UNIVERSITY OF AGDER

Modeling, Simulation and Experimentation of a Hydrostatic Transmission

By
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Abstract

Modeling and simulation of real systems is necessary for the industry in order to improve or develop new products quickly. Simulation is less time consuming and cheaper compared with experimental testing and prototyping. Different modeling tools are on the market today. This thesis presents the results of modeling and simulation of a physical hydrostatic transmission with three different modeling tools; Simulink, SimHydraulics and SimulationX. The aim has been to get the simulations from the different models to be as similar as possible to the two measured pressures and the rotational speed of the load. The four frictional losses in the hydraulic motor have also been tried estimated in order to create a more detailed model. The SimulationX model gave the best results compared with the measurements. The largest challenge has been to simulate the model in Simulink and to find the frictional losses in the hydraulic motor by performing different tests. The solver in Simulink could not solve the equations and it was difficult to find the tests for finding two of the friction parameters.

Preface

This master thesis is submittet in partial fulfillment of the requirements for the degree Master of Science in Mechatronics at the University of Agder, Faculty of Engineering and Science. This work was carried out under the supervision of professor Geir Hovland at the University of Agder, Norway.

First of all we wish to thank our supervisor Geir Hovland. We thank him for his follow up and support through the whole project period.

We also wish to thank professor Michael R. Hansen for his assistance in the field of hydraulics and modeling of hydraulic systems.

Grimstad, May 2010

Per Willy Lauvli Bjørn Victor Lund

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Chapter 1

Introduction

1.1 Topic Description

The purpose of this thesis is to perform modeling and simulation of an existing hydrostatic transmission bench with three different simulation concepts/tools; SimulationX, SimHydraulics and Matlab/Simulink. Physical measurements from the bench will be done in order to verify the results from the simulations. The motor dependent constants for calculating the 4 different hydro-mechanical losses for the hydraulic motor are estimated by using specific tests on the transmission.

1.2 Background

Modeling and simulation have become more and more important for the industry in order to optimize and improve product design, perform dynamically analysis and to easily test new concepts instead of building costly and time consuming prototypes. Different simulation tools are on the market today. Some tools include finished developed model libraries that include physically models of elements in e.g. hydraulics, mechanics and electronics. These tools require different inputs from the user in order to give the components correct properties. Other tools require that the user set up the mathematical equations for the whole system. To create a model of a hydraulic motor and pump the hydro-mechanical losses must be included. Often the hydro-mechanical efficiency for the pump and motor, as a function of the pressure drop or rotational speed, is known. The hydro-mechanical efficiency includes 4 different friction losses. By including the hydro-mechanical efficiency it is not possible to know how large the different friction losses are. Therefore it is necessary to perform some experiments to find all the friction losses.

1.3 Motivation

If the simulation results shall correspond to the results from a real system it assumes that the model describes the system well. The only way to get an idea if the model describes the system well or not is to compare the simulated results with measurements from the physical system. In order to create an accurate mathematical model of a hydraulic system, which includes a hydraulic motor, the friction losses due to the output torque in the motor needs to be found. This project shows the simulation results for three different modeling concepts/tools when a hydrostatic transmission is modeled. The results can give an indication of which concepts/tools that will give the most accurate results compared with the real system. The tests for finding the 4 hydro-mechanical losses in pumps and motors can be useful; when someone wants to create a more detailed model or when the hydro-mechanical efficiency curves are not known and when to optimize the design of a pump or motor.

1.4 Project Scope

This project covers the development of three models of a hydrostatic transmission by using different tools and the simulation results. The purpose of this project is to create an accurate model of a hydrostatic transmission and to evaluate the results from three different tools. The results will be confirmed by physical experiments. It also describes an experimental method for finding the friction losses in a hydraulic motor. This project includes following:

- Modeling of a hydrostatic transmission with standard component models in SimulationX and SimHydraulics.
- Mathematical modeling of a hydrostatic transmission in Matlab and Simulink.
- Comparison of the different simulation results with the experimental results from hydrostatic bench.
- Investigating different experiments methods to find the different constants for the friction losses in a hydraulic motor.

Chapter 2

Literature Review

Hydrostatic transmissions are widely used and a well known transmission system. It is easy to find research articles, notes and books about this subject. Hydrostatic transmissions are used in cranes, winches and vehicles. It is also in the research area for wind turbines. A lot of the research articles describe the mathematical modeling of hydrostatic transmissions for vehicular driving systems. The models are often made to come up with different control strategies. The mathematically equations that describes the hydrostatic transmission for this project is established and solved by using the methods found in the note ”Hydraulic Components and Systems” [1].

Chapter 3

Nomenclature

Parameter	Description
ρ	Density of oil$\left[\frac{kg}{m^3}\right]$
m_L	Mass of the load $[kg]$
P_1, P_2, P_3, P_4	Pressure of each node $[bar]$
$P_{cr1},\ P_{cr2},\ P_{cr\_check}$	Crack pressures for relief valves and check valves $[bar]$
$\Delta P_P,\ \Delta P_M,\ \Delta P_{pipe},$ $\Delta P_{bend},\ \Delta P_{T-junc}$	Pressure difference over pump, motor, pipe, bend and t-junction$[bar]$
V_1, V_2, V_3, V_4	Volume of each pressure node $[l]$
$D_P,\ D_M, D_{BP}$	Displacement for motor, pump and boost pump $\left[\frac{cm^3}{rev}\right]$
β	Bulk modulus $[bar]$
$\mu,\ \mu_{ref}$	Dynamic viscosity $[bar \cdot min]$
J_{eff}	Effective mass moment of inertia $[kgm^2]$
$n_M,\ n_P,\ n_{BP}$	Rotational speed of motor, pump and boost pump $\left[\frac{rev}{min}\right]$
$\ddot{\theta}_M$	Angular acceleration of the motor $\left[\frac{rad}{s^2}\right]$
$Q_1,\ Q_8$	Theoretical pump flow, main pump and boost pump $\left[\frac{l}{min}\right]$
Q_6	Theoretical motor flow $\left[\frac{l}{min}\right]$
$Q_2,\ Q_{3,}\ Q_{4,}\ Q_5$	Flow over check valves $\left[\frac{l}{min}\right]$
$Q_{7,}\ Q_9$	Flow over relief valves $\left[\frac{l}{min}\right]$
$Q_{leak,M}, Q_{leak,P}$	Leakage flow, motor and pump $\left[\frac{l}{min}\right]$
$k_{leak,M},\ k_{leak,P}$	Leakage coefficient, motor and pump $\left[\frac{l}{min \cdot bar}\right]$
$\eta_{vM},\ \eta_{vP}$	Volumetric efficiency, hydraulic motor and pump
$\eta_{mhM},\ \eta_{mhP}$	Hydro-mechanical efficiency, hydraulic motor and pump
T_{loss_M}	Total losses in the output torque due to friction for hydraulic motor $[Nm]$
$T_{mM},\ T_{vM},\ T_{hM},\ T_{sM}$	Losses in the output torque of the hydraulic motor due to mechanical, viscous, turbulent and static friction $[Nm]$
T_{tM}	Theoretical output torque from the hydraulic motor $[Nm]$
T_{tP}	Theoretical input torque to the hydraulic pump $[Nm]$
$T_{mP},\ T_{vP},\ T_{hP},\ T_{sP}$	Input torque of the pump required to overcome mechanical, viscous, turbulent and static friction $[Nm]$
$K_{mM},\ K_{mP}$	Motor and pump dependent constant for mechanical friction $\left[\frac{\frac{cm^3}{rev}}{62{,}83}\right]$
K_{vM}, K_{vP}	Motor and pump dependent constant for viscous friction $\left[\frac{m^3}{10^5}\right]$
$K_{hM},\ K_{hP}$	Motor and pump dependent constant for turbulent friction $\left[\frac{kgm^2}{\frac{1}{60}}\right]$
v_{line}	Fluid velocity in line $\left[\frac{m}{s}\right]$

Chapter 4

Hydrostatic Transmission

The main purpose for a hydrostatic transmission is to transfer the mechanical input power to a mechanical output power by using a hydraulic system consisting of a pump and a motor. The pump is directly connected to a hydraulic motor. By using a variable displacement pump or motor it is possible to adjust the output speed of the motor in order to get the required output speed or torque from the motor since $power = momentum \cdot rotationalspeed$. It is then possible to adjust the speed ratio/gearing between the pump and motor. The hydrostatic transmission can be either open or closed. For an open hydrostatic system the pump gets its oil from the tank before it is transported to the motor and returns back to the tank. A directional control valve have to be used in order to change direction of the speed of the motor. For a closed hydrostatic circuit the pump is directly connected to the motor without any direct connection to tank. The hydrostatic transmission bench is a closed loop transmission as shown in Figure 4.1.

The variable displacement pump (1) is driven by the electro motor (14) and is directly connected to the hydraulic motor (3). The pump and motor have a drain connection, (2) and (4), which is connected to the tank (13). The connection to tank is because of the external leakage from the chambers to the housing of the motor and pump as shown in Figure 4.2. A boost pump (5) refills the closed circuit with oil since the external leakage of the pump and motor returns to tank. It also makes sure that the oil will be cooled, since it changes the oil in the closed

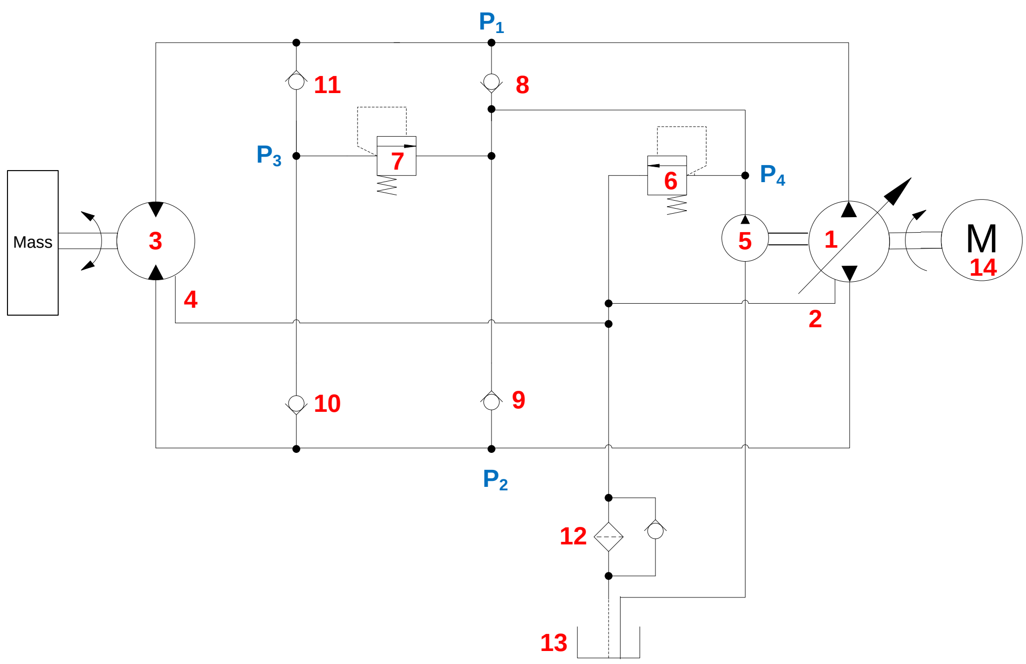


Figure 4.1: Closed hydrostatic transmission

circuit with oil that has been cooled in the tank. The boost pump is driven by the electromotor (14) with a belt transmission. The motor and pump have also an internal leakage from the high pressure chamber to the low pressure chamber. The pressure relief valve (6) makes sure that the minimum system pressure is 10 bar. If the pressure P_1 comes below a minimum pressure at 9,65 bar (10 bar - 0,35 bar) the check valve (8) will open and refill oil. When the pressure P2 comes below the minimum pressure the check valve (9) will open and refill oil. The pressure relief valve (7) make sure that the pressures P_1 and P_2 will not become larger than 75bar when the motor is braking or accelerating. If the pressure P_1 exceeds the maximum pressure the pressure relief valve (7) will open. The pressure P_4 will increase and the check valve (9) will open and fill oil to the low pressure branch. When the pressure P_2 exceeds the maximum pressure the pressure relief valve (7) will open, and the check valve (8) will refill oil to the low pressure branch.

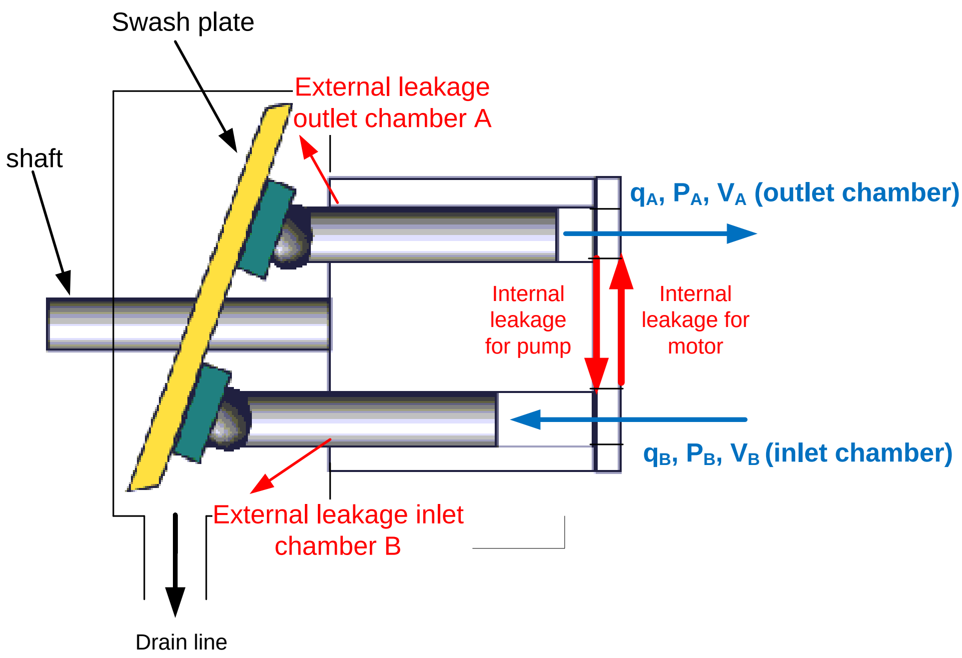


Figure 4.2: Leakage in motor and pump

4.1 Pump and Motor Efficiency

The pump efficiency tells how much of the rotational power on the input shaft which will be transferred to hydraulic power. The motor efficiency tells how much of the hydraulic power on the input port which will be transferred to rotational power. There are three different efficiencies: Hydro-Mechanical, volumetric and total.

The following list were found in ”Hydraulic Components and Systems”[1]

- Hydro-Mechanical efficiency: is how much the theoretical torque is to the actual torque needed. There are 4 different hydro-mechanical losses:
  - Mechanical friction: due to mechanical contact between parts of the pump moving relative to each other. Proportional to the pressure rise. Calculated with equation (4.1).
  - Viscous friction: due to shearing of fluid films between parts of the pump moving relative to each other. Proportional to the speed of the

moving parts and the viscosity. Calculated with equation (4.2).

- Hydro-kinetic friction: due to turbulent pump flow around restrictions, bends, etc. within the pump. Proportional to the square of the flow. Calculated with equation (4.3).
- Static friction: mainly due to friction in seals. Constant.

- Volumetric efficiency: is how much of the theoretical flow the pump actually delivers. This leakage is mainly due to laminar clearance flow from the high pressure chamber to the low pressure chamber within the pump. See figure 4.2
- Total efficiency: is the combination of the volumetric and the Hydro-mechanical efficiency.

$$T_{mP/M} = K_{mP/M} \cdot \Delta P_{P/M} \tag{4.1}$$

$$T_{vP/M} = K_{vP/M} \cdot \mu \cdot n_{P/M} \tag{4.2}$$

$$T_{hP/M} = K_{hP/M} \cdot n_{P/M}^2 \tag{4.3}$$

4.2 Hydraulic Transmission Bench

A LabView program had to be constructed in order to control and perform measurements on the bench. One of the pressure transducers had to be replaced with a new one, the actuator had to be mounted back on, a second tachometer were installed on the shaft of the AC-motor and the connection between the control member on the pump and its position sensor had to be remade.

Since the Labview program were not available, it had to be reconstructed and adjusted to the need of the project. See more in section 4.16.

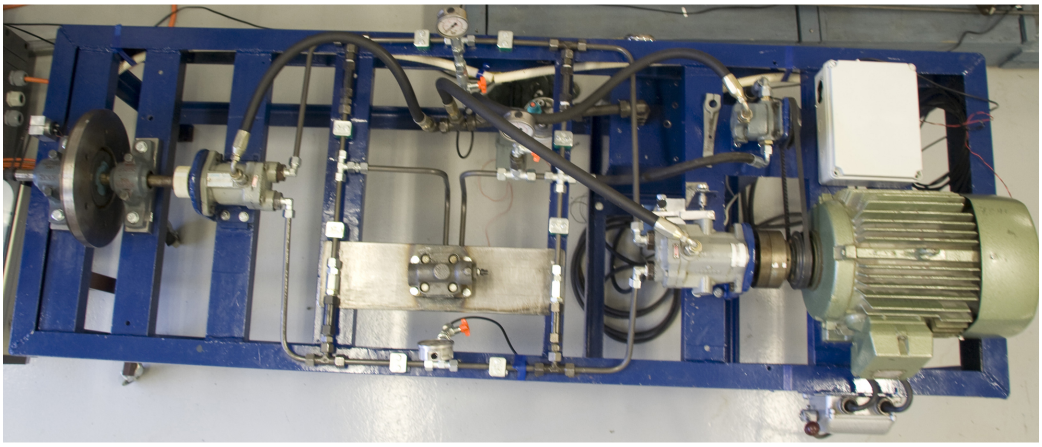


Figure 4.3: Picture of hydrostatic transmission

The bench were used for collecting measurement data to confirm the models and to estimate some friction parameters.

4.3 Main Pump

The main pump is a ”Vickers PVB5” variable displacement pump. This pump has a maximum displacement of $10,55\frac{cm^3}{rev}$ and a maximum outlet pressure of $210bar$.

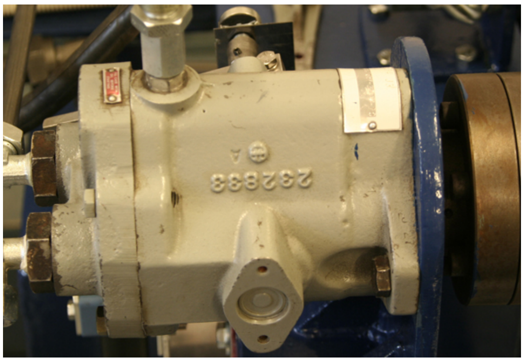


Figure 4.4: Picture of the main pump

The flow equation for the pump are presented in equation (4.4)

$$Q_P = \eta_{vP} \cdot \frac{D_P \cdot n_P}{1000} \tag{4.4}$$

The torque equation for the pump are presented in equation (4.5)

$$T_P = \frac{1}{\eta_{mhP}} \cdot \frac{D_P \cdot \Delta P_P}{62,832} \tag{4.5}$$

The volumetric efficiency η_{vP} are calculated with the equation (4.6)

$$\eta_{vP} = 1,0 - \frac{k_{leak,P} \cdot \Delta P_P}{\mu \cdot D_P \cdot n_M} \tag{4.6}$$

The hydro-mechanical efficiency η_{mhP} are calculated with the equation (4.7)

$$\eta_{mhP} = \frac{T_{tP}}{T_{tP} + T_{mP} + T_{vP} + T_{hP} + T_{sP}} \tag{4.7}$$

For the data sheet see appendix A

4.4 Boost Pump

The boost pump is a ”JSB GPA1” gear pump with internal pressure relief valve. This pump has a maximum displacement of $1,76\frac{cm^3}{rev}$ and an adjustable outlet pressure between $5-60bar$. On the transmission the pressure is adjusted to $10bar$.

For the data sheet see appendix C

4.5 Motor

The motor is a ”Vickers MFB5” Fixed displacement pump. This motor has a displacement of $10,55\frac{cm^3}{rev}$, a maximum output torque of $30,5Nm$, a maximum operating speed of $3600rpm$ and a maximum inlet pressure of $210bar$.

The flow equation for the motor are presented in equation (4.8)

$$Q_M = \frac{\eta_{vM}}{\cdot} \frac{V_M[cm^3/rev] \cdot n_M[rpm]}{1000} \tag{4.8}$$

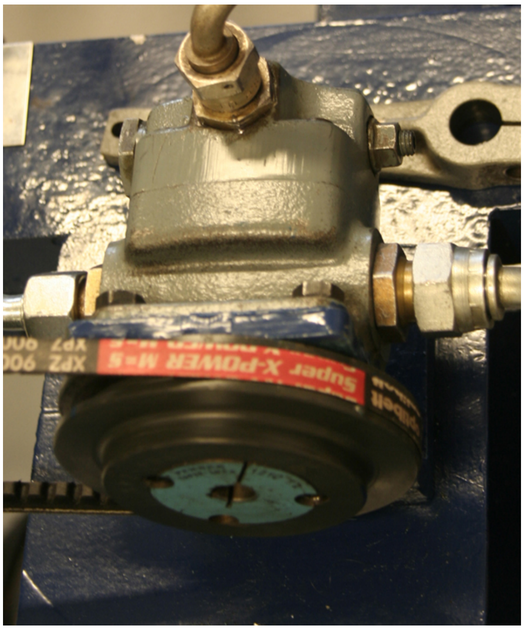


Figure 4.5: Picture of Boost pump

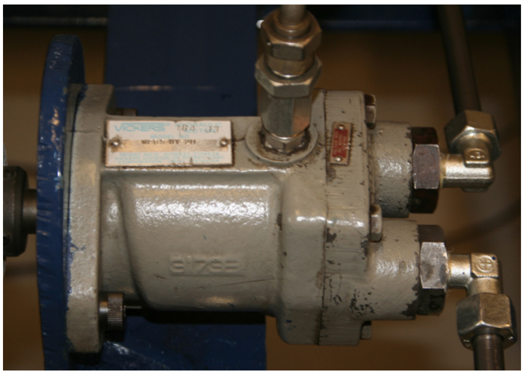


Figure 4.6: Picture of the hydraulic motor.

The torque equation for the motor are presented in equation (4.9)

$$T_M = \eta_{mhM} \cdot \frac{D_M \cdot \Delta P_M}{62,832} \tag{4.9}$$

The volumetric efficiency η_{vM} are calculated with the equation (4.10)

$$\eta_{vM} = \frac{1,0}{1,0 + \frac{k_{leak,M} \cdot \Delta P_M}{\mu \cdot D_M \cdot n_M}} \tag{4.10}$$

The hydro-mechanical efficiency η_{mhP} are calculated with the equation (4.11)

$$\eta_{mhM} = \frac{T_{tM} - T_{mM} - T_{vM} - T_{hM} - T_{sM}}{T_{tM}} \tag{4.11}$$

For the data sheet see appendix B

4.6 Check Valve

The four check valves are "Vickers DT8P1-03-5-10" and are poppet check valves with a crack pressure of $0,345bar$ and have a nominal flow of $30\frac{l}{min}$.

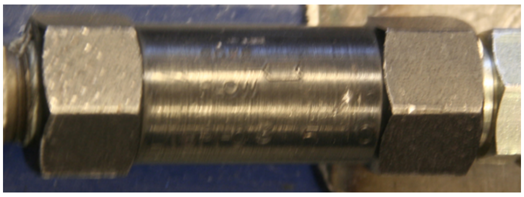


Figure 4.7: Picture of check valve

For the data sheet see appendix D

4.7 Pressure Relief Valve

The pressure relief valve is a "Vickers CG-03-CV-10" valve. It has a pressure range of $35 - 140bar$. The rated flow is $30\frac{l}{min}$ On the transmission the pressure is adjusted to $65bar$.

