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Master’s Programme in Mechanical Engineering

Design of temperature control system for a test bench of servo motor-operated pumps and motors

Subtitle: Designing a system for thermal dissipation through a PID-controlled heat exchanger for a test bench for servomotor-operated pumps and motors.
Building a Simscape/SIMULINK-based simulator for the system.

Francesco Santoro

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Abstract

This thesis addresses the design of a temperature control system in a hydraulic test bench. Temperature control is done by flushing part of the oil flow in the main system, which will be filtered and cooled by a freshwater flow in a heat exchanger. The amount of flushing and flow rates will be dependent on the temperature of the fluid we want to test the equipment with. The control is based on a PID controller.

Precise temperature control is fundamental for hydraulic component testing because the efficiency of components greatly varies with fluid viscosity, which in turn varies significantly as a function of temperature.

The obtained system exhibits the following properties:

- the control seemed stable and changes in some system parameters did not deteriorate the operability
- the target value for the pump's inlet temperature was able to achieve
- the use of closed-loop control in thermal control may speed up the actual pump testing, which has previously been a slow process because of long transition times between the operational points (rotational speed, pressure).
- Even though the heat exchanger performance parameters were not accurate, the thermal control system's performance seems to be adequate for the intended usage.

Keywords Test bench, Temperature control, Heat exchanger, Design.

Contents

Preface

I want to thank Instructor Kajaste Jyrki, professor Petri Kuosmanen and Supervisor Roberto Finesso for their guidance and patience.

I also want to thank my family and my partner for their support.

Francesco Santoro

Symbols and abbreviations

Symbols

η_{hmp} = hydromechanical efficiency of the pump

ε_p = displacement setting of the pump

D_p = displacement of the pump

Δ_p = pressure difference of the pump

M_{in} =input torque to pump

η_{volp} = volumetric efficiency of the pump

q_{ep} = effective flow of the pump

n_p = speed of the pump

Re = Reynolds number

Pr = Prandtl number

de = hydraulic diameter

L = length of the plate

η = dynamic viscosity of fluid

ηw = dynamic viscosity of the fluid at heat exchanger plate temperature, an average of hot and cold side fluid temperatures

h_{H2O}=water heat transfer coefficient

$h_{OIL.corr}$=oil heat transfer coefficient

$h_{OIL.corr.turb}$=oil heat transfer coefficient in case of turbulent flow

$Nu_{OIL.corr}$=Nusselt number oil laminar flow

$Nu_{OIL.corr.turb}$ = Nusselt number oil turbolent flow

1 Introduction

Nowadays, the hydraulic industry has been more and more concerned about efficiency. To study effectively new solutions, we have to be able to measure precisely the efficiency of a component in very controlled working conditions, such as fluid temperature (viscosity), fluid type, flow rates, and pressures.

The control of the temperature in a hydraulic machine testing rig is what we will focus on in this work, going through:

- why we want to do our research,
- what are the best solutions now available in terms of motor and pump architecture,
- the usual practices in testing.
- the heat exchanger choice, and its related control strategies,
- the explanation of the main parts of the design in Simulink.
- results we had in an 'easily' presented summary graph that shows how we can control the temperature of the fluid in the main testing system and its response time.

The research questions in this work are:

- Is it possible to build a reliable Simulink model of the system?
- Will the system be stable?
- Will the heat exchanger be enough?
- Will the cooling and heating times be acceptable?

The targets of this study were to:

- design of temperature control system for a test bench of servo motor-operated pumps and motors

- make a simulation model by using Simulink Simscape

- study the stability, performance, and cooling/heating times

The considered heat exchanger has been chosen because it is at disposal of the University of Aalto, so we can easily realize a real circuit by it.
This work has not been straightforward, I started initially designing the main testing rig, then I decided to focus on the temperature control system, being the temperature a really important factor in testing. I realized that designing such a system is not straightforward, and requires a lot of trial and error, starting from the architecture, to the PID tuning to the characterization of the heat exchanger.

1.1 Why energy efficiency of fluid power is so important

We will here consider the motives of our study.
(1) The analysis conducted by the U.S. Department of Energy (Office of Scientific and Technical Information) study shows that "fluid-powered systems consumed between 2.0 and 2.9 Quadrillion (1015) Btus (Quads) of energy per year; producing between 310 and 380 million metric tons (MMT) of Carbon Dioxide (CO_2). In terms of efficiency across all industries, fluid power system efficiencies range from less than 9% to as high as 60% (depending upon the application), with an average efficiency of 22%." This is a very low efficiency compared to what is achieved by thinking seriously about efficiency in the design phase of our hydraulic systems, using best practices and new control strategies.
Applications sectors of the fluid power system are very wide and their potential is huge.
A useful scheme of the main losses in architecture is in figure 1.

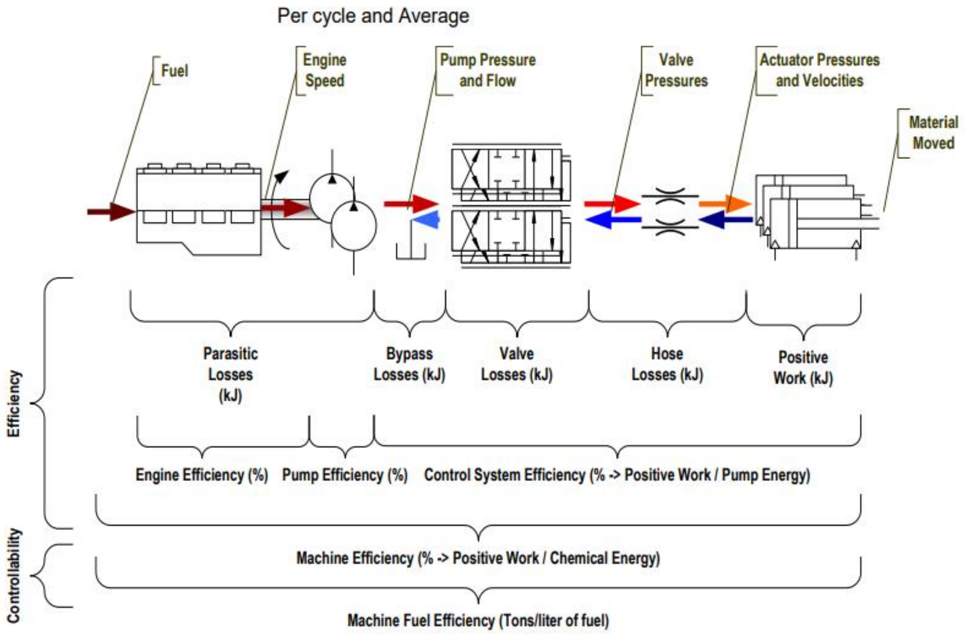


Figure 1: System losses

We can see where their proportions come from in figure 2:

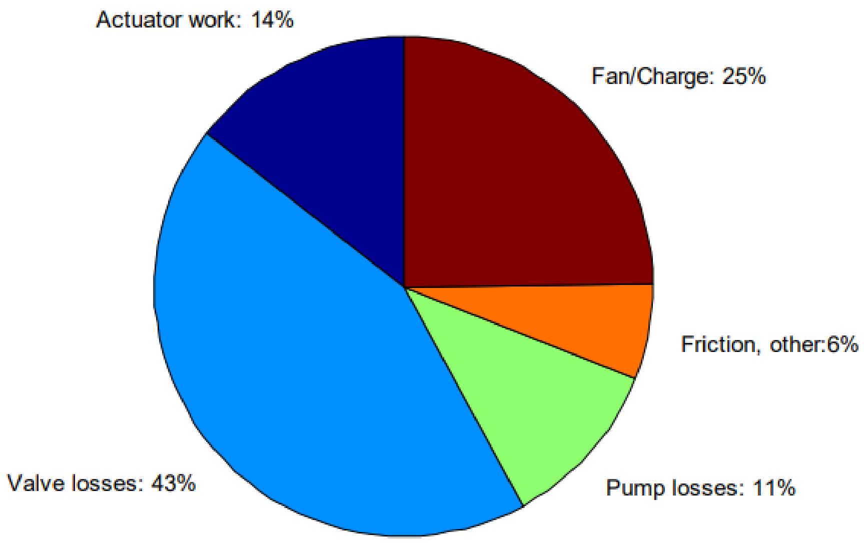


Figure 2: Energy losses in mobile load sensing hydraulic application (1)

The main strategies to reduce consumption are: (1)

- Load Sensing (LS) Systems – For constant pressure systems, the energy required to raise a light load (pressure times displaced actuator volume) is the same as the energy required to raise a light load.
- Energy Recuperation - Conventional valve-controlled systems use energy to both raise, and lower, a load. Palmberg is exploring mode switching and energy recuperation by replacing conventional spool valves with programmable valves that enable more flexibility in the direction of energy. His study suggests an additional 5% to 10% increase in efficiency
- Hydraulic Transformers – Figure 5 shows that there are tremendous losses through the control valves. Throttling losses introduce both energy losses as well as generation of heat. There is growing interest in the area of valveless controls eliminating throttling losses. To achieve this goal, there must be a variable displacement actuator or hydraulic transformer.
- Hybrid Hydraulics – Hybrid hydraulic systems size the primary power source for the average power demand and use hydraulic accumulators for energy storage. The accumulator can store energy during negative power flow (e.g., when a load is being lowered or a vehicle is braking) that can be efficiently used when accelerating or raising a load. Eaton demonstrated a 50% improvement in fuel economy and a 30% reduction in emissions by transforming a United Postal Service delivery truck into a hybrid hydraulic drive train.
- Weight Reduction – Eaton developed a new 5000 pounds per square inch hydraulic power system for the Airbus A380. The increased operating pressure reduced the overall weight of the hydraulic power system by a metric ton.

  The National Fluid Power Association conducted a workshop in 2010 focusing on energy efficiency. Subject matter experts from eighteen fluid power manufacturers projected that a 5-year effort focusing on

Best Practices could increase this efficiency by 5% to an average efficiency of 27%.

The impact of this improvement (increasing efficiency from 22% to 27% for an industry that consumes more than 2.0 Quads) would save the U.S. industry and consumers approximately 0.4 Quads/year.

This group also projected that a more aggressive 15-year research and development (R&D) effort, focusing on sensing, design, controls, and advanced materials could increase this efficiency by 15% to an average efficiency of 37%. The impact of this long-term improvement with an efficiency improvement from 22% to 37% for an industry consuming more than 2.0 Quads, would save U.S. industry and consumers approximately 0.8 Quads/year.

(1)

Fluid power (hydraulics and pneumatics), is a fundamental technology with unique capabilities. It is used pervasively in applications of great importance such as transportation, construction, agriculture, manufacturing, aerospace, and medical devices. Although a large consumer of energy, the technology is also typically low in energy efficiency. While there have been some attempts to replace fluid power with higher efficiency electric systems, fluid power's high performance and rugged operating conditions limit the impact of simple part replacement. Therefore, there are tremendous opportunities to improve efficiency through both Best Practices and a focused R&D program.

(1)

2 Literature review

2.1 The increasing importance of energy efficiency in hydraulics.

The energetic factor has become fundamental for the industry nowadays.
Electric motor-driven systems are used more and more in fluid power, also. Electro-hydraulic actuators (EHA) are examples of systems whereby avoiding throttling of fluid flow, large pressure losses can be avoided, and very good energy efficiency can be achieved.
Proficiency in an electric motor-driven system (EMDS) (such as a pump, compressor, or fan) is defined by the whole system, *i.e.* the multiplication of efficiencies for each component.
Energy savings options are available for single components and even for integrated systems.

A ***squirrel cage induction motor*** is a type of motor which functions based on the principle of electromagnetism and allows the conversion of electrical energy into rotational mechanical power; therefore, and is a powerful way to operate pumps, compressors, and conveyors, at established speed. These kinds of motors are broadly manufactured, available in standard catalog types and sizes and are easy to replace. These products are worldwide traded like commodities and have the asset of standardized features (such as frame size, output power or torque, rotational speed, insulation, and protective coatings).
(2,3) The standard IEC/EN 60034-2-1, which came into force in September 2007, introduces new rules concerning the testing methods to be used for determining losses and efficiency.
It provides a minimum energy performance standard (MEPS) as a specification, containing several performance requirements for an energy-using device, that effectively limits the maximum amount of

energy that may be consumed by a product in performing a specified task. MEPS can be matched, and this stimulates competition between companies to improve the efficiency of their products.

- The past standard, IEEE 112B, has discussed the repeatability coefficient and accuracy of the method.

It has now been fixed with the new IEC standard that offers a variety of testing methods, all of which account for stray load losses, i.e. the difference of the total iron losses and harmonic rotor cage losses between the no-load and load conditions.

Current IEC 60034-30:2008 specifies efficiency classes for single-speed, three-phase, 50 Hz, and 60 Hz, cage-induction motors.

The energy efficiency of the AC motor was classified by IEC 60034-30 (October 2008), then has been replaced by IEC 60034-30-1:2014.

This standard establishes a set of limit efficiency values based on frequency, number of poles, and motor power. No distinction is made between motor technologies, supply voltage, or motors with increased insulation designed specifically for converter operation even though these motor technologies may not all be capable of reaching the higher efficiency classes. This makes different motor technologies fully comparable concerning their energy efficiency potential.

Compared with IEC/EN 60034-30: 2008, it significantly expands the range of products covered with the inclusion of 8-pole motors and introduces the IE4 efficiency performance class for electric motors.

What are the efficiency classes defined by IEC/EN 60034-30-1: 2014?

The standard defines four IE (International Efficiency) efficiency classes for single-speed electric motors that are rated according to IEC 60034-1 or IEC 60079-0 (explosive atmospheres) and designed for operation on sinusoidal voltage.

1. Super-Premium efficiency IE4

2. Premium efficiency IE3
3. High efficiency IE2
4. Standard efficiency IE1

The new standard covers a wider scope of products.

The power range has been expanded to cover motors from 120 W to 1000 kW.

The coverage of the new standard includes:

Single-speed electric motors (single and three-phase), 50 and 60 Hz

- 2, 4, 6, or 8 poles
- Rated output PN from 0.12 kW to 1000 kW
- Rated voltage UN above 50 V up to 1 kV
- Motors, are capable of continuous operation at their rated power with a temperature rise within the specified insulation temperature class
- Motors, marked with any ambient temperature within the range of -20 °C to +60 °C
- Motors, marked with an altitude up to 4000 m above sea level

In short, as we can see in figure 3, the higher the output the higher the efficiency, with a significant factor being the efficiency class.

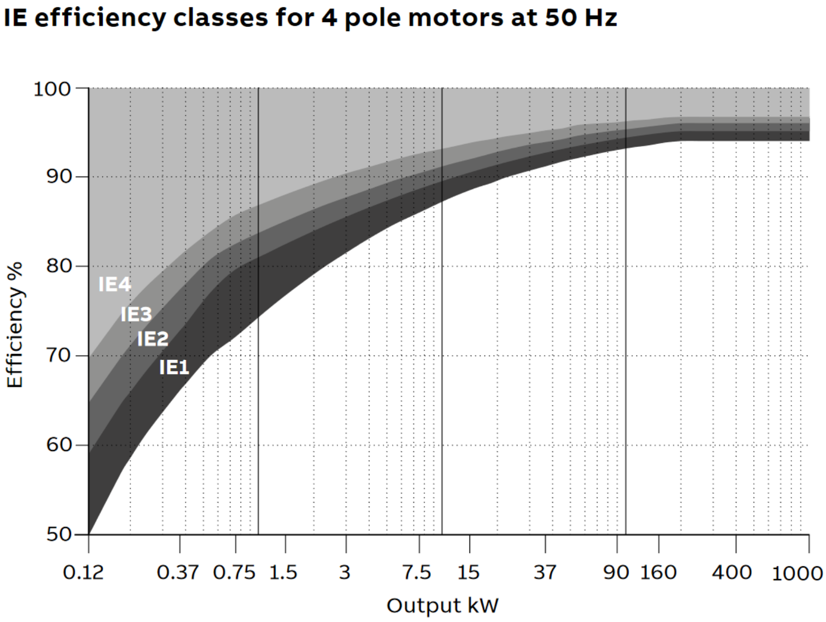


Figure 3. efficiency vs output[KW] per IE efficiency class

(3)

2.1.1 Motor control technologies

(4) **VFDs or ASDs controller -**Minimize Adverse Motor and Adjustable Speed Drive Interactions

A variable-frequency drive (VFD) also referred to as Adjustable Speed Drives (ASD), is a type of motor drive used in electro-mechanical drive systems to control AC motor speed and torque by varying motor input frequency and to control associated voltage or current variation.

