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RP-200

Design of PD pump for pumping of molasses

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Problem Description

Background

Presently there is a market for pumping molasses. For the time being six orders for eight molasses pump are placed; the first order with a scheduled delivery in October 2008. Molasse is a liquid with special properties regarding density and viscosity index that are challenging for the pump and pumping equipment. This thesis work is a continuation of the student's project last fall.

The purpose of this thesis is to document the new molasses pump design where the different solutions are clearly explained, this in order to ease future work along the same lines.

Aim

The goal is to make a prototype of a submerged pump specifically made for pumping molasses that can fulfill the customer requirements for flow and pressure. Obtaining reliable test result and demonstration of the pump is desirable before the new product is set into production.

The master thesis will contain the following tasks:

1. Investigate the differences between internal and external gear pump regarding pumping and decide if it necessary to make prototypes of both internal and external gear pump
2. Develop a design of the cog shape of the pinion and gear
3. Estimate maximum theoretical flow as a function of the speed of the pump
4. Make a proposal for a test rig of the pump and describe instrumentation for proper documentation of performance.
5. Design a pump in Pro Engineer and make drawings and production list for the different parts
6. Assembly and test functionality of the prototype
7. If the student has time it is desirable to make a revision of the software developed through the student's project in fall 2007 with additional features

Assignment given: 01. April 2008
Supervisor: Morten Kjeldsen, EPT

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I especially would like to thank Framo that has helped me through this entire project and allowed me the pleasure to work for them during the period from January through October 2007 .

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Abstract

Motivation

There is, at the present time, no submerged molasses pump on the market that is designed specifically for cargo tankers. Due to this I, find it interesting to look into the possibilities of installing a molasses pump in cargo tankers to transport molasses instead of transporting molasses in containers as it is done today. It is challenging to come up with a new product, and the motivation of actual be able to release a pump for the international marked is indescribable.

Problem

The goal is to make a prototype of a submerged pump specifically made for pumping molasses that can fulfill the customer requirements for flow and pressure. Obtaining reliable test result and demonstration of the pump is desirable before the new product is set into production.

Approach

Molasses is a very special and complex cargo, due to the complexity, $8 \cdot 10^3 kg$ of molasses was ordered from Australia. Then it was possible to do several tests on the actual molasses which the current market is for.

Different pump designs have to be evaluated and then some design can be put into prototyping. The prototypes needs to go through several test so as much knowledge as possible can be gained before the pump is released on the market.

Conclusion

There is definitely a large market for transporting molasses by cargo tankers. There are already several orders for a molasses pumping system. Molasses seems to be a more complicated cargo pump then first assumed because of its big variations in viscosity due to temperature and different batches.

There are many unknown factors involved in pumping molasses and as further it was dogged in to the problems new ones occurred. But the problems have been solved, some has been hard to solve. After three prototypes the customers requirements were finally achieved, and then all the hard work has finally given result.

Even if the pump design is ready for the first order, many new question have arrived and this is the motivation to continue with the process that has already started. Especially interesting is the new technology that will be available next year regarding CFX a motivation factor to keep trying to rise the efficiency.

Samandrag

Motivasjon

Det er for augeblikket ingen nedykka pumpe for melasse på marknaden spesielt designa for supertankarar. Grunna dette finn eg det svært interessant å sjå nærmare på moglegheitene å installere melasse pumper ombord supertankarar for å kunne transportere melasse i staden for å transportere det i behaldare slik som det blir gjort i dag.

Det er utfordrande å komme opp med eit nytt produkt og faktisk få lov til å lansere det på verdsmarknaden er ein stor motivasjonsfaktor.

Problem

Målet er å kunne lage ein neddykka pumpe spesielt for melasse som oppfyller kundens krav for volumstrøm og trykk. Vis det er mulig er det sterkt ønskeleg å oppnå gode test resultat og demonstrere pumpa før den blir satt i produksjon.

Framgangsmåte

Melasses er ein menet spesiell type væske og difor vart $8 \cdot 10^3$kg med melasse bestilt frå Australia. på denne måten var det mulig å gjennomføre fleire testar på den faktiske melassen som det aktuelle marknaden er for.

Ulike pumpe design har blitt evaluert før eventuelle prototypar kan bli produsert. Prototypane må gå gjennom ein rekke testar slik at så mykje kunnskap som mulig kan bli etablert før pumpa blir lansert på verdens marknaden.

Konklusjon

Det er definitivt eit stort marknaden for å transportere melasse med supertankarar. Det er allereie fleire ordrar for melasse pumpesystem. Melasse er ein meir komplisert last den fyrst antatt grunna melassen sin store variasjon i viskositet grunna temperatur endringar.

Det er mange ukjente faktorar knytte til pumping av melasse, og når ein undersøkte problema knytte til dette grundigare dukka det opp nye problem. Problema har blitt løyst, nokon problem har vore svært vankelege å løyse. Etter tre prototypar har kundens krav endeleg blitt oppfylt, og det harde arbeid har gitt gode resultat.

Sjølv om pumpe designa er klart for den fyrste ordren, har det dukka opp mange nye spørsmål som er med på å auke motivasjonen for vidare arbeid. Spesielt interessant er den nye teknologien som ligger i den nye versjonen av CFX som kommer i 2009 ein stor motivasjonsfaktor for å vidareføre prosessen.

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Nomenclature

Multi Archieve system software

Pro Engineer CAD software

CFD Calculation of Fluid Dynamics

CAD Computer aided design

RP 200 Rotary pump 200 cm^3

Catman Logging system

CFX A CFD sofware by ANSYS coroporation

T Temperature [Ľ]

a Constant $[cSt^2]$

b Constant $[cSt]$

Q_{motor} Flow to the motor $[m^3/h]$

ηh_{motor} Hydraulic efficiency of the hydraulic motor $[-]$

ηmh_{motor} Hydraulic mechanical efficiency of the hydraulic motor $[-]$

ηm_{motor} Mechanical efficiency of the hydraulic motor $[-]$

M_{motor} Moment of the motor $[Nm]$

N_{motor} Speed of the motor $[rpm]$

n_{motor} Speed of the motor $[rev/s]$

P_{motor} Power of the motor $[W]$

ηt_{motor} Total efficiency of the hydraulic motor $[-]$

ν Viscosity $[cSt]$

V_{motor} Volume of the motor $[m^3]$

ηv_{motor} Volumetric efficiency of the hydraulic motor $[-]$

Q	Actual flow $[m^3/s]$
h_a	Addendum [mm]
$d\phi_1$	Angle that the moment for gear 1 works over $[-]$
$d\phi$	Angle that the moment works over $[-]$
ω_2	Angular speed for gear 2 $[rad/s]$
ω_1	Angular speed for gear 1 $[rad/s]$
ω	Angular speed $[rad/s]$
A_w	Area of wall $[m^2]$
A	Area where hydrostatic pressure works over $[m^2]$
A	Area $[m^2]$
Q_{Avg}	Average volume flow $[m^3/h]$
p_b	Base pitch [mm]
a_w	Center distance between gear $[m]$
$d\phi$	Change in rotation angle $[rad]$
$d\phi_1$	Change in rotation angle for gear 1$[rad]$
$d\phi_2$	Change in rotation angle for gear 2$[rad]$
dti	Change in time $[s]$
L_D	Characterstic length $[m]$
p	Circular pitch [mm]
e_α	Contact ratio [-]
z_{cu}	Critical permissable tooth number [-]
z_c	Critical tooth number [-]
f	Darcy friction factor $[-]$
h_f	Dedendum [mm]
ρ	Density $[kg/m^3]$
d	Diameter of gear/pinion $[m]$
Δp	Differential pressure $[Pa]$

ΔT Differential temperature $[K]$

Va Displaced volume $[m^3]$

μ Dynamic viscosity $[Pa \cdot s]$

x_p Factor for offfset of tool due to ermissable undercut [-]

x Factor for offset of gear production tool [-]

$d\theta$ Finite angle $[rad]$

dL_D Finite characterstic length $[m]$

dM Finite moment $[N \cdot m]$

dr Finite radius $[m]$

dw Finite width $[m]$

dW Finitie work $[J]$

F_{p1} Force due to pressure at gear 1 $[N]$

$F_{ball\,bearing}$ Forces on ball bearing $[N]$

$\mu_{ball\,bearing}$ Friction coeffisient for ball bearing $[-]$

b Height of gear $[m]$

h_{Gear} Height of gear $[m]$

h Height of gear/pinion $[m]$

h_{Pinion} Height of pinion $[m]$

β Helix angle [°]

D_h Hydraulic diameter $[m/s]$

p_2 Hydrostatic pressure on gear $[Pa]$

r_i Inner radius $[m]$

p_i Internal pressure $[Pa]$

ν Kinematic viscosity $[m^2/s]$

$Q_{leakage}$ Leakage flow $[m^3/s]$

L_{ab} Line of action [mm]

$D_{gear\,max}$ Maximum outer diameter of gear $[m]$

$D_{pinion\,max}$ Maximum outer diameter of pinion $[m]$

$N_{motor\,max}$ Maximum speed of the motor $[rpm]$

$\sigma_{allowed}$ Maxuimum allowable stress $[Pa]$

$\eta_{m_{pump}}$ Mechanical efficiency for the pump $[-]$

$D_{C\,Min}$ Minimum recomanded diameter of cargo pipe $[mm]$

m_1 Module for gear set 1 $[m]$

m_2 Module for gear set 2 $[m]$

m_3 Module for gear set 3 $[m]$

m Module [mm]

$M_{ball\,bearing}$ Moment due to ball bearing $[N \cdot m]$

$M_{bushing}$ Moment due to bushing $[N \cdot m]$

$M_{rotating\,cylinder}$ Moment for roating cylinder $[N \cdot m]$

$M_{rotating\,disc}$ Moment for roating disc $[N \cdot m]$

M_{shaft} Moment on shaft $[N \cdot m]$

M Moment working on gear $[Nm]$

M_1 Moment working on gear 1 $[Nm]$

M_2 Moment working on gear 2 $[Nm]$

M_2 Moment working on gear 2 $[Nm]$

M Moment working on gear $[Nm]$

M Moment $[N \cdot m]$

j_n Normal backlash [mm]

z_{Gear} Number of cogs of gear $[-]$

z_{Pinion} Number of cogs of pinion $[-]$

z_{pinion_1} Number of teeth for pinion in gear set 1 $[-]$

z_{pinion_2} Number of teeth for pinion in gear set 2 $[-]$

z_{pinion_3} Number of teeth for pinion in gear set 3 $[-]$

z_G Number of teeth for the gear [-]

z_P Number of teeth for the pinion [-]

z Number of teeth on the given gear $[-]$

x_L Offset of gear production tool [-]

A_w Offset of the motor center axis and the center axis of the trunk $[mm]$

$d_{a_{gear}}$ Outer diameter of gear $[m]$

$d_{a_{pinion}}$ Outer diameter of pinion $[m]$

D_p Outer diameter of pump $[mm]$

d_a Outer diameter [mm]

r_o Outer radius $[m]$

θ_w Part covered by wall $[-]$

d Pitch diameter [mm]

P_{shaft} Power at shaft $[W]$

$P_{rotating\,cylinder}$ Power for roating cylinder $[W]$

$P_{rotating\,disc}$ Power for roating disc $[W]$

P_{cargo} Power for the cargo $[W]$

$P_{leakage}$ Power loss due to leakage flow $[W]$

$P_{mechanical}$ Power loss due to mechanical factors $[W]$

$P_{ball\,bearing}$ Power losses due to ball bearing $[W]$

$P_{bushing}$ Power losses due to bushing $[W]$

P_{heat} Power transfered to heat up fluid $[W]$

α Pressure angle [°]

p_{motor} Pressure for the hydraulic motor $[W]$

c Radial clearance factor [-]

r_{h1} Radius of addendum circle of gear 1 $[m]$

r_{h2} Radius of addendum circle of gear 2 $[m]$

r_{shaft} Radius of shaft $[m]$

r Radius $[mm]$

p_r Return pressure $[Pa]$

t_s Rim thickness of support ring $[mm]$

d_f Root diameter [mm]

Ω Rotation speed $[rad/s]$

ω_{in} Rotation speed of the input shaft [rpm]

ω_{out} Rotation speed of the output shaft [rpm]

τ Shear stress $[Pa]$

N_{Gear} Speed for gear $[rpm]$

N_{Pinion} Speed for pinion $[rpm]$

Cp Spesific thermal capasity $[kJ/(kg \cdot K)]$

t_w Stress in θ direction for a cylinder under internal pressure $[mm]$

p_s System pressure $[Pa]$

v_t Tangential speed $[m/s]$

v_ci Tangential velocity for rotating cylinder $[m/s]$

t_i Thickness $[m]$

t Time $[s]$

A_{Gear} Area between cogs of the gear $[m^2]$

A_{Pinion} Area between cogs of the pinion $[m^2]$

A_{tot} Total area between one pair of cogs $[m^2]$

e Total contact ratio [-]

$N_{parameters}$ Total number of parameteres $[-]$

N_{points} Total number of points $[-]$

N_{tot} Total number of test $[-]$

N_{times} Total number of timest $[-]$

V_{tot} Total volume between one pair of cogs $[m^3]$

$\eta_{t_{pump}}$ Total efficiency for the pump $[-]$

i Transmission ratio [-]

D_t Trunk diameter $[mm]$

v Velocity $[m/s]$

$\eta_{v_{pump}}$ Volumetric efficiency for the pump $[-]$

t_w Wall thickness of housing $[mm]$

v_w Wall velocity $[m/s]$

e Width of space between teeth [mm]

s_2 Width where force from gear2 works over $[m]$

s_1 Width where hydrostatic pressure works over $[m^2]$

w Width $[m]$

c Working distance of the pressure $[m]$

b_w Working face width of gear [mm]

1 Introduction

At first sight molasses, a thick syrup, might seem impossible to pump it into cargo tankers in order to make transportation faster and cheaper. So far it has only been transported by railroad and in containers, which means high cost and rather low efficiency. Due to the fact that the demand for molasses will increase during the next years, other ways of transportation is needed soon.

1.1 Aim

The aim is to see how molasses can be transported in vesselships using a submerged gear pump. In addition it needs to withstand several strict limitations.

1.2 Motivation

There is, at the present time, no molasses pump on the market that is designed specifically for vessel ships. Due to this I, find it interesting to look into the possibilities of installing a molasses pump on a vessel ship instead of transporting molasses in containers as it is done today. In fall 2006 Frank Mohn Fusa A/S was asked to come up with a solution of transporting molasses from one land based reservoir to another one by vessel ship. The customer, Dorval Kaiun wants to transport molasses from one place of the coast of Australia to another place of the coast of Australia or to New Zea-land. The whole concept of being a part of developing a such special product as this from the start in 2006 to October 2008 when the first pumps is scheduled to be delivered is a motivation factor itself.

1.3 Problem

Molasses is a complex cargo and therefore several possible solutions needs to be evaluated. These proposal then needs be put into prototyping before the final product can be released on the market. This involves several task, among investigate performance, geometry, test configuration, production papers.

1.4 Approach

The first approach is to look into technology that already exist and investigate problem with these technologies. When this is done then it is first possible to come up with good improvements and new suggestions. This is not always easy, and therefore there will most likely be a lot of testing and failure. In Frank Mohn Fusa A/S there is already a lot knowledge regarding pumping, and by using this there should be possibly to come up with a good submerged pump especially designed for pumping molasses.

2 Molasses

2.1 Introduction

Molasses or treacle is a thick syrup by-product from the processing of the sugar cane or sugar beet into sugar wik [2007a]. Nowadays it is used in animal food production and fermentationfer [2007], K.H. Steinkraus [2002] (ethyl alcohol, rum, yeast, lysine and monosodium glutamate)wik [2007b] Molasses is becoming more and more important these days due to its advantage of requiring very little further processing for animal feed, as it is the case for many of the competing food productsAustralia [2007]

The use of molasses by the intensive cattle feeding industries depends on availability and relative cost compared with alternative products available. Transport is a significant factor in determining the overall cost for molassesCogo [2006]

Today the transport is mainly organized via road and railroad. In order to decrease the costs for transport, it would be convenient to use vessels instead.

Australia has an expanding sugar cane industry located principally in coastal Queensland, but also in northern New South Wales and northern Western Australia Canegrower [2006], Australia [2007], Cogo [2006] . The product from these sugar canes is Australian blackstrap molasses, which is used as a raw material in the stock feed industries supplying the domestic and export marketsCogo [2006], Australia [2007].

2.2 Cane molasses

The sugar cane plant is harvested and stripped of its leaves. Its juice is then extracted from the canes, usually by crushing or mashing. The juice is boiled to concentrate and to promote the crystallization of the sugar. The results of this first boiling and removal of sugar crystal is first molasses, which has the highest sugar content because comparatively little sugar has been extracted from the juice. Second molasses is created from a second boiling and sugar extraction, and has a slight bitter tinge to its taste.

The third boiling of the sugar syrup gives blackstrap molasses. The majority of sucrose from the original juice has been crystallized but blackstrap molasses is still mostly sugar by calorieswik [2007b]. Unlike refined sugars, it contains significant amounts of vitamins and minerals. Blackstrap molasses is a source of calcium, magnesium, potassium, and iron. One tablespoon provides up to 20% of the daily value of each of those nutrientswik [2007a]. Blackstrap is often sold as a health supplement, as well as being used in the manufacture of cattle feed, and for other industrial uses.

There is also another kind of molasses, sugar beet molasses. This, however, is not relevant for us as it is supposed to be easier to pump than the black strap molasses and has therefore not necessarily to be taken into account at this stage.

2.3 Frank Mohn's specific molasses

To be able to design a pump specific for molasses it was necessary to get molasses, that would be similar to the molasses that later on will be transported by Dorval Kaiun.Therefore, Frank Mohn Fusa A/S Dalen [2007] ordered $8000kg$ of black strap molasses from Australia. This molasses was delivered with a data sheet where the main parameters are given in table 2.1.

Description	Value	Unit
Heat transfer- still	100	$\frac{W}{m^2 \cdot K}$
Heat transfer- flowing	20	$\frac{W}{m^2 \cdot K}$
Specific heat	2.3	$\frac{kJ}{kg \cdot K}$
Density	1420-1450	kg/m^3
Viscosity	30000-3000	cSt
Brix	80	%
Max handling temperature	40	℃
Recommended handling temperature	38	℃

Table 2.1: Technical data for molasses.

2.4 Requirements for pumping molasses

Molasses that will be pumped by this pump has to fulfill several requirements acquired by the customer. According to Oceanic Maritime Services,Services [2006] in Townsville the following points have to be followed.

1. Tanks have to be steam sterilized at $0.35[bar]$ for 30 minutes prior to arrival at load port.

2. Heating of the cargo will commence four days prior to vessel's arrival at discharge port to ensure a uniform temperature of 38℃.

3. Heating should be done gradually, rapid heating must be avoided. Excessive use of steam on molasses will cause caramelizing around pipe work and could lead to degradation of sugars.

4. Copy of heating logs showing actual temperature have to be supplied to receivers at discharge port(s).

2.5 Viscosity

2.5.1 Non-Newtonian fluid

A non-Newtonian fluid is a fluid in which the viscosity changes with the applied strain rate. As a result, non-Newtonian fluids may not have a well-defined viscosity.

Although the concept of viscosity is commonly used to characterize a material, it can be inadequate to describe the mechanical behavior of a substance, particularly of non-Newtonian fluids. They are best studied through several other theoretical properties which relate to the connections between the stress and strain tensors under many different flow conditions, such as oscillatory shear, or extensional flow, which are measured by using different devices or rheometers.

Oobleck, is a typical example of a non-Newtonian fluid. Molasses is supposed to be non-Newtonian fluid, which creates a problem because then the viscosity is no longer only dependent of the temperature, but also the shear rate. There is inadequate evidence available proving molasses having non-Newtonian properties. It has therefore been assumed that our specific molasses used is a Newtonian fluid in the temperature range provided by the customer requirements. This means that in this temperature rate there is little change in viscosity due to the applied shear stress, or that the applied shear stress is small. At these temperatures the molasses behaves more or less like thick oil. This assumption needs to be fully tested, in order to be validated. A method to test this is to pump the molasses through a rheometer for several shear rates and validate that the viscosity is relatively unaffected by these changes..

2.5.2 Different models for viscosity of molasses

Three different models for modeling how the viscosity of the molasses changes regarding a temperature change have been made

- Rule of thumb,labratory A. Verwey [1982]
- Data sheet that were supplied with the molasses, Services [2006]
- Measurements by Chemlab, Christensen [2007]

2.5.2.1 Rule of thumb

The molasses viscosity is alleged halve due to an increase of temperature of 5.5 °Cas a well known rule of thumb.

For the rule of thumb, the data points for 20°C,table 2.2, were used and then new points for every 5.5°Cwere calculated by halving the viscosity. At the end, a spline was fitted on top of these points to generate a smooth curve.

Description	Value	Unit
ν at 20 ℃	20000	cSt
ν at 30 ℃	8000	cSt

Table 2.2: Our specific molasses technical data

2.5.2.2 Given data sheet

The data sheet has to known data points and it is assumed that there is an exponential connection between the data points, equation 2.1.

$$\nu = a \cdot \exp^{-b \cdot T}[cSt] \tag{2.1}$$

By implementing this on the data set, table 2.2,the constants can be found as shown in equation 2.2 to 2.5.

$$\begin{aligned} \nu_1 &= a \cdot \exp^{-b \cdot T_1}[cSt] && (2.2) \\ \nu_2 &= a \cdot \exp^{-b \cdot T_2}[cSt] && (2.3) \end{aligned}$$

$$\begin{aligned} a &= \sqrt[\left(1-\frac{T_2}{T_1}\right)]{\nu_2 \cdot \nu_1^{-\frac{T_2}{T_1}}}[cSt^2] && (2.4) \\ b &= \frac{\ln \frac{a}{\nu_1}}{T_1}[cSt] && (2.5) \end{aligned}$$

From equation 2.7 the value for a can be evaluated and from equation 2.9 b can be found.

$$\begin{aligned} a &= \sqrt[\left(1-\frac{30}{20}\right)]{8000 \cdot 30000^{-\frac{30}{20}}}[cSt^2] && (2.6) \\ a &= 421 \cdot 10^3[cSt^2] && (2.7) \\ b &= \frac{\ln \frac{421 \cdot 10^3}{30000}}{20}[cSt] && (2.8) \\ b &= 0.1322[cSt] && (2.9) \end{aligned}$$

2.5.2.3 Chemlab

Two bottles with samples of the molasses were sent into the ChemlabChristensen [2007] for analyses.

Chemlab provided three data points and a numerical spline curve was fitted to the data.

2.5.2.4 Comparison between the models

The comparison of the three methods for the viscosity of the molasses when shear stress are neglected can be found in figure 2.1. As it can been seen from figure 2.1, the changes of the viscosity of molasses is very dependent of temperature, but for the temperatures around the pump design point, the three models are similar.

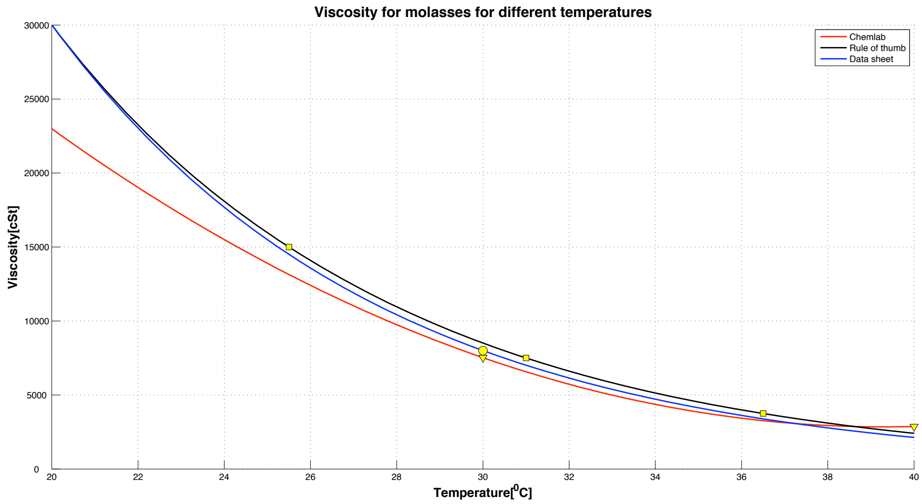


Figure 2.1: Viscosity for the three different viscosity models for molasses.

2.6 Heating of molasses

The recommended handling temperature of molasses is roughly 32 to 38 °C. The maximum advisable handling temperature is 40°C. At temperatures above 40 °Cthe destruction of sugar molecules may take place, which will reduce the feed value of the molasses.

To heat the molasses it is recommended to use a liquid in pipe such as water, at a temperature of 38 °C. Also other methods of heating might be possible as long as uniform temperature is achieved and that the molasses do not exceed the maximum handling temperature.

2.7 Conclusion

The benefits of feeding molasses have been quantified by many research trials and while it was almost always beneficial the actual response varied depending on a number of factors, including animal diet, stage of production, and level of consumption. There can be no doubt, however, that molasses is a great source of energy and minerals for ruminants. It can be fed in a number of ways and is very beneficial in many situations.

Molasses is supposed to be a non-Newtonian fluid, but for normal handling temperature it is assumed to behave like a Newtonian fluid. Therefore the viscosity can be treated as a function of temperature. The three different models have almost identical result for the viscosity around the handling temperature.

At lower temperature it seems that the Frank Mohn's molasses has a lower viscosity. This could be because of the non-Newtonian behavior or slightly different structure of the molasses.