For the data sheet see appendix E

4.8 Pressure sensor

The two pressure sensors are "Parker SCP-400-44-07" sensors. The sensors have a pressure range of $0 - 400bar$. The sensors needs a external voltage in the range

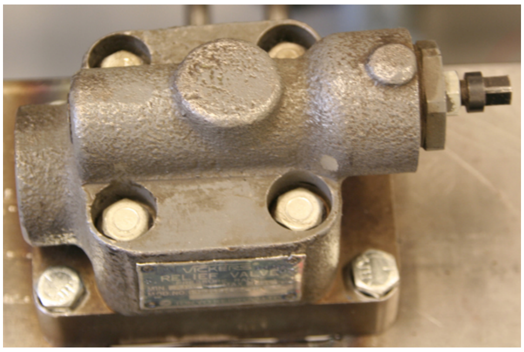
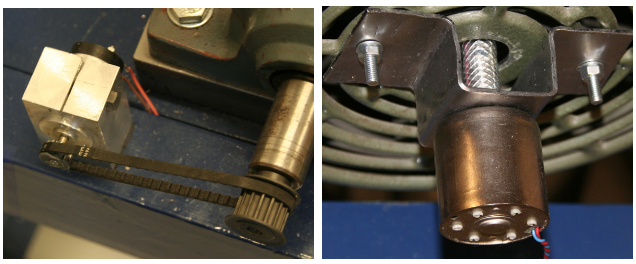


Figure 4.8: Picture of pressure relief valve

$12 - 30V$ and gives a output signal in the range $0 - 10V$.
For the data sheet see appendix F

4.9 Tachometer

The two tachometers are two DC-motors with low internal torque. The tachometer which measure the load is connected to the load with a belt and the tachometer that measures the AC-motor is connected direct to the shaft of the AC-motor i.e. it runs with the same speed as the AC-motor. This is positive because then no calculation of any gear ratios are necessary. Signals from both tachometers are going through a voltage divider to limit the voltage measured by the DAQ unit.



(a) Load tacho (b) AC tacho

Figure 4.9: Pictures of tachometers

4.10 Linear Actuator

The actuator runs on $24V$ and is controlled with the DAQ unit through a operational amplifier circuit.

4.11 DAQ unit

The DAQ unit is a ”NI USB-6008”. It has two analog output ports that can provide $0 - 5V$, and 4(differential) or 8(single-ended) analog input ports that can read voltages up to $10V$. It also have 12 digital I/O, a 32 bits counter and two outputs with constant voltage of $5\ and\ 2,5V$

For the data sheet see appendix G

4.12 Oil

The oil used in the transmission is ESSO UNIVIS n32. This is an oil with a kinematic viscosity of 32 at 40^oC

For the data sheet see appendix H

4.13 Calculation of Pressure Losses

It is important to calculate the pressure drop in the pipe to check if it is significant

To calculate the pressure drop in pipes and fittings, the flow from the pump must be calculated. Using the theoretical flow because this gives the highest speed in the pipe and the largest pressure drop. The pressure drop is only calculated for the high pressure side because it is the high pressure that drives the hydraulic motor.

$$\begin{aligned} Q_1[l/min] &= \frac{D_p[cm^3/rev] \cdot n_P[rpm]}{1000} \\ &= \frac{10,55[cm^3/rev] \cdot 1500[rpm]}{1000} = 15,825[l/min] \end{aligned} \tag{4.12}$$

Then Reynolds number are calculated to:

$$Re = 21,22 \cdot 10^3 \cdot \frac{Q[l/min]}{d[mm] \cdot \nu[cSt]} = 21,22 \cdot 10^3 \cdot \frac{15,825}{9,0 \cdot 32} = 1166 \tag{4.13}$$

The friction factor, λ, are calculated to:

$$\lambda = \frac{64}{Re} = \frac{64}{1166} = 54,89 \cdot 10^{-3} \tag{4.14}$$

The fluid speed are then calculated to:

$$v_{line}[m/s] = \frac{16,667 \cdot Q_1}{A[mm^2]} = \frac{16,667 \cdot 15,825[l/min]}{\pi \frac{9[mm]}{2}^2} = 4,146[m/s] \tag{4.15}$$

The pressure drop for the straight pipe are calculated to:

$$\begin{aligned} \Delta p_{pipe} &= \lambda \cdot \frac{l[m]}{d[m]} \cdot \rho \cdot \frac{v^2[m/s]}{2} \\ &= 54,89 \cdot 10^{-3} \cdot \frac{1,41[m]}{0,009[m]} \cdot 875,2 \cdot 10^{-6}[kg/m^3] \cdot \frac{4,146^2[m/s]}{2} \\ &= 633,55 \cdot 10^{-3}[bar] \end{aligned} \tag{4.16}$$

The pressure drop per 90^o bend is calculated to:

$$\begin{aligned} \Delta p_{bend} &= \xi \cdot \frac{\rho}{2} \cdot v_{line}^2 = 1,3 \cdot \frac{875,2[kg/m^3]}{2} \cdot 4,146^2[m/s] \\ &= 97,787 \cdot 10^{-3}[bar] \end{aligned} \tag{4.17}$$

The pressure drop per T-junction is calculated to:

$$\Delta p_{T-junc} = \xi \cdot \frac{\rho}{2} \cdot v_{line}^2 = 0,1 \cdot \frac{875,2[kg/m^3]}{2} \cdot 4,146^2[m/s]$$
$$= 7,522 \cdot 10^{-3}[bar] \tag{4.18}$$

There are 4 90^o bends and 3 T-junctions on the high pressure line the total pressure drop are calculated to:

$$\sum \Delta p_{loss} = \Delta p_{pipe} + 4 \cdot \Delta p_{bend} + 3 \cdot \Delta p_{T-junc} = 1,047[bar] \tag{4.19}$$

The total pressure drop over the high pressure line is $1bar$. This value is too large to neglect, but since this value is calculated with the theoretical flow from the pump at maximum displacement, it is a worst case value. If the volumetric efficiency of the pump is taken in to the consideration the true value is smaller.

4.14 Calculation of Inertia

The volume of the inertia are calculated in equation (4.20)

$$\begin{aligned} V &= h \cdot A \\ &= h \cdot \pi \cdot r^2 \\ &= 3cm \cdot \pi \cdot 14,5cm^2 \\ &= 1981,56cm^3 \\ &= 1981,56 \cdot 10^{-6}m^3 \end{aligned} \tag{4.20}$$

The mass of the inertia are calculated in equation (4.21)

$$\begin{aligned} m &= V \cdot \rho \\ &= 1981,56 \cdot 10^{-6} m^3 \cdot 7850 \frac{kg}{m^3} \\ &= 15,555 kg \end{aligned} \tag{4.21}$$

The moment of inertia are calculated in equation (4.22)

$$\begin{aligned} J &= \frac{m \cdot r^2}{2} \\ &= \frac{15,555 kg \cdot (0,145 m)^2}{2} \\ &= 0,1635 kgm^2 \end{aligned} \tag{4.22}$$

4.15 Calculation of the Dynamic Viscosity

The dynamic viscosity are calculated in equation (4.23)

$$\begin{aligned} Dynamic\ viscosity &= Kinematic\ viscosity \cdot density \\ \mu &= \nu \cdot \mu \cdot density \\ &= 31,8106 \cdot 10^{-6} \frac{m^2}{s} \cdot 857,2 \frac{kg}{m^3} \cdot density \\ &= 0,000016 [bar \cdot min] \end{aligned} \tag{4.23}$$

4.16 LabVIEW Implementation

As mention in chapter 4.2, the control system were programmed in LabVIEW. LabVIEW is a platform and development environment for a visual programming language from National Instruments. LabVIEW is commonly used for data acquisition, instrument control, and industrial automation.

The outputs from the program are rotational speed of the load and the AC-motor, the A and B pressures and the position of the control member. There are implemented a saving function which saves the outputs to a file.

The control of the actuator can either be controlled manually or by a generated sine wave. The control signal to the actuator is a voltage between 0V and 5V were 2,5V being the ”stop” signal

4.16.1 The Program

The program consists of 3 different while-loops.

The first loop, shown in figure 4.10, is the reading loop. This loop reads in the measurements from the rig, scale and corrects offset errors in the measurements, and project them in instruments and/or graphs.

Since each measurements consists of 100 samples, the measurement needs to be taken a mean of. The measurement needs to be filtered to remove high frequency measurement noise. Each of the pressure measurements needs to be scaled up by multiplying by 40. The speed of the load needs to be scaled up by 205 and offset by 2,5. The AC motor needs to be scaled up by 151 and offset by 12,4. The measurement needs to be offset by the value in the block ”Fine Tune”. This is to set the measurement to 2,5 when the pump is set to zero displacement. The displacement is calculated from the value of the potentiometer.

The second loop, shown in figure 4.11, is the saving loop. This loop saves a set of parameters to a text file.

The ”Delay time” value sets the time step between each data-point, the ”Lagre” switch starts and stop the saving procedure. In the case structure, the ”Time Target” value sets the timespan of the saving. The time, in addition with the different measurements, are combined in an array and written to a text file.

The last loop, shown in figure 4.12, is the writing loop. This loop writes a signal to the rig which controls the actuator.

The control signal can be either a sine wave or a manually controlled signal. This is chosen by the switch ”Sinus?”. The ”start” switch restarts the sine wave

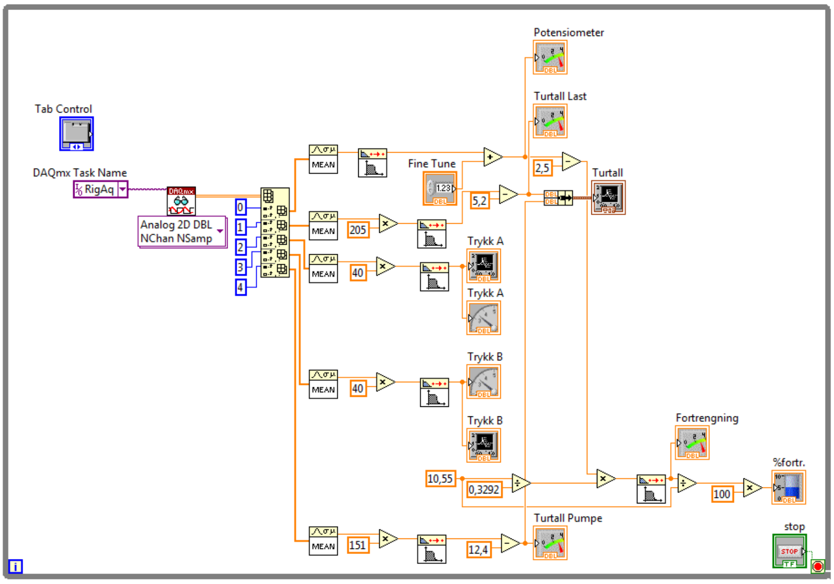


Figure 4.10: Reading loop

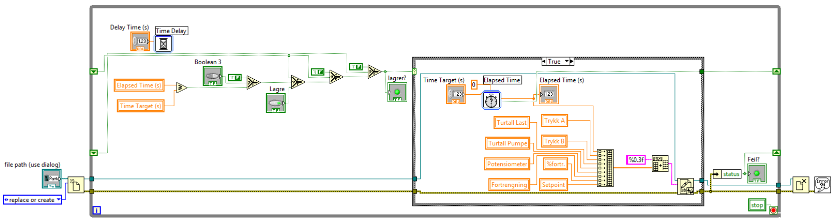


Figure 4.11: Saving loop

at zero. In the case structure the generation of the sine wave is performed. To the ”Elapsed Time” block there are connected a zero constant, to set the start time to zero, and a boolean false constant , to prevent the block from auto resetting. It also gets a reset signal from the ”Start” switch. To the ”Sine Wave PtByPt” block there are connected three signals; Amplitude, frequency and the signal from the ”Elapsed Time” block. The output from the ”Sine Wave PtByPt” is added with 2,5 to offset the signal.

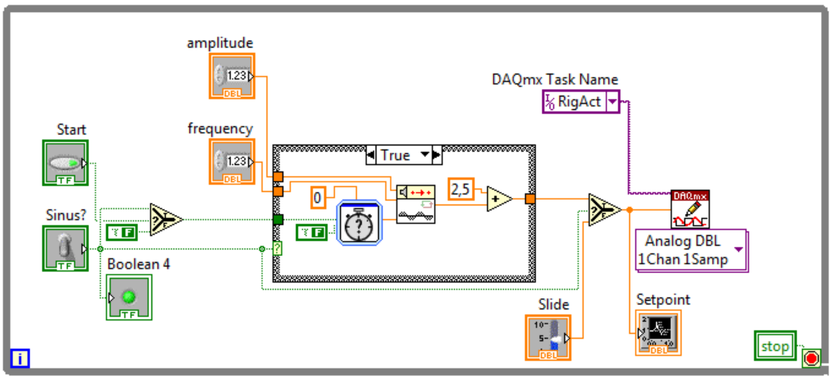


Figure 4.12: Writing loop

Chapter 5

Simulation Models

A dynamic model of the hydrostatic transmission has been created with three different simulation tools; SimulationX, SimHydraulics and Simulink/Matlab. The simulation results from the different models will be compared to see if they will give the same results.

A steady-state model of the hydrostatic bench has also been created in order to do steady-state simulation of the hydrostatic bench. The results from the steadystate simulation will give an indication if the dynamic models give a rational result or not for different operating conditions. The results are also used to set the initial conditions for the unknown parameters for the dynamic models. The steady-state simulation does not include any dynamic behavior of the system such as compressibility of fluid and pressure peaks.

5.1 SimHydraulics

SimHydraulics is a block package under the Simscape toolbox in Simulink which is used to model up a hydraulic system. There are also used blocks from SimMechanics, another block package under Simscape, to model the mechanical components connected to the hydraulic system.

The SimHydraulics model is shown in figure 5.1 to figure 5.8. The model is constructed with the same components as the actual bench, with some exceptions like the flow and torque measurements which are not included on the bench. The low pressure measuring is only an analog gauge, not a pressure transducer. The boost pump includes the pressure relief valve on the actual rig, but in the model they are two separate blocks (see figure 5.7) The parameters for the model can be found in appendix J

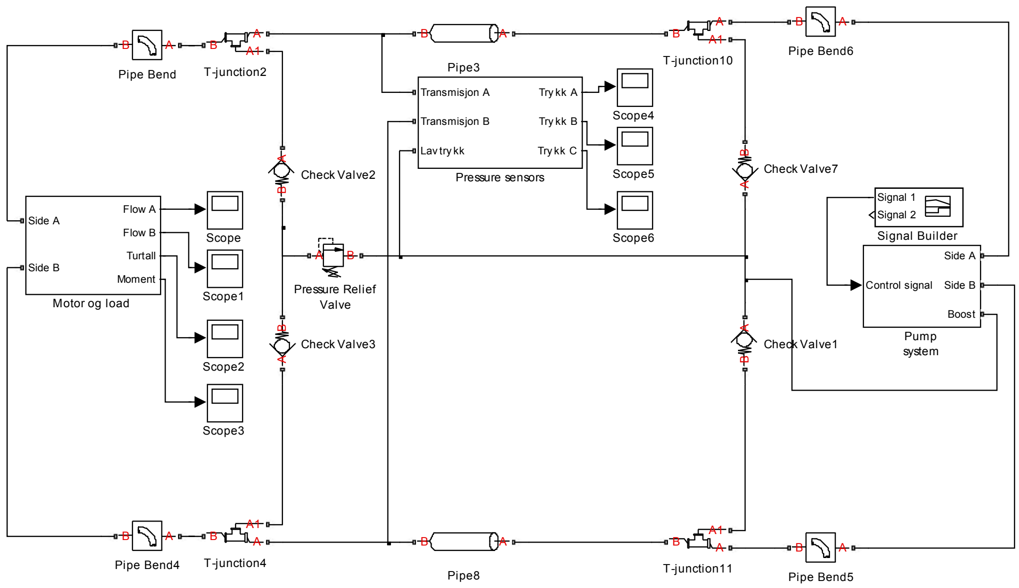


Figure 5.1: The main model.

Some gains are included in the model which are used to convert from SI units to hydraulic units so that the results are easier to compare.

The motor and load subsystem contains the hydraulic motor, the flow measurement subsystem and the load subsystem. The motor gets flow from the flow measurement subsystem and delivers rotation to the load sub system.

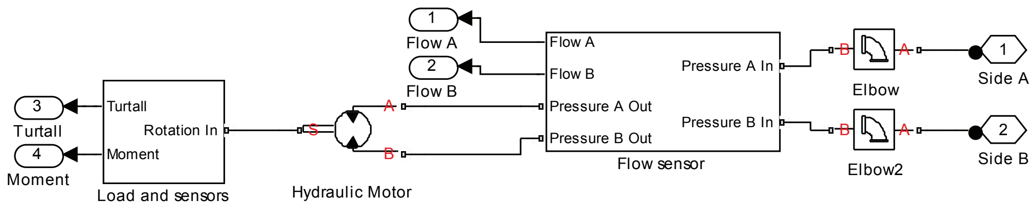


Figure 5.2: The motor and load subsystem

In the load and sensor subsystem the inertia is provided and the torque and rotational speed are measured.

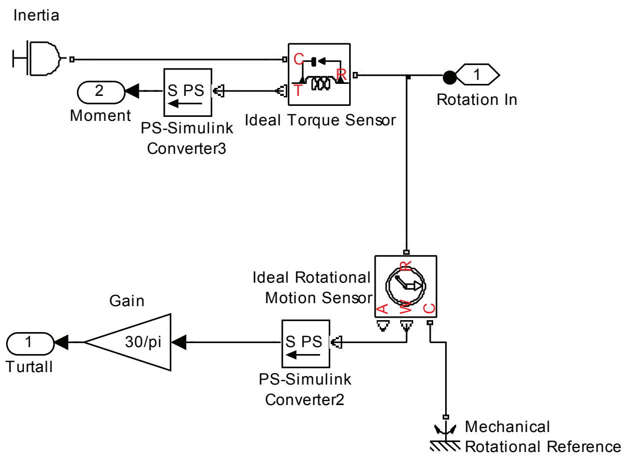


Figure 5.3: The load and sensor subsystem.

The flow sensor subsystem measures the flow to and from the hydraulic motor.

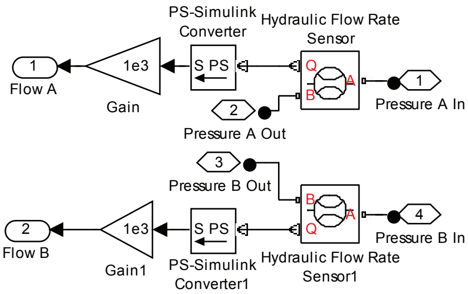


Figure 5.4: The flow sensor subsystem.

The Pressure sensor subsystem measures the 4 pressures in the system.

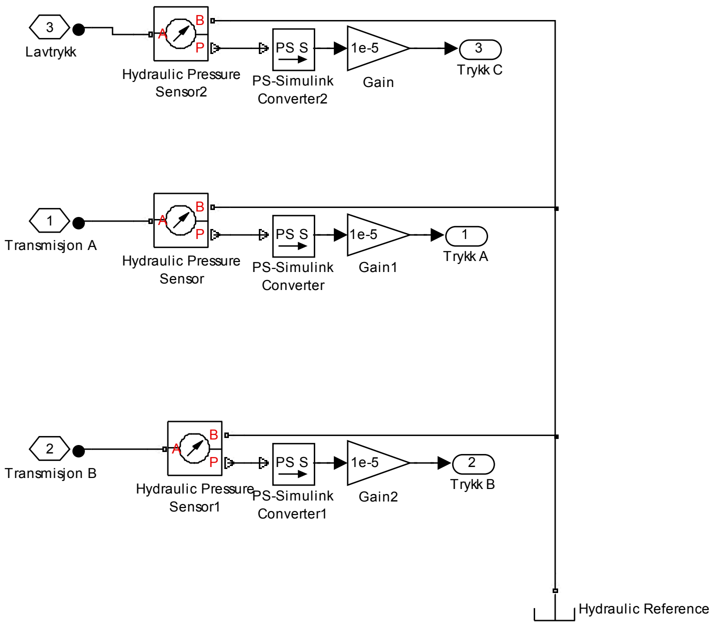


Figure 5.5: The pressure sensor subsystem.

The pump system contains the variable displacement pump, the boost pump subsystem and the electro motor subsystem. The pump get rotational speed and control signal in and provides the system with flow.

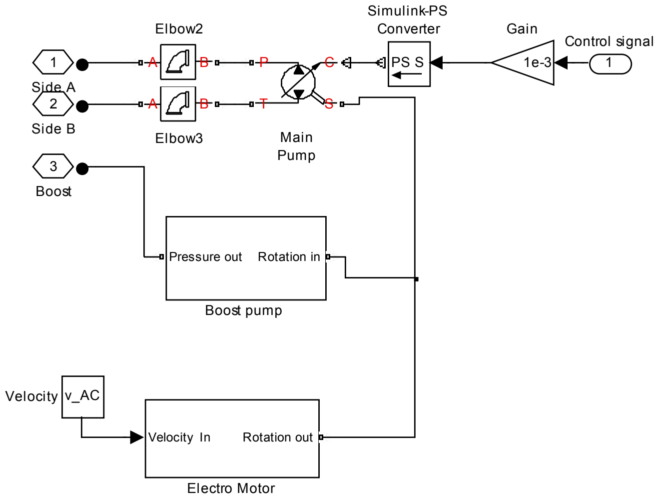


Figure 5.6: The pump system subsystem.

The Boost pump subsystem contains the boost pump and its pressure relief valve. It also contains the hydraulic fluid reference and the solver configuration block.

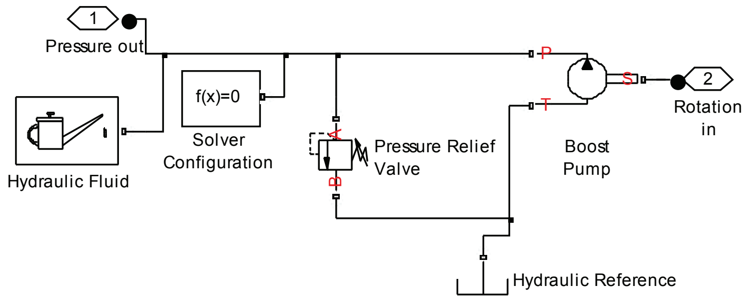


Figure 5.7: The boost pump subsystem.

The electro motor subsystem is constructed with the ”Ideal Angular Velocity Source” and the ”mechanical reference” blocks. The Angular velocity source is connected to the mechanical reference and the angular velocity reference signal.

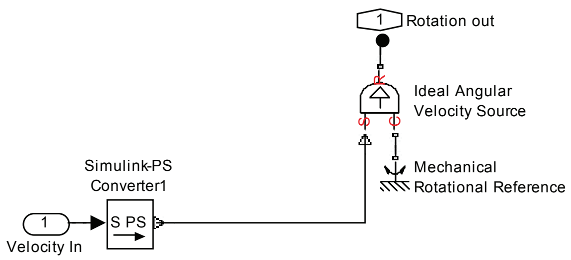


Figure 5.8: The electro motor subsystem.

5.2 SimulationX

SimulationX is a physical system simulation program which is well suited for simulating advanced mechatronic systems. SimulationX include libraries for among other hydraulics, mechanics, pneumatics and electronics. SimulationX also supports the creation of self created components.

The SimulationX model is build up by the same components as the physical transmission. Another difference is that on the physical transmission the boost pump has the pressure relief valve built into the unit, but in the model the boost pump and the pressure relief valve is two separate blocks. The model does not include bends and t-junctions because these components are not included in "Proffesional" edition of SimulationX. The model is shown in fig.5.9 The parameters for the model can be found in appendix I.

