UNIVERSITETET I AGDER

Title

Rotating tools for quick connection of drill pipe (Spinner)

By

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Subject area
Mechatronics

This Master's Thesis is carried out as a part of the education at the University of Agder and is therefore approved as a part of this education. However, this does not imply that the University answers for the methods that are used or the conclusions that are drawn.

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Abstract

In this project a rotating tool for quick connection of drill pipe has been designed. It satisfies the following three operation cases:

First, is to grip the different sizes of drill pipe.
Second, is to spin of the upper joint of drill pipe.
Third, is to make up and break out a torque.

In addition the connecting system of drill pipe is dimensioned and set up as a simulation model.

The following have been accomplished:

Product specification and generation of concepts.
Dimension of the hydraulic system.
Selection of hydrauilc component.
Selection of mechanical component .
Synchronization of two hydraulic motors.
Synchronization of two hydraulic cylinders.
Design and FEM analysis of the spinner.
Building a prototype of the spinner.

The spinner is designed to handle all normal ranges of drill pipe from 3˵ up to 10˵. The motor with gearing system ensures that the spinner is able to make up and break out at a torque of 7700N.m with a speed of 93 rpm for 10˵drill pipe and a torque of 2309N.m with a speed of 313 rpm for 3˵drill pipe. This high torque is higher than that which can be found at the present, and this may eliminate the need for the torque wrench as the gap between the drill pipes will be reduced or possibly eliminated. The main purpose is to reduce operating time and compact the tool.
The spinner includes the four hydraulically synchronized driven rollers to give a better grip on the drill pipe. The machine made to provide optimal grip and permit limited rotation along the vertical connection axis. This rotation allows the rollers to adjust onto the drill pipe as they clamp onto it providing optimal grip.

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Nomenclature

F	=	Force
F_{thp}	=	Theoretical periphery force, Pinion
F_{thw}	=	Theoretical periphery force, Gear
F_{cl}	=	Clamping force
P_b	=	Equivalent dynamic bearing load
σ_{bp}	=	Bending stress, Pinion
σ_{bg}	=	Bending stress, Gear
σ_{op}	=	Contact stress, Pinion
σ_{og}	=	Contact stress, Gear
T_s	=	Spinning torque
T_m	=	Motor moment
N_{sp}	=	Spinning speed
n_m	=	Motor speed
Cr	=	Chrome
Ni	=	Nickel
Mo	=	Molybdenum
MF	=	Main Function
SF	=	Sub Feature
DS	=	Different Solutions
LVDT	=	Linear Variable Differential Transformer
CVG	=	Control Valve Group
DCV	=	Directional Control Valve
CNC	=	Computer Numerical Control

List of Contents

Chapter 1

1. Introduction

Connecting pipe down the well bore on a drilling rig is a tough, hazardous job and typically done in a dangerous working environment best done by unmanned tools to avoid casualties.

The traditional way of connecting pipe on drilling floor was by manual labourer called *Floorhands* [9]. Of the 2 *Floorhands*, the *lead tong hand* or "worm" operates the break out tongs used to break apart the threaded connections of the drill pipe. The *chainhand*, who is one of the *floorhands*, operates the "make-up tongs" which are used to retighten and torque the connection. These activities are the reason for many human accidents and casualties. The solution to avoid these problems is to remove the traditional human roughnecks, and replace them with machines operated from a safe location.
The automated connection and disconnection of a wide variety of drill pipe can be as efficient as if done by human labor. This machine will be safe and reliable.
Furthermore a machine failing or not being able to operate up to the specified requirements can cause the rig to come to a halt. A small breakdown will have a large economic impact. More important than economic aspect is that by replacing manual labour with machines the safety of the rig workers will be maintained.

NOV (National Oilwell Varco) is one of the leading worldwide providers of drilling equipment to the oil and gas industry producing equipment to drill ships, semi submersibles and fixed installations. One such equipment is the roughneck. A NOV Roughneck consists of two parts, the upper part is called spinner and the lower part is called the torque wrench. The spinner spins the upper joint in/out of the drill pipe and then the torque wrench makes up torque to give a better connection of the drill pipe. The upper part (spinner) cannot give enough torque to spin the drill pipe.
To spin the drill pipe in/out, the spinner must have a high grip force and a high torque to be able to spin the drill pipe. This is solved by using two motors with gearing system.

A spinning wrench engages the stem of the upper joint of the drill pipe and spins the upper joint of the drill pipe until it is connected/ disconnected from the lower joint.
This tool allows the drill pipe to be spun up to get the smallest possible gap between the pipes.
The spinner is hydraulically operated and it incorporates a roller drive.

This work has the following primary objectives:

- Calculation of the hydraulic and the mechanical system.
- The design part includes the dimensioning of the mechanical and the hydraulic circuit of the griper and the spinner design system.
- The virtual design and simulation of a spinning wrench.
- Design of the spinner wrench and verification of the structural calculation by using *Solid Works Simulation*.
- To develop and produce the prototype of the spinner machine.

Chapter 2

2. Spinner wrench

2.1 Product specification

The product specification shall respond to most issues related to the spinner machine, and should be used as a memo. The product specification table shows the product *demand* as “must” and the product *desire* as “should”.
The spinner needs to fulfill certain requirements as listed in (table 2.1). The main demand is to achieve spinning torque as high as possible with an acceptable speed to possibly eliminate the toque wrench of the roughneck. The tool must handle drill pipe with diameter of $2\ ^1/_2{}'' - 10''$ and should be designed as compact as possible.

Product Specification		
	Must	Should
Function:		
• Spin in/out the drill pipe	X	
Functional properties/ demands:		
Process:		
• Spinning velocity: min. 0-100 [rpm]	X	
• Spinning torque: min. 8000 [N.m]	X	
Product:		
• Lifetime: 5 year	X	
• Reliability	X	
• Accuracy	X	
• Complexity (Less component)	X	
• Robust and fast reconstruction		X
• Low maintenance operation	X	
• Small size and low weight		X
Service:		
• PLC control panel	X	
Boundary conditions:		
Spin the pipe:		
• $(2\ ^1/_2{}'' - 10'')$	X	
• Thread movement 0-200mm	X	
• Interface, Hydraulic pipe and signal cable	X	

Product Specification		
	Must	Should
Project plan:		
• Project start: 13.01.2011	X	
• Design: 4 weeks		X
• Simulation and calculation 6 weeks		X
• Prototyping: 4 weeks		X
• Hand over report : 31.05.2012	X	
Cost:		
• Development cost: Not specified		
(Dis)Assembly/ Manipulation:		
• Change of roller	X	
Standard:		
NS-EN 1993-1-1 Design of steel structures- General rules	X	
NS-EN 1993-1-8 Beregning av knutepunkter og forbindelser	X	
Safety:		
Safe to avoid human and material damage	X	
Environment:		
Withstand harsh environment (outdoor)	X	

Table 2.1 – Product specification

2.2 Function alternatives

2.2.1 Objectives

To generate dozens of concepts, and find a unique solution to this challenge, a concept tree has been made. This section describes function alternatives with the selected option that has been used for the spinner wrench. By defining the design problem in terms of a Main Function (MF) the construction work will start. The complex design problem will be split into manageable pieces in terms of Sub Function (SF).

2.2.2 Approach

The main function (MF) of the spinner wrench is to make up and break out drill pipe. The necessary functions to be able to spin the different size of pipes are divided into Sub Features (SF). Different Solutions (DS) to each Sub Feature is presented in (Figure 2.1)

The main function table needs to be made to generate the different concept.

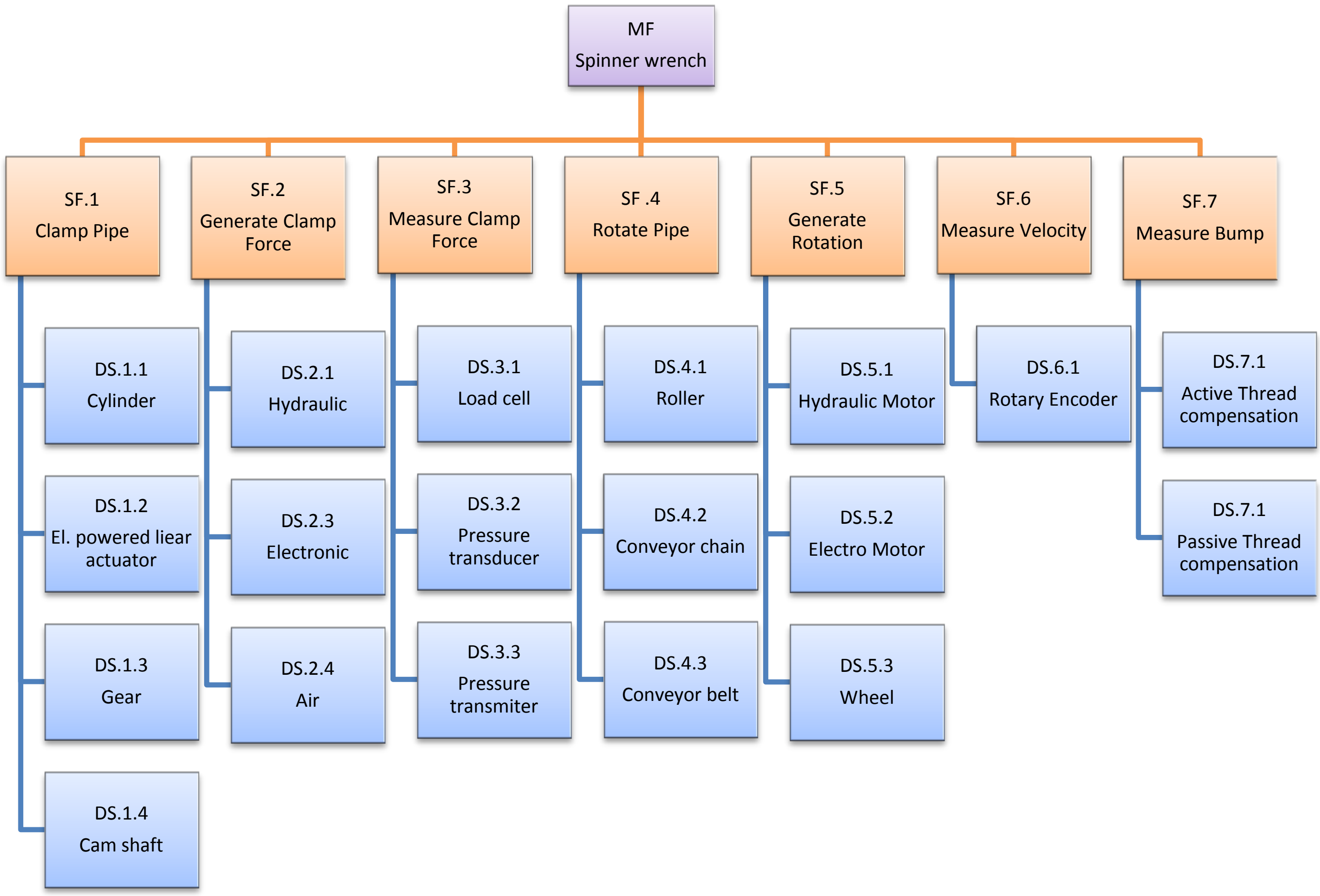


Figure 2.1 – Function alternatives

2.3 Partial Solution description and grading

2.3.1 Objectives

The spinner structure is constructed according to the features that the spinner will perform. To understand and evaluate of each sub function it is necessary to define the most independent sub function which is possible to generate different solutions.
In the following tables (table 2.2 – 28) each different solution to each sub function is described and evaluated. Furthermore, each different solution has been graded from 0 to 10 points where 10 is the best grade. In addition the best solutions are marked with (*).

SF1: Clamp Pipe:

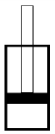
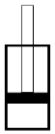
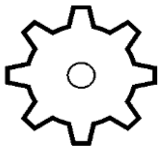
Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS.1.1	Cylinder		• High force • Long stroke • Do not require other movable parts	• Need Hydraulic power unit • Need hydraulic Control valve	8	*
DS.1.2	Linear Actuator		• Long stroke • Do not require other movable parts	• Low force • Expensive • Need cyclo-inverter/ PWM	4	
DS.1.3	Gear		• Reliable	• Require motor • wear and tear	5	*
DS.1.4	Cam Shaft		• Reliable	• Short stroke • Low force	1	

Table 2.2 - Partial solution description and grading of Sub Feature 1

SF2: Generate Clamp Force:

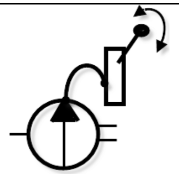
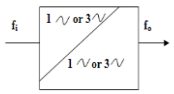
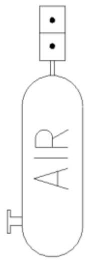
Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS.2.2	Hydraulic		• No manual labour • Accurate force	• Requires power • Requires help of mechanic	9	*
DS.2.3	Electric		• No manual labour • Can be used in high power application	• Require power • Complex (electronic device)	8	*
DS.2.4	Air		• No manual labour	• Requires power for compressor • Requires help of mechanic • Not accurate force	5	

Table 2.3 - Partial solution description and grading of Sub Feature 2

SF3: Measure Clamp Force:

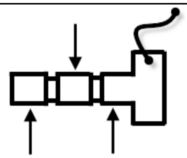
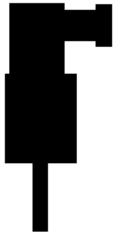
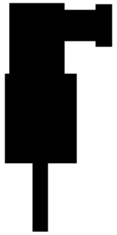
Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS.3.3	Force and Load cell		• Integrated signal conditioning circuitry. • Standard output signal 4-20 mA	• Requires mechanical accommodation to place the load cell in the structure	5	
DS.3.2	Pressure transmitter		• Feature additional reset and calibration options • Calibrate the dumping of the output signal • Integrated signal conditioning circuitry. • Standard output signal 4-20 mA	• Not interchangeable between different manufacturers	8	*
DS.3.1	Pressure transducer		• Changes the physical variable pressure into a quantity that can be processed electrically • Interchangeable between different manufacturers	• Gives an unamplified signal requiring signal conditioning circuitry	7	*

Table 2.4 - Partial solution description and grading of Sub Feature 3

SF4: Rotate Pipe:

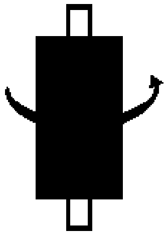
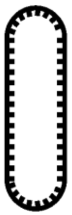
Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS.4.1	Roller		• Reliable • Flexible rotation • Easy to change • Better grip • Do not require other movable parts	• wear and tear	7	*
DS.4.2	Conveyor chain		• Reliable • Makes noise	• Require other movable parts • wear and tear	4	*
DS.4.3	Conveyor belt		• Easy to use • Reliable	• Require other movable parts • Not safe • wear and tear	3	

Table 2.5 - Partial solution description and grading of Sub Feature 4

SF5: Generate Rotation:

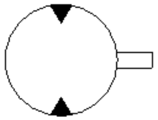
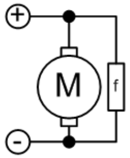
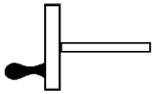
Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS. 5.1	Hydraulic motor		• No manual labour • Reliable	• Requires power	8	*
DS.5.2	Electric motor		• Reliable • Full torque at 0 RPM • No transmission is required • Power configurations can be done in software	• Heavy • Requires power • Inability to operate at low speed	7	*
DS.5.3	Wheel		• Reliable	• Manual labour	0	

Table 2.6 - Partial solution description and grading of Sub Feature 5

SF6: Measure Velocity:

Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS. 6.1	Rotary Encoder		• Reliable • Output is a binary value proportional to the angle of the shaft • Supplies a certain number of pulses for each shaft revolution.	•	8	*

Table 2.7 - Partial solution description and grading of Sub Feature 6

SF7: Measure Bump:

Solution	Name	Figure	Advantages	Disadvantages	Grade [1/10]	Relevant solution
DS. 7.1	Active thread compensation		• Safety and performance • Cost saved due to not needing a physical prototype	• Complex • Needs Hardware-in-the-loop	8	*
DS.7.2	Passive thread compensation		• Safety and performance • Simple	• Needs hydraulic accumulator	8	*

Table 2.8 - Partial solution description and grading of Sub Feature 7

2.3.2 Approach

To generate a different idea it is necessary to generate several concepts which there are also used different solutions for the various partial functions.
The best graded alternative solution for each sub feature in the previous tables, has been chosen and evaluated as an acceptable basis to generate the different concept**.**

- SF1: Clamp Pipe
  - DS.1.2 - Hydraulic cylinder
  - DS.1.4 - Gear

- SF2: Generate Clamp Force
  - DS.2.2 - Hydraulic
  - DS.2.5 - Electronic

- SF3: Measure Clamp Force
  - DS.3.1 - Pressure transducer
  - DS.3.2 - Pressure transmitter

- SF4: Rotate Pipe
  - DS.4.1 – Roller
  - DS.4.1 – Conveyor chain

- SF5: Generate Rotation
  - DS.5.1 – Hydraulic motor
  - DS.5.1 – Electro motor

- SF6: Measure Velocity
  - DS.6.1 – Rotary Encoder

- SF7: Measure Bump
  - DS.7.1 – Active thread compensation

2.4 Concepts

A design concept is more specific than an idea, but less specific than a layout or drawing of a product. Usually, concepts are best described by an annotated sketch.

A full concept for a product defines all the key features, functions and characteristics of a design. The concept of the spinner captures the essential purpose of the product, with enough detail that all other engineers can work out the product.

To generate the spinner wrench concepts the solutions found suitable in the evaluation of the partial solution (ref. Table 3.1 to 3.10) has been used. Below there are three concepts generated to make spinner equipment:

For concept 1, the spinner wrench machine consists of two hydraulic cylinders equipped with rollers in the rod end. These cylinders are mounted to a U-formed frame (Fig. 2.2). The object of using a hydraulic cylinder on each side is to provide a firm clamp in order to keep the drill pipe in the well center. The holder of the rollers is formed to adapt to a variety of drill pipe dimensions.

The cylinders are actuated by means of a HPU and control valve, while the motor can either be actuated by a HPU and a control valve, or electrically. The force and velocity are measured by a pressure transmitter and rotary position sensor respectively.

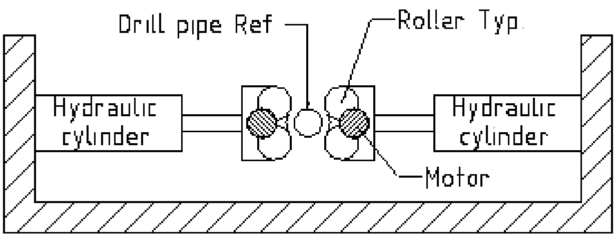


Figure 2.2 – Schematic connection of concept one

For concept 2, the spinner wrench machine consists of two gears equipped with rollers on the end of retention brackets that are mounted on the sprocket side, (Fig. 2.3). Both gears and rollers are driven by a hydraulically or electrically actuated motor. The force and velocity are measured by a pressure transmitter and a rotary position sensor respectively.

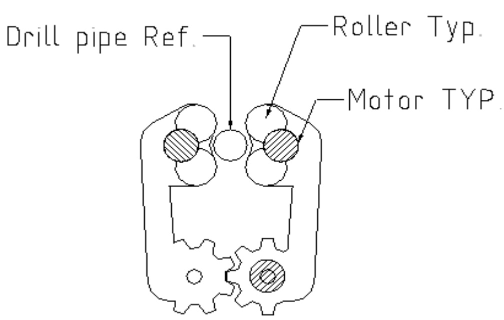


Figure 2.3 - Schematic connection of concept Two

For concept 3, the spinner machine utilizes a hydraulic cylinder to grab onto the drill pipe, and a chain drive system to rotate it. The hydraulic cylinder is connected at the far end of two interconnected frames (Fig. 2.4). These frames can be described as elongated and asymmetrical with a rectangular shape on one side and a bulge in the middle of the other side. The bulged part of the two frames is connected to each other using a bolt allowing rotation of both frames around the longitudinal axis of the bolt.

On the opposite side of where the cylinder is mounted, one roller is placed on each frame. While gripping onto the drill pipe, these rollers will be in contact with it. A third roller is mounted on the motor axle and a fourth smaller roller is mounted on the right hand side above the motor for the purpose of tightening the chain.

During operation the spinner machine is moved into position (Fig. 2.4) and the cylinder extrudes pushing the far end sides of the frames resulting in a motion which clamps the drill pipe between the two rollers. This clamping mechanism is reminiscent of how scissors work. With the drill pipe firmly in contact with the chain, the motor will run the chain drive and thus rotate the drill pipe. The force and velocity are measured by a pressure transmitter and rotary position sensor respectively.

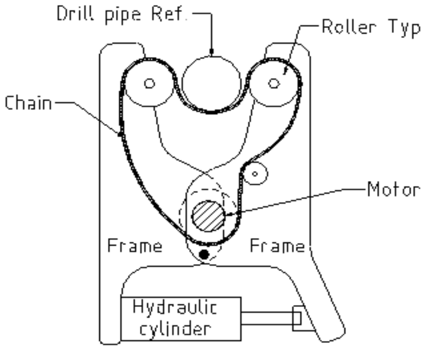


Figure 2.4 - Schematic connection of concept three

2.5 Evaluation of concepts

The different concepts were evaluated with regard to function and construction.
The concepts were given points from 1 to 5 for prioritized criteria's where 1 represent "not acceptable" and 5 represent "very good".

Evaluation of Function:

Criteria evaluation	**Concept 1**	**Concept 2**	**Concept 3**	**Ideal**
Reliability	4	3	3	5
Complexity	4	3	2	5
Frame rigidity	4	3	4	5
Adaptivity	4	2	4	5
Sum:	16	11	13	20
Relative value:	**0.80**	0.55	0.65	1

Table 2.9 – Function evaluation

Concept 1 scored best in the function evaluation.

Evaluation of Construction:

Criteria evaluation	**Concept 1**	**Concept 2**	**Concept 3**	**Ideal**
Size	4	4	2	5
Stiffness	4	3	3	5
Assembly	4	3	3	5
Sum:	8	7	5	10
Relative value:	**0.80**	0.70	0.5	1

Table 2.10 – Construction evaluation

Concept 1 scored best in the construction evaluation.

Concept one scored best in each case, so the **"Concept One"** will be chosen.

After the evaluation is finished and the concept is selected, the construction work will continue by varying the structure and shape to achieve the most optimal solution and design. The change of structure and design will be described in the next section.

2.6 From concept to final design

Overall this design was chosen due to its compactness while in storage and its ability to provide optimum grip whilst providing all the necessary functionalities for connecting drill pipes.

The final design incorporates the best features of the previously proposed concepts into one compact and functional design. This design include the 4 hydraulically driven rollers from concepts 1 and 2, as this setup was deemed to give better grip on the drill pipe. Like the other two concepts the motor is placed on top of two rollers driving two gears attached to the rollers themselves.

In both ends of the rollers two heart-shaped plates are attached. The plates are then mounted on retention brackets with bearings to permit limited rotation along the vertical connection axis. This rotation allows the rollers to adjust onto the drill pipe as they clamp onto it providing optimal grip.

The retention brackets in this design are movable by two hydraulic cylinders. This saves space compared with concept 1 due to it being able to retract and thus decrease its overall width. The brackets are mounted on rods which function as tracks allowing the brackets to move linearly with the cylinders' motion. The rods are a part of the supporting structure for the whole machine and are stationary.