VFDs are used in applications ranging from small appliances to large compressors.
Electronically adjustable speed drives (ASDs) are extremely efficient and valuable assets to motor systems. They allow precise process

control and provide energy savings within systems that do not need to operate continuously at full output.

Even with the relatively flat efficiency curve of larger motors (between 50% and 125% load), large efficiency gains can be obtained by adapting motor speed and torque to the required load.
The largest benefit comes with pumps in closed loops because power consumption varies as a cubic power of their rotational speed.
In traditional equipment, the load adjustment is made by introducing artificial brakes, such as control valves, dampers, or throttles.

VSDs can be used in air-conditioning systems, where it allows the temperature and flow control of pumps and fans, the reduction of on/off cycles, and offer stabilization of indoor climate.
For air compressors and horizontal conveyors, that's to say in constant torque loads, an adjustable speed control also has efficiency benefits by functioning the system with modulation, more stably therefore than with on/off cycles.
Traditional speed and torque control use either two-speed or multi-speed motors, with several motors working in parallel or with changing gears.
A s*tar-triangle* starter *switch* for engines with brakes can also be used.

Formerly to alter speed continuously DC motors were used.
For a *DC motor* to rotate *continuously*, *it* must-have *brushes and* commutators in its structure. However, due to the increased *wear,* they are not used much today.

“A variable frequency drive (VFD) is a type of motor controller that drives an electric motor by varying the frequency and voltage of its power supply. The VFD also can control ramp-up and ramp-down of the motor during start or stop, respectively.

Even though the drive controls the frequency and voltage of power supplied to the motor, we often refer to this as speed control, since the result is an adjustment of motor speed.

There are many reasons why we may want to adjust this motor speed. For example, to:

- Save energy and improve system efficiency
- Convert power in hybridization applications
- Match the speed of the drive to the process requirements
- Match the torque or power of a drive to the process requirements Improve the working environment Lower noise levels, for example from fans and pumps
- Reduce mechanical stress on machines to extend their lifetime Shave peak consumption to avoid peak-demand prices and reduce the motor size required”

(5)

A variable-frequency drive (VFD) is a device used in a drive system, consisting of the following three main sub-systems: AC motor, main drive controller assembly, and drive/operator interface.

The VFD controller is a solid-state power electronics conversion system consisting of three distinct sub-systems: a rectifier bridge converter, a direct current (DC) link, and an inverter.

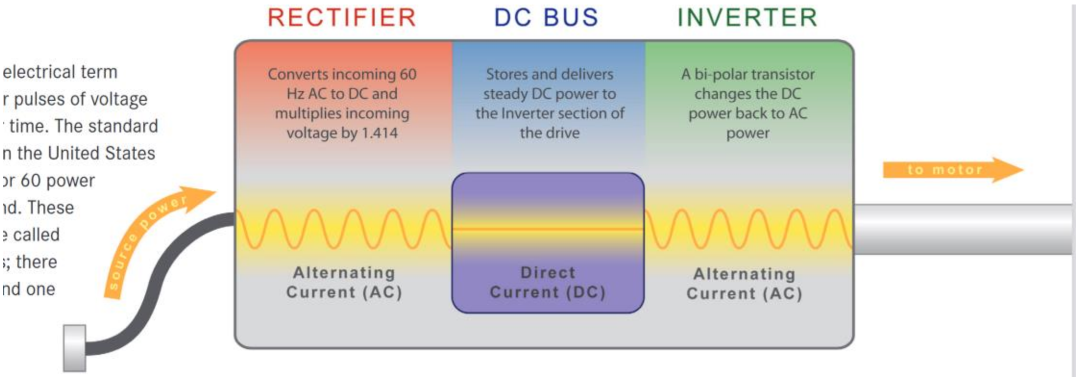


Figure 4: variable-frequency drive (VFD) structure
(6)
The most common kind today is the pulse-width-modulated (PWM) VFD with a fast-rise-time insulated gate bipolar transistor (IGBT) to reduce switching losses and noise levels.
However, these devices can produce voltage spikes or overshoots, that can stress motor windings and bearings. But these problems can be removed through proper design and equipment selection.
They rectify the 60-hertz (Hz) fixed-voltage alternating current (AC) to direct current (DC) and use an inverter to simulate an adjustable frequency and variable voltage AC output.
Transistors, or electronic "switches," create the AC voltage output but have very high losses when they create wave shapes other than square waves.

Most of the new motor technologies operate with variable speeds: they don't have a fixed speed, but
electronically adapt the speed, offering several advantages concerning energy efficiency:

- They root out to eliminate the main origin of partial-load losses, such as mechanical resistance
    - elements (throttles, dampers, bypasses).
- Adjustable speed can be used for direct drive, removing needless gears, transmissions, and clutches and cutting back losses.

2.1.2 Losses in VFDs

(7) VFDs consume energy for motor control, network connection, and input/output logic controllers, and lose energy, mostly in the output switches.

Actual VFDs for low voltage, that's to say less than 1 000 V, apply an integrated gate bipolar transistor (IGBT) three-terminal power semiconductor device, which combines high efficiency and fast switching with frequencies between 1 kHz and 20 kHz.

Total losses of the common inverters are divided into three main parts, conductive and joule losses, switching losses, and additional losses.

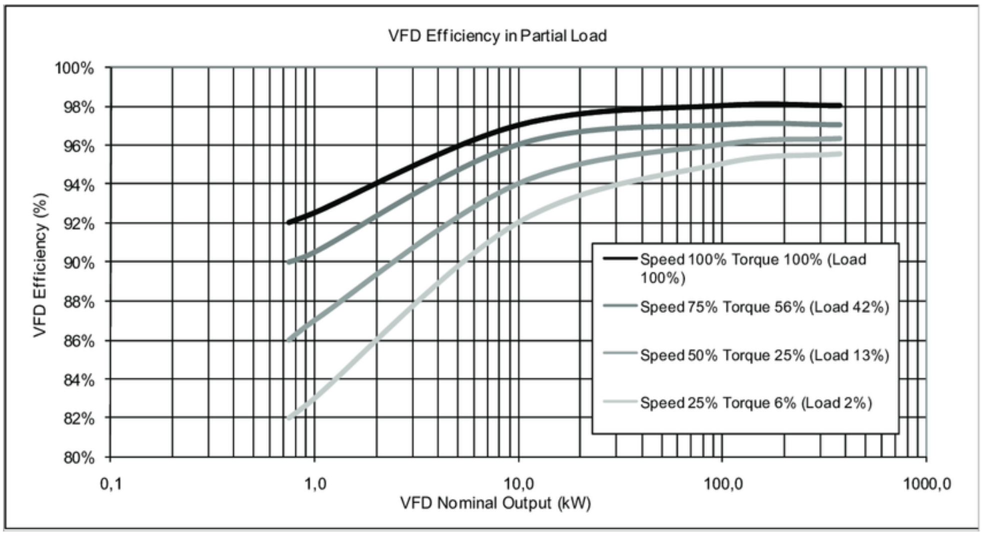
The losses of these inverters are quite low and their efficiency in partial load is better than cage-induction motors.

A VFD is considered a non-linear load because it only draws current from the power line, therefore it causes harmonic distortion - a measure of the amount of deviation from a pure sinusoidal waveform that can be caused by a non-linear load.

The primary collected factors on total losses are the switching frequency and the output current, linked to output power and load.

An interesting new approach to testing VFD is the Eu Directive on ecodesign of energy-using products (2005/32/EC) which aims at reducing the environmental impact of energy-related products (ERPs), including energy consumption throughout their entire life cycle. The Directive establishes a framework for setting Ecodesign requirements (such as energy efficiency requirements) for all ERPs in the residential, tertiary, and industrial sectors.

If a VFD is used below (Figure 5) at 50% speed or 50% torque, it will get heavy additional losses. In a typical application for rotodynamic pumps, a reduction of speed (e.g. down to 25%) will invariably reach very small torques (only 6.3%) and thus result in a very low load (1.6%) with severe losses inefficiency: down to <50%. Thus, the correct sizing of square torque machines is still critical to avoid many operation hours with speeds <50%. Rotodynamic pumps are not the ones used in conventional hydraulics(positive displacement ones.)



1 Figure 5:Variable-frequency drive efficiency at full and partial load (8)

To establish the energy savings that are possible when a variable frequency drive is applied to a variable torque load, you must determine the load duty cycle - that's to say the percentage of time that the pump works at each system operating point. We need to know the efficiency of the VFD and the drive motor when the motor is operating partially loaded and at a reduced speed, to fulfill variable flow regulations.
The efficiency of pumps ranges with size (flow, diameter, power) and kind of fluid. A major impact is the operation point versus the optimal point (**Error! Reference source not found.**6). (9)

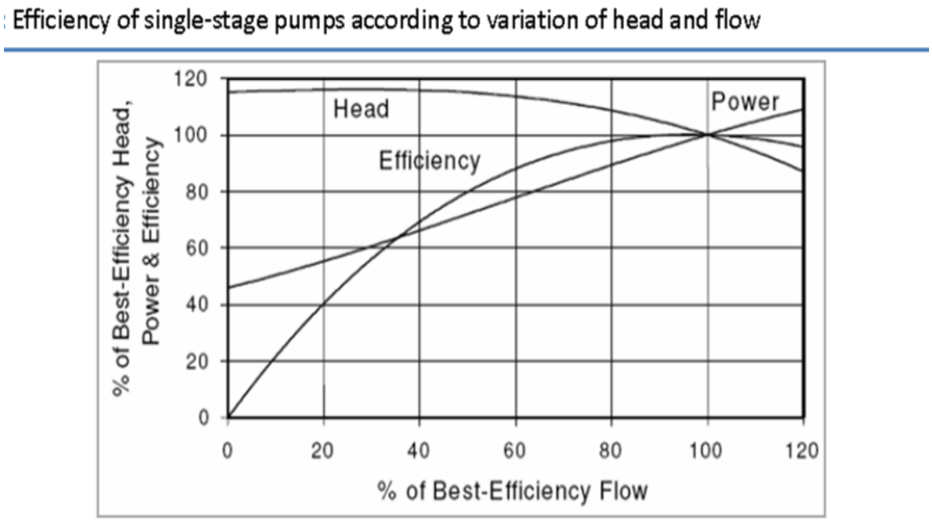


Figure 6: (Efficiency of single-stage pumps vs head and power)
(9)

A study has shown that by applying all available efficiency measures, savings of about 80-90% can be achieved for heating-system pumps and about 40-75% for industrial pumps.

Best enhancement can be found for broader duct pump, permanent magnet motor with a variable frequency drive, and however, manufacturers properly brought down.

Since the sizes of pipes and valves in existing systems cannot be changed, it is very important to pick the ideal size at the time of purchase.

In existing systems, some pipes can be undersized at the time of replacement or added to the existing item.

This factor has to be considered, but this will impact energy costs.

For industry cooling, heat exchangers can source a speedy fall of pressure. For every reassessment or intervention, it's useful to consider the substitution with a low-pressure kind of heat exchanger or adding a minor pump in high-pressure manufact.

Both in domestic heating and industrial pumps, we can gain substantial energy sparing through a correct calibration of the pumps and new larger piping.

A performance specification states requirements in terms of the required results with criteria for verifying compliance, and defining the functional requirements for the item, but without pointing out the methods for reaching the required results.

For electric motors, the most significant standards are IEEE 112 (2004), concerning the more applicable and acceptable tests to determine the performance and characteristics of polyphase induction motors and generators.

- phase, induction motors the standard IEEE 114 (2010) is now the Inactive Standard (2021-03-25) for single-phase induction motors

For Pumps, European ISO 9906:2012 specifies hydraulic performance tests for rotodynamic pumps i.e. centrifugal, mixed flow, and axial pumps.

It is intended to be used for pump acceptance testing at pump test facilities, such as manufacturers' pump test facilities or laboratories.

It specifies three levels of acceptance:

- grades 1B, 1E, and 1U with tighter tolerance.
- grades 2B and 2U with broader tolerance.
- grade 3B with even broader tolerance.

It applies either to a pump itself without any fittings or to a combination of a pump associated with all or part of its upstream and/or downstream fittings.

This standard is for rotodynamic pumps.

For positive displacement pumps, we are using ISO 4409:2019. (ISO - ISO 4409:2019 - Hydraulic fluid power — Positive-displacement pumps, motors, and integral transmissions — Methods of testing and presenting basic steady-state performance, no date a).

Spreading knowledge on the correct choice of the manufact is important because pumps are intrinsically efficient, but often they are conducted in an ineffectual manner.
Several tools have been introduced to enhance the better use of pumps. In the United States, has been established Pump Systems Matter (PSM), an educational foundation whose aim is to promote pumping systems' energy efficiency.

(11)

PSM brings together a wide range of stakeholders to advocate for the benefits gained by optimizing pumping systems.

The **Pumping System Assessment Tool** (PSAT)(12) is a free online software recently reviewed in the MEASURE tool suite, to support industrial users' assessment and to improve the efficiency of pumping system operations. The system uses achievable pump performance data settled by Hydraulic Institute standards to calculate potential energy and allow the determination of electrical energy and money savings.

In **Europe**, Performance curves for Single Stage Centrifugal Pumps published in manufacturers' catalogs have been used to produce six plots, published by the Manufacturers Association Europump/SAVE,(Guides and Guidelines | Europump European Association of Pump Manufacturers - download Guide Click Europump energy Directive EUROPUMP industry requirements application guide guideline Pumps amending Motors field, no date) to help decide which type of pump is suitable, considering the flow and head at which the maximum efficiency is needed.

Old Continent has even a Programme named ENERGYPlus PUMPS: Technology procurement for very energy-efficient circulation pumps. This project fosters market adoption of higher-efficiency circulators endorsed by the Intelligent Energy Europe Programme (IEE), which provides training on new techniques that can lead to significant energy savings purchasing solutions.

We have to mention here that servomotors nowadays have higher efficiency and are more and more used.
Permanent magnet motors, also called servo motors allow for easy control of motion and higher accelerations and maximum speed than induction motors. They also have higher torque density(around 50 percent higher) than an equivalent-sized induction motor, so they are more compact. A useful summary of the advantages and disadvantages of permanent magnet synchronous motors is ((14):
Advantages
■ Excellent torque-speed curve and dynamic
response
■ High efficiency and reliability, low maintenance requirements
■ Variable speed capability
■ Higher efficiency at partial loads
■ Longer lifetime
■ Low acoustical noise
■ High-speed capability
■ High torque-to-volume ratio (high power density).
Disadvantages
■ High cost (volatile because of rare earth magnet prices)
■ Need for a controller

In figure 7 we can see that the permanent magnet motor easily overcomes any induction motor in terms of efficiency:

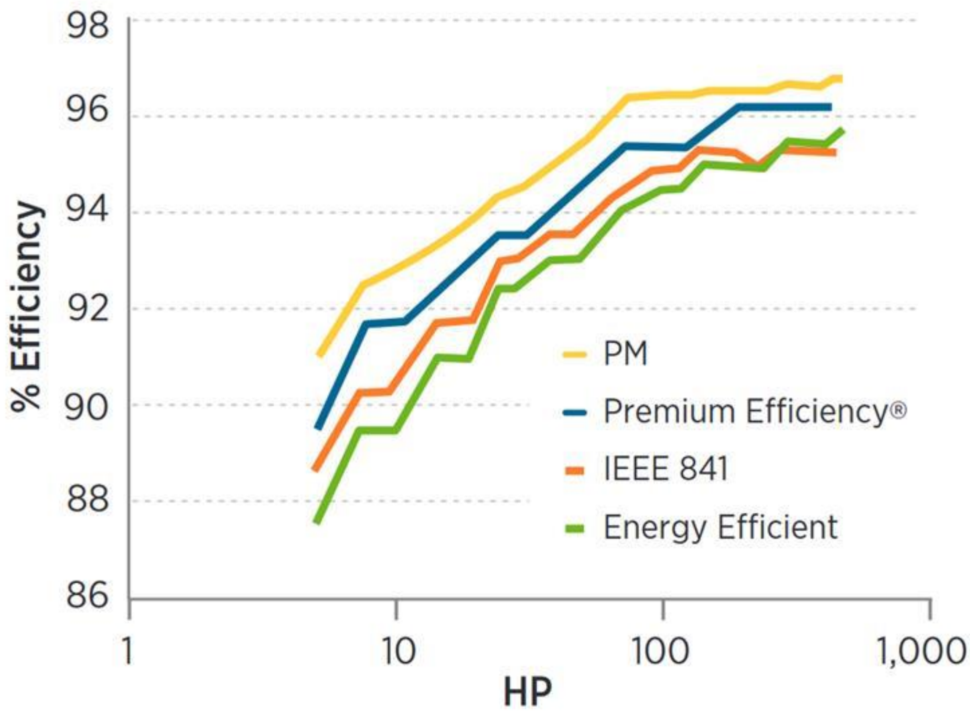


Figure 7. Full load efficiency values for PM versus NEAM premium efficiency motor models.