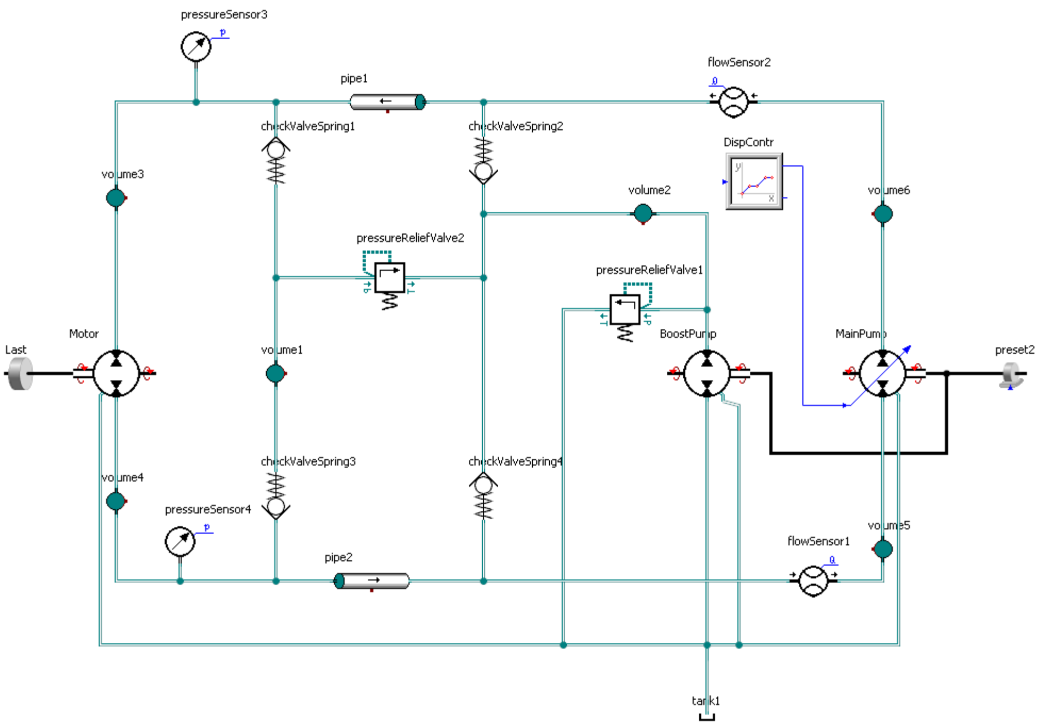


Figure 5.9: SimX model 1.

5.3 Steady-State Modeling

Following simplifications was made to create a steady-state model:

1. The fluid is incompressible
2. The spring of the check valves are not included
3. The pressure relief valve characteristics are not included

In the steady-state modeling the p - Q characteristics of the check valves and the pressure relief valves are ignored. The volumetric efficiencies of the pump and motor are included through the leakage flow within the pump $Q_{leak,P}$ and across the motor $Q_{leak,M}$. The hydro-mechanical efficiency of the motor is included through the losses in the output torque due to mechanical, viscous, turbulent and static friction.

The inputs to steady-state model are the displacement of the pump D_p and the rotational speed of the pump n_P. The 4 pressures, 11 flows and the rotational speed of the motor n_M are the outputs. The system is described by 16 equations. The equations are solved to find the 16 unknowns; 11 flows + 4 pressures + rotational speed for the hydraulic motor, n_M. The configuration parameters for all the valves have been chosen, since they have two operation modes.

A step wise approach as described in "Hydraulic Components and Systems chapter 6"[1] is used to perform a steady-state analysis of the hydrostatic transmission shown in Figure 5.10.

Step 1: Pressure nodes

The first step is to identify pressure nodes. The 4 pressure nodes (P1, P2, P3 and P4) are shown in Figure 5.10.

Step 2: Identify components

Identify the components that surround each pressure node and the different flow variables.

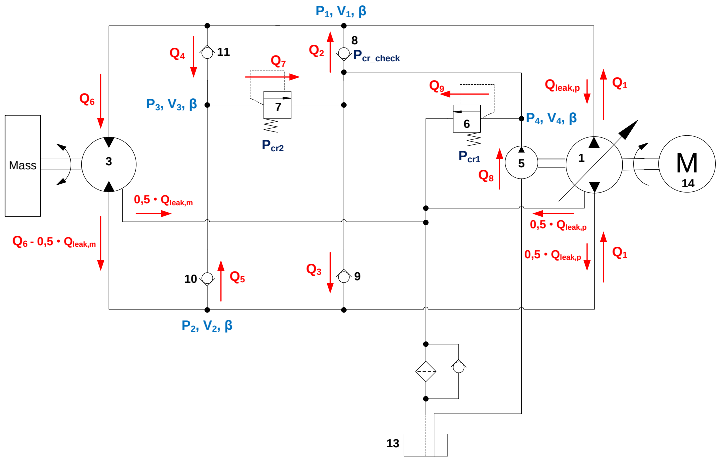


Figure 5.10: Hydrostatic transmission with flows, volumes and pressure nodes

Table 5.1: Pressure nodes

Pressure node P1	Pressure node P2	Pressure node P3	Pressure node P4
Pump, Q1	Pump, Q1	Relief valve, Q7	Boost pump, Q8
Check valve, Q2	Check valve, Q3	Check valve, Q4	Check valve, Q2
Check valve, Q4	Check valve, Q5	Check valve, Q5	Check valve, Q3
Motor, Q6	Motor, Q6		Relief valve, Q7
			Relief valve, Q9

Step 3: configuration parameters

Choose configuration parameters for all the components with more than one mode of operation. It is guessed that the relief valve (6) is open in order to keep a minimum system pressure of 10 bar. The pressure P4 is therefore set equal the crack pressure $P_{cr1} = 10bar$. The check valve (9) is open to refill the half leakage flow from both pump and motor to the low pressure side, P_2, when the pressure P_2 will fall under 9,65 bar $(P_4 - Pcr_check = 10bar - 0,35bar = 9,65bar)$. For the steady state condition both acceleration and braking are not considered. High pressure peaks will therefore not occur, the check valves, (11) and (10), and the relief valve (7) will be closed.

Table 5.2: Configuration parameters

Component	Configuration	Equation
Relief valve, Q9	Open	P4 = Pcr1 = 10 bar
Relief valve, Q7	Closed	Q7 = 0
Check valve, Q2	Closed	Q2 = 0
Check valve, Q3	Open	$P_2 = P_4 - P_{cr_check}$
Check valve, Q4	Closed	Q4 = 0
Check valve, Q5	Closed	Q5 = 0

Step 4: Equations

1. Equations for flow continuity for each pressure node:

$$P_1 : Q_1 - Q_{leak,P} + Q_2 - Q_4 - Q_6 = 0 \tag{5.1}$$

$$P_2 : Q_6 - 0,5 \cdot Q_{leak,M} - Q_5 + Q_3 - Q_1 + 0,5 \cdot Q_{leak,P} = 0 \tag{5.2}$$

$$P_3 : Q_4 + Q_5 - Q_7 = 0 \tag{5.3}$$

$$P_4 : Q_8 + Q_7 - Q_9 - Q_2 - Q_3 = 0 \tag{5.4}$$

2. Equations for flow continuity for pump and actuator (hydraulic motor):

$$Pump : Q_1 = \frac{n_P \cdot D_P}{1000} \Rightarrow Q_1 - \frac{n_P \cdot D_P}{1000} = 0 \tag{5.5}$$

$$Boost\ pump : Q_8 = \frac{n_{BP} \cdot D_{BP}}{1000} \Rightarrow Q_8 - \frac{n_{BP} \cdot D_{BP}}{1000} = 0 \tag{5.6}$$

$$Hydraulic\,motor : Q_6 = \frac{n_M \cdot D_M}{1000} \Rightarrow Q_6 - \frac{n_M \cdot D_M}{1000} = 0 \tag{5.7}$$

3. Static equilibrium equation for the actuator (hydraulic motor):

   Since there is no acceleration at steady-state the sum of torques will be equal zero. The theoretical output torque from the motor T_{tM} will then be equal the internal friction losses for the motor, $T_{loss\_M} = T_{tM}$. It is assumed that the losses in bearings and the tachometer are very small, therefore they are not considered. The internal friction losses include mechanical, viscous, turbulent and static friction.

$$J_{eff} \cdot \ddot{\theta}_M = T_{tM} - T_{loss_M} = 0 \tag{5.8}$$

$$\begin{aligned}\frac{D_M \cdot (P_1 - P_2)}{62,83} &= T_{mM} + T_{vM} + T_{hM} + T_{sM} \\ &= K_{mM} \cdot (P_1 - P_2) + K_{vM} \cdot \mu \cdot n_M \\ &+ K_{hM} \cdot n_M^2 + T_{sM}\end{aligned} \tag{5.9}$$

$$\frac{D_M \cdot (P_1 - P_2)}{62,83} - K_{mM} \cdot (P_1 - P_2) - K_{vM} \cdot \mu \cdot n_M - K_{hM} \cdot n_M^2 - T_{sM} = 0 \tag{5.10}$$

   The motor dependent constants for friction $K_{mM},\ K_{vM},\ K_{hM}\ and\ T_{sM}$ are estimated in chapter 7

4. Equations for leakage flow for pump and motor:

$$Q_{leak,M} = k_{leak,M} \cdot \frac{(p_1 - p_2)}{\frac{\mu}{\mu_{ref}}} \Rightarrow Q_{leak,M} - k_{leak,M} \cdot \frac{(p_1 - p_2)}{\frac{\mu}{\mu_{ref}}} = 0 \tag{5.11}$$

$$Q_{leak,P} = k_{leak,P} \cdot \frac{(p_1 - p_2)}{\frac{\mu}{\mu_{ref}}} \Rightarrow Q_{leak,P} - k_{leak,P} \cdot \frac{(p_1 - p_2)}{\frac{\mu}{\mu_{ref}}} = 0 \quad (5.12)$$

5. Equations related to the configuration parameters in step 3:

$$P_4 = P_{cr1} \Rightarrow \; P_{cr1} - \; P_4 = 0 \quad (5.13)$$

$$Q7 = 0 \quad (5.14)$$

$$Q2 = 0 \quad (5.15)$$

Set the pressure P_3 to be 1 bar less than the crack pressure of the relief valve (7), since the relief valve (7) should be closed.

$$P_3 = P_{cr2} - 1 \Rightarrow \; P_3 - P_{cr2} + 1 = 0 \quad (5.16)$$

$$Q4 = 0 \quad (5.17)$$

$$P_2 = P_4 - P_c r_c heck \Rightarrow \; P_2 + P_c r_c heck - P_4 = 0 \quad (5.18)$$

The result from step 4 is 16 equations. These equations are solved in Matlab (See appendixK for the code) to find the 16 unknown parameters (11 flows + 4 pressures + rotational speed for the hydraulic motor, n_M). The results from the steady-state simulation are shown in chapter 6.

The leakage coefficients for the hydraulic motor $k_{leak,M}$ and pump $k_{leak,P}$ were calculated to find good starting values before tuning the parameters.

The pressures $P1$ and $P2$ are found by taking the average values of the measurements from the hydrostatic bench, $P1$ = 19,72 bar and $P2$ = 8,67 bar. Find the leakage flow for the pump when the displacement is at the maximum, $D_p = 10,55\frac{cm^3}{rev}$. The volumetric efficiency η_{vP} for the pump is found to be 99% from the datasheet in appendix A for a rotational speed of 1500 $\frac{rev}{min}$ when the outlet pressure $P1$ is approximately 20 bar.

The leakage flow for the pump with maximum displacement is:

$$\begin{aligned} Q_{leak,P} &= \frac{n_p \cdot D_p}{1000} \cdot (1-\eta_{vP}) \\ &= \frac{1500\ \frac{rev}{min} \cdot 10,55\ \frac{cm^3}{rev}}{1000} \cdot (1-0,99) = 0,1583\ \frac{l}{min} \end{aligned} \tag{5.19}$$

Find the leakage coefficient for the pump from equation 3.10:

$$k_{leak,P} = \frac{Q_{leak,M}}{p_1 - p_2} = \frac{0,1583\ \frac{l}{min}}{19,72\ bar - 8,67\ bar} = 0,0143\ \frac{l}{min \cdot bar} \tag{5.20}$$

Found the leakage flow for the hydraulic motor when the displacement for the pump is at the maximum, $D_p = 10,55\frac{cm^3}{rev}$. The rotational speed of the motor $n_M = 1454\frac{rev}{min}$ are found by taking the average of the measurements from the hydrostatic bench when $D_p = 10,55\frac{cm^3}{rev}$.

The volumetric efficiency for the motor is found to be 96% from the datasheet in appendix B when the rotational speed of the motor $n_M = 1454\frac{rev}{min}$.

The leakage flow for the pump with maximum displacement is:

$$\begin{aligned} Q_{leak,M} &= \frac{n_M \cdot D_M}{1000} \cdot (1-\eta_{vM}) \\ &= \frac{1454\ \frac{rev}{min} \cdot 10,55\ \frac{cm^3}{rev}}{1000} \cdot (1-0,96) = 0,6136\ \frac{l}{min} \end{aligned} \tag{5.21}$$

Find the leakage coefficient for the motor from equation (5.11):

$$k_{leak,M} = \frac{Q_{leak,M}}{p_1 - p_2} = \frac{0,6136\ \frac{l}{min}}{19,72\ bar - 8,67\ bar} = 0,0555\ \frac{l}{min \cdot bar} \tag{5.22}$$

Approximately value of the hydro-mechanical losses, T_{loss_M}, for the hydraulic motor was found to be equal 1,69 Nm by using SimulationX. In SimulationX it is possible to find the effective torque for the hydraulic motor. The hydro-mechanical losses were found by subtracting the effective torque from the theoretical torque. The value of the hydro-mechanical losses from SimualtionX was trusted since the simulation results from SimulationX were quite close to the measurements from the hydrostatic transmission as shown in chapter 6. The value of the hydro-mechanical losses will be used as a starting value when tuning the models after the measurements.

5.4 Simulink

To create a dynamic model of the hydrostatic system in Simulink several equations must be established. The losses in pipes and bends are ignored. The dynamical model includes the acceleration of the load and the dynamics of the valves. The fluid is considered compressible. This equation describes the volume of fluid for each pressure node.

$$\dot{p} = \frac{\beta \cdot (Q - \dot{V})}{V} \tag{5.23}$$

For the hydrostatic transmission the displacements flow $\dot{V} = 0$. Equation (5.23) can then be rewritten to

$$\dot{p} = \frac{\beta}{V} \cdot Q \tag{5.24}$$

Pressure build up equations for each pressure node:

$$\begin{aligned}\dot{p}_1 &= \frac{\beta}{V_1} \cdot (Q_1 + Q_2 - Q_4 - Q_6 - Q_{leak,P}) \\ \rightarrow p_1 &= \int \frac{\beta}{V_1} \cdot (Q_1 + Q_2 - Q_4 - Q_6 - Q_{leak,P})\end{aligned} \tag{5.25}$$

$$\begin{aligned}\dot{p}_2 &= \frac{\beta}{V_2} \cdot (Q_6 + Q_3 - 0,5 \cdot Q_{leak,M} + 0,5 \cdot Q_{leak,P} - Q_5 - Q_1) \\ \rightarrow p_2 &= \int \frac{\beta}{V_2} \cdot (Q_6 + Q_3 - 0,5 \cdot Q_{leak,M} + 0,5 \cdot Q_{leak,P} - Q_5 - Q_1)\end{aligned} \tag{5.26}$$

$$\begin{aligned}\dot{p}_3 &= \frac{\beta}{V_3} \cdot (Q_4 + Q_5 - Q_7) \\ \rightarrow p_3 &= \int \frac{\beta}{V_3} \cdot (Q_4 + Q_5 - Q_7)\end{aligned} \tag{5.27}$$

$$\begin{aligned}\dot{p}_4 &= \frac{\beta}{V_4} \cdot (Q_8 + Q_7 - Q_9 - Q_2 - Q_3) \\ \rightarrow p_4 &= \int \frac{\beta}{V_4} \cdot (Q_8 + Q_7 - Q_2 - Q_3 - Q_9)\end{aligned} \tag{5.28}$$

The respective block diagram from Simulink for calculating the pressure P4 is shown below. Use the same way to find the three other pressures.

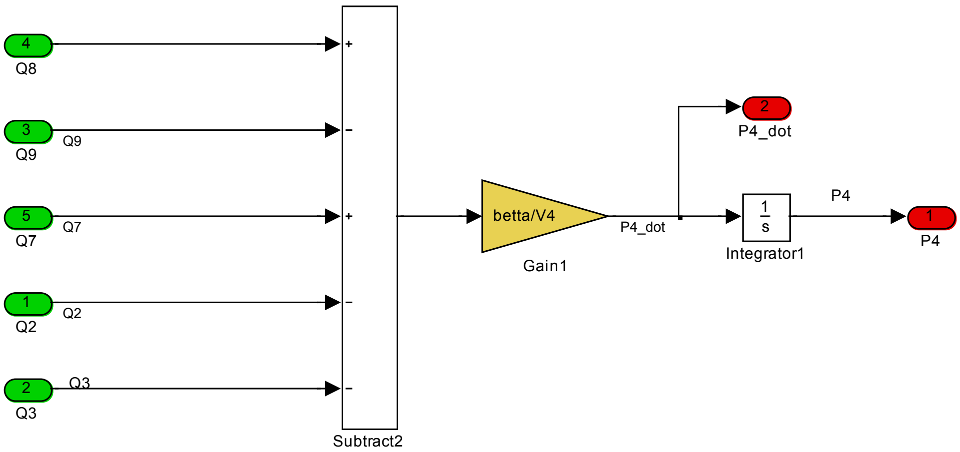


Figure 5.11: Pressure calculation

Equations for flow continuity for pumps and hydraulic motor:

$$Pump : Q_1 = \frac{n_P \cdot D_P}{1000} \tag{5.29}$$

The block diagram in Simulink for calculating the flow Q_1:

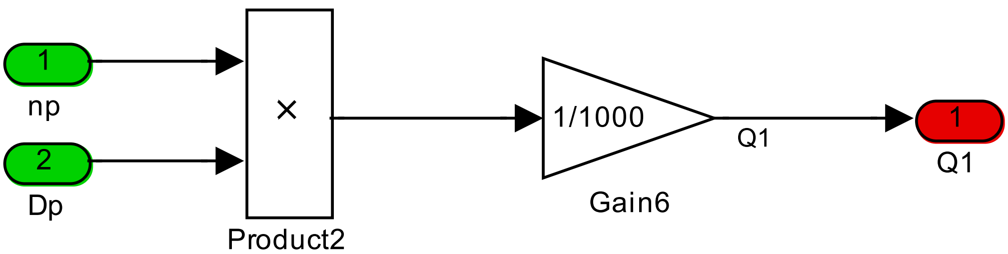


Figure 5.12: Q_1 calculation in Simulink

$$Boost\ pump : Q_8 = \frac{n_{BP} \cdot D_{BP}}{1000} \tag{5.30}$$

$$Hydraulic\, motor : Q_6 = \frac{n_M \cdot D_M}{1000} \tag{5.31}$$

Dynamic equilibrium equation for the hydraulic motor:

Hydraulic motor:

$$
\begin{aligned}
J_{eff} \cdot \ddot{\theta}_M &= T_{tM} - T_{loss\_M} \\
&= \frac{D_M \cdot (P_1 - P_2)}{62,83} - [T_{mM} + T_{vM} + T_{hM} + T_{sM}] \\
&= \frac{D_M \cdot (P_1 - P_2)}{62,83} \\
&\quad - \left[K_{mM} \cdot (P_1 - P_2) + K_{vM} \cdot \mu \cdot n_M + K_{hM} \cdot n_M^2 + T_{sM}\right] \\
\rightarrow \ \dot{\theta}_M &= \frac{1}{J_{eff}} \int \left[\frac{D_M \cdot (P_1 - P_2)}{62,83} - T_{loss\_M}\right] \qquad (5.32)
\end{aligned}
$$

The block diagram in Simulink for calculating the rotational speed of the motor, n_M, is shown below:

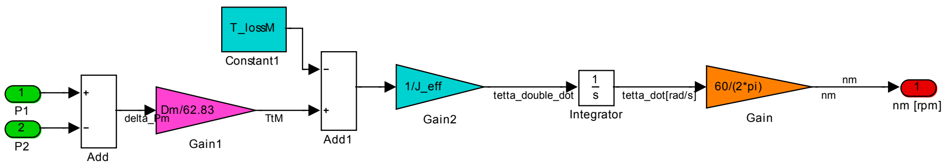


Figure 5.13: Calculation of n_m in Simulink

The slope of the $\Delta P - Q$ curves of the four check valves (See appendix D) can be included in order to get the correct flow from the check valves when the crack pressure is reached. These curves can be included in the Simulink model by using the function "Lookup table".

$$P_2 - P_3 > P_{cr\_check} \ \Rightarrow \ Q_5 = (P_2 - P_3) \cdot K_{\Delta P - Qcurve} \qquad (5.33)$$

$$P_2 - P_3 \ \leq \ P_{cr\_check} \ \Rightarrow \ Q_5 = 0 \qquad (5.34)$$

$$P_4 - P_2 > P_{cr\_check} \;\Rightarrow\; Q_3 = (P_4 - P_2) \cdot K_{\triangle P-Q\ curve} \tag{5.35}$$

$$P_4 - P_2 \leq \; P_{cr\_check} \;\Rightarrow\; Q_3 = 0 \tag{5.36}$$

$$P_4 - P_1 > P_{cr\_check} \;\rightarrow\; Q_2 = (P_4 - P_1) \cdot K_{\triangle P-Q\ curve} \tag{5.37}$$

$$P_4 - P_1 \leq \; P_{cr\_check} \;\Rightarrow\; Q_2 = 0 \tag{5.38}$$

The Simulink block for calculating the flow over a check valve is shown below:

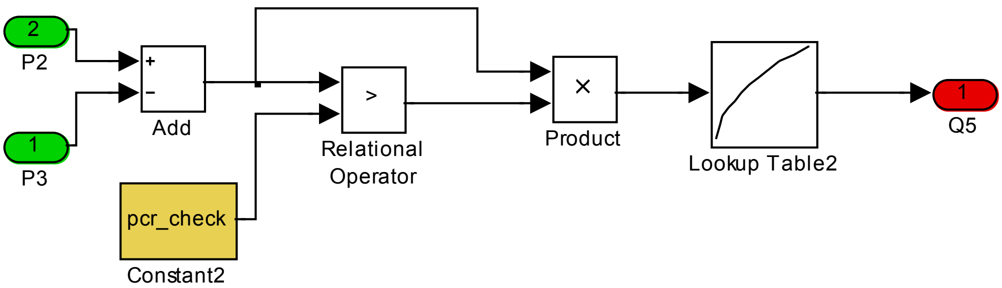


Figure 5.14: Flow over check valve calculation in Simulink

The $\Delta P - Q$ curves of the two pressure relief valves are not found in any data sheet. Since the simulation results from the SimulationX model is very close to the measurements of the real system. The pressure drops for the pressure relief valves and corresponding flows are found in SimulationX. The slope coefficients of the valve characteristics can then be calculated.

Pressure relief valve (7):

$$P_3 > P_c r2 \Rightarrow Q_7 = K \cdot \sqrt{(frac2\rho) \cdot (P_3 - p_4)} \tag{5.39}$$

$$P_3 \leq P_{cr2} \Rightarrow Q_7 = 0 \tag{5.40}$$

Where K = 15,4847

The Simulink model for calculating the flow Q_7 is shown below:

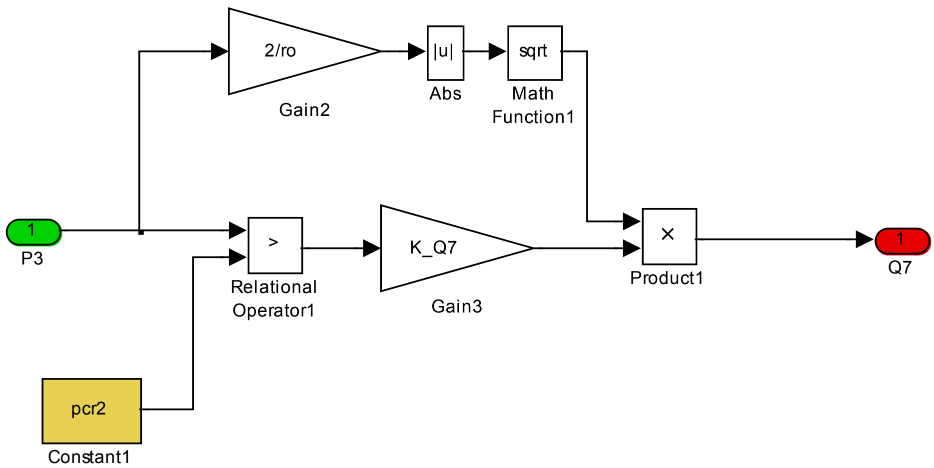


Figure 5.15: Flow Q_7 calculated in Simulink

Pressure relief valve (6):

$$P_4 > P_c r1 \Rightarrow Q_9 = K \cdot \sqrt{(frac2\rho) \cdot p_4} \tag{5.41}$$

$$P_4 \leq P_{cr1} \Rightarrow Q_9 = 0 \tag{5.42}$$

Where K = 13,03

The volumetric efficiencies for the motor and pump are found from the datasheets. These curves can be included in the Simulink model by using ”Lookup table”.