During operation, the main structure, as seen in Figure 3.5 and as described above, moves into position with the retention brackets in the wide open position. Then the clamping procedure starts whereby the hydraulic cylinders retract to grip onto the drill pipe. Sensors will feedback the force to a controller in order to determine when the rollers are firmly in contact with the drill pipe. The rollers will now start spinning and sensors will again feedback torque data to a controller for the motor. When a certain level a torque is reached, the motor stops, and the retention brackets are brought out again into the wide open position.

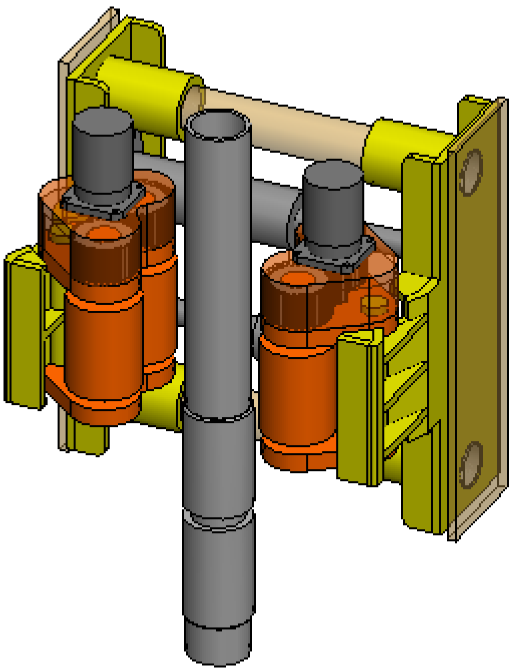


Figure 2.5 - Final design

2.7 Choice of components

After choosing the final concept the following components were chosen.

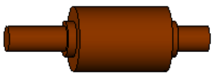
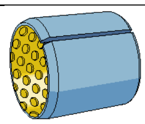
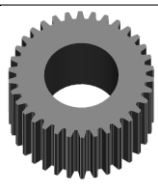
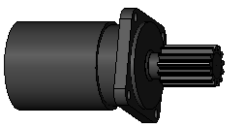
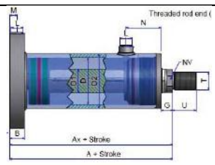
Name	Description:	Image:	Qty.
Roller	Stainless Steel		4
Pressure transmitter [1]	Pressure transmitter		2
Roller Bearing [6]	Bearing series number 22313 CC		8
SKF dry sliding bearing [13]	M material 2mm wall thickness, 220mm length		4
Gear	15 CrNi 6		4
Hydraulic Motor with Pinion [12]	OMT 315 cm^3 from Sauer Danfoss		2
Hydraulic Cylinder [11]	Cylinder LHA25 B/G 110mm from pmc servi		2

Table 2.11 – Choice of Components

Chapter 3

3. Hydraulic System

Hydraulic systems can be found today in a wide variety of applications, from small assembly processes to integrated steel and most aircraft use hydraulics in the braking systems and landing gear. Hydraulic systems use an incompressible fluid, such as oil or water, to transmit forces from one location to another within the fluid. Hydraulics enable the operator to accomplish significant work (lifting heavy loads, turning a shaft, drilling precision holes, etc.) with a minimum investment in mechanical linkage through the application of Pascal's law, which states:
"Pressure in a confined fluid is transmitted undiminished in every direction, and acts with equal force on equal areas, and at right angles to the container walls". [2]
Static analyses are first conducted for the configuration of the hydraulic system. This gives a quick overview of system performance and component dimensions. The calculations will also serve as references during the dynamic analysis in which simulation software is used. The aim of the dynamic analysis is to validate the dimensions calculated in the static analysis.

3.1 Hydraulic System Components

The hydraulic system consists of two hydraulic cylinders, servo valve, flow divider and two hydraulic motor which is driven by hydraulic supplier existing on the rig.

A hydraulic cylinder is a mechanical actuator that is used to give a unidirectional force through a unidirectional stroke. The hydraulic cylinder consists of a cylinder barrel, in which a piston connected to a piston rod moves back and forth.

A hydraulic motor is a mechanical actuator that converts hydraulic pressure and flow into torque and angular displacement (rotation). The hydraulic motor uses pressurized fluid to rotate the active drum in this project to compensate heave motion and hoisting the payload to the seabed.

To assure the same speed of two hydraulic systems (actuators or motors) independently of their loads, the flow dividers [7] must be used. Flow dividers split flow in one direction according to some ratio, into two flow paths independent of pressure in each path and combine flow in the opposite direction.

An electro-hydraulic servo valve is an electrically operated valve that controls how hydraulic fluid is ported to the hydraulic motor. Hydraulic valves are used in a system to start, stop and direct fluid flow to compensate for the load disturbance and keep the regulated process variable as close as possible to the desired velocity and torque.

A gear is a rotating machine part having cut teeth, which mesh with another toothed part in order to transmit torque. To achieve a desired torque to spin the drill pipe, the spur gear has been used.

A bearing is a machine element designed to fix, guide or hold moving parts and to reduce friction. Bearing allow constrained relative motion between two or more parts, typically rotation or linear movement. The spinning and clamping drill pipe, gives both radial and axial load applied on bearing.

3.2 Static Analysis

The static analysis begins with the hydraulic system configuration fitted for the mechanical system. Since the variable in the mechanical system is determining the required force to clamp the pipe, the hydraulic performance configuration is found according to these. This chapter is about the dimensioning and modeling of these elements.

The method of analysis is using the required torque which is given in the spinner specification, to calculate and determine components fitted for this configuration in following order gear, bearing, hydraulic cylinder(s), hydraulic motor(s) and servo valve(s). Calculation equations are used from a hydraulic compendium [2]. This section will briefly describe the calculation of these components.

To determine the size of the hydraulic motor D_M, calculation of the motor displacement $D_{M,min}$ (cm^3/rev) is needed. The equation below is used to determine this component:

$$D_{M,\min} = \frac{2 \cdot \pi \cdot M}{\Delta \mathrm{P}} \qquad 3.1$$

To determine the diameter of the hydraulic cylinder, calculation of the cylinder piston area A(m^2) is needed. The equation bellow is used to determine this component:

The necessary cylinder piston area:

$$\mathrm{A} = \frac{F}{\Delta \mathrm{P}} \qquad 3.2$$

The necessary cylinder diameter:

$$d = \sqrt{\frac{4\mathrm{A}}{\pi}} \qquad 3.3$$

The size of the servo valve is determined by calculating the needed motor flow, Q_M (l/min). The motor flow is calculated by the following calculation:

$$Q_M = \frac{D_M \cdot n_M}{\eta_{vM}} \qquad 3.4$$

Where: n_M : The motor speed (rpm)

η_{vM} : Volumetric efficiency of the motor

The servo valve is chosen according to motor flow. The choice is CVG30 (300 l/min). The valve will operate with a valve pressure (P, A, B, LS) 350 bar. The valve has *spool position control* (LVDT).

3.2.1 Calculation and dimension of spur gear

To achieve a desire torque of 8000N.m and spin the different size of drill pipes, it is necessary to use gear to transfer the torque. The roller which has been used to spin the different size of drill pipes has a diameter of 164mm, so the outside diameter of gear should not exceed this diameter. To design a suitable gear which adapts a roller, the gear module has assumed to 4mm.

3.2.1.1 Geometrical conditions of spur gear

The gear module, number of tooth of the pinion and number of tooth of gear wheel has been assumed to (4mm, 13 and 34) respectively, and according to gear compendium capital 9. [3], the dimension of pinion and gear was found.

Modul	$m = 4mm$	
Width factor	$\lambda = 18$	
Tooth width	$b = \lambda \cdot m$	$b = 72mm$
Number of tooth of pinion	$Z_1 = 13$	
Number of tooth of wheel	$Z_2 = 34$	
Circular Pitch	$p = m \cdot \pi$	$p = 1.57mm$
Addendum	$h_a = 1.0m$	$h_a = 4mm$
Dedendum	$h_b = 1.0m$	$h_b = 4mm$
Tooth high	$h = 2 \cdot m$	$h = 8mm$
Circular pitch	$S_n = \frac{P}{2} - 0.05m$	$S_n = 6.083mm$
Pitch diameter of the pinion	$d_{P1} = m \cdot Z_1$	$d_{P1} = 52mm$
Outside diameter of the pinion	$d_{O1} = m \cdot Z_1 + 2 \cdot h_b$	$d_{O1} = 60mm$
Root diameter of the pinion	$d_{R1} = m \cdot Z_1 - 2 \cdot h_b$	$d_{R1} = 41mm$
Pitch diameter of the wheel	$d_{P2} = m \cdot Z_2$	$d_{P2} = 136mm$
Outside diameter of the wheel	$d_{O2} = m \cdot Z_2 + 2 \cdot h_b$	$d_{O2} = 144mm$
Root diameter of the wheel	$d_{R2} = m \cdot Z_2 - 2 \cdot h_b$	$d_{R2} = 128mm$

3.2.1.2 Spur gear tooth bending stress

There are two types of high stress on the teeth, the tensile and compressive stresses due to bending of the tooth. The compressive stress has a greater magnitude due to the radially inward component of the tooth force F_t . The bending stress is cyclic as it occurs once per revolution of the gear and will, thus, lead to a potential fatigue failure.

The Lewis formula has been used to calculate the bending stress.
According to table 4.6 from compendium (MASKINKONSTRUKSJON II) [4], the overload factor is:

Over load factor $k_a = 1$

According to table 4.8 from compendium (MASKINKONSTRUKSJON II) [4], the tooth factor is:

Tooth factor pinion $\gamma_p = 2.9$

Tooth factor wheel $\gamma_g = 2.6$

When a gear wheel is rotating the gear teeth come into contact with some degree of impact. To allow for this a velocity factor (k_v) is to be find.

Pitch line velocity $\upsilon = 2.52 \frac{m}{s}$

According to table 4.6 from compendium (MASKINKONSTRUKSJON II) [4], the drive factor is:

Drive factor $\mathrm{A} = 10 \frac{m}{s}$

Dynamic factor $k_v = \frac{\mathrm{A} + \upsilon}{\mathrm{A}}$ $k_v = 1.25$

The motor type OMT 315 cm^3 from Sauer Danfoss [5] see page 36, has been chosen. This motor has a max. output torque of 950 Nm and 1140 Nm intermittent operation.

Motor moment $T_m = 950 N \cdot m$

Motor speed $n_m = 380 rpm$

Radius of drill pipe $R_{pipe} = 127 mm$

Radius of the wheel $R_g = \frac{d_{P2}}{2}$ $R_g = 68 mm$

Radius of the pinion $R_P = \frac{d_{P1}}{2}$ $R_g = 26 mm$

Theoretical periphery force, pinion $F_{thp} = \frac{T_m}{2 \cdot R_P}$ $F_{thp} = 18269N$

Theoretical periphery force, wheel $F_{thw} = \frac{T_m}{2 \cdot R_P}$ $F_{thw} = 18269N$

Bending stress pinion $\sigma_{bp} = \frac{F_{thp} \cdot k_a \cdot k_v}{b \cdot m} \cdot \gamma_P$ $\sigma_{bp} = 230.3Mp$

Bending stress gear $\sigma_{bg} = \frac{F_{thw} \cdot k_a \cdot k_v}{b \cdot m} \cdot \gamma_g$ $\sigma_{bp} = 206.5Mp$

According to table 4.12 from compendium (MASKINKONSTRUKSJON II) [4], the bending stress of material 15 CrNi 6 is:
Limited bending stress $\sigma_{b\lim} = 460Mp$ Hence OK

3.2.1.3 Spur gear contact stress

There is a contact stress situation as the two, approximately cylindrical surfaces roll and slid on each other during tooth contact. This stress may lead to a surface fatigue of the tooth. To control the contact stress that occur on contact surface between to cylinders, Hertzian theory has been used

Leading edge factor $f_c = 1.76$

Material factor of cooperative tooth $f_w = 271\sqrt{\frac{N}{mm^2}}$

Tooth ratio $i = \frac{Z_1}{Z_2}$

Contact stress pinion $\sigma_{op} = f_w \cdot f_c \sqrt{\frac{F_{thp} \cdot (i+1)}{b \cdot d_{P1} \cdot i} \cdot k_a \cdot k_v}$ $\sigma_{op} = 1386Mpa$

Contact stress gear $\sigma_{og} = f_w \cdot f_c \sqrt{\frac{F_{thw} \cdot (i+1)}{b \cdot d_{P2} \cdot i} \cdot k_a \cdot k_v}$ $\sigma_{og} = 827Mpa$

According to table 4.10 from compendium (MASKINKONSTRUKSJON II) [4], the drive factor is:
Lubricating factor $K_L = 1.1$

According to table 4.11 from compendium (MASKINKONSTRUKSJON II) [4], the drive factor is:
Speed factor $Z_v = 0.877$

Contact stress for unlimited live time for material 15 CrNi 6, pinion $\sigma_{o\lim.p} = 1900Mp$

Contact stress for unlimited live time for material 15 CrNi 6, pinion $\sigma_{o\lim.g} = 1900Mp$

Safety factor $V_o = 1.25$

Allowable contact stress pinion, gear $\sigma_{ap,g} = \frac{\sigma_{thw}}{V_o} k_L Z_v$ $\sigma_{ap,g} = 1466.4Mpa$

The contact stress for both pinion and gear is lower than the allowable stress Hence OK

3.2.2 Calculation of spinning torque and speed

The required torque to spinn the drill pipe is 8000N.m,

Spinning Torque $T_s = 4 \cdot T_m \cdot \frac{R_g}{2 \cdot R_p} \cdot \frac{R_{pipe}}{R_{rolle}}$ $T_s = 7696.3N \cdot m$

Spinning Speed $n_{sp} = n_m \cdot \frac{R_p}{2 \cdot R_g} \cdot \frac{R_{rolle}}{R_{pipe}}$ $n_{sp} = 93.81N \cdot m$

3.2.3 Calculation of clamping force

From the maximum torque for the largest pipe dimension, the necessary force to clamp the drill pipe could be found.

Friction coefficient, lubricated surfaces $\mu = 0.16$

Clamp force $F_{cl} = \frac{T_s}{R_{pipe} \cdot \mu}$ $F_{cl} = 348.75kN$

3.2.4 Calculation and dimension of cylinder

To determine the cylinder diameter equation 3.2 – 3.3 respectively has been used. We assume the pressure to be 250bar.

Pressure $\mathrm{P} = 250bar$

Areal of cylinder $\mathrm{A}_c = \frac{F_{cl}}{\mathrm{P}}$ $\mathrm{A}_c = 0.015m^2$

Cylinder diameter $d_c = \sqrt{\frac{4 \cdot \mathrm{A}_c}{\pi}}$ $d_c = 139mm$

From standard cylinder data blade pmc servi [10]

Piston length $\mathrm{A}_x = 243mm$

Cylinder Threaded head $U = 75mm$

To clamp the different drill pipe the cylinder must be able to stroke 275mm

Cylinder Stroke $S_t = 275mm$

The overall cylinder length at open position can determine by:

Overall cylinder length $Z = A_x + 2 \cdot S_t + U$ $Z = 868mm$

The size of the cylinder is large compared to the spin structure.

To minimize the length of cylinder which can provide a clamping force of 378kN, two cylinders with diameter of 110mm must be used.
From standard cylinder data blade pmc servi [10]
New piston diameter $d_{cnew} = 110mm$

Rod diameter $d_R = 56mm$

Area piston side $A_{cP} = \dfrac{\pi \cdot d_{cp}^{\ 2}}{4}$ $A_{cP} = 0.01m^2$

Area rod side $A_{cR} = \dfrac{\pi \cdot (d_{cp}^{\ 2} - d_{cR}^{\ 2})}{4}$ $A_{cR} = 0.01m^2$

The clamping force on drill pipe can be calculated by a combination of force from piston and rod side

Force piston side $F_{cP} = A_{cP} \cdot P$ $F_{cP} = 237.6kN$

Force piston side $F_{cR} = A_{cR} \cdot P$ $F_{cR} = 176kN$

Force acting $F_{clamp} = F_{cP} + F_{cR}$ $F_{clamp} = 413.6kN$

The new dimension of the cylinder will be as follows:

Piston length $A_x = 217mm$

Cylinder Threaded head $U = 56mm$

The cylinder stroke should remain constant 275mm

The new overall cylinder length at open position will be:

Overall cylinder length $Z = A_x + 2 \cdot S_t + U$ $Z = 823mm$

3.2.5 Calculation of bearing

There are two types of loads acting on roller during the spinning operation Radial and Axial load. A radial load: is a load producing by clamping force and applied perpendicular to the shaft axis. An axial (thrust) load: is a load producing by the weight of the spinner and applied parallel to the shaft axis. The bearing must resist these loads.

From rolling lager compendium (RULLELAGER) Kapittel 7 [5]

Theoretical life time	$L_{10h} = 1200$ drive hour	
Shaft speed	$n_s = 94 \frac{\text{O}}{\text{min}}$	
Life time (mill.rev)	$L_{10} = \frac{L_{10h} \cdot 60 \cdot n_s}{10_6}$	$L_{10} = 67.68$
Number of bearing	$N_b = 8$	
Number of carry bearing	$N_{ba} = 4$	
Weight of Spinner assumed to 1000kg	$W_s = 1000kg$	
Design load	$F_s = W_s \cdot 9.81$	$F_s = 10kN$
Radial load component	$F_r = \frac{F_{cl}}{N_b}$	$F_r = 47.34kN$
Axial load component	$F_a = \frac{F_s}{N_{ba}}$	$F_a = 2.5kN$
Relation between Axial and radial load	$R_{a,r} = \frac{F_a}{F_r}$	$R_{a,r} = 0.053$

According to table spherical roller bearing [6] for bearing with 65mm inner diameter:

Relation between Axial and radial load	$e = 0.35$
Bearing radial factor	$X = 1$
Bearing axial factor	$Y_1 = 1.9$

From equation of equivalent dynamic bearing load for *spherical roller bearing* [6]

Equivalent dynamic bearing load	$P = F_r + Y_1 \cdot F_a$	If $\frac{F_a}{F_r} \le e$
	$P = 0.67 \cdot F_r + Y_2 \cdot F_a$	If $\frac{F_a}{F_r} \rangle e$

In this case $\frac{F_a}{F_r} \leq e$ $\qquad P = 52.1kN$

Dynamic carry number $\qquad C = P \cdot \sqrt[3]{L_{10}}$ $\qquad C = 212.3kN$

To choose the bearing which can resist this dynamic load and not have a larger diameter than the roller, *Double row spherical roller bearing* should be used. From SKF bearing data sheet we choose bearing series number 22313 CC which has the outer diameter of 140mm, inner diameter of 65 and C = 253kN. The term consisting of five layers numbers describing the series, where the two last indicates the inner diameter x5 and after codes indicate the bearing characteristics.

3.3 Dynamic Analysis

The aim of the dynamic analysis is to be able to analyze the system under dynamic conditions. This allows the validation of the initial component dimensions from the static analysis. Wherever important discrepancies occur, the design must be adjusted for by increasing or decreasing component dimensions.

Specific objectives in this section include:

- To use the desired torque to spin the drill pipe
- Set up a simulation model to test the hydraulic system
- Validate the initial dimensions found in the static analysis

The approach to analyzing the system is using simulation software called SimulationX. A *Simulation X* model is set up based on the dimensions calculated by hand. Having done this, simulations of model are performed to see if the components calculated in the static analysis hold up.

3.3.1 Simulation model

This section is about creating the *Simulation X* model of the hydraulic motor which is controlled by a proportional directional control valve(s). The purpose of the hydraulic circuit is to choose the suitable servo valve to operate the motor, which gives the required torque to spin the upper joint of drill pipe. To do so the motor(s) and cylinder(s) must operate according to the command signal of the proportional directional control valve CVG. Figure 3.1-a illustrates these circuits.

In the simulation model of the motor, the model has been simplified by using a flow source and a variable throttle valve instead of the CVG valve as shown in Figure 3.1. The highest and lowest required torque of 7696N.m and 2309N.m respectively to spin the drill pipe is modeled by a source component. The gar ratio is also modeled for the largest and smallest diameter of drill pipe. The control signal block control the flow in to the flow source and throttle valve respectively

The flow through the valve can be calculated by equation 3.1

$$Q_A = C_d \cdot A_d(u) \cdot \sqrt{\frac{2}{\rho} \cdot \Delta p} \qquad 3.1$$

Where

C_d : Discharge coefficient

ρ : Density

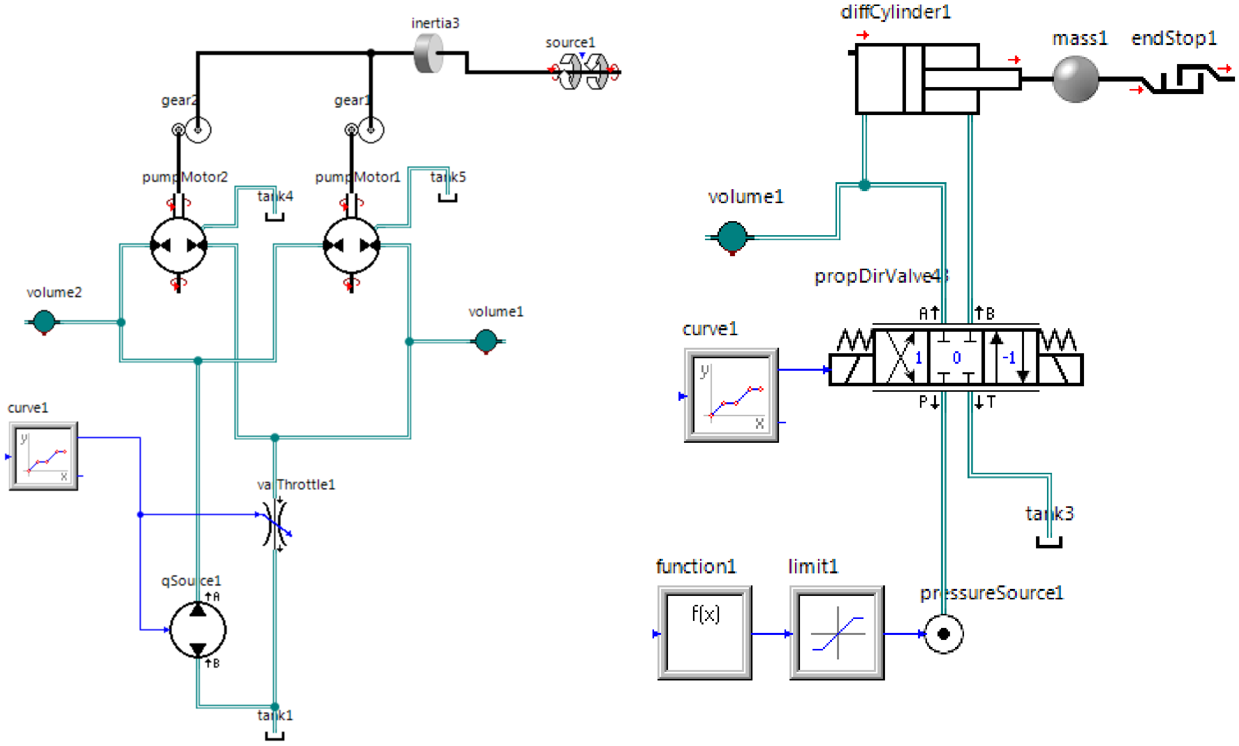
ΔP : Pressure drop across the valve (54, compensator spool) [8]

A_d : Discharge area

(u) : Control signal

In the simulation model of the cylinder seen in Fig. 3.1-b, the CVG has been modeled in an alternative way by using a 4/3-way directional control valve (DCV) and making sure the pressure drop over it is constant which also ensures a constant flow through it. This is done by sending a command to the pressure source (“pressuresource1”) to always be 3 bar above the cylinder piston side pressure. This command is written in the “function1” block and sent through the “limit1” block which limits the pressure command from 0 to 210 bar which is the maximum pressure delivered by the actual pump. In this case we don’t have to model the return orifice separately since it is modeled within the 4/3-way directional control valve.

