cycle is shown in *Fig. 8.1*. The last graph shows input charge pump power for the standard FCP system (Variant A, blue line) and for VCP system with bypass orifice (Variant B, red line). The VCP power saving for the Y cycle is up to 0.4 kW, depending on the vehicle performance

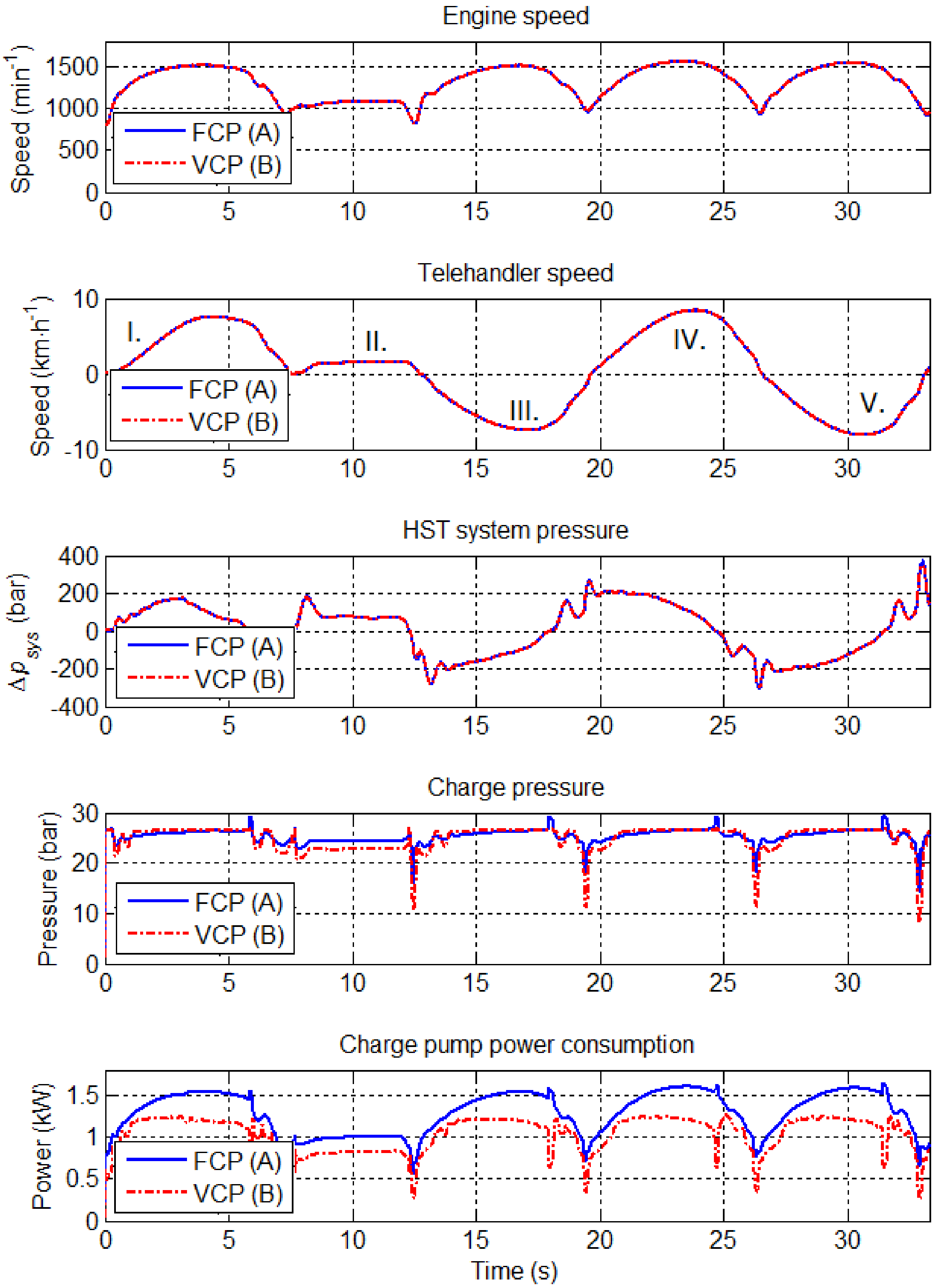


Fig. 8.1: Telehandler Y cycle simulations for the standard FCP system (A) and VCP system (B) with the standard LF system and pump bypass orifice (1.8 mm) with constant oil temperature.

8.1 RECOMMENDATIONS FOR SIZING OF A VARIABLE CHARGE PUMP IN HST

In the past, the standard FCP was normally sized as a 10% of pump and motor total displacement [20]. According to this rule, HST circuit with the 78 cm^3 pump and 110 cm^3 requires the 19 cm^3 charge pump. A more detailed approach quantifies all leakages and flow consumption which occurs in the circuit. Together with the detail knowledge of the vehicle application, approach in the thesis was proposed.