Leakage flow motor:

$$Q_{leak,M} = \frac{1}{\eta_{vM}} \cdot Q_6 - Q_6 \tag{5.43}$$

The volumetric efficiency for the motor is found in appendix B and included in the Simulink model by using Lookup table.

Leakage flow pump

$$Q_{leak,P} = Q_1 - \eta_{vP} \cdot Q_1 \tag{5.44}$$

The Simulink block for finding the leakage flow Qleak,p is shown below:

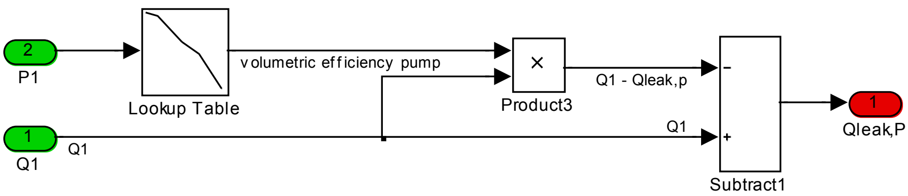


Figure 5.16: $Q_{leak,p}$

The volumetric efficiency for the pump is found in appendix A and included in the Simulink model by using Lookup table.

Chapter 6

Simulation Results

This chapter includes the results from the steady state simulation and the simulations in SimulationX, SimHydraulics and Matlab/Simulink.

6.1 Steady State Verification

The results from the steady state simulation are compared with the chosen configuration parameters in chapter 5.3 to verify if the parameters are fulfilled or not. The Steady state results for a pump displacement, $D_P = 5,275cm^3/rev$ (50% of max displacement) are shown in Table 6.1. The result verifies that all the chosen configurations parameters are fulfilled.

Table 6.1: Results steady state simulation

Parameter	**Value**
$Q1$	7,91 $\frac{l}{min}$
$Q2$	0 $\frac{l}{min}$
$Q3$	0,73 $\frac{l}{min}$
$Q4$	0 $\frac{l}{min}$
$Q5$	0 $\frac{l}{min}$
$Q6$	7,02 $\frac{l}{min}$
$Q7$	0 $\frac{l}{min}$
$Q8$	2,55 $\frac{l}{min}$
$Q9$	1,82 $\frac{l}{min}$
Q_leak, P	0,896 $\frac{l}{min}$
Q_leak, M	0,559 $\frac{l}{min}$
$P1$	19,72 bar
$P2$	9,65 bar
$P3$	64 bar
$P4$	10 bar
n_M	665,12 $\frac{rev}{min}$

The measured values for the pressures, P_1 and P_2, and the rotational speed n_M for a pump displacement equal $5,275\frac{cm^3}{rev}$ are shown in Table 6.2. These measurements were used to find the correct leakage coefficients for pump and motor, and the total hydro-mechanical loss for the motor. Table 6.3 contains the parameters that were found. These values were found by adjusting them until the pressure difference $(P_1 - P_2)$ and the rotational speed, n_M, were equal the measured values

To verify the leakage coefficients and the hydro-mechanical loss, a simulation

Table 6.2: Measurements, pump displacement equal $5,275cm^3/rev$

Parameter	Value
P_1	19,4 bar
P_2	8,6bar
n_M	665,1 $\frac{rev}{min}$

Table 6.3: Leakage coefficients for the pump and motor and total hydro-mechanical loss

Parameter	Value
k_leak, P	0,896 $\frac{l}{min \cdot bar}$
k_leak, M	0,559 $\frac{l}{min \cdot bar}$
$T_{loss,M}$	1,691 Nm

with a pump displacement equal $2,6375\frac{cm^3}{rev}$ was done. The simulation results are shown in Table 6.4. The results from the measurement for the same displacement are shown in Table 6.5. There is a difference of $24\frac{rev}{min}$ between the simulated and measured value of the rotational speed. This difference may be caused by inaccurate measurements. The pressure differences, $P_1 - P_2$, are almost equal for all the simulation and the measurement results. The pressure differences become constant for any displacement in steady state, since there is no acceleration.

Table 6.4: Simulations results, pump displacement equal $2,6375cm^3/rev$

Parameter	Value
P_1	19,72 bar
P_2	9,65bar
n_M	290,12 $\frac{rev}{min}$

Table 6.5: Measurements, pump displacement equal $2,6375cm^3/rev$

Parameter	Value
P_1	18,98 bar
P_2	8,5bar
n_M	314 $\frac{rev}{min}$

6.2 Comparison

Since there are differences in which parameters each model needs, how they are added and how each model is configured, it is difficult, if not impossible; to give each model the exact same running conditions hence giving somewhat different results.

The simulations results from SimHydraulics, SimulationX and the measurements from the hydrostatic transmission with a ramp are shown in the figures 6.1-6.3. The results from the simulink model is not included since there was simulation problems with that model.

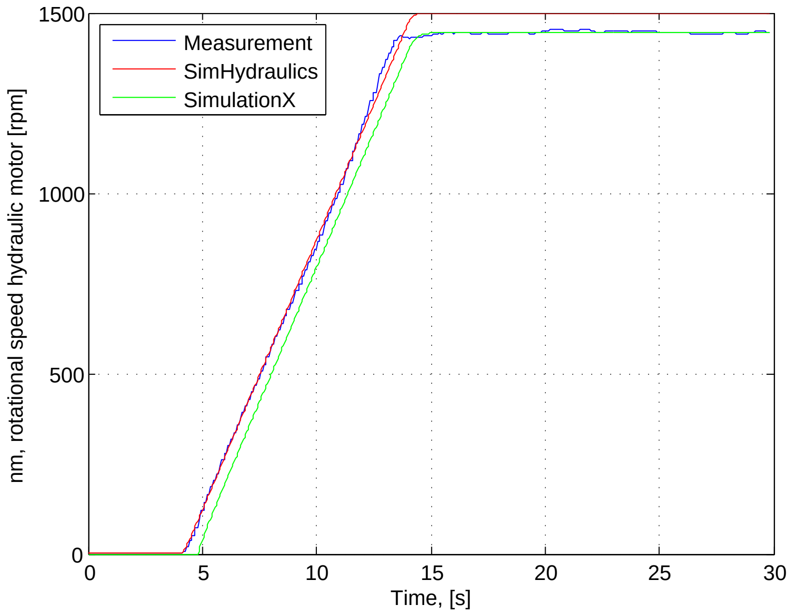


Figure 6.1: Velocity comparison with ramp signal

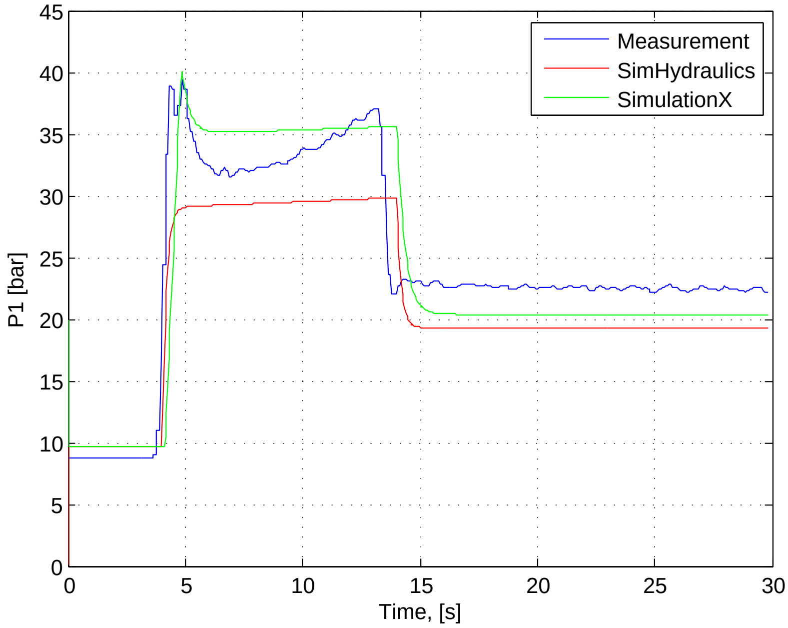


Figure 6.2: P1 comparison with ramp signal

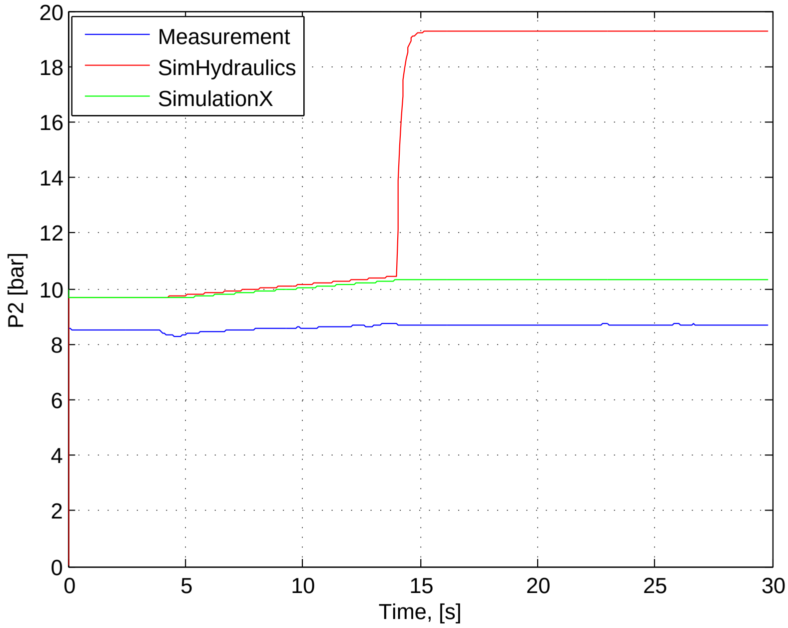


Figure 6.3: P2 comparison with ramp signal

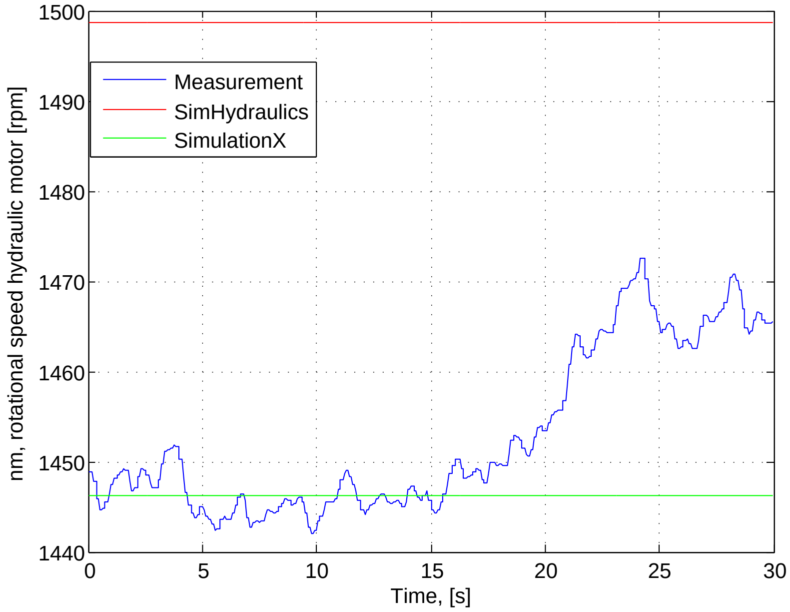


Figure 6.4: Velocity comparison with full speed

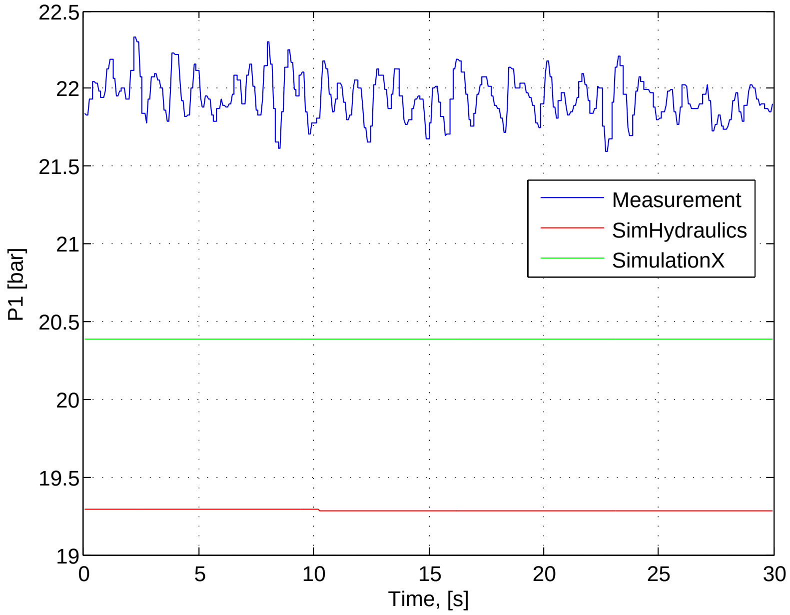


Figure 6.5: P1 comparison with full speed

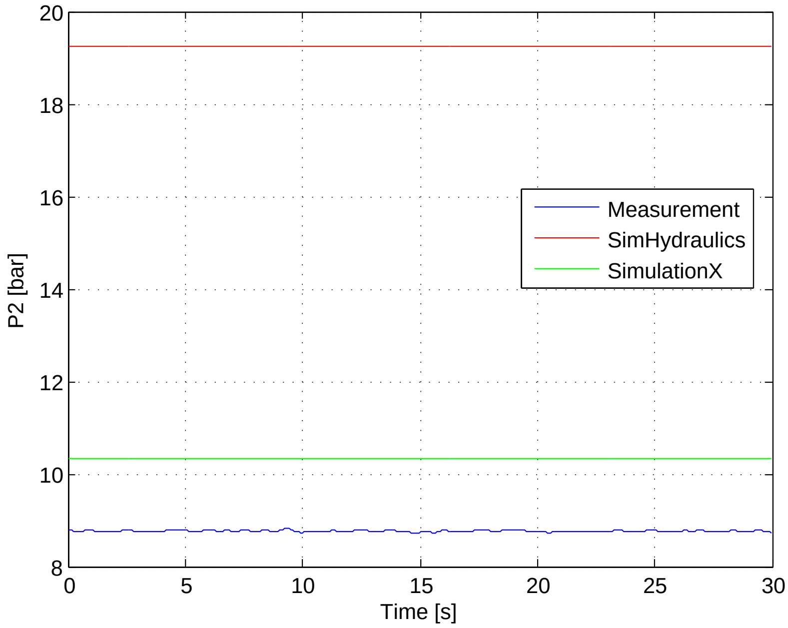


Figure 6.6: P2 comparison with full speed

Chapter 7

Parameter Estimation

The four unknown Friction losses described in section 4.1 should be possible to find by four tests and with the moment equilibrium equation (7.1).

$$
\begin{aligned}
J \cdot \ddot{\theta} = & T_{tM} - K_{mM} \cdot \int (\Delta p) dt - K_{vM} \cdot \int (n \cdot \mu) dt \\
& - K_{hM} \cdot \int (n^2) dt - T_{sP} \cdot \int (sign(n)) dt
\end{aligned} \tag{7.1}
$$

7.1 First Estimation Attempt

7.1.1 Test 1

The first test is to run the transmission with the velocity curve shown in figure 7.1. The important points are that the positive and negative velocity is equal and last for the same amount of time. Because of these two points K_{vM} and T_{sP} becomes zero in eq. (7.2).

$$T_{tM} - K_{mM} \cdot \int (\Delta p_1) dt - K_{hM} \cdot \int (n_1^2) dt = 0 \tag{7.2}$$

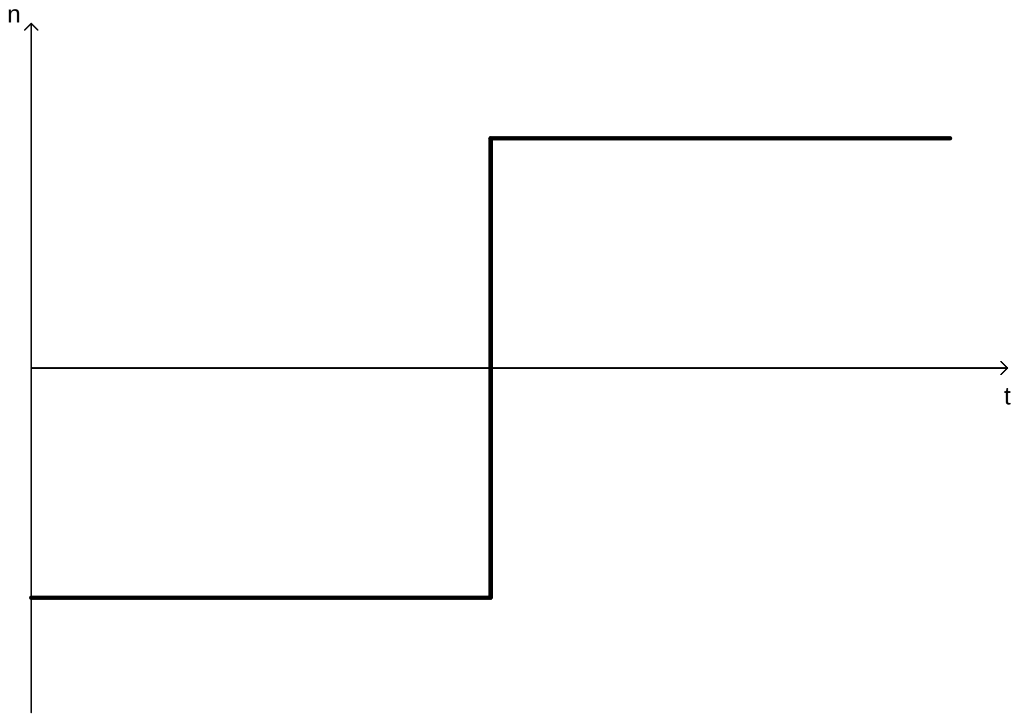


Figure 7.1: The velocity curve for Test 1.

7.1.2 Test 2

The second test is to run the transmission with a sine wave around zero velocity, see figure 7.2. The important point in this test is to start and stop at zero velocity.

$$J \cdot \int (\ddot{\theta}_2 \cdot \dot{\theta}_2)dt = T_{tM} - K_{mM} \cdot \int (\Delta p_2 \cdot \dot{\theta}_2)dt - K_{vM} \cdot \int (n_2 \cdot \mu \cdot \dot{\theta}_2)dt$$
$$- K_{hM} \cdot \int (n_2^2 \cdot \dot{\theta}_2)dt - T_{sP} \cdot \int (sign(n_2) \cdot \dot{\theta}_2)dt \quad (7.3)$$

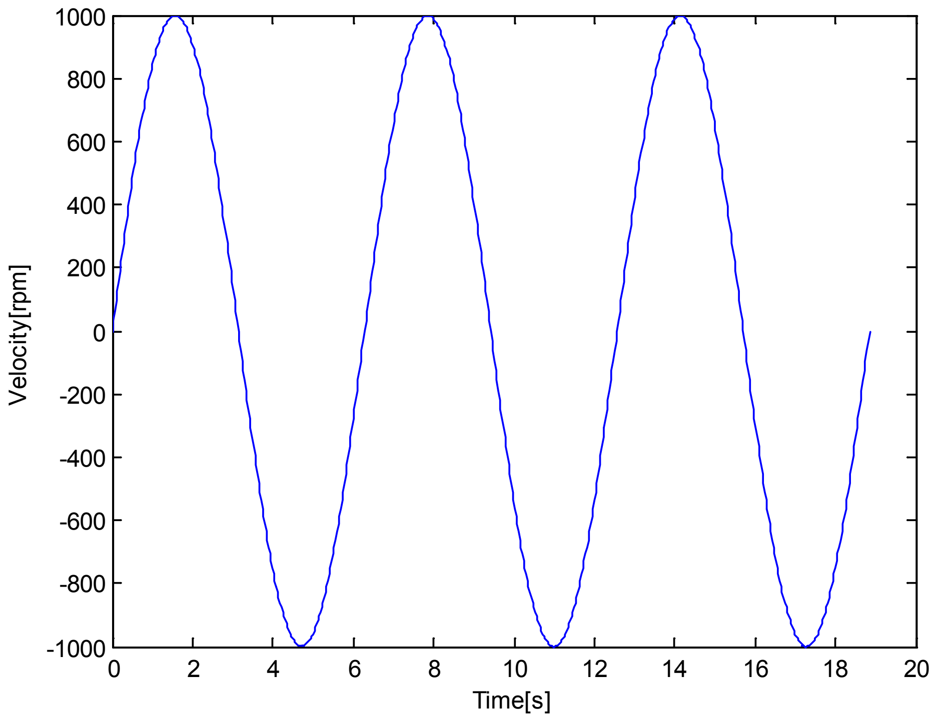


Figure 7.2: The velocity curve for Test 2.

7.1.3 Test 3

This test is to run the rig with a constant positive velocity for a given period of time. See figure 7.3

$$
\begin{aligned}
&T_{tM} - K_{mM} \cdot \int (\Delta p_3)dt - K_{vM} \cdot \int (n_3 \cdot \mu)dt \\
&- K_{hM} \cdot \int (n_3^2)dt - T_{sP} \cdot \int (sign(n_3))dt = 0
\end{aligned} \tag{7.4}
$$

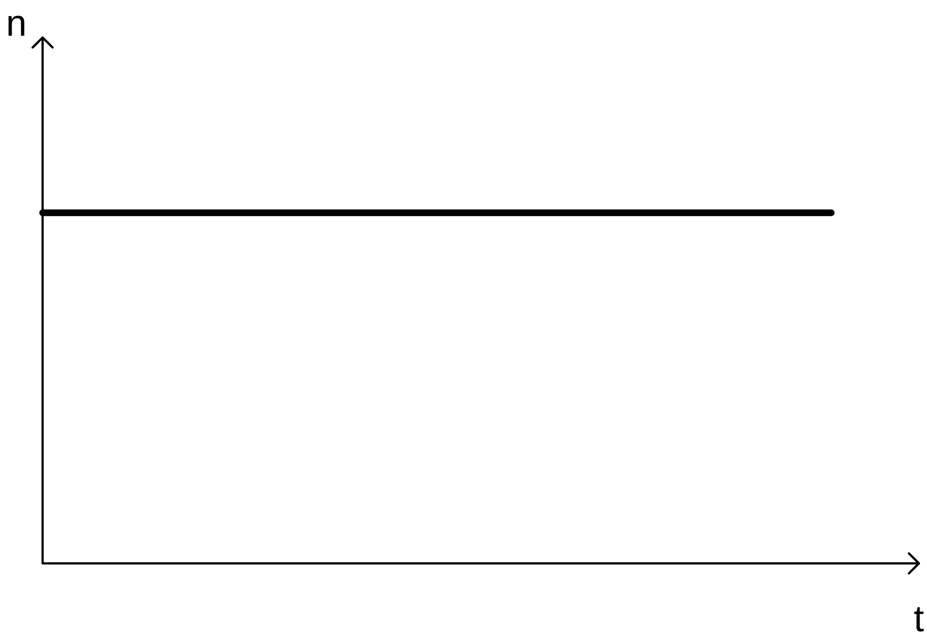


Figure 7.3: The velocity curve for Test 3.