3 Gear pump

Selecting correct pump for pumping molasses is not obvious, at certain combinations of flow and pressure centrifugal pump are inherently inefficient, A positive displacement pump, PD pump, by contrast, is very well suited for low flow conditions. Centrifugal pumps, by contrast, tend to do very well in high flow conditions. $60[m^3/h]$ is a low flow condition, the high viscosity and the foam on top of the molasses make a PD pump a clear choice for specifically pump for molasses, pum [2007b,a], White [1994], Chapple [2002], Phd. Jaroslav Ivantysyn [1993]

3.1 Internal gear pump

Internal gear pumps, figure 3.1 carry fluid between the gear teeth from the inlet to outlet ports. The outer gear drives the pinion on a stationary shaft. The gears create voids as they come out of mesh and liquid flows into the cavities. As the gears come back into mesh, the volume is reduced and the liquid is forced out at the outlet. The fluid seperator, part 4 in figure 3.1 prevents liquid from flowing backwards from the outlet to the inlet port.

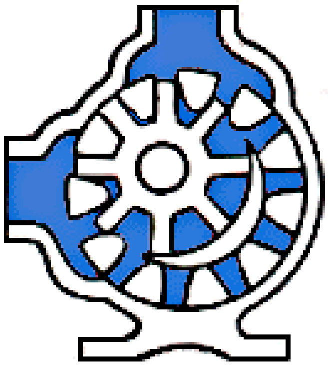


Figure 3.1: Internal gear pump, courtesy of pum [2007b]

3.1.1 Bearings

Depending on shaft sealing arrangements, the pinion shaft support bearings may run in the pumped liquid. This is an important consideration when handling molasses and can wear out the support bearing. The gear is in most cases installed on an overhang shaft and might lead to very large forces on the gear shaft bearing. The choice of bearing will therefore be critical regarding fatigue.

3.1.2 Size

An internal gear pump is physically small and is more or less symmetric around the gear shaft, which means that the gear can be made large and still fit inside the support rings and the trunk. This will then allow the internal gear pump run at a lower speed or that it can be made with less thickness than an external gear pump and still displace the same amount of liquid.

3.2 External gear pump

External gear pumps, figure 3.2 use gears which come in and out of mesh. As the teeth come out of mesh, liquid flows into the pump and is carried between the teeth and the casing to the discharge side of the pump. The teeth come back into mesh and the liquid is forced out the discharge port. External gear pumps rotate two gears against each other, which can be identically but it is not necessary. Both gears are on a shaft with bearings on both sides of the gears.

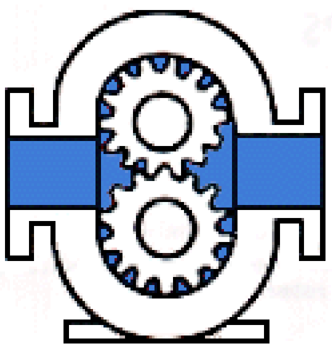


Figure 3.2: External gear pump, courtesy of pum [2007b]

3.2.1 Bearings

Each gear is supported by a shaft with bearings on both sides of each gear. Typically, all four bearings operate in the pumped liquid. Because the gears are supported on both sides, external gear pumps are used for high pressure applications such as hydraulics. An external gear pump will therefore handle larger shaft loadings than an equivalent internal gear pump with the same shaft dimension.

3.2.2 Size

An external gear pump is larger than an internal gear pump. This might lead to that it needs to run on a higher speed than an equivalent internal gear pump.

3.3 Conclusion

Since molasses viscosity is really dependent on temperature, it is desirable to use a pump that can handle both high viscous fluids as well as thinner liquids. A lobe pump is also able to handle this very well, but the problem with a lobe pump is the need of timed gear, Phd. Jaroslav Ivantysyn [1993]. This will make a gear pump a natural choice instead of a lobe pump.

At current stage there is no clear choice between internal and external gear pump for which is the most suitable for handling molasses.

An external gear pump has one bearing on each side of the shaft, while an internal only has bearings on one side of the shaft. This might be a problem if the shafts loadings turns out to be large. The design of external gear pumps allows them to be made to closer tolerances than internal gear pumps, this might not be an issue but its worth to take into account.

The internal gear pump might run on lower speed due to that it can be made physically smaller, see chapter 6.

4 Basic gearing dimension

4.1 Introduction

Before the molasses pump could be modeled up in Pro Engineer, it was necessary to investigate basic gearing dimension in order to select a proper gearing set for the pump. The gearing dimension needs to fit the requirements given in chapter 6.

A gear is a component within a transmission device that transmits rotational force to another gear or device. A gear is different from a pulley in that a gear is a round wheel which has linkages ("teeth" or "cogs") that mesh with other gear teeth, allowing force to be fully transferred without slippage. Depending on their construction and arrangement, geared devices can transmit forces at different speeds, torques, or in a different direction, from the power source. The most common situation is for a gear to mesh with another gear, but a gear can mesh with any device having compatible teeth, such as linear moving racks. A gear's most important feature is that gears of unequal sizes (diameters) can be combined to produce a mechanical advantage, so that the rotational speed and torque of the second gear are different from that of the first. This report will explain some of the basically parameters for a spur gearing with straight and helical toothing. This report can be used as a documentation for the software MITcalc, mit [2008] or similar software.

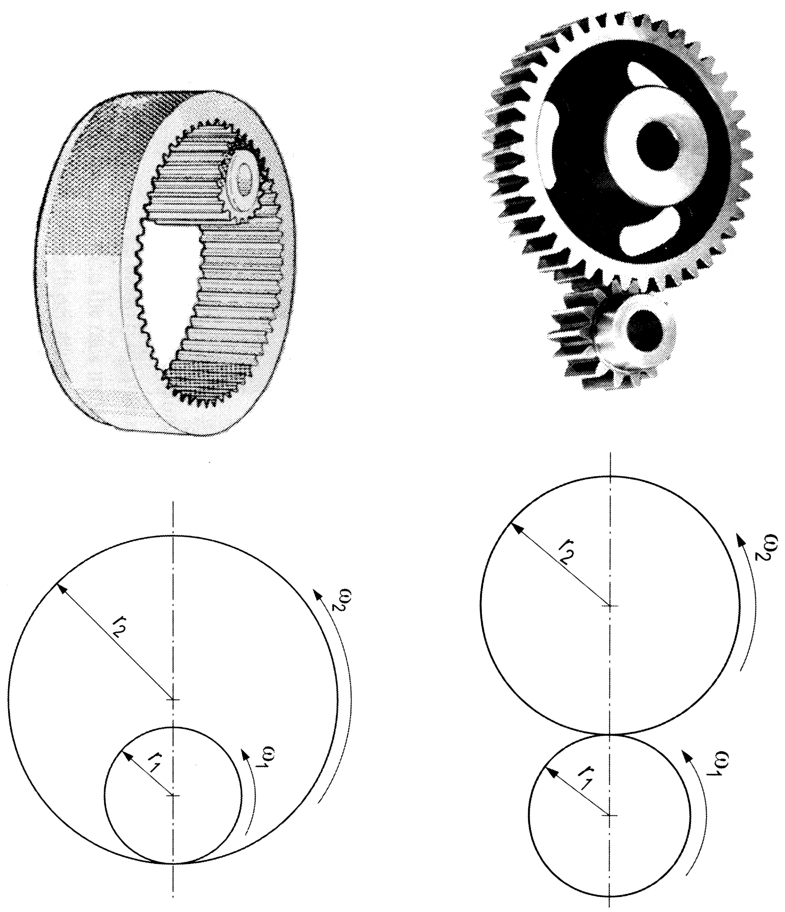
4.2 Types of gears

4.2.1 Spur gears

As mentioned in chapter 3 there are both internal and external gear pumps, and it is therefore also two different types of spur gears as illustrated in figure 4.1.

- Internal gear - rotating the same way

- External gear - rotating opposite way



(a) Internal spur gear (b) External spur gear

Figure 4.1: Spur gears, courtesy of Herkegard [2002]

4.2.2 Straight toothing

Spur gears with straight toothing, illustrated in figure 4.2, have the teeth parallel to the axis of rotation and are used to transmit motion from one shaft to another, parallel shaft.

Of all types, the straight gearing is the simplest and for this reason will be used to develop the primary kinematic relationship of the tooth form.

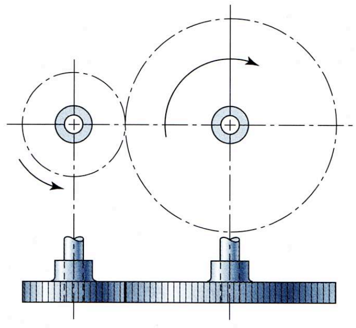


Figure 4.2: Straight toothing, courtesy of Shigley et al. [2003].

4.2.3 Helical toothing

Helical gears, shown in figure 4.3 have the teeth inclined to the axis of rotation. Helical gears can be used for the same applications as straight gears and, when so used, are not as noisy, due to the more gradual engagement of the teeth during meshing. The inclined tooth also generates thrust loads and bending moments, which are not present with a straight gearing. In some cases the helical gears can be used to transmit motion between non parallel shafts.

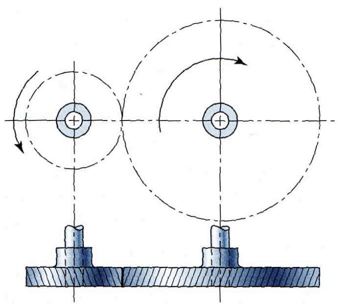


Figure 4.3: Helical toothing,courtesy of Shigley et al. [2003].

4.3 Design of a module geometry of toothing

4.3.1 Transmission ratio

A gearing is a mechanism where at least two gears are in mesh, where mechanical work from the input shaft is transferred to the output shaft with a given transmission rate, as seen in equation 4.1.

$$i = \frac{\omega_{in}}{\omega_{out}} [-] \tag{4.1}$$

This can also be evaluated out from the number of teeth or the diameters as shown in equation

$$i = \frac{z_G}{z_P} \tag{4.2}$$

4.3.2 Teeth number

The gear has along the circumsphere an evenly spaced number of teeth, z.

A generally applicable rule states that increasing the number of teeth (with the same axis distance) leads to:

- increase in loading capacity of the surface (contact, seizure, wearing)
- improvement in the gearing coefficient
- decrease in loading capacity in bend
- reduction in production costs

Recommended values:

1. For both gears annealed normalisationally/improved by heat - soft gears
    - Straight toothing, helical toothing, lower output power, 15 to 30 teeth.
    - Helical toothing, higher output powers, 20 to 40 teeth.
2. For a hardened pinion and non-hardened gear (or both gears nitrided)
    - Straight toothing, helical toothing, lower output powers, 15 to 35 teeth.
    - Helical toothing, higher output powers, 18 to 40 teeth.
3. Both gears surface hardened

- Straight toothing, helical toothing, lower output power, 10 to 30 teeth.
- Helical toothing, higher output powers, 15 to 30 teeth.

The rule is that higher numbers of teeth are chosen for higher output powers and lower transmission ratios.

4.3.3 Normal pressure angle

This angle determines parameters of the basic profile and is standardized to an angle of 20°. Changes in the pressure angle, α affect functional and strength properties. Changes in the meshing angle, however, require non-standard production tools. In case there is no special need to use another meshing angle, use the value of 20°.

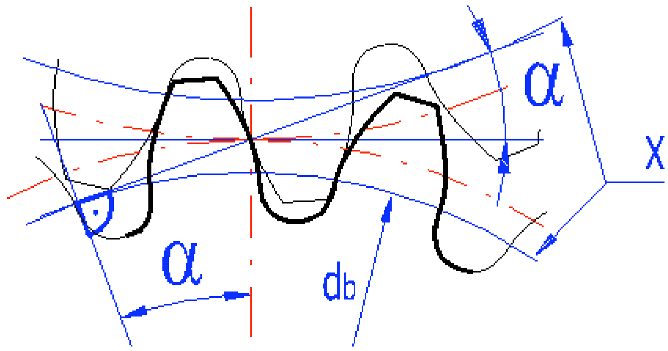


Figure 4.4: Pressure angle, the letter X marks the base circle, courtesy of MIT [2007].

Increasing the meshing angle allows:

- reduction in the danger of undercutting and interference
- to reduce slipping speeds
- increased loading capacity in contact, seizure and wear
- increased rigidity of the toothing
- increased noise and radial forces

Option of values

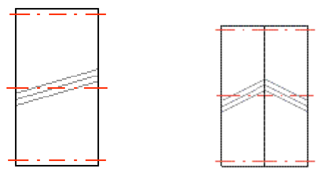
- Straight toothing with increased loading capacity requirement - 25 to 28°
- Helical toothing - up to 25°
- Gearing with a special requirement for quietness - 15 to 17.5°

Recommended values: In case you do not have any special requirements for the designed gearing, it is recommended to use 20°.

4.3.4 Base helix angle

Toothing with the slope of teeth = 0° (straight toothing) is used with slow speed and highly loaded gearing. With high speed gearing, where catching of axial forces could be difficult and where increased noise does not cause any problems. Toothing with the slope of teeth > 0° (helical toothing) is used with high speed gearing; it is characterized by lower noise and higher loading capacity, enabling the use of a lower number of teeth without undercutting.

Recommended values The angle beta is chosen from the sequence 6,8,10,12,15,20°, figure 4.5(a), In case of a double or herringbone gear ,figure 4.5(b), values 25,30,35,40°can also be used.



(a) Single helix gear

(b) Double helix gear

Figure 4.5: Helix angle,, courtesy of MIT [2007].

4.3.5 Normal backlash

It is necessary to make tangential clearance between the unloaded face of the driven teeth in mesh to the unloaded face of the next driven teeth. This tangential clearance can be made by making teeth thickness at the pitch diameter smaller and the width between the cogs larger.

A backlash is necessary to create a continuous layer of lubricant on sides of teeth and to overcome production inaccuracies, deformations and thermal expansion of individual elements of the mechanism. Very small clearances are required in gearing of control systems and instruments and if it is not possible to eliminate it, gearing with automatic take up of backlash is usually used. Great backlash must be chosen with heavily loaded gearing (thermal expansion) and high-speed gearing (hydraulic resistance and shocks with pushing of oil off the inter-tooth gaps.

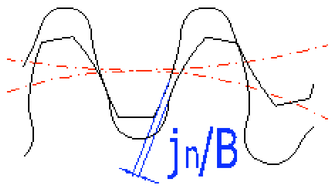


Figure 4.6: Normal backlash, courtesy of MIT [2007].

It is normal to select the tangential clearance by equation 4.3.

$$j \leq 0.1 \cdot m[mm] \tag{4.3}$$

4.3.6 Width of gear

In axial direction is the gear is limited by two parallel planes, that are normal cylinder axis. The distance between these planes is the the teeth width.

The teeth width can also be evaluated as a function of the module, by implementing the factor λ, which is the recommended maximum teeth width, equation 4.4.

$$w \leq \lambda \cdot m[mm] \tag{4.4}$$

For roughly made gears the $\lambda = 6$ and for high precision gears the $\lambda = 30$.

Width of toothing, w of individual gears is measured on a pitch cylinder. Width of toothing of the pinion is usually greater than the width of the gear by the size of one module.

4.3.7 Working face width

Working face width. This is a common width of both gears on rolling cylinders. If the gears are not in offset positions, figure 4.7, it is mostly the width of the gear. This width is used for strength checks of toothing. In case the check box in this row is enabled, the "Working width of toothing" is automatically with the lower value of the width of toothing.

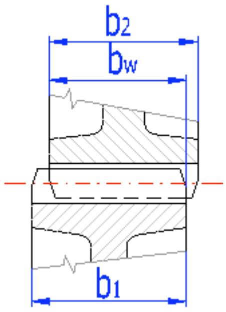


Figure 4.7: Working face width,courtesy of MIT [2007].

4.3.8 Module

If two gears are going to work together they need to have the same module.

This is the most important parameter, which determines the size of the tooth and thereby the gearing itself. It is generally applicable that for a higher number of teeth it is possible to use a smaller module and vice verse.

Recommended values: The module is normally picked from an international standard table of modules, see table 4.1.

Module [mm]	1	1.25	1.5	2	2.5	3	4	5
	6	8	12	14	16	18	20	25

Table 4.1: International standard modules

4.4 Basic gearing dimensions

There are several basic gearing dimension that describes the geometry of the gearing as shown in illustration 4.8.

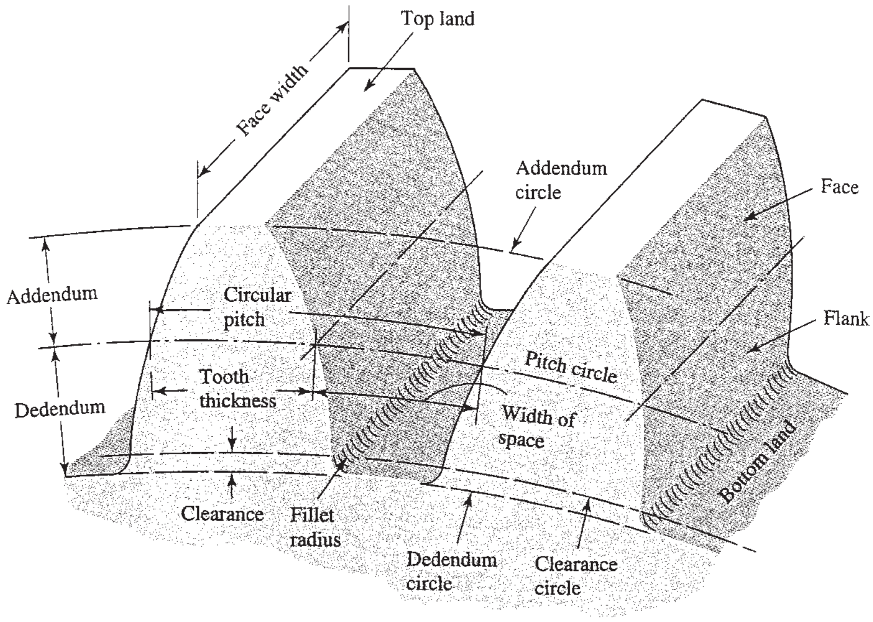


Figure 4.8: Basic gearing dimensions, courtesy of MIT [2007].

4.4.1 Pitch diameter

Equal to twice the perpendicular distance from the axis to the pitch point, equation 4.5. The nominal gear size is usually the pitch diameter.

$$d = m \cdot z[mm] \tag{4.5}$$

4.4.2 Addendum

The radial distance from the pitch surface to the outermost point of the tooth, equation 4.6.

$$h_a = \frac{d_a - d}{2}[mm] \tag{4.6}$$

For normal gear the addendum is set in equation 4.7.

$$h_a = m[mm] \tag{4.7}$$

4.4.3 Dedendum

The radial distance from the depth of the tooth trough to the pitch surface, equation 4.8.

$$h_f = \frac{d - d_f}{2}[mm] \tag{4.8}$$

By choosing the dedendum greater than the addendum will there be a radial clearance, equation 4.9.

$$c = h_f - h_a[mm] \tag{4.9}$$

The radial clearance is normally set to be, equation 4.10.

$$c = 0.25 \cdot m[mm] \tag{4.10}$$

For normal gear the addendum is set in equation 4.11.

$$h_f = 1.25 \cdot m[mm] \tag{4.11}$$

4.4.4 Teeth height

Teeth height is the total depth of a tooth space, equal to addendum plus dedendum, also equal to working depth plus clearance, equation 4.12.

$$h = h_f + h_a[mm] \tag{4.12}$$

For normal gear this will then be, equation 4.13.

$$h = 2.25 \cdot m[mm] \tag{4.13}$$

4.4.5 Outside diameter

Diameter of the gear, measured from the tops of the teeth, equation 4.14.

$$d_a = d + 2 \cdot h_a[mm] \tag{4.14}$$

4.4.6 Root diameter

Diameter of the gear, measured from the base of the tooth space, equation 4.15.

$$d_f = d - 2 \cdot h_f[mm] \tag{4.15}$$

4.4.7 Circular pitch

The distance from one face of a tooth to the corresponding face of an adjacent tooth on the same gear, measured along the pitch circle, equation 4.16.

$$p = \pi \cdot m[mm] \tag{4.16}$$

4.4.8 Base pitch

The distance from one face of a tooth to the corresponding face of an adjacent tooth on the same gear, measured along the base circle, equation 4.17.

$$p_b = m \cdot \pi \cdot \cos\alpha[mm] \tag{4.17}$$

4.4.9 Width of space between teeth

Pitch thickness is the distance from the face on teeth to the face on the next teeth by equation 4.18.

$$e = \frac{p + j}{2}[mm] \tag{4.18}$$

4.4.10 Pitch thickness

Pitch thickness is the teeth thickness at the pitch diameter and is given by equation 4.19.

$$s = \frac{p - j}{2}[mm] \tag{4.19}$$

4.5 Correction of toothing

4.5.1 Principle of corrections, use of corrections

Approaching and withdrawal of the production tool from the gear center changes the shapes and therefore also properties of the involute toothing. This creates corrected toothing. The illustration 4.9 shows:

A. Production tool

B. Produced gear

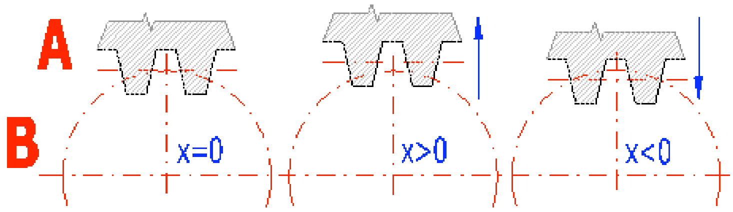


Figure 4.9: Correction of toothing, courtesy of MIT [2007].

Correction of toothing enables:

- Achieve the exact axis distance.
- Prevent undercutting of teeth (roots of a small number of teeth might be undercut; this decreases the coefficient of duration of the meshing and reduces loading capacity of the teeth)
- Eliminate sharpness of teeth Prevent creation of production and operational interference of teeth
- Improve the contact ratio (achieve a contact ratio >1) Reduce noise and vibrations of the gearing
- Improve efficiency
- Increase loading capacity of the gearing (contact, bend, seizure, wear)

The shift affects geometric and kinematic and strength characteristics as well. When designing corrections, first it is necessary to fulfill functional requirements and then optimize the corrections to improve some of the other parameters of the gearing.

Example of a tooth profile (z=10, a=20;b=0), where at X=0 the teeth are undercut and the value x=0.7 causes sharpness of teeth is shown in figure 4.10.

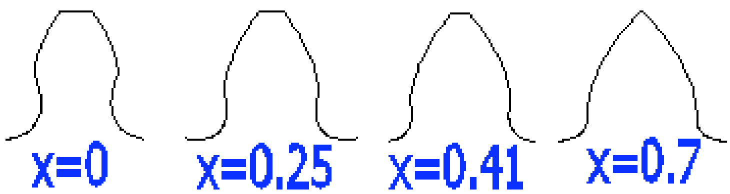


Figure 4.10: Effects of changing the correction factor,X, courtesy of MIT [2007].

Recommended values - optimization When determining values of corrections, first it is necessary to fulfill functional requirements for toothing, where the most important items include:

- Desired axis distance (given sum of both corrections)
- Elimination of teeth undercutting
- Elimination of teeth sharpness

4.5.2 Eliminate undercutting of teeth

Undercutting, figure 4.11 is a phenomena that is desirable to avoid to eliminate fatigue issues.

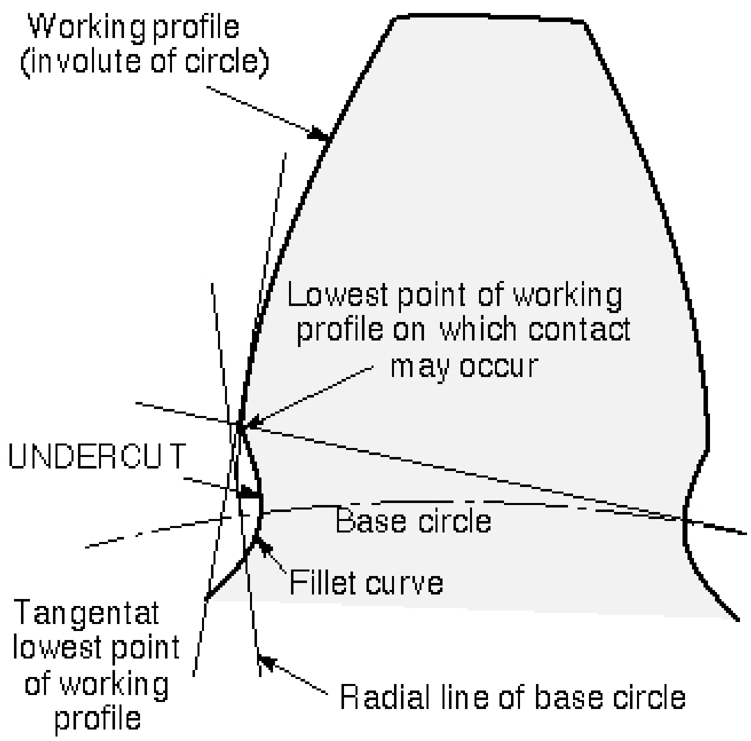


Figure 4.11: Effects of changing the correction factor,X, courtesy of MIT [2007].

Elimination of undercutting can be done by offsetting the tool away from the center, equation 4.20.

$$x_L = x \cdot m[mm] \tag{4.20}$$

The minimum factor x is given in equation 4.21.

$$x = 1 - \frac{z}{z_c}[-] \tag{4.21}$$

Where z_c is given by equation 4.22.

$$z_c = ceil\left(\frac{2}{\sin^2 \alpha}\right)[-] \tag{4.22}$$

4.5.3 Permissible undercutting of teeth

Permissible undercutting of teeth is normally done by allowing a smaller number of z_c as shown in equation can be done by offsetting the tool away from the center, equation 4.20.

$$z_{cu} = ceil\left(\frac{10}{6 \cdot \sin^2 \alpha}\right)[-] \tag{4.23}$$

The minimum factor x_p is given in equation 4.24.

$$x_p = 1 - \frac{z}{z_{cu}}[-] \tag{4.24}$$

4.6 Performance of gearing

4.6.1 Contact ratio

For smooth meshing of gears , it is necessary that the other pair of teeth enters in meshing before the first pair is released, equation 4.25.

$$e_\alpha = \frac{\sqrt{\left(\frac{z+2}{\cos \alpha}\right)^2 - z^2}}{2 \cdot \pi}[-] \tag{4.25}$$

4.6.2 Total contact ratio

The length of action is illustrated in figure 4.12.