The elements required for these circuits are shown in table 3.1-3.2



a) b)

Figure 3.1 – Hydraulic circuit of motor in Simulation X using a) a flow source, and b) 4/3-way DCV.

The element required for the hydraulic circuit is shown in table 4.1:

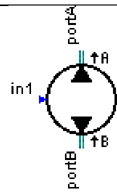
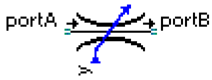
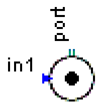
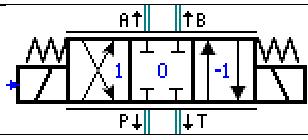
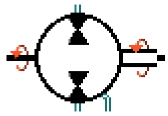
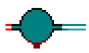
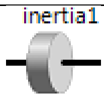
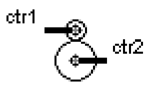
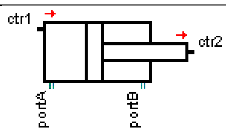
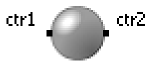
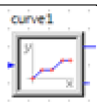
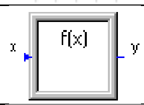
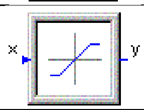
Component	Library name	Symbol
Tank	Hydraulics/ Basic Elements	
Flow Source	Hydraulics/ Basic Element/ QSource	
Variable Throttle Valve	Hydraulic/valves/flow valves/ VarThrottle	
Pressure Supply	Hydraulic/ Basic Element/ Pressure Source	
Proportional Directional Control Valve	Hydraulics/ Valves	
Constant Displacement Pump/Motor	Hydraulics/ Actuators	
Volume	Hydraulics/ Basic Elements	
Inertia	Mechanics/ Rotational Mechanics	
gear	Mechanics/ Rotational Mechanics/Transmission	
Differential Cylinder	Hydraulics/ Actuators	
Mass	Mechanics/Translation	
End Stop	Mechanics/Translation	
Curve	Signal blocks/Signal Sources	
Function	Signal blocks	
Limit	Signal blocks/NonLinear. Limit	

Table 3.1 – The element required for the hydraulic circuit.

The desired parameters are mostly taken from the data catalogues, however there are also some assumptions. The parameters that are to be input in the model are set in table 4.2. The figures of each component refers to able 4.1

Component	Parameter input
Pressure Sources	Set a pressure to 600 bar
Proportional Directional Control Valve	- In the slecetion box Stroke Signal select Normalized Signal The Normalized Signal means that the valid range of the input signal must be from -1 to +1 and at an input signal of zero the valve will be in its center position. - In the Dynamics Options set the Consider Dynamics to true - In the dynamics Parameter set the Natural Frequency to 60 Hz - In the Q(y) function dialog box the Type of Edges must be select to Identical Edges. - Set the Pressure Drop at Valve edge to 35 bar, and density of 0.85 g/cm^3 - The Flow per Stroke must have 500 l/min It means that the valves have a flow of 500 l/min at a pressure drop of 35 bar *at one signal edge* for full valve opening stroke.
Constant Displacement Pump/Motor	- From Geomery dialog page set the displacement volum to 315 cm^3 - From Friction dialog box set the Hydro-Mechanical Efficiency to 92% - From Leakage dialog box set the Volumetric Efficiency to 92%
Flow Source	The maximum flow is 250 l/min. Because SI-units are required when entering an expression, typ in: Volume Flow –self.in1*250/6e4
Variable Throttle valve	In the Paramters dialog box, choose: Flow description, Reference measurement - Reference flow 680 l/min - Presure Drop 10 bar - Stroke signal –self.y
Volumes	The volumes are not required in SimulationX. However a pump volume of 0 would resualt in an infinitely fast pressure change, which is not realistic. The purpose of using the volum is to considerate of the compressible fluid behavior in a motor and valves. Since there is no data about the hydraulic hoses . The volume are set to between 1 and 2 dm^3.
Inertia-inertia1	In the Paramters dialog box, type in: - Moment of Inertia : 1 kgm^2 The value represent motor inertia. This value are displayed in the data sheet.
Gear	In the Paramters dialog box, type in: Constant Ratio (ctr1/ctr2) i_max 4.05, min 2.42
Cylinder	In the Paramters dialog box, type in: Maximum Stroke 290mm Pisto Diameter 110mm Rod Diameter 56mm
Mass	In the Paramters dialog box, type in: Mass 100
End Stop	In the Paramters dialog box, type in: Stop 1: 280mm Stop 2:1000m
Curve	In the Paramters dialog box choose: Simulation Time t [s]
Function	In the Paramters dialog box, type in: Volum2.p+1.8e5
Limit	In the Paramters dialog box, type in: Lower Rage Limit, x min: 0 Upper Range Limit, x max: 210e5

Table 3.2 – Component parameter for the hydraulic circuit

3.3.2 Simulation Results

The result in fig 3.2 shows that at the maximum torque to spin the largest drill pipe, the CVG valve must adjust to 36% to give enough oil flow to the motor. At the minimum torque the valve must adjust to 100% to give enough oil flow to the motor.

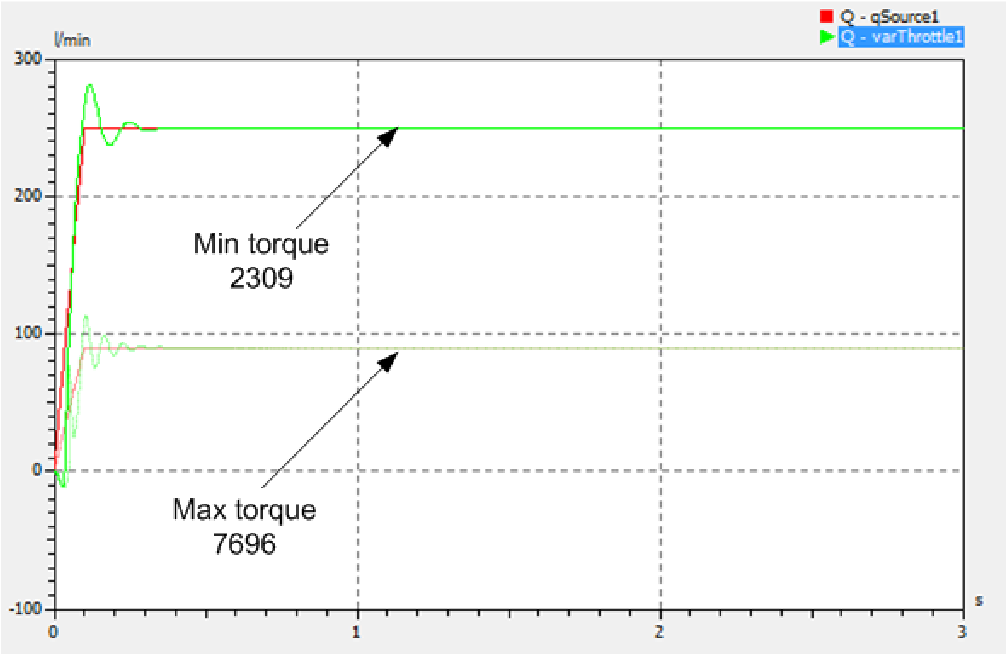


Figure 3.2 – Flow source and variable throttle valve

The result in fig 3.3 shows that to achieve the maximum clamping force it is necessary to use a servo valve which has a flow rate of 300l/min. The velocity will be 0.45m/s

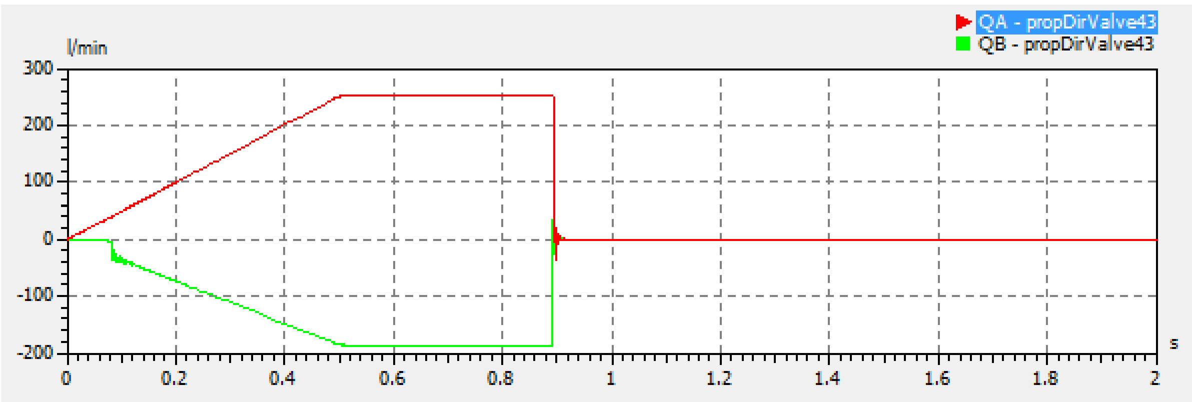


Figure 3.3 – 4/3 Directional control valve

Chapter 4

4. Control System

The control system is responsible for the clamping and spinning of the drill pipe. To do this, it must control the servo valve in the hydraulic circuit. This chapter will briefly describe the control of the main function of the spinner.

4.1 Utility Control

The oil supply to the spinner is available on the rig and it is enabled by powering the shutoff valve. The principle is that when the valve is tripped, the flow is quickly stopped and an indicator tells the operator that the electrical circuit has been opened by a failure somewhere in the system. To de-energize the shutoff valve, the emergency stop relay is needed.

4.2 Control of CVG Proportional Valve

The philosophy of electro hydraulic actuation is integration of electronics, hydraulics, sensors and actuators into a single unit that interfaced to the spool of the proportional valve.

The valve has a closed loop control system [8] as shown in figure 5.1. To ensure the system behaves as desired. A sensor captures the actual system output and this value is subtracted from the desired reference value to create the error signal that is used in the controller to adjust system input.

The proportional actuators feature an integrated feedback transducer that measures spool movement in relation to the input signal, and by means of a solenoid valve bridge, controls the force, position, speed and direction of the main spool of the valve.

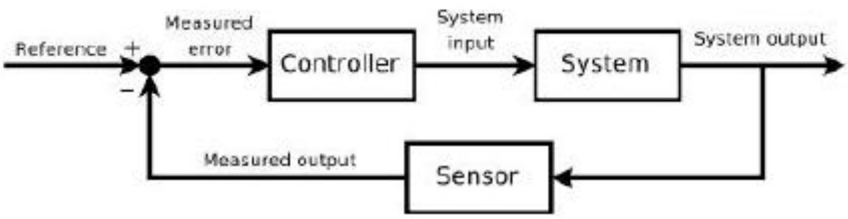


Figure 4.1 – Closed loop architecture

In closed loop valve position of main spool is measured by LVDT. Position signal is send to amplifier card that compares position signal to command signal and corrects the control signal to solenoids accordingly.

4.3 Control of Hydraulic Motor

To spin the drill pipe clockwise and counter clockwise, two hydraulic motors have been used. The spinner should run at a constant speed regardless of pipe type in automatic mode for the defined number of seconds for spin in/out. To assure the same speed of two motors independently of their loads, the flow dividers must be used.

To follow the drill pipe being rotated in or out, a gas accumulator must be used. The motor should be engaged once both clamps are on, and spinner rotate clockwise or counter clockwise.

For two hydraulic motors rotating at the same speed and directions a flow divider is needed. The circuit in Figure 4.2 shows a flow divider synchronizing two hydraulic motors. As the motors turn in right-hand rotation, they stay almost perfectly synchronized. Pressure to each motor may vary but flow from each flow-divider outlet remains near constant. If the directional control valve shifts to turn the motors in left-hand rotation, the flow divider may get equal flow and the hydraulic motors may stay synchronized. However, if one hydraulic motor meets more resistance than it can overcome and stalls all pump flow goes to the running hydraulic motor. The second motor then turns twice as fast. During this scenario, one flow-divider motor overspeeds while the opposite one cavitate. The only way to make sure both hydraulic motors stay synchronized in both directions of rotation is to install flow dividers at both valve ports. However, if the application allows the motors to become mechanically linked during operation, it is necessary that relief valves, anti-cavitation check valves, or slip orifices be added to the circuit.

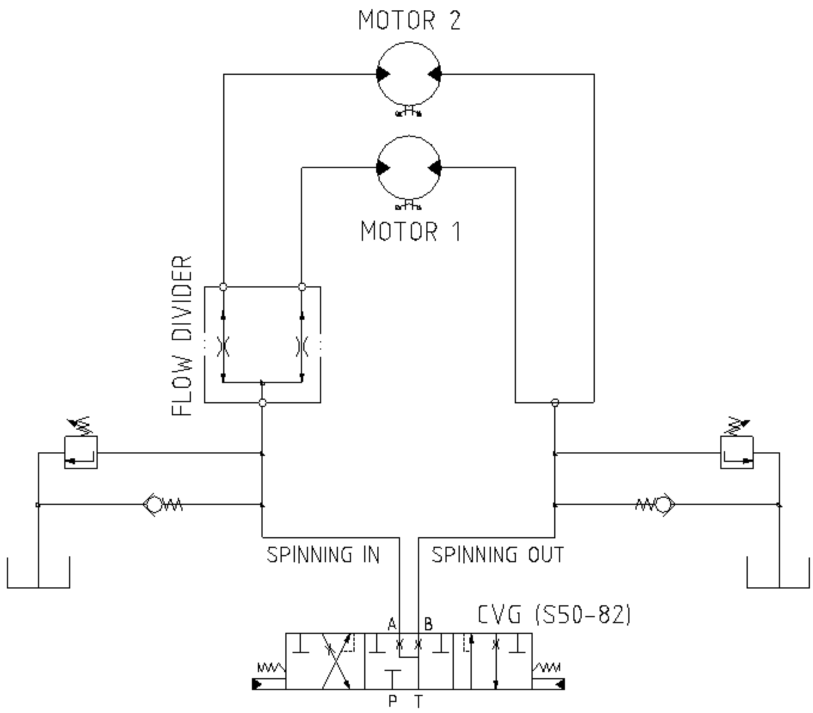


Figure 4.2 – Hydraulic System of motor

4.4 Control of hydraulic cylinder

The two cylinders have been used to provide a necessary force to clamp the drill pipe, and they mechanically connected to the spinner arm. There is no need to synchronizing tow cylinder which have a rigid mechanism ties them together and sliding on a rigid body via bushing.

4.5 Control of Torque

Torque control is achieved by adjusting the pressure in the CVG valve to correspond with the desired toque. The pressure over pressure transmitters will correspond to the actual torque.

The output torque and speed varies with regard of pipe type. These calculated to spin the drill pipe in section 3.2.3 by using the equation below.

Spinning Torque $$T_s = 4 \cdot T_m \cdot \frac{R_g}{2 \cdot R_p} \cdot \frac{R_{pipe}}{R_{rolle}}$$

The output torque and speed during spinning the different drill pipe are shown in table 5.1.

Nom. Bore inch	O.D. mm	Torque Nm	Speed rpm
3	76.2	2309	313
3 ½	88.9	2693	268
4	101.6	3078.5	234.5
5	127	3848	187.5
5 ½	139.7	4233	170.5
6	152.4	4618	156
8	203,2	6158	117
9	228.6	6927	104
9 ½	241.3	7311.5	99
10	254	7696.5	94

Table 4.1 – Torque and speed table

4.6 Control of vertical movement and bump

The spinner will be mounted to the Roughneck which is includes the elevator. The elevator should be controlled by testing the spinner for different pipe and determining a parameter for function speed in mm/sec for both *clock/counter clockwise* directions and stored in the controller table. These calculated seconds representing the vertical movement of the elevator. The bump can be controlled by either active thread compensation or passive compensation system. Active thread compensation will be achieved by controlling of the elevator which is including speed and friction of the elevator. The passive thread compensation can be controlled by using an accumulator to composite the pressure.

Chapter 5

5. Solid Works simulation FEM analysis

This chapter will briefly describe the structural analyses which have been carried out for the spinner support and appurtenant components for the clamping force during operation.

The objective of the structural analysis have been

- To provide sufficient and accurate information about the stress distribution within the structural components due to the clamping force applied on the structure.
- To achieve an optimal construction with respect to weight, global strength and deflection.
- To achieve an economic structural solution with respect to fabrication, installation and service.

To present that the steel design of the spinner structure is adequate with respect to strength and functionality for the given loading conditions and that the design complies with the statuary regulations and required standard as stated.
Solid Works-Simulation Program verifies the structural strength for the considered loading conditions constituent clamping and In-place analysis when clamping the drill pipe.
The criterion for adequate structural strength is that the stress level/Interaction ratio in the spinner structure is below the design yield strength of the material.

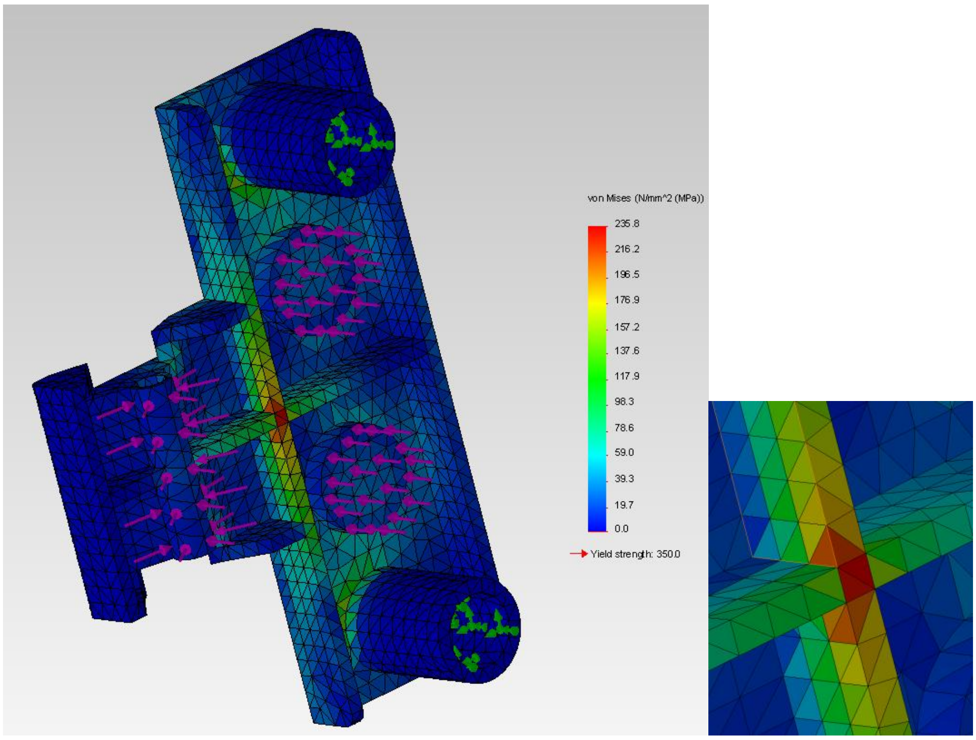


Figure 5.1 – Von Mises Stress Retraction bracket

Figure 6.1 is presenting Von Mises stresses for the spinner arm assembly including clamping force of 190kN. The result shows that the maximum yield stress is 235MP. As the yield stress for the material is 355MPa the material capacity is adequate.

The previous left hand figure show the Von Mises Stress for the arm assembly and the right hand show a detail of the maximum stress level detail in the crosection of plates. A load of 190kN is applied to the surface plate of spinner arm in horizontal direction. The spinner arm has been considered fixed with the interface of bushing.
A static linear analysis is carried out applying 2. order isotropic tetrahedral elements. All parts are modeled as one part, the bush hole is restricted to translate in X,Y and Z direction. Uniformly distributed pressure is applied at a cylinder bracket and hanger of rollers. The maximum stress level located in the crosection of the plates 20mm as shown in the detail below. This is expected as the detail has a sharp corner combined with high load.

Figure 6.2 is presenting Von Mises stresses for the motor bracket assembly including spinner moment of 8000N.m. The result shows that the maximum yield stress is 79MP. As the yield stress for the material is 355MPa the material capacity is adequate. The bottom of plate is restricted to translate in X,Y and Z direction.

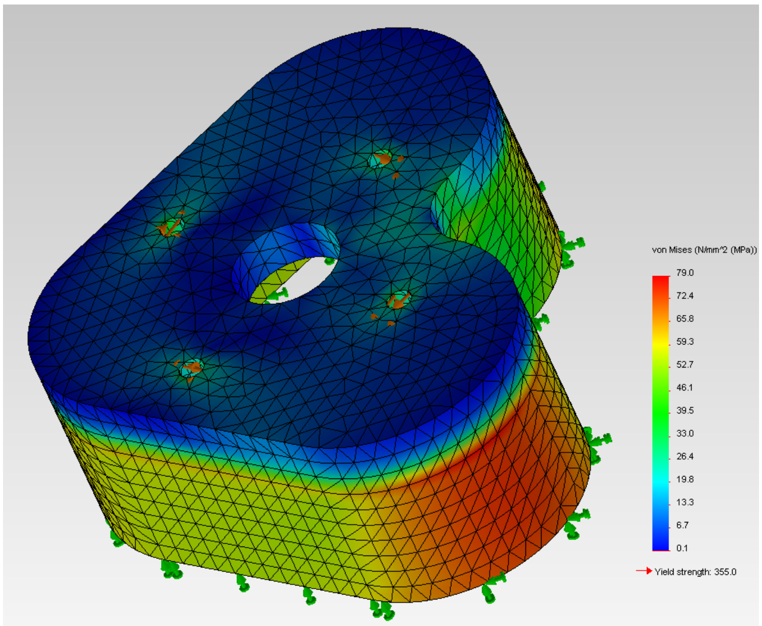


Figure 5.2 – Von Mises Stress Heart-shaped cover

The following general material properties are used in the analyses:

Steel density	ρ	7850	kg/m^3
Modulus of elasticity	E	210000	N/mm^2
Shear modulus	G	$0.8\cdot10^5$	N/mm^2
Poisson ratio	µ	0.3	[-]
Yield strength structural steel	f_y	355	N/mm^2

Chapter 6

6. Prototype

This chapter will briefly describe the prototyping of the spinner model. The spinner has been fabricated by quickly fabricating a scaled model of the physical assembly, using *MicroStationV8i* to design the model. *MicroStation* is a CAD software product for 2 and 3 dimensional design, its native format is the DGN format. The DGN file has been converted to STL file which is used for rapid prototyping.

The prototype model has been scaled1:2 and in the someplace the model scaled more than this scale to adapt to 3D printer board and to avoid welding of the parts. CNC machine has been used to print the prototype. The material which was used is the ABS plastic in a quantity of 3380ccm. The purpose of this prototyping is to visualize and test the mechanical function of the model.