(14)

2.2 Evolution of PUMP EFFICIENCY in time

In this section, we want to consider the *best technologies available* at the moment regarding positive displacement pump efficiency.

In an Innas study where it is possible to compare efficiencies of the major contemporary models of pumps, we can understand what current best technologies are in this field.

We can observe in figure 8 a Comparison between the measured power loss for all of the machines with p_2 = 200 bar.

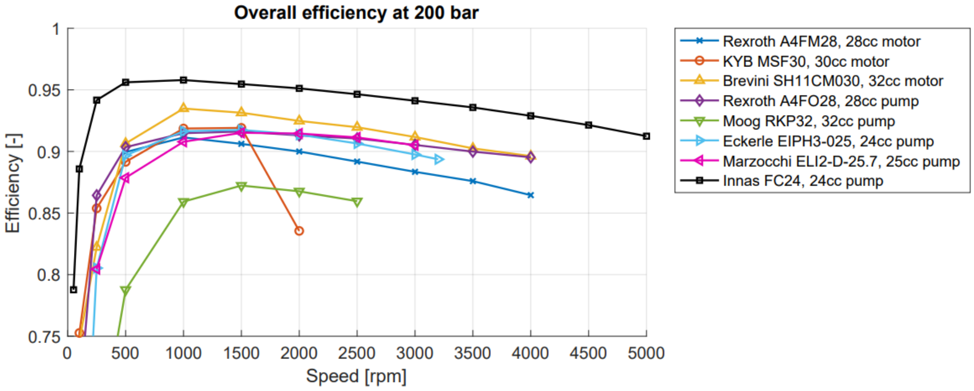


Figure 8. Overall efficiencies of pumps and motors at 200 bar

(15)

As we can see, the Innas FC24 is the clear winner across the spectrum. Its technology is the Floating Cup. It has a very good efficiency in the large operating area, speed, and pressure and it can be connected to a very good electric motor. The combined efficiency would be great, much better than the traditional electric motor and pump combinations. We will describe this design technology in a subsection.

To better describe figure 8, we have to consider the type of each pump and motor considered:

- Rexroth A4FO28 Slipper type, axial piston pump
- Moog RKP32 Slipper type, radial piston pump
- Eckerle EIPH3-025 Internal gear pump
- Marzocchi ELI2-D-25.7 External gear pump
- Innas BV FC24 Floating cup, axial piston pump

Another very promising design is the Digital Displacement Pump from Danfoss. It offers very high efficiency in a broad range, as we can see from the Digital Displacement Pump Gen 1 DDP096 and DPC12 Technical sheet in figure 9.

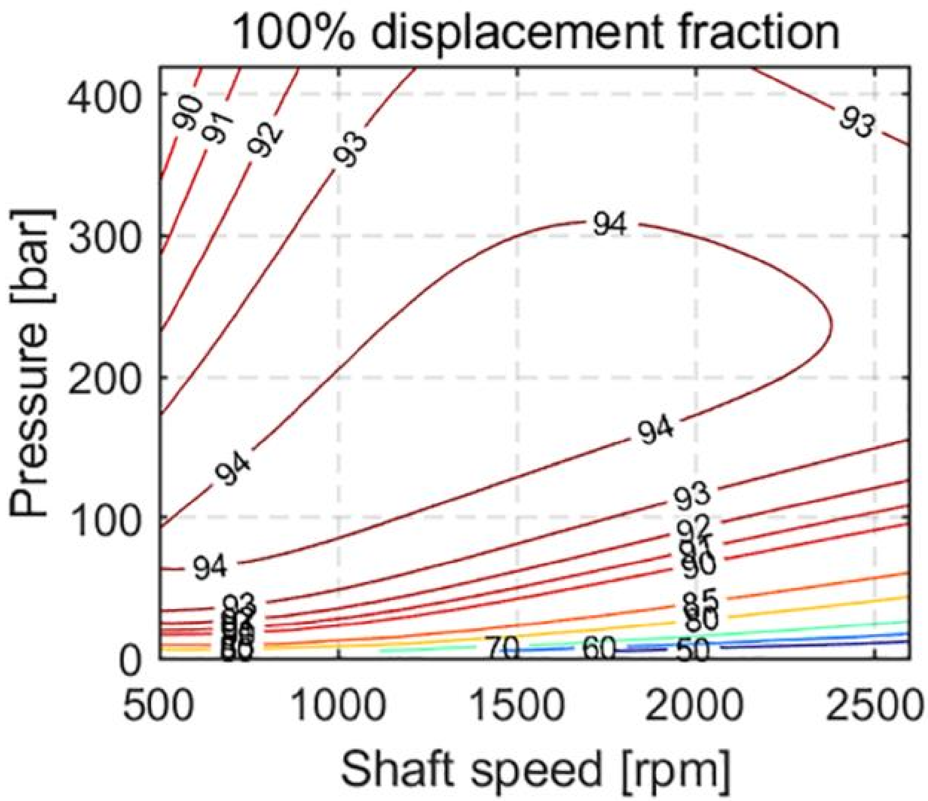


Figure 9. The overall efficiency of Danfoss digital displacement pump (reference)
(16)

2.2.1 Floating Cup design pumps

They are mainly designed by Innas at the moment. Their producer is Bucher AX. We'll consider figures from INNAS website for this reason.(17)
The easiest way to understand their working mechanism is to look at figure 10.

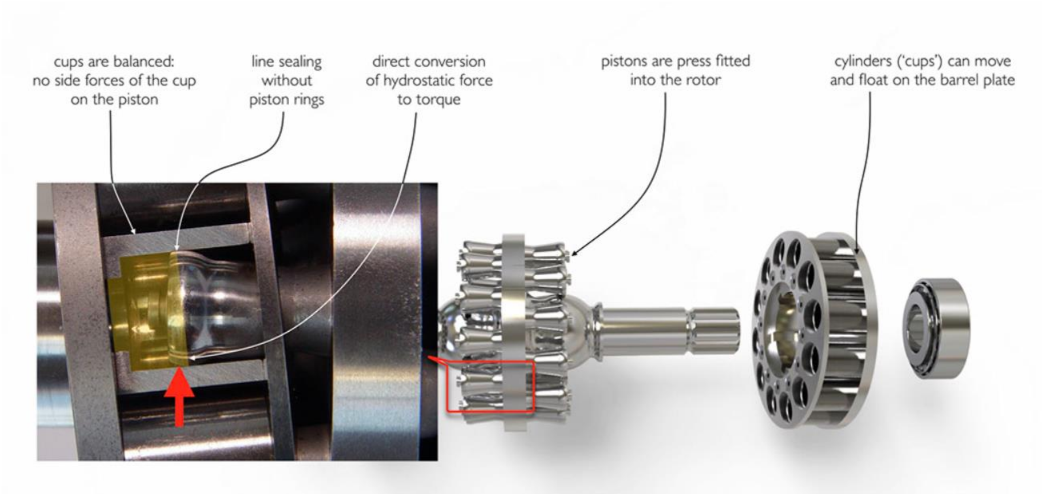


Figure 10. Structure of floating cup pump (17)

This generates a static force, expressed in figure 11

Figure 11. Structure of floating cup piston
(17)

The necessary shaft torque is created by the radial components of these piston forces. The oil column creates a hydrostatic force on the piston, having the same tilted position as the cup. The conversion of hydraulic power to mechanical output power is direct: there are no moving interfaces or linkages. As a result, there are also no principal losses.
Moreover, the cup is fully balanced, as we can see in figure **12**.

Figure 12. Structure of the cup

(17)

The ball-shaped piston crown has the same outer diameter as the inner cylinder of the cup. The resulting sealing line is always and by definition perpendicular to the main axis of the cup, irrespective of the tilt position of the piston. Consequently, the radial pressure load on the cup is equal in all directions. The cup is therefore completely balanced and does not create a hydrostatic load on the piston, which minimizes friction and wear.

Also, the global geometry is balanced, as highlighted in figure 13.

Figure 13. Pistons and the shaft (reference)
(17)

The Floating Cup design allows for a much higher number of pistons in the same envelope. This fundamentally improves performance. The mirrored design allows for a balance of the high large hydrostatic forces, resulting in a low bearing load.
We can say that the advantages of this technology are that it is:

- Highly efficient, as we have seen previously.
- Smooth and quiet, because of the symmetrical design and a high number of pistons.
- Versatile, as it can be applied to pumps and motors, at constant and variable displacement and in hydraulic transformers.
- Ready for cost-effective production: Automotive bucket tappets are realized similarly, so it is possible to use their existing production process knowledge.
  (17)
  Overall, I think we will see this technology applied a lot in the future.

2.2.2 Digital Displacement pumps

As we can learn from (18) the new technology surrounding the Digital Displacement® pumps (DDP) and motors (DDPM) created and patented in 1994 by Stephen Salter, co-founder of Artemis Intelligent Power, appears to be generating a lot of interest. By replacing mechanical gearboxes with hydraulic transmissions, Digital Displacement® has shown significant energy savings by harvesting energy from wind and waves. When used on an excavator, the technology showed up to a 20% reduction in fuel consumption and, perhaps more importantly, a nearly 30% improvement in productivity. Because the pump can be started with very little hydraulic stress, the DDP is expected to eliminate the requirement for a variable frequency drive (VFD) or a soft starter. For many, digital technology might result in significant cost savings as well as increased productivity.

If the unit can be employed as a pump or a motor Dual Dynamic Power Management (DDPM), each piston has two solenoids. Each pump can be on or off in 30 milliseconds, and this short e time can restrict the fluid amount that gets in the power stream.
In effect, it is a multistage transmission, in which we can adjust every output flow. We can arrange several numbers, positions, and sizing of the pistons.
A 68 pistons device was built to get the energy coming from a wind turbine.
A most used unit, as figure 14 shows, employs 12 pistons, sorted in 3 packs of 4.

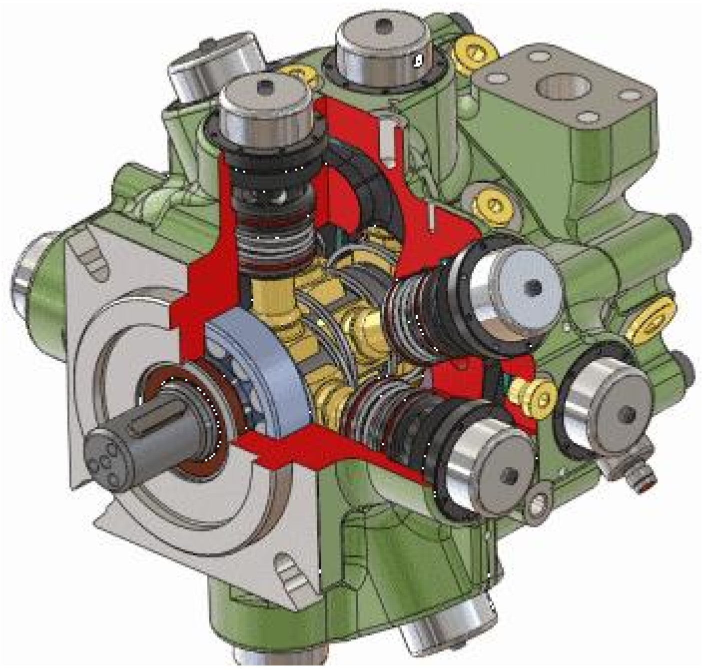


Figure 14. Digital displacement pump structure
(18)

In a six-piston pump (Figure 15) as the cam revolves, the pistons are sequentially pulled in and pushed out. A check valve isolates the high and low-pressure zones of the pump; a pull solenoid valve locks and unlocks the track from the low-pressure zone.
Each reciprocating pump can be considered a specific source.

If the none solenoid valve is activated (Fig. 2) the low-pressure source would keep on ready for the piston during the whole cam spin.
The piston would merely move fluid out and back into the low-pressure zone, requiring very little energy.
If an electrical power failure happens, the DDP would face up to a no-flow, low-pressure failing condition.

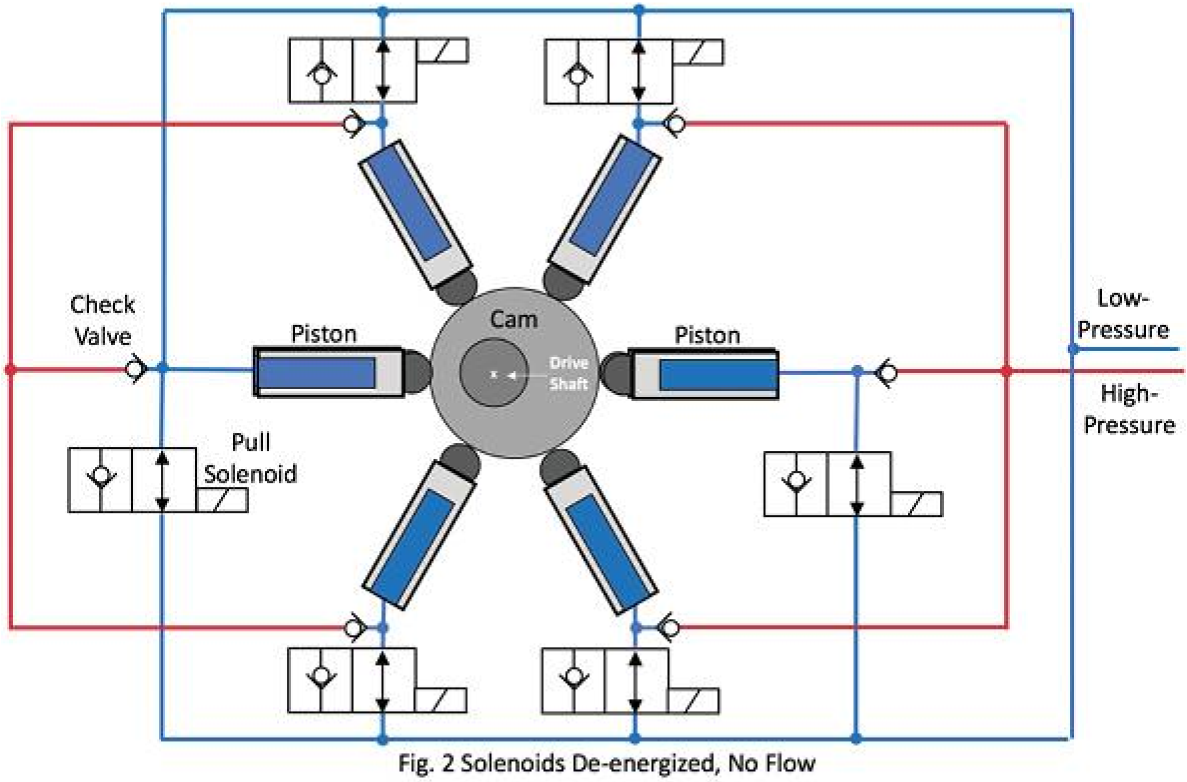


Figure 14. Digital displacement pump operating principle (reference) (18)

When a solenoid valve is activated, the piston would get from the low-pressure zone and then drain into the high-pressure core.
For example: for a shift of 10 cc for every piston, the whole displacement will be 60 cc for the 6 pistons (3.7 in^3).
At 1800 RPM, the flow potential would be 108 lpm (28.5 gpm).