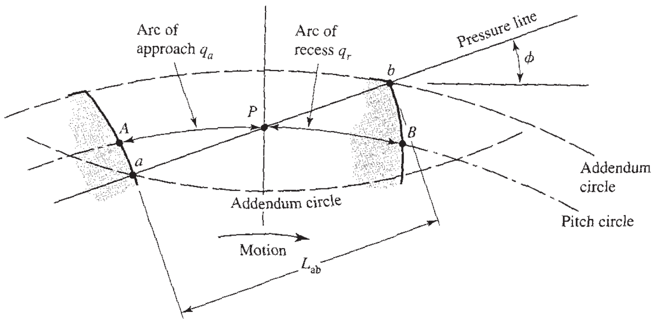


Figure 4.12: Contact ratio, courtesy of Shigley et al. [2003].

The term total contact ratio is defined by equation 4.26.

$$e = \frac{L_{ab}}{p_b}[-] \tag{4.26}$$

The total contact ratio is the sum of the contact ratio for the two meshing gears, equation 4.27.

$$e = \frac{\sqrt{\left(\frac{z_P+2}{\cos\alpha}\right)^2 - z_P^2}}{2\cdot\pi} + \frac{\sqrt{\left(\frac{z_G+2}{\cos\alpha}\right)^2 - z_G^2}}{2\cdot\pi}[-] \tag{4.27}$$

If the value $e = 1$, is the limit case when only one pair of teeth is in meshing at once. In case of $e = 2$, then two teeth are in meshing simultaneous. In case $1 < e < 2$, the meshing will include partly one pair of teeth and partly two pair of teeth.

4.7 Conclusion

It is necessary to know some basic gearing dimension, in order to obtain geometrical and performance data for gears that can be used in a gear pump. Software's like MitCalc, will be able to generate the involute coordinates, but the user still needs to know how the different basic gearing parameters affects the performance and the design of the gearing.

The module and number of teeth is the most important basic gearing parameters, because these parameters affect most of the other parameters. By varying these parameters it is possible to come up with many suggestions that might fulfill the limitations of the pump, see chapter 6. It is therefore necessary to look into how these parameters affect other parameters than just the basic gearing dimensions.

5 Theoretical volume flow

5.1 Introduction

A Positive displacement pump (PD-pump) is moving a given volume from the inlet to the outlet when gears comes into mesh. The leakage flow is dependent of the differential pressure and the internal gaps in the PD-pump. In figure 5.1 is a typically pump curve for PD-pump and centrifugal pump represented.

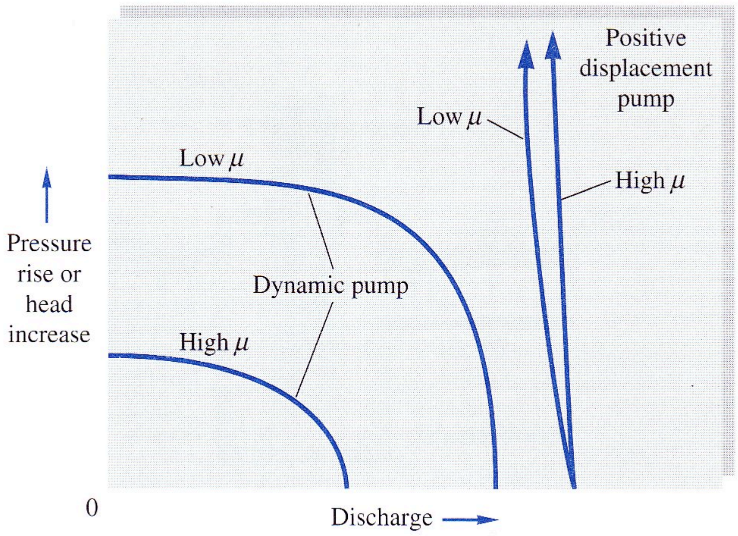


Figure 5.1: Pump curves, courtesy of White [1994].

As seen in figure 5.1 the volume flow of a PD-pump is in many cases almost independent of the differential pressure, which leads to that the formulas in this chapter can be used as a good guideline for designing a new pump when the assumption of high volumetric efficiency is valid.

5.2 Area consideration

If there is no leakage in the pump, the volume flow should only be dependent on the speed of the pump and the volume it moves from the inlet to the outlet as shown in figure 5.2.

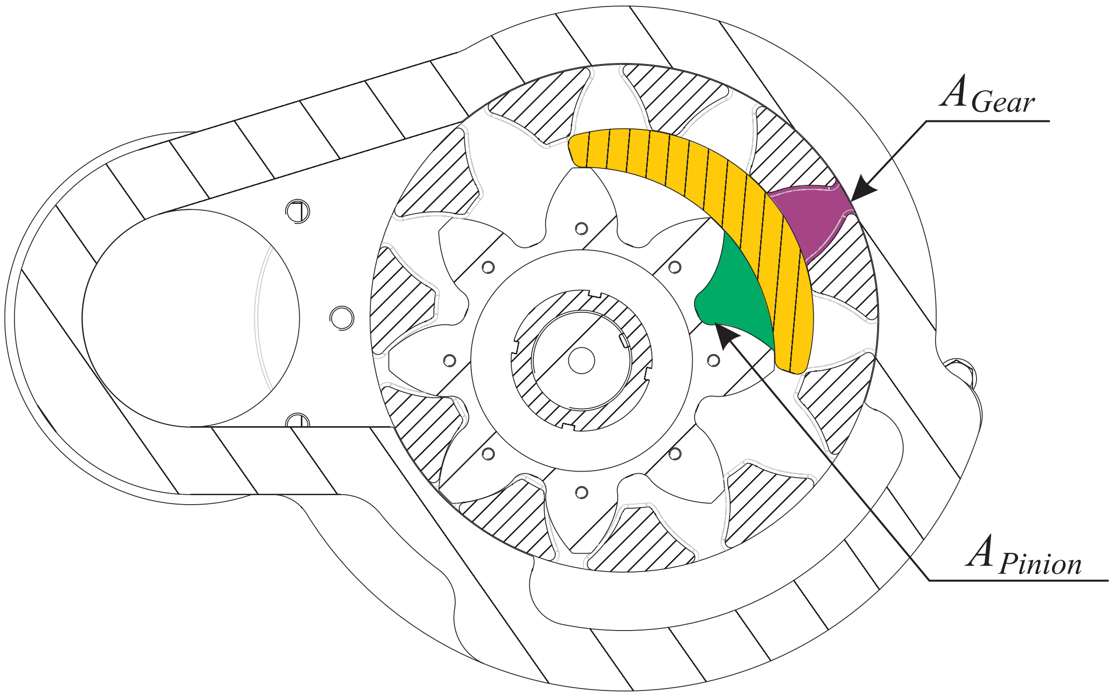


Figure 5.2: Area that moves from inlet to outlet

The volume flow can then be described by equations 5.1-5.3.

$$
\begin{aligned}
A_{tot} &= A_{Gear} + A_{Pinion}[m^2] && (5.1)\\
V_{tot} &= A_{tot} \cdot min(h_{Gear},\ h_{Pinion})[m^3] && (5.2)\\
Q_{avg} &= \frac{V_{tot} \cdot z_{Pinion} \cdot N_{Pinion}}{60}[m^3/s] && (5.3)
\end{aligned}
$$

In equation 5.3 the values for the gear can be used instead as well by using equation 5.4.

$$N_{Pinion} = \frac{N_{Gear} \cdot z_{Gear}}{z_{Pinion}} \tag{5.4}$$

5.3 Spur gearing

The volume flow can be calculated theoretically when a standard spur gear, see chapter4, is used in the PD-pump. The formulas are valid for both internal and external gear pumps, Phd. Jaroslav Ivantysyn [1993].

5.3.1 Work consideration

The theoretical volume flow for PD-pump is only dependent of revolution and geometrical data. If the flow is incompressible and rotational free, then the work is given by the displaced volume and the pressure, as shown in equation 5.5.

$$dW = dV_a \cdot \Delta p \tag{5.5}$$

The work that is transferred to the pump is given by equation 5.6

$$dW = M \cdot d\varphi \tag{5.6}$$

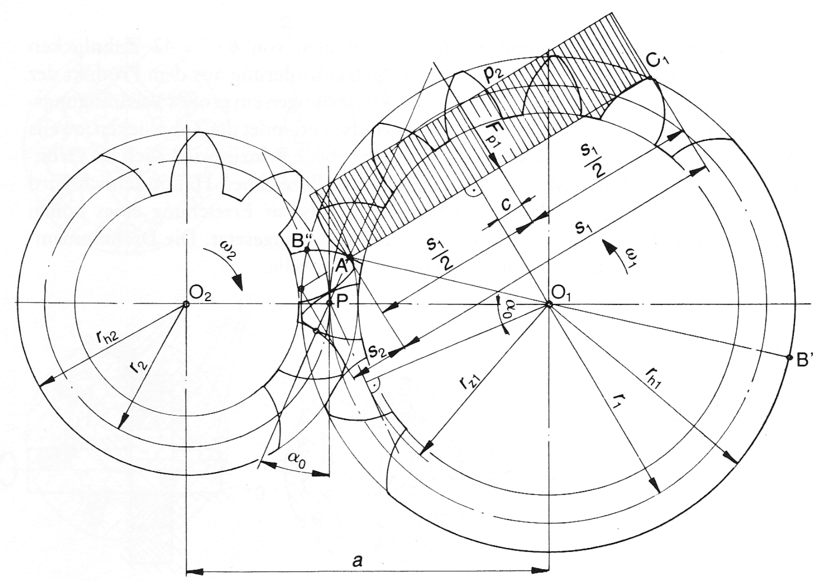


Figure 5.3: Figure to calculate the geometrical volume flow, courtesy of Phd. Jaroslav Ivantysyn [1993].

When a change in the volume V_a by both cogs splits, as shown in figure 5.3 then the work can be expressed as the sum of the work for each gear, as shown in equation 5.7.

$$dW = M_1 \cdot d\varphi_1 + M_2 \cdot d\varphi_2 \tag{5.7}$$

5.3.2 Moment consideration

In the next deduction only one of the gears are taken in to account, since the same rules also applies for gear 2. On gear 1 works the moment given by equation 5.8

$$M_1 = F_{p1} \cdot c \tag{5.8}$$

This result in the following force, equation 5.9

$$F_{p1} = p_2 \cdot A = p_2 \cdot s_1 \cdot b \tag{5.9}$$

The moment is then given by, equation 5.10

$$M_1 = p_2 \cdot s_1 \cdot b \cdot c \tag{5.10}$$

From figure 5.3, it can be seen that equation5.11 is correct.

$$c = \frac{1}{2} \cdot s_2 \tag{5.11}$$

If equation 5.11 is substituted into equation 5.10 the moment became, equation 5.12

$$M_1 = p_2 \cdot s_1 \cdot b \cdot \frac{s_2}{2} \tag{5.12}$$

$$s_1 \cdot s_2 = \overline{AB}' \cdot \overline{AB}" \tag{5.13}$$

$$s_1 \cdot s_2 = (r_{h1} + \overline{O_1A}) \cdot (r_{h1} - \overline{O_1A}) \tag{5.14}$$

$$s_1 \cdot s_2 = r_{h1}^2 - \overline{O_1A}^2 \tag{5.15}$$

$$M_1 = \frac{p_2 \cdot b}{2}(r_{h1}^2 - \overline{O_1A}^2) \tag{5.16}$$

And analog for the second gear as well

$$M_2 = \frac{p_2 \cdot b}{2}(r_{h2}^2 - \overline{O_2A}^2) \tag{5.17}$$

The angular speed is given by equation 5.18.

$$\omega = \frac{d\varphi}{dt} \tag{5.18}$$

From equation 5.18 can the finite angle rotation be found as shown in equation 5.19 and 5.20

$$d\varphi_1 = \omega_1 \cdot dt \tag{5.19}$$

$$d\varphi_2 = \omega_2 \cdot dt \tag{5.20}$$

The two speeds are directly connected either through the radius or through the numbers of cogs, as shown in equation 5.21.

$$\omega_2 = \omega_1 \cdot \frac{r_1}{r_2} \tag{5.21}$$

The volume flow can be approximated by using equation 5.22.

$$\begin{aligned}
Q_{avg} &= h \cdot \pi \cdot n \cdot \left[r_{hG}^2 + \frac{r_G}{r_P} \cdot r_{hP}^2 \right. \\
&\left. -r_G \cdot (r_G + r_P) - \left(1 + \frac{r_G}{r_P}\right) \cdot \frac{t_0^2}{12} \right] [m^3/s] \qquad (5.22)
\end{aligned}$$

The different parameters are given in equations 5.23-5.28.

$$t_0 = m \cdot \pi \cdot cos\,(\alpha_0)\,[m] \qquad (5.23)$$

$$r_G = m \cdot \frac{z_G}{2}[m] \qquad (5.24)$$

$$r_P = m \cdot \frac{z_P}{2}[m] \qquad (5.25)$$

$$r_{hG} = m \cdot \left(\frac{z_G}{2} + 1\right)[m] \qquad (5.26)$$

$$r_{hP} = m \cdot \left(\frac{z_P}{2} + 1\right)[m] \qquad (5.27)$$

$$n = \frac{rpm}{60}[rps] \qquad (5.28)$$

5.4 Involute gearing

Equation 5.22 can be simplified by implementing the standard gearing dimensioning,see chapter4,

Equations 5.29-5.34 describe how the volume flow can be calculated when an involute teeth shape is chosen.

$$
\begin{aligned}
Q_{avg} &= h \cdot \pi \cdot n \cdot \left[r_{hG}^2 + \frac{z_G}{z_P} \cdot r_{hP}^2 - r_G \cdot (r_G + r_P) \right. \\
&\qquad \left. - \left(1 + \frac{z_G}{z_P}\right) \cdot \frac{t_0^2}{12} \right] [m^3/s] && (5.29) \\
Q_{avg} &= h \cdot \pi \cdot n \cdot \left[m^2 \cdot \left(\frac{z_G^2}{4} + z_G + 1 \right) \right. \\
&\qquad + \frac{z_G}{z_P} \cdot m^2 \cdot \left(\frac{z_P^2}{4} + z_P + 1 \right) - \frac{m^2 \cdot z_G}{4} \cdot (z_G + z_P) \\
&\qquad \left. - \left(1 + \frac{z_G}{z_P}\right) \cdot \frac{m^2 \cdot \pi^2 \cdot \cos^2(\alpha_0)}{12} \right] [m^3/s] && (5.30) \\
Q_{avg} &= h \cdot \pi \cdot n \cdot m^2 \cdot \left[\frac{z_G^2}{4} + z_G + 1 + \frac{z_G}{z_P} \cdot \left(\frac{z_P^2}{4} + z_P + 1 \right) - \right. \\
&\qquad \frac{z_G}{4} \cdot (z_G + z_P) [m^3/s] \\
&\qquad \left. - \left(1 + \frac{z_G}{z_P}\right) \cdot \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12} \right] && (5.31) \\
Q_{avg} &= h \cdot \pi \cdot n \cdot m^2 \cdot \left[\frac{z_G^2}{4} + z_G + 1 + \frac{z_G \cdot z_P}{4} + z_G \right. \\
&\qquad + \frac{z_G}{z_P} - \frac{z_G^2}{4} - \frac{z_G \cdot z_P}{4} - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12} \\
&\qquad \left. - \frac{z_G}{z_P} \cdot \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12} [m^3/s] \right] && (5.32) \\
Q_{avg} &= h \cdot \pi \cdot n \cdot m^2 \cdot \left[2 \cdot z_G + \frac{z_G}{z_P} \cdot \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12} \right) \right. \\
&\qquad \left. + 1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12} \right] [m^3/s] && (5.33) \\
Q_{avg} &= h \cdot \pi \cdot n \cdot m^2 \cdot [2 \cdot z_G \\
&\qquad + \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12} \right) \cdot \left(1 + \frac{z_G}{z_P} \right)] [m^3/s] && (5.34)
\end{aligned}
$$

5.4.1 Estimate

A closer look at equation 5.34 leads to that it can be simplified as shown in equation 5.35.

$$
Q_{avg} = h \cdot \pi \cdot n \cdot m^2 \cdot [2 \cdot z_G + c_1 + c_2] [m^3/s] \qquad (5.35)
$$

The value for c_2 is given in equations 5.36-5.37.

$$c_2 = 1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12}[-] \quad (5.36)$$

$$c_2 = 0.274[-] \quad (5.37)$$

The c_1 part of the equation 5.35 is dependent on the transmission ratio, this can then be combined to one part as shown in equations 5.38-5.39.

$$c_1 = \frac{z_G}{z_P} \cdot \left(1 - \frac{\pi^2 \cdot cos(\alpha_0)^2}{12}\right)[-] \quad (5.38)$$

$$c_1 \approx 0.274 \cdot \frac{z_G}{z_P}[-] \quad (5.39)$$

The constants c_1 and c_2 can be combined into one constant as shown in equations 5.40-5.42.

$$c = c_1 + c_2[-] \quad (5.40)$$

$$c = \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12}\right) \cdot \left(\frac{z_G}{z_P} + 1\right)[-] \quad (5.41)$$

$$c \approx 0.274 \cdot \left(\frac{z_G}{z_P} + 1\right)[-] \quad (5.42)$$

If equation 5.39 and 5.37 is implemented into equation 5.35 then the result is shown in equations **??**-5.44.

$$Q_{avg} = h \cdot \pi \cdot n_G \cdot m^2 \cdot [2 \cdot z_G + \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12}\right) \cdot \left(\frac{z_G}{z_P} + 1\right)\Bigg][m^3/s] \quad (5.43)$$

$$Q_{avg} \approx 2 \cdot h \cdot \pi \cdot n_G \cdot m^2 \cdot \left[z_G + 0.14 \cdot \left(1 + \frac{z_G}{z_P}\right)\right][m^3/s] \quad (5.44)$$

As long as the transmission ratio is not to large $\frac{z_G}{z_P} < 10$ then equation 5.44 can be simplified even further without the error getting to0 large as shown in equation 5.45.

$$Q_{est} \approx 2 \cdot h \cdot \pi \cdot n_G \cdot m^2 \cdot z_G[m^3/s] \quad (5.45)$$

If this is converted into metric units equation becomes the equation given in equation 5.46.

$$Q_{est} \approx 120 \cdot h \cdot \pi \cdot rpm \cdot m^2 \cdot z_G[m^3/h] \quad (5.46)$$

This is a very simple equation where only two basic gearing parameters is included, z_G and m. The volume flow is quadtratic with the module while it is only proportional with the other parameters as shown in equation 5.47-5.50.

$$
\begin{aligned}
Q_{avg} &\propto n && (5.47)\\
Q_{avg} &\propto h && (5.48)\\
Q_{avg} &\propto m^2 && (5.49)\\
Q_{avg} &\propto z_G && (5.50)
\end{aligned}
$$

5.4.2 Constant diameter

The outer diameter of the gearing is given in equation 5.51.

$$D = m \cdot (z + 2) \tag{5.51}$$

If equation 5.51 is substituted into 5.46, a different form of the solution is obtained in equation 5.52.

$$Q_{est} \approx \frac{120 \cdot h \cdot \pi \cdot rpm \cdot D^2 \cdot z_G}{(z_G + 2)^2} [m^3/h] \tag{5.52}$$

If the number of gears is getting large, $z > 30$ equation 5.52 can even be more simplified as shown in equation 5.53

$$Q_{est} \approx \frac{120 \cdot h \cdot \pi \cdot rpm \cdot D^2}{z_G} [m^3/h] \tag{5.53}$$

In figure 5.4 is the equation 5.52 plotted for different number of teeth, to illustrate that the volume flow is drastically increased when number of teeth is reduced.

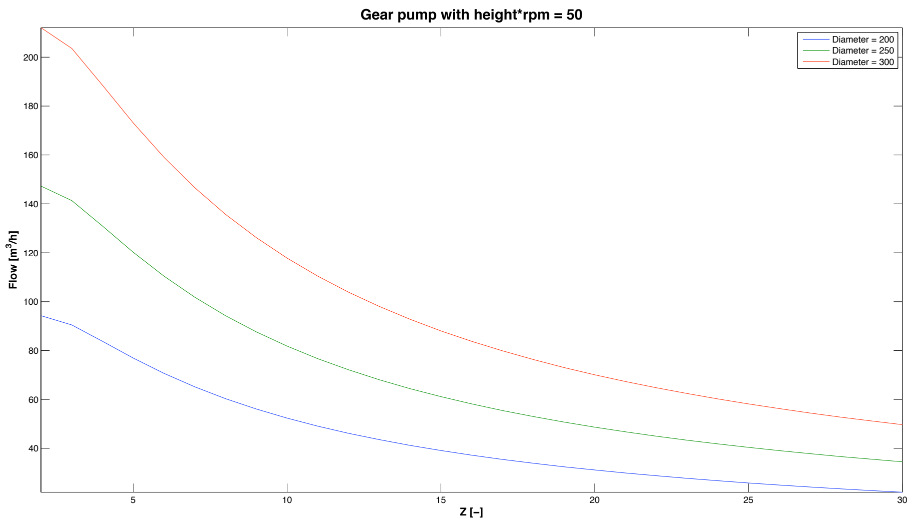


Figure 5.4: Volume flow for different teeth numbers and diameters.

5.4.3 Error

This is a conservative assumption, $Q_{est} < Q_{avg}$ which then will give a lower volume flow than which theoretically is correct.,see equation

In these formulas the actual path of action, axis offset and back lash is not taken into account, these factors will decrease the volume flow and then the estimate will even be closer to the actual flow.

In equation 5.29 should have been as shown in equation 5.54 when the working center distance of the gears are taken into account.

$$\begin{aligned} Q_{avg} &= h \cdot \pi \cdot n \cdot \left[r_{hG}^2 + \frac{z_G}{z_P} \cdot r_{hP}^2 \right. \\ &\left. -r_G \cdot \frac{(r_{kG} + r_{kP})^2}{(r_G + r_P)} - \left(1 + \frac{z_G}{z_P}\right) \cdot \frac{t_0^2}{12} \right] [m^3/s] \end{aligned} \tag{5.54}$$

The value for r_{kG} and r_{kP} can be calculated by using equations 5.55 - 5.58.

$$r_k = \frac{r \cdot \cos(\alpha_0)}{\cos(\alpha_k)} [m] \tag{5.55}$$

$$r_k = \frac{r \cdot \cos(\alpha_0)}{\frac{\cos(\alpha_0) \cdot (r_G + r_P)}{r_G + r_P + x}} [m] \tag{5.56}$$

$$r_{kG} = z_G \cdot \frac{m}{2} + \frac{x}{z_G + z_P} [m] \tag{5.57}$$

$$r_{kP} = z_P \cdot \frac{m}{2} + \frac{x}{z_G + z_P} [m] \tag{5.58}$$

5.4.4 Undercutting

As it might be noticed the undercutting of teeth is not included in these formulas, this is because the undercutting implements a dead volume which always will be transferred back to the inlet as illustrated in figure 5.5.

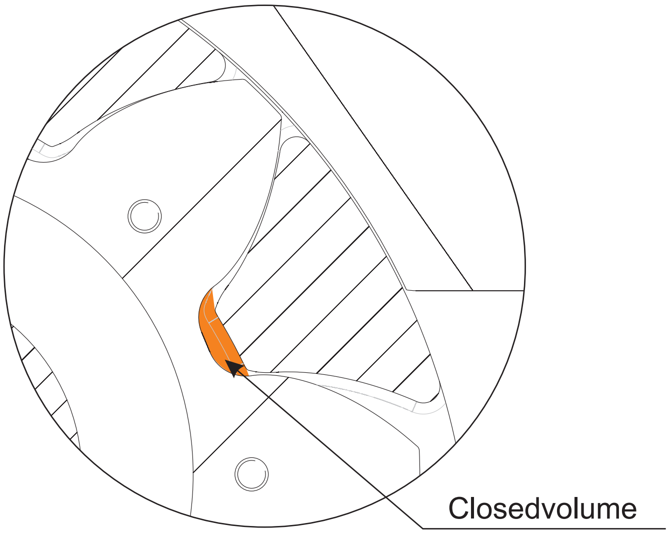
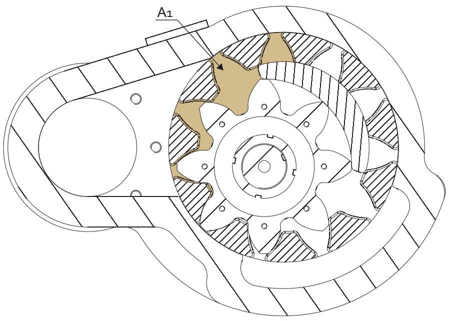


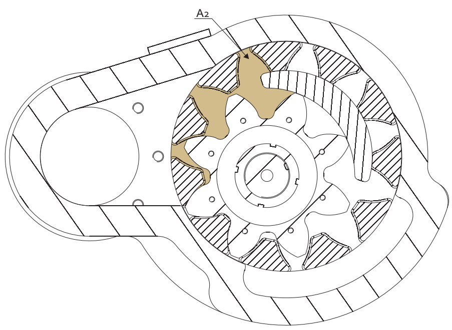
Figure 5.5: Closed volume

5.4.5 Pulsation in flow

As shown in figure 5.6 the gear and pinion will open the liquid filled space between the teeth differently for different angle of rotation.