The total demand for cooling flow can be calculated from all power losses which occur in a HST. The power losses can be specified in three ways; experimentally, by rough estimation (about 25-30% of HST input power) or simulated with Polymod [35].The specification of the power losses in the HST has to consider not only the losses in the pump and motor but it has to consider all losses which occur in hoses, fittings and also valves. The used approach assumed that all power losses in a HST convert to increased oil temperature. Normally, the hottest spot in HST is a motor case but for the calculation it is better to use the oil temperature in a system hose (loop temperature) T_{Loop}. For each application, we have also a defined maximum temperature of the hydraulic oil. Then the rough estimation of the cooling flow Q_{CF}, demanded for heat removal from the HST, can be written as:

$$Q_{CF} = \frac{(P_{lossPTotal} + P_{lossMTotal} + P_{lossHose})}{\left(T_{Loop} - T_{Tank}\right) \cdot \rho_{oil} \cdot c_{oil}} \tag{8.1}$$

where the total pump loss $P_{lossPTotal}$ is a loss which occurs in the pump housing, which means it is the sum of kit and pressure drop losses, control losses, charge pump losses, CPRV (bypass, in the case of the VCP) losses, check valve losses and HPRV losses. Similarly, total motor losses $P_{lossMTotal}$ include kit losses and pressure drops, loop flushing losses and motor control losses. These losses used in the numerator of the equation (8.1) can be measured, calculated using efficiency data or simply estimated with overall efficiencies. Losses, which occur in hoses, are included in $P_{lossHose}$. The sum of all losses in the HST is represented by the numerator of the equation (8.1). The denominator contains the oil density ρ_{oil}, the oil specific heat c_{oil} and the difference between the tank and loop oil temperatures. This temperature difference (T_{Loop}-T_{Tank}) provides information about the cooler effectivity. The hot oil is removed from the HST through the case flows. Thus the sum of the pump case flow and the motor case flow has to be equal to the cooling flow Q_{CF}. The equation (8.1) can be plotted as a function of the pump speed and it has to be calculated for some typical system delta pressures and maximal allowed temperatures. The calculated cooling flow Q_{CF} includes pump case leakages, motor case leakages. The difference between calculated cooling flow and case leakages (Q_{contP}, Q_{contM}, Q_{leakP}, Q_{leakM}, Q_{caseCP}) has to be divided between the loop flushing flow Q_{LF} and the CPRV flow Q_{CPRV} (bypass flow in the case of the VCP). The LF valve setting and bypass orifice sizing have to be based on simulations or measurements. The VCP sizing depends on the whole cooling flow Q_{CF} demand, not on the cooling flow redistribution between the pump case and motor case.

9 CONCLUSION

The presented thesis gives a holistic view about variable roller charge pumps. At the beginning, the work deals with the theory connected to the selected pump and then continue with the volumetric efficiency modelling and measurements. The work continues with the pump control force analysis and the control comparison. These analyses approach the end with system investigations and finish with the evaluation of the variable charge pump power saving potential in selected mobile machines. The whole thesis can be separated into two parts. The first part is primary focused on the roller pump and the second part is focused on the system application and power saving evaluation.

At the first part of this thesis, the roller pump functionality was described and the methodology of a 1-D simulation model was developed. The developed simulation model could be used for pressure profile prediction, roller force prediction and cross port leakage prediction which has direct impact on the total volumetric efficiency. The simulation model was successfully used as a tool for optimization of the port plates which was confirmed by several tests. The measurements confirmed the tendency of the effective flow behavior between simulation results and measurements. The developed tool could help to predict the variable roller charge pump volumetric efficiency and in the end, it provides better port plate designs for the roller pumps and solves possible future problems related to the pressure profile. This work provides a good base for next developments in the area of roller pump simulations.

The next model improvement could be achieved by including the external leakage into the model. It should help to calculate more accurately the volumetric efficiency in the simulation model. The implementation of the torque calculation and total efficiency calculation should be a future step in the model development.

Another advantage of the developed model is the cam ring force prediction which could help to improve the performance of actual controls. Also this part of the model was verified. The work deals also with possible controls which can be applied on the variable roller charge pump. Performed basic measurements compared two types of the control and confirmed that the variable roller charge pump is able work in transmissions with both types of the control. Simulation models provided also explanation of the root reasons for different control performances.