Each piston is activated by its control solenoid, therefore the potential flow will be enhanced by 18 lpm (4.8 gpm).
If part of a work cycle requires only 54 lpm (14.3 gpm), half of the 6 solenoids can be turned on to deliver the needed output.

We can use different scheduled activation timing of the solenoids, aiming to have only a share of the piston displacement sent to the high-pressure core.

Coming back to a 6-piston pump, if the flow demand is 63 lpm (16.6 gpm) the displacement of 3.5 pistons would be enough. This could be achieved by activating 3 solenoids to put 3 pistons on, and 1 solenoid to close the track to the low-pressure side as the piston reached half stroke; half of the piston translation would introduce the pressure flow.

Another technique would be to activate all the solenoids for 1,044 revolutions and then drop them off for 756 revolutions.

To avoid power flutters, we can get the same stream scheduling the 6 solenoids to induce the pistons to eject only 5.8 cc per cycle, switching them on at 58% stroke.

These devices allow us to obtain not 6 incremental flows, but lots of increments.

The DDP can simulate an infinitely variable displacement pump, and without needing to get a constant core pressure. The single pistons can be at inlet pressure, despite the burden at the pump outlet.

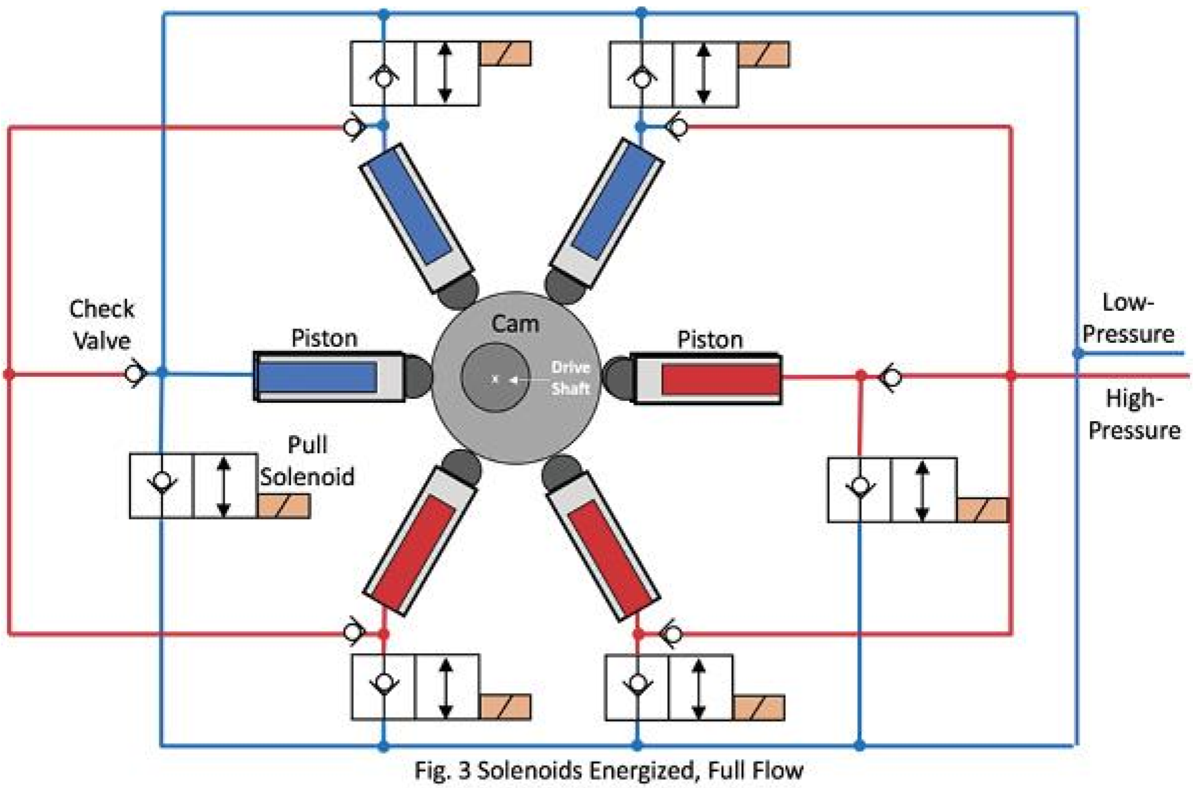


Figure 15. Usage of solenoid valves in digital displacement pump

(18)

Another property is that for each piston, the output can be segregated to allow flow to several circuits.
The reasoning behind Digital Displacement is linear to understand, but the device isn't very simple to use, at least not yet.
This is due to the lability of the moving parts, which don't have the high tolerances as usual piston pumps, and they also require a higher degree of fluid purity.

Furthermore, having been designed to be computer-directed, they require advanced skill in fluid power command, to achieve the complete gain the method can offer. Less experienced technicians could prefer to work with pressure-compensated or load-induced pressure-sensing pumps.

It wouldn't be correct to compare the price of a traditional unit pump with a Digital Displacement one. We would have to consider the whole system including prime mover and heat exchanger, electronic controls, and programming. We would have to calculate the power savings and the higher productivity, too.
The chance of using individual cylinders calling only if needed into action is really valuable because means reduced losses, and longer work lives these days.
Danfoss promotes combining their Editron and Digital Displacement products in building up electro-hydraulic powertrains. The Editron system includes electric machines, electric converters, and energy storage.
This permits high integration of their systems, revolutionizing the idea of design.
(19) Electric converters … "can act as a motor inverter, active front end, create a microgrid or as a DC/DC-converter depending on the options selected."

They are generally moving towards system integration also in their EDITRON program(figure 16)

Figure 16. Usage of solenoid valves in digital displacement pump
(20)

With electric and hybrid powertrain solutions for heavy-duty and commercial vehicles and machinery on land and sea, Danfoss Editron aims to redefine the way the world moves.

The characteristics of this program are:

- The Editron system is available in both fully electric and hybrid versions.
- Fuel and energy usage are reduced.
- Emissions of carbon dioxide and tiny particles are reduced.
- Because of the lightweight and small hardware, there is more design freedom.
- Simple integration into a variety of machinery.

2.3 TESTING USUAL PRACTICES

2.3.1 Testing PUMPS

(21)

To enhance the energy economics of hydraulic systems, power losses in two primary entities, energy converting components and energy regulating and conveying components, should be addressed. To achieve the former, components must be operated within their most energy-efficient working range. Several operational parameters influence the energy conversion efficiency of a pump, which is the fundamental energy converter in a hydraulic system. Only pressure and rotational speed are generally addressed, but with variable displacement pumps, fluid temperature and derived capacity have a significant impact on efficiency. If these considerations are ignored, the pump may be operated outside of its most efficient operating range, resulting in significant energy losses. However, operating the pump in its optimal range necessitates a thorough understanding of its performance parameters, which are rarely made public by pump makers. The performance measurement findings of a variable displacement axial piston pump are presented in the form of efficiencies as a function of pressure, rotating speed, derived capacity, and inlet fluid temperature in this research. The findings reveal that all of these parameters have a substantial influence on the energy conversion efficiency of a hydraulic pump and should be considered while operating one.

One testing configuration we can use is presented in figure 17:

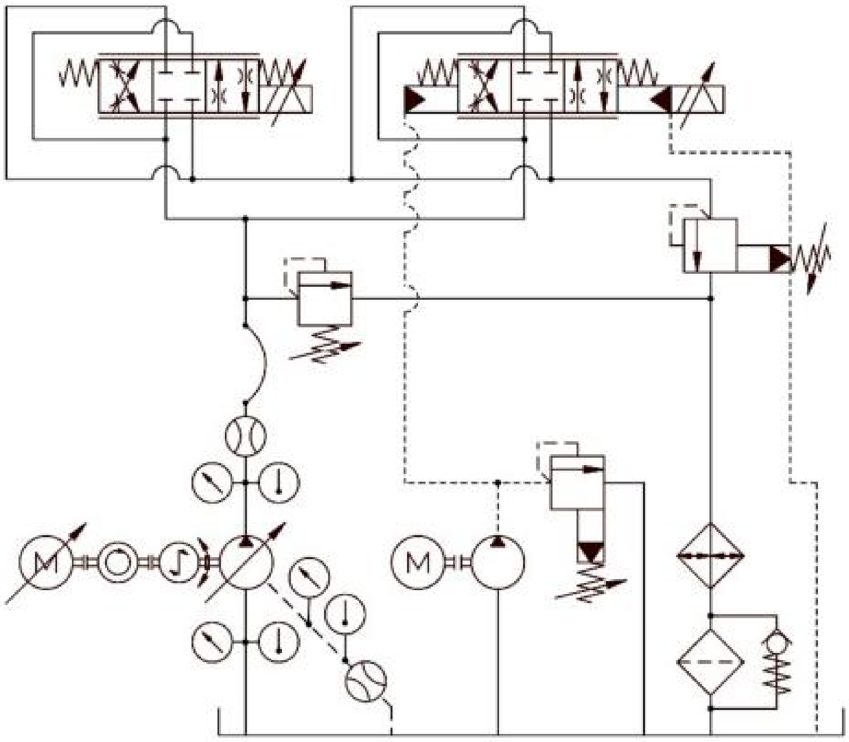


Figure 17: Measurement system example
(2)

The main problem in this system is the pump load is produced by throttling the flow, so we generate heat with all the pump's power. In other words, all the pump's power becomes heat, so we would need to cool down all the pump's power.
Instead, using a motor, we have to cool down only the power related to the pump's and hydraulic motor's inefficiencies(around 10 to 40 percent of the testing power, depending on the efficiency of components).
To detect the performance of a pump it is significant to evaluate the volumetric and hydromechanical efficiency. During the test, the inlet pressure of the pump must be held unchanged, according to standard More precise information can be found in ISO 4409:2019

Hydraulic fluid power — Positive-displacement pumps, motors, and integral transmissions — Methods of testing and presenting basic steady-state performance, especially if you want to have *strict control of the pump's inlet pressure*. The pump inlet line should not exceed 0.25 bar. Usually, the pump inlet pressure at the inlet fitting should be under 0.034 bar.

(22)The parameters that must be recorded are:

1. Input torque.
2. Outlet flow of the pump.
3. Fluid temperature.
4. Drainage flow

To test the pump, there are two points to highlight:

- Maintaining constant speed and varying the outlet pressure of the pump.
- Maintaining constant the pump outlet pressure and varying the speed of the pump

The pump efficiencies can be obtained by the equations (1) and (2):

$$\eta_{hmp} = \frac{\varepsilon_p \; D_p \; \Delta_p}{2\pi \; M_{in}} \tag{1}$$

$$\eta_{volp} = \frac{q_{ep}}{\varepsilon_p \; D_p \; n_p} \tag{2}$$

Where:

η_{hmp} = hydromechanical efficiency of the pump

ε_p = displacement setting of the pump

D_p = displacement of the pump

Δ_p = pressure difference of the pump

M_{in} =input torque to pump

η_{volp} = volumetric efficiency of the pump

q_{ep} = effective flow of the pump

n_p = speed of the pump

Test circuit for pumps

1. Driver
2. Pressure transducer
3. Temperature sensor
4. Temperature sensor
5. Pressure transducer
6. Torque meter
7. Speed meter

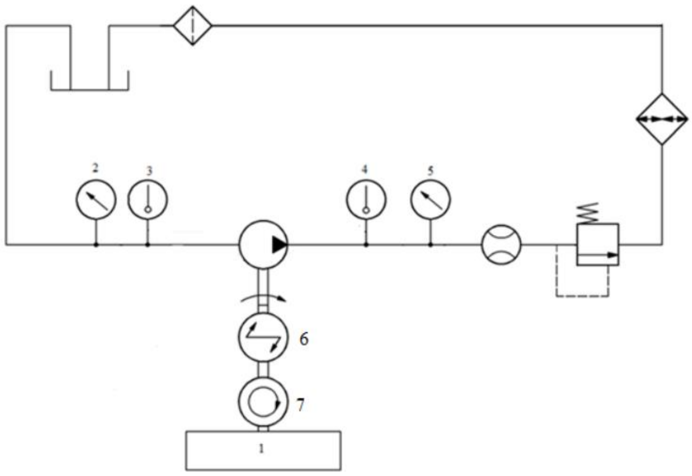


Figure 18: test circuit configuration for pumps

(2)

In this circuit the position of sensors is the one that allows the conditions:

Being *d* the diameter of the hose or pipe,

- Sensors for temperature are located at a distance of 2d-4d, upstream and downstream, from the component to be evaluated
- Pressure transducers are located at a minimum distance of 5d upstream and 10d downstream

We can see that A level standard requires strict tolerances:

Measurement accuracy class (see Annex A)	**A**	**B**	**C**
Temperature tolerance (°C)	±1,0	±2,0	±4,0

Table 1:test fluid temperature tolerances

(10)

Table 4 — Permissible variation of mean indicated values of controlled parameters

Parameter	Permissible variation for classes of measurement accuracy[a] (see Annex A)		
	A	B	C
Rotational frequency, %	±0,5	±1,0	±2,0
Torque, %	±0,5	±1,0	±2,0
Volume flow rate, %	±0,5	±1,5	±2,5
Pressure, Pa ($p_e < 2 \times 10^5$ Pa)[b]	$\pm 1 \times 10^3$	$\pm 3 \times 10^3$	$\pm 5 \times 10^3$
Pressure, % ($p_e \geqq 2 \times 10^5$ Pa)	±0,5	±1,5	±2,5

[a] The permissible variations listed in this table concern deviation of the indicated instrument reading and do not refer to limits of error of the instrument reading; see Annex A. These variations are used as an indicator of steady state and are also used where graphical results are presented for a parameter of fixed value. The actual indicated value should be used in any subsequent calculation of power, efficiency or power losses.

[b] 1 Pa = 1 N/m^2.

Table 2: Permissible variation of the mean indicated values of controlled parameters

(10)

Level standards are quite difficult to achieve, especially in terms of temperature, requiring an accurate temperature sensor and control system.

2.3.2 TESTING MOTORS

(2) To test a motor, the outlet pressure must be held constant.
The parameters that must be recorded are:

1. Inlet flow to the motor.
2. Output torque.
3. Fluid temperature.
4. Drainage flow

The motor must be tested at different speed ranges, changing the flow, and at variable input pressures.

Test circuit for motors(18) :
1. Regulated flow supply
2.Pressure transducer
3 Temperature sensor
4. Load
5. Temperature sensor
6. Pressure transducers

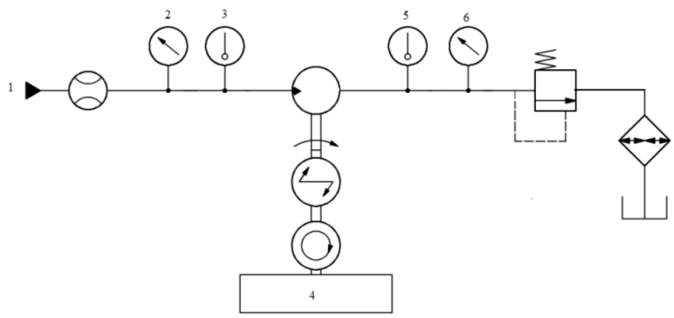


Figure 18: Test circuit for motors
(2)

Some rules must be taken into account during the test:

- The temperature of the fluid must be kept as indicated by the producer, with a tolerance of ± 4°C
- Other parameters such as speed and outlet pressure also must be in the range stated by the producer.
- If the pumps, or the motor, is variable displacement, the displacement will be according to the specification.