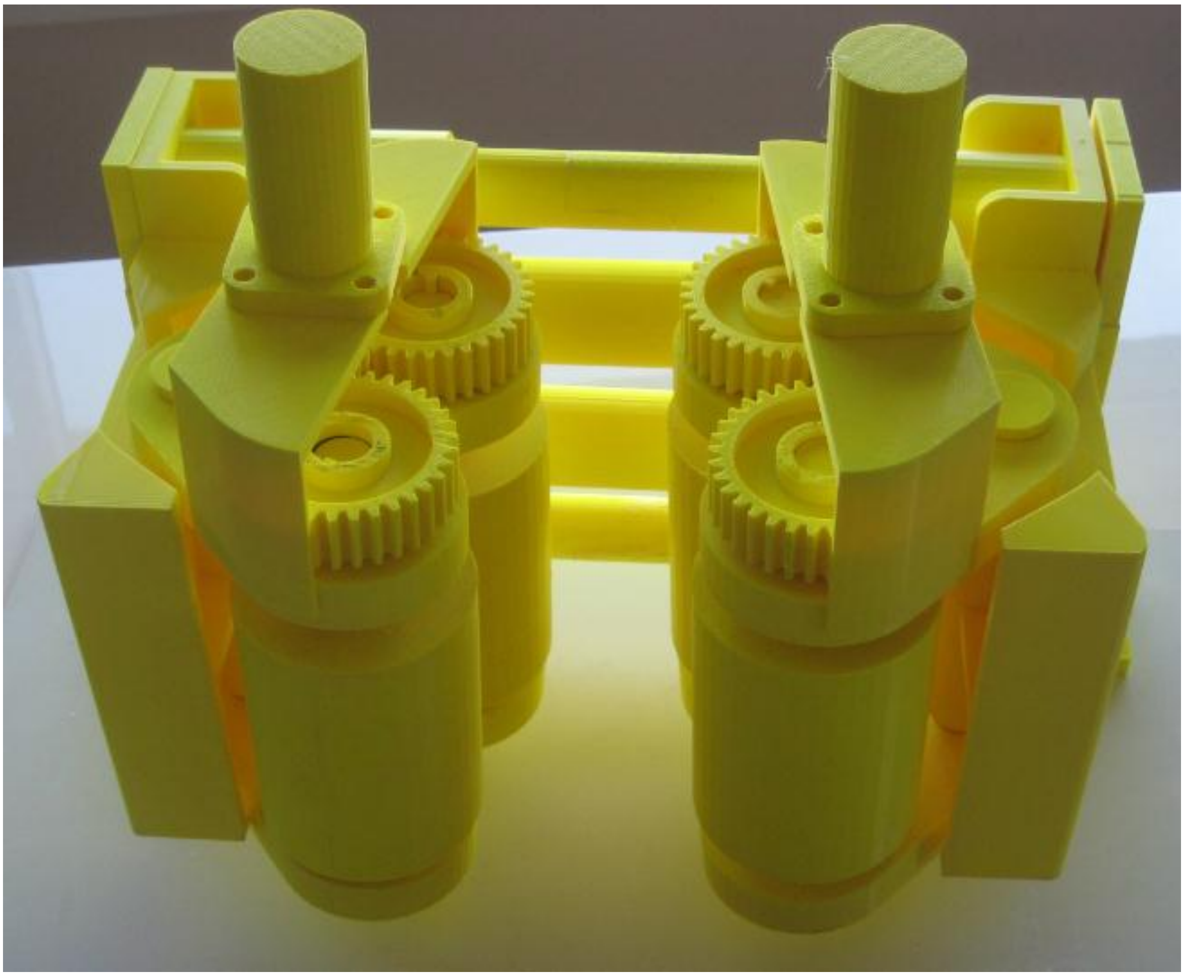


Figure 6.1 – prototype of spinner wrench

Chapter 7

7. Conclusion

The new spinner wrench system of this thesis has using to motors was developed to spin inn/out the drill pipe. Important component and calculation of these components have been dimensioned, modeled, and set up in simulation X. the Solid works simulation has been used to verify a structural strength of the material.

To achieve a required torque of 8000Nm the gearing system has been developed and the gear wheel has been dimensioned. The spinner wrench has an output torque 7700Nm with a speed of 93rpm for the largest drill pipe 10". The tool can handle drill pipe with diameter of 3" – 10", its compact design and ability to provide optimal grip of drill pipe compared to previous tool make it unique .

Furthermore, the hydraulic component was chosen and two hydraulic motors were synchronized by using the flow divider.

Acknowledgements

This work has been completed by the support of NOV, and the author thanks for this support. The authors thank Professor Kjell G. Robbersmyr for constructive technical advices, follow-up and providing the experimental data of a product development. Acknowledgement and thanks is also given to Professor Michael R. Hansen who was instrumental in starting the author on the road to success in the project. The author thanks Professor Anne Mueller at UiA for advice on academic writing, design and structure of thesis. I also appreciate the proof reading of this thesis by my colleague Thomas Liland at Nymo AS. I wish to thank my wife for essential encouragement and support.

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[13]http://www.skf.com/portal/skf/home/products?lang=en&maincatalogue=1&newlink=3_4_32
[14] http://calc.homepage.dk/pasningere.htm

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Appendix

Appendix A

Appendix A - Project Description

The University of Agder
Faculty of Engineering and Science
Master education in Mechatronics

Master-thesis spring 2012
Student: Lawk Fryad Farji

The Spinner – the rotating machinery.

National Oilwell Varco Inc. (NOV) is a worldwide leader (more than 50 000 employees in more than 60 countries), in design and manufacturing of equipment, systems and components used in oil and gas drilling and production. The head quarter is in Houston, Texas, but they have also a cocompany in Norway, National Oilwell Varco Norway AS (100% owned of NOV), with head quarter in Kristiansand and with branch offices in Stavanger, Molde, Asker and Arendal, totally 3000 employees in Norway.

NOVN is a leading global provider of drilling equipment to the oil and gas industry producing equipment to drill ships, semi submersibles and fixed installations. One such equipment is the spinner, the machine for rapid “spinning” in or out the thread of the drill pipe before the torque wrench making up the torque.

NOVN has already such equipment for sale, but they have asked for an alternative solution to the existing equipment which can spin drill pipes and drill collars from 2,5” to 10” OD, and which can make up and break out a torque of 8000 Nm without damage to the drill pipe. The spinner will be installed into a commercial torque wrench today, so this equipment should be small in size. The equipment should give an indication of the torque, the speed, the bump and so on when the operation is ended.

The project is expected to consist of:

1. Introduction / Literature study.
2. Product specification and generation of konsepts
3. Develope the spinner.
4. Develop the control system / oil hydraulic
5. Design and FEM analysis.
6. Building a prototype

Expenses related to the project must approved by the supervisors to be covered.

Delivering of the project is to be done in accordance with the guidelines published in the Fronter room for MAS500.

The project is carried out for: National Oilwell Varco Norway AS, Kristiansand

Supervisors:
Kjell G. Robbersmyr phone +47 3725 3207, email kjell.g.robbersmyr@uia.no
Henning Grosaas, phone +47 9948 7811, email henning.grosaas@nov.com
Kjetil Nyvold, phone + 47 3819 4272, email kjetil.nyvold@nov.com

Appendix B

Appendix B - Motor Catalogue Sauer – Danfoss

Technical data
for OMT, OMTW, OMTS, OMT FX OMT FL and OMT FH

Type			OMT OMTW OMTS OMT FX OMT FL OMT FH	OMT OMTW OMTS OMT FX OMT FL OMT FH	OMT OMTW OMTS OMT FX OMT FL OMT FH	OMT OMTW OMTS OMT FX OMT FL OMT FH	OMT OMTW OMTS OMT FX OMT FL OMT FH	OMT OMTW OMTS OMT FX OMT FL OMT FH
Motor size			**160**	**200**	**250**	**315**	**400**	**500**
Geometric displacement	cm^3 [in^3]		161.1 [9.83]	201.4 [12.29]	251.8 [15.37]	326.3 [19.91]	410.9 [25.07]	523.6 [31.95]
Max. speed	min-1 [rpm]	cont.	625	625	500	380	305	240
		int[1)]	780	750	600	460	365	285
Max. torque	Nm [lbf·in]	cont.	470 [4160]	590 [5220]	730 [6460]	950 [8410]	1080 [9560]	1220 [10800]
		int.[1)]	560 [4960]	710 [6280]	880 [7790]	1140 [10090]	1260 [11150]	1370 [12130]
Max. output	kW [hp]	cont.	26.5 [35.5]	33.5 [44.9]	33.5 [44.9]	33.5 [44.9]	30.0 [40.2]	26.5 [35.5]
		int.[1)]	32.0 [42.9]	40.0 [53.6]	40.0 [53.6]	40.0 [53.6]	35.0 [46.9]	30.0 [40.2]
Max. pressure drop	bar [psi]	cont.	200 [2900]	200 [2900]	200 [2900]	200 [2900]	180 [2610]	160 [2320]
		int.[1)]	240 [3480]	240 [3480]	240 [3480]	240 [3480]	210 [3050]	180 [2610]
		peak[2)]	280 [4060]	280 [4060]	280 [4060]	280 [4060]	240 [3480]	210 [3050]
Max. oil flow	l/min [USgal/min]	cont.	100 [26.4]	125 [33.0]	125 [33.0]	125 [33.0]	125 [33.0]	125 [33.0]
		int.[1)]	125 [33.0]	150 [39.6]	150 [39.6]	150 [39.6]	150 [39.6]	150 [39.6]
Max. starting pressure with unloaded shaft	bar [psi]		10 [145]	10 [145]	10 [145]	10 [145]	10 [145]	10 [145]
Min. starting torque	at max. press. drop cont. Nm [lbf·in]		340 [3010]	430 [3810]	530 [4690]	740 [6550]	840 [7430]	950 [8410]
	at max. press. drop int.[1)] Nm [lbf·in]		410 [3630]	520 [4600]	630 [5580]	890 [7880]	970 [8590]	1060 [9380]

1) Intermittent operation: the permissible values may occur for max. 10% of every minute.
2) Peak load: the permissible values may occur for max. 1% of every minute.

For max. permissible combination of flow and pressure, see function diagram for actual motor.

Appendix C

Appendix C - Flow divider sun hydraulic

Capacity
12 - 60 gpm (45 - 240 L/min.

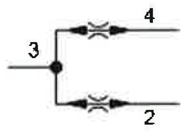
Closed center, flow divider-combiner valve

Functional Group:
Products : Cartridges : Flow Divider : Divider/Combiner : Closed Center

Model
FSF/

Product Description

Closed-center flow divider/combiners are sliding-spool, pressure-compensated devices used to split flow in one direction and combine flow in the opposite direction. These valves may be used to accurately control two or more cylinders or hydraulic motors where bidirectional operation is required.



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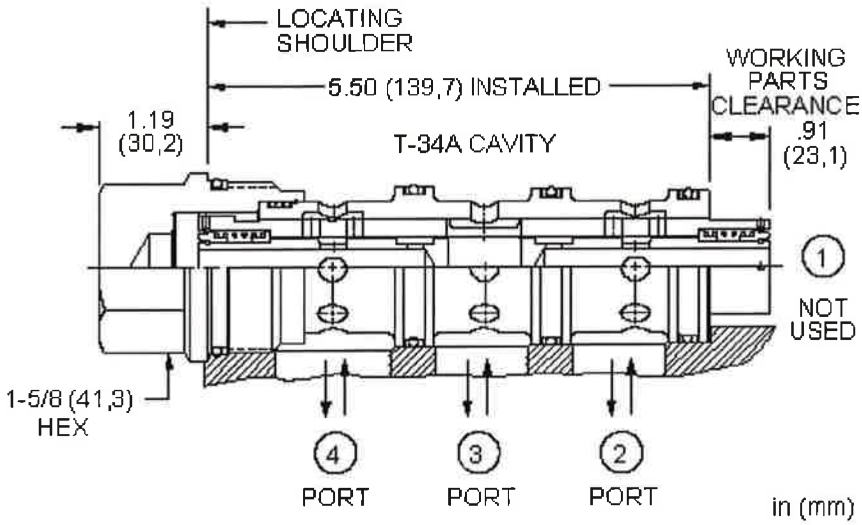


Technical Features

- All flow divider and divider/combiner cartridges are physically interchangeable (i.e. same flow path, same cavity for a given frame size).
- Operating characteristics cause the leg of the circuit with the greatest load to receive the higher percentage of flow in dividing mode. If a rigid mechanism is used to tie actuators together, the lead actuator may pull the lagging actuator and cause it to cavitate.
- In combining mode, compensating characteristics will cause the leg of the circuit with the lowest load to receive the higher percentage of flow. If a synchronization feature is not included, an additive accuracy error will be experienced with each full stroke of the actuator.
- In applications involving rigid mechanisms between multiple actuators, operating inaccuracy will cause the eventual lock-up of the system. If the mechanical structure is not designed to allow for the operating inaccuracy inherent in the valve, damage may occur.
- In motor circuits, rigid frames or mechanisms that tie motors together, and/or complete mechanical synchronized motion of the output shaft of the motors, either by wheels to the pavement or sprockets to conveyors, will contribute to cavitation, lock-up and/or pressure intensification.
- Variations in speed and lock-up can be attributed to differences in motor displacement, motor leakage, wheel diameter variance and friction of wheels on the driving surface.
- Extreme pressure intensification can occur on multiple wheel drive vehicles.
- Flow between ports is limited to spool leakage. This does not provide leak proof holding capability, but can be useful in minimizing cross flov and drift.
- Divisional and combining accuracy are equal.
- Below the minimum flow rating there is not enough flow for the valve to modulate. It is effectively a tee. If flow starts at zero and rises, there will be no dividing or combining control until the flow reaches th€ minimum rating.
- Incorporates the Sun floating style construction to minimize the possibility of internal parts binding due to excessive installation torque and/or cavity/cartridge machining variations.

Technical Data

	U.S. Units	Metric Units
Cavity	T-34A	
Capacity	12 - 60 gpm	45 - 240 L/min.
Divisional Accuracy at Max Input Flow	50% ±2.5%	
Divisional Accuracy at Minimum Input Flow	50% ±4.5%	
Maximum Operating Pressure	5000 psi	350 bar
Pressure Drop at Maximum Rated Input Flow	350 psi	24 bar
Pressure Drop at Minimum Rated Input Flow	30 psi	2 bar
Series (from Cavity)	Series 4	
Valve Hex Size	1 5/8 in.	41,3 mm
Valve Installation Torque	350 - 375 lbf ft	475 - 500 Nm
Seal Kits - Cartridge	Buna: 990-034-007	
Seal Kits - Cartridge	Viton: 990-034-006	
Model Weight	2.98 lb.	1.35 kg.

Split	Input Flow		Rated Accuracy	Maximum Possible Flow Variation
50:50	Max Rated	60 gpm	±2.5%	28.5 - 31.5 gpm
		240 L/min		108 - 119 L/min
	Min rated	12 gpm	±4.5%	5.5 - 6.5 gpm
		45 L/min		21 - 25 L/min

The maximum possible variation is at 5000 psi (350 bar) differential between legs with the high pressure leg being the higher flow in dividing mode and the lower flow in combining mode.

FSFA-XAN

Control

Standard Options

X Not Adjustable

Flow Split

Standard Options

A 50/50

Seal Material

Standard Options

N Buna-N

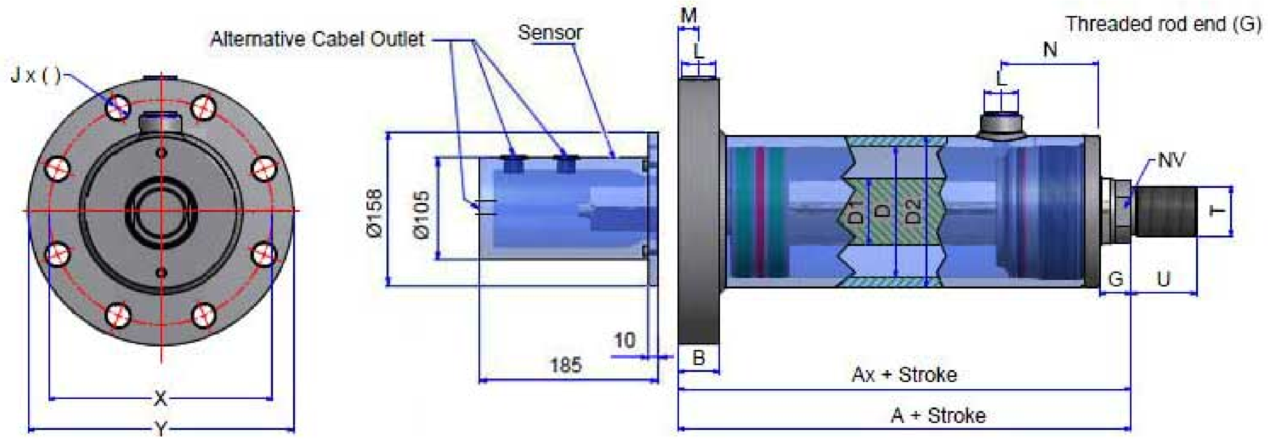
V Viton

Appendix D

Appendix D - Pmc servi cylinder servic

Cylinder LHA25 B/G

Threaded Rod



Order No.

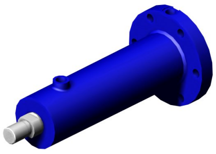
LHA25-110/56x0275-BG-HC-SSN-NNN-0

Type

Cylinder LHA25

Created: 2012-03-26 11:57:14

Address:	PMC Servi Cylinderservice AS Kvithyll 7100 Rissa Norway	Phone: Fax: Web: Email:	47 73 85 05 00 47 73 85 05 01 www.servi.no cylinderservice@pmcservi.no



Fixing cylinder	Flange bottom side
Fixing rod	Threads
Bore diameter D (mm)	110
Rod diameter D1 (mm)	56
Pressure (bar)	250
Rod surface	20-30my chromium
Rod material	Carbon Steel
Seals	Standard
Bearing	Standard
Damping	None
Sensor/switch	None
Atex/EX	No
Ice scraper	No
Stroke (mm)	275
Piston rod extension G (mm)	0
Surface treatment	Standard
Third party certification	DNV
A (mm)	217
Ax (mm)	217
D2 (mm)	130
X (mm)	200
Y (mm)	240
B (mm)	35
J	Ø22x6
G (mm)	30
L	G 3/4
M (mm)	17.5
N (mm)	85
T	M42x2
U (mm)	56
NV (mm)	42

Created: 2012-03-26 11:57:14

Address: PMC Servi Cylinderservice AS
Kvithyll
7100 Rissa
Norway

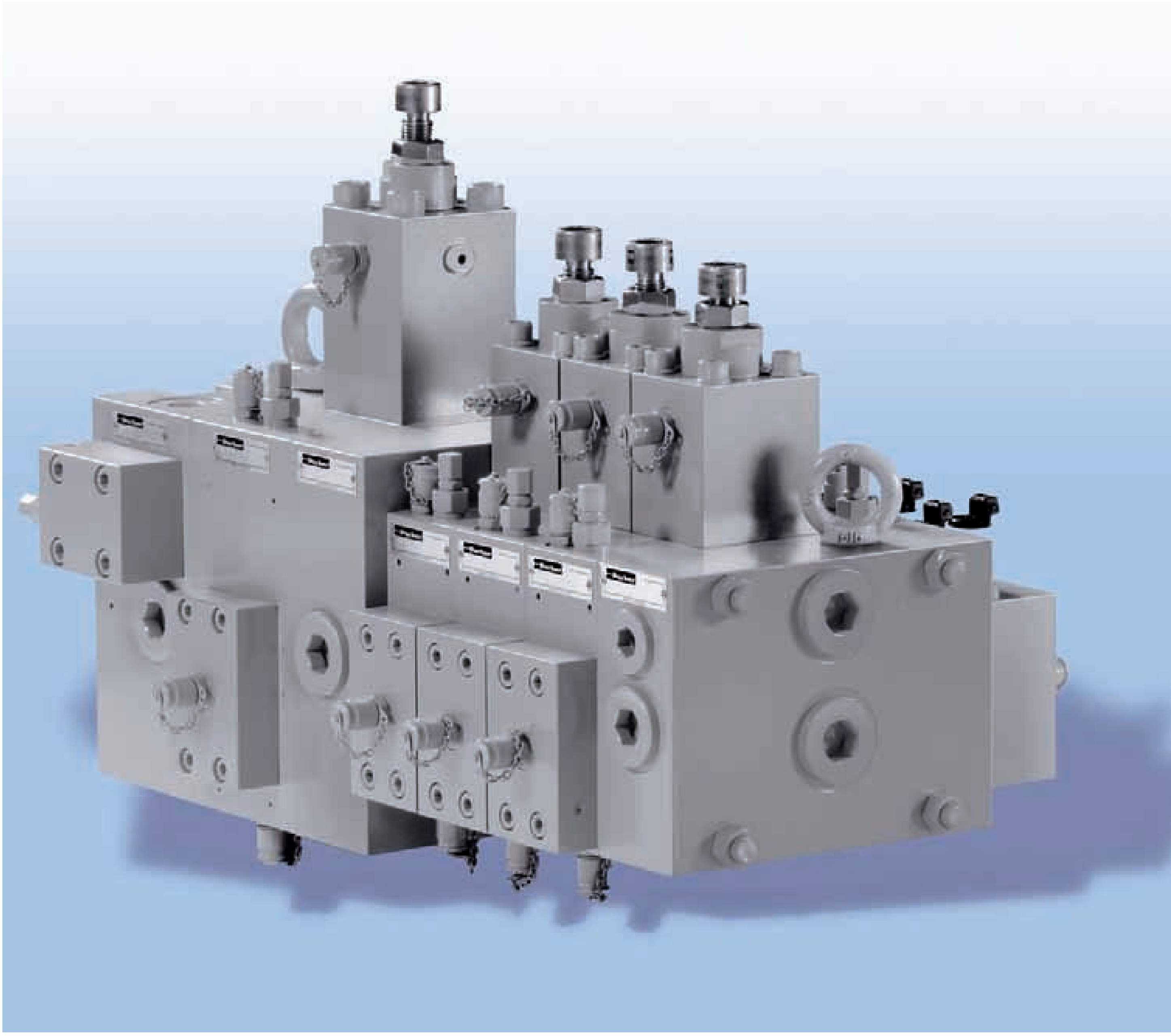
Phone: 47 73 85 05 00
Fax: 47 73 85 05 01
Web: www.servi.no
Email: cylinderservice@pmcservi.no

Appendix E

Appendix E - CVG Control Valve Group

CVG
Control Valve Group

Catalog FI-EN104-C
November 2007



FEATURES

Proportional control valve group CVG with sandwich plate design.
Accurate control of high flows up to 1200 l/min.
Two nominal sizes available: CVG30 and CVG50.
CVG30 and CVG50 sections can be joint in one block.
Very compact design.
Load sensing.
Load pressure compensated.
Closed loop version with spool position control LVDT available.
Separately adjustable pressures in A and B ports.
Main pressure relief valve available.
Hydraulic and electro-hydraulic controls.
Excellent linearity and repeatability, low hysteresis.
Every valve group factory tested and adjusted.
World wide Parker service.

DESCRIPTION

CVG valve group can be used to control wide range of different machines like cranes, large mobile vehicles, lifting equipment, drilling equipment and stationary applications. High flows can be accurately controlled with low pressure drop and low hysteresis. CVG is a proportional load sensing valve group with sandwich plate design. Control sections are available in two nominal sizes: CVG30 and CVG50, and with different spools. CVG30 and CVG50 sections can be joined in one block. Each main spool is load pressure compensated to keep speed constant independent on load. Maximum flow can be adjusted separately for each section. Electrically open and closed loop versions are available. Closed loop version has improved linearity and repeatability and extremely low hysteresis. Maximum pressures in ports A and B are separately adjustable in each section. Inlet sections are available with or without main pressure relief valve. Control signal can be either hydraulic or electric or both signals can be used parallel.