(a) Area at angle 1



(b) Area at angle 1

Figure 5.6: Opens at different times

This will lead to a pulsation in the flow as illustrated in figure 5.7.

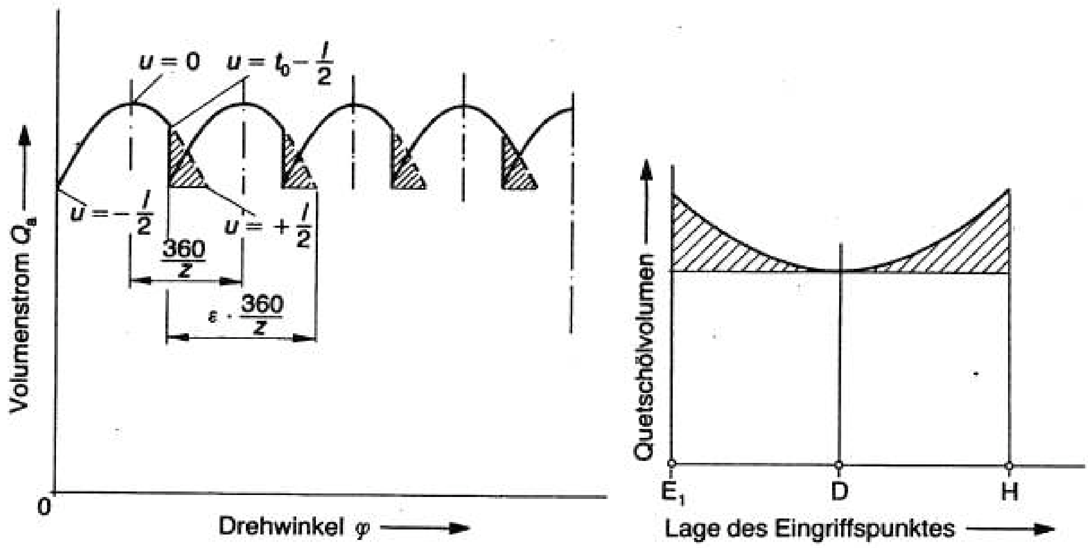


Figure 5.7: Pulsation in flow, courtesy of Phd. Jaroslav Ivantysyn [1993].

5.4.6 Minimum flow

The minimum flow that the pump will produce is given in equations 5.59-5.61.

$$\begin{aligned}
Q_{min} &= h \cdot \pi \cdot n \cdot \left[r_{hG}^2 + \frac{r_G}{r_P} \cdot r_{hP}^2 \right. \\
&\left. -r_G \cdot (r_G + r_P) - \left(1 + \frac{r_G}{r_P}\right) \cdot \frac{t_0^2}{4} \right] [m^3/s] \qquad (5.59) \\
Q_{min} &= h \cdot \pi \cdot n \cdot \left[m^2 \cdot \left(\frac{z_G^2}{4} + z_G + 1 \right) \right. \\
&+ \frac{z_G}{z_P} \cdot m^2 \cdot \left(\frac{z_P^2}{4} + z_P + 1 \right) - \frac{m^2 \cdot z_G}{4} \cdot (z_G + z_P) - \\
&\left. \left(1 + \frac{z_G}{z_P}\right) \cdot \frac{m^2 \cdot \pi^2 \cdot \cos^2(\alpha_0)}{4} \right] [m^3/s] \qquad (5.60) \\
Q_{min} &= h \cdot \pi \cdot n_G \cdot m^2 \cdot \left[2 \cdot z_G + \right. \\
&\left. \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{4}\right) \cdot \left(\frac{z_G}{z_P} + 1\right) \right] [m^3/s] \qquad (5.61)
\end{aligned}$$

5.4.7 Maximum flow

The maximum flow that the pump will produce is given in equations 5.62-5.66.

$$
\begin{aligned}
Q_{max} &= h \cdot \pi \cdot n \cdot \left[r_{hG}^2 + \frac{r_G}{r_P} \cdot r_{hP}^2 - r_G \cdot (r_G + r_P) \right] [m^3/s] && (5.62)\\
Q_{max} &= h \cdot \pi \cdot n \cdot \left[m^2 \cdot \left(\frac{z_G^2}{4} + z_G + 1 \right) \right. && (5.63)\\
&\quad + \frac{z_G}{z_P} \cdot m^2 \cdot \left(\frac{z_P^2}{4} + z_P + 1 \right) && (5.64)\\
&\quad \left. - \frac{m^2 \cdot z_G}{4} \cdot (z_G + z_P) \right] [m^3/s] && (5.65)\\
Q_{max} &= h \cdot \pi \cdot n \cdot m^2 \cdot \left(2 \cdot z_G + 1 + \frac{z_G}{z_P} \right) [m^3/s] && (5.66)
\end{aligned}
$$

5.4.8 Differential in the flow

The difference in the flow will be the difference between the maximum and the minimum flow as shown in equations 5.67-5.69.

$$
\begin{aligned}
\Delta Q &= Q_{max} - Q_{min} [m^3/s] && (5.67)\\
\Delta Q &= h \cdot \pi \cdot n_G \cdot m^2 \cdot \left(2 \cdot z_G + 1 + \frac{z_G}{z_P} \right) \\
&\quad - h \cdot \pi \cdot n_G \cdot m^2 \cdot \left[2 \cdot z_G + \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{4} \right) \cdot \right. \\
&\quad \left. \left(\frac{z_G}{z_P} + 1 \right) \cdot \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{4} \right) \cdot \left(\frac{z_G}{z_P} + 1 \right) \right] [m^3/s] && (5.68)\\
\Delta Q &= h \cdot \pi \cdot n_G \cdot m^2 \cdot \frac{\pi^2 \cdot \cos^2(\alpha_0)}{4} \cdot \left(1 + \frac{z_G}{z_P} \right) [m^3/s] && (5.69)
\end{aligned}
$$

Often it is better to scale this factor according to the average volume flow as shown in equations 5.70-5.73.

$$\delta_Q = \frac{\Delta Q}{Q_{avg}}[-] \quad (5.70)$$

$$\delta_Q = \frac{h \cdot \pi \cdot n_G \cdot m^2 \cdot \frac{\pi^2 \cdot \cos^2(\alpha_0)}{4} \cdot \left(1 + \frac{z_G}{z_P}\right)}{h \cdot \pi \cdot n_G \cdot m^2 \cdot \left[2 \cdot z_G + \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12}\right) \cdot \left(1 + \frac{z_G}{z_P}\right)\right]}[-] \quad (5.71)$$

$$\delta_Q = \frac{\frac{\pi^2 \cdot \cos^2(\alpha_0)}{4} \cdot \left(1 + \frac{z_G}{z_P}\right)}{2 \cdot z_G + \left(1 - \frac{\pi^2 \cdot \cos^2(\alpha_0)}{12}\right) \cdot \left(1 + \frac{z_G}{z_P}\right)}[-] \quad (5.72)$$

$$\delta_Q = \frac{3 \cdot \pi^2 \cdot \cos^2(\alpha_0)}{\frac{24 \cdot z_G}{1 + \frac{z_G}{z_P}} + \left(12 - \pi^2 \cdot \cos^2(\alpha_0)\right)}[-] \quad (5.73)$$

This can be estimated with high accuracy when $\alpha_0 = 20$ as given in equations 5.74-5.75.

$$\delta_Q \approx \frac{\pi^2 \cdot \cos^2(\alpha_0)}{8} \cdot \left(\frac{1}{z_G} + \frac{1}{z_P}\right)[-] \quad (5.74)$$

$$\delta_Q \approx \frac{1}{z_G} + \frac{1}{z_P}[-] \quad (5.75)$$

5.4.9 Speed

The speed that the pump will be running on is then dependent on the leakage flow and the geometry of the pump. The leakage flow is not considered in this paper, due to its complexity and that this normally requires empirical data. Empirical data for leakage flow can easily be found when running the pump against closed valve. Then the leakage flow is normally dependent of speed and pressure which then can be logged and later used to find the leakage flow by using equation 5.76.

$$N_{motor} \approx \frac{Q_{est}}{120 \cdot h \cdot \pi \cdot m^2 \cdot z_G}[rpm] \quad (5.76)$$

The flow in equation 5.77 is given by equation 5.77.

$$Q_{est} = Q + Q_{leakage}[rpm] \quad (5.77)$$

5.5 Conclusion

Measuring the area between the cogs is time consuming task and therefore an analytical formula for the volume flow is very important in an evaluation process of selecting correct height, speed, module and teeth number for a gear pump.

The analytical formula is an complex formula that involves many gearing parameters, but when the assumption of small leakage flow and small error is allowed then the volume flow becomes only dependent of four parameters. Then it is proportional with how many teeth that is chosen by one gear and it is proportional with the speed of that gear and the height of the gear. The most significant factor is however the module, where a doubling in the module will lead to 4 doubling in the flow, but only doubling in the outer diameter of the gear.

The pulsation in the flow might be critical when small numbers of teeth for the different gear is chosen. This might then lead to pressure waves at the outlet that might have impact far away from the outlet as well, which can be critical if the pressure pulsation is getting large.

6 Limitations

There are several limitations that have been set both by the customer, the law and Frank Mohn Fusa A/S. These limitations gives the boundary of the design of a submerged pump for molasses, which reduces the number of unknowns parameters in the aim of developing the most efficient pump for pumping of molasses. It is also therefore expected that the final product will have a lower efficiency than other pump on the market that are used for pumping molasses.

6.1 Customers requirements

The customer have given Framo the requirements given in table 6.1.

Description	Value	Unit
Differential pressure	10	$[Bar]$
Flow	60	$[m^3/h]$
Viscosity	3200	$[cSt]$

Table 6.1: Customer requirements

In addition it is desirable if the viscosity can change without that the other two customer requirements are largely affected by this change.. This is a reasonable requirement as stated in chapter 2. Therefore it is desirable to evaluate the performance of the pump for changes in viscosity.

6.2 Maximum outer diameter

6.2.1 Trunk diameter

The pump is ordered with a standard SD200 trunk, A/S [2006], which therefore, adds a limitation of the maximum diameter of the pump as given in equation 6.1.

$$D_t = 635[mm] \tag{6.1}$$

6.2.2 Support ring

Practically this can not be fully used since the support ring needs to have some material left at its rim, as shown in equation 6.2.

$$t_s = 15[mm] \tag{6.2}$$

6.2.3 Cargo pipe

In addition the outlet needs to be connected to the cargo pipe, this pipe is recommended to be not smaller than the diameter given in equation 6.3,Services [2006].

$$D_{C\ Min} = 150[mm] \tag{6.3}$$

This will only be a limitation for the internal versions, because for the external versions it is possible to move the cargo pipe around.

6.2.4 Offset of shaft

The shaft of the motor has an offset compare to where the high pressure pipe enters, figure 6.1.

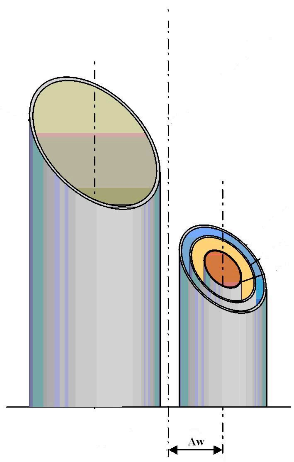


Figure 6.1: Axis distance for the different hydraulic motors.

The offset for the SD200 motor, the motor suitable for the customer requirements,Skåtun [2007] is given in equation 6.4.

$$A_w = 95[mm] \tag{6.4}$$

The motor can be placed in two positions, the motor can be turned 180^0 and still match the holes on the 8 flange. Therefore the shaft of the motor can be aligned with the center axis of the trunk or with an offset given by the equation 6.5.

$$\begin{aligned} A_w &= 2 \cdot A_m \\ A_w &= 190[mm] \end{aligned} \tag{6.5}$$

6.2.5 Housing

In addition the wall of the housing needs to withstand the internal pressure with a given safety factor. The wall thickness can be estimated by using equations for a cylinder with an internal pressure as given in equation 6.6 and 6.7 when the assumption $t_w \ll D_i$ is valid.

$$\sigma_\theta = \frac{r \cdot p_i}{t_w}[Pa] \tag{6.6}$$

$$t_w = \frac{D_p \cdot p_i}{2 \cdot \sigma_{allowed}}[mm] \tag{6.7}$$

As a quick estimate the following values where set,

$$\begin{aligned} p_i &= 20[bar] \\ D_p &= 600[mm] \\ \sigma_{allowed} &= 150[MPa] \end{aligned}$$

This gives the minimum required wall thickness which is given equation 6.8.

$$\begin{aligned} t_w &= \frac{600 \cdot 10^{-3} \cdot 2 \cdot 10^6}{2 \cdot 150 \cdot 10^6}[m] \\ t_w &= 4[mm] \end{aligned} \tag{6.8}$$

This dimension is smaller than the commonly used bolt width of 12$[mm]$,Dalen [2007], the limitations is therefore going to be the bolt diameter. The width have therefore been set to value given by equation 6.9

$$t_w = 15[mm] \tag{6.9}$$

6.2.6 Internal gear pump

The limitations that is valid for the internal gear pump is given in figure 6.2.

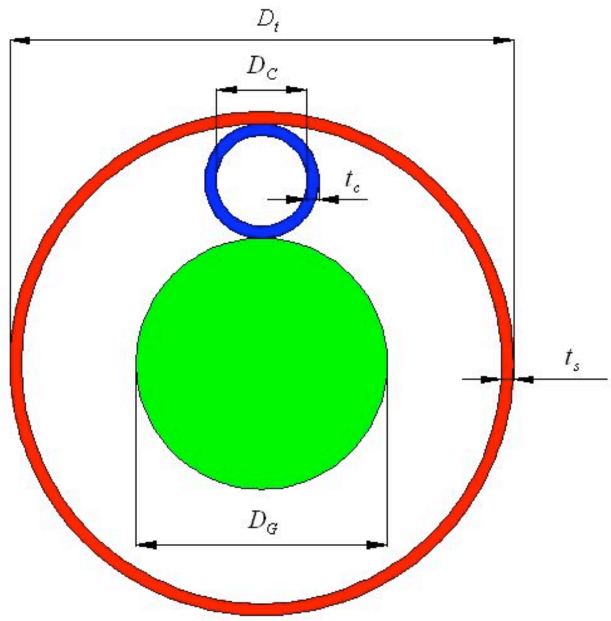


Figure 6.2: Limitations for internal gear pump

Maximum practical outer diameter for the internal version is given in equation 6.10.

$$
\begin{aligned}
D_{gear\ max} &= D_t - 2(t_s + 2 \cdot t_c + D_{C\ Min})[m] \\
D_{gear\ max} &= 316.4[mm]
\end{aligned} \tag{6.10}
$$

6.2.7 External gear pump

For the external version there will be two gears that will affect the maximum outer diameter as shown in figure 6.3. It is assumed that the pinion, the driven gear is smaller than the gear.

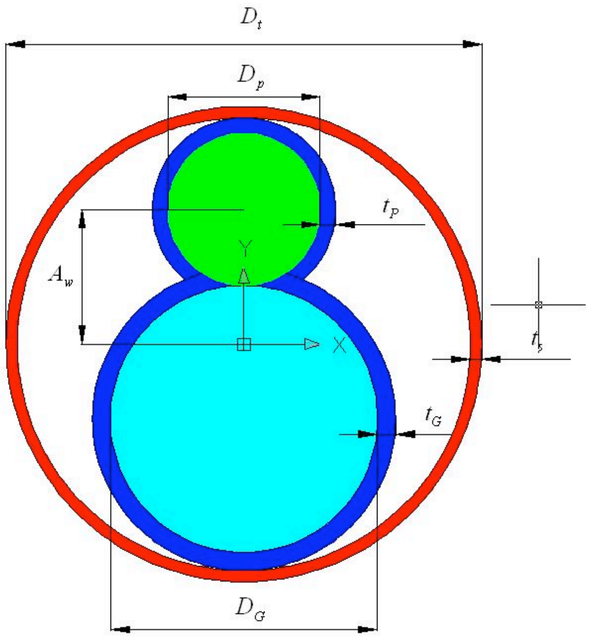


Figure 6.3: Limitations for external gear pump

Maximum practical outer diameter for the external version is given in equation 6.11 and 6.12.

$$\begin{aligned} D_{pinion\ max} &= D_t - 2(t_s + t_P + A_w)[m] \\ D_{pinion\ max} &= 195[mm] \end{aligned} \tag{6.11}$$

$$\begin{aligned} D_{gear\ max} &= D_t - 2 \cdot t_s - t_G - t_P - D_P[m] \\ D_{gear\ max} &= 2 \cdot A_w - t_G + t_P[m] \\ D_{gear\ max} &= 380[mm] \end{aligned} \tag{6.12}$$

This gives a transmission ratio of $i < 1.95$, when the pinion is as large as possible. If the pinion gets smaller the gear diameter can be increased i.e transmission ratio is increased, $i > 1.95$.

6.3 Height of gears

The height of the gear is limited by two factors:

- Machining limitations
- Hydraulic limitations

6.3.1 Machining

Machining of high gear might become a problem if the height of the gear is high combined with small radius-es at the gear. If the gear is molded then this effect might be eliminated.

6.3.2 Hydraulic

The gear needs to be filled with molasses and it is not necessary to make the gears higher than what is possible to fill the empty chamber in the available opening time. The chamber will be empty when it enters the inlet and should ideally be full when it leaves the inlet. It is assumed that when the chamber enters the chamber it will be no air in the chamber, while there will be molasses outside the chamber ready to enter the chamber, differential pressure is therefore assumed to be around $1[bar]$. This is illustrated in figure 6.4.

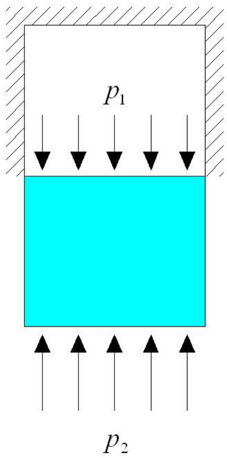


Figure 6.4: Filling of chamber as gears comes in and out of mesh, pressure difference

This can be simplified by using newton second law as shown in equation 6.13-6.16.

$$F = m \cdot a[N] \tag{6.13}$$

$$a = \frac{F}{m}[m^2/s] \tag{6.14}$$

$$a = \frac{\Delta p \cdot A}{m}[m^2/s] \tag{6.15}$$

$$a = \frac{\Delta p}{\rho \cdot h}[m^2/s] \tag{6.16}$$

If this is combined with the assumption that the filling of the volume happens under constant acceleration and that it ends with the velocity v, figure 6.5.

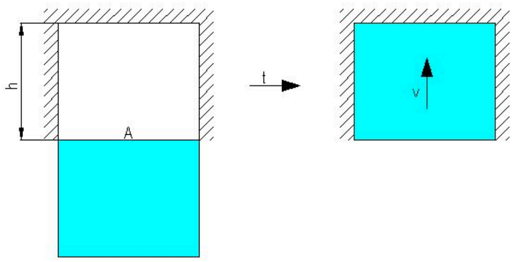


Figure 6.5: Filling of chamber as gears comes in and out of mesh, pressure difference

This means that the equations for constant accelerations can be used and this is shown in equation 6.17 and 6.18

$$v = v_0^2 + a \cdot t[m/s] \tag{6.17}$$
$$v^2 - v_0^2 = 2 \cdot a \cdot h[m/s] \tag{6.18}$$

In this case the start velocity can be set zero,$v_0 = 0[m/s]$ which means that the acceleration can be found from equations 6.17 and 6.18and is given in equation 6.19.

$$\begin{aligned} v^2 &= (a \cdot t)^2[m/s] \\ (a \cdot t)^2 &= 2 \cdot a \cdot h[m/s] \\ a &= \frac{2 \cdot h}{t^2}[m/s^2] \end{aligned} \tag{6.19}$$

By combing these two assumptions it is possible to find a limitations for the maximum height of the gear as shown in equation 6.20 and 6.21.

$$\frac{2 \cdot h}{t^2} = \frac{\Delta p}{\rho \cdot h}[m^2/s] \tag{6.20}$$
$$h = \sqrt{\frac{\Delta p \cdot t^2}{2 \cdot \rho}}[m] \tag{6.21}$$

The time it takes to from the gears comes out of mesh until they come back into mesh, can be approximated by using the angle of the inlet. The time that chamber is open for filling can then be estimated by the equation 6.22.

$$\begin{aligned} rps &= \frac{N}{60}[rps] \\ t &= \frac{\theta}{rps \cdot 360}[s] \\ t &= \frac{\theta}{N \cdot 6}[s] \end{aligned} \tag{6.22}$$

This means that the height is inverse proportional with the speed, as shown in equation 6.23.

$$h \propto \frac{1}{N} \tag{6.23}$$

6.4 Maximum speed

The motor and the bearing has a maximum speed that the pump can not exceed without putting the pump in danger for failure. The maximum speed is dependent of several different parameters:

- Bearings
- Back stop unit
- STC valve
- Hydraulic motor
- Mechanical seal

Which one of these factors that occurs first is design dependent, they might all occur more or less at the same time or one of them is the only one that effects the design.

6.4.1 Bearings

There are several types of bearings on the market, for instance ball bearings, bushings, magnetic bearing among many other types of bearings. For pumping molasses ball bearing and bushing is most likely the best choice for this purpose.

Ball bearing Ball bearings exist in several variants, the common for them all is that they are not recommended to be run in the pumping medium, especially not in molasses. So when a ball bearing is chosen a mechanical is also recommended and of course this also have have a limitations regarding speed. The bearings used by Frank Mohn Fusa A/S today is listed in table 6.3.

Bearing	Type	d	D	B	rpm	rpm_{max}
N 309 ECP	Cylindrical roller bearings, single row	45	100	25	7500	8500
N 213 ECP	Cylindrical roller bearings, single row	65	120	23	6300	6700
N 216 ECP	Cylindrical roller bearings, single row	80	140	26	5300	5600
3213 A/CR	Angular contact ball bearings, double row	65	120	38.1	4500	4800
3314 A	Angular contact ball bearings, double row	70	150	63.5	4300	4300
3317 A	Angular contact ball bearings, double row	85	180	73	3200	3400
3319 A	Angular contact ball bearings, double row	95	200	77.8	2800	3000
3310 A-2RS1/C3MT33	Angular contact ball bearings, double row	50	90	30.2	4800	4800

Table 6.3: Bearing data, courtesy of SKF [2006]

Bushings Bushing have the benefit that they can be used for moving parts that are in contact with the molasses, as long as the temperature rise in the bushings are not getting so high that the molasses crystallizes. The bushing that are allowed to use in molasses and that are suitable for the task is SKF bushings in the PSMF series and the properties for these bushings are given in table 6.4.

Description	Value	Unit
Permiss. load(dyn/stat)	10/50	$[Pa]$
Permiss. sliding velocity	10	$[m/s]$
Friction coefficient, μ	$0.05 \ldots 0.10$	$[-]$
Temperature range	$-10 \ldots +100$	[°C]
Shaft tolerance f7-f8	$-30 \ldots -60$	$[\mu m]$
Housing tolerance H7	$0 \ldots +30$	$[\mu m]$
Shaft roughness Ra, μ	$0.2 \ldots 0.8$	$[-]$
Shaft hardness	$200 \ldots 300$	$[HB]$

Table 6.4: Characteristic data for PSMF 506035 A51, courtesy of SKF [2006] .

The maximum speed the bushing can have is given in equation 6.24 when the tangential speed is converted into revolutions. The largest SKF bushing have a diameter of $50[mm]$ and this will then give a maximum speed which are indicated in equation 6.25. When a smaller bushing is selected then the maximum speed will increase.

$$
\begin{aligned}
v_t &= w \cdot r[m/s] \\
v_t &= \frac{N_{motor} \cdot 2 \cdot \pi \cdot r}{60}[m/s] && (6.24) \\
N_{motor\,max} &= \frac{60 \cdot v_t}{2 \cdot \pi \cdot r_{shaft}}[rpm] \\
N_{motor\,max} &= 1910[rpm] && (6.25)
\end{aligned}
$$

6.4.2 Mechanical seal

The mechanical seal can be evaluated the same way as for the bushing, with the exeption that maximum allowed tangential speed is $20[m/s]$. This then means that the maximum speed for the motor is given in equation 6.26.

$$N_{motor\,max} = 3820[rpm] \qquad (6.26)$$

6.4.3 Hydraulic motor

The motors that is chosen comes from Bosch Rextrot and they have some limitations regarding the maximum speed they can operate on. The data for maximum speed for the different sizes is listed in table 6.5.

Motor $[cm^3]$	$N_{motor\,max}[rpm]$
125	4000
200	2900
250	2500
355	2240

Table 6.5: Maximum speed for the different hydraulic motors, courtesy of Group [2004].

6.4.4 STC-valve

The STC-valve can deliver a certain amount of hydraulic oil and this might therefore limit the maximum speed. The Bosch Rextrot motor have a high volumetric efficiency and therefore by using $\eta_v = 0.95$ the result will be very conservative. The motors speed can be calculated by using equation 6.28.

$$
\begin{aligned}
n_{motor} &= \frac{Q_{motor} \cdot \eta_{v_{motor}}}{V_{motor}}[\frac{rev}{s}] && (6.27) \\
N_{motor} &= n_{motor} \cdot 60[rpm] && (6.28)
\end{aligned}
$$

The different STC sizes for different motors are listed in table 6.6.

STC/Motor	25	30	40
$125[cm^3]$	$1216[rpm]$	$3040[rpm]$	$4940[rpm]$
$200[cm^3]$	$760[rpm]$	$1900[rpm]$	$3088[rpm]$
$250[cm^3]$	$608[rpm]$	$1520[rpm]$	$2470[rpm]$
$355[cm^3]$	$428[rpm]$	$1070[rpm]$	$1739[rpm]$

Table 6.6: Maximum volume flow for different STC sizes

6.4.5 Backstop unit

The back stop unit is necessary to avoid the pump of running the wrong way under flushing, this would lead to that the hydraulic motor would start working as pump and build up an pressure which then might lead to failure of part of or the whole system.