2.4 Heat exchanger choice

Those are the main reasons to have a heat exchanger in our system:

• The fluid must be filtered and in a closed hydraulic system (traditional hydrostatic drive system) the fluid must be drained. Filtration and temperature control are performed in a subsystem connected to the system reservoir. These measurements can be performed in low-pressure circuits, allowing lower-cost components to be used.
• The use of a closed system in pump measurements, where the hydraulic and electric motors (generators) generate the load allows for a lower heat load and a reduced need for cooling (power of the exchanger).
. • ISO standards require precise fluid temperature control. Therefore, the thermal control system must be advanced and preferably a closed-loop control system e.g. PID.
• The heat control system can be for example in the return line (after the relief valve), in the inlet or outlet pipe of the intake pump, or it can be a… Separate offline thermostatic circuit.
• Heat exchanger is hydraulic fluid type (mineral oil) - water type. Water will be used due to the efficiency of the media and the quiet operation of the device.

As regards the filter, we have many design tactics possible: (23):

- Pressure filtration: Placing filtering media in the pressure line gives the best protection for downstream components. Because of the pressure available to drive fluid through the medium, filtration rates of 2 microns or less are conceivable. However, high flow velocities, pressure, and flow transients, which can unsettle trapped particles, can diminish filter efficiency.

- Return filtration: If the reservoir and the fluid it contains start clean, and all air entering the reservoir and returning fluid is effectively filtered, then fluid cleanliness will be maintained. Another benefit of employing the return line as a filter site is that there is adequate pressure to drive fluid through tiny media (usually 10 microns) without complicating filter or housing construction.

- Off-line filtration allows for continuous multi-pass filtration with a regulated flow velocity and pressure drop, resulting in great filtering efficiency. Filtration rates of 2 microns or less are achievable, and heat exchangers and polymeric (water-absorbent) filters can be incorporated into the circuit for entire fluid conditioning. Off-line filtration's major disadvantage is its high initial cost, which can typically be justified on a life-of-machine basis.

- Suction filtration: The pump intake is an attractive site for filtering material from a filtration standpoint. The lack of both high fluid velocity, which can upset trapped particles, and high-pressure drop across the element, which can induce particle migration through the medium, improves filter performance. The limitation the element generates in the intake line, as well as the detrimental impact this might have on pump life, outweigh these benefits.

In our project, we will use offline cooling and filtering through the use of an offline system to enable efficient cooling and filtering. Cooling can be regulated both by flow rate and coolant fluid’s temperature control. This system can be used to closed-loop control the system pump’s input flow temperature by using a feedback signal from the pump inlet.

2.5 .Plate heat exchanger

The plate heat exchanger is a suitable device to be used in thermal control and its operation principle and structure are discussed in the following section.

A heat exchanger is a device that transfers heat from one medium to another.

A plate heat exchanger is a type of heat exchanger that uses metal plates to shift heat between two fluids, offering a conventional heat exchanger the advantage that the fluids are exposed to a much larger surface area, spreading the fluids over the plates.
Other kinds of exchangers are Shell and Tube, or Spiral heat Exchangers, but they are much less used because Plate types are well-functioning, solid, and simple-to-use and for maintenance.
(24)
The plate heat exchanger (PHE) is a specialized design well suited to transferring heat between medium- and low-pressure fluids. We can have also high-pressure solutions (Vahterus). (Therefore, they need inlet and outlet points, because the flowing fluids (liquid or gas) must go in and exit the exchanger.

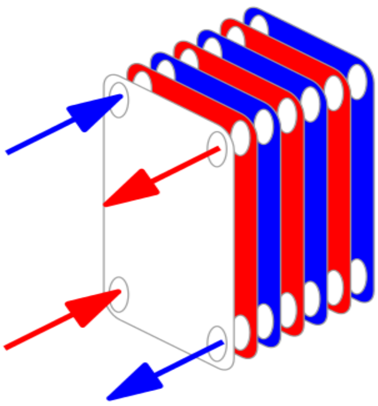


Figure 19: Plate heat exchanger's working principle
(24)

The plates are spaced by rubber sealing gaskets, which are wedged around the edge of the plates.

Gaskets and plates are used to separate the flowing medium and prevent them from mixing and are adherent to one side of each plate only.

Gaskets are adherent to one side of each plate only.
They may also be:

- Soldered, like in the case of (25)
- Welded(26)

The plates hang upon a carrying bar and are pressed together using clamping bolts.

The plates are compressed together in a rigid frame to form an arrangement of parallel flow channels with alternating hot and cold fluids. The arrangement of the gaskets allows flow in single channels. This enables the primary and secondary media in a counter-current flow.

The plates compressed one another (with minimal distance- called Clearance) to form a "plate stack".
They are thin and with a wide surface area and are made of materials that allow a high thermal conductivity, to raise the transfer rate.
The plate surface is corrugated (herringbone) to strengthen and avoid laminar flow and aims to create turbulence flow in the fluids, setting up an effective heat transfer coefficient.

There is a guide bar designed to allow the perfect alignment of plates during the opening and closing operation of the stack.
Of the two covers, one is movable (frame plate) and one is fixed (pressure plate).
At this latter cover are mounted the inlets and outlets.
More detailed anatomy is given in Figure 20.(27)

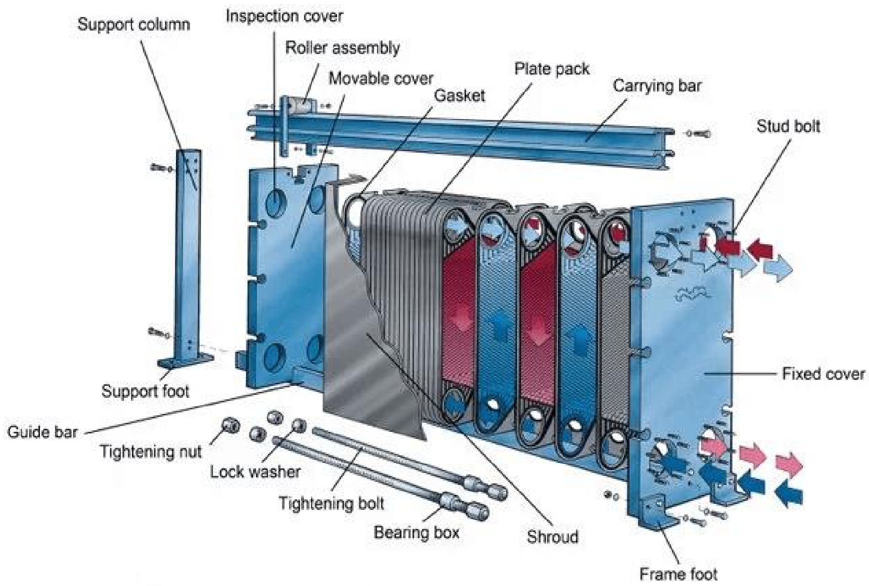


Figure 20: Plate heat exchanger's anatomy
(27)

2.6 Heat transfer – The theory

(27) The driving energy in a system tends to flow until equilibrium. When there is a temperature difference, there is a dissipation of heat. Heat transfer between mediums and fluids happens to obey some rules:

- Heat is always transferred from a hot medium to a cold medium. There must always be a temperature difference between the mediums.
- The heat lost from the hot medium is equal to the amount of heat gained by the cold medium.
- In a heat exchanger following the equalization principle, with a plate heat exchanger, heat cuts through the surface and separates the hot medium from the cold. In this way, heating and cooling fluids and gasses need minimal energy levels.

- Plates seal the channels and direct the mediums into alternate channels.

- The upper carrying bar supports the channel and pressure plate. They are then fixed in a position by a lower guiding bar on the support column.

- The hot medium comes in through the opposite inlet, and gaskets direct the hot fluid into the space between two plates and avoid the flow into the space between the next couple of plates. The process goes on, and each second set of plates is loaded with the hot medium.

Meanwhile, the cold fluid gets in through its inlet; this time the gaskets allow the cold fluid to stream into the space where there is no hot fluid.
At this point, the heat exchanger is replete with flows of hot and cold fluid; each fluid flows emerge from its point of outlet, and the process goes on.
For heat-sensitive or viscous media, cold fluid convenes with hot fluid. This minimizes the risk of the media overheating or freezing.

There are several ways to modify the cooling power of a plate heat exchanger.
For instance, we can act on the valves that allow the outlet flow, aiming to raise or reduce it. No need to disassemble anything. We cannot act on inlet valves to avoid overcooling zones.
If we need a reduction of cooling capacity we will ease the plate amount, while if we add plates, we obtain an increase in cooling capacity.

Most plate heat exchangers have a single-pass design, fluids pass each other only once. Multi-pass heat exchangers have fluids that pass each other more than once using U tubes and baffles.

The three categories of flow types are (Figure 21) :

- **parallel flow** - the two fluids enter the exchanger at the same end and travel in parallel to one another to the other side.
- **counterflow -** the fluids enter the exchanger from opposite ends
- **cross flow -** the fluids travel roughly perpendicular to one another through the exchanger.

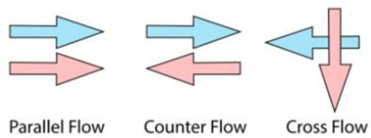


Figure 21. Parallel, counter, and cross-flow principles
(27)

2.6.1 Advantages of plate heat exchanger:
(28)

1. Heat transfer precision – improved temperature approach, true counter-current flow, very compact
2. Low cost - low capital investment, installation costs, limited maintenance, and operating costs. Save weight and space.
3. Maintenance and cleaning are easy.
4. Greatest reliability - less fouling, stress, wear, and corrosion.

Different materials (metals, alloys, Teflon, or other polymer-based materials) produce exchangers fitting for different applications.

5. Responsible - least energy consumption for most process effect, reduced cleaning.
6. Easy to expand capacity – adjustable plates on existing frames
7. No need for more space for decommissioning

2.6.2 Disadvantages of plate heat exchangers:
(28)

1. Easy leaking in extreme environments
2. Replacement of leaking gaskets in situ is almost impossible. They must be replaced by the manufacturer, with the waste of time and money
3. Small clearance, raises the risk of fouling
4. More expensive than other design
5. Limited pressure use, generally not more than 1.5 MPa.

Exceptions for special designs to this last point are :

(29)In special cases high-pressure plate heat exchangers have pressures as high as 15 MPa)

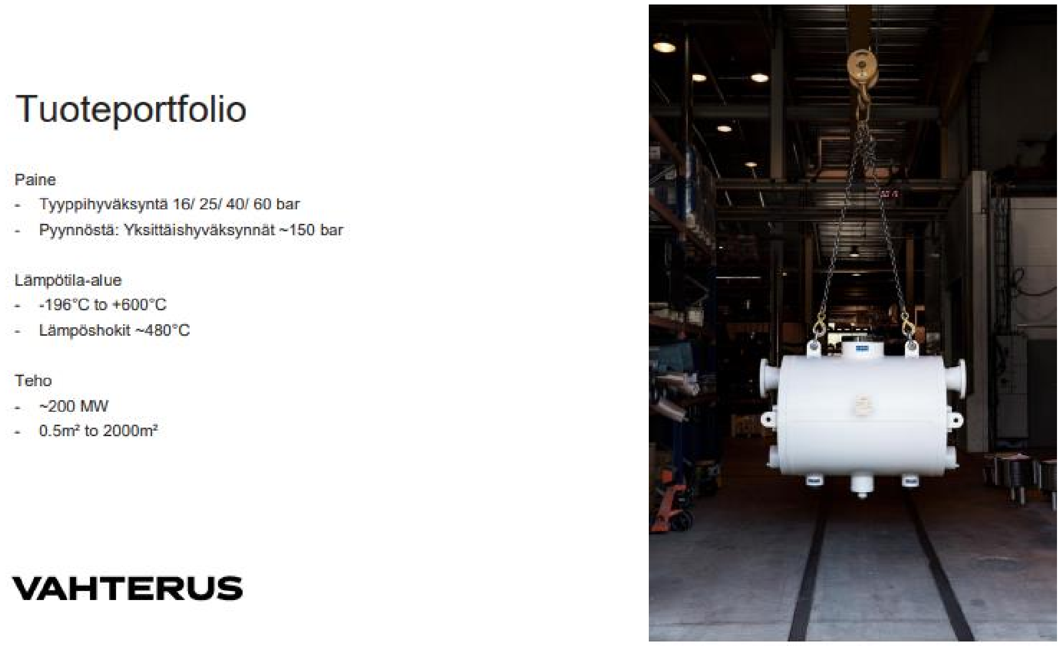


Figure 22. Plate heat exchangers with exceptional operational conditions by Vahterus
(References to the seminar:
Author: Ville Kesälä
Seminar: Kuumaöljy Käyttäjäpäivät 20.3.2019)

Translations in English:

Product portfolio

Pressure

- Qualification approval for 16/25/40/60 bar
- On request: separate approval for circa 150 bar

Temperature ranges:

- from - 196 °C to + 600 °C
- thermal shockscirca 480 °C

Power

- circa 200 MW
- plate areas from 0.5 m2 to 2000 m2

6. Limited operating temperature, due to finite temperature resistance of the gasket material.
7. Small flow path, and not suited for gas-to-gas heat exchange or steam condensation.
8. High blockage occurrence especially with suspended solids in fluids.
9. The flow resistance is larger than the shell and tube.

3 Research material and methods

3.1 Heat control simulation system

The scheme of the main pump testing system is in figure 23:

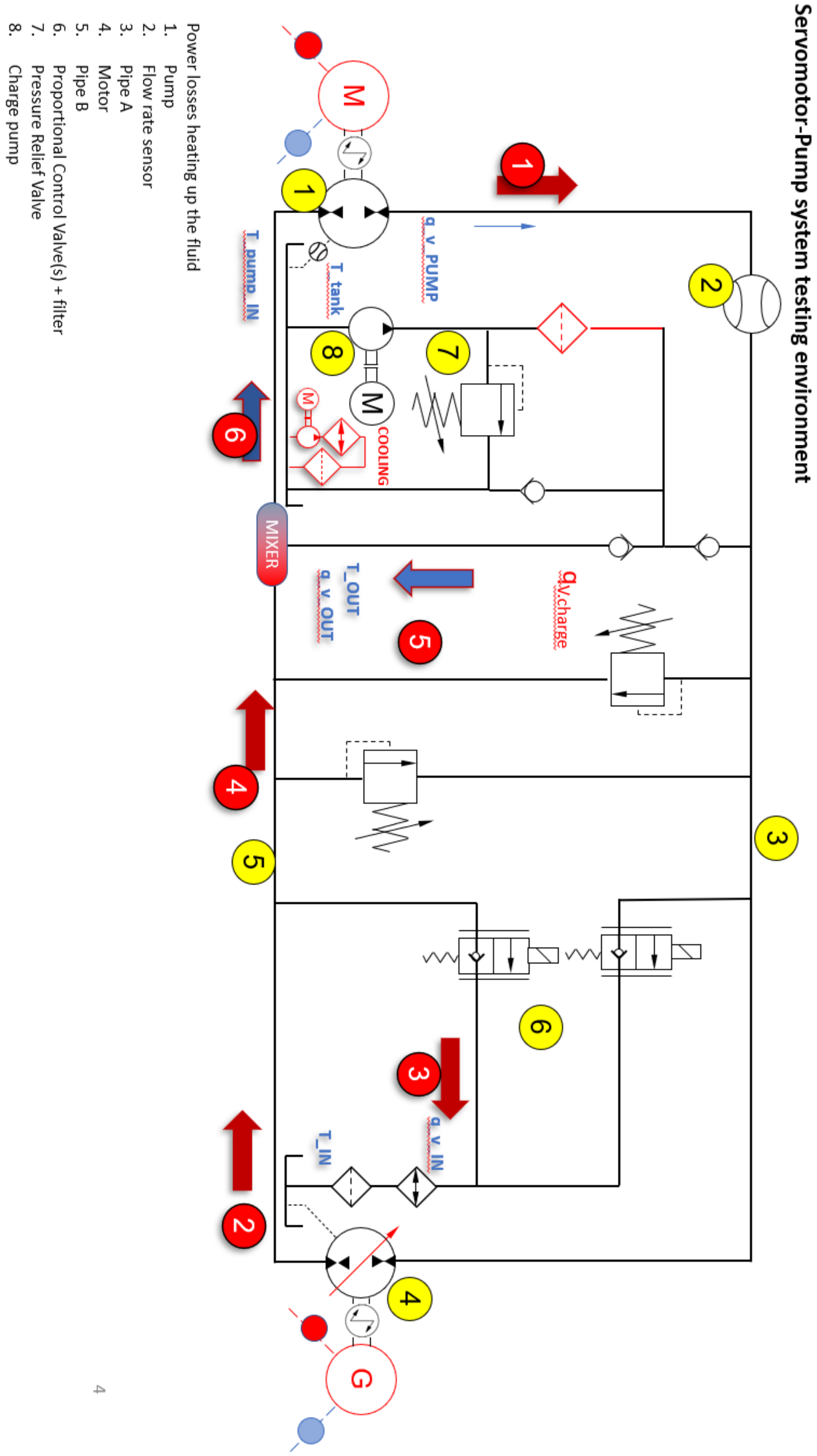


Figure 23: System scheme

System components (yellow):

1. Pump
2. Flow rate sensor
3. Pipe A
4. Motor
5. Pipe B
6. Proportional Control Valve(s) + filter
7. Pressure Relief Valve
8. Charge pump

The RED numbers are heat flows.