This second part of the work analyzed the potential of a variable charge pump for two typical mobile applications. This part required to perform a 1-D drivetrain simulation model together with thermal behaviour analysis of the HST. The drivetrain simulation of the VCP charging system with a bypass orifice confirms higher power savings only in cases when the pump speed was significantly higher than normal speeds and a relatively constant flushing flow through the bypass orifice to the pump case still ensures suitable cooling. The best simulation results were achieved with variable flushing flows, where the demand for charging flow was adjusted (decreased) according to the HST cooling requirements.

Performed simulations show maximal steady state power savings about 2.8 kW for combine harvester (maximal engine power 300 kW) and about 1.4 kW for telehandler (maximal engine power 75 kW) at simulated conditions. Theoretically further acceptable oil temperature increase could bring further VCP power savings (savings are paid by oil temperature increase). This idea has to be investigated more because of the leakage increase due to the oil temperature increase.

Based on previous simulation results, the VCP can only be effective or offer an added value in systems (vehicles) where the full (or a large amount) of the charge flow is not needed in the majority of machine operating time. For example the suitable application could be a HST system where a significant amount of the operating time, the pump runs at zero or very small angle, for example differential steering at track type vehicles or suitable HST application with an oil temperature dependent intelligent flushing (cooling) system. Especially the combination of a suitable HST system and the intelligent variable flushing flow could make with the VCP an interesting solution aimed to decrease the power consumption and in the end the fuel consumption. Theoretically, next system improvements can be achieved with an electronically adjustable PC control, which is able to adjust charge pressure according to the HST demands.

These VCP system and intelligent flushing (cooling) systems are not used in today's standard HST. The next step is to design and simulate the VCP with the concrete intelligent variable flushing flow and compare fuel savings and VCP system costs, if the VCP implementation is profitable. The electronically controlled PC control for the VCP requires further investigation too.

9.1 CONTRIBUTION OF DOCTORAL THESIS

From the state of the art point of view, this work brings relatively complex investigation of the variable charge pump design and its system application together with a power saving evaluation. The developed theory and tools will extend the existing state of the art connected not only with the roller pump topic but mainly with variable charge pump systems. The main contribution of this work can be separated into the following points:

- Extension of the variable roller pump theory, which was confirmed by comparison between measurements and simulations [42].
- Development of a methodology for the variable roller pumps optimization with focus on the volumetric efficiency [42].
- Cam ring forces (control force) derivation, validated by measurements [31].
- Comparison of selected pressure compensator controls for a variable charge pump and their analysis through linearized VCP control models.
- Novel proposal of variable charge pump implementation into HST [41].
- Power saving potential evaluation for two types of vehicles [41]:

- Telehandler.
- Combine harvester.

- Recommendations for pump sizing and proposal of a system layout together with suitable charge systems for maximization of the variable charge pump power savings (variable flushing flow).

The whole work presents a guideline for a sophisticated variable (roller) charge pump development and covers empty spots in the theory. Additionally, methodologies dealing with the variable charge pump design and system applications were shown. Kinematic equations and tool for cross section areas calculation were also implemented into graphic user interface, which is successfully used for VCP design calculations. The value of the knowledge included in this work is highlighted by a real market demand.

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ABSTRACT

Presented doctoral thesis deals with an extensive hydraulic variable roller pump analysis and the power saving prediction of hydrostatic drivetrains in the mobile machines achieved with a variable roller charge pump implementation.

At the first part of the work, the roller pump functionality was described and the theory of a 1-D simulation model was developed. Based on this developed simulation model is suitable for pressure profile prediction, roller force prediction and cross port leakage prediction which has a direct impact on the total volumetric efficiency. The simulation model was successfully used as a tool for optimization of the port plates, which was confirmed by measurements. The first part of the work includes the pump control force analysis validated by measurements and also the basic pressure compensator controls comparison. Developed control force prediction could help to improve the control performance. The measurements confirmed that the variable roller charge pump is able to successfully work in transmissions with measured types of the control.

The second part of the work analyzed the power saving potential of a variable charge pump for two selected typical mobile applications: telehandler (9 ton) and combine harvester (20 ton). This part required a 1-D drivetrain simulation model together with thermal behaviour of the hydrostatic transmission. Two different modifications of the charging systems were compared with the conventional charging system in simulations performed for the working and transporting mode. The drivetrain simulation of the variable roller charge pump with a bypass orifice confirms higher power savings only in cases when the pump speed was significantly higher than normal speeds and a relatively constant flushing flow through the bypass orifice to the pump case still ensures suitable cooling. The highest power savings were achieved with variable flushing flows, where the demand for charging flow was adjusted according to the hydrostatic transmission cooling requirements. At the end of the second part, this thesis deals with a variable charge pump sizing.