7.1.4 Test 4

In the last test, the velocity should follow the velocity curve shown in figure 7.4. The important point in this test is to not stop the test at zero acceleration.

$$J \cdot \dot{\theta}_4(T) = T_{tM} - K_{mM} \cdot \int (\Delta p_4)dt - K_{vM} \cdot \int (n_4 \cdot \mu)dt - K_{hM} \cdot \int (n_4^2)dt - T_{sP} \cdot \int (sign(n_4))dt \tag{7.5}$$

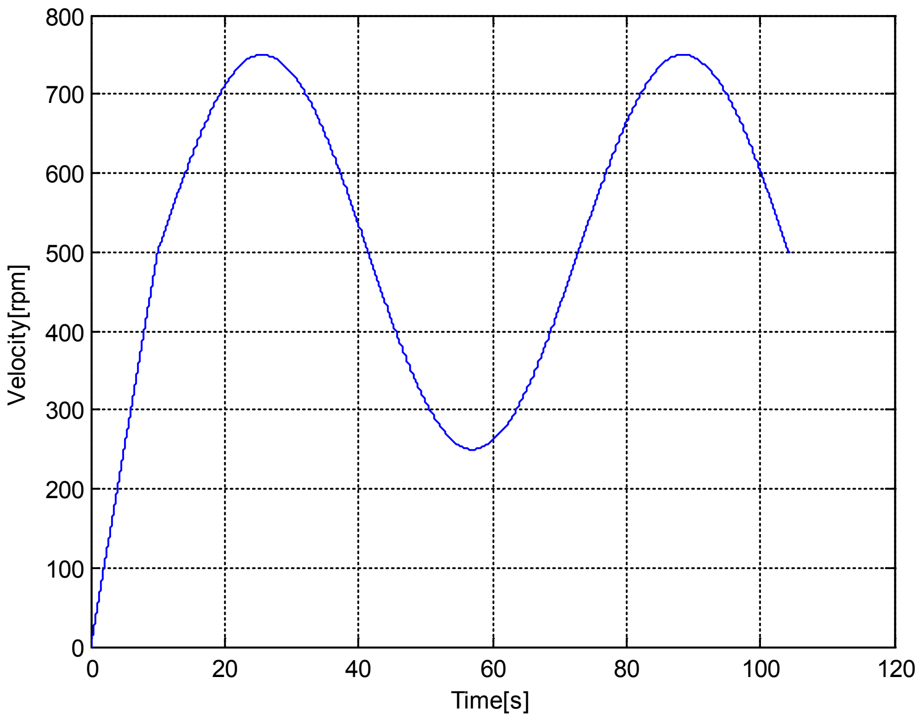


Figure 7.4: The velocity curve for Test 4.

7.1.5 Solution

The equations are then converted into the three matrices A,b and c below.

$$A = \begin{bmatrix} -\int \Delta p_1 dt & 0 & -\int n_1^2 dt & 0 \\ -\int \Delta p_2 \cdot \dot{\theta} dt & -\int (\mu \cdot n_4) \cdot \dot{\theta} dt & -\int n_2^2 \cdot \dot{\theta} dt & -\int sign(n_1) \cdot \dot{\theta} dt \\ -\int \Delta p_3 dt & -\int (\mu \cdot n_3) dt & -\int n_3^2 dt & -\int sign(n_3) dt \\ -\int \Delta p_4 dt & -\int (\mu \cdot n_4) dt & -\int n_4^2 dt & -\int sign(n_4) dt \end{bmatrix} \tag{7.6}$$

$$b = \begin{bmatrix} K_{mM} \\ K_{vM} \\ K_{hM} \\ T_{sP} \end{bmatrix} \tag{7.7}$$

$$c = \begin{bmatrix} -T_{tM} \\ -T_{tM} \cdot \int \dot{\theta} dt \\ -T_{tM} \\ J \cdot \dot{\theta}_4(T) - T_{tM} \end{bmatrix} \tag{7.8}$$

The A-matrix needs to be scaled to get each equation to the same scale. Then the equations are solved in Matlab by calculating $b = A^-1 \cdot c$

Table 7.1: Parameters from the transmission

Parameter	Parameter value
K_{mM}	0,1746
K_{vM}	-3,7036e-7
K_{hM}	-2,7037e4
T_{sP}	0,1933

Since the values of K_{vM} and K_{hM} are negative something is wrong. K_{vM} and K_{hM} can not be negative because friction can not be negative.

The condition number of the A-matrix are checked and it is $3.5731e11$. This number is too large since the condition number should be close to 1. This is because each row of the A-matrix needs to be as independent of each other as possible (the identity matrix has a condition number of one since each row is total independent of each other).

Since the test results clearly are wrong, the test parameters have to be tuned. The basic shape of the velocity curve are kept, but the velocities, and frequency of the oscillations in test 2 and 4, are changed. To make the testing quicker, the tests are performed in SimulationX, because the model are the most like the physical transmission.

After testing with many different combinations of velocities and frequencies, the condition number could not be reduced below $5,5329e9$

Table 7.2: Parameters from the SimulationX model

Parameter	Parameter value
K_{mM}	-1,7845e-4
K_{vM}	-5,9751
K_{hM}	3,0265e-10
T_{sP}	6,6666e-5

This set of tests were discarded because the condition number only were reduced by about 2 orders of magnitude from the test on the physical transmission and some of the k's are negative.

7.2 Second Estimation Attempt

In this estimation each parameter should be found separately if possible.

The tests are first run in SimulationX to find good velocity curves for the tests. This is accomplished by setting the k-parameters to random values and recover them by running the four test presented below. The recovered parameters and comparison with the actual parameters are presented in table 7.3 See appendix K for the code.

7.2.1 Test 1

The first test is to run the transmission with the velocity curve shown in figure 7.5. The important point is to have the same steady state speed in both rotational directions and that the positive and negative velocity last for exact the same amount of time. This test is to find the parameter K_{hM}. Since the curve is symmetrical, it is assumed that the K_{vM} part of the equation is zero since $\int (n_1)dt = 0$. And the T_{sP} part of the equation is assumed zero because $\int sign(n_1)dt = 0$. K_{mM} can not be assumed zero because $\int (\Delta p)dt \neq 0$, so test 1 has to be combined with test 2.

$$J \cdot \int (\ddot{\theta}_1)dt = T_{tM} - K_{mM} \cdot \int (\Delta p_1)dt - K_{hM} \cdot \int (n_1^2)dt \tag{7.9}$$

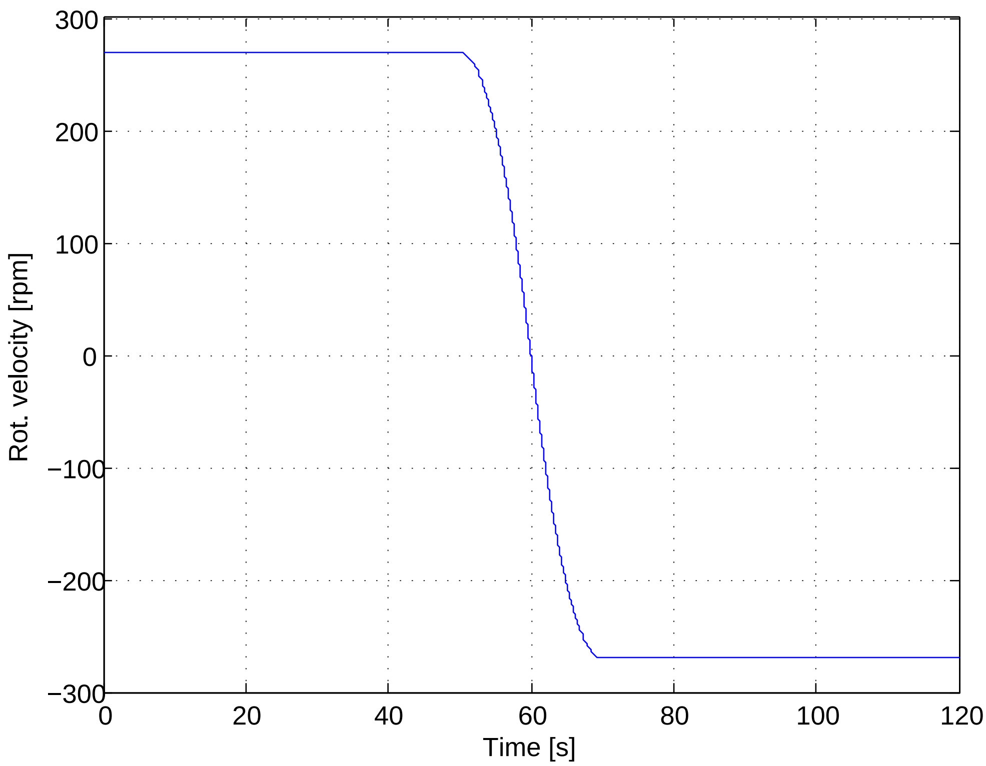


Figure 7.5: The velocity curve for Test 1.

7.2.2 Test 2

No test were found to meet both the valid test criterion(stated in subsection 7.2.5 and low condition number criterion in combination with test 1. Both test 1 and test 2 were variated to try to meet the criterion.

7.2.3 Test 3

The third test is to run the transmission with first a low positive velocity for a chosen time and then a larger negative velocity for the same amount of time. See figure 7.6. In this test T_{tM} and K_{mM} is assumed to be zero because $\int(\Delta p_2)dt = 0$. And the T_{sP} part of the equation is assumed zero because $\int sign(n_2)dt = 0$. The $J \cdot \int \ddot{\theta}_2$ part is assumed zero because the velocity is constant and therefor $\ddot{\theta}_2 = 0$. This test provides the K_{vM} parameter.

$$K_{vM} = \frac{-K_{hM} \cdot \int (n_3^2)dt}{\mu \cdot \int (n_3)dt} \tag{7.10}$$

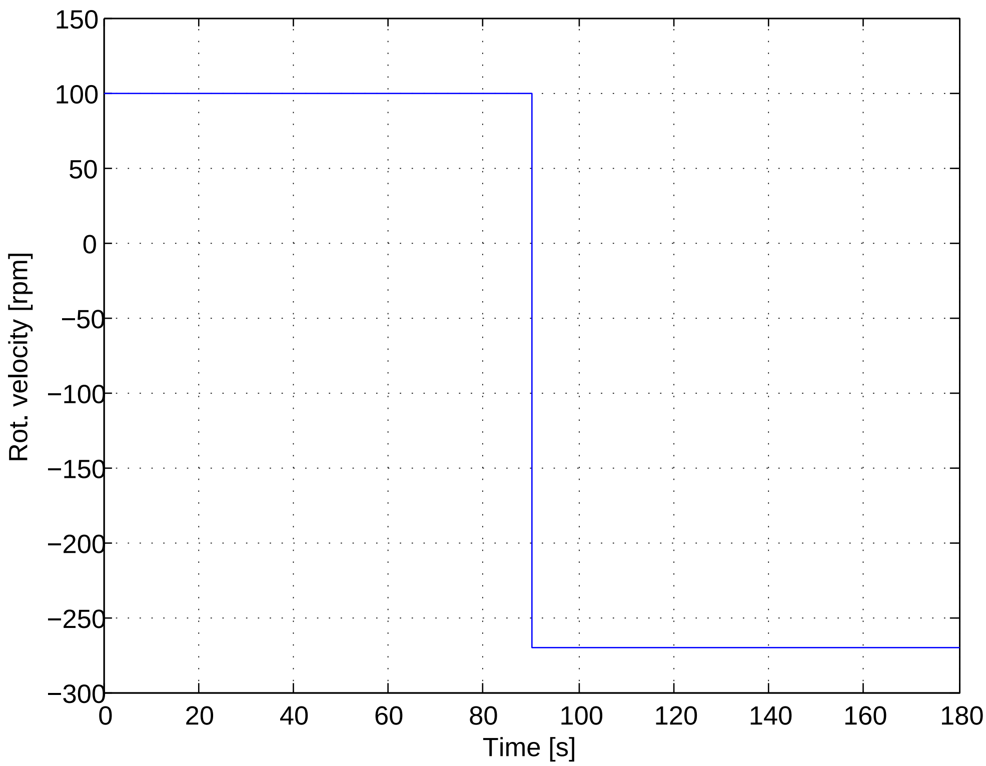


Figure 7.6: The velocity curve for Test 3.

7.2.4 Test 4

In the last test, the velocity should be a high positive for a short time and then low negative for a long time. See figure 7.7. The area under the positive part of the curve must be equal to the area of the negative part, so that the K_{vM} part becomes zero. The shape of the curve prevents the T_{sP} parameter being zero, since this test provides this parameter.

$$T_{sP} = \frac{T_{tM} - K_{mM} \cdot \int(\Delta p_4)dt - K_{hM} \cdot \int(n_4^2)dt}{\int sign(n_4)dt} \tag{7.11}$$

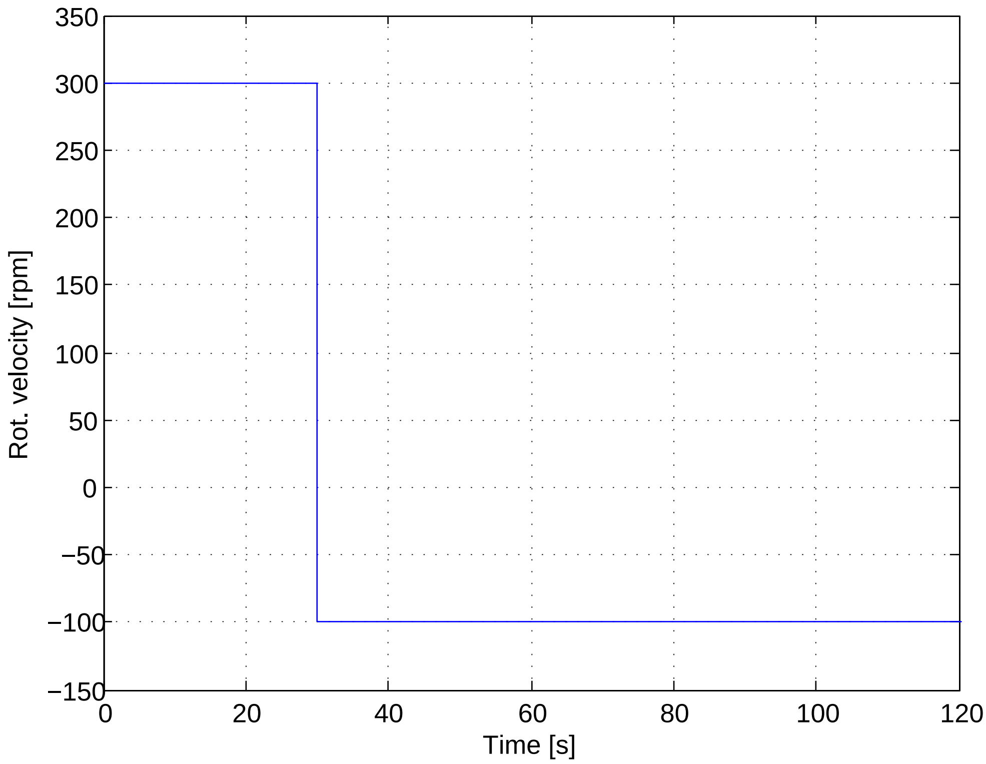


Figure 7.7: The velocity curve for Test 4.

7.2.5 Solution

Test 1 and 2 has to be combined in a matrix equation to find K_{mM} and K_{hM}. This is because it is impossible to make a test where Δp is zero and $\ddot{\theta}$ is different from zero. The tests must be checked to see if it is a valid test. This is done by using the true k values in the equation for the test and check if both right and left side is equal to each other.

Test 3 and 4 gives the correct value for K_{vM} and T_{sP} (see table 7.3) if the true value for K_{mM} and K_{vM} are used.

Table 7.3: SimulationX test results

Parameter	Real parameter value	Calculated parameter value	Ratio
K_{mM}	0,013		
K_{vM}	2345	2346,5	1,00
K_{hM}	5,689e-8		
T_{sP}	0,0233	0,0250	1,07

Chapter 8

Conclusion and further work

The greatest challenges of this thesis have been the Simulink model and the parameter estimation of the frictions losses in the hydraulic motor. The solver in Matlab got problems when to solve the algebraic equations for the Simulink model. Several methods to model the transmission in Simulink/Matlab were tried, but the solver could not deal with the algebraic loops. Because of the lack of time this problem could not be solved. A possible solution to this problem is to simplify the transmission model as much as possible and compare the results from each block with the equivalent blocks in SimHydraulics to see if the results will be the same. Several tests were tried out in order to find the different frictions constants for the hydraulic motor. The two tests for finding the static and viscous friction constants were found, but they can only be found after first finding the mechanical and the turbulent friction constants. No good tests for these constants could be established since the tests did not fulfill the requirements of being valid tests for the constants and being independent enough of each other. A possible solution to this problem is to add a dynamic load to be able to control the pressure drop and the speed of the motor. Good simulation results from the SimulationX have been obtained. The model in SimulationX was the easiest to create, and gave the best results compared with the measurements.

Bibliography

[1] M. Hansen and T. Andersen, “Hydraulic components and systems,” Lecture notes in MAS402.

Appendices

Appendix A

Data Sheet Main pump

Operating Data

Pressure and Speed Limits

Basic model designation	Geometric dispalcement, cm^3/r (in^3/r)	Maximum shaft speed (r/min)			Maximum outlet pressure, bar (psi)		
		Anti-wear hydraulic oil	Water-in–oil emulsion (40%/60%)	Water-glycol	Anti–wear hydraulic oil	Water glycol	Water-in–oil emulsion (40%/60%)
PFB5 PFB10 PFB20	10,55 (0.64) 21,10 (1.29) 42,80 (2.61)	3600 3200 2400	1800	1800	210 (3000) 210 (3000) 175(2500)	175 (2500)	175 (2500)
PVB5 PVB6 PVB10 PVB15 PVB20 PVB29 PVB45	10,55 (0.64) 13,81 (0.84) 21,10 (1.29) 33,00 (2.01) 42,80 (2.61) 61,60 (3.76) 94,50 (5.76)	1800	1800	1800	210 (3000) 140 (2000) 210 (3000) 140 (2000) 210 (3000) 140 (2000) 210 (3000)	140 (2000) 100 (1500) 140 (2000) 100 (1500) 140 (2000) 100 (1500) 140 (2000)	140 (2000) 100 (1500) 140 (2000) 100 (1500) 140 (2000) 100 (1500) 140 (2000)
PVB90	197,50 (12.0)	1800	1200	1200	210 (3000)	140 (2000)	140 (2000)

Maximum Inlet Pressure

All pumps except PVB5/6/10/15 with H, M or V controls . . . . 1,0 bar (15 psi)
PVB5/6/10/15 with H, M or V controls . . . . As "Max. outlet pressure" above for appropriate size.

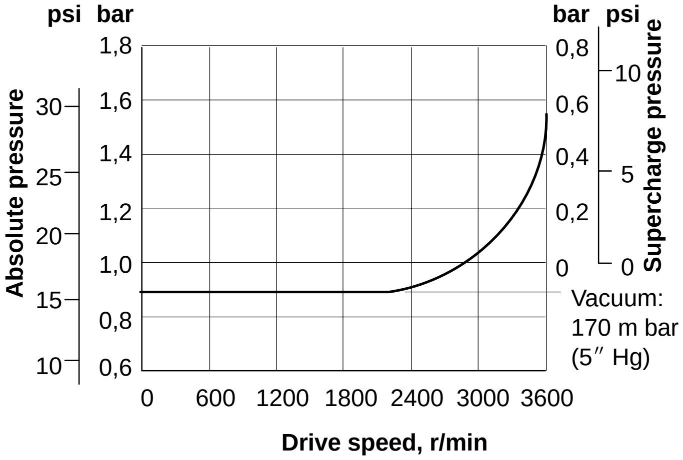
Case Drain Pressure

See "Installation data" section, on page A.33.

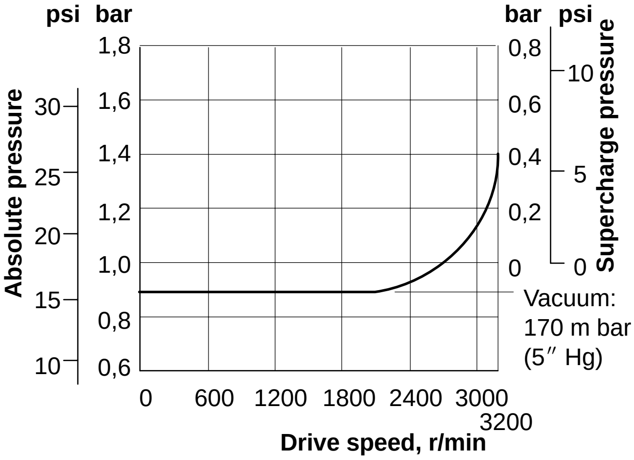
Minimum Inlet Pressure

See following graphs.
Based on oil viscosity of 21 cSt (102 SUS) and at 50°C (120°F).