Of which simplified schematic is in figure 24 :

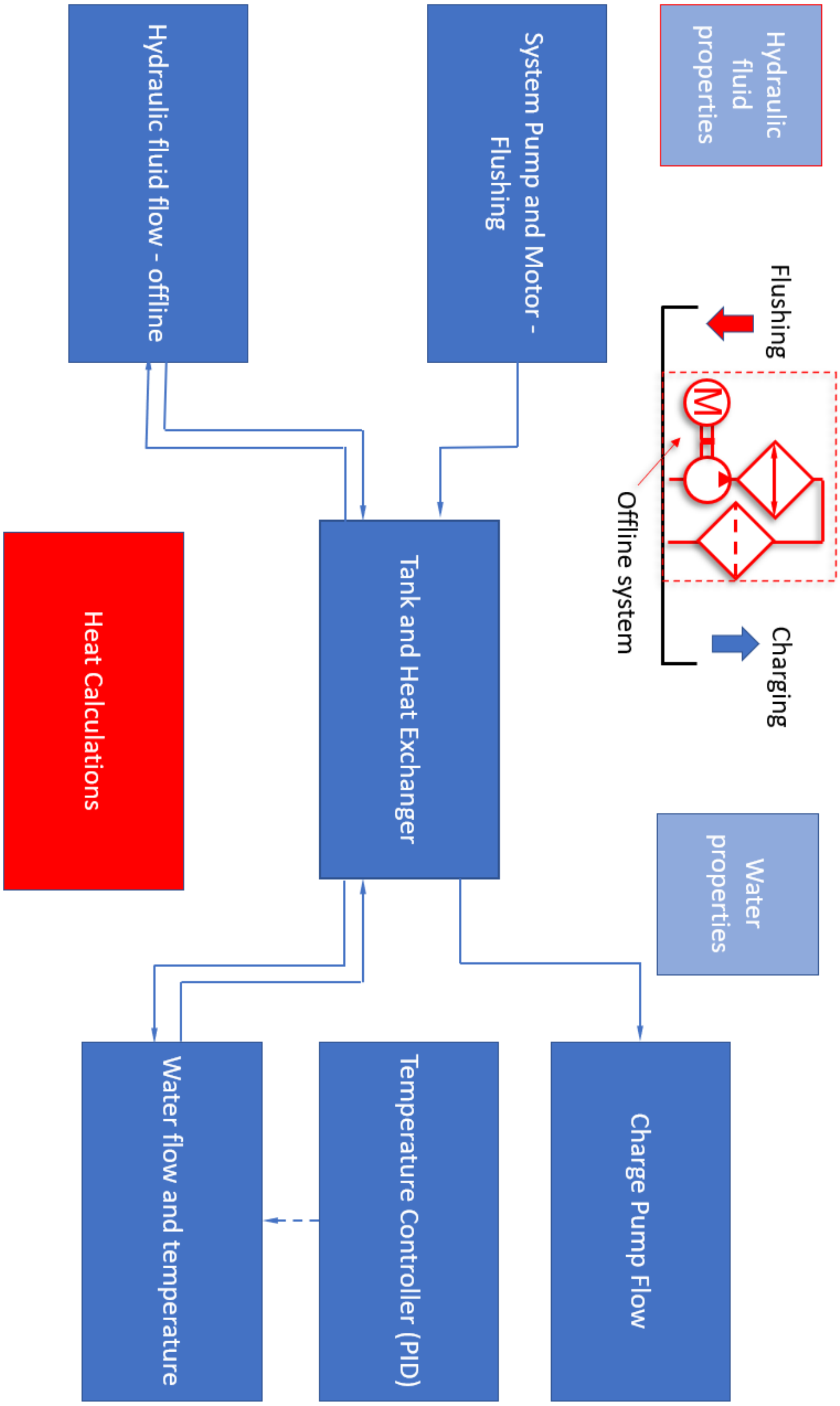


Figure 24: Simplified schematic of the main system

We will use an Offline cooling and filtering system, schematized in figure the flows **into** the tank are:

- flushing flow from the main system
- pump's and motor's case drain flows
- offline system's return flow

the flows **from the** tank are:

- charge flow back to the main system
- possible excess flow from the charge pump through the low-pressure pressure relief valve
- offline system's inlet flow

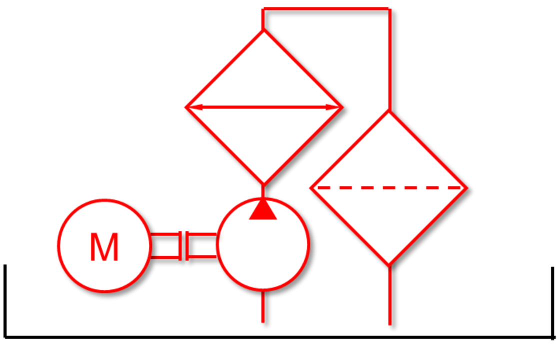


Figure 25:subsystem scheme

The System's Flows and Subsystems are schematized in figure 26:

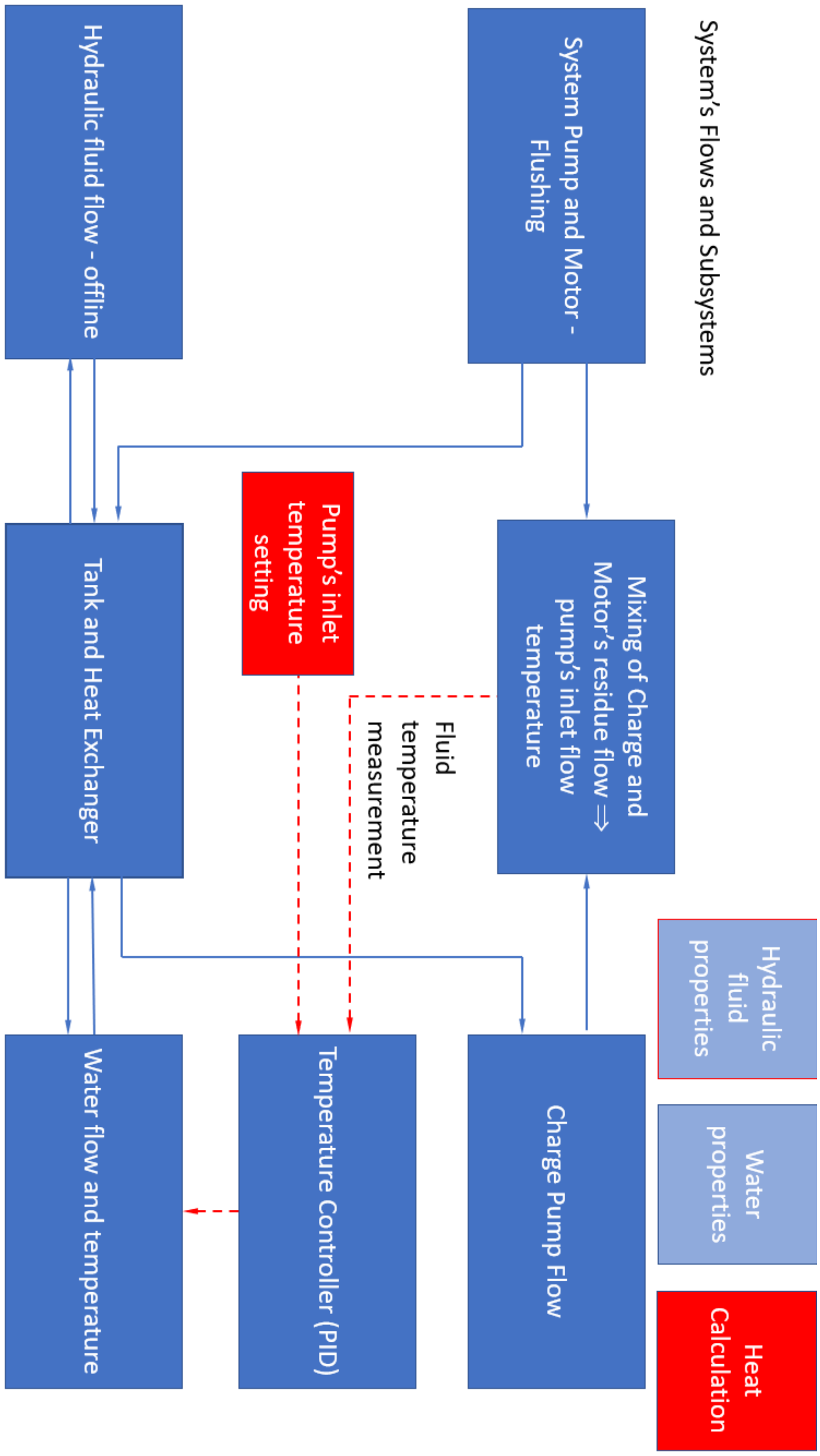


Figure 26: Flows and subsystem

The general scheme in Simulink is in figure 27:

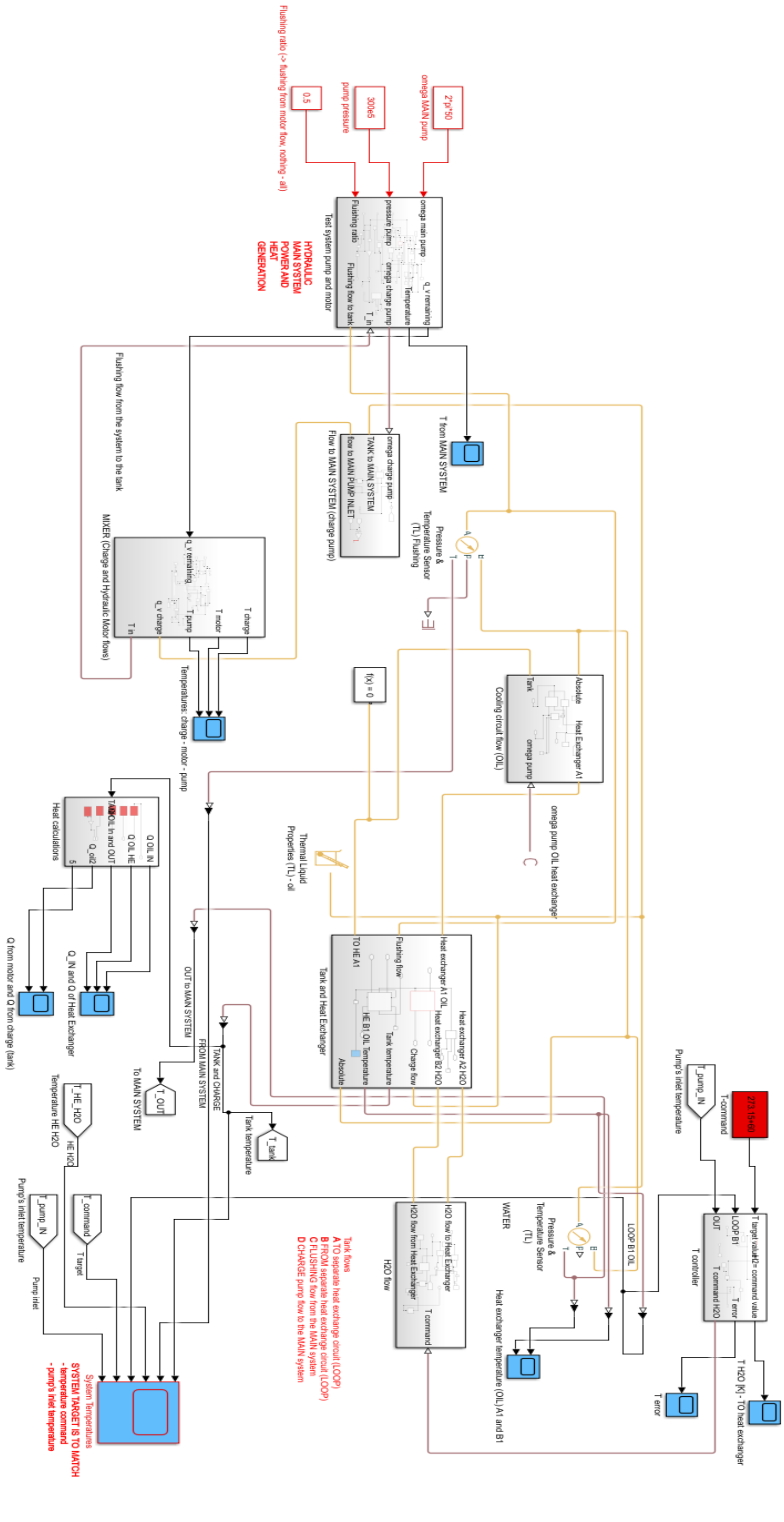


Figure 27: Simulink scheme

This part will describe the system in its various parts. The scheme is complex, so we will start to describe it from the tank flows:
Figure 8

3.2 Flows from and to tank

3.2.1 D flow

D flow is the one from the tank to the main system, highlighted in fi

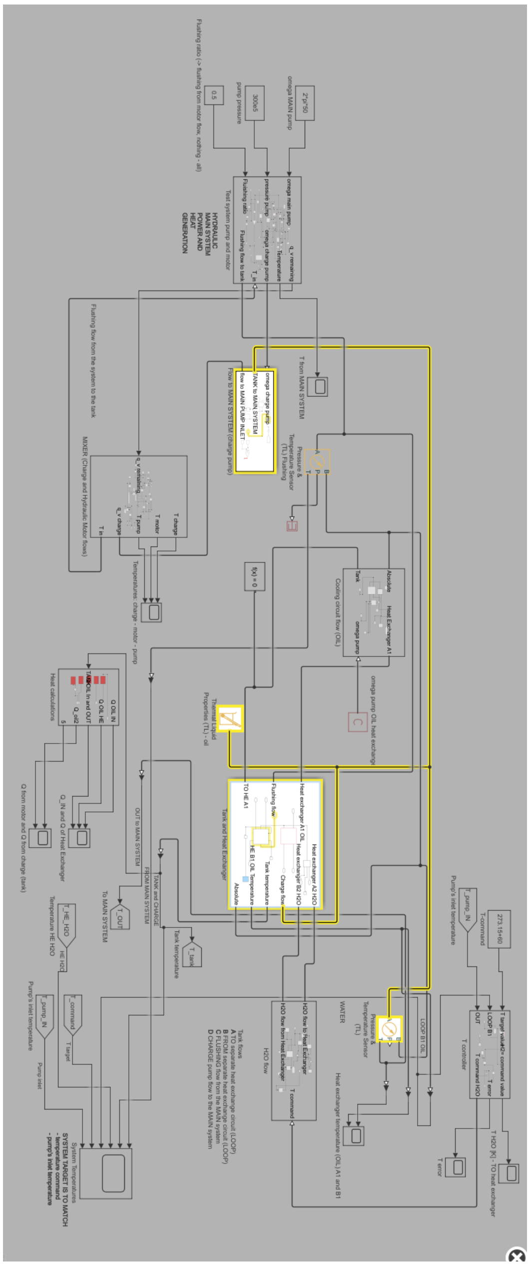


Figure 28: D flow from the tank to the main system

3.2.2 A flow

A flow delivers fluid to the heat exchange circuit (Figure 29)