OPERATION

Speed and direction of movements are proportionally controlled by spring centered main spools. Pilot pressure moves the main spool against spring force. In electric version pilot pressure is adjusted by electro-proportional 3-way pressure reduction valves. In hydraulic version pilot pressure is adjusted by external control valve. Pilot oil flow to electro-proportional pilot valves is arranged either internally or externally. Internal signal is taken from port P through pressure reduction valve in the inlet section. External pilot pressure source is connected to port PP. Pressure difference over each main spool is kept constant by 2-way pressure compensators. Flow (and speed of actuator) with certain opening of the main spool is constant independent on load variations. Pressure compensators are equipped with adjustable springs (version P2). In the version P2 the spring force is steplessly adjustable. Spring force of the pressure compensator sets also a limit for maximum speed of each actuator. Maximum pressures in ports A and B are adjusted separately by pilot valves (6). These valves are piloting the pressure compensator cartridge. When pressure in port A or B exceeds setting of the pilot valve the pressure compensator starts to operate like pressure reduction valve. CVG is a load sensing valve group. Every main spool is equipped with LS signal channels. The highest load pressure in system is always connected to LS port. This signal can be used to control variable load sensing pump. In open loop valve version (F0) position of the main spool follows electric control signal coming from amplifier card. In closed loop valve version (F1) position of main spool is measured by LVDT. Position signal is sent to amplifier card that compares position signal to command signal and corrects the control signal to solenoids accordingly. Spool position (and flow) is accurately controlled. Influences of friction and flow forces are compensated and linearity, repeatability and hysteresis of the valve are improved.

CHARACTERISTICS

Design	Control valve group with sandwich plate design
Mounting positions	Optional
Ambient temperature range	- 30 °C … + 50 °C
Operating pressure (P, A, B, LS)	350 bar
External pilot pressure (PP) (optional)	45 bar
Permissible tank line (T) pressure	30 bar
Permissible drain line (DR) pressure	10 bar
Nominal ?ow (with 5 bar pressure drop over one control edge of the spool)	CVG30: 500 l/min / section CVG50: 800 l/min / section
Maximum ?ow/control section	CVG30: 750 l/min / section CVG50: 1200 l/min / section
Maximum ?ow/main pressure relief valve	NS30 inlet section: 800 l/min NS50 inlet section: 1200 l/min
Solenoids	24 V, 0 – 750 mA, 100% ED, IP65
Spool position control (optional)	LVDT
Ampli?er card (ask for separate data sheets)	Parker amplifier programs. Consult your contact in Parker Hannifin
Hydraulic remote control signal	0 - 35 bar
Step response time	CVG30: 300ms CVG30 high responce: 110ms CVG50: 800ms CVG50 high responce: 280ms
Fluid	Mineral oil according to DIN 51524 and DIN 51525
Fluid temperature range	- 20 °C … + 70 °C
Contamination level	Max. permissible contamination level according to NAS 1638 Class 8 (Class 9 for 15 Micron and smaller) or ISO 19/17/14
Weight	CVG30 - inlet section P30-1 22 kg - inlet section P30-2 22 kg - control section 40 kg - outlet section T30 22 kg - outlet section T31 16 kg CVG50 - inlet section P50-1 60 kg - inlet section P50-2 60 kg - control section 84 kg - outlet section T50 55 kg - outlet section T51 33 kg Adapter plate (to connect CVG30 and CVG50) 39 kg
Surface treatment	2-component epoxy primer

Ordering code

CVG

ORDERING FORM OF COMPLETE VALVE GROUP

Example:

Code of valve group	CVG	51	33									
Code of inlet section	P50	11	A	N	1							
Code of control section 1	S50	81	8	C1	P2	F1	A	N	1	720	A350	B300
Code of adapter	A53	0	A	N								
Code of control section 3	S30	81	5	C2	P2	F1	A	N	1	400	A350	B200
Code of control section 4	S30	81	3	C2	P2	F1	A	N	2	300	A300	B300
Code of control section 5	S30	81	1	C3	P2	F1	A	N	2	80	A250	B100
Code of outlet section	T30	A	N									

ORDERING CODE OF COMPLETE VALVE GROUP

Code of the complete valve group: **CVG-51-33**

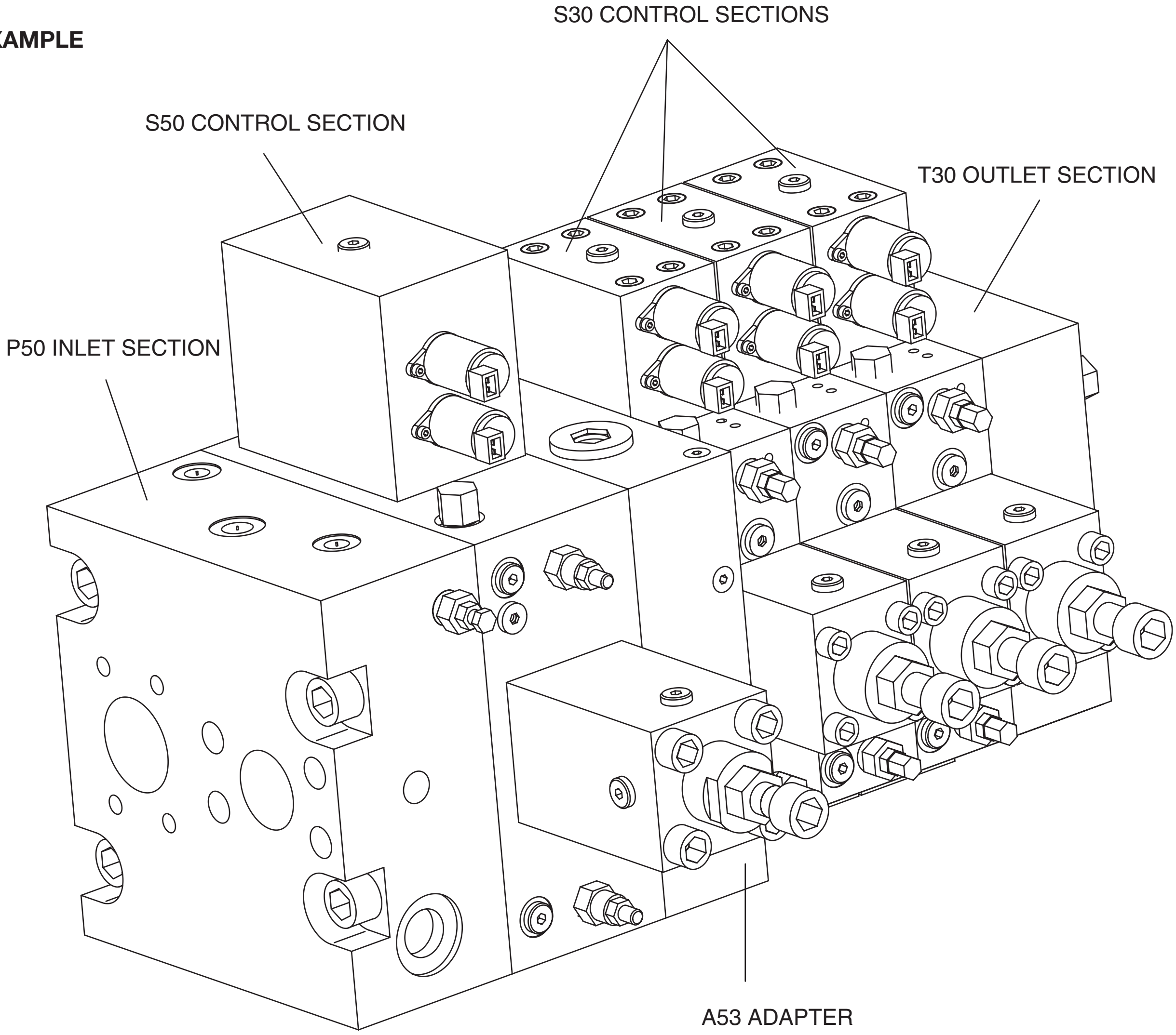
Control valve group

Number of nominal size 50 sections
5X = X pcs size 50 sections (omit if no size 50 sections)

Number of nominal size 30 sections
3X = X pcs size 30 sections (omit if no size 30 sections)

Note, that the maximum number of spool sections is 5 pcs/size. If more sections required, please contact supplier.

EXAMPLE



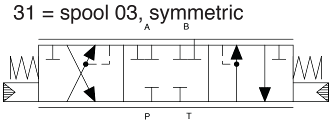
Parker Hannifin Oy
Lokomec
Tampere, Finland

Code of control sections: **S50–81–8–C1–P2–F1–AN1/720–A350B300**

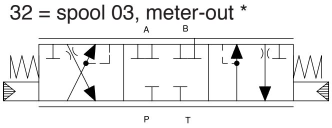
Valve selection
S 30 = size 30 control section
S 50 = size 50 control section

Spool symbol

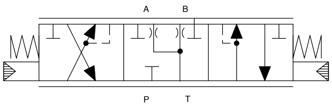
31 = spool 03, symmetric



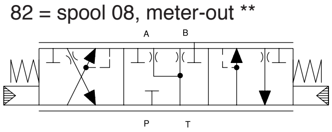
32 = spool 03, meter-out *



81 = spool 08, symmetric



82 = spool 08, meter-out **



* Available spool sizes 300 l/min and 500 l/min. } Other sizes
** Available spool sizes 500 l/min and 800 l/min. } consult Lokomec.

Spool size (dp = 5 bar / edge)
1 = 100 l/min (for size 30 only)
3 = 300 l/min (for size 30 only)
5 = 500 l/min (for sizes 30 and 50)
8 = 800 l/min (for size 50 only)

Controls
C1 = electro-proportional *
C2 = electro-proportional and hydraulic *
C3 = hydraulic
C4 = electro-proportional high response (more information contact Parker) **

* Suitable connectors AMP Junior-Timer type C
** Plug-in connectors according to ISO 4400

Pressure compensator adjustment
P2 = adjustable spring force

Feed back of the main spool position
F0 = no feedback
F1 = feedback with LVDT

Design letter

Seal class
N = N.B.R. (Buna N)

Pressure measuring connectors
1 = no connectors
2 = carbon steel connectors
3 = stainless steel connectors

Flow setting (standard setting maximum ?ow)
720 = 720 l/min (see ?ow table on page 10)

Pressure settings (standard setting 250 bar)
A250B300 = 250 bar in port A, 300 bar in port B

Code of inlet sections: P50-11-AN1-365

Valve selection
P30 = size 30 inlet section
P50 = size 50 inlet section

Main flow valves in the inlet section
1 = no main pressure relief valve
2 = with main pressure relief valve

Pilot pressure supply
1 = internal supply through pressure reducing valve (4)
2 = external supply through port PP

Design letter

Seal class
N = N.B.R. (Buna N)

Pressure measuring connectors
1 = no connectors
2 = carbon steel connectors
3 = stainless steel connectors

Pressure setting (standard setting 300 bar)
setting of the main pressure relief valve

Code of outlet sections: T50-AN

Valve section
T 30 = Size 30 outlet section
T 31 = Size 30 blind cover without connections
T 50 = Size 50 outlet section
T 51 = Size 50 blind cover without connections

Design letter

Seal class
N = N.B.R. (Buna N)

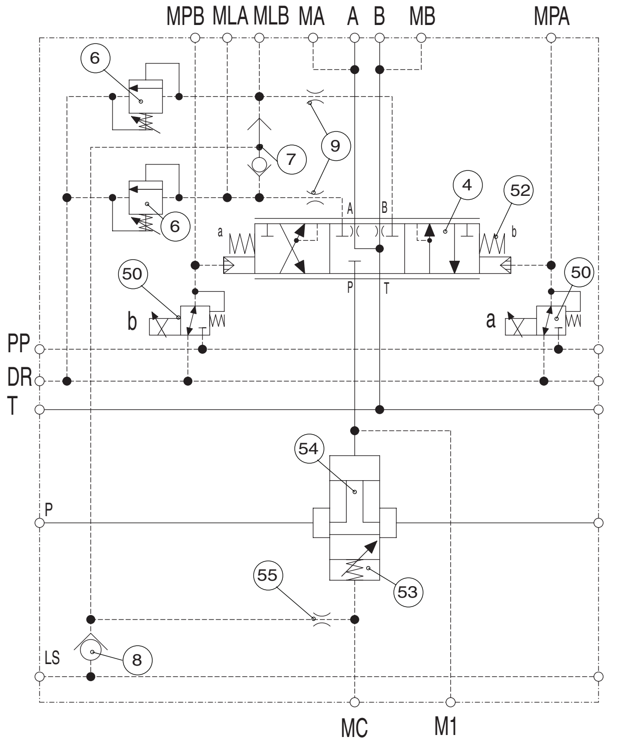
Code of adapter: A530-AN

Adapter between CVG 50 and CVG 30 sections

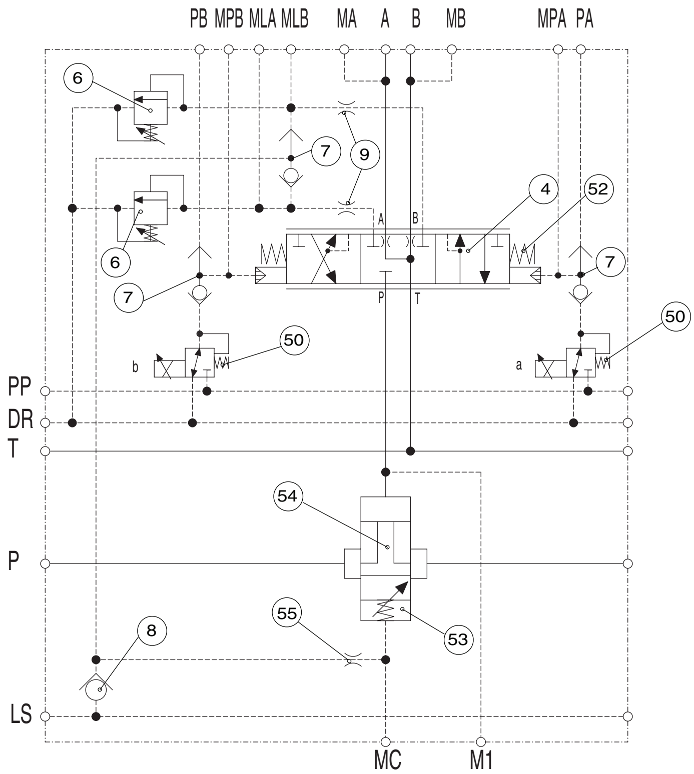
Design letter

Seal class
N = N.B.R. (Buna N)

CVG30 / CVG50 CONTROL SECTION (C1 = electro-proportional)

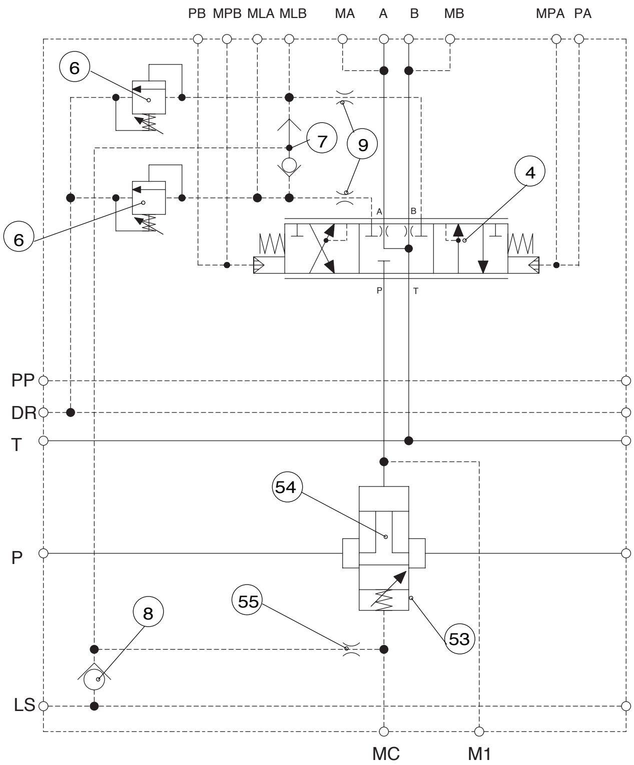


CVG30 / CVG50 CONTROL SECTION (C2 = electro-proportional and hydraulic)

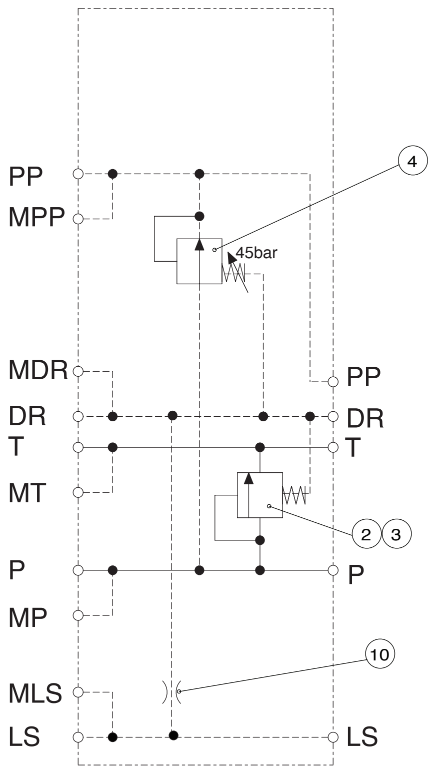


Parker Hannifin Oy
Lokomec
Tampere, Finland

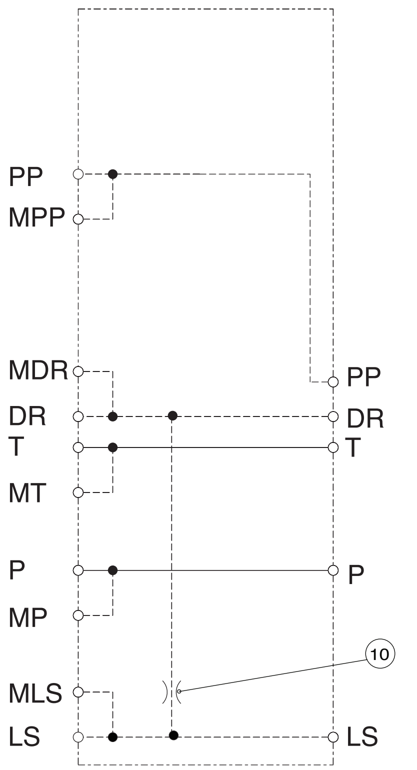
CVG30 / CVG50 CONTROL SECTION (C3 = hydraulic)



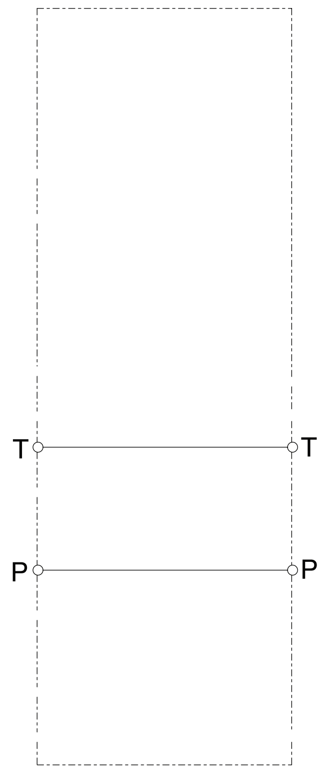
CVG30 / CVG50 INLET SECTION / OUTLET SECTION



Inlet section (P** - 21) with main pressure relief valve (2 and 3) and pilot pressure reducing valve (4).



Inlet section (P** - 12) without main pressure relief valve (2 and 3) and pilot pressure reducing valve (4).



Outlet section T30/T50.

Parker
Parker Hannifin Oy
Lokomec
Tampere, Finland

Appendix F

Appendix F - Tannhjul Kapittel 9

9. TANNHJUL

I dette kapittel gjennomgås sentrale definisjoner og viktige mål på et tannhjul. Det er også vist typiske konstruksjoner hvor tannhjul benyttes. I tillegg vises flere eksempler på tannhjulsberegninger.

9.1. INTRODUKSJON

Tannhjul benyttes der hvor avstand mellom aksler er forholdsvis liten, til forskjell fra reimdrift hvor avstanden mellom aksler er stor. To aksler og to tannhjul satt sammen i et hus kalles en veksel. Når flere aksler og flere tannhjul er satt sammen i et system kalles dette en girkasse.

En veksel benyttes til å:

- overføre rotasjon
- overføre moment
- endre turtall / moment

På markedet finnes flere typer tannhjul, og det skilles i prinsippet mellom:

- sylindriske tannhjul
- koniske tannhjul

Fortanning beskriver flankelinjenes (skjæringslinjen mellom tannflanke og deleflate (definert langs delesirkelen, fig 9.5)) form. Sylindriske tannhjul kan ha forskjellig fortanning, eksempler er vist i fig 9.1. Eksempler på typer fortanning for koniske tannhjul er vist i fig 9.2.

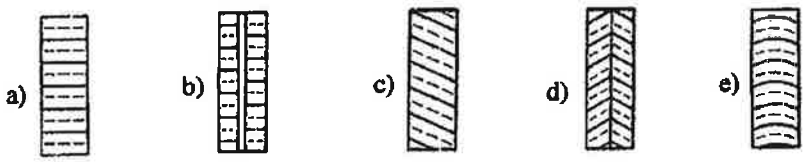


Fig 9.1. Fortanning på sylindriske tannhjul. a) Rettfortanning, b) Trinnfortanning, c) Skråfortanning, d) Pilfortanning, e) Buefortanning.

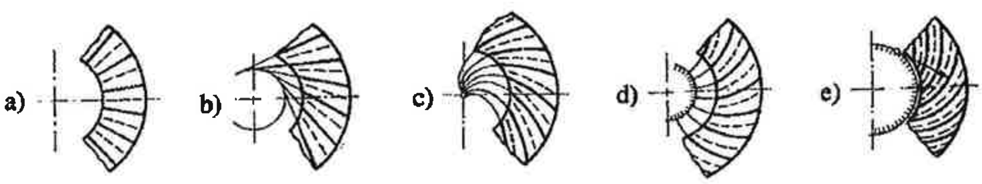


Fig 9.2. Fortanning på koniske tannhjul. a) Rettfortanning, b) Skråfortanning, c) Spiralfortanning, d) Evolventfortanning, e) Buefortanning.

Avhengig av hvordan akslene er montert i forhold til hverandre, skilles mellom:

- parallelle aksler; her benyttes sylindriske tannhjul med rette eller skrå tenner
- skjærende aksler; her benyttes koniske tannhjul
- kryssende aksler; dette er typisk konstruksjon for såkalte snekkeveksler

Typiske aksel-oppstillinger er vist i fig 9 3 for forskjellige typer tannhjul og forskjellig fortanning. Både parallelle og skjærende aksler ligger i samme plan. Rette sylindriske tannhjul er de mest benyttede tannhjul. Avhengig av akslenes og tannhjulenes form og montering skilles det mellom tannhjulsveksel og snekkeveksel.