The back stop unit is an advanced bearing that only allows the shaft to rotate one way, but it has its limits regarding the speed, table 6.7.

Pipe stack	Maximum speed[rpm]
SD 100	10000
SD 125/150	6600
SD 200	4500
SD 300	4000
SD 350	2000

Table 6.7: Backstop unit.

6.5 Maximum motor pressure

The maximum motor pressure is given by the system pressure and the return pressure. The system pressure is supposed to be $p_s = 225[bar]$and the return pressure around $p_r = 5[Bar]$. This yields to the motor pressure given in equation 6.29.

$$\begin{aligned} p_{motor} &= p_s - p_r[Pa] \\ p_{motor} &= 220 \cdot 10^5[Pa] \end{aligned} \tag{6.29}$$

6.6 Maximum differential pressure

The flanges at the pipe stack are following DNV certification DN16, which implements that the maximum pressure at the outlet must not exceed 16[bar]. How this is solved is either by a buy pass valve or make sure that such a high pressure never will occur by design the geometry of the pump in such a way, see Skåtun [2007].

6.7 Conclusion

There are several factors that gives the limitations on the design of the new pump. The limitations will affect the performance of the pump, but it will also reduce the number of variants that are possible to make. This will then hopefully decrease the development time. Here it has not manufacturing limitations been taken into account, these needs to be taken into account when evaluating a proposal. Not everything is possible to make and other might be time consuming or expansive solutions. It is therefore desirable to use as many standard component as possible, as long as this does not reduce the performance drastically. The outer diameters of the gear seems to be an important factor together with the customers requirements. Even if there is several limitations, it should still be a solvable task.

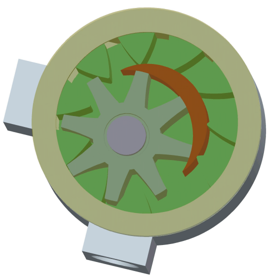
7 Different proposal for design

7.1 Introduction

When the limitations for the design had been laid out then the process of coming up with different proposals of how to design a submerged molasses pump. There are already pumps for molasses on the market but non of these pumps are submerged. As a starting point a pump made by Framo in the 80's where used to come up with different proposal, versions.

7.2 Version 1

This was the original "Framo pump" that were used back in the 80's for lower volume flow than required today. The principal might still be suitable for larger volume flows as well and it was therefore considered as a possible solution. The model was modeled up in Pro Engineer from the original drawing and is illustrated in figure 7.1.



(a) Pro Engineer model

(b) Model of RP100

Figure 7.1: Version 1

From the ProE model the data given in table 7.1 could be found.

Description	Value	Unit
Number of teeth Pinion	8	[−]
Number of teeth Gear	11	[−]
Tip diameter Pinion	203.7	[mm]
Tip diameter Gear	184.0	[mm]
Root diameter Pinion	104.0	[mm]
Root diameter Gear	280.0	[mm]
Tip width of Pinion	20.0	[mm]
Tip width of Gear	2.37	[mm]
Root width of Pinion	33.3	[mm]
Root width of Gear	54.0	[mm]
Offset between gear	37.0	[mm]
Height of Pinion/Gear	90.0	[mm]
Inlet Area	16382	[mm^2]
Outlet Area	16383	[mm^2]
Area between cogs of Pinion	1759	[mm^2]
Area between cogs of Gear	1600	[mm^2]
Distance between cogs of gear	22.0	[mm]

Table 7.1: Data for Version 1.

It already existed a model of the “Framo pump” which where used to study the principle of a gear pump. From the model it came clear that it would be large wear of the gear and pinion due to sliding effects when the gears comes in and out off mesh. Due to this a new gear shape would be desirable.

7.3 Version 2

To reduce the wear on the gears it was desirable to change the gear shape to involute cog shape, which is standard in most gearing design these days. The module and number of teeth was selected in such a way that the outer diameter was sufficient regarding the size limitations for the design. The new proposal are presented in figure 7.2.

(a) Pro Engineer model (b) Model of Version 2

Figure 7.2: Version 2

A special software, MitCalc, mit [2008]to genearate the gear shape when the basic gearing dimensions had been calculated. The basic data for gear can be found in table 7.2 .

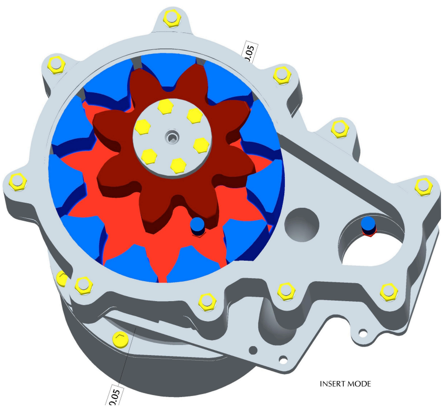
Description	Value	Unit
Number of teeth Pinion	9	[−]
Number of teeth Gear	12	[−]
Tip diameter Pinion	187.1	[mm]
Tip diameter Gear	183.7	[mm]
Root diameter Pinion	122.6	[mm]
Root diameter Gear	280.0	[mm]
Tip width of Pinion	8.2	[mm]
Tip width of Gear	12.8	[mm]
Root width of Pinion	29.0	[mm]
Root width of Gear	57.7	[mm]
Offset between gear	26.6	[mm]
Height of Pinion/Gear	80.0	[mm]
Inlet Area	13412	[mm^2]
Outlet Area	11137	[mm^2]
Area between cogs of Pinion	904	[mm^2]
Area between cogs of Gear	732	[mm^2]
Distance between cogs of gear	8.7	[mm]

Table 7.2: Data for Version 2.

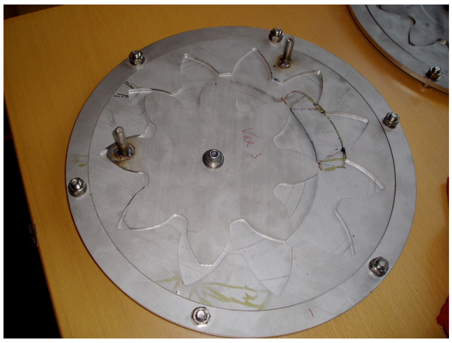
The problem considering this version would be manufacturing due the distance between cogs of gear, then a 8 [mm] mill needed to be used. On the gear which a large overhang this would be critical to obtain the needed accuracy. Also the displaced area is drastically reduced compared to version 1, which then means a higher rotating speed of the pump is necessary in order to obtain the same volume flow when the height is kept the same.

7.4 Version 3

In order to increase the displaced volume number of teeth was reduced and the module increased. The teeth shape is still the same as for version 2, see figure 7.3.



(a) Pro Engineer model



(b) Model of version 3

Figure 7.3: Version 3

The basic gearing dimension is given in table 7.3..

Description	Value	Unit
Number of teeth Pinion	8	[−]
Number of teeth Gear	11	[−]
Tip diameter Pinion	191.4	[mm]
Tip diameter Gear	1186.1	[mm]
Root diameter Pinion	119.8	[mm]
Root diameter Gear	251.4	[mm]
Tip width of Pinion	9.4	[mm]
Tip width of Gear	14.5	[mm]
Root width of Pinion	31.7	[mm]
Root width of Gear	61.2	[mm]
Offset between gear	30	[mm]
Height of Pinion/Gear	80.0	[mm]
Inlet Area	15089	[mm^2]
Outlet Area	12530	[mm^2]
Area between cogs of Pinion	1137	[mm^2]
Area between cogs of Gear	916	[mm^2]
Distance between cogs of gear	10.0	[mm]

Table 7.3: Data for Version 3.

The problem considering this version would be manufacturing due the distance between cogs of gear, then a 8 [mm] mill needed to be used. On the gear which a large overhang this would be critical to obtain the needed accuracy. Also the displaced area is drastically reduced compared to version 1, which then means result in a higher rotating speed of the pump in order to obtain the same volume flow.

7.5 Version 4

Since there was no clear choice between internal and external gear pump, an external version was evaluated as well. Due to prototyping a version using the same pinion as version 3 was considered as illustrated in figure 7.4

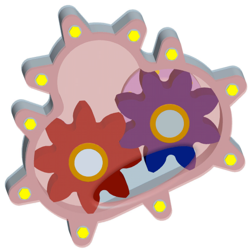


Figure 7.4: Pro engineer model of version 4

The basic gearing gearing dimensions for version 4 is given in table 7.4 .

Description	Value	Unit
Number of teeth Pinion	8	[−]
Number of teeth Gear	11	[−]
Tip diameter Pinion	191.4	[mm]
Root diameter Pinion	119.9	[mm]
Tip width of Pinion	9.4	[mm]
Root width of Pinion	31.7	[mm]
Offset between gear	160.0	[mm]
Height of Pinion/Gear	80.0	[mm]
Inlet Area	10996	[mm^2]
Outlet Area	26560	[mm^2]
Area between cogs	1137	[mm^2]

Table 7.4: Data for Version 4.

7.6 Version 5

Instead of having number of cogs the same on pinion and gear a new design proposal was suggested. Since there was no clear choice of the optimal number of cogs at current stage, the idea was to make a house that could fit several different set of gears. To be able to so it is necessary that the total outer diameter is the same and that the diameter of the rotating pinion is the same.

The outer diameter of the pinion is given by equation 7.1.

$$d_{a_{pinion}} = m \cdot (z_{pinion} + 2)\,[m] \tag{7.1}$$

Equation 7.1 needs to be the same for all the gear set that is desirable to use in the same gearing house, this yields to equation 7.2.

$$\begin{aligned}\frac{m_1}{m_2} &= \frac{z_{pinion_2}+2}{z_{pinion_1}+2}[-] \\ \frac{m_1}{m_3} &= \frac{z_{pinion_3}+2}{z_{pinion_1}+2}[-]\end{aligned} \tag{7.2}$$

At the same time the total outer diameter needs to be the same for each set. This is achieved by varying the gear, both in diameter, equation 7.3, and axis working distance, equation 7.4.

$$d_{a_{gear}} = m \cdot (z_{gear}+2)\,[m] \tag{7.3}$$

$$a_w = m \cdot (z_{pinion}+z_{gear})\,[m] \tag{7.4}$$

The total outer diameter is then given in equation 7.5.

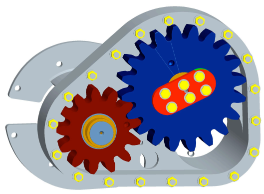
$$\begin{aligned} d &= \frac{d_{a_{pinion}}+d_{a_{gear}}}{2} + a_w[m] \\ d &= m \cdot (z_{pinion}+z_{gear}+2)\,[m]\end{aligned} \tag{7.5}$$

The numbers needs to real and the only way to find suitable set that can fit in the same housing is by trial and error. This is a time consuming task but after some time three different gear set was found which is given in table 7.5.

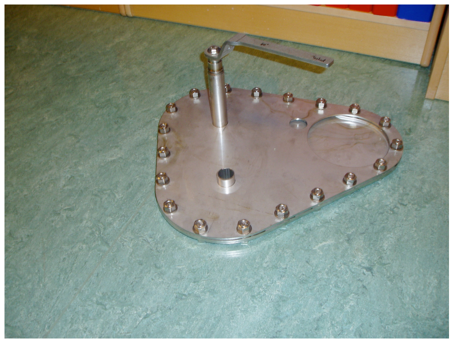
Description	Set1	Set2	Set3	Unit
Module	20	12	10	[mm]
Number of teeth Pinion	7	13	16	[-]
Number of teeth Gear	12	20	24	$[-]$
Tip diameter Pinion	180.3	180.3	180.3	[mm]
Root diameter Pinion	90.0	126.0	135.0	[mm]
Tip diameter Gear	280.3	264.3	180.3	[mm]
Root diameter Gear	190.0	210.0	215.0	[mm]
Tip width of Pinion	9.95	7.41	6.5	[mm]
Root width of Pinion	21.5	17.1	15.3	[mm]
Tip width of Gear	12.2	8.2	7.0	[mm]
Root width of Gear	27.7	19.6	17.3	[mm]
Offset between gear	190.2	198.1	200.1	[mm]
Height of Pinion/Gear	80.0	80.0	80.0	[mm]
Area between cogs of pinion	1328	482	335	[mm^2]
Area between cogs of pinion	1238	483	337	[mm^2]

Table 7.5: Data for Version 5.

After recommendations from Iversen and Burkow [1978] it was decided that probably gear set two would be the most efficient gear set, this gear set is shown in figure 7.5.



(a) Pro Engineer model



(b) Model of version 5

Figure 7.5: Version 5

7.7 Comparison

The different version have their benefits and disadvantages so below is a list of some if them:

1. Ver 1

Advantages

- 88 teeth passing until same pair of cogs comes into to mesh again.
- Physically small, can easily be fitted inside the support ring
- Large area between cogs, resulting in a lower speed
- Large splits in gear, reducing the outlet loss

Disadvantages

- Pure sliding of cog along path of action, resulting in high wear of cogs.
- Forces on the cog, is at the tip
- Only one pair of cogs in mesh at all time.
- Small inlet and outlet area

- Difficult to manufacture due to high requirements of tolerances
- High requirements to axis distance
- Large radial forces on bearings
- Overhang of gear and pinion

2. Ver 2

   Advantages

   - Involute cog shape , accuracy of center distance not critical
   - Between one and two gear in mesh.
   - Small amount of sliding, which means low wear factor.
   - Large inlet and outlet area.
   - Low friction

   Disadvantages

   - Difficult to manufacture due to high requirements of tolerances.
   - Large radial forces on bearings.
   - Overhang of gear and pinion.
   - Small radius of gear and pinion.
   - Undercutting of teeth

3. Ver 3

   Advantages

   - Involute cog shape , accuracy of center distance not critical
   - Between one and two gear in mesh.
   - Small amount of sliding, which means low wear factor.
   - Large inlet and outlet area.
   - 88 teeth passing until same pair of cogs comes into to mesh again.
   - Low friction

   Disadvantages

   - High requirements to tolerances.
   - Difficult to manufacture fluid separator due to high requirements of tolerances.
   - Large radial forces on bearings.

- Overhang of gear and pinion.
- Small radius of gear and pinion.
- Large undercutting of teeth

4. Ver 4

   Advantages

   - Involute cog shape , accuracy of center distance not critical
   - Between one and two gear in mesh.
   - Small amount of sliding, which means low wear factor.
   - Large inlet and outlet area.
   - Low friction
   - Pressure difference over several gears.
   - Bearing support on both side of gears
   - Easy to manufacture pinion and gear.
   - Gear and pinion are the same, just with different shaft attachment

   Disadvantages

   - Small radius of gear and pinion.
   - 8 teeth passing until same gears comes into mesh
   - Physically large

5. Ver 5

   Advantages

   - Involute cog shape , accuracy of center distance not critical
   - Between one and two gear in mesh.
   - Small amount of sliding, which means low wear factor.
   - Large inlet and outlet area.
   - Low friction
   - Pressure difference over several gears.
   - Bearing support on both side of gears
   - Easy to manufacture pinion and gear.
   - 3 sets of gears can be fitted in the same housing

Disadvantages

- Small radius of gear and pinion.
- Physically large.
- Complicated shaft support to allow three sets of gears to be fitted inside the same house.

7.8 Conclusion

The different version have their different benefits, and disadvantages. After evaluating the pros and cons for the different proposal it was decided to make a prototype of version 3 and 5. This was due to that there was no obvious choice which version would be the most suitable for pumping of molasses.

8 Performance estimate

8.1 Introduction

It is important to find out what parameters that might effect the perfomance of the pump. Every part of the pump will have some tolerances, and it always desirable to set as large tolerances as possible due to machining. But at some places it is necessary to set fine tolerances, and it is therefore important to get an idea of how these parameters influence the performance of the pump.

8.2 Simplifications

The tolerances will influence the leakage flow and the shear moment in the pump. To find out how the tolerances effects the overall performance a CFX simulation in combination with a complete test is necessary. This is a complex task and therefore an estimate needs to be made to find some appropriate dimension for the different parts. Most important is it to find out the connections between the parameters.

The parameters that can be changed is assumed to influence three effects:

- Shear stress
- Leakage flow
- Bearing friction

8.3 Gap flow

As an estimate the leakage flow and shear moment over and under the gear, and between the gear and the housing where taken into account. This due to that it is here the largest areas can be found

The gears is simplified by using a cylinder instead of the actual gear shape. This is most likely to be a conservative assumption, which will do some compensation of the gaps that was ignored.

8.4 Parameters

The parameters after the simplification and assumption has been taken into account is listed in table and shows which parameters that applies for the different design

- Diameter pinion
- Shaft diameter for pinion
- Height pinion

- Diameter gear
- Shaft diameter for gear
- Height pinion gear
- Thickness of housing

These parameters is valid for both version, the height of the gears for the external versions have to be almost identical for the pinion and the gear and can therefore in this case be combined to one height.

8.5 Shear stress

There will occur shear stresses between each moving part, but it is possible to categories this into to two categories:

- Cylinder rotating inside another cylinder - between the gear/pinion and the housing
- Between rotating disk and fixed surface - between the gear/pinion and the lid of the housing

The shear stress τ is given by the equation 8.1, White [1994].

$$\tau = \mu \cdot \frac{d\theta}{dr}[Pa] \tag{8.1}$$

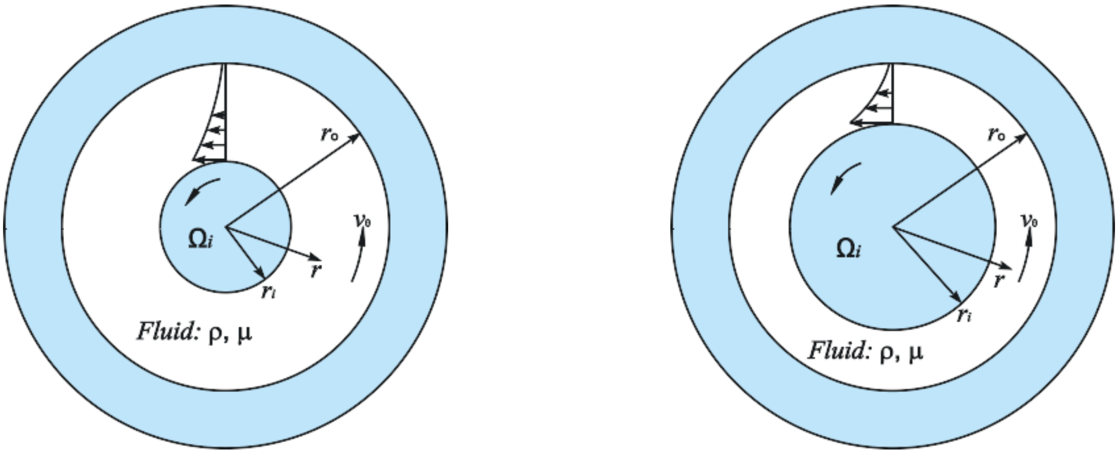
Integration of the shear stress over the working area gives the moment and is shown in equation 8.3,White [1994].

$$dM = \tau \cdot dA \cdot r[N \cdot m] \tag{8.2}$$
$$M = \int dM[Nm] \tag{8.3}$$

The problem is to find out how the velocity changes and over what area it works. This has been simplified for the two cases mentioned above.

8.5.1 Rotating cylinder

The shear stress between a rotating cylinder and a fixed wall is illustrated in figure 8.1.



(a) Small inner diameter

(b) Large inner diameter

Figure 8.1: Shear stress between rotating cylinder and fixed wall

If the diameter of the rotating cylinder is much larger then clearance then the problem can be estimated by using Couchette flow as shown in figure 8.2.

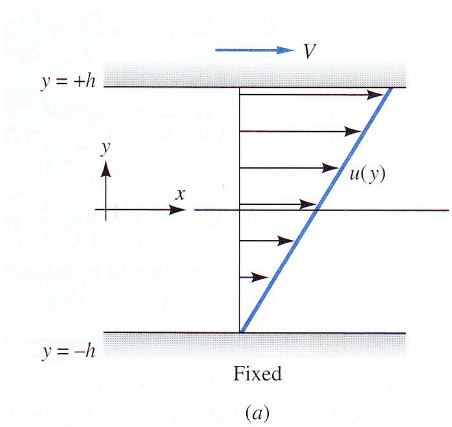


Figure 8.2: Couchette flow

This means that equation 8.1 can be simplified, the change in velocity $\frac{d\theta}{dr}$ will be constant when no slip conditions are assumed. At the inner cylinder the velocity is given in equation 8.4 and at the wall the velocity is set to $v_w = 0\left[\frac{m}{s}\right]$.

$$v_c = \Omega \cdot r_i \left[\frac{m}{s}\right] \tag{8.4}$$

When equation is implied together with the boundary condition at the wall then the expression for the change in velocity is shown in equation 8.5.

$$\begin{aligned} \frac{d\theta}{dr} &= \frac{v_c - v_w}{r_o - r_i}\left[\frac{m}{s^2}\right] \\ \frac{d\theta}{dr} &= \frac{\Omega \cdot r_i}{t_i}\left[\frac{m}{s^2}\right] \end{aligned} \tag{8.5}$$

The area that the shear force works over is given in equation 8.6.

$$A_w = 2 \cdot \pi \cdot r_i \cdot h \cdot \theta_w [m^2] \tag{8.6}$$

Since the shear stress is assumed to be constant over the area the total expression for the moment is given in equation 8.7.

$$\begin{aligned} M_{rotating\,cylinder} &= \mu \cdot \frac{d\theta}{dr} \cdot r_i \cdot A_w [N \cdot m] \\ M_{rotating\,cylinder} &= \frac{2 \cdot \pi \cdot \nu \cdot \rho \cdot \Omega \cdot h \cdot \theta_w \cdot r_i^3}{t_i} [N \cdot m] \end{aligned} \tag{8.7}$$

This yields that the power loss can be found in equation 8.8.

$$\begin{aligned} P_{rotating\,cylinder} &= M_{rotating\,cylinder} \cdot \Omega [W] \\ P_{rotating\,cylinder} &= \frac{2 \cdot \pi \cdot \nu \cdot h \cdot \theta_w \cdot \Omega^2 \cdot r_i^3}{\rho \cdot t_i} [W] \\ P_{rotating\,cylinder} &= \frac{8 \cdot \pi^3 \cdot \nu \cdot \rho \cdot h \cdot \theta_w \cdot N_{motor}^2 \cdot r_i^3}{3600 \cdot t} [W] \end{aligned} \tag{8.8}$$

This means that the power consumption has the following dependencies given in equation 8.9 and 8.12.

$$P_{rotating\,cylinder} \propto r_i^3 \tag{8.9}$$

$$P_{rotating\,cylinder} \propto N_{motor}^2 \tag{8.10}$$

$$P_{rotating\,cylinder} \propto h \tag{8.11}$$

$$P_{rotating\,cylinder} \propto \mu \tag{8.12}$$

8.5.2 Rotating disc

There will also occur shear stresses over and under the gear, but here the velocity is not constant as shown in equation 8.13which means it is necessary to integrate over the working area for the shear stress, figure 8.3.

$$v_c = \Omega \cdot r \left[\frac{m}{s}\right] \tag{8.13}$$

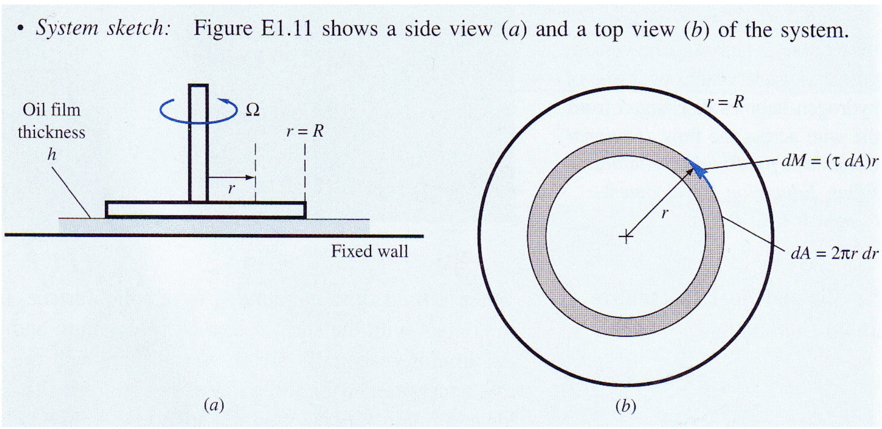


Figure 8.3: Rotating disc, courtesy of White [1994]

When the velocity is deviated it is found that the change in velocity is constant, equation 8.14 so that the shear stress is constant as shown in equation 8.15.

$$\frac{d^2\theta}{dr^2} = \Omega \left[\frac{m}{s^2}\right] \tag{8.14}$$

$$\tau = \frac{\nu \cdot \rho \cdot \Omega \cdot r}{t} [Pa] \tag{8.15}$$

The moment is then found by integrating from the inner radius to the outer radius and is shown in equation 8.16.

$$\begin{aligned} dM &= \frac{\nu \cdot \rho \cdot \Omega \cdot 2 \cdot \pi \cdot r^3}{t_i} \cdot dr[N \cdot m] \\ M_{rotating\,disc} &= \int_{r_i}^{r_o} \frac{2 \cdot \nu \cdot \rho \cdot \Omega \cdot \pi \cdot r^3}{t_i} \cdot dr[N \cdot m] \\ M_{rotating\,disc} &= \frac{\nu \cdot \rho \cdot \Omega \cdot \pi}{2 \cdot t} \cdot \left(r_o^4 - r_i^4\right) \end{aligned} \tag{8.16}$$

The power can than easily be found and is given in equation 8.17.

$$\begin{aligned} P_{rotating\,disc} &= M_{rotating\,disc} \cdot \Omega [W] \\ P_{rotating\,disc} &= \frac{\nu \cdot \rho \cdot \Omega^2 \cdot \pi}{2 \cdot t_i} \cdot \left(r_o^4 - r_i^4 \right) [W] \\ P_{rotating\,disc} &= \frac{2 \cdot \nu \cdot \rho \cdot N_{motor}^2 \cdot \pi^2}{3600 \cdot t_i} \cdot \left(r_o^4 - r_i^4 \right) [W] \end{aligned} \tag{8.17}$$

8.6 Leakage flow

There will be leakage flow in the gaps in the pump, this leakage flow is simplified in such way that it the flow only happens in the plane, and it is also simplified to only flow over the parts as illustrated in figure 8.4.