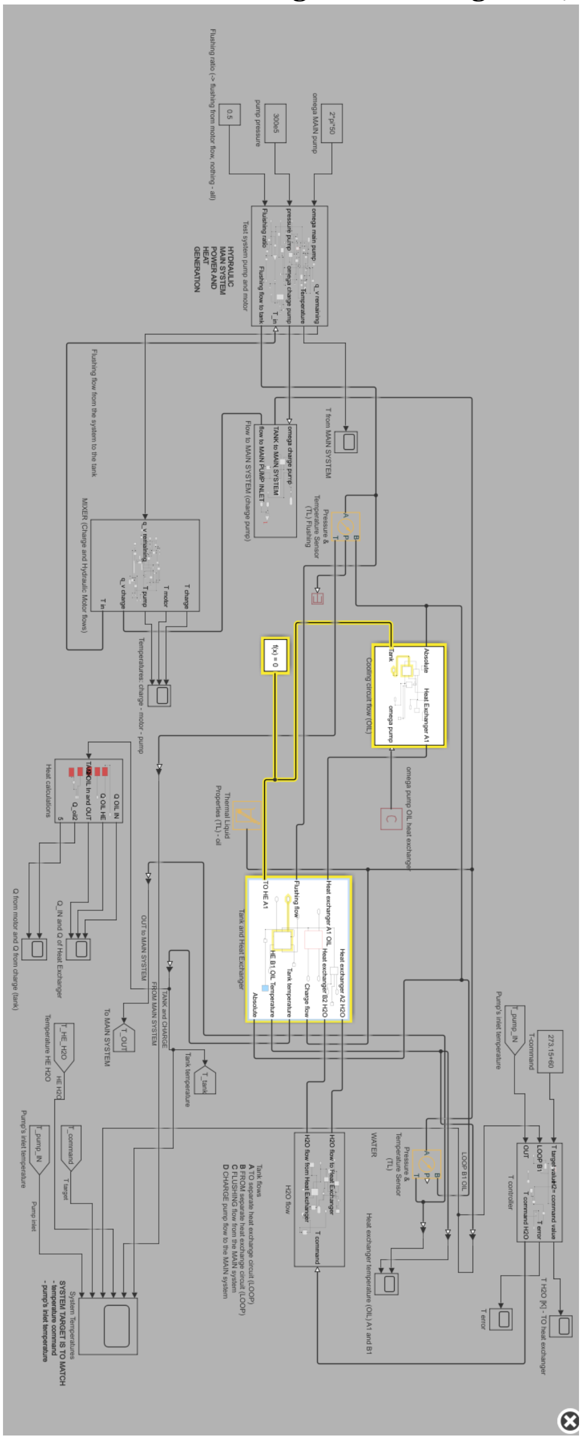


Figure 29: A flow from the tank to the heat exchanger

And goes into the tank through the subsystem in Figure 30.

The oil-side fluid flow is generated and controlled with a fixed-displacement pump and an electric motor. The electric motor (speed) could be controlled with an inverter in practice.

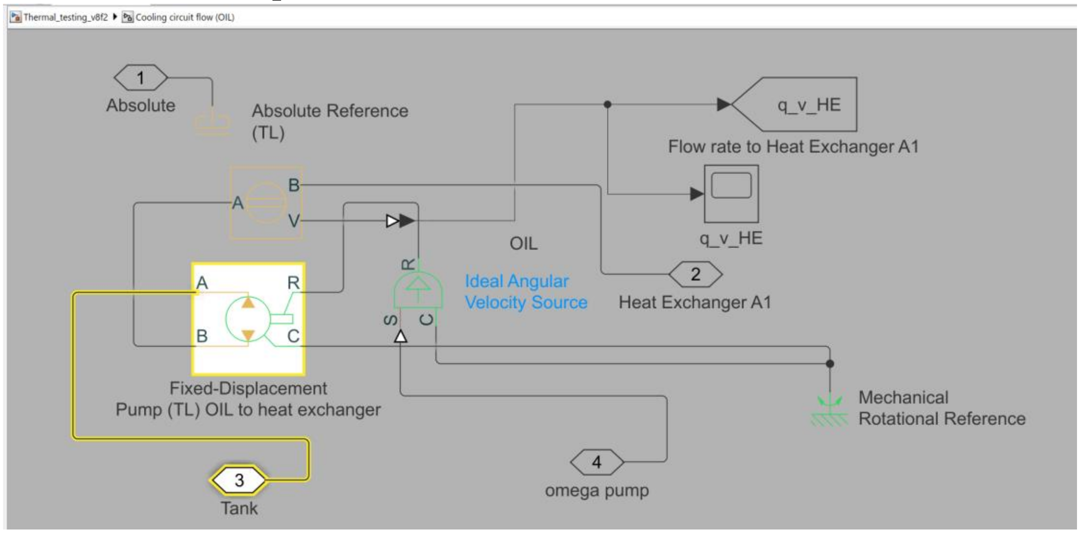


Figure 30: Fixed-displacement pump and an electric motor controlling heat exchanger oil flow

3.2.3 B flow

B flow brings fluid back to the tank from the heat exchanger. (Figure 31)

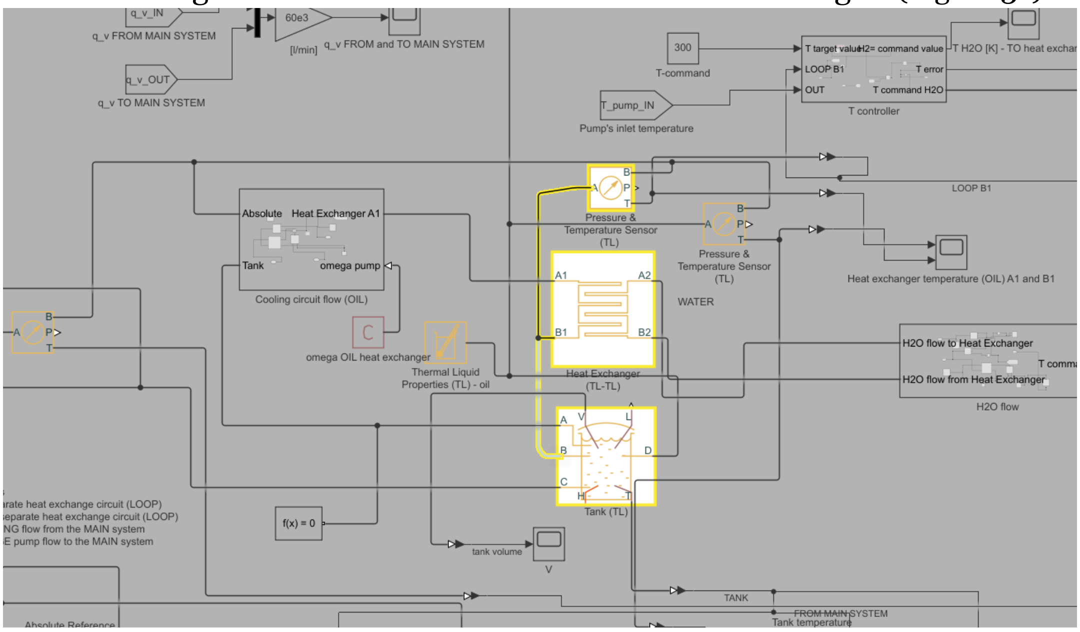


Figure 31: Flow back to the tank from the heat exchanger

3.2.4 C flow

Tank's C flow is the flushed one from the main testing system (Figure 32).

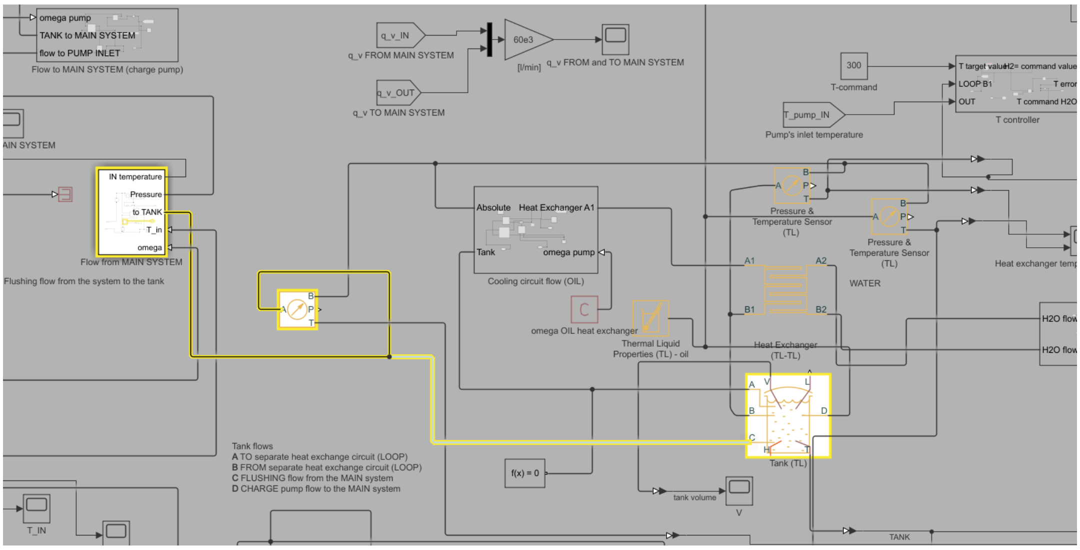


Figure 32 Flushed flow to tank

3.3 PID Control system

3.3.1 Temperature control block

In the Temperature control block (Figure 33), the temperature of the fluid is compared with the desired testing temperature in red and a command for the water temperature to the H2O flow block is sent. The control could have been operated also by controlling the water flow rate.

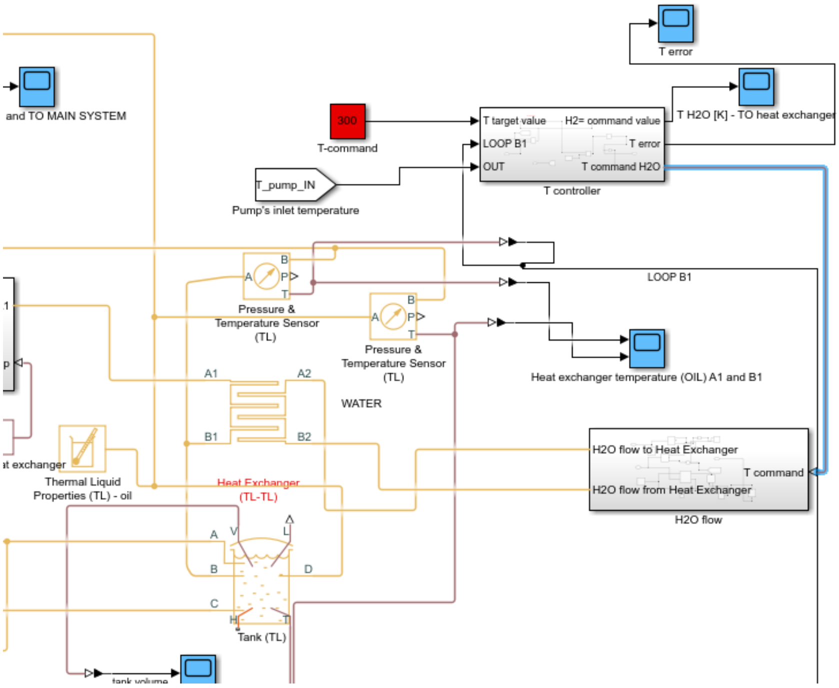


Figure 33 Controlling the pump

We can see the internal configuration of the "T controller" subsystem in figure 15. The target temperature value and the measured pump's inlet temperature are compared and the error signal is sent to the PID controller. The control signal is saturated based on the achievable temperature limits for the water flow the minimum being 280 K and the maximum 350 K. In the actual system the fluid temperature can be controlled for example with two proportional control valves, on

e for hot water and the other for cold water.

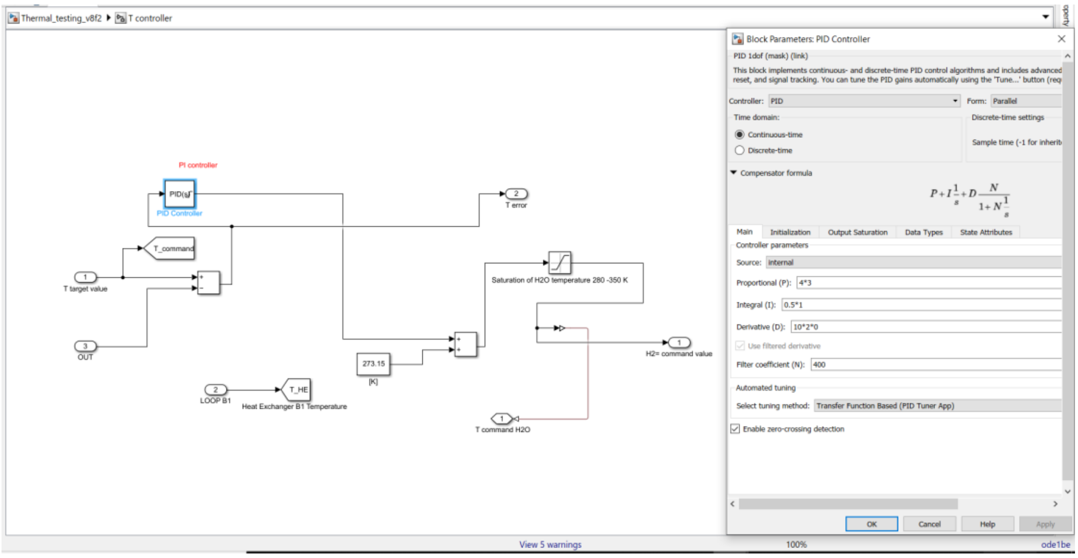


Figure 34: PID controller parameters used in the study

The PID has been tuned to achieve an accurate target value for the fluid and to minimize oscillations in the final temperature signal.

3.4 Heat exchanger modeling

We can model the heat exchanger based on the correlation equation for corrugated plates of heat exchangers.(30)

As an example heat exchanger a plate heat exchanger Funke FP 10-29-1-NH – 25.0 bar was used as it was available in the laboratory.

The heat exchanger's performance data was limited, and thus the accurate prediction of its operation was practically impossible. The heat exchanger model was used merely to study the applicability of the temperature control principle. However, realistic characteristics were aimed at.

The heat exchanger has been tested with only water as fluid. As the other fluid in the system will be hydraulic fluid (mineral oil), the accuracy of the parameters related to this side of the system is likely to be less accurate.