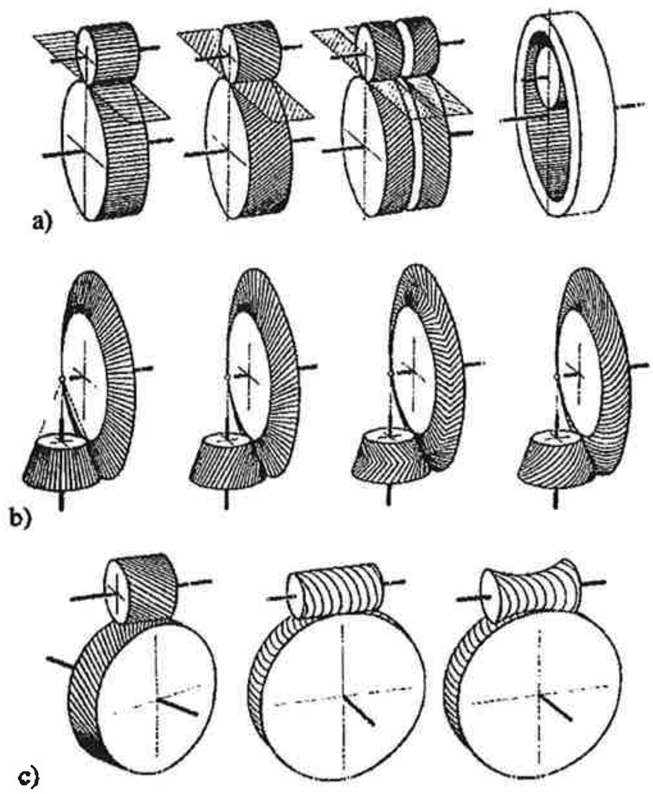


Fig 9 3. **Typiske akseloppstillinger for ulike typer tannhjul med forskjellig fortanning a) Parallelle aksler**, b) Skjærende aksler, c) Kryssende aksler (snekkeveksler)

Legg merke til at dobbel skråfortanning i prinsippet er identisk med pilfortanning. For å oppnå lik dreieretning på to tannhjul kan innvendig fortanning på største tannhjul benyttes. Andre typer tannhjulsveksler er vist i fig. 9 4

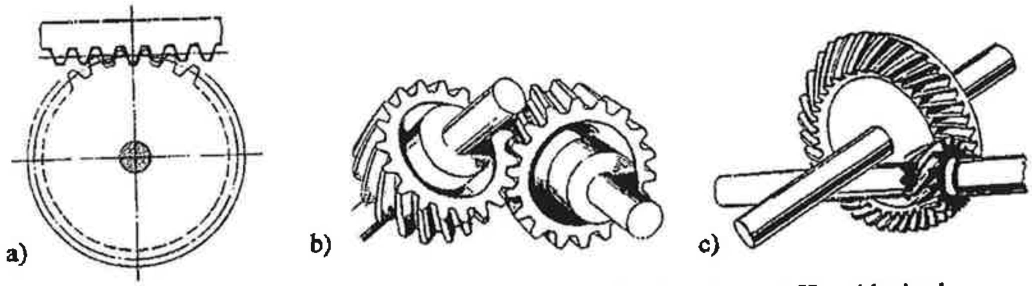


Fig 9 4. **Andre typer tannhjulsveksler** a) Tannstang, b) Skruehjul, c) Hypoidveksel

Mindre tannhjul og aksel kan tilvirkes av ett emne, vist i fig. 9 5a). Normalt feste av tannhjul til aksel er bruk av kile, vist i fig. 9 5c) og i sjeldnere tilfeller bruk av sveis, fig. 9 5b)

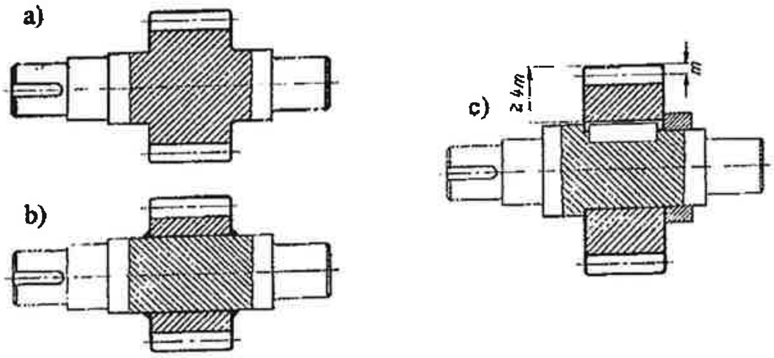


Fig 9 5 Feste av tannhjul til aksel a) Tannhjul og aksel tilvirket av ett emne b) Tannhjulet sveist til akselen. c) Bruk av kile

Når to riktig konstruerte tannhjul overfører rotasjon eller moment, vil de to tannhjulene ha et felles tangeringspunkt. For hvert tannhjul trekkes en sirkel gjennom tangeringspunktet, de såkalte delesirklene, vist i fig. 9 6

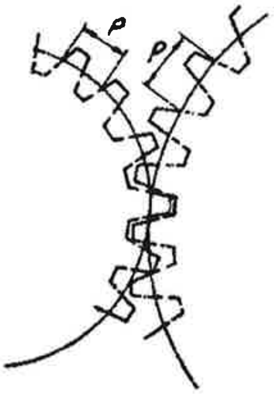


Fig 9 6 Delesirkler

Langs delesirkelen defineres tannhjulets delingen som summen av tann pluss luke

Ved å modellere tannhjul i plexiglass og benytte spenningsoptisk metode (polarisert lys sendes gjennom modellen) vil typiske spenningslinjer oppstå. Feltene på hver side av disse linjene har forskjellige spenningsnivå. For tannen vist i fig. 9 7 a) er spenningen stor ved belastningens angrepspunkt og ved tannfoten. Fig 9 7 b) viser spenningsfordelingen ved kraftens angrepspunkt, mens fig. 9 7 c) viser stor bøyespenning ved tannfoten pga. momentarmen

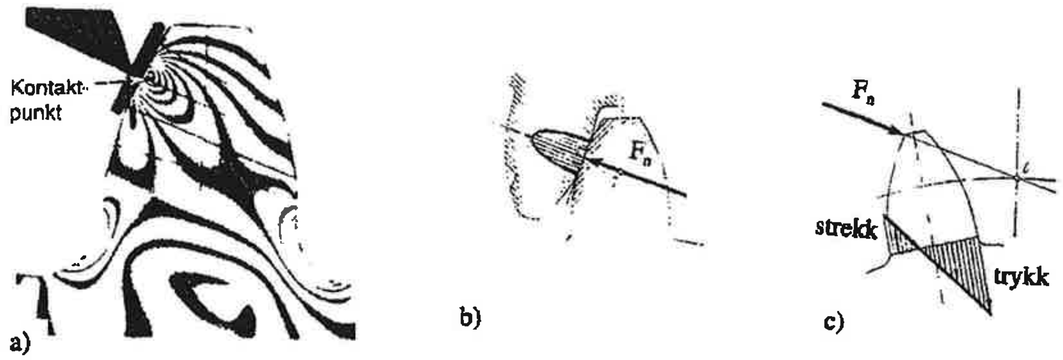


Fig. 9.7 a) Spenningsoptisk modell av belastet tannhjul, b) Spenningsfordeling ved kraftens angrepspunkt c) Bøyespenning ved tannfot.

Detaljer om tannhjul finnes i Norsk Standard:

- NS 5000:Tannhj Modulrekker for sylindriske tannhj. og for koniske tannhj med rette tenner
- NS 5001:Tannhjul Basisprofil for sylindriske tannhjul.
- NS 5005:Tannhjul Sylindriske tannhjul.
- NS 5006:Tannhjul Koniske tannhjul med rette tanner
- NS 5010:Tannhjul Terminologi
- NS 5011:Tannhjulsveksler Sylindriske tannhjul Rette eller skrå tenner Beregning av bæreevne.
- NS 5012:Tannhj Evolventtannhj med parallelle aksler Kontrollmetoder og toleransesystemer

9.2. BEREGNINGSUNDERLAG

I en tannhjulsveksel er det to forhold som har betydning ved beregninger:

- oversetningsforholdet, i : forholdet mellom inngående og utgående turtall:

$$i = n_1 / n_2 \tag{9.1}$$

hvor

n_1 : turtall på det lille hjulet Dette hjulet betegnes **drevet**.
n_2 : turtall på det store hjulet Dette hjulet betegnes **hjulet**

Dette forholdet kan være både større, lik eller mindre enn 1:

$i > 1$; turtallsreduserende oversetning, såkalt " nedveksling "
(dette er den mest vanlige oversetning i praksis)
$i = 1$; konstant turtallsoversetning
$i < 1$; turtallsøkende oversetning, såkalt " oppveksling "

- utvekslingsforholdet, u : storhjulets (hjulets) tannantall (z_s) dividert på lillehjulets (drevets) tannantall (z_l) uten hensyn til hvilket av hjulene som er drivende:

$$u = z_s / z_l \qquad (9.2)$$

Forholdet kan være større eller lik 1

I praksis benyttes oversetning og utveksling om hverandre.

Oversetningsforholdet er tidligere definert som forholdet mellom inngående turtall og utgående turtall, men det er også lik forholdet mellom inngående vinkelhastighet og utgående vinkelhastighet. Andre varianter er at oversetningsforholdet er lik forholdet mellom utgående delesirkelen og inngående delesirkel og tilsvarende mellom utgående tanntall og inngående tanntall:

$$i = n_1 / n_2 = \omega_1 / \omega_2 = d_2 / d_1 = z_2 / z_1 \qquad (9.3)$$

hvor

n : turtall
ω : vinkelhastighet
d : delesirkel
z : tanntall
1 : inngående, drevet
2 : utgående, drevne

Bruk av mellomhjul kan forekomme i praksis, og rotasjonsretningene for tannhjulene blir da som vist i fig. 9.8.

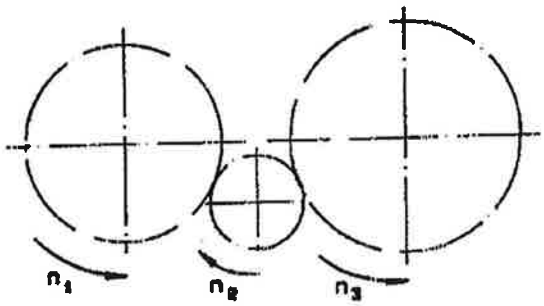


Fig. 9.8. Rotasjonsretningen på tannhjulene ved bruk av mellomhjul.

Ved bruk av mellomhjul blir det totale oversetningsforholdet (forholdet mellom inngående og utgående turtall) :

$$i_{tot} = n_1 / n_3 = z_3 / z_1 \qquad (9.4)$$

hvor

n : turtall
z : tannantall
1 : inngående
3 : utgående

Fig. 9.9 viser en tannhjulsoverføring bestående av 6 tannhjul og 4 aksler (en større tannhjulsveksel)

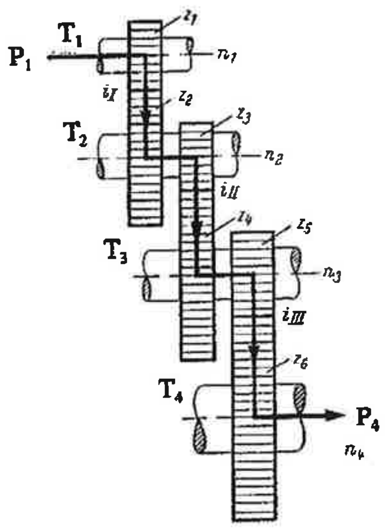


Fig. 9.9. En større tannhjulveksel.

I fig. 9.9 gjelder følgende definisjoner:

- P : effekt [W]
- T : torsjonsmoment / dreiemoment [Nm]
- n : turtall [o/min]
- z : antall tenner
- i : oversetning pr. tannpar

Virkningsgrad defineres som:

$$\eta = P_2 / P_1 = T_2 / (T_1\ i) \qquad (9.7)$$

Total virkningsgrad:

$$\eta_{tot} = \eta_1\ \eta_2\ \eta_3 \ldots \qquad (9.8)$$

Total oversetning:

$$i_{tot} = i_1\ i_2\ i_3 \ldots \qquad (9.9)$$

Og dermed blir

$$P_1 = P_n / \eta_{tot} \qquad (9.10)$$

$$T_1 = T_n / (i_{tot}\ \eta_{tot}) \qquad (9.11)$$

I praksis gjelder følgende erfaringsverdier pr inngrep:

* Virkningsgrad for
- ubearbeidede tenner: $\eta = 0.92 - 0.94$
- finbearbeideing og god smøring: $\eta = 0.96$
- slipte tenner og væskesmøring: $\eta = 0.98$

* Oversetning for:
- drivverk: $i = 4 - 6$
- løfteinnretninger: $i = 7 - 10$

Modulen for et tannhjul er definert som:

$$m = p / \pi \ [mm] \quad (9.12)$$

hvor

p : delingen (summen av tann pluss luke definert langs delesirkelen)

Diameter for tannhjulets delesirkel er gitt av:

$$d = m\,z \quad (9.13)$$

hvor

m : modul
z : antall tenner

Modulrekken er standardisert, og følgende rekke gis 1 prioritet:

1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40, 50

For standard moduler etter 2 og 3 prioritet henvises til NS 5000.

9.3. VIKTIGE MÅL PÅ TANNHJUL

I fig. 9.10 er vist et par tenner av et tannhjul med definerte størrelser påført.

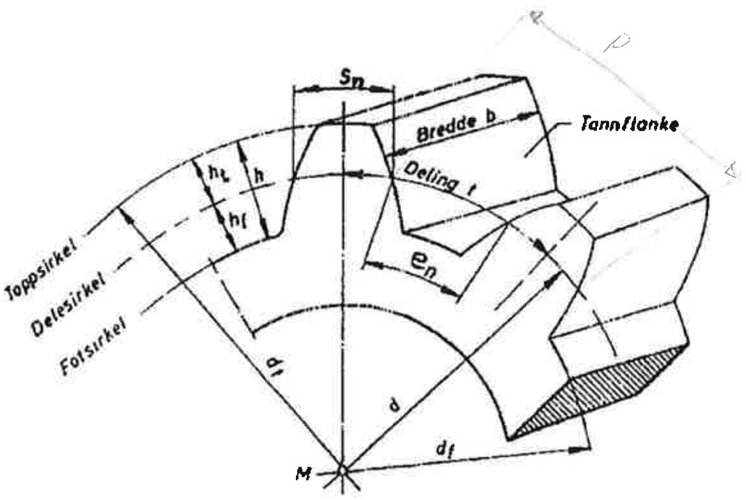


Fig. 9.10 Definerte størrelser

De ulike betegnelser er definert som:

* topphøyden:	$h_t = 1\ m$	(9.13)
* fothøyden:	$h_f = 1.25\ m$	(9.14)
* tannhøyden:	$h = 2.25\ m$	(9.15)
* tanntykkelsen:	$s_n = p / 2 - 0.05\ m$	(9.16)
* lukevidde:	$e_n = p / 2 + 0.05\ m$	(9.17)

hvor

m : modul
p : deling

Fothøyden h_f er over satt lik 1.25 m (standard verdi), men kan variere mellom 1.1 m og 1.3 m. Teoretisk verdi for tanntykkelsen s_n og lukevidde e_n er p / 2. Beregning av s_n og e_n etter formlene over gir en flankeklaring lik 0.1 m. Flankeklaringen kontrolleres med bladsøkerblad (føler), mens for å kontrollere inngrepet benyttes merkestift.

For et tannhjul defineres følgende tre diametere:

* delediameter:	$d = m\ z$	(9.18)
* toppdiameter:	$d_t = m\ z + 2\ h_t$	(9.19)
* bunndiameter:	$d_f = m\ z - 2\ h_f$	(9.20)

hvor

m : modul
z : tannantall
h : tannhøyde

Et annet viktige mål er tannhjulets maksimale bredde:

$$b_{maks} = \lambda\ m \qquad (9.21)$$

hvor

λ : breddefaktor
m : modul

Breddefaktoren velges ut fra tennenes bearbeiding og tannhjulets opplagring:

* Ubearbeidede tenner:	$\lambda \leq 6$
* Bearbeidede tenner og vanlig opplagring:	$\lambda = 8 - 12$
* Slipte tenner og nøyaktig opplagring:	$\lambda \leq 30$

Praktisk bruk av formelen for toppdiameteren d_t er å måle diameteren og deretter telle antall tenner på tannhjulet. Modulen kan da bestemmes fra formelen:

$$m = \frac{d_t}{z + 2} \qquad (9.22)$$

Senteravstanden a mellom to tannhjul er gitt av formelen:

$$a = \frac{m(z_1 + z_2)}{2} \qquad (9.23)$$

hvor

m : modul
z_1 : tannantall for tannhjul 1
z_2 : tannantall for tannhjul 2

9.4. KRAFTBILDET I ET TANNHJULSPAR.

Fig. 9.11 viser to rettfortannede tannhjul i inngrep der tannhjul 1 er drevet og tannhjul 2 er hjulet.

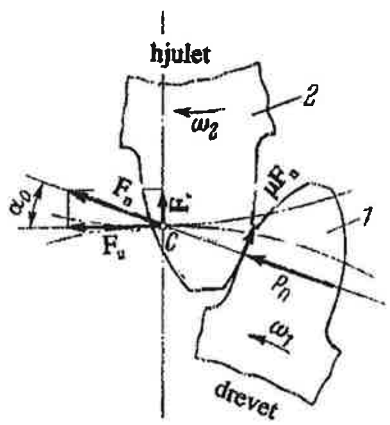


Fig 9.11 **Kraftbildet i et tannhjulspar**

Den overførte kraften er F_n gir friksjonskraften μF_n. Den overførte kraften dekomponeres i det felles tangeringspunktet (tangeringspunktet for delesirklene) til en periferikraft og en radialkraft hhv :

$$F_u = 2\,T / d \qquad (9.24)$$

$$F_r = F_u \tan \alpha_0 \qquad (9.25)$$

hvor

T : dreiemoment [Nm]
d : delediameter [m]

Radialkraften F_r virker alltid i retning av hjulets opplagring (akselsenter)

For skråfortannede tannhjul er kraftbildet vist i fig 9.12

Appendix G

Appendix G - MASKINKONSTRUKTION II, Beregning av sylindriske tannhjul med rette tenner

BEREGNING AV SYLINDRISKE TANNHJUL MED RETTE TENNER

UTDRAG FRA
MASKINKONSTRUKSJON II
C.ZEGVELD
J. MEIJER
(NKI)

b Med hensyn til kontaktspenning:

$$m = 1{,}84 \sqrt[3]{\frac{M_w \cdot K_a \cdot K_v}{k \cdot z^2 \cdot \lambda} \cdot \frac{i+1}{i}}$$

$$m = 1{,}84 \sqrt[3]{\frac{6369{,}42 \cdot 10^3 \mathrm{N} \cdot \mathrm{mm} \cdot 1{,}25 \cdot 1{,}3}{7 \mathrm{N/mm}^2 \cdot 6400 \cdot 15} \cdot \frac{2+1}{2}}$$

$m = 5{,}2$ mm, som vi runder av til $m = 6$ mm.

En modul på 6 mm ser ut til å holde både når det gjelder tannbrudd og kontaktspenning.

4.29 Oppgaver

Kontrollspørsmål

1 Nevn fire typer av skader som tannhjul kan bli utsatt for.

2 *a* Hvorfor oppstår slitasje på tannflankene generelt?
b Nevn noen faktorer som bidrar til å minske denne slitasjen.

3 Nevn årsaken til riving av tannflankene.

4 *a* Hva mener vi med tanntretthetsbrudd?
b Hva mener vi med voldsomt brudd?

5 Når oppstår groptæring på tannflankene til samvirkende tannhjul, og hva kan det føre til?

6 Hvilke krefter består den resulterende periferikraften av som vi bruker ved beregninger?

7 *a* Hvordan oppstår den ytre dynamiske kraften i roterende tannhjul?
b Hvordan oppstår den indre dynamiske kraften?

Øvingsoppgaver

1 I en tannhjulsoverføring har drivhjulet 25 tenner og det drevne hjulet 80 tenner. Inngrepsvinkelen er 20°, modulen 8 mm og tannbredden 80 mm. Effekten *P*, som skal overføres, er 8 kW når drivhjulet har en rotasjonsfrekvens $n = 2\ \mathrm{s}^{-1}$. Materialet er C 60 N i begge tannhjulene. Viskositeten til oljen som tannhjulene blir smurt med, er 19 °E. Driftsfaktoren $A = 5$ (se tabell 4.7). Driftstiden er 8 til 10 timer per dag. Belastningsfaktoren $K_a = 1{,}25$ og tannformfaktoren $\gamma = 2{,}73$. Kontrollregn med hensyn til tannbrudd.

2 Bruk dataene fra oppgave 1, beregn kontaktspenningen og kontroller om den er tillatt.

3 Et tannhjul av materialet C 45 N har 60 tenner med en modul på 8 mm og overfører en effekt på 10 kW. Rotasjonsfrekvensen til tannhjulet er $n = 4\ s^{-1}$. Driften er en sterkt støtaktig, og driftstiden er 8 til 10 timer per dag. Tannhjulene er nøyaktig framstilt. Tannbredden $b = 120$ mm. Kontrollregn med hensyn til tannbrudd.

4 Tannhjulsoverføringen i oppgave 3 blir smurt med olje med viskositeten 19 °E. Det drevne tannhjulet har 120 tenner, og rotasjonsfrekvensen $n = 2\ s^{-1}$. Finn kontaktspenningen.

5 Drivverket til en mobilkran blir drevet av en elektromotor med en effekt på 12 kW og en rotasjonsfrekvens $n = 24\ s^{-1}$. Drivakselen i tannhjulsvekselen blir drevet direkte av motoren. Drivhjulet har 22 tenner og det drevne hjulet 130 tenner. Belastningsfaktoren $K_a = 1{,}12$. For hastighetsfaktoren bruker vi på delesirkelen $K_v = 3$. Tannhjulene er laget av krom-molybdenstål 42 CrMo 4 V med $\sigma_o = 185\ N/mm^2$ og flankestyrkefaktoren $k = 7{,}8\ N/mm^2$. Tannformfaktoren $\gamma = 2{,}8$ og tannbreddefaktoren $\lambda = 10$. Finn:

a modulen med hensyn til styrke

b modulen med hensyn til kontaktspenning

6 Figur 4.61 viser en lukket tannhjulsveksel. Tannhjulene 1 og 2 har skrå tenner med en normalmodul $m_n = 2$ mm og med henholdsvis $z_1 = 36$ og $z_2 = 45$ tenner. Senteravstanden til hjulene med skrå tenner er 83 mm. Tannhjulene 3 og 4 er av stål og har rette tenner. $z_3 = 20$, og $z_4 = 60$ tenner. Senteravstanden til disse hjulene er 80 mm. Tannbredden er 20 mm. Inngangseffekten $P = 2$ kW ved en rotasjonsfrekvens $n = 15\ s^{-1}$. Belastningsfaktoren $K_a = 1{,}25$, og den dynamiske belastningsfaktoren $K_v = 1{,}15$. Finn:

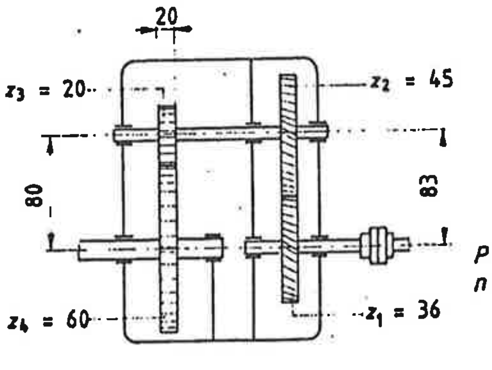
a den utgående rotasjonsfrekvensen til tannhjulsvekselen

b modulen til tannhjulene 3 og 4

c skråvinkelen β til hjulene 1 og 2

d bøyespenningen i tannfoten til hjul 3 og finn ut om den blir overskredet når $\sigma_b = 300\ N/mm^2$

e kontaktspenningen som opptrer mellom tannflankene til hjulene 3 og 4, og kontroller at den ikke blir overskredet når $\sigma_o = 900\ N/mm^2$



Figur 4.61 Tannhjulskasse

4.29

1	5,866
2	431 N/mm^2
3	14,34
4	195 N/mm^2
5*a*	2,906 mm
b	3,252 mm
6*a*	4 s^{-1}
b	2 mm
c	12,6°
d	138,2 N/mm^2
e	850 N/mm^2

4.27

Beregning av sylindriske tannhjul med rette tenner

I avsnitt 4.12 kom vi inn på kraftvirkningen mellom to samvirkende, sylindriske tannhjul med rette tenner. Se figur 4.29.