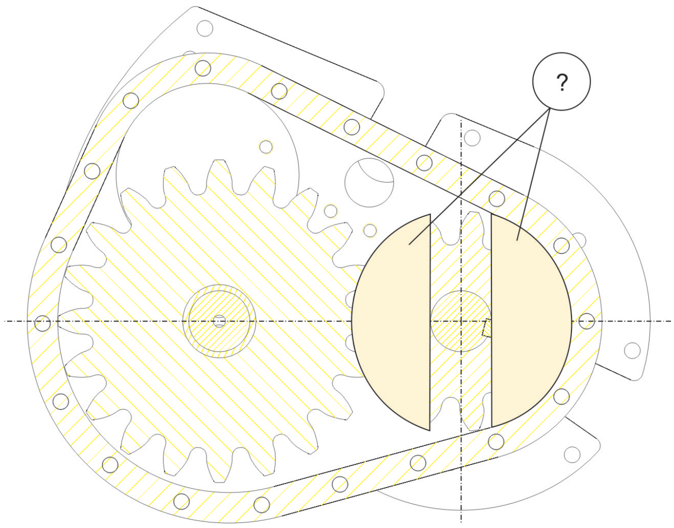


Figure 8.4: Assumption of how the fluid flows over a rotating disc

8.6.1 Laminar flow

It is assumed that the flow is laminar for the whole gap and that Bernoulli's equation can be used, equation 8.18-8.22. The hydraulic diameter is used, since the gap has a rectangular shape.

$$\begin{aligned}
\Delta p &= \frac{v^2 \cdot \rho \cdot f \cdot L_D}{2 \cdot D_h}[Pa] && (8.18)\\
v^2 &= \frac{2 \cdot \Delta p \cdot D_h \cdot Re_D}{\rho \cdot 64 \cdot L_D}[(m/s)^2] && (8.19)\\
v &= \frac{D_h^2 \cdot \Delta p}{32 \cdot \nu \cdot L_D \cdot \rho}[m/s] && (8.20)\\
Q_{leakage} &= \frac{D_h^2 \cdot \Delta p}{32 \cdot \nu \cdot L_D \cdot \rho} \cdot w \cdot h[m^3/s] && (8.21)\\
Q_{leakage} &= \frac{\Delta p \cdot h^3 \cdot w^3}{8 \cdot \nu \cdot L_D \cdot \rho \cdot (h^2 + 2 \cdot h \cdot w + w^2)}[m^3/s] && (8.22)
\end{aligned}$$

If the assumption $w \gg h$ is valid then the expression for the back flow is dependent of the clearance in third power as shown in equation 8.23.

$$Q_{leakage} = \frac{h^3 \cdot \Delta p}{8 \cdot \nu \cdot L_D \cdot \rho} \cdot w[m^3/s] \tag{8.23}$$

The power consumption is then given by equation 8.24.

$$\begin{aligned}
P_{leakage} &= Q_{leakage} \cdot \Delta p[W]\\
P_{leakage} &= \frac{h^3 \cdot \Delta p^2}{8 \cdot \nu \cdot L_D \cdot \rho} \cdot w[m^3/s] && (8.24)
\end{aligned}$$

8.6.1.1 Variable length and width

If the length and the width varies, but the assumption that height is much smaller than the overall width then the length can be expressed as a function of the width and the overall leakage flow and power loss can be estimated.

$$\begin{aligned}
Q_{leakage} &= \frac{h^3 \cdot \Delta p}{8 \cdot \nu \cdot \rho} \cdot \int \frac{dw}{L_D}[m^3/s] && (8.25)\\
P_{leakage} &= \frac{h^3 \cdot \Delta p^2}{8 \cdot \nu \cdot \rho} \cdot \int \frac{dw}{L_D}[W] && (8.26)
\end{aligned}$$

For the flow over and under the cylinder the length needs to be estimated, this was estimated by using figure 8.5.

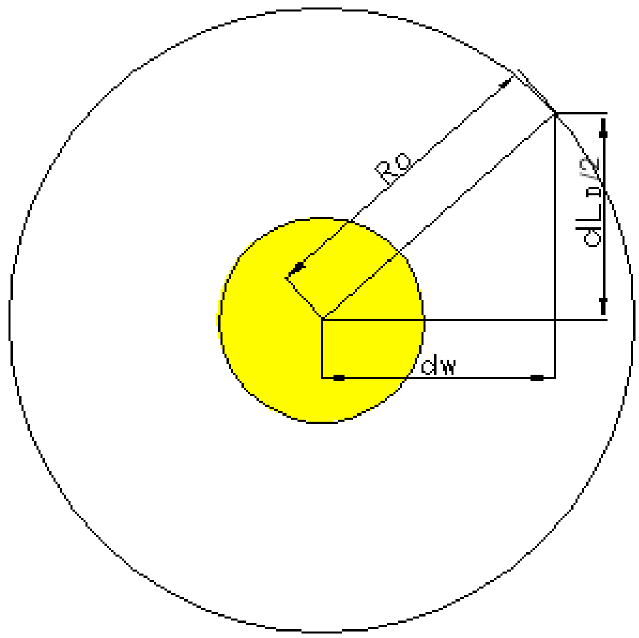


Figure 8.5: Assumption of how the fluid flows over a rotating disc

$$r_o = dw^2 + \left(\frac{dL_D}{2}\right)^2 [m] \tag{8.27}$$

$$dL_D = 2 \cdot \sqrt{r_o^2 - dw^2}[m] \tag{8.28}$$

$$L_D = \arcsin{(1)} - \arcsin\left(\frac{r_i}{r_o}\right) [m] \tag{8.29}$$

The different connections is then shown in equation 8.30 - 8.33.

$$Q_L \propto h^3 \tag{8.30}$$

$$Q_L \propto \Delta p \tag{8.31}$$

$$Q_L \propto \frac{1}{L_D} \tag{8.32}$$

$$Q_L \propto w \tag{8.33}$$

8.7 Bearings

The bearing can either be a ball bearing together with a mechanical seal or a bushing which can be run in the pumping medium. Therefore two cases have been evaluated.

8.7.1 Ball bearings

The problem is to find the forces that are acting on the bearing. This can be estimated when the output power and is known together with the diameters of the gears. This yields that an estimate of the total efficiency of the pump is necessary to obtain an estimate of the forces. In this case it is important to estimate a conservative efficiency so that the forces are not underestimated, equation 8.34-8.38.

$$P_{shaft} = \frac{\Delta p \cdot Q}{\eta_{t_{pumpe}}}[W] \quad (8.34)$$

$$P_{shaft} = \frac{M_{shaft} \cdot 2 \cdot \pi \cdot N_{motor}}{60}[W] \quad (8.35)$$

$$M_{shaft} = \frac{\Delta p \cdot Q \cdot 60}{\eta_{t_{pumpe}} \cdot 2 \cdot \pi \cdot N_{motor}}[N \cdot m] \quad (8.36)$$

$$F_{ball\,bearing} = \frac{M_{shaft}}{r_{shaft}}[N] \quad (8.37)$$

$$F_{ball\,bearing} = \frac{\Delta p \cdot Q \cdot 60}{\eta_{t_{pumpe}} \cdot \pi \cdot N_{motor} \cdot 2 \cdot r_{shaft}}[N] \quad (8.38)$$

This means that the power consumption can be evaluated by using equation 8.39 and 8.40.

$$M_{ball\,bearing} = \frac{\Delta p \cdot Q \cdot 60 \cdot \mu_{ball\,bearing}}{2 \cdot \eta_{t_{pumpe}} \cdot \pi \cdot N_{motor}}[N \cdot m] \quad (8.39)$$

$$P_{ball\,bearing} = \frac{\Delta p \cdot Q \cdot 60 \cdot \mu_{ball\,bearing}}{2 \cdot \eta_{t_{pumpe}} \cdot \pi \cdot N_{motor}} \cdot \frac{2 \cdot \pi \cdot N_{motor}}{60}[W]$$

$$P_{ball\,bearing} = \frac{\Delta p \cdot Q \cdot \mu_{ball\,bearing}}{\eta_{t_{pumpe}}}[W] \quad (8.40)$$

If then these equations are combined a the power loss for the bearing can be found as shown in 8.41.

$$P_{ball\,bearing} = P_{shaft} \cdot \mu_{ball\,bearing}[W] \quad (8.41)$$

This means that the power loss is proportional with μ_B and that $\eta_{tot}$is not necessary for a quick estimate to find out how many % the power loss in the bearings are, equation 8.42 and 8.43.

$$P_{ball\,bearing} \propto \mu_{ballbearing}[W] \quad (8.42)$$

$$P_{ball\,bearing} \propto \frac{1}{\eta_{t_{pumpe}}}[W] \quad (8.43)$$

8.7.2 Bushings

It is desirable to use bushing for the moving parts that are in contact with the molasses, as long as the temperature rise in the bushings are not getting so high that the molasses crystallizes and that the friction is not much higher than for a ball bearing. The bushing that are allowed to use in molasses and that are suitable for the task is SKF bushings in the PSMF series and the properties for these bushings are given in table 6.4.

There will be some molasses leaking into the bushing and this will yield to some additional friction due to shear stresses. This can be evaluated as the same way as for cylinder rotating inside a fixed cylinder as shown in equation

The bushing needs to be evaluated by equation 8.44 and 8.45.

$$M_{bushing} = \frac{4 \cdot \pi^2 \cdot \nu \cdot N_{motor} \cdot h \cdot r_{shaft}^3}{60 \cdot \rho \cdot t} [N \cdot m] \tag{8.44}$$

$$P_{bushing} = \frac{8 \cdot \pi^3 \cdot \nu \cdot N_{motor}^2 \cdot h \cdot r_{shaft}^3}{3600 \cdot \rho \cdot t} [N \cdot m] \tag{8.45}$$

From these equations it can been seen that the connections between the are given in equations 8.46-8.48.

$$P_{bushing} \propto \nu \tag{8.46}$$

$$P_{bushing} \propto N_{motor}^2 \tag{8.47}$$

$$P_{bushing} \propto r_{shaft}^3 \tag{8.48}$$

8.8 Conclusion

The performance estimate is a complex task and should ideally be done by CFD calculations. To do it by CFD is a complex and time consuming task and therefore a simple estimate was done. This showed that by changing one parameter, influence both the shear stress and the leakage flow. This means that there is no clear choice of the optimal values of the parameters, and that these needs to be found experimentally.

9 Prototypes

To be able to test the functionality of the pump and to decide which one of internal and external gear pump that is most suitable for pumping molasses two prototypes needs to be made. In order to be able to make prototypes several drawings of each part of the design might be necessary to make:

- Material drawing
- Pre machining drawing
- Fine machining drawing
- Welding drawing
- Pre machining drawing
- Fine machining drawing
- Assembly drawing
- Handling and shipping drawing

Not all these drawing is necessary for each part but most of them applies to all parts, to make these drawings the CAD tool Pro Engineer was used which helps shortening the drawing process, but it is still a time consuming process. It is important to check the drawings very good before the drawings are approved. This is normally done by an other person that created the drawings.

In addition it is also necessary to make a production three, where each part get an unique ID number, quantity, material, storage position. This is in the Frank Mohn AS company organized by a software called Multi, and is a great help when it comes to planning the production time of each part and when expected delivery of each part is.

9.1 Complete pump

One of the challenges by doing this was to make sure that it was only the pump head that needed to be exchanged for the prototypes, see figure 9.1.

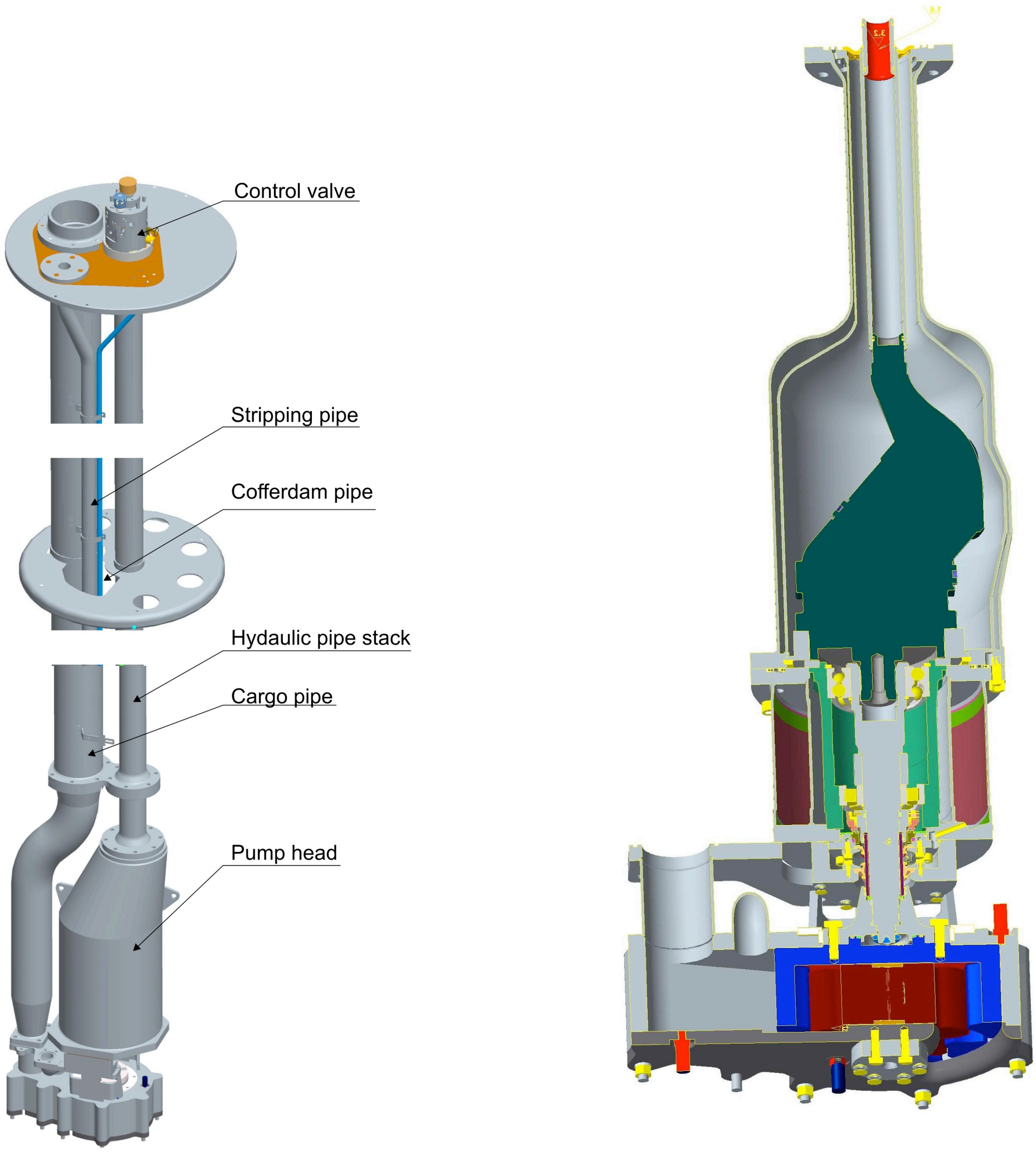


Figure 9.1: Complete pump

This will reduce the production time drastically and it also make it easier to compare the two version, since then the motor will be the same. For the prototype a special short pipe stack needs to be made, that way a smaller test tank can be used. As seen in figure 9.1, the rest are standard Frank Mohn AS components, which is a great benefit.

9.2 Pump head

The two different pump head, figure 9.2, is quite different from each other regarding size and how the bearing support is solved. This led to several challenges to make the two prototypes fit on the same pipe stack.

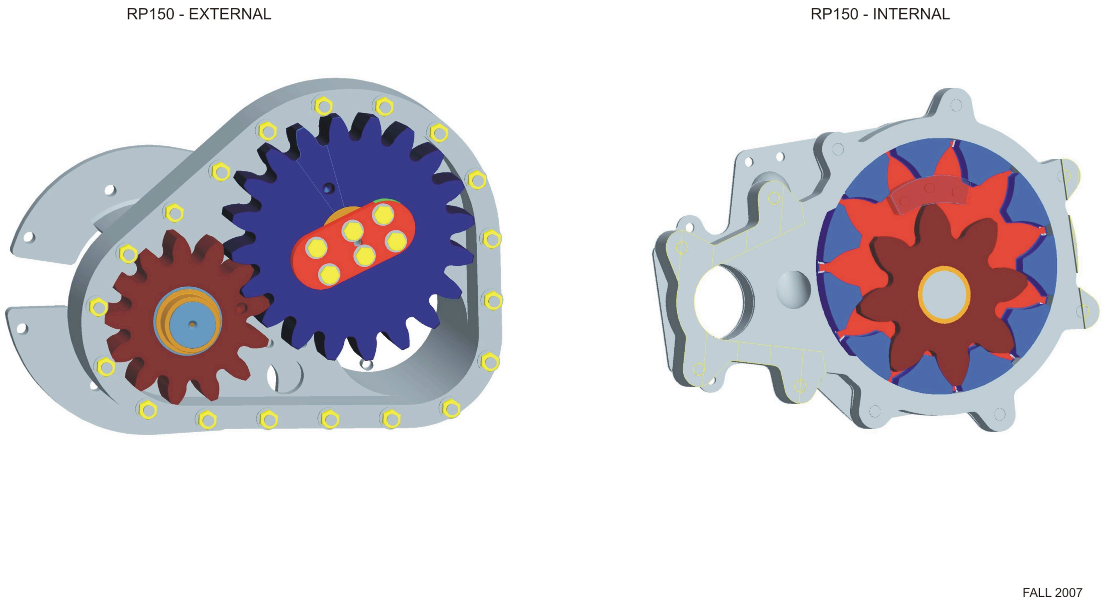
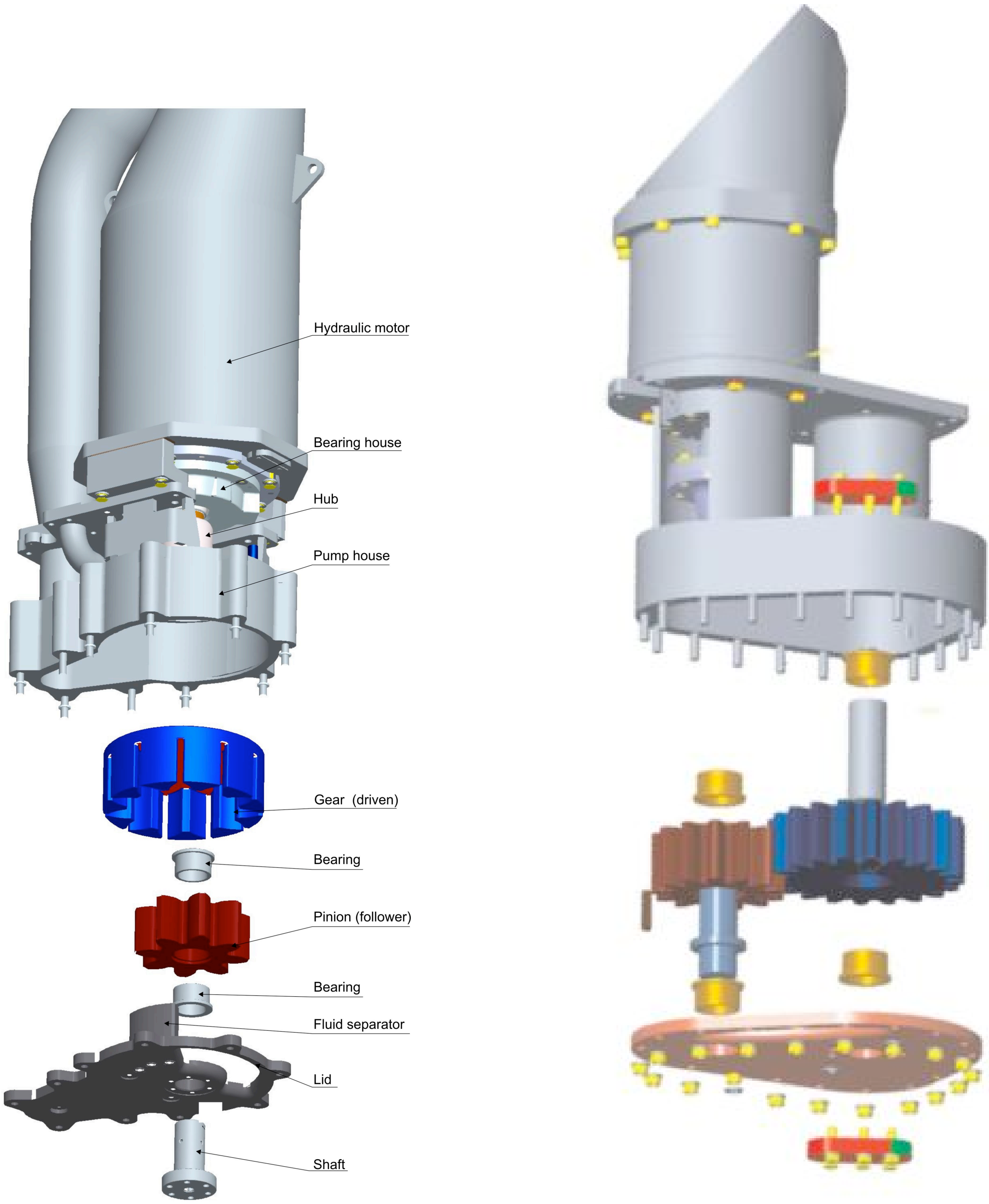


Figure 9.2: Pump heads

The pump head consist of several parts, figure 9.3 and the tolerances was therefore very complex, since each part effect the tolerances for the whole pump.



(a) Internal gear pump

(b) External Gear pump

Figure 9.3: Exploded pump heads

A cross section of the pump is illustrated in figure 9.4, to better explain how the different parts operate inside the pump.

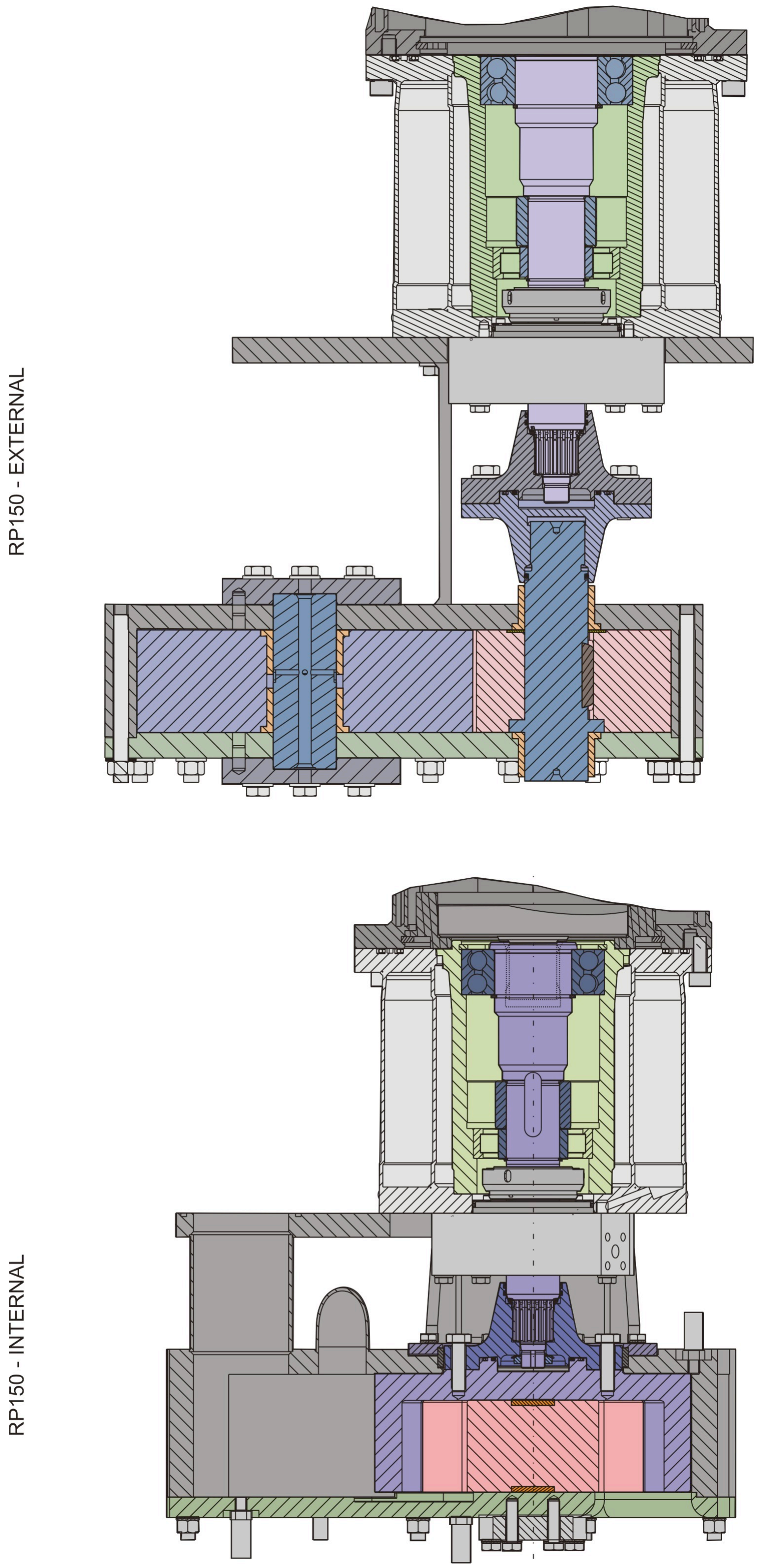


Figure 9.4: Exploded pump heads

9.3 Manufacturing

Most of the parts where made by Frank Mohn Fusa A/S, but some few parts needed to be made outside of the company. This was in order to be able to get the parts quicker, due to an extraordinary high production in 2007. The gear were milled at the company, but in the future it is recommended that this will be done at a gearing company where there is proper tools for making the cog shape correct.