PFB5 and PVB5



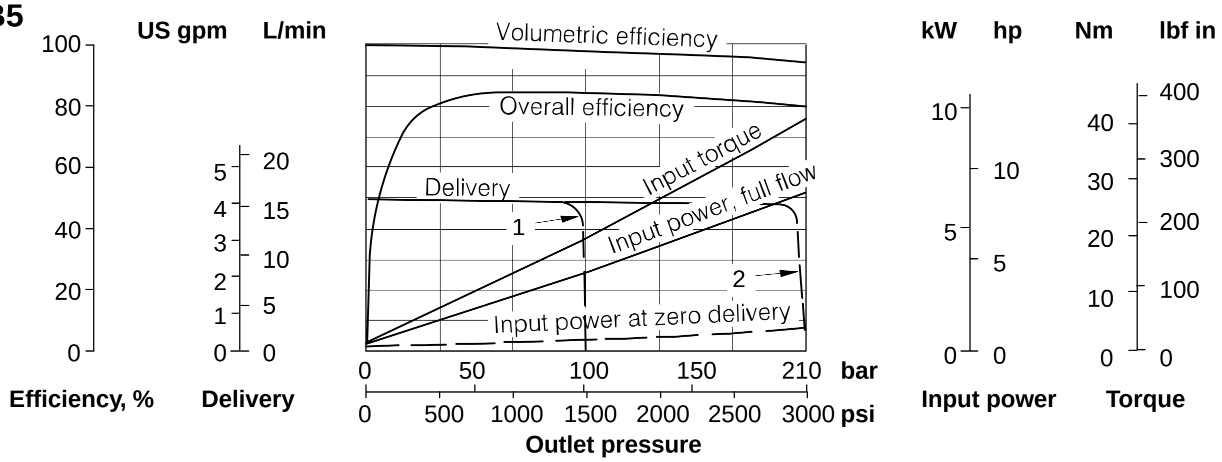
PVB6



Performance Data at 1500 r/min Drive Speed (cont'd)

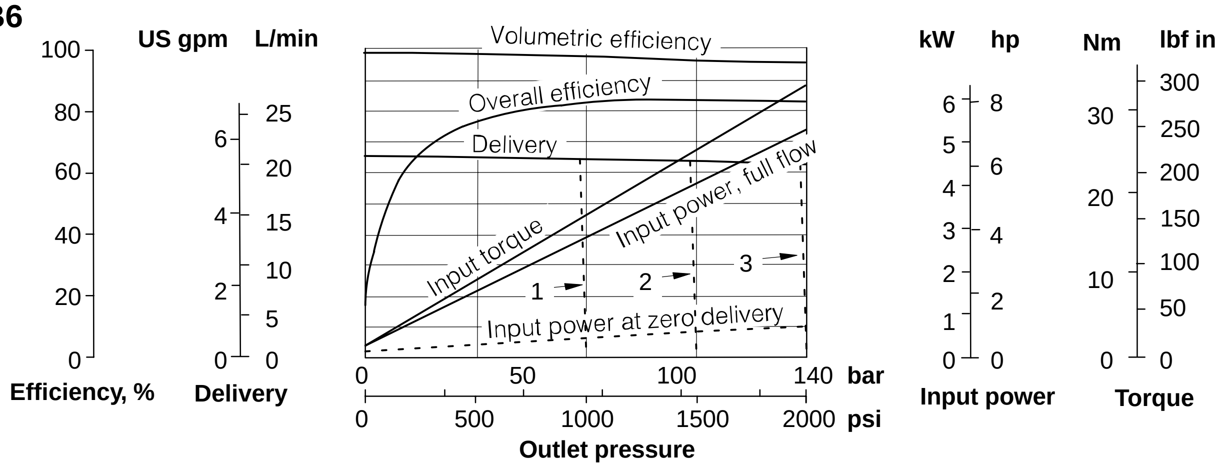
With oil at 21cSt (102 SUS) and at 49°C (120°F): Atmospheric inlet
For data at drive speed of 1800 r/min, see pages A.11 to A.14

PVB5



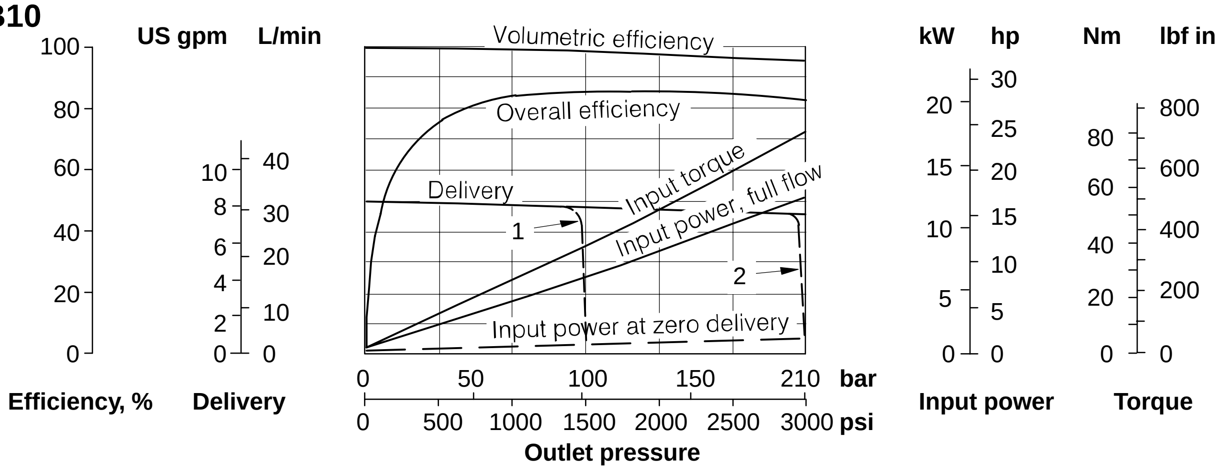
1 = Delivery with compensator setting of 100 bar (1500 psi)
2 = Delivery with compensator setting of 200 bar (3000 psi)

PVB6



1 = Delivery with compensator setting of 70 bar (1000 psi)
2 = Delivery with compensator setting of 100 bar (1500 psi)
3 = Delivery with compensator setting of 140 bar (2000 psi)

PVB10



1 = Delivery with compensator setting of 100 bar (1500 psi)
2 = Delivery with compensator setting of 200 bar (3000 psi)

Appendix B

Data Sheet Hydraulic Motor

MFB Model Series

Model Code

F3 – MFB ** –(F) U(*)– **

1 2 3 4 5 6 7

1 **Special Seals**

F3 – Seals for use with mineral oil or fire resistant fluids.
Blank – Omit if not required

2 **Model Series**

M – Motor
F – Fixed displacement
B – Inline type

3 **GPM Rating @ 1800 rpm**

5 – 19 L/min (5 USgpm)
10 – 37,9 L/min (10 USgpm)
20 – 75,7 L/min (20 USgpm)
29 – 109,8 L/min (29 USgpm)
45 – 170,3 L/min (45 USgpm)

4 **Mounting Type**

F – Foot bracket
(For separate foot bracket kit, order model model FB–A–10)
Blank – Omit for flange mounting

5 **Rotation**

U – Either direction

6 **Shaft End (MFB5/10 only)**

Y – Standard Shaft
Blank – Optional shaft

* Optional shaft is available only to provide interchangeability with earlier (–10 design) units. (Not recommended for operation above 1800 r/min and 100 bar (1500 psi).

6 **Port Connections (MFB 45)**

F – SAE 4-bolt Flanged Ports

7 **Design Number**

Subject to change

21 – 21 Design (MFB5)
31 – 31 Design (MFB10)
10 – 10 Design (MFB 20, 29, 45)

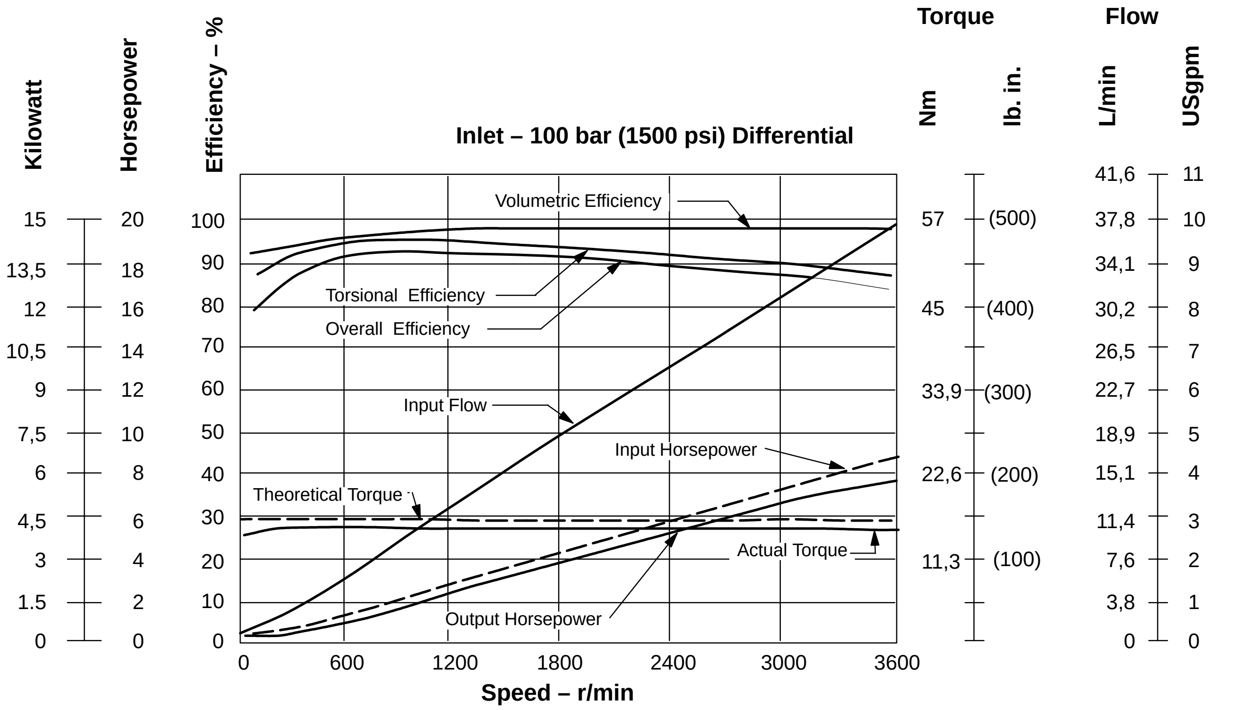
Specifications

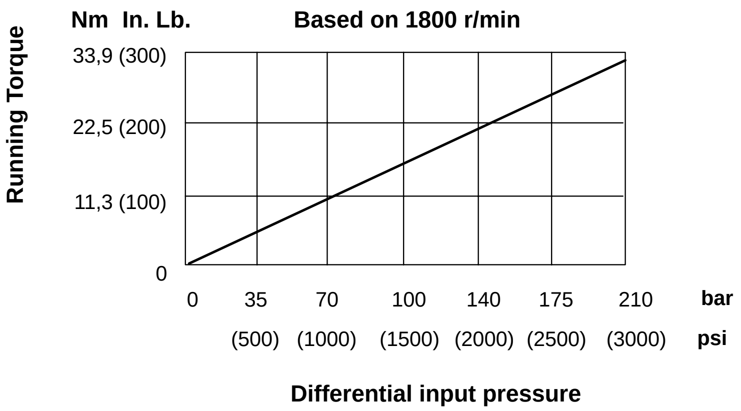
Model	Theoretical Displ. cm^3/rev (in^3/rev)	Flow L/min (USgpm) @ Rated r/min	Operating Speed r/min		Pressure bar (psi)		Output Torque Nm (lb in)		Dry Weight kg (lb)
			Rated	Max	Rated	Max	Rated	Max	
MFB5	10,5 (0.643)	19.0 (5.0)	1800	3600	100 (1500)	210 (3000)	15,25 (135)	30,5 (270)	5,0 (11)
MFB10	21,12 (1.29)	37,9 (10.0)	1800	3200	100 (1500)	210 (3000)	32,1 (284)	64,2 (568)	9,5 (21)
MFB20	42,8 (2.61)	75,7 (20)	1800	2400	100 (1500)	175 (2500)	50,85 (450)	101,7 (900)	18,5 (49)
MFB29	61,6 (3.76)	109,8 (29)	1800	2400	70 (1000)	140 (2000)	58,75 (520)	117,5 (1040)	18,5 (49)
MFB45	94,4 (5.76)	170,3 (45)	1800	2200	100 (1500)	210 (3000)	135,6 (1200)	271,2 (2400)	33 (73)

MFB5 Model Series

Performance Characteristics

Based on oil temperature of 49°C (120°F) – Atmospheric Outlet





Appendix C

Data Sheet Boost Pump

Data sheet GPA-H-E-02/96

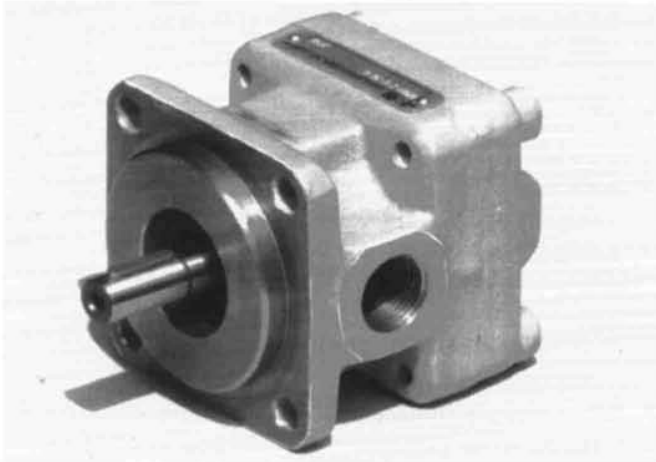
JSB Hesselman
GARPHYTTAN GROUP

Quiet Internal Gear Pump

GA and GPA Series 30

General description

Fixed displacement internal gear pumps that can be driven by fixed or variable speed prime movers. Available in single, double, triple and quadruple configuration to suit a wide variety of applications.



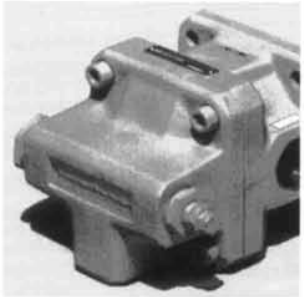
Basic characteristics

Displacements	1.7 to 63 cm^3 per single pump or section
Max. pressure	100 bar
Max. speed	up to 4000 rev/min
Types	single and multiple models

Features

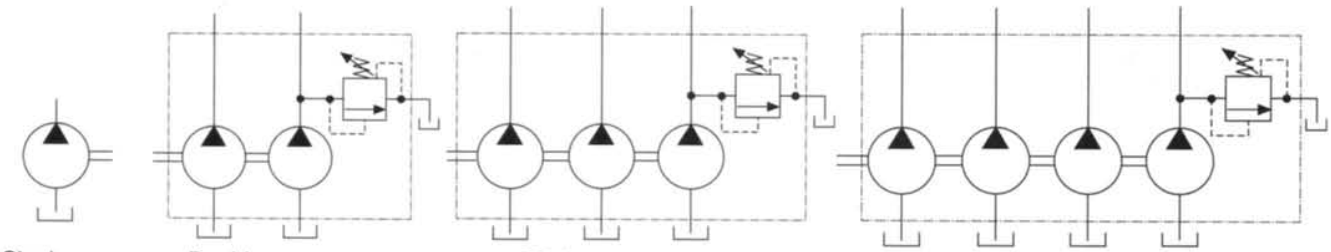
- Quietness
- Internal pressure relief valve option
- Heavy duty bearing option for indirect drives
- Choice of ISO metric or SAE mounting
- Choice of clockwise or counter-clockwise rotation
- G(BSPF) threaded ports

with pressure relief valve



without pressure relief valve

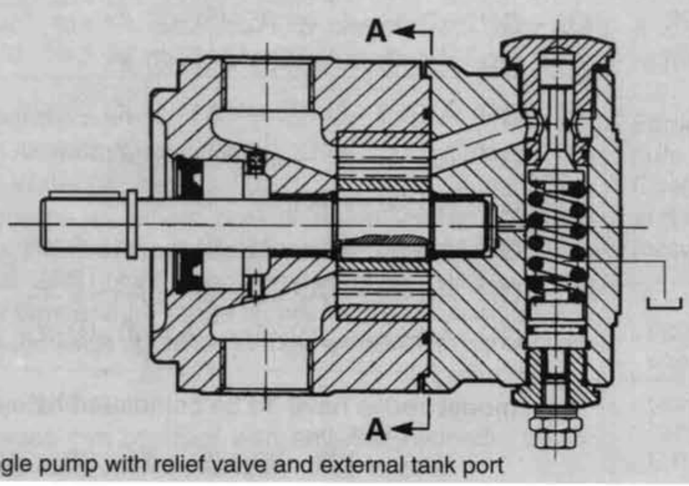
Functional symbols



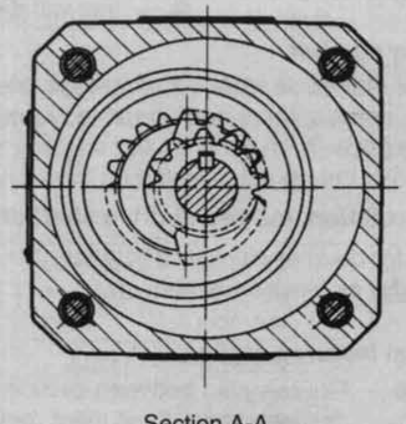
Single pump Double pump ... Triple pump ... Quadruple pump ..

... equipped with pressure relief valve (optional) for cover end pump section and external tank port

Cross section



Single pump with relief valve and external tank port



Section A-A

Model code

① **Seals for phosphate ester fluids** = Viton
Omit if not required.

② **Monting flange**
P = 4-hole square mounting flange to ISO 3019/2, standard for frame size **A3,** so always fill in for **A3**
w/o = SAE 2-hole oval mounting flange (ISO 3019/1) Option for frame sizes by code **A1** and **A2** only.

③ **Frame size**

A1	=	Displacement	1.76 cm^3 ... 4.40 cm^3
A2	=	Displacement	6.90 cm^3 ... 17.30 cm^3
A3	=	Displacement	25.50 cm^3 ... 63.60 cm^3

④,⑤,⑥ **Displacement of single pump or pump section**
For multiple pumps of only one frame size the displacements increase from shaft end to cover end

④	⑤	⑥
1 = 1.76 cm^3	**6** = 6.90 cm^3	**25** = 25.50 cm^3
2 = 2.75 cm^3	**10** = 11.00 cm^3	**40** = 40.80 cm^3
4 = 4.40 cm^3	**16** = 17.30 cm^3	**63** = 63.60 cm^3

⑦ **Mounting flange**
A = SAE 'A' size 2-hole oval mounting flange (ISO 3019/1). For frame sizes **A1** and **A2** only. Omit for 4- hole metric square flanges to ISO 3019/2, for all GPA models.

⑧ **Front bearing arrangement**
E = for plain bearing for direct drives
F = roller and plain bearings, for indirect drives. (always fill in for frame size **A1** SAE "A" SAE 2-hole oval mounting flange (ISO 3019/1)

⑨ **Integral adjustable relief valve**
(Omit if not required).

	H	=	5 ... 25 bar
Adjustment range	**K**	=	5 ... 60 bar
	M	=	5 ... 100 bar

⑩ **Relief valve discharge**
(Omit if no relief valve is specified).
1 = External discharge
2 = Internal discharge

⑪ **Design number**
30 = Series 30. Subject to change. Installation dimensions unchanged for design numbers 30 to 39.

⑫ **Shaft rotation, viewed at drive shaft end**
L = for counter-clockwise rotation
R = for clockwise rotation

⑬ **Special features suffix**
GE330 = Fluid sealing between sections of a multiple pump. Omit if not required.

Model keys for pump variations by both number and frame sizes of pump sections. The keys have to be completed by displacement code(s) and from key ⑦ on according to requirements.

Single pumps

① - **G**② A1-④ ✍
① - **G**② A2-⑤ ✍
① - **GP** A3-⑥ ✍

Double pumps

① - **G**② A1-④ -④ ✍
① - **G**② A2-⑤ -⑤ ✍
① - **G**② A2-⑤ — A1-④ ✍
① - **GP** A3-⑥ -⑥ ✍
① - **GP** A3-⑥ — A2-⑤ ✍
① - **GP** A3-⑥ — A1-④ ✍

Triple pumps

① - **G**② A1-④ -④ -④ ✍
① - **G**② A2-⑤ -⑤ -⑤ ✍
① - **G**② A2-⑤ -⑤ — A1-④ ✍
① - **G**② A2-⑤ — A1-④ -④ ✍
① - **GP** A3-⑥ -⑥ -⑥ ✍
① - **GP** A3-⑥ -⑥ — A2-⑤ ✍
① - **GP** A3-⑥ — A2-⑤ -⑤ ✍
① - **GP** A3-⑥ -⑥ — A1-④ ✍
① - **GP** A3-⑥ — A1-④ -④ ✍
① - **GP** A3-⑥ — A2-⑤ — A1-④ ✍

Quadruple pumps

① - **G**② A1-④ -④ -④ -④ ✍
① - **G**② A2-⑤ -⑤ -⑤ -⑤ ✍
① - **G**② A2-⑤ -⑤ -⑤ — A1-④ ✍
① - **G**② A2-⑤ -⑤ — A1-④ -④ ✍
① - **G**② A2-⑤ — A1-④ -④ -④ ✍
① - **GP** A3-⑥ -⑥ -⑥ -⑥ ✍
① - **GP** A3-⑥ -⑥ -⑥ — A2-⑤ ✍
① - **GP** A3-⑥ -⑥ — A2-⑤ -⑤ ✍
① - **GP** A3-⑥ — A2-⑤ -⑤ -⑤ ✍
① - **GP** A3-⑥ -⑥ -⑥ — A1-④ ✍
① - **GP** A3-⑥ -⑥ — A1-④ -④ ✍
① - **GP** A3-⑥ — A1-④ -④ -④ ✍
① - **GP** A3-⑥ -⑥ — A2-⑤ — A1-④ ✍
① - **GP** A3-⑥ — A2-⑤ -⑤ — A1-④ ✍
① - **GP** A3-⑥ — A2-⑤ — A1-④ -④ ✍

✍ **model codes have to be completed by**

— ⑦ - ⑧ - ⑨ - ⑩ - **30** - ⑫ - ⑬

Operating data

Pressure limits

Inlet ports	min. continuous	– 0.25 bar
	min. intermittent	– 0.40 bar
	maximum	+ 2.00 bar
Outlet ports, maxi.:	with antiwear hydraulic oils	100 bar
	with burner fuel oils	50 bar
	with other fluids	consult ***JSB***

Shaft speed limits, max. speed (rev/min) ▲

Frame size	Operating pressure 20 bar	100 bar
G(P)A 1	4000	3000
G(P)A 2	3500	3000
G P A 3	2300	2000

▲ For burner fuel oils max. speed for all sizes n = 1800 rev/min

Shaft speed limits, min. speed (rev/min)

Frame size	Oil viscosity [cSt]	Operating pressure 60 bar	80 bar	10 bar
G(P)A 1	14.5	500	600	800
	9.0	600	800	
	7.5	800		
G(P)A 2	14.5	< 500 *	500	600
	9.0	500	600	
	7.5	600		
G P A 3	14.5	< 500 *	< 500 *	500
	9.0	< 500 *	500	
	7.5			

* For specific applications, consult ***JSB*** representative

Performance data

Typical at 1500 rev/min with oil at 40 cSt and at 38°C

Pump size	7 bar L/min	kW	25 bar L/min	kW	50 bar L/min	kW	70 bar L/min	kW	100 bar L/min	kW
G(*)A1-1	2,6	0,1	2,5	0,15	2,4	0,3	2,3	0,4	2,1	0,6
G(P)A1-2	4,1	0,1	4,0	0,25	3,9	0,5	3,8	0,7	3,6	0,9
G(P)A1-4	6,6	0,15	6,4	0,40	6,2	0,7	6,0	0,9	5,7	1,3
G(P)A2-6	10,3	0,25	10,0	0,6	9,7	1,1	9,3	1,4	8,9	2,0
G(P)A2-10	16,5	0,4	16,1	0,9	15,7	1,6	15,3	2,3	14,8	3,2
G(P)A2-16	25,9	0,6	25,5	1,5	25,0	2,6	24,5	3,7	23,8	5,1
GPA3-25	36,4	0,85	35,2	2,2	34,0	3,8	32,7	5,1	31,3	7,1
GPA3-40	60,4	1,4	58,7	3,0	57,3	5,5	55,9	7,5	54,3	10,8
GPA3-63	94,3	2,2	92,8	4,4	91,2	8,8	89,5	12,0	87,7	16,8

Temperature limits

Antiwear hydraulic oil .... 0 to 68 °C
Water glycol .... 0 to 60 °C

For phosphate esters consult manufacturer and ***JSB*** where limits are outside those for hydraulic oil. Also consult ***JSB*** before using burner fuel oils. Whatever the actual temperature range is, ensure that viscosities stay within the limits specified in the 'Hydraulic fluids' section.

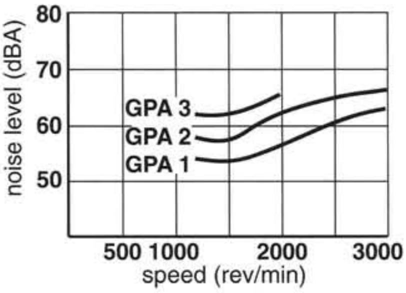
Hydraulic fluids.

All pumps can be used with antiwear hydraulic oils or water glycols. (continued right colum above).

Burner fuels to BS.2869 Class D or equivalent can be pumped but will necessitate lower max pressures and speeds, consult ***JSB.***
Add prefix 'F3' to model designation when phosphate ester (alkyl based types not permitted) or chlorinated hydrocarbons are to be used.
The extreme operating viscosity is from 1000 to 16 cSt but the recommended running range for hydraulic fluids is from 45 to 30 cSt.
However, for burner fuels the range is 5.5 to 1.5 cSt.

Noise levels

Typical levels when operating at 100 bar with oil at 28 cSt and 65°C. Inlet pressure minus 0.16 bar. Measured in accordance with ISO 4412



Filtration requirements: 25 µm absolute, or finer.

Drive recommendations

Direction of rotation

Clockwise or counter-clockwise (viewed at shaft end). To order see also 'Model code' and 'Installation dimensions' sections.

Load and torque limits

For direct drives, shafts for all single pumps and common frame size double pumps are designed to operate at rated pressure.
Double and triple pumps must be used within the following limits where:

p_1, p_2, p_3 & p_4 = Max. pressures [bar] of individual sections (referenced from the shaft end) for the application.
V_1, V_2, V_3 & V_4 = Displacements (cm³) of the same sections.

Shaft load* = $(p_1 V_1)+(p_2 V_2)+(p_3 V_3)+(p_4 V_4)$

Internal coupling load* = $(p_2 V_2)+(p_3 V_3)+(p_4 V_4)$

* according to number of pump sections and max. hydr. load

Frame size	Check Shaft load	Internal coupling load
G(P)A 1	≤ 1,32 x 10³	≤ 0,88 x 10³
G(P)A 2	≤ 5,2 x 10³	≤ 3,5 x 10³
G P A 3	≤ 19,1 x 10³	≤ 13,0 x 10³

If axial loading is envisaged or a quadruple pump is required, consult your ***JSB*** representative.