For å få litt innsikt i hvorfor tenner slites eller blir skadd, skal vi se på de vanligste årsakene til skader.

Ved beregninger av *tannbrudd* og *kontaktspenninger* holder vi oss til gjeldende ISO-normer, angitt i DIN 3990.

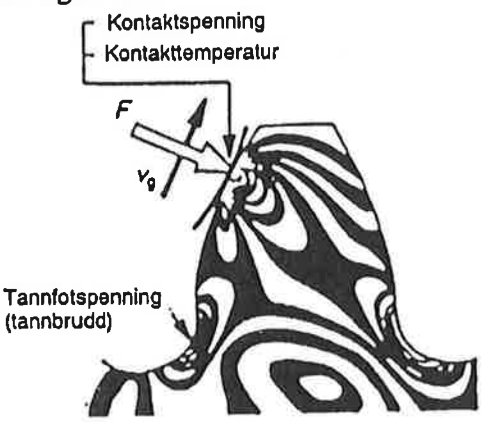
4.27.1

Typer av skader

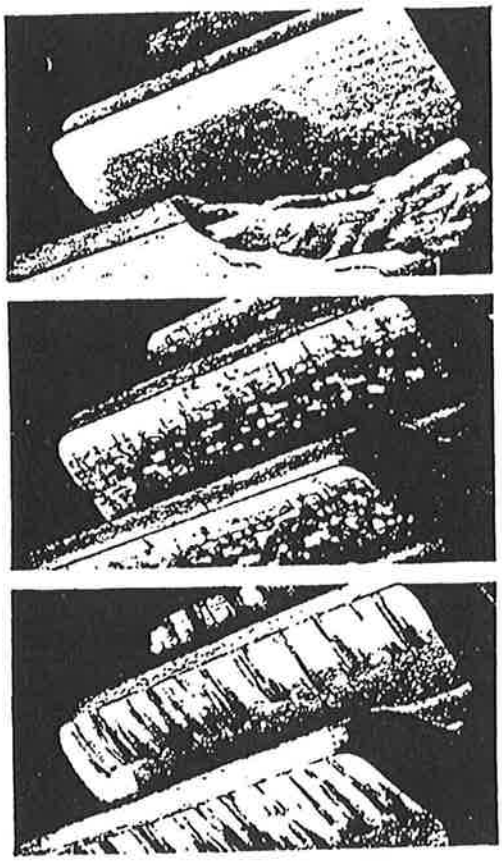
De typer av skader som vanligvis opptrer på tannhjul, er:

- slitasje
- riving
- tannbrudd
- groptæring

Se figur 4.59.



a Spenninger forårsaket av kreftene som virker på tennene, og av glidehastigheten



b Skader som kan forekomme

Figur 4.59 Skader som følge av tannbelastning

- *Slitasje*

Fordi tannflankene til to samvirkende tannhjul ikke ruller 100 % over hverandre, men også til dels gnisser mot hverandre, blir tennene etter hvert slitt. Graden av slitasje avhenger blant annet av disse faktorene:
- overflatekvaliteten til tannflankene
- smøringen
- tannbelastningen
- tannmaterialet

Smøring av tannhjulene er en effektiv måte å motvirke slitasje på. Det er bare mulig å danne en oljefilm dersom tilstrekkelig med smøremiddel blir tilført, og *glidehastigheten* er stor nok. Det framgår av dette at vi ikke kan vente nevneverdig slitasje dersom overflatekvaliteten til tannflankene er god, tannhjulsvekselen er innkapslet og smøremiddelet har den rette viskositeten.

Åpne tannhjulsoverføringer er imidlertid svært følsomme for slitasje fordi de som oftest er saktegående, utilstrekkelig smurt og utsatt for forurensninger gjennom smøremiddelet (fett).

- *Riving*

Sterkt belastede tannhjul med stor rotasjonshastighet er ofte utsatt for riving. Skaden oppstår når det blir gjennombrudd i oljefilmen, slik at det blir mekanisk kontakt mellom tannflankene på de samvirkende tannhjulene. Temperaturen kan da bli så høy at tannflankene blir sveiset sammen. Når tannhjulet fortsetter å rotere, blir sveisen revet opp. Det kan føre til alvorlige skader på tannflankene. Se figur 4.59.

- *Tannbrudd*

Tannbrudd er den alvorligste formen for skade på tannhjul. Vi skiller mellom to typer av tannbrudd: tretthetsbrudd og voldsomt brudd. Tretthetsbrudd henger i stor grad sammen med tannmaterialet og varigheten av belastningen på tannhjulet. Voldsomt brudd oppstår når spenningen i tannfoten overstiger bruddspenningen til materialet, for eksempel fordi den drevne maskinen bråstopper.

- *Groptæring*

Groptæring er den vanligste skaden på tannhjul. Den forekommer på tannflanken mellom delesirkelen og fotsirkelen. Groptæring er en form for materialtretthet som oppstår når *kontaktspenningen* mellom tannflankene er for høy. På grunn av den høye kontaktspenningen blir det dannet hårfine sprekker i materialet. Sprekkene blir fylt med olje, og når tannflankene ruller over hverandre, blir oljen blir presset sammen. Til slutt faller deler av tannflanken ut.

4.27.2 Resulterende periferikraft

Som vi har nevnt tidligere, er rullesirklene til to samvirkende tannhjul de sirklene hvor periferihastigheten er den samme. Periferikraften F_t som angriper i rullesirklene til tannhjul, må vi kjenne når vi skal foreta tannberegninger. Denne kraften er i prinsippet en resultant av tre krefter, nemlig den *teoretiske* eller *statiske periferikraften* (som vi beregner av det dreiemomentet tannhjulene overfører), den *ytre dynamiske kraften* som den drivende og den drevne mekanismen forårsaker, og den *indre dynamiske kraften* som tannhjulene selv forårsaker.

- *Teoretisk periferikraft*

Den teoretiske periferikraften F_{th} beregner vi av det dreiemomentet tannhjulene overfører:

$$F_{th} = \frac{T}{r}$$

$$F_{th} = \frac{M_w}{r}$$

$$\left[r = \frac{\text{delesirkel dia}}{2} = \frac{d}{2} \text{ i mm} \quad F_{th} \text{ i N} \quad M_w \text{ i Nmm} \right]$$

- *Ytre dynamisk belastning*

Den ytre dynamiske belastningen på et tannhjul skyldes at drivmekanismen ikke avgir et konstant moment til tannhjulsoverføringen, og at den drevne mekanismen ikke mottar et konstant moment fra tannhjulsoverføringen. Det er med andre ord

Tabell 4.6 Belastningsfaktor K_a

Driv-mekanisme	Driftstimer per dag	Jevn	Lette støt	Tunge støt
Elektromotor eller hydromotor	8-10 timer	1	1,25	1,75
	2-3 timer	0,8	1	1,50

jevn	**lette støt**	**tunge støt**
lette transportenheter	betongblandere	steinknusere
blandeapparater	blandeverk	veihøvler
sorteringsbånd	traverskraner	presser
vinsjer	svingkraner	kulemøller
sentrifugalpumper	stempelpumper med flere sylindre	sakser
		grabber
		mudderapparater
		gravemaskiner

graden av *uensartethet* mellom drivmekanismen og den drevne mekanismen som bestemmer størrelsen av den ytre dynamiske belastningen. Fordi det som regel ikke er mulig eksakt å bestemme belastningsvariasjonene, bruker vi ved beregninger en faktor K_a, som vi multipliserer den teoretiske periferikraften F_{th} med.

I tabell 4.6 er belastningsfaktoren K_a angitt ved forskjellige driftsforhold og for ulike maskiner.

- *Indre dynamisk belastning*

Den indre dynamiske belastningen skyldes selve tannhjulene og er avhengig av periferihastigheten til tannhjulene, nøyaktigheten ved tilvirkningen av tennene, eksentrisiteten til fortanningen i forhold til boringen og massen til hjulene. Ved beregninger multipliserer vi den teoretiske periferikraften F_{th} med en faktor K_v. Den dynamiske faktoren blir beregnet med formelen:

$$K_v = \frac{A + v}{A} \tag{4.19}$$

hvor:

K_v = den dynamiske belastningsfaktoren
A = driftsfaktoren (m/s)
v = periferihastigheten til delesirkelen i m/s

De vanligste verdiene for driftsfaktoren A er angitt i tabell 4.7.

Tabell 4.7 Driftsfaktor A

Tannhjulstype	A (m/s)
Svært nøyaktig tilvirket (slipt) tannhjul	10
Nøyaktig tilvirket tannhjul	5
Vanlig tannhjul, inkl. åpen drift	3

Størrelsen på den retningsgivende periferikraften ved beregninger får vi ved å multiplisere den teoretiske periferikraften F_{th} med den ytre dynamiske belastningsfaktoren K_a og den indre dynamiske faktoren K_v. Det vil si:

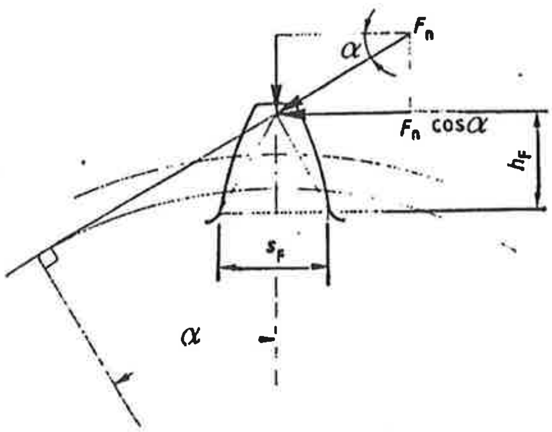
$$F_t = K_a \cdot K_v \cdot F_{th} \tag{4.20}$$

hvor:

F_t = den retningsgivende periferikraften
K_a = den ytre dynamiske belastningen
K_v = den indre dynamiske belastningen
F_{th} = den teoretiske periferikraften

4.27.3 Tannbruddsberegninger

I et drivende tannhjul forflytter tannkraften F_n seg via tannflanken fra tannfoten til tanntoppen. Tannkraften *bøyebelaster, trykkbelaster* og *skjærbelaster* tannen. Bøyemomentet på en tann er maksimalt når den alene må overføre den totale effekten, og når kraften angriper lengst fra tannfoten. Selv om belastningen som regel er fordelt på to tannpar i inngrep, går vi av sikkerhetsmessige grunner ut fra at den tangentielle komponenten av tannkraften $F_n (F_t = F_n \cdot \cos \alpha)$ angriper jevnt fordelt på toppen av tanna. Se figur 4.60.



Figur 4.60 Tannkrefter

Beregningsmetoden til professor Niemann legger belastningene på tannfoten til grunn. I motsetning til denne metoden fastslår den forenklede beregningsmetoden ifølge DIN 3990 at den beregnede og den målte spenningen stemmer best overens når bare bøyebelastningen på tannfoten blir beregnet.

Ifølge figur 4.60 er bøyebelastningen på tannfoten lik $F_n \cdot \cos \alpha$. Kraften F_n er her i kraftretningen forskjøvet til den vertikale senterlinjen til tanna slik at $F_n \cdot \cos \alpha$ angriper i en høyde h_F, regnet fra avrundingen av tannfoten.

$$F_n = \frac{F_{th} \cdot K_a \cdot K_v}{\cos \alpha}$$

Bøyemomentet som tannfoten blir belastet med, er:

$$M_b = \frac{F_{th} \cdot K_a \cdot K_v \cdot \cos \alpha \cdot h_F}{\cos \alpha}$$

Spenningen i tannfoten får vi av formelen:

$$\sigma_b = \frac{F_{th} \cdot K_a \cdot K_v \cdot \cos\alpha \cdot h_F \cdot 6}{b \cdot S_F^2} \text{ som } W_b = \frac{1}{6} \cdot b \cdot S_F^2$$

Multipliserer vi nå med modulen m i teller og nevner, får vi:

$$\sigma_b = \frac{F_{th} \cdot K_a \cdot K_v}{b \cdot m} \cdot \frac{6 \cdot m \cdot h_F \cdot \cos\alpha}{S_F^2 \cdot \cos\alpha}$$

Deretter setter vi:

$$\gamma = \frac{6 \cdot m \cdot h_F \cdot \cos\alpha}{S_F^2 \cdot \cos\alpha}$$

Her er γ tannformfaktoren. Formelen blir da slik:

$$\sigma_b = \frac{F_{th} \cdot K_a \cdot K_v}{b \cdot m} \cdot \gamma \qquad (4.21)$$

hvor:
σ_b = bøyespenningen i tannfoten i N/mm^2
b = tannbredden i mm
m = modulen til fortanningen i mm
γ = tannformfaktoren. Se tabell

Tannformfaktoren er avhengig av antall tenner og av profilforskyvningsfaktoren x. Vi skal imidlertid begrense oss til å beregne ukorrigerte tenner. Tabell 4.8 inneholder derfor bare tannformfaktoren for ukorrigerte tenner.

Tabell 4.8

Antall tenner z	Tannformfaktor γ	Antall tenner z	Tannformfaktor γ
15	–	40	2,45
20	2,90	60	2,30
25	2,73	80	2,24
30	2,60	100	2,21

Den maksimalt tillatte fotspenningen for ubegrenset levetid er empirisk bestemt. I tabell 4.12 er verdien på denne spenningen for ulike materialer angitt som $\sigma_{b(lim)}$.

Når vi skal fastsette den tillatte fotspenningen, innfører vi sikkerhetsfaktoren V. Den maksimalt tillatte fotspenningen for samvirkende tannhjul, som ikke må overskrides, er da:

$$\bar{\sigma}_b = \frac{\sigma_{b_{lim}}}{V_b} \quad (4.22)$$

$\sigma_b < \bar{\sigma}_{b\ \text{tillatt}}$

hvor: $\bar{\sigma}_b$ = den tillatte fotspenningen i N/mm^2
$\sigma_{b(lim)}$ = den eksperimentelt bestemte maksimale fotspenningen ved ubegrenset levetid. Se tabell 4.12
V_b = sikkerhetsfaktoren (er som regel 1,7)

- *Modulstørrelse*

Når vi skal beregne modulen, går vi ut fra spenningsformelen:

$$\sigma_b = \frac{F_{th} \cdot K_a \cdot K_v}{b \cdot m} \cdot \gamma$$

Uttrykker vi deretter den effektive tannbredden i forhold til modulen, får vi:

$$b = \lambda \cdot m$$

hvor: λ = tannbreddefaktoren. Se tabell 4.9

Tabell 4.9 Tannbreddefaktor

Tannhjulsoverføring	λ
Innebygde tannhjul	30
Tannhjul montert mellom lagre med en begrenset senteravstand	15
Tannhjul for enkle konstruksjoner	8

Vi multipliserer først med $\frac{d}{2}$ på begge sider av likhetstegnet, og setter inn $b = \lambda \cdot m$, slik at vi får:

$$\frac{d}{2} \cdot \sigma_b = \frac{F_{th} \cdot \frac{d}{2} \cdot K_a \cdot K_v}{\lambda \cdot m \cdot m} \cdot \gamma$$

Deretter setter vi inn vrimomentet $M_w = F_{th} \cdot d/2$ og så $d = z \cdot m$, slik at vi får:

eller:

$$z \cdot m \cdot \lambda \cdot m \cdot m \cdot \sigma_b = 2 \cdot M_w \cdot K_a \cdot K_v \cdot \gamma$$

Det vil si:

$$m^3 = \frac{2 \cdot M_w \cdot K_a \cdot K_v}{z \cdot \lambda \cdot \sigma_b} \cdot \gamma$$

Da får vi:

$$m = 1{,}26 \sqrt[3]{\frac{M_w \cdot K_a \cdot K_v}{z \cdot \lambda \cdot \sigma_b} \cdot \gamma} \qquad (4.23)$$

hvor: z = antall tenner

4.27.4 Beregning av kontaktspenningen

Ved kontrollregning av spenningen som opptrer i kontaktflaten mellom to sylindere (tannflanker) som blir trykt mot hverandre, går vi ut fra en formel stilt opp av Hertz. Ifølge en utvidet variant av formelen til Hertz tilpasset kontaktspenningen mellom tannflankene til to samvirkende tannhjul, gjelder:

$$\sigma_o = f_w \cdot f_c \sqrt{\frac{F_{th}}{b \cdot d_1} \cdot \frac{i+1}{i} \cdot K_a \cdot K_v} \qquad (4.24)$$

hvor:

σ_o = kontaktspenningen i N/mm²
f_w = materialfaktoren for samvirkende tannhjul
f_c = flankeformfaktoren (for en evolvent tannform gjelder $f_c = 1{,}76$)
d_1 = delesirkeldiameteren til drivhjulet
i = oversetningsforholdet

Når det gjelder materialfaktoren f_w, kan vi for to samvirkende stålhjul slå fast:

$$f_w = \sqrt{0{,}35 \cdot E}$$

eller:

$$f_w = \sqrt{0{,}35 \cdot 210 \cdot 10^3 \text{ N/mm}^2} = 271\sqrt{\text{N/mm}^2}$$

Når et stålhjul virker sammen med et hjul av et annet materiale, gjelder:

$$f_w = \sqrt{0{,}35 \cdot \frac{2 \cdot E_1 \cdot E_2}{E_1 \cdot E_2}}$$

Når vi skal beregne kontaktspenningen mellom to tannflanker, dividerer vi den maksimalt bestemte empiriske kontaktspenningen med en sikkerhetsfaktor. ~~I tabell 4.12 er sikkerhetsfaktoren angitt for en del forskjellige materialer.~~ Den tillatte kontaktspenningen blir:

$$\overline{\sigma_o} = \frac{\sigma_{o_{lim}}}{V_o} \cdot K_L \cdot Z_v \qquad (4.25)$$

hvor:

$\overline{\sigma_o}$ = den tillatte kontaktspenningen i N/mm^2

$\sigma_{o(lim)}$ = den eksperimentelt bestemte kontaktspenningen for ubegrenset levetid

V_o = sikkerhetsfaktoren (Vanlig 1,25)

K_L = smøreoljefaktoren, som har sammenheng med viskositeten til smøreoljen. Se tabell 4.10

Z_v = hastighetsfaktoren, som har sammenheng med periferihastigheten til hjulene. Se tabell 4.11

Tabell 4.10 Smøreoljefaktor

Viskositet i °E	5	9	13,5	19	26
K_L	0,9	0,95	1	1,05	1,1

Tabell 4.11 Hastighetsfaktor

v i m/s	0,25	1	2	3	4	5
Z_v	0,835	0,842	0,855	0,877	0,905	0,932
v i m/s	6	7	8	9	10	12
Z_v	0,960	0,980	1	1,015	1,033	1,058

Tabell 4.12

Material-betegnelse ifølge DIN	Brinell-hardhet i kgf/mm^2	σ_B i N/mm^2	$\sigma_{b_{lim}}$ * i N/mm^2	$\sigma_{o_{lim}}$ * i N/mm^2
Fe 430	125	450	160	430
Fe 590	180	650	210	620
C 45 N	185	650	220	540
C 60 N	220	800	250	610
34 Cr 4 V	260	900	300	715
42 CrMo 4 V	280	950	310	760
16 MnCr 5	650	1000	410	1600
15 CrNi 6	650	1600	410	1900

2100

* Empirisk bestemte grenseverdier for ubegrenset levetid

910

- *Beregning av modulen med hensyn til kontaktspenningen*

Ved beregning av modulen bruker vi også avledning av en formel av Hertz. Den er slik:

$$m = 1{,}84 \sqrt[3]{\frac{K_a \cdot K_v \cdot M_w}{k \cdot z^2 \cdot \lambda} \cdot \left(\frac{i+1}{i}\right)} \qquad (4.26)$$

hvor: k = flankestyrkefaktoren i N/mm^2. Se tabell 4.13.

Dette er en verdi av flankestyrken $\left(k = \dfrac{\sigma^2_{o_{lim}}}{0{,}35 \cdot E}\right)$ ved ubegrenset levetid som er funnet ved prøver.

Tabell 4.13 Flankestyrkefaktor

Material-betegnelse ifølge DIN	Tillatt fotspenning i N/mm^2 $\left(\frac{\sigma_{b\,lim}}{1{,}7}\right)$	Tillatt flankestyrkefaktor k i N/mm^2
Fe 430	95	2,5
Fe 590	125	5,2
C 45 N	130	4,0
C 60 N	150	5,0
34 Cr 4 V	180	7,0
42 CrMo 4 V	185	7,8
16 MnCr 5	240	35,0
15 CrNi 6	240	49,0

Appendix H

Appendix H - Rullingslager Kapittel 7

7. RULLINGSLAGER

Det finnes i hovedsak to ulike lagerprinsipper: **glidelager** (akseltapp løper direkte på lagerflaten dog med smøreolje mellom) og **rullingslager** (mellomliggende lag av herdede stålkuler eller -ruller). I dette kapittel er det kun rullingslager som blir behandlet. Det vises typer av rullingslager og nødvendig underlag for å velge riktig type og størrelse på lageret

7.1. ULIKE LAGERTYPER

Rullingslager er felles betegnelse for kulelager og rullelager. Av begge lager finnes flere typer. Disse deles gjerne i følgende grupper:

* Radiallager; benyttes primært ved radiell belastning, et utvalg er vist i fig. 7.1.
* Aksiallager; benyttes primært ved aksiell belastning og er vist i fig. 7.2.
* Y - lager; disse opptar relativt store oppretningsfeil, men tillater ikke aksielle forskyvninger, og er vist i fig. 7.3.

Av disse lager finnes flere varianter av både kule- og rullelager. Lagerne vist i fig. 7.1., 7.2 , og 7.3. viser et lite utvalg av mulige lagervarianter.