When the molasses pump is set into production most of the parts will be made by the company. In addition templates and support will be made for some parts to ease the production.

Several problem occurred when assembling the pump, first of all the gears did not match up. The gear were first cutout from a massive circular steel rod with water jet and then after wards it needed to be machined from to sides. The alignment when turning the gears after the first machining led to small error is in the alignment, and also the surface finish of the machining was not as good as desirable. Therefore it was hard to get the gear to run smoothly in start, this were fixed by manually sanding the surface at the places were it was needed.

In addition were the vertical tolerances off their limits which also made the lid hard to get on. This were fixed by adding several thin pieces of steel were it was needed, the fit was slightly better for the external version which were put together during testing of the internal version.

9.4 Conclusion

The prototypes have several common parts, which eased the production and was a major achievement for this project. By using Pro Engineer it was possible to get feedback and input from the production crew, which have helped a lot regarding coming up with smart solutions to ease the work for the producers. The drawings were taken directly out from the CAD model, and this work could never have been done without Atle Hope, who made many drawings, and also approved all the drawings. He also helped out with making the structures of the two prototypes and put them into Multi. This gave benefits later, when there only a few drawings that did not got approved by the production due to a missing dimension.

The production of the two prototypes went smoothly with just a few delays of some parts compared to the timetable for this project. Some of the parts had some errors, but none had such high errors that they could not bee used. All these small inaccuracy of the parts put this project behind schedule, but at the end there were two prototypes ready for testing.

10 Evaluation of prototype

10.1 Introduction

To evaluate the the different prototypes in order to find the most suitable pump for pumping molasses a proper test rig was necessary. Also instrumentation and logging of data is important in order to be able to compare the two different versions. Testing is a time consuming process and is therefore, important to evaluate what to test so that the desirable data can be extrated from the test results.

10.2 Test rig

Frank Mohn Fusa AS had a tank for demonstration of their SD125 pump, Bergfjord [2006]. This tank was modified in such a way that the internal and external gear pump could be tested in this tank, see figure 10.1.

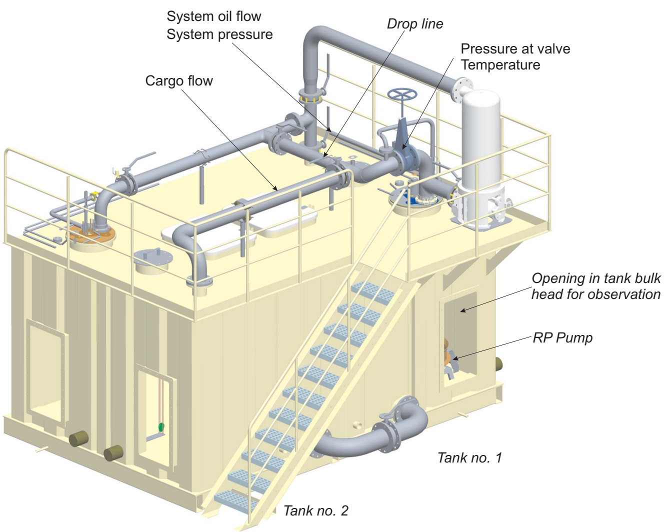


Figure 10.1: Test rig for molasses pumps

10.3 Pre face

It is assumed that the different input parameters is independent of each other such that superposition can be used to find a solution that includes all the input parameters, by varying the input parameters one by one. This will reduce the number of required test drastically, see equation 10.2 compared to if the parameters is not independent of each other, equation 10.1.

$$N_{test} = m^n \tag{10.1}$$
$$N_{test} = m \cdot n \tag{10.2}$$

To make sure that data is consistent from one run to another run, it is important to vary the same parameters under the same conditions at least twice to see that there is no fluctuation in the data. Most likely there will always be some external noise which makes it important to run the same test several times, preferably at least five times in order to find out that there is no connection between the noise in the result results.

10.4 Input parameters

From theory explained earlier it is assumed that the parameters that can be changed during normal operation of the pump is:

- Hydraulic oil pressure to the hydraulic motor
- Hydraulic oil flow to the hydraulic motor
- Temperature of the fluid
- Position of cargo valve, simulation of pipe friction at different terminals

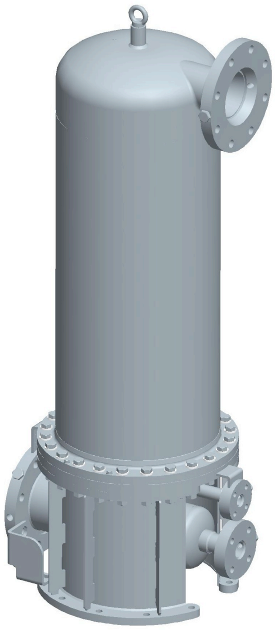
10.4.1 Control of the hydraulic motor

The motor can be controlled with to parameters, the hydraulic oil pressure gives the hydraulic motor a given torque, while the hydraulic oil flow gives the hydraulic motor a given speed. The hydraulic oil pressure was at the test tank controlled through a potentiometer which gave an output between 0-10 voltage. This was controlled manually with a handle and therefore there is inaccuracy of its position.

10.4.2 Cargo temperature

The cargo temperature was controlled through two different system, before a test could start the molasses needed to be heated and during a test the power loss generated heat and therefor cooling was necessary. The heating was done by inserting several plastic tubes at the bottom of the test tank, figure 10.2, which then was connected to a normal house heating system. The cooling was done by running the molasses through a normal cargo heater which were connected to cold water rather than steam.

(a) Heating of cargo



(b) Cooling of cargo

Figure 10.2: Heating arrangement in tank.

10.4.3 Pipe friction

To simulate pipe friction a normal vane valve was installed on the test cargo pipe. This valve was maneuvered with a wheel that could be turned to open and close the valve.

10.5 Output

To monitor the performance of the pump several different types of sensors were attached to the pump:

- pressure sensors
- temperature sensors
- rpm sensors
- distance sensors
- strain gauges

The location of these test points can bee seen in figure 10.3, here is shown for the internal version but the same is valid for the external version as well.

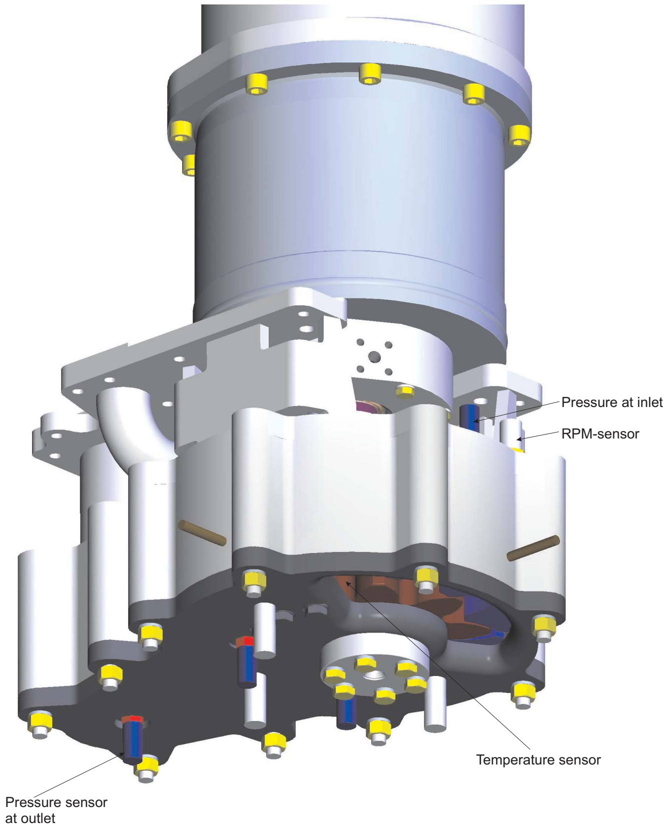


Figure 10.3: Sensor for internal version

The data from the sensors were logged with Catman, so the performance of the two different versions could be evaluated afterwords.

10.6 Function testing

10.6.1 Water test

At the start the pumps were run in water in order to easily test if they working as expected, figure 10.4Filling the tank with molasses is a time consuming process due to a small transfer pump that transfer the molasses from its original containers to the test tank. Molasses is a thick syrup which is not very pleasant to work with because its sticky and have a rather distinct smell. Therefore it was decided to test the pump as much as possible with water first, in case it needed adjustment of the design.

Figure 10.4: Function test in water.

10.6.2 Tolerances

In this early stage it was found out that the tolerances inside the pump was hard to achieve to get the pumps to operate properly. This was especially a problem on the internal version, which needed to be disassembled several times and re machined in order to be able to run it without internal touch. Later investigation, and fem analysis than showed that the lid was not strong enough to eliminate small displacement. These small deformation led to large enough displacement at the tip of the fluid separator which then yielded to touch with the pinion.

Also the roughness of the gears itself led to some problems. The gear were first cutout from a massive circular steel rod with water jet and then after wards it needed to be machined from to sides. The alignment when turning the gears after the first machining led to small error is in the alignment, and also the surface finish of the machining was not as good as desirable. Therefore it was hard to get the gear to run smoothly in start, this were fixed by manually sanding the surface at the places were it was needed.

10.7 Test procedure

The pump where supposed to be tested for five values for each input parameters and five times to eliminate external noise. This would yield to 100 test points as indicated in equation 10.3.

$$\begin{aligned} N_{tot} &= N_{parameters} \cdot N_{points} \cdot N_{times}[-] \\ N_{tot} &= 100[-] \end{aligned} \tag{10.3}$$

In an early test stages from the test runs in water and the problems with the tolerances, it was founded out that this would be difficult achieve due to time restrictions. In addition it was also founded out that the transferring of the molasses took longer time then expected, at one point the transfer pump went down as well, which even put the test further behind its schedule.

After some adjustment the pumps could be run in molasses as shown in figure 10.5.

Figure 10.5: Test run in molasses.

10.7.1 Heating

The hardest point to achieve was the five temperature points, since heating was time consuming process as shown by equation 10.4.

$$t = \frac{Cp \cdot \Delta T}{P_{heat}} \tag{10.4}$$

The pump could not be run in order to speed up the process of the heating, due to the high risk of breaking the pump when the viscosity of the molasses were high, temperatures below 30 °C.

Therefore it was agreed on running the test only for three points instead of five, 33, 36 and 39°C. By doing this it was possible to run test for all the temperatures during one extended working day.

When the test was started it was soon realized that the temperature increased during the test sessions. This then yielded to some unnecessary long waiting time between each test in order to allow the molasses to cool down. To avoid this a cargo heater was installed on the test rig, but instead of sending hot steam through it cold water was connected to it. After some testing of how to adjust the cold water valve a more constant temperature of the molasses could be achieved.

10.7.2 Oil flow

The oil flow to the hydraulic motor was in the beginning adjusted through a normal compensator, later this was exchanged to an adjustable compensator. This then meant that the hydraulic pressure needed to be adjusted when the position of the valve was changed in order to keep constant speed of the pump. This then allowed for setting the maximum oil flow the motor could get. The setting of the compensator was relatively easy to set accurate after some training with adjusting the compensator.

10.7.3 Hydraulic pressure

The hydraulic pressure was easy to adjust within a high accuracy of the desired value, the hydraulic pressure needed to be increased slowly due to a large time delay in the response of the system. This was partly due to an unknown system error, the diagnostic of this error was not proceeded due to high time pressure regarding the deadline of test period.

10.7.4 Back pressure

The valve was really hard to adjust, this mainly due to a high inaccuracy of the valve itself. An motor with belt to drive the valve was fitted, this made the valve easier to adjust for low pressure. Unfortunately the motor was not strong enough.

10.8 Test result

Since there were no direct method at current stadium to measure the volume flow and since this also required to pump from one tank to the other tank, the volume flow needed to be estimated using the formula obtained in chapter 5. This were done under the assumption that the leakage flow was low, and for current being it was only necessary to find out which version that should be further improved.

The efficiency can then be estimated by using equation 11.8.

$$\eta_{t_{pump}} \approx \frac{Q_{theory} \cdot \Delta p}{Q_{motor} \cdot \Delta p_{motor} \cdot \eta_{t_{motor}}}$$
$$\eta_{t_{pump}} \approx \frac{h \cdot \pi \cdot m^2 \cdot z \cdot \Delta p}{2 \cdot V_{motor} \cdot \Delta p_{motor} \cdot \eta_{t_{motor}}} [m^3/h] \tag{10.5}$$

The graph for the test when operated in molasses can be found in figure 10.6 for internal version and in figure 10.7 for the external version.

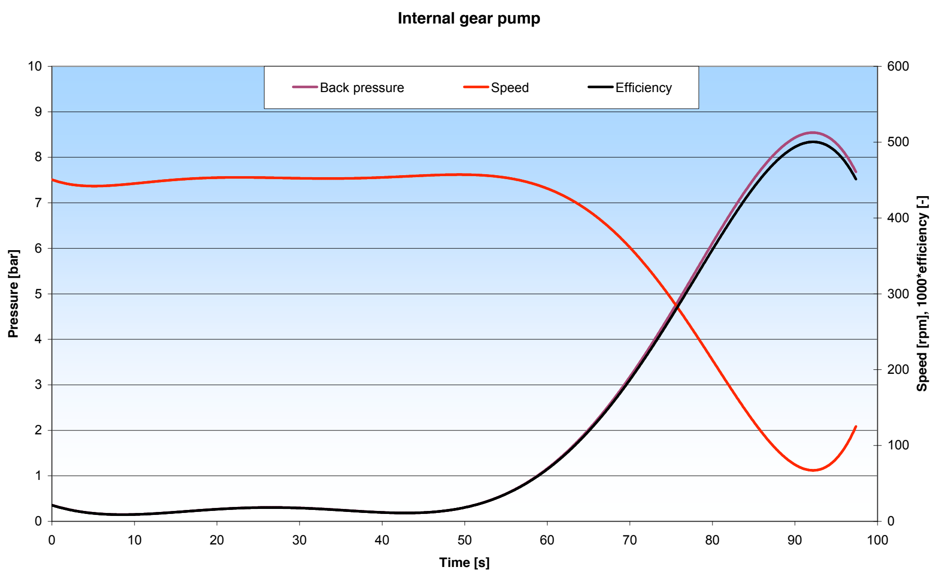


Figure 10.6: Test result for internal version, version 3

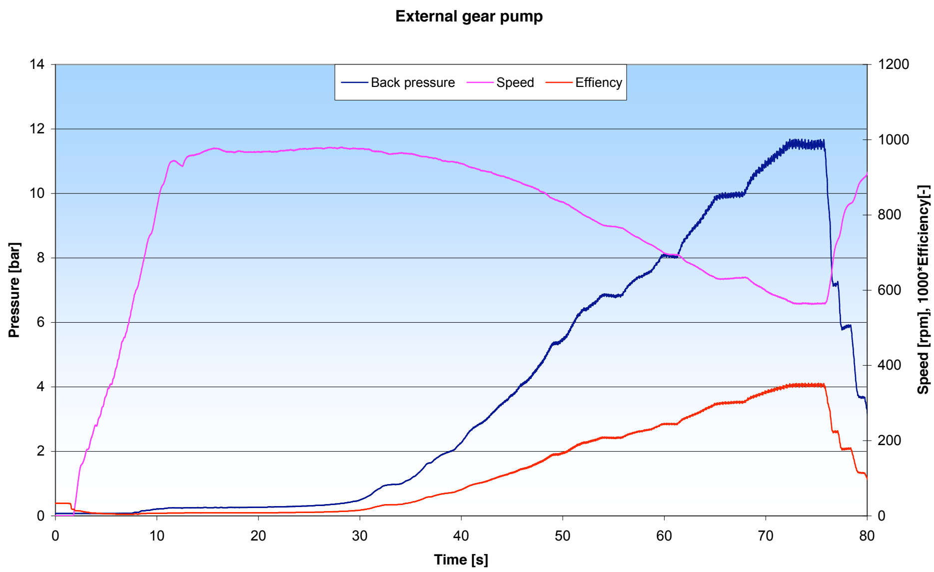


Figure 10.7: Test result for external version, version 5

From the graphs the efficiencies can be found, the internal version have its best efficiency, 50.7% , while the external version had its best efficiency, 35.3% . It is important to remember that these numbers does not take leakage flow into account and therefore the actual efficiency will be lower.

The maximum power was estimated to $4.8[kW]$ for the internal version and the external version $10.4[kW]$.

In addition it also worth mentioning that the external version had a considerably higher noise and vibrations, which might be an issue for the workers at the vessel ships.

10.9 Disassemble

10.9.1 Internal version

After testing the two version where disassembled for inspection, and for the internal version there had been touch between the gear and the wall, see figure 10.8.

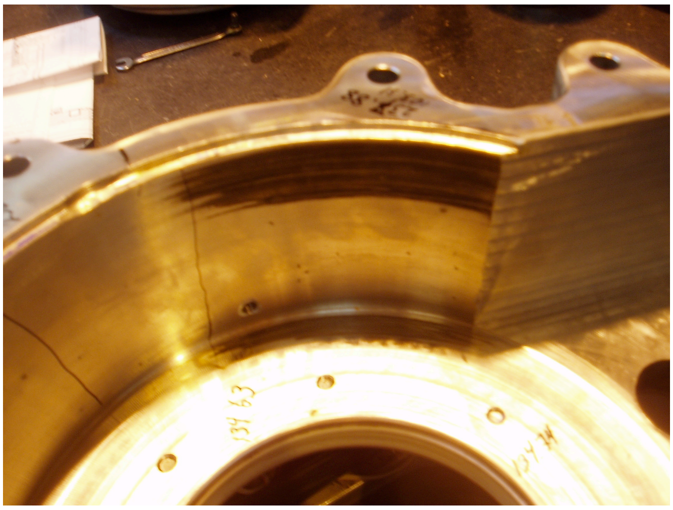


Figure 10.8: Touch between gear and housing.

Based on these observation an FEM analyses of the design were performed, see figure 10.9.

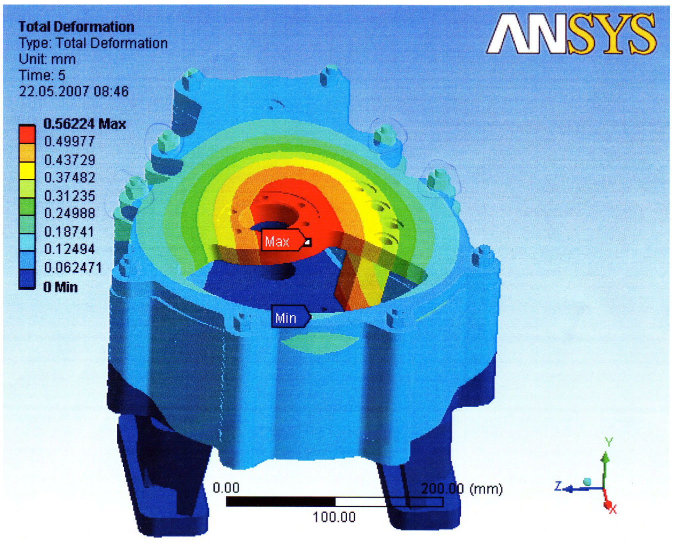


Figure 10.9: FEM analysis of the pump.

The shaft and the lid were to weak and needs to be changed on a new prototype to avoid touch.

10.9.2 External version

Dissemble of the external version had no sign of touch, on this version the bushing seemed to be a big issue. These had thin layer of crystallized molasses between the shaft and the wall of the bushing, figure 10.10.

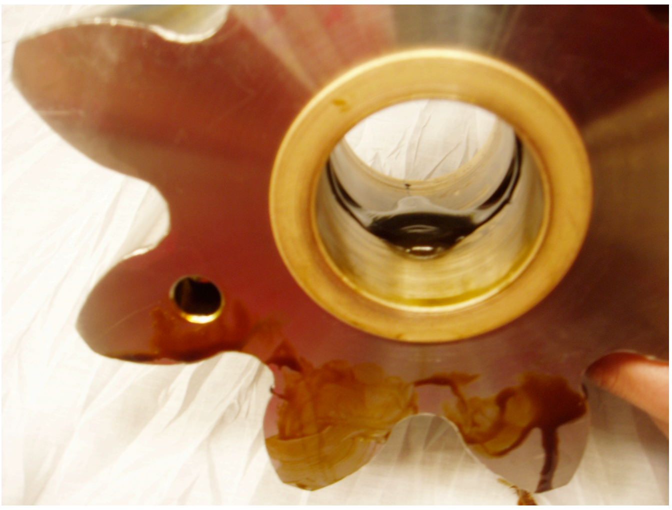


Figure 10.10: Bushing problems.

10.10 Conclusion

There were several problem with the tolerances with both version which meant a lot of adjustment, disassembling and assembling. This was a time consuming process, that was a bit frustrating. But after several adjustment the pumps were able to run smoothly in water.

When the pumps were tested in molasses new problem occurred due to increased forced compared to operation in water. These effects probably affected the results strongly together with small inaccuracy instrumentation.These effects could have been avoided if there were more time available.

A great amount of knowledge were obtained during this test and failure procedure, that was taken into account when the new version were developed. After an evaluation process it was decided to improve the internal version. It has to be assumed that the efficiency will be much better when the strength of the pump is increased in addition with better bearing system.

11 New improved version

11.1 Introduction

After been evaluating both internal and external gear pump regarding pumping of molasses it was assumed that internal gear pump would be the most suitable for pumping of molasses. There had encured several problems during the evaluation of the prototypes and therefore a new design, based on version 3 was necessary.

11.2 CFD

In order to find out how the fluid flows inside the pump a simplified model was evaluated in CFX for different parameters changes. This have been challenging, because even the simple model was complicated to get a high quality of the mesh with a proper boundary layer. A large mesh with close to 2.5 millions elements were evaluated.

The total pressure, velocity compensated pressure is shown in figure 11.1 and corresponding stream lines are shown in figure 11.2.

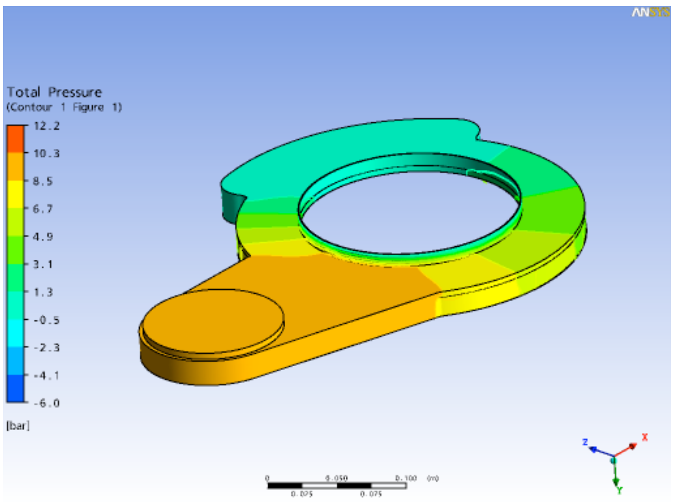


Figure 11.1: CFD model of total pressure.

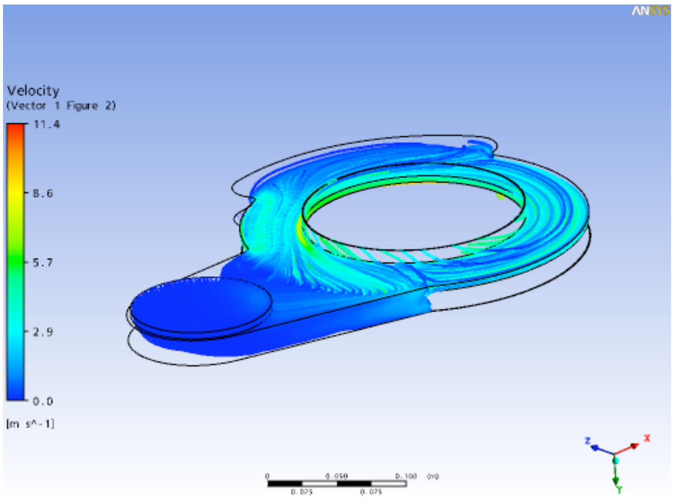


Figure 11.2: CFD model of stream lines.

11.3 Model

The new model were created in the same way as for the previous prototypes, with Pro Engineer and Multi. Some of the parts from the previous prototypes could be used for the new version which decreased the time of generating a new design. Some changes were made based on the experience from the prototypes, others from CFD calculations and some because of new ideas occurred. A draft of some of the changes are given in figure 11.3.