Table 3: Heat exchanger data

Fluid		**Water**	**Water**
Mass flow rate	[kg/s]	3.59	3.59
Volume flow rate	[m3/h]	13.121	12.988
Inlet temperature	[°C]	60	20
Outlet temperature	[°C]	40	40
Dynamic viscosity	[cP] IN/OUT	0.468 / 0.655	0.988 / 0.655
Density	[kg/m3] IN/OUT	981.4 / 990.3	997.3 / 990.3
Specific heat capacity	[kJ/kgK] IN/OUT	4.173 / 4.177	4.191 / 4.177
Thermal conductivity	[W/mK]	0.645 / 0.635	0.624 / 0.635
Heat duty	[kW]	300	300
Eff. heat transfer area	[m2]	2.49	2.49
Log./effective ΔT	[K]	20	20
Heat transfer coeff. req./act.	[W/m2K]	6028/6353	6028/6353
Fouling factor, total	[m2K/W]	0.0000085	0.0000085
Overdesign	[%]	5	5
Pressure loss	[kPa]	24.195	23.389
Channel velocity	[m/s]	0.44 / 0.76	0.45 / 0.73
Connection velocity	[m/s]	1.56	1.55
Number of passes in series	[-]	1	1
Number of plates total	[-]	29	19
Mix of Channeltype	[-]	9·HH+5·HL	9·HH+5·HL
Weight	[kg] DRY / WET	127 / 134	
Volume	[dm3]	3.511	3.511
Plates	[-]	AISI 316L / 1.4404 (0.60 mm)	
Min/Max design/test press.	[bar, gauge]	0 / 25 / 32.5	0 / 25 / 32.5
Min/Max design temp.	[°C]	0 / 110	0 / 110

The correlation equations for turbulent and laminar flows are(1 and 2):

$$Nu_{\text{turbulent}} = 0.2\,(Re)^{0.67} Pr^{0.4} \left(\frac{\eta}{\eta_w}\right)^{0.1} \tag{1}$$

$$Nu_{\text{laminar}} = 1.68\,(Re\,\mathrm{Pr}\frac{d_e}{L})^{0.4} \left(\frac{\eta}{\eta_w}\right)^{0.1} \tag{2}$$

in which:

Re	Reynolds number
Pr	Prandtl number
d_e	hydraulic diameter
L	length of the plate
η	dynamic viscosity of fluid
η_w	dynamic viscosity of the fluid at heat exchanger plate temperature, average of hot and cold side fluid temperatures

Assumptions were:

- For water, the heat transfer coefficient in a turbulent regime would be about(3):

  $$h_{H2O} = (6.812 \cdot 10^3)\frac{kg}{s^3 \cdot K} \tag{3}$$

- For oil, the flow pattern would presumably be laminar and the heat transfer coefficient has been assumed(4):

  $$h_{OIL.corr} := \frac{k_{OIL}}{D_{hydr}} \cdot Nu_{OIL.corr} = 444.304\,\frac{\mathrm{kg}}{\mathrm{s^3 \cdot K}} \tag{4}$$

  In the case of turbulent oil flow, we could count on a higher heat transfer coefficient(5):

  $$h_{OIL.corr.turb} := \frac{k_{OIL}}{D_{hydr}} \cdot Nu_{OIL.corr.turb} = (1.243 \cdot 10^3)\frac{\mathrm{kg}}{\mathrm{s^3 \cdot K}} \tag{5}$$

  In the case of oil laminar flow, the Nusselt number would be (6):

  $$Nu_{OIL.corr} := 1.68 \cdot \left(Re_{OIL} \cdot \mathrm{Pr}_{OIL} \cdot \frac{D_{\mathrm{hydr}}}{L_{\mathrm{plate}}}\right)^{0.4} = 11.548 \tag{6}$$

  Instead, in case of turbulent oil flow, the Nusselt number would be (7):

  $$Nu_{OIL.corr.turb} := 0.2 \cdot \mathrm{Re}_{OIL}^{0.67} \cdot \mathrm{Pr}_{OIL}^{0.4} = 32.307 \tag{7}$$

4 Results

The system can handle controlling the temperature of the fluid sufficiently well in most situations. There exist more tricky situations at the extremes, for example:

- If the temperature we want to reach is high (around 60 degrees) and pump power is low (e.g. 10 revs/s at 50 bars) and the starting temperature is low (around 20 degrees), naturally it takes quite a long time to reach the testing temperature (around 10 minutes). This is quite inevitable because we would need a heating system for it to be fast, and the investment is not worth it, since this is not a common usage situation.
- In the case of high power pump testing (e.g. 50 revs/s at 300 bars) we need quite high flows of water to cool down effectively.

We will test the system varying:

- Target temperature
- Pump power
- Flushing ratio (the proportion of flow that goes to the tank instead of recirculating in the main system)

To understand its behavior in different requirements and its flexibility. Starting from a low temperature, With a high Starting power pump, varying the flushing ratio from 0.5 in figure 35 to 1 in figure 36 we reach the stabilization of the pump inlet temperature (the most important parameter for testing) in a quicker time (340 to 250 s). The results are presented in Figures 35 and 36.

The signals presented in the plot are:

T_target	the target temperature for the pump's inlet flow
Pump inlet	pump's inlet flow temperature
LOOP B1 OIL	heat exchanger's outlet temperature for hydraulic fluid
HE H2O	heat exchanger's water temperature for inlet flow
FROM MAIN SYSTEM	the temperature of flushing flow from the main system
TANK and CHARGE	the temperature of charge flow and fluid in the tank

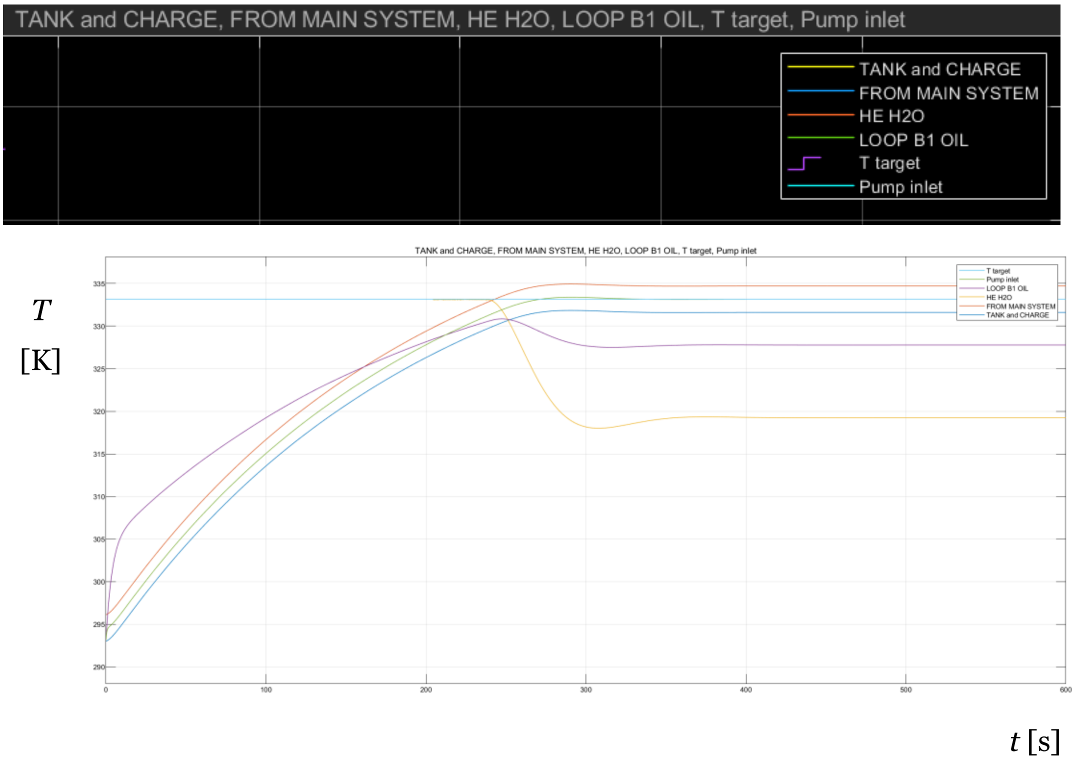


Figure 35. Temperatures with a flushing ratio of 0.5 and a target temperature of 60 °C

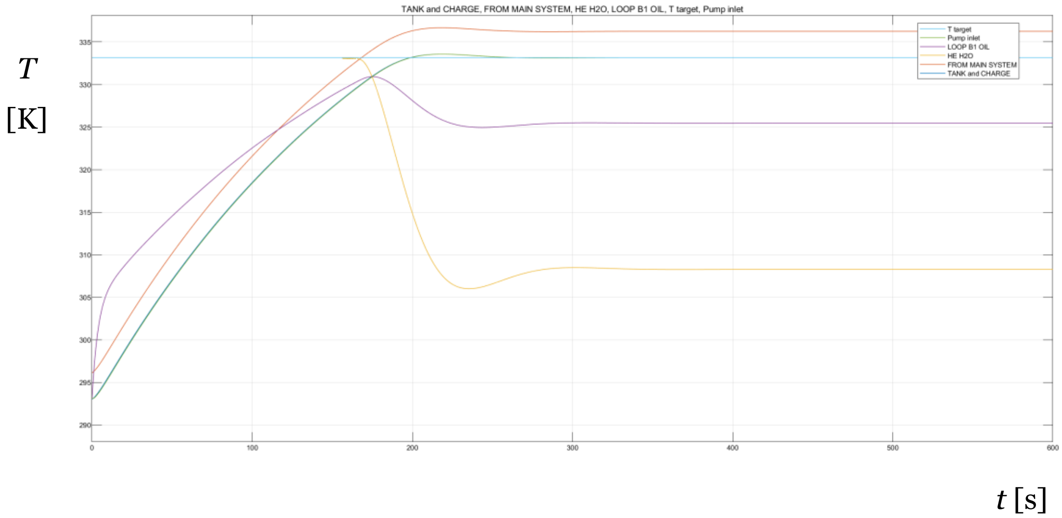


Figure 36. Temperatures with a flushing ratio of 1 and a target temperature of 60°C

Instead, using a flushing ratio of 0.1, the settling time becomes 480 s (Figure 37).

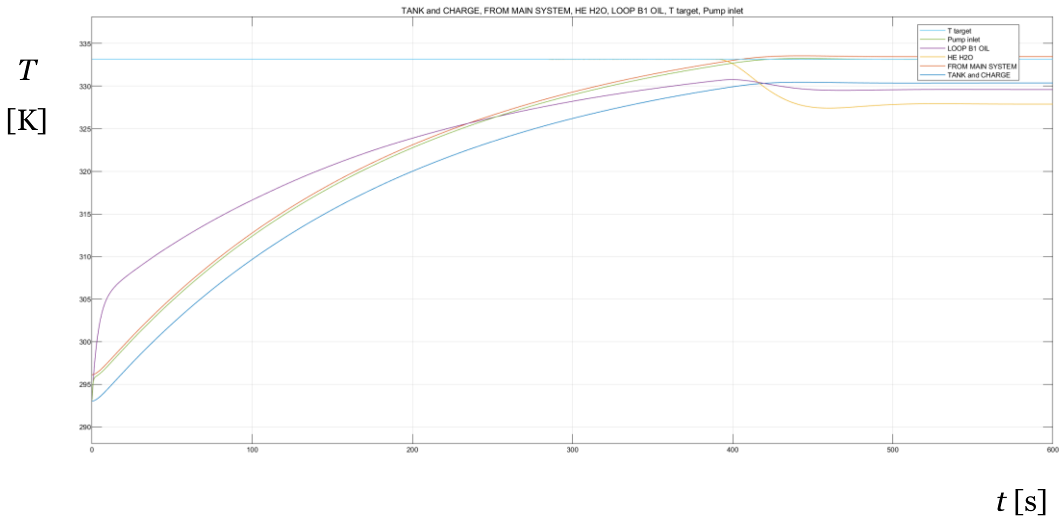


Figure 37. Temperatures with a flushing ratio of 0.1 and a target temperature of 60°C

Also, the oscillations in temperatures become less. This is because of the different proportions of flow recirculating.

Using a flushing ratio of 0.5, but lowering the desired temperature to 40 degrees, the system reaches the target value much faster (170 s for the pump's inlet temperature) and can maintain it stable, as we can see in figure 38.

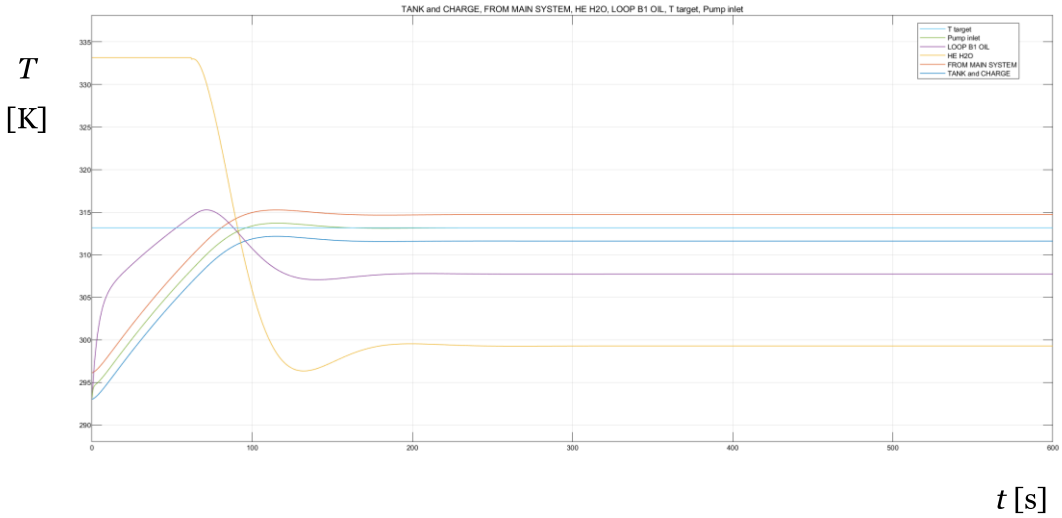


Figure 38. Temperatures with a flushing ratio of 0.5 and target temperature of 40°C

Maintaining the same parameters but having a less powerful pump, takes a longer time for the system to reach the desired temperature. In the case of 40 degrees the difference is quite small, but in the case of 60 degrees becomes very consistent (the small pump takes 600 s and the more powerful one 320 s).

To account for uncertainties in the OIL side heat exchanger's heat exchange coefficient due to the quality of oil and flow type (laminar or turbulent), some similar tests have been performed. The oil in the heat exchanger will be in a laminar regime in most testing cases, decreasing the heat exchange coefficient to around 235 W/(m^2 K) with 52.5 l/min flow. Anyway, the results showed the system handles well this uncertainty, also the cases if the heat exchange coefficient changes considerably.

However, the temperature control system always reacts in a very predictable and steady way, which means it is reliable at least in these simulated cases.

5 Summary/Conclusions

Targets (from the Introduction) were:

- Is it possible to build a reliable Simulink model of the system?
- Will the system be stable?
- Will the heat exchanger be enough?
- Will the cooling and heating times be acceptable?

- A Simscape-based simulator was built to study the functionality of a test rig controlling the hydraulic pump's inlet flow temperature.
- The estimated heat exchanger performance was based on an existing plate heat exchanger for which tested data transfer data for water flow was available.
- The hydraulic fluid (mineral oil) side of the heat exchanger performance of the heat exchanger was estimated by using a correlation equation for the corrugated plate heat exchanger
- According to the numerical analysis, the flow pattern in the hydraulic fluid side is presumably laminar or transitional. Because not all the heat exchanger-related parameters were available, accurate estimates for the hydraulic fluid side of the heat exchanger performance were not able to achieve. That is why the study concentrated on the functionality of the thermal control system. The heat exchange parameters and amount of flushed system fluid were altered to study the robustness of the control.
- The simulations showed that:
    - the control seemed stable and changes in some system parameters did not deteriorate the operability
    - the target value for the pump's inlet temperature was able to achieve
    - the use of closed-loop control in thermal control may speed up the actual pump testing, which has previously been a slow

process because of long transition times between the operational points (rotational speed, pressure).
   - In the tests, the settling times were in order of several minutes because of large changes in command values (target temperatures), but in practical tests, the changes in target temperature are rare, which might make reaching the end value faster.
   - Even though the heat exchanger performance parameters were not accurate, the thermal control system's performance seems to be adequate for the intended usage.

Possible improvements to this research are:

- Realizing the system with real components and seeing what are the problems regarding cost, construction, occupied space, and support systems. This will probably be done by the Aalto Department in the future.
- Measuring the performance of the real system to evidence the errors in modeling to acquire better modeling skills to apply in the future.
- Improving the system with a heating system for the fluid.
- Designing the system in a way that permits to use of smaller and easily replaceable heat exchangers and tanks.
- Putting on the market the project with the help of a company.

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