For indirect drives use only pumps fitted with the heavy duty bearing option; see 'Model code' section.
Check drive shaft torque limits as for direct drives above and that transverse and axial loads do not exceed the following limits:

Frame size	Maximum transverse force	Maximum transverse moment	Maximum axial force
G(P)A 1	440 N	15.8 Nm	300 N
G(P)A 2	820 N	41.0 Nm	600 N
G P A 3	1600 N	102.4 Nm	1000 N

Appendix D

Data Sheet Check Valve

Operating Data

Maximum Operating Pressure, bar (psi)

C5G-805 valves	250 (3600)
C5GV-815/825 valves	350 (5000)
All other valves	210 (3000)
E-C4GM-815 subplates	210 (3000)
E-C5GM-825 subplates	210 (3000)

Cracking Pressure, bar (psi)

C2-8**	0,35 (5)
C2-8**-S3	3,5 (50)
C2-8**-S8	5,2 (75)
C2-8**-S12	0,35 (5)
C5G(V)-8**(-S12)	0,35 (5)
C5G(V)-8**-S3	3,5 (50)
C5G(V)-8**-S8	5,2 (75)
DT8P1-**-05	0,35 (5)
DT8P1-**-30	2,1 (30)
DT8P1-**-65	4,5 (65)

Nominal Flow, L/min (USgpm)

C2-800-(S**)	12 (3)
C2-805-(S**)	23 (6)
C2-815-(S**)	60 (16)
C2-820-(S**)	105 (28)
C2-825-(S**)	170 (45)
C2-830-(S**)	245 (65)
C5G-805-(S**)	38 (10)
C5GV-815-(S**)	76 (20)
C5GV-825-(S**)	380 (100)
DT8P1-02-**	12 (3)
DT8P1-03-**	30 (8)
DT8P1-06-**	76 (20)
DT8P1-10-**	190 (50)

Hydraulic Fluids

The C2 and C5G(V) models can be used with hydraulic oils, water-in-oil emulsions and water glycols. Add prefix “F3” to model designation when phosphate ester (not alkyl-based) are to be used.

DT8P1 models can be used with all of the above mentioned fluids.

The extreme operating viscosity range is from 13 to 860 cSt (70 to 4000 SUS) but the recommended range is 13 to 54 cSt (70 to 245 SUS).

For further information about fluids see leaflet 920.

Contamination Control Requirements

Recommendations on contamination control methods and the selection of products to control fluid condition are included in Vickers publication 9132 or 561, “Vickers Guide to Systemic Contamination Control”. The book also includes information on the Vickers concept of “ProActive Maintenance”. The following recommendations are based on ISO cleanliness levels at 2 µm, 5 µm and 15 µm. For products in this catalog the recommended levels are:

Up to 210 bar (3000 psi)	20/**18**/**15**
Above 210 bar (3000 psi)	20/**18**/**15**

Temperature Limits

Ambient

Min.	–20°C (–4°F)
Max.	+70°C (+158°F)

Fluid Temperature

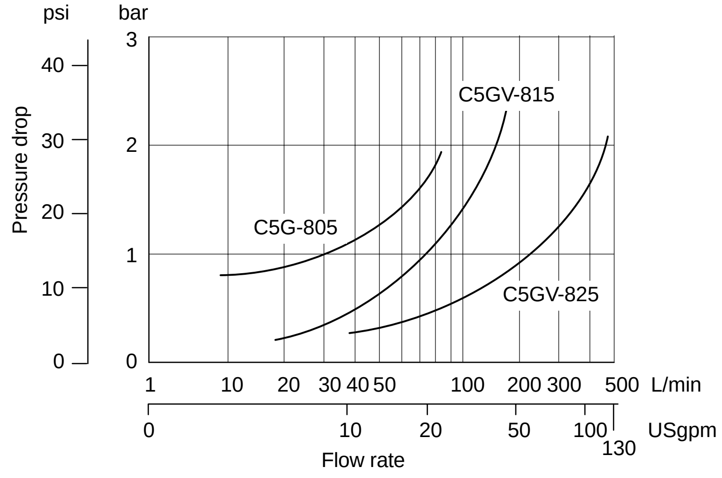
	Petroleum oil	Water-containing
Min.	–20°C (–4°F)	+10°C (50°F)
Max.*	+80°C (+176°F)	+54°C (130°F)

* *To obtain optimum service life from both fluid and hydraulic system 65°C (150°F) normally is the maximum temperature except for water-containing fluids.*

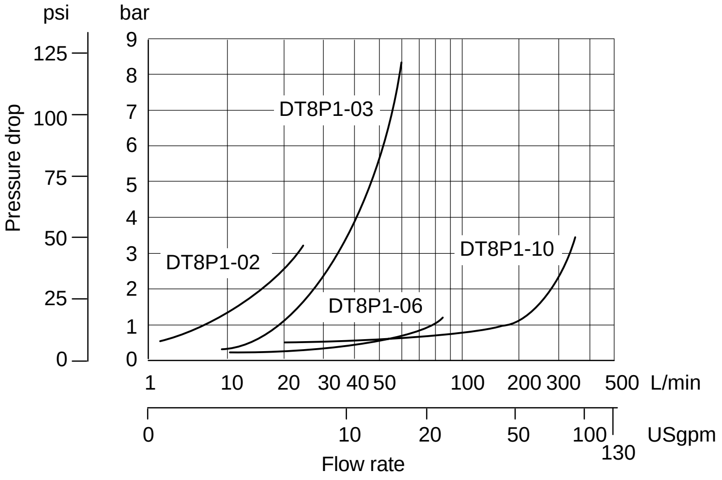
For synthetic fluids consult manufacturer or Vickers representative where limits are outside those for petroleum oil. Whatever the actual temperature range, ensure that viscosities stay within the limits specified in the “Hydraulic Fluids” section.

Pressure Drop

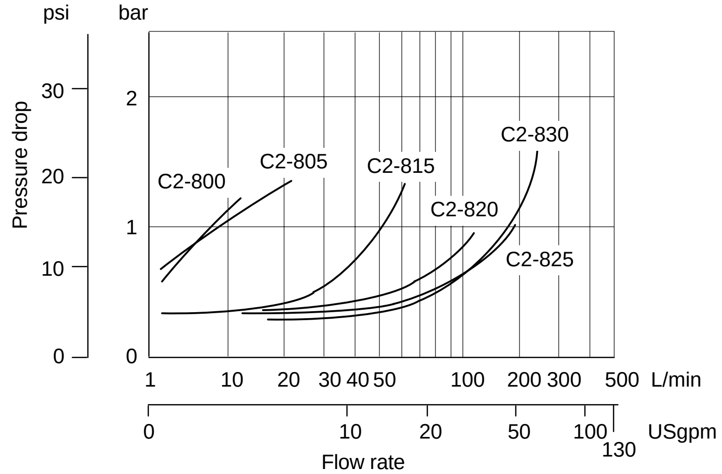
Models C5G(V)-8**



Models DT8P1



Models C2-8**



Appendix E

Data Sheet Pressure Relief Valve

Series CG-03 Relief Valves

Model Code

(F3) - CG - 03 - C (V) - 10 - (S81)

1 2 3 4 5 6 7

1 Seals

F3 – For mineral oil and fire resistant fluids
Blank – Omit if not required

2 Model Series

C – Relief valve
G – Subplate mounted

3 Valve Size

03 – 9,525 mm ($^{3}/_{8}$ in) nominal size

4 Pressure Range

B – 5 to 70 bar (75 to 1000 psi)
C – 35 to 140 bar (500 to 2000 psi)
F – 100 to 210 bar (1500 to 3000 psi)

5 High Vent Spring

V – Optional high vent spring installed
Blank – Omit if not required

6 Design Number

Subject to change. Installation dimensions same for designs 10 through 19.

7 Special Feature Suffix

S81 – Optional handwheel control installed

General Information

Series CG-03 valves utilize balanced piston type construction. They are designed for applications requiring an adjustable pressure relief valve to limit system pressure to a desired maximum.

Pressure Range

The available pressure ranges for this valve are from 5 to 70 bar (75 to 1000 psi), from 35 to 140 bar (500 to 2000 psi), and from 100 to 210 bar (1500 to 3000 psi). Select an appropriate pressure range that will prevent excessively high working pressures from being imposed on the pump and other equipment.

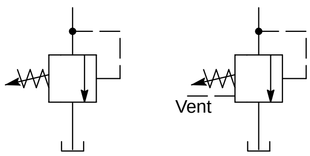
Pressure Adjustment

Pressure is adjusted by loosening a jam nut and turning an adjustment screw or optional handwheel. Turning clockwise increases pressure, and turning counterclockwise decreases pressure.

Tank Connection

If tank line back pressure exceeds system pressure by 7 bar (100 psi), a malfunction may occur. Pressure in the tank line is additive to the pressure setting. Contact your Vickers representative for alternatives.

Graphical Symbols



Ratings

The CG-03 is rated for a maximum pressure of 210 bar (3000 psi). Rated capacity is 0 to 30 l/min (0 to 8 USgpm). The following table lists approximate minimum venting pressures for the three available pressure ranges of the CG-03 with and without high vent springs.

Model	Pressure Range bar (psi)	Percentage of Maximum Rated Capacity			
		25%	50%	75%	100%
		Minimum Venting Pressure bar (psi)			
CG-03-B-10	5 to 70 (75 to 1000)	25	27	28	32
CG-03-BV-10		74	75	78	81
CG-03-C-10	35 to 140 (500 to 2000)	25	27	28	32
CG-03-CV-10		74	75	78	81
CG-03-F-10	100 to 210 (1500 to 3000)	25	27	28	32
CG-03-FV-10		74	75	78	81

Appendix F

Data Sheet Pressure Sensor

1.1 SCP Mini pressure sensors

- ✓ **Stainless steel cell**
- ✓ **Small construction**
- ✓ **High burst pressure**
- ✓ **Resistant to pressure peaks**
- ✓ **Shock and vibration-proof**
- ✓ **Wide media resistance**
- ✓ **High linearity**
- ✓ **Long-term stability**

The Mini-SCP pressure sensor was designed for industrial application requirements and is used in control, regulation and monitoring systems where rapid pressure-dependent analogue signals are needed.
The SCP-Mini pressure sensor is outstanding because of its compact construction, high linearity and excellent interference resistance.

Construction
The SCP-Mini includes only a few active components – the sensor element, a signal-processing ASIC and a converter switch.
The ASIC is a programmable precision CMOS-ASIC with EEPROM data memory and analogue signal path, which is qualified for an extended working temperature range. Because of electronic calibration, a small total error and high long-term stability are achieved. The electronics are resistant to the effects of electromagnetic interference.
Pressure is captured with a zero-point and long-term stable measurement cell.
The hermetically welded stainless steel membrane is vacuum tight and highly resistant to bursting.
The standardised G1/4 BSPP corrosion-resistant stainless steel process connection, in so far as it is compatible with stainless steel, guarantees wide-ranging media resistance.

Applications
Plenty of electrical output signals and plug-in connectors guarantee a wide spectrum of applications.
This sensor is eminently suitable for permanent series usage in hydraulic and pneumatic applications, thanks to its long durability, high accuracy, high reliability and rugged stainless steel construction.

SCP Mini	004	006	010	016	025	040	060	100	160	250	400	600
pressure range * **P_N (bar)**	0...4	0...6	0...10	0...16	0...25	0...40	0...60	0...100	0...160	0...250	0...400	0...600
overload pressure **P_{max} (bar)**	2 times											
burst pressure **P_{Burst} (bar)**	3- times											2,5-times

Pressure connection	
pressure connection	G1/4A BSPP
	DIN 3852 T11, form E
erosion bore	0,6 mm
	ED-seal FKM
Material	
parts in contact with media	FKM; stainless steel 1.4542,1.4548; 17-4PH
housing	stainless steel 1.4301
protection class	IP67 DIN EN 60529 (with DIN EN 175301-803 form A plug IP65)
Plug-in connection	
4-pole; M12x1; IP67	
4-pole; DIN EN 175301-803 form A; IP65	
Electrical connection	
short circuit protect'n; reverse polarity protect'n; protect'n class 3	
Accuracy	
characteristic curve deviation	± 0,5 % FS start point setting
General	
response time	≤ 1 ms
long-term stability	< 0,1 % FS/a
weight	ca. 80 g
load reversals	≥ 20 Mio.

Environmental conditions	
environmental temperature range	-40…+85 °C
fluid temperature range	-40…+125 °C
compensated range	-20…+85 °C
storage temperature	-40…+125 °C
temperature coefficient	≤ ± 0,3 % FS/10 K
vibration resistance	IEC 60068-2-6; ± 5 mm; 10 Hz...32 Hz 200 m/s²; 32 Hz...2 kHz
shock resistance	IEC 60068-2-29: 500 m/s²; 11 ms IEC 60068-2-32: 1 m (free fall onto steel plate)
Electromagnetic compatibility	
interference emissions	DIN EN 61000-6-3
interference resistance	DIN EN 61000-6-2

Output signal	0…20 mA 3-core	4…20 mA 3-core	4…20 mA 2-core	0...10 V 3-core
auxiliary energy $+U_b$ (U_{DC})	9…30 V	9…30 V	12...30 V	12...30 V
working resistance max.	(U_b-9 V)/28 mA	(U_b-9 V)/30 mA	(U_b-12 V)/20 mA	35 kΩ

* see page 82, 6.3

Appendix G

Data sheet NI USB-6008

Technical Sales
Norway
66 90 76 60
ni.norway@ni.com

NI USB-6008

12-Bit, 10 kS/s Low-Cost Multifunction DAQ

- 8 analog inputs (12-bit, 10 kS/s)
- 2 analog outputs (12-bit, 150 S/s); 12 digital I/O; 32-bit counter
- Bus-powered for high mobility; built-in signal connectivity
- OEM version available
- Compatible with LabVIEW, LabWindows/CVI, and Measurement Studio for Visual Studio .NET
- NI-DAQmx driver software and NI LabVIEW SignalExpress LE interactive data-logging software

Overview

The National Instruments USB-6008 provides basic data acquisition functionality for applications such as simple data logging, portable measurements, and academic lab experiments. It is affordable for student use, but powerful enough for more sophisticated measurement applications. Use the NI USB-6008 with the included ready-to-run data logger software to begin taking basic measurements in minutes, or program it using LabVIEW or C and the included NI-DAQmx Base measurement services software for a custom measurement system.

To supplement simulation, measurement, and automation theory courses with practical experiments, NI developed a USB-6008 Student Kit that includes a copy of the LabVIEW Student Edition. These kits are exclusively for students, giving them a powerful, low-cost, hands-on learning tool. Visit the NI academic products page for more details.

For faster sampling, more accurate measurements, and higher channel count, consider the NI USB-6210 and NI USB-6211 high-performance USB data acquisition devices.

Every USB data acquisition module includes a copy of NI LabVIEW SignalExpress LE so you can quickly acquire, analyze and present data without programming. In addition to LabVIEW SignalExpress, USB data acquisition devices are compatible with the following versions (or later) of NI application software – LabVIEW 7.x, LabWindows™/CVI 7.x, or Measurement Studio 7.x. USB data acquisition modules are also compatible with Visual Studio .NET, C/C++, and Visual Basic 6.

Specifications

Specifications Documents

- Detailed Specifications (2)
- Data Sheet

Specifications Summary

Specifications Summary

General

Product Name	USB-6008
Product Family	Multifunction Data Acquisition
Form Factor	USB
Operating System/Target	Windows , Linux , Mac OS , Pocket PC
DAQ Product Family	B Series
Measurement Type	Voltage
RoHS Compliant	Yes
Analog Input	
Channels	8 , 4
Single-Ended Channels	8
Differential Channels	4
Resolution	12 bits
Sample Rate	10 kS/s
Throughput	10 kS/s
Max Voltage	10 V
Maximum Voltage Range	-10 V , 10 V
Maximum Voltage Range Accuracy	138 mV
Minimum Voltage Range	-1 V , 1 V
Minimum Voltage Range Accuracy	37.5 mV
Number of Ranges	8
Simultaneous Sampling	No
On-Board Memory	512 B
Analog Output	
Channels	2
Resolution	12 bits
Max Voltage	5 V
Maximum Voltage Range	0 V , 5 V
Maximum Voltage Range Accuracy	7 mV
Minimum Voltage Range	0 V , 5 V
Minimum Voltage Range Accuracy	7 mV

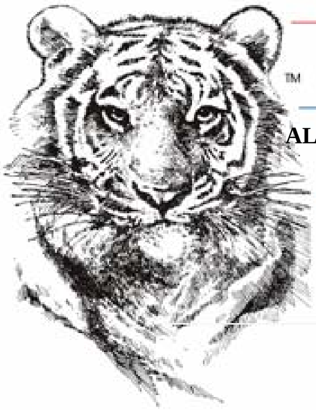
Update Rate	150 S/s
Current Drive Single	5 mA
Current Drive All	10 mA
Digital I/O	
Bidirectional Channels	12
Input-Only Channels	0
Output-Only Channels	0
Number of Channels	12 , 0 , 0
Timing	Software
Logic Levels	TTL
Input Current Flow	Sinking , Sourcing
Output Current Flow	Sinking , Sourcing
Programmable Input Filters	No
Supports Programmable Power-Up States?	No
Current Drive Single	8.5 mA
Current Drive All	102 mA
Watchdog Timer	No
Supports Handshaking I/O?	No
Supports Pattern I/O?	No
Maximum Input Range	0 V , 5 V
Maximum Output Range	0 V , 5 V
Counter/Timers	
Counters	1
Buffered Operations	No
Debouncing/Glitch Removal	No
GPS Synchronization	No
Maximum Range	0 V , 5 V
Max Source Frequency	5 MHz
Minimum Input Pulse Width	100 ns
Pulse Generation	No
Resolution	32 bits

Timebase Stability	50 ppm
Logic Levels	TTL
Physical Specifications	
Length	8.51 cm
Width	8.18 cm
Height	2.31 cm
I/O Connector	Screw terminals

Appendix H

Data sheet ESSO UNIVIS n32

Imperial Oil*



Product Data Sheet

UNIVIS™ N

ALL-WEATHER ANTI-WEAR HYDRAULIC OILS FOR MOBILE AND INDUSTRIAL EQUIPMENT

January 2007

UNIform VIScosity; an essential quality from which the name UNIVIS N was derived. These premium-quality hydraulic oils offer the following features and benefits:

- ♦ **Stable, high-performance anti-wear additives**
- ♦ **Excellent rust and oxidation control**
- ♦ **High VI optimized low temperature fluidity, low pour points for excellent all weather performance**
- ♦ **Designed to resist viscosity change with temperature for mobile marine and outdoor applications**
- ♦ **Designed to meet major hydraulic pump manufacturer performance requirements**
- ♦ **Excellent shear stability for stay in grade performance**
- ♦ **Fast air release, foaming control and water demulsification**
- ♦ **Good hydrolytic stability**

Primary Applications

The sophisticated hydraulic systems of today rely on fluids that can resist changes in viscosity as temperatures rise or fall. This is particularly important in hydraulic systems used outdoors on both fixed and mobile equipment.

UNIVIS N oil is available in six viscosity grades, thus providing a product for all the climatic conditions experienced in Canada. All grades conform to the ISO viscosity classification system and are entirely compatible with one another. They can be mixed together in any ratio with no adverse effects.

The high, all-weather efficiency of UNIVIS N oils has enabled many operators to standardize on one grade in all outdoor hydraulic equipment year round, without seasonal grade changes.

Selecting the Correct Viscosity
For optimum hydraulic system efficiency, the fluid should conform to the pump manufacturer's recommended viscosity. This will result in:

- ♦ maximum power
- ♦ extended component life
- ♦ rapid response to the operator's commands

To select the correct grade of UNIVIS N oil, first refer to the operator's manual of the equipment for the proper viscosity grade. The Machine Builders' Guidelines (page 3) lists five major pump manufacturers' recommendations.

Referring to Vickers, note that fluids with viscosities of 13 to 54 cSt at the operating temperature, are recommended. Assuming an operating temperature of 80°C, the viscosity/ temperature chart on the next page indicates

UNIVIS N68 has a viscosity of 17 cSt @ 80°C, nicely inside the 13 - 54 cSt range recommended.

The viscosity/temperature chart can also be used to determine that, for UNIVIS N68, the maximum viscosity of 54 cSt is reached at an operating temperature of about 45°C, and the minimum viscosity recommended (13 cSt) occurs at about 92°C.

Assuming under no-load conditions, a start-up viscosity of 9300 cSt can be tolerated (see note under equipment builders' table), UNIVIS N68 reaches this viscosity at about -22°C. Similarly, for start-up under load, a maximum viscosity of 860 cSt is recommended (see table). UNIVIS N68 reaches this viscosity at about 0°C.

Performance Features

Wear Protection
UNIVIS N oils contain a powerful anti-wear additive which protects pumps, motors, valves and hydraulic cylinder assemblies against undue wear - even under very high pressure conditions.

Rust and Corrosion Control
All grades pass the rust test ASTM D665A and B (distilled water and synthetic sea water). As a result, all oil-wetted parts, ferrous and non-ferrous, are protected from rust and corrosion.

Stable Viscosity
As a hydraulic oil, the stable viscosity of UNIVIS N means more uniform functioning of the hydraulic system over a wider temperature range - whether in a small hydraulic door check, or in modern tree harvesting machinery including feller-bunchers and skidders. The high viscosity index values can be seen in the Typical Properties section of this data sheet.

Shear Stability
UNIVIS N oils are blended to provide “Stay-in-Grade” viscosity performance through the use of leading edge, shear stable viscosity improver additive packages. Other viscosity improver additives mechanically break down during use and cause the oils to lose some of their wide-temperature range properties. UNIVIS N oils, using superior additive technology, experience only marginal shear-down even under the most extreme conditions, thus retaining their excellent viscosity/temperature properties.

Foam Resistance
Foaming can cause a system to react in a “spongy” manner. It can also cause cavitation and rapid wear. UNIVIS N oils contain an effective foam inhibitor to ensure optimum system performance.

Demulsibility
Water is often present in hydraulic systems. Changes in temperature cause condensation to take place, and water formation results. UNIVIS N oils separate water quickly.

Hydrolytic Stability
UNIVIS N oils are hydrolytically stable. They maintain their effectiveness even in the presence of small amounts of water. Hydrolysis is a decomposition reaction caused by water, which results in the formation of products which can be very corrosive to copper and its alloys. It also reduces the effectiveness of some anti-wear additives.

Hydraulic Pump Performance
UNIVIS N is recommended for most hydraulic pump manufacturer application requirements, including Dension, Vickers, and Cincinnati-Milacron.

Operating Precautions

Because of the excellent demulsibility qualities inherent in the UNIVIS product line, water tends to settle quickly. It is highly recommended that operators drain water daily from fluid reservoirs.

Severe hydraulic service applications, combined with wide seasonal temperature changes will, in most cases, require a seasonal grade change of UNIVIS. This will ensure that equipment viscosity demands are not compromised and that high system efficiency will result.

Precautions

UNIVIS N oils are manufactured from high quality petroleum base stocks, carefully blended with selected additives. As with all petroleum products, good personal hygiene and careful handling should always be practiced. Avoid prolonged contact to skin, splashing into the eyes, ingestion or vapour inhalation. Please refer to our ESSO Material Safety Data Sheet for further information. Note: This product is not controlled under Canadian WHMIS legislation.