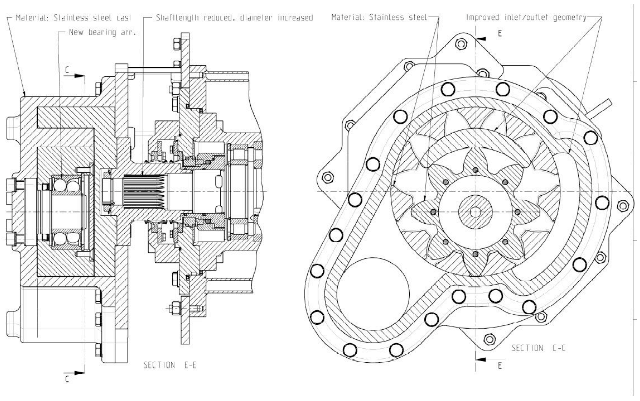


Figure 11.3: Improvements made to the internal version.

The new complete pump is shown in figure 11.4, here all the new bearing can been seen clearly.

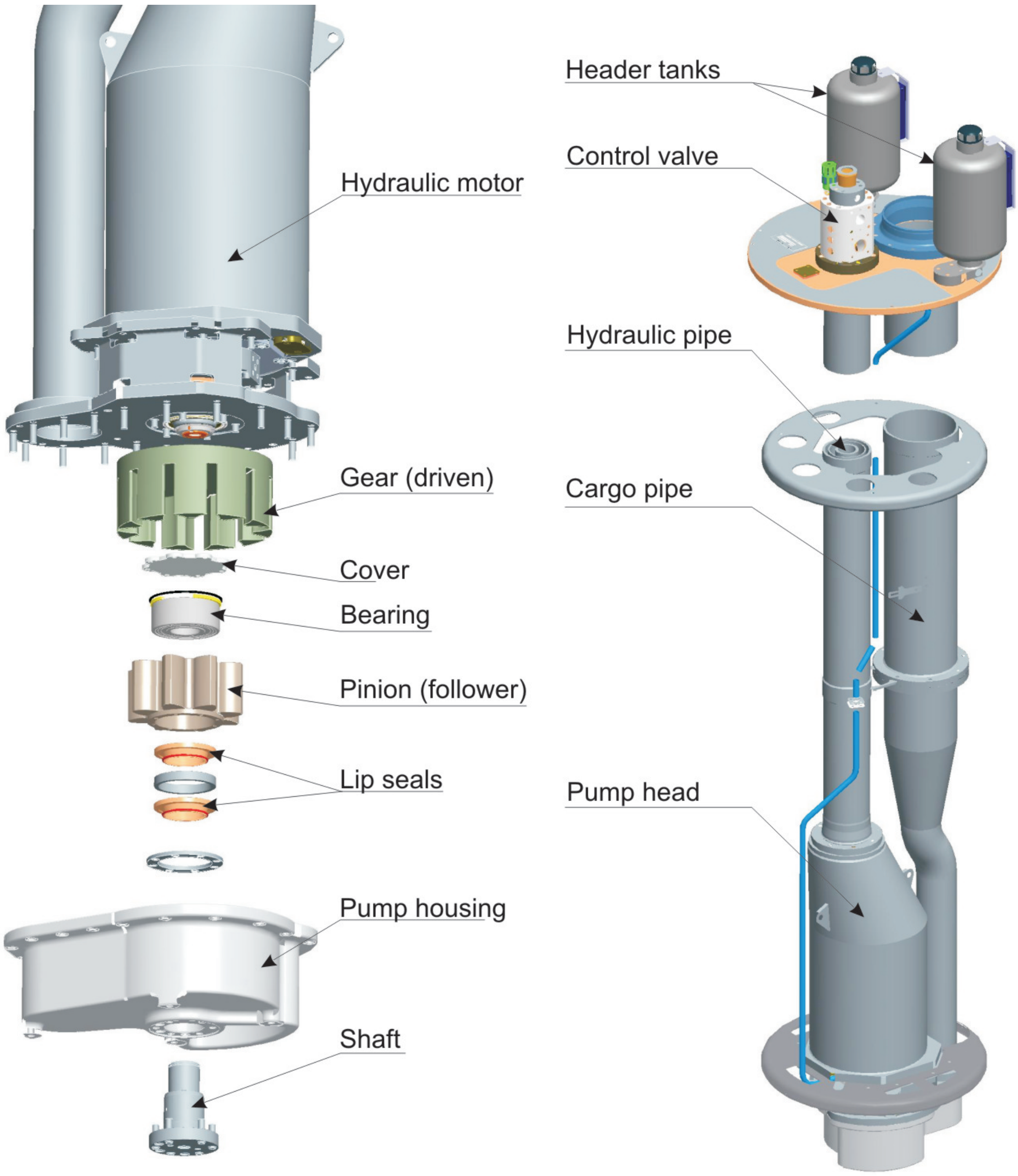
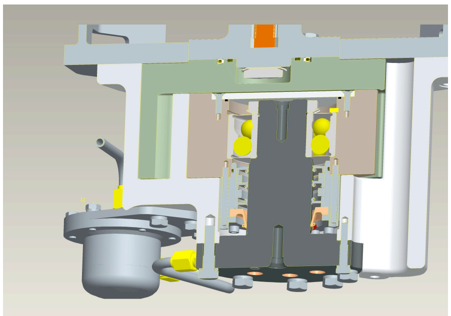


Figure 11.4: RP 200.

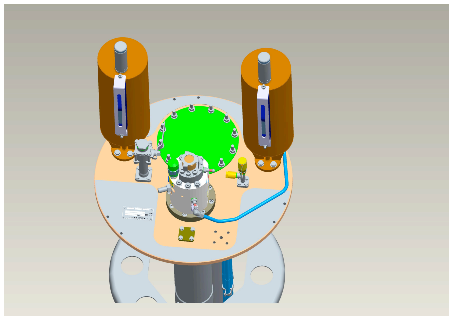
11.4 Manufacturing

The lid and shaft was strengthen as mentioned in chapter 9. The shaft got an ceramic coating which was done in England and the gears were now made in Denmark with a proper hub. After the gears had been fine machined at Frank Mohn Fusa A/S they were sent to the Netherlands to improve the surface roughness. This meant that the manufacturing is now dependent on several companies, but hopefully the quality are better on these specifics parts.

The new bearing system, which is shown in figure 11.5, meant several new parts. These part added some extra manufacturing time to the product, but the result is a good and study design.



(a) Mechanical seal and bearing suport

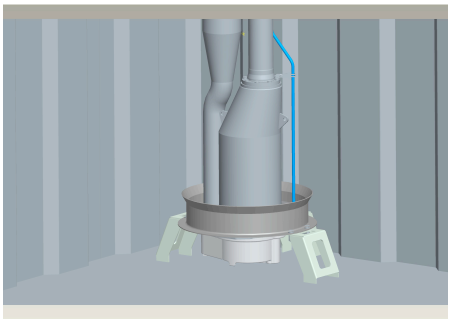


(b) Header tanks

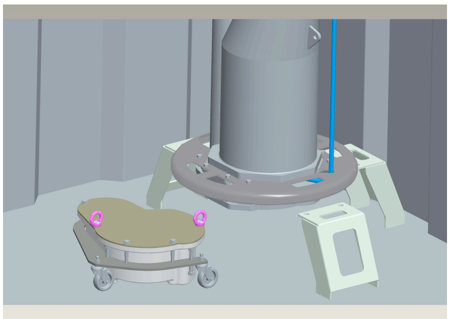
Figure 11.5: New bearing system

11.5 Handling of the pump

When the pump is not in use, it is necessary to disassemble the pump head so the vessel ship can use other cargoes as well. This were solved by making a small trolley for the pump head, which is illustrated in figure 11.6.



(a) Pump installed in tank



(b) Pump head disassembled

Figure 11.6: Handling of the pump

11.6 Testing

Before the new version were tested a cargo flow sensor were installed on the pipe stack, in addition to several new temperature sensors. By doing this more accurate test result could hopefully be obtained.

It is also desirable to find out the different efficiencies to understand more were the losses are, a long time test is also necessary before the product can be released in October 2008.

11.6.1 Long time test

In order to make sure that the pump remain functionality over a longer time span with start and stop a web camera was set up to monitor the run of the pump, figure 11.7.

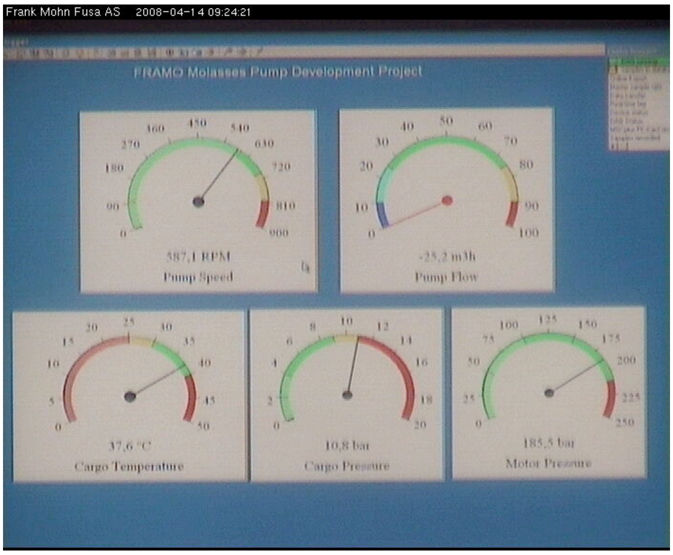


Figure 11.7: Long run test.

The new version have now run for more than 100 hours without any problems, after the new bearing system had been adjusted.

11.6.2 Volumetric loss

The volumetric losses can be found using the speed of the pump to obtain the theoretical flow without leakage, this means that the leakage flow then will be as shown in equation

$$P_{shaft} = \frac{V_{motor} \cdot N_{motor} \cdot p_{motor} \cdot \eta_{t_{motor}}}{60} [W] \tag{11.1}$$

$$\begin{aligned} Q_{leakage} &= Q_{theory} - Q[m^3/s] && (11.2)\\ P_{leakage} &= Q_{leakage} \cdot \Delta p[W] && (11.3) \end{aligned}$$

By using the formula given in equation 11.1 together with equation 11.3, the volumetric efficiency can be found in equation 11.4.

$$\eta_{v_{pump}} = 1 - \frac{P_{leakage}}{P_{shaft}} [-] \tag{11.4}$$

11.6.3 Mechanical loss

The mechanical losses can be approximated by leaving the back pressure totally open and run the pump in water. There will here be some error, because there will be some hydraulic losses connected to the mechanical losses. By implementing the formulas for the motor, equation 11.1 in combination with the small power produced by the pump, equation 11.5, the mechanical power can be found , equation 11.6. When the power is known then the efficiency can be found as well, equation 11.7.

$$P_{cargo} = Q \cdot \Delta p [W] \tag{11.5}$$

$$P_{mechanical} = P_{shaft} - P_{cargo} - P_{leakage} [W] \tag{11.6}$$

$$\eta_{m_{pump}} = \frac{P_{mechanical}}{P_{shaft}} [-] \tag{11.7}$$

11.6.4 Total efficiency

The efficiency can then be estimated by using equation 11.8.

$$\begin{aligned} \eta_{t_{pump}} &\approx \frac{Q \cdot \Delta p}{Q_{motor} \cdot \Delta p_{motor} \cdot \eta_{t_{motor}}} [-] \\ \eta_{t_{pump}} &\approx \frac{60 \cdot Q \cdot \Delta p}{V_{motor} \cdot N_{motor} \cdot \Delta p_{motor} \cdot \eta_{t_{motor}}} [-] \end{aligned} \tag{11.8}$$

11.6.5 Results

The test result are presented as normal for pumps, Q-H plot is given in figure 11.8, and shows that the volume flow only have small variations regarding changes in back pressure. This were as expected, White [1994]. For water the internal leakages becomes to dominant and the flow is no longer laminar and therefore this curve is different from the others.

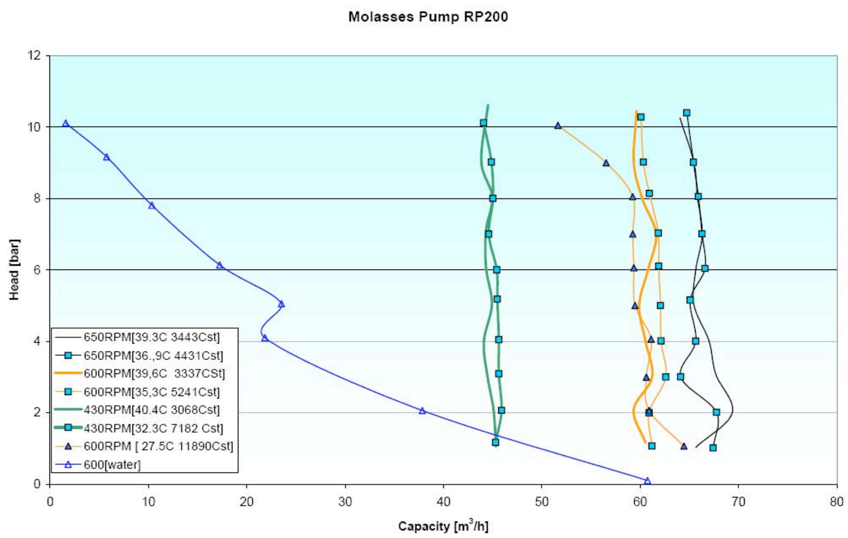


Figure 11.8: Q-H test result for the new version.

The Q-p_{motor} plot is given in figure 11.9, and shows the same effects as for the Q-H plot.

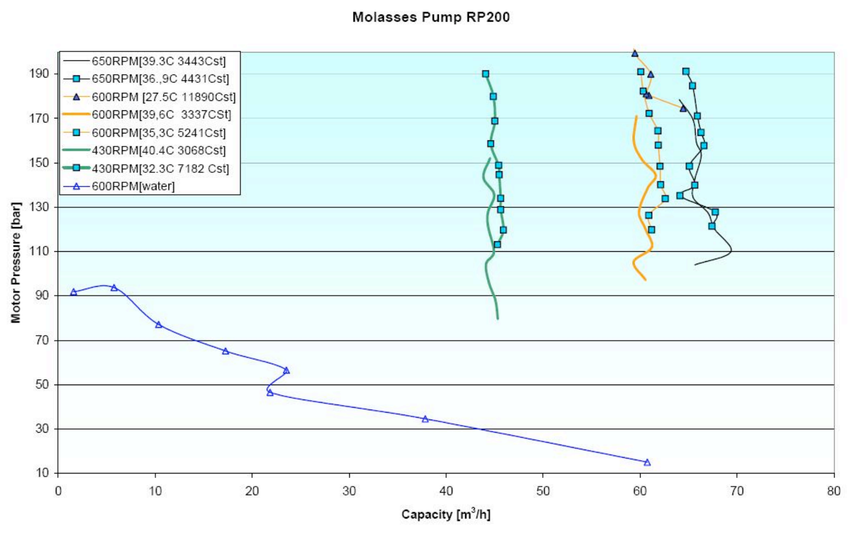


Figure 11.9: Q-p_{motor}test result for the new version.

The efficiency plot,figure 11.10 shows that a maximum efficiency will occur at a lower speed and an higher handling temperature of the molasses than in the customer requirements.

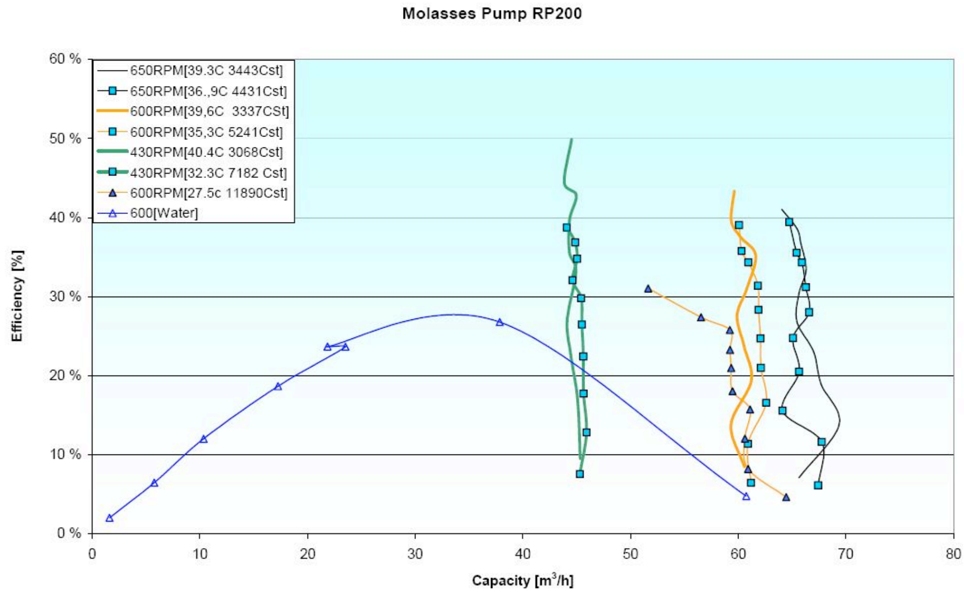


Figure 11.10: Efficiency test result for the new version.

Ideally this point should be in the design point, but the efficiency is only slightly lower for customer requirement point. These result are very promising with a maximum efficiency between 40 and 50%. This efficiency will hopefully be improved even further in a second generation of these pumps.

It is worth noticing that this efficiency is better than the one founded for version 3 and 5, since this efficiency have taken the leakage flow into account as well, but important is that the new version is able to reach the design point and even with a safety margin.

11.7 Errors

The results are more accurate for the new version than when evaluating for the version 3 and 5. Still the pump needs to be manually tested which means that there will be some inaccuracy. The added cargo flow sensor makes it now possible to separate the efficiencies, but there are still some issues regarding the flow sensor accuracy when measuring molasses. The largest error is still the inaccuracy in the temperature, to avoid this the heating cooling needs to be computer controlled, and the test should also have been run for liquids that are not so temperature dependent regarding the viscosity.

11.8 Conclusion

The pump now works fine over long time run, it runs smoothly and produces very little noise. The pump now fulfill the customer requirements, and customer are satisfied with the result.

There are still critics to how the test have been committed, a new test coordinator have started to work for Frank Mohn Fusa AS. This hopefully means that better temperature sensors can be installed, together with in flow viscosity-meter, cargo flow meter for the hydraulic motor and a new electrical controlled valve for the back pressure will then hopefully be installed. An automated testing procedure will make it possible to then get very accurate test result.

The new version does not require a 250ccm motor and will be installed with a 200ccm motor, which fits in the same pump housing. The name have therefore been decided to RP 200,(Rotary pump 200 cm^3). The design point will then yield to a motor pressure around 200 bar which then still allows for a safety margin.

12 Future work

12.1 Introduction

Even if the molassses pump has proofed its concept, there is still several issues that needs to be solved. The molasses pump is an ongoing project, which means that it has to be a lot of work in the future as well.

12.2 Production of the first series

Especially now when the product is put into production, there will probably be some issues that needs to be solve. Also it needs to be made jigs and production tool to increase the production efficiency. This was not needed for the prototype, but needs to be made now when there is closer to 50 pumps that is going to be produced before march 2009.

12.3 Improvement of performance

In order to improve the performance it is desirable to perform a full CFD analysis of the complete model. This is a complicated task, the mesh is already generated see figure 12.1.

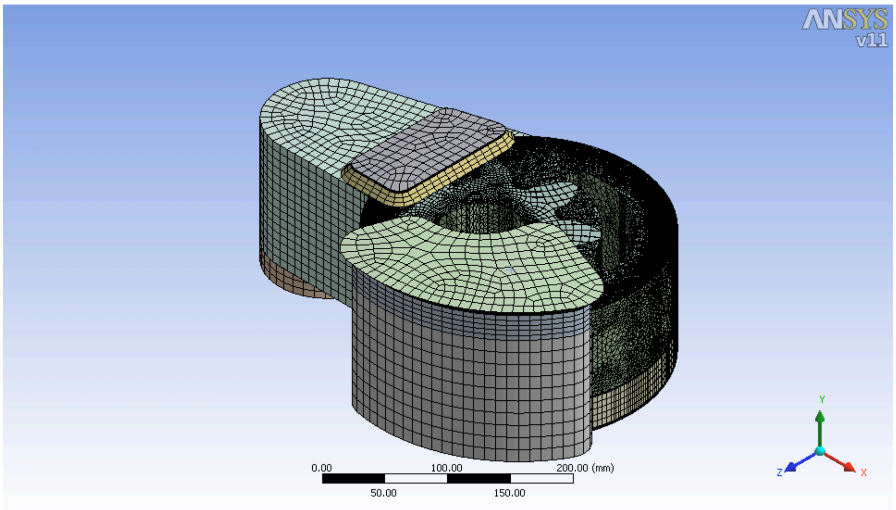


Figure 12.1: Mesh for CFD calculations.

The CFD calculation will be performed as soon as a new computer is installed, because at current stadium it is not enough memory to solve this complex model.

12.4 Documentation

There is many aspects that are omitted for this report, most of these aspect are documented in separated documents. It is desirable at some point to connect all these document together to make it easy to navigate between the different aspects. An operation manual for the pump is also necessary for the customer.

12.5 Software

The work of a new software to make it easier design gear pumps in the future has started. This software is connected with earlier software made for molasses, see Skåtun [2007]. It is for time being no clear choice which platform this software should be on.

12.6 Conclusion

There is a lot of work that still needs to be done, but the tough time limit is no longer present. This makes it easier to be motivated for continues work on this project. There are still some issues to be solved, but these issues will hopefully be solved quickly.

13 Conclusion

13.1 Molasses

The viscosity of the molasses is very dependent of temperature, therefore heating is necessary to ensure that the molasses viscosity is low enough to archive a good flow rate. Also the viscosity is very dependent from different load of molasses. Overall this means that the tolerances for the molasses viscosity need to be great, from 3000 to 7000cSt seems to be a reasonable range for pumping the molasses.

13.2 Gear pump

There are several PD- pumps available and selecting the correct type for the desirable flow, differential pressure and viscosity range that is required pumping molasses is a complex task.

Gear pump comes in two versions, internal and external gearing. The largest disadvantage with internal gear pump is the overhang of the gears which means that the gears will only have one supported bearing on one side of the gear. The advantage with external gearing is that the gears will have supported bearing on each side of the gears.

There were no clear solution of which gear pump that is the best pump design for pumping molasses, therefore, it was decided to make two different prototypes to test the concept.

13.3 Basic gearing dimensions

The module and number of teeth is the most important basic gearing parameters, because these parameters affect most of the other parameters. By varying these parameters it is possible to come up with many suggestions that will fulfill the limitations of the pump. It is therefore necessary to look into how these parameters affect other parameters than just the basic gearing dimensions.

13.4 Theoretical volume flow

In the development process it was very important to find an expression for the theoretical volume flow, the theory is complex to obtain this from basic geometry. Further investigation showed that this could be simplified with high accuracy just by using four parameters. This was very useful when selecting correct module and number of teeth. There is some fluctuation in the flow, due to uneven areas at different times at the outlet region. These fluctuations are small and will most likely not make any pressure shocks in the pipe stack.

13.5 Limitations

The customer have several requirements, they should all be achievable. The wish that there will only be small variation in the pump performance for large temperature changes might be hard to achieve.

The trunk diameter set several constraints on the sizes of the pump, this will probably reduce the efficiency of the pump, since then higher speed is necessary to obtain the same flow. There is most likely no point in making the gear as high as possible to reduce the speed of the pump, because it hard to fill high slim gears with molasses.

There seems to be no critical speed limits, as long as a bushing combined with a large shaft diameter is selected. Then speeds above 2000 rpm would be possible. The maximum differential pressure that pump can deliver has not be higher than $16[bar]$, which means that is necessary with either a mechanical or electrical safety valve.

13.6 Different proposal for design

Several different design proposal have been evaluated, and two of these have been put into prototyping. There were no clear choice between internal and external gear pump, each of them had their own benefits. The internal version are physically smaller, while the external version have a better bearing support for instance.

13.7 Performance estimate

It was harder to estimate the performance of the pumps than first thought. This was because of the complex shape with many tolerances affecting the performance of the pump. The performance estimate were divided into three different part, shear stress, leakage flow and bearings. No numerical value of how to set the parameters exact was obtained, but the connection between the parameters for a simplified case were found. This can then be used in the prototype design.

13.8 Prototypes

The two prototypes were made with only a few problems. The problems were mainly with time limits and the tolerances that were set on each part. Hardly any drawings was declined because of improper drawings, which was a major achievement. The reuse of the same pipe stack saved a lot of time in making prototypes and will also have benefits regarding testing of the prototypes.

13.9 Evaluation of prototypes

The different prototypes were tested at special tank, which showed its excellent performance for this task. The test tank needed several modification, but these modification will also be useful for other test in the future. The tests showed several problems that had not been encountered for under the design process. The bushing that were used turned out not to be useful at all. In addition the forces were much larger than expected on internal version which then led to touch between moving parts. The time consumption of performing the test were much longer than expected, and the way the test needed to be done were not ideal. This meant that no qualified data could be gained from these test, but a lot of knowledge on how to improve the design were gained. After an evaluation it was found that internal gear pump were most suitable to lower noise, and that this most likely will have a higher efficiency when the deign is improved.

13.10 New improved version

The new improved version fulfills the customer requirements, and it also possible to reduce the motor size to 200 cm^3. This was a great achievement after been working hard with this pump for more than a year continuously. This achievement were only possibly due to technology gained by the previous prototypes together with CFD calculations.

The new version has also proved its functionality over more than 100 hours, and the pump should no be ready for release on the international market even if the efficiency could have been higher.

13.11 Final words

The paper has fulfilled the aims even though there were a tight time limit at the end. It has been interesting working with the problem situation, which make motivation for further work possible. There has been some minor problems during the work of the paper. It has to be admitted that it has been tough in some periods when combining work, school and customer inspection. Now its possible to look back and see that the product idea that started almost two years ago is ready for the international market. There is still motivation to improve the current design and to follow up the customer and the production of RP 200. This will be an ongoing process in the next few years and it will be interesting to see which way the market will take.

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