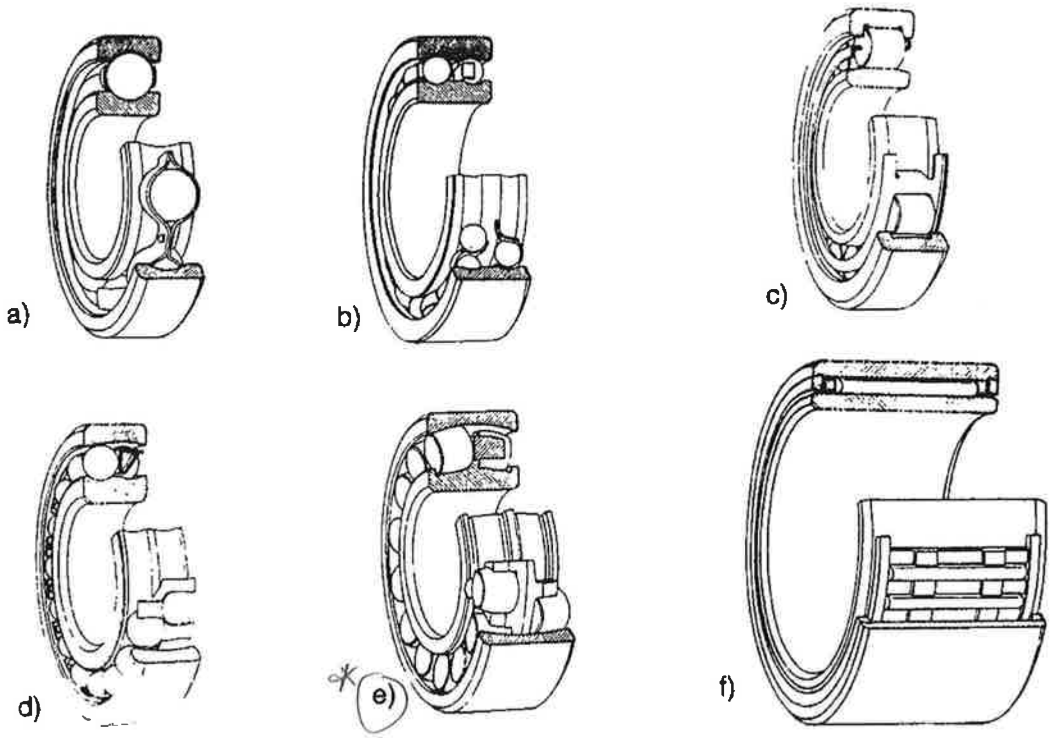


Fig. 7.1. Et utvalg av radiallager a) Enrads sporkulelager, b) Torads vinkelkontaktkulelager, c) Sylindrisk rullelager, d) Sfærisk kulelager, e) Sfærisk rullelager, f) Nålelager.

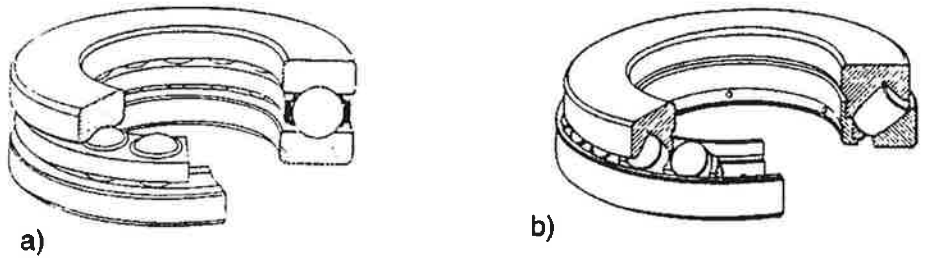


Fig. 7.2. Et utvalg av aksiallager. a) Enkeltvirkende aksialkulelager, b) Sfærisk aksial rullelager.

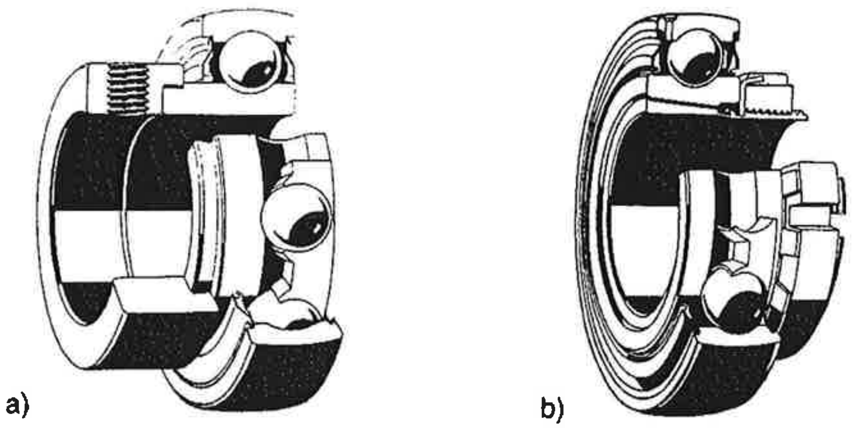


Fig. 7.3. Y-lager. a) Med eksentrisk låsering, b) Med klemhylse

7.2. VALG AV LAGERTYPE

For valg av lagertype må både belastningens størrelse og retning være kjent. Generelt gjelder for:

* Belastningens størrelse:

- Rullelager kan belastes mer enn kulelager med samme yttermål.
- Lager med maksimalt antall kuler/ruller (uten holder) kan belastes mer enn tilsvarende lager med holder.
- Ved små og mellomstore belastninger benyttes for det meste kulelager, mens rullelager er vanligst ved større belastninger og for store akseldiametere.

* Belastningens retning:

- Radiell belastning: her benyttes spor -, sfæriske -, vinkelkontakt kule-lager, sylindriske -, nål-, sfæriske -, og koniske rullelager.
- Aksiell belastning: de fleste radiallager (unntatt nålrullelager) kan oppta aksiell belastning, de best egnede er imidlertid sfæriske aksial-, aksial-, nål-, sylindriske aksial-, sylindriske aksial- og koniske rullelager.

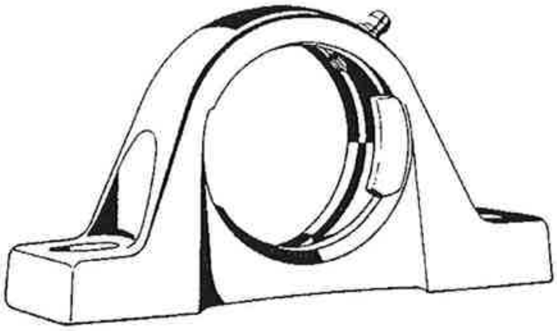
- Kombinert belastning (både radiell og aksiell belastning) : sporkulelager, vinkelkontaktlager, sfæriske - og koniske rullelager

For mer informasjon og hvilke typer lager som bør velges ved ulike belastninger henvises til produsentenes varekataloger.

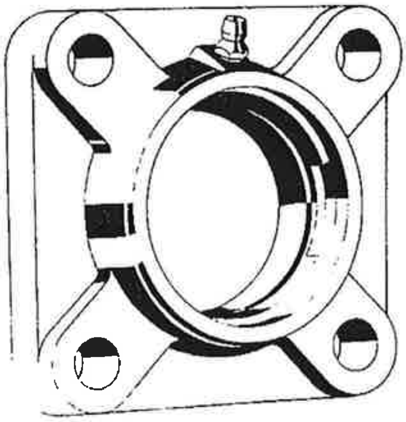
Konstruktørens oppgave blir dermed å:

* velge **riktig lagertype** med krav til den funksjon lageret skal oppfylle.
* velge **riktig lagerstørrelse** med krav til lagerets levetid.
* velge **innbygning av lageret** (lagerhus) avhengig av konstruktiv utforming og miljø.

I fig. 7.4 er vist to typer lagerhus avhengig av monteringen Det ene monteres på horisontalt underlag mens det andre benyttes ved vertikal montasje



a)

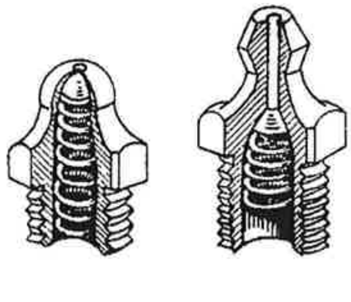


b)

Fig. 7.4. Lagerhus a) For horisontal montasje, b) For vertikal montasje.

På lagerhusene, vist i fig. 7.4., er det montert fettnipler for smøring av lageret med fett. Typiske fettnipler er vist i fig.7.5.a).

a)



b)

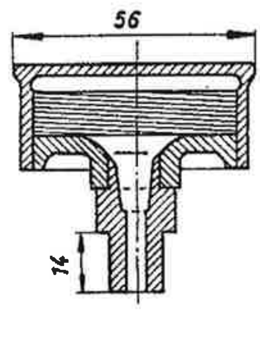


Fig 7.5. a) Fettnipler, b) Smørekopp for fett.

I fig.7.5.b) er vist en smørekopp som også kan benyttes ved fettsmøring. Konstruktørens oppgave er i tillegg til å velge lagerhus, også å velge type smørepunkt og sørge for at disse er tilgjengelige også etter at konstruksjonen er innstallert

7.3. KONSTRUKTIVE UTFØRELSER

I fig. 7.6 er vist et utvalg av mulige løsninger for feste / fiksering av lager på aksel.

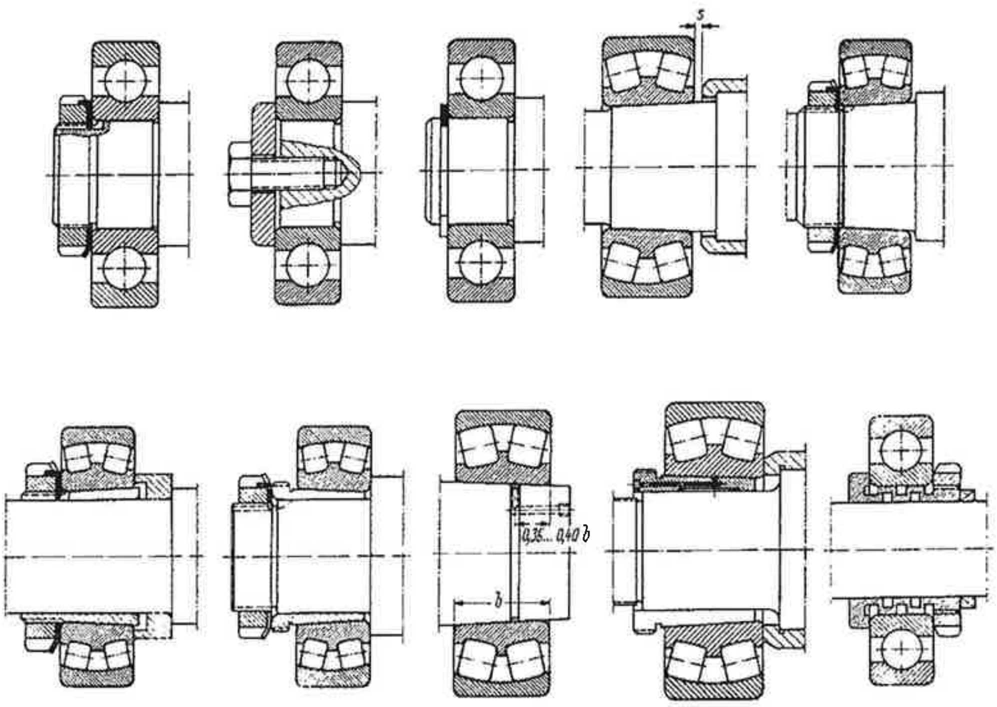


Fig. 7.6. Fiksering av lager på aksel.

I fig. 7.7 er vist et utvalg av mulige løsninger for fiksering av lagerets ytterring.

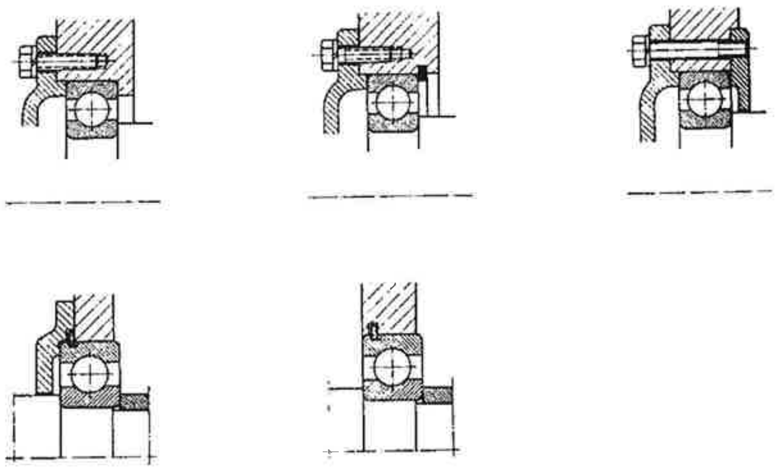


Fig. 7.7. Fiksering av ytterring.

For å beskytte lagerene mot støv og andre partikler benyttes tetninger, de vanligste er filttetning, fjærbelastet mansjettetning og aksial labyrinttetning. De to første inngår i gruppen slepende tetninger, mens den siste inngår i gruppen ikke slepende tetninger.

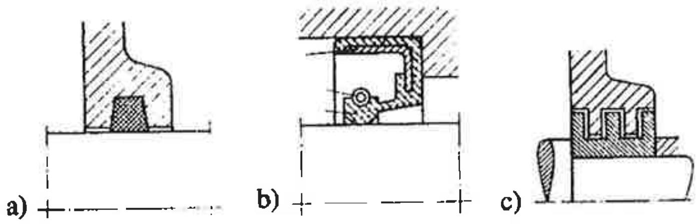


Fig. 7.8 Utvalg av tetninger. a) Filttetning, b) Fjærbelastet mansjettetning, c) Aksial labyrint-tetning

7.4. VALG AV LAGERSTØRRELSE

For valg av lagerstørrelse med hensyn på levetid kan ISO's formel for nominell levetid benyttes:

$$L_{10} = \left(\frac{C}{P} \right)^{\cancel{p}\ 3} \qquad (7.1)$$

FA/FB

hvor

L_{10} : nominell levetid [mill omdr.]

C : dynamisk bæretall [N]; definert som den last som gir lageret en levetid på 1 mill. omdreininger ved 90 % pålitelighet.

P : ekvivalent lagerlast [N]

p : eksponent som settes lik 3 for kulelager og 10/3 for rullelager

Den ekvivalente lagerlast beregnes på følgende måte:

Radiallast	: $P = F_r$	(7.2)
Aksiallast	: $P = F_a$	(7.3)
* Kombinert last	: $P = X\,F_r + Y\,F_a$	(7.4)
	hvor X:radialfaktor og Y:aksialfaktor	

For å kunne bestemme faktorene X og Y, må forholdet F_a/C_0 beregnes. Det statiske bæretallet C_0 [N] er definert som den last som gir en varig deformasjon for kule/rulle og rullebane lik:

$$\Delta d = 0.0001\ d \qquad (7.5)$$

hvor

d : diameteren av kule/rulle

Når forholdet F_a/C_0 er beregnet finnes en karakteristisk størrelse e fra tabell 7.1. For ensporede lager kan den ekvivalente lagerlast bestemmes utfra følgende:

$$P = F_r \qquad \text{når } F_a/F_r \le e \qquad (7.6)$$

$$P = X\,F_r + Y\,F_a \qquad \text{når } F_a/F_r > e \qquad (7.7)$$

For lager i O- eller X-anordning henvises til produsentens varekatalog. Disse anordningene benyttes ved forspenning av lager.

F_a / C_0	e	X	Y
0.025	0.22	0.56	2.0
0.04	0.24	0.56	1.8
0.07	0.27	0.56	1.6
0.13	0.31	0.56	1.4
0.25	0.37	0.56	1.2
0.50	0.44	0.56	1.0

Tabell 7.1 Beregningsfaktorer for enradede sporkulelager (enkeltstående eller i tandem-anordning). Ved andre anordninger henvises til lagerprodusentens varekatalog

For et lager med konstant turtall kan levetiden i driftstimer beregnes av formelen:

$$L_{10h} = \frac{L_{10}\ 10^6}{n\ 60} \quad \text{timer} \tag{7.7}$$

hvor

n : turtallet [omdr./min.]

For både kule- og rullelager gjelder nomogramet, vist i fig. 7.9. Nomogrammet består av

n : turtallet [omdr./min.]
C : dynamisk bæretall [N]
P : ekvivalent lagerlast [N]
L_{10} : levetid [mill.omdr.]
L_{10h} : levetid [driftstimer]

Appendix I

Appendix I – Sfåfiska rullager

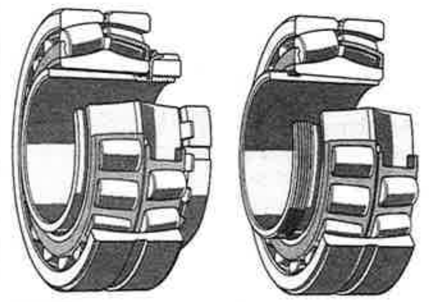
Sfäriska rullager

Det sfäriska rullagret har två rader rullar med gemensam sfärisk löpbana i ytterringen. På innerringen har rullraderna var sin löpbana, som är snedställd i förhållande till lageraxeln. Lagret är självinställande och därför okänsligt för de snedställningar av axeln i förhållande till lagerhuset som kan uppstå vid montering eller till följd av axelns utböjning. Förutom radialbelastningar kan lagret också överföra axialbelastningar i båda riktningarna.

SKF sfäriska rullager har ett stort antal långa symmetriska rullar med stor diameter, vilket ger en mycket stor bärförmåga. Den inre konstruktionen hos dessa lager har genom åren genomgått en kontinuerlig förbättring. Speciella löpbaneformer med optimerad ytjämnhet säkerställer minimal friktion hos SKF sfäriska rullager, speciellt utförandena E, CC och CAC. Lagren har lägre driftstemperaturer eller kan överföra större axialbelastningar eller kan arbeta vid högre varvtal än andra sfäriska rullager. SKFs nya E-utförande införs successivt som standard för sfäriska rullager och introduktionen börjar med de mindre storlekarna ur serierna 222 och 223.

De sfäriska rullagren kan levereras med cylindriskt eller koniskt hål. Lagren ur serierna 240 och 241 med koniskt hål har koniciteten 1:30 (efterbeteckning K30), medan övriga lager har koniciteten 1:12 (efterbeteckning K).

SKF levererar kläm- och avdragshylsor med vars hjälp man på ett enkelt sätt kan montera sfäriska rullager med koniskt hål på släta axlar eller axlar med ansatser. Uppgifter om sfäriska rullager med passande kläm- och avdragshylsor finns i tabellerna med början på sidorna 490 respektive 500. Ytterligare information om hylsorna finns i avsnittet "Tillbehör".



Lager på klämhylsa — Lager på avdragshylsa

Utföranden
SKF sfäriska rullager tillverkas i något av de utföranden som beskrivs nedan, beroende på storlek och serie.

Utförande CC, C och EC

Dessa lager är försedda med symmetriska rullar, en innerring utan flänsar och en pressad stålhållare för varje rullrad. Styrringen är centrerad på innerringen. Lagren i EC-utförande har förstärkta rullsatser för högre bärförmåga. Ytjämnheten på rullar och löpbanor hos CC-utförandet är optimerad för att förbättra styrningen av rullarna och reducera friktionen.

Utförande CAC, ECAC, CA och ECA

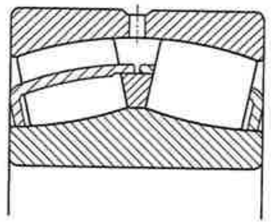
De större storlekarna av SKF sfäriska rullager tillverkas i dessa utföranden. Rullarna är symmetriska och innerringen har fasta styrflänsar. Styrringen är centrerad på innerringen mellan de två rullraderna och hållaren är en massiv s.k. kamhållare av mässing eller stål. Utförandena CAC och ECAC har samma förbättringar beträffande ytjämnheten som CC-lagren, och ECAC och ECA-utförandena är försedda med förstärkta rullsatser för högre bärförmåga.

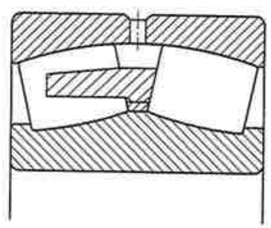
Utförande E

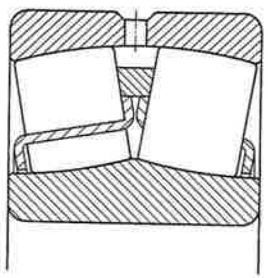
Lagret i SKFs nya standardutförande E är försett med symmetriska rullar, en innerring utan flänsar och en sintrad styrring, centrerad på hållarna, samt en pressad stålhållare för varje rullrad.

E-lagret har alla de fördelar som kännetecknar SKFs väl beprövade CC-lager och är dessutom ytterligare förbättrat. De pressade stålhållarna t.ex. är omkonstruerade och rymmer ett större antal och/eller längre rullar med större diameter, vilket ytterligare ökar lagrets bärförmåga.

Genom den nya placeringen av styrringen förbättras smörjningen i kontakten mellan rullända och styrring. Styrringen bidrar till att reducera friktionen i lagret genom att styra rullarna dels i den obelastade zonen och dels vid inträdet i den belastade zonen.


CC, C och EC-utföranden


CAC, ECAC, CA och ECA-utföranden


E-utförande

Smörjspår och smörjhål

För att smörjningen skall bli enkel och effektiv har SKF sfäriska rullager försetts med ett smörjspår och tre smörjhål i ytterringen som standard (utom lager ur serie 213 CC och lager i CC-utförande med ytterdiameter mindre än 150 eller 180 mm, beroende på lagerserie). Efterbeteckningen W33 används för att identifiera detta utförande hos lagertyperna CC, C, EC, CAC, ECAC, CA och ECA. Däremot har lagren i utförande E ingen sådan efterbeteckning eftersom smörjspår och tre smörjhål ingår i standardutförandet. Om man önskar E-lager utan smörjspår och smörjhål i ytterringen skall efterbeteckningen W läggas till lagerbeteckningen, t.ex. 22312 EW eller 22312 EKW.

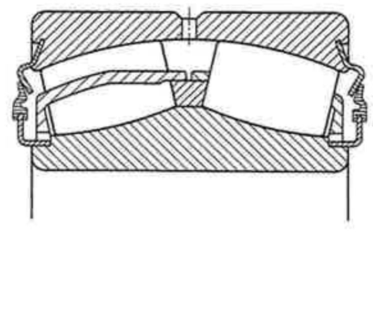
Lager med tätningar
För lagringar som utsätts för mycket svåra driftsförhållanden, och där speciella krav ställs på tätning, kan SKF leverera sfäriska rullager försedda med frikterande gummitätningar på båda sidor. Tätningarna består av två brickor, en är fästad i ytterringen och en i innerringen och båda är korrosionsskyddade. Innerringens tätningsbricka har en vulkaniserad tätningsläpp av fluor-gummi som effektivt skyddar lagret mot inträngande föroreningar och när innerringen roterar fungerar denna tätningsbricka dessutom som avkastare. Tätningsbrickorna skjuter visserligen ut från lagrets sidplan så att det behövs ett något större axiellt utrymme. Ändå kan standardlager i många fall ersättas med SKF tätade sfäriska rullager i befintliga lagerinbyggnader.

SKF tätade sfäriska rullager tillåter snedställningar av innerringen i förhållande till ytterringen på upp till 0,5° i inbyggnader där innerringen roterar. Tätningsläppens material begränsar driftstemperaturen för dessa lager till området −30 till +150 °C. Lagren är fyllda med en lämplig mängd av ett korrosionsskyddande, litiumbaserat fett avsett för temperaturområdet −30 till +110 °C. På begäran kan lagren levereras med andra fetter.

I många fall behöver inte tätade sfäriska rullager någon eftersmörjning. När lagren skall arbeta under stora belastningar, vid höga varvtal eller vid temperaturer över +70 °C, behöver de däremot eftersmörjas, vilket sker via smörjspåret och smörjhålen i ytterringen.

Ytterligare uppgifter om tätade sfäriska rullager lämnas på begäran.

Sfäriskt rullager med tätningar



Sfäriska rullager för vibrerande inbyggnader

För skaksiktar och andra vibrerande inbyggnader har SKF konstruerat speciella lager. Dessa lager har samma tabelldata som lagren ur serie 223 CC(K). De tillverkas i två olika utföranden beroende på storlek och skiljer sig från det ursprungliga CC-utförandet genom att de är försedda med ythärdade pressade stålhållare med extra stor slitstyrka. De större lagren med håldiametern d ≧ 75 mm har en styrring som centreras mot ytterringens löpbana medan de mindre lagrens styrring centreras mot innerringen. Vidare är radialglappet avpassat för de speciella driftsförhållandena och