Typical Properties

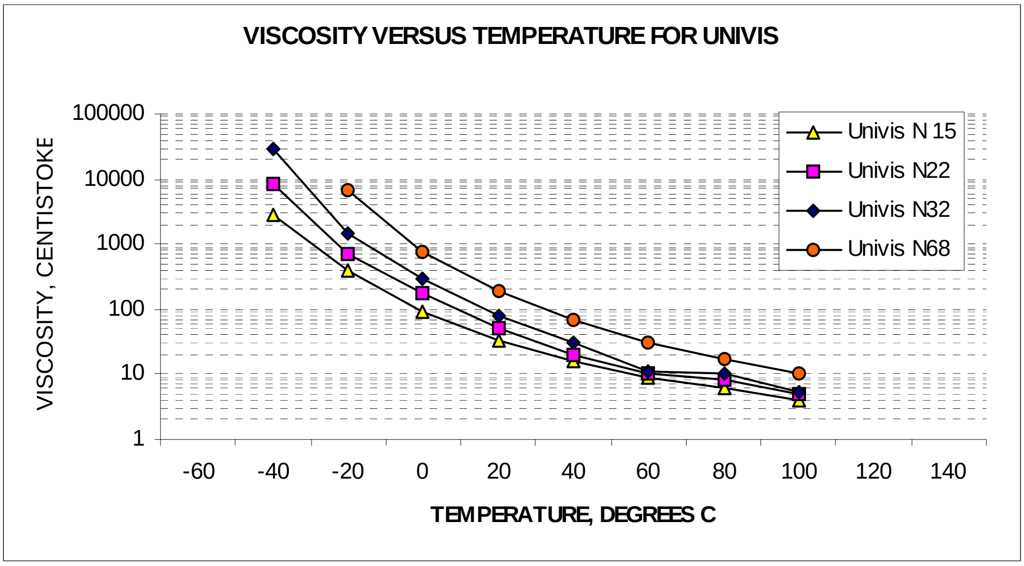
UNIVIS	N15	N22	N32	N46	N68	N100
Viscosity cSt @ 40°C cSt @ 100°C	 15 3.9	 22 5.1	 32 6.6	 46 8.2	 68 10.7	 100 14.9
Viscosity Index	163	175	172	158	151	155
Pour Point, °C	-57	-54	-48	-45	-39	-36
Flash Point, °C	150	156	204	218	226	242
Rust Protection, 24 hrs @ 60°C in synthetic sea water	pass	pass	pass	pass	pass	pass
FZG , fls	11	11	12	12	12	12
Air Release, min.	-	2.4	2.7	2.8	3.8	-
Demulsibility after 15 minutes	40-40-0	40-40-0	40-40-0	40-40-0	40-40-0	40-40-0

The values shown above are representative of current production. Some are controlled by manufacturing and performance specifications while others are not. All may vary within modest ranges.

Machine Builders' Viscosity Guidelines For Hydraulic Oils

		Operating Viscosity		Start-Up (under load) *	Optimum
Manufacturer	Equipment	Min cSt	Max cSt	cSt	cSt
Hägglunds Denison Bulletin 2002-I	Piston Pumps	10	162	1618	30
	Vane Pumps	10	108	862	30
Racine Form No. S-106	FA; RA; K vane pumps	15	216	864	26-54
	Q; Q6; SV-10,15,20,25 vane pumps	21	216	864	32-54
	SV-40, 80 and 100 vane pumps	32	216	864	43-65
	Radial piston pumps	10	65	162	21-54
	Axial piston pumps	14	450	647	32-65
Vickers Data Sheet I-286	In-line piston pumps & motors	13	54	220	--
	Angle piston, vane and gear pumps & motors	13	54	860	--
	MHT vane motors	13	54	110	--
Sundstrand	Piston pumps, Series 30	6.4	--	--	13
	Other piston units	9.0	--	--	13
Mannesmann Rexroth	Axial piston pumps	16	100	1000	16-36
	V2 vane pumps, MZ motors	16	160	800	--
	V3, V4 vane pumps	25	200	800	25-160
	V5 vane pumps	16	200	800	--
	R4 radial piston pumps	10	200	--	--
	G2, G3, G4 gear pumps & motors	10	300	1000	--
	G8, G9, G10 gear pumps	10	1000	--	25-85
	GM gear pumps	20	300	1000	--

** Imperial Oils experience shows that most systems can be started at 8000 cP (9300 cSt) under no load conditions. Full load can be applied when the start-up (under load) viscosity is reached (see above)*



VISCOSITY VERSUS TEMPERATURE FOR UNIVIS
VISCOSITY, CENTISTOKE
100000
10000
1000
100
10
1
-60
-40
-20
0
20
40
60
80
100
120
140
TEMPERATURE, DEGREES C
Univis N 15
Univis N22
Univis N32
Univis N68

Appendix I

SimulationX Components

The value of the different parameters are presented under each component in this chapter.

I.1 Preset

This component represents the electrical motor.

Parameter	Parameter value
Kind	Rotational Speed
Rotational speed	1500 rpm
Initial angle of rotation	0 rad

Table I.1: Preset Parameter

I.2 Variable Displacement Pump/Motor

This component is the main pump in this model.

Parameter	Parameter value
Max. Displacement Volume	10,55 cm^3/rev
Upper Limit (> 0)	100%
Lower Limit	-100%
Dead Volume Port A	50 cm^3
Dead Volume Port B	50 cm^3
Angular Difference	0 rad
Friction Description	No Friction Losses
Leakage Description	Volumetric Efficiency ηvol
Speed (at Working point)	1500 rpm
Pressure Difference (at WP)	13,176 bar
ηvol (at WP)	99 %
Consider External Leakage	”Checked”
Ratio of Ext./Int. Leak.	50%

Table I.2: Main pump Parameter

I.3 Constant Displacement Pump/Motor

This component is used as the boost pump and the motor in this model.

Parameter	Parameter value
Displacement Volume	1,7 cm^3/rev
Dead Volume Port A	50 cm^3
Dead Volume Port B	50 cm^3
Angular Difference	0 rad
Friction Description	No Friction Losses

Table I.3: Boost pump Parameter

The data table for the hydro-mechanical efficiency are extracted from the data sheet in appendix A

Parameter	Parameter value
Displacement Volume	10,55 cm^3/rev
Dead Volume Port A	50 cm^3
Dead Volume Port B	50 cm^3
Angular Difference	0 rad
Friction Description	Hydro-Mechanical Efficiency ηhm
Sticking Friction Torque	5 Nm
Coulumb Friction Torque	1,68 Nm
Consider Speed-Dependency	”Checked”
Maximum Speed	3000 rpm
ηhm(n) Data Table (at dp Max)	
Leakage Description	Volumetric Efficiency ηvol
Speed (at Working point)	1453 rpm
Pressure Difference (at WP)	13,176 bar
ηvol (at WP)	98 %
Consider External Leakage	”Checked”
Ratio of Ext./Int. Leak.	50%

Table I.4: Motor Parameter

I.4 Volume

This is used to give the pipes a volume. It is based on the length of the pipe and the inner diameter of the pipe. Since there are two volumes on both the A and B side, the volume size would be half of the total volume of that side.

Parameter	**Parameter value**
Volume 3 and 6	0.04485015 l
Volume 4 and 5	0.0375342 l
Volume 1	0.0693428 l
Volume 2	0.0947898 l
Wall Capacity	0

Table I.5: Volume Parameter

I.5 Flow Sensor

This sensor is not on the actual rig, but is added in the model for analysis purposes.

I.6 Pipe

This component is used to include the effect of pipelines in the model.

Parameter	**Parameter value**
Pipe Length	141 and 118 cm
Inner Diameter	9 mm
Average Wall Roughness	10 μm

Table I.6: Pipe Parameter

I.7 Check Valve

This component represents the check valves on the rig.

Parameter	Parameter value
Cracking Pressure	0,35 bar
Full Opening A-> B	0,5 bar
Pressure Drop(Throttle Range)	10 bar
Flow (Throttle Range)	60 l/min
Reference Density	0,89 g/cm^3
Critical Reynolds Number	20
Turb. Flow Coefficient	0,7

Table I.7: Check Valve Parameter

I.8 Pressure Relief Valve

There are two pressure relief valves in this model. The one near the boost pump is actually build into the pump on the actual rig. since the datasheets for the motor and the valve have not been found, the standard values is used.

Parameter	Parameter value
Set Pressure	65 bar
Static Pressure Behavior	0,01 $bar/(l/min)$
Pressure Drop(Throttle Range)	100 bar
Flow (Throttle Range)	120 l/min
Reference Density	0,89 g/cm^3

Table I.8: Pressure Relief Valve Parameter

I.9 Inertia

This component represents the rotating mass connected to the motor.

Parameter	**Parameter value**
Moment of inertia	0,1635 kgm^2
Initial Angle of Rotation	0 rad
Initial Rotational Speed	0 rad/s

Table I.9: Inertia Parameter

I.10 External Torque

This component represents all the different friction losses in the system

Appendix J

SimHydraulics Components

The different components used in the model explained here. A short description of the parameter is provided. The description is based on the SH manual.

J.1 Pipes

Table J.1: Pipe Parameters

Parameter	**Parameter value**	**Description**
Pipe cross section type	Circular	Choose if the pipe is circular or not
Pipe internal diameter	9 mm	The inner diameter of pipe
Geometrical shape factor	64	The parameter is used for computing friction factor at laminar flow and depends of the shape of the pipe cross section
Pipe length	1,18 and 1,41 m	The Length of the pipes
Aggregate equivalent length of local resistances	4 cm	This parameter represents total equivalent length of all local resistances associated with the pipe
Internal surface roughness height	$10\mu m$	The roughness of the inside of the pipe
Laminar flow upper margin	2300	Reynolds number at fully developed laminar flow
Turbulent flow lower margin	2301	Reynolds number at fully developed turbulent flow
Pipe wall type	Rigid	Choose if the pipe is rigid or flexible
Specific heat ratio	1,4	Gas-specific heat ratio

J.2 T-Junction

Table J.2: T-junction Parameters

Parameter	**Parameter value**	**Description**
Main pipe diameter	8 mm	The internal pipe diameter of the main run
Branch pipe diameter	8 mm	The internal pipe diameter of the branch
A-B pressure loss coefficient	0,1	The pressure loss coefficient between ports A and B when fluid flows in the direction from A to B
B-A pressure loss coefficient	0,1	The pressure loss coefficient between ports A and B when fluid flows in the direction from B to A
A-A1 pressure loss coefficient	1,3	The pressure loss coefficient between ports A and A1 when fluid flows in the direction from A to A1
A1-B pressure loss coefficient	1,3	The pressure loss coefficient between ports A1 and B when fluid flows in the direction from A1 to B
B-A1 pressure loss coefficient	1,3	The pressure loss coefficient between ports A1 and B when fluid flows in the direction from B to A1
Critical Reynolds number	2300	The maximum Reynolds number for laminar flow

J.3 Check Valves

Table J.3: Check Valve Parameters

Parameter	**Parameter value**	**Description**
Maximum passage area	32,4 mm^2	Valve passage maximum cross-sectional area
Cracking pressure	0,345 bar	Pressure level at which the orifice of the valve starts to open
Maximum opening pressure	0,5 bar	Pressure differential across the valve needed to fully open the valve.
Flow discharge coefficient	0,7	Semi-empirical parameter for valve capacity characterization. Its value depends on the geometrical properties of the orifice
Leakage area	$1 \cdot 10^{-12} m^2$	The total area of possible leaks in the completely closed valve
Critical Reynolds number	20	The maximum Reynolds number for laminar flow

J.4 Pressure Relief Valve

Table J.4: Pressure Relief Valve Parameters

Parameter	Parameter value	Description
Maximum passage area	10,04mm^2	Valve passage maximum cross-sectional area
Valve pressure setting	65 bar	Preset pressure level, at which the orifice of the valve starts to open
Valve regulation range	1 bar	Pressure differential across the valve needed to fully open the valve.
Flow discharge coefficient	0,7	Semi-empirical parameter for valve capacity characterization. Its value depends on the geometrical properties of the orifice, and usually is provided in textbooks or manufacturer data sheets
Leakage area	$1 \cdot 10^{-12} m^2$	The total area of possible leaks in the completely closed valve
Critical Reynolds number	20	The maximum Reynolds number for laminar flow

J.5 Main Pump

Table J.5: Main Pump Parameters

Parameter	**Parameter value**	**Description**
Model parameterization	By maximum displacement and control member stroke	
Maximum displacement	10,55 cm^3/rev	Displacement at full control member stroke
Maximum stroke	15 mm	Maximum control member stroke
Volumetric efficiency	0,75	
Total efficiency	0,75	The volumetric efficiency multiplied with the hydro-mechanical efficiency
Nominal pressure	20	Pressure at working point
Nominal angular velocity	1500 rpm	Angular velocity at working point
Nominal kinematic viscosity	31,81 cSt	Kinematic viscosity at working point

J.6 Boost Pump

Table J.6: Boost Pump Parameters

Parameter	Parameter value	Description
Pump displacement	1,7cm^3/rev	See main pump
Volumetric efficiency	0,99	See main pump
Total efficiency	0,75	See main pump
Nominal pressure	10 bar	See main pump
Nominal angular velocity	1500 rpm	See main pump
Nominal kinematic viscosity	31,81 cSt	See main pump

J.7 Hydraulic Fluid

Used to specify the hydraulic fluid.

Table J.7: Hydraulic Fluid Parameters

Parameter	**Parameter value**	**Description**
Hydraulic fluid	ISO VG 32	
Relative amount of trapped air	0	Amount of entrained, nondissolved gas in the fluid
System temperature	40^oC	Fluid temperature in Celsius
Viscosity derating factor	1	Proportionality coefficient that you can use to adjust fluid viscosity, if needed

J.8 Elbow

Simulates a elbow in the pipe.

Table J.8: Elbow Parameters

Parameter	**Parameter value**	**Description**
Elbow internal diameter	12 mm	The internal diameter of the pipe
Elbow angle	90^o	The angle of the bend
Elbow type	Smoothly curved elbow	bend type
Critical Reynolds number	2300	The maximum Reynolds number for laminar flow

J.9 Pipe Bend

Simulates a bend in the pipe.

Table J.9: Pipe Bend Parameters

Parameter	**Parameter value**	**Description**
Pipe internal diameter	9mm	The internal diameter of the pipe
Bend angle	90^o	The angle of the bend
Bend radius	30mm	The radius of the bend
Critical Reynolds number	2300	The maximum Reynolds number for laminar flow
Internal surface roughness height	10 μm	See pipe

J.10 3-phase Electro Motor

To simulate the electro motor, a block who simulates angular speed is used. It just need a desired speed compared to the reference. See figure 5.8.

J.11 Solver Block

The solver block is needed to give information about the global surroundings.

J.12 Hydraulic Motor

Table J.10: Hydraulic Motor Parameters

Parameter	**Parameter value**	**Description**
Motor displacement	10,55 cm^3/rev	See main pump
Volumetric efficiency	0,95	See main pump
Total efficiency	0,75	See main pump
Nominal pressure	100 bar	See main pump
Nominal angular velocity	1453 rpm	See main pump
Nominal kinematic viscosity	31,81 cSt	See main pump

J.13 Sensors

The sensors that are used in this model is:

- Flow sensors
- Tachometer
- Torque sensor
- Pressure sensor

Appendix K

Matlab code

M-script for importing data from Excel

```
clear all
%% Global parameters
k = 10.55/62.83;
J = 0.1635;
my = 11.826e-9;

%% Reading in measurement data for Test 1
t_1 = xlsread('test1','A2:A1203')';
p1_1 = xlsread('test1','C2:C1203')';
p2_1 = xlsread('test1','B2:B1203')';
n_1 = xlsread('test1','D2:D1203')';
dp_1 = p1_1-p2_1;
n_1_r = n_1.*(pi/30);
%% Reading in measurement data for Test 2
t_2 = xlsread('test2','A2:A524')';
p1_2 = xlsread('test2','C2:C524')';
p2_2 = xlsread('test2','B2:B524')';
n_2 = xlsread('test2','D2:D524')';
dp_2 = p1_2-p2_2;
```

```
n_2_r = n_2.*(pi/30);
%% Reading in measurement data for Test 3
t_3 = xlsread('test3','A2:A602')';
p1_3 = xlsread('test3','C2:C602')';
p2_3 = xlsread('test3','B2:B602')';
n_3 = xlsread('test3','D2:D602')';
dp_3 = p1_3-p2_3;
n_3_r = n_3.*(pi/30);
%% Reading in measurement data for Test 4
t_4 = xlsread('test4','A2:A1122')';
p1_4 = xlsread('test4','C2:C1122')';
p2_4 = xlsread('test4','B2:B1122')';
n_4 = xlsread('test4','D2:D1122')';
dp_4 = p1_4-p2_4;
n_4_r = n_4.*(pi/30);
```

M-script for checking if a test can be used to find the true parameters used in section 7.2.

```
%% Reading in measurement data
t_1 = xlsread('test1_v8_','A2:A10001')';
dp_1 = xlsread('test1_v8_','C2:C10001')';
n_1 = xlsread('test1_v8_','B2:B10001')';
%%
clc
dt=10e-3;                         %setting the timestep

J = 0.1635;                       %Load inertia
ktrue=10.55/62.83;                %True value for k
k1true=0.013;                     %True value for k_mM
k3true=5.689e-8;                  %True value for k_hM

nddot = [0 diff(n_1)/dt];         %Derivating the speed
```

```
%%Plotting of data
plot(t_1,n_1); grid on;
hold on;
plot(t_1,cumsum(n_1)*dt,'r');
plot(t_1,cumsum(sign(n_1))*dt,'g');
plot(t_1,cumsum(nddot)*dt,'k');
hold off;



%%Integrating
THETA=sum(nddot*pi/30)*dt;
DP=sum(dp_1)*dt;
N1=sum(n_1)*dt;
N2=sum(n_1.^2)*dt;

%%Constructing A-matrix
A=[DP -N2];

%% calculating right and left side of test equation
R = A*[(ktrue-k1true); k3true]
L = THETA*J
%% comparing each side against each other
R/L
```

M-script for calculating the friction constants in section 7.2.

```
dt = 10e-3;

%% Test 1 and Test 2
theta_1_dd = [0 diff(n_1_r)/dt];

A=[-trapz(t_1,dp_1),-trapz(t_1,n_1.^2);...
   -trapz(t_2,dp_2),-trapz(t_2,n_2.^2)]
```

```
C=[J*trapz(t_1 ,theta_1_dd)-k*trapz(t1 ,dp_1 );...
    J*trapz(t_2 ,theta_2_dd)-k*trapz(t2 ,dp_2 )];

B = A\C;
k_mM = B(1)
k_hM = B(2)
%% Test 3

k_vM = (-k_hM*trapz(t_2 ,n_2.^2))/(my*trapz(t_2 ,n_2 ));

%% Test 4
k_sM = ((k-k_mM)*trapz(t_4 ,dp_4 )...
         -k_hM*trapz(t_4 ,n_4.^2))...
         /trapz(t_4 ,sign(n_4 ));

%% Printing values
k_mM
k_vM
k_hM
k_sM

%Checking the error from the actual value
pk1 = (k_mM/0.013)
pk2 = (k_vM/2345)
pk3 = (k_hM/5.689e-8)
pk4 = (k_sM/0.0233)
```

M-script for calculating the friction constants in section 7.1.

```
%% Scaling factors
g_1 = 1e5;
g_2 = 1e9;
g_3 = 1e6;
```

```
g_4 = 1e7;

%% A-matrix
A = [-trapz(t_1,dp_1),0,-trapz(t_1,n_1.^2),0;
    -trapz(t_2,dp_2.*n_2_r),...
    -trapz(t_2,my.*n_2.*n_2_r),...
    -trapz(t_2,n_2.^2.*n_2_r),...
    -trapz(t_2,sign(n_2).*n_2_r);...
    -trapz(t_3,dp_3),-trapz(t_3,my.*n_3),...
    -trapz(t_3,n_3.^2),-trapz(t_3,sign(n_3));...
    -trapz(t_4,dp_4),-trapz(t_4,my.*n_4),...
    -trapz(t_4,n_4.^2),-trapz(t_4,sign(n_4))];

A(1,1:4) = A(1,1:4)./g_1;
A(2,1:4) = A(2,1:4)./g_2;
A(3,1:4) = A(3,1:4)./g_3;
A(4,1:4) = A(4,1:4)./g_4;

%% C-matrix
C = [-k*trapz(t_1,dp_1);...
    -k*trapz(t_2,dp_2.*n_2_r);...
    -k*trapz(t_3,dp_3);...
    J*n_4_r(length(n_4_r))-k*trapz(t_4,dp_4)];

C(1,1) = C(1,1)./g_1;
C(2,1) = C(2,1)./g_2;
C(3,1) = C(3,1)./g_3;
C(4,1) = C(4,1)./g_4;

%% Matrix check
s = svd(A)
c = cond(A)
```

```
%% Calculation of parameters
B = A\C;
D = B';
k_1 = D(1)
k_2 = D(2)
k_3 = D(3)
k_4 = D(4)
```

M-script for static simulation.

```
function res = static_simulation_bench_1(x)
%Parameters:
Dp=10.55; %Displacement pump [cm^3/rev]
np=1500; %Rotational speed pump/AC motor [rev/min]
Dm=10.55; %Displacement motor [cm^3/rev]
Dbp=1.7; %Displacement boost pump [cm^3/rev]
nbp=np; %Rotational speed boost pump [rev/min]
Klp=0.088921; %leakage coefficient pump [(l/(min*bar)]
Klm=0.055528; %leakage coefficient motor [(l/(min*bar)]
% KmM=0.17458; %motor constant for mechanical friction, k1
% KvM=-27037.36524; %motor constant for viscous friction, k2
% KhM=0; %motor constant for turbulent fiction, k3
% TsM=0.1933; %losses in the static friction [Nm], k4
my=1.635e-5; %Dynamic viscosity oil [bar*min]
myRef=1.635e-5; %reference Dynamic viscosity oil [bar*min]
pcr1=10; %crack pressure relief valve boost pump [bar]
pcr2=65; %crack pressure relief valve circuit [bar]
pcr_check=0.35; %pressure drop check valve [bar]
T_loss=1.691; %losses in motor & bearings
%unknown variables:
% x(1)=Q1   [l/min];
% x(2)=Q2   [l/min];
% x(3)=Q3   [l/min];
% x(4)=Q4   [l/min];
% x(5)=Q5   [l/min];
% x(6)=Q6   [l/min];
% x(7)=Q7   [l/min];
% x(8)=Q8   [l/min];
% x(9)=Q9   [l/min];
% x(10)=Qlp leakage flow pump [l/min];
% x(11)=Qlm leakage flow hydraulic motor[l/min];
```

```
% x(12)=P1   [bar];
% x(13)=P2   [bar];
% x(14)=P3   [bar];
% x(15)=P4   [bar];
% x(16)=nm  Rotational speed hydraulic motor [rev/min];
%Flow continuity, pressure nodes:
res(1)=x(1)-x(10)+x(2)-x(4)-x(6);%press node 1, P1
res(2)=x(6)-0.5*x(11)+x(3)-x(5)-x(1)+0.5*x(10);%pressnode 2, P2
res(3)=x(4)+x(5)-x(7);%Pressure node 3, P3
res(4)=x(8)+x(7)-x(9)-x(2)-x(3);%Pressure node 4, P4
%Flow continuity, pumps and actuator(hydraulic motor):
res(5)=x(1)-((np*Dp)/1000);%theoretical flow from pump
res(6)=x(8)-(nbp*Dbp)/1000;%theoretical flow from boost pump
res(7)=x(6)-((x(16)*Dm)/1000);%actual flow to hydraulic motor
%Static equilibrium of actuator(hydraulic motor):
res(8)=(Dm*(x(12)-x(13))/62.83) - T_loss;
%Configuration parameters:
res(9)=pcr1-x(15);
res(10)=x(7);
res(11)=x(2);
res(12)=x(14)-pcr2+1.0; %just below crack pressure
res(13)=x(4);
res(14)=x(13)-x(15)+pcr_check;
%Leakage flows:
res(15)=x(11)-((Klm*(x(12)-x(13)))/(my/myRef));
res(16)=x(10)-((Klp*(x(12)-x(13)))/(my/myRef));


clc;
%Solve the 16 equations to find the 16
%unknown variables (11 flows
%+ 4 pressures + rotational speed motor)
%The solutions are in hydraulic units:
```

```
options=optimset('MaxFunEvals'...
,1e3,'MaxIter',1e3);
x=fsolve('static_simulation_bench_1',...
ones(1,16)',options);
Q1=x(1)
Q2=x(2)
Q3=x(3)
Q4=x(4)
Q5=x(5)
Q6=x(6)
Q7=x(7)
Q8=x(8)
Q9=x(9)
Qlp=x(10)
Qlm=x(11)
P1=x(12)
P2=x(13)
P3=x(14)
P4=x(15)
nm=x(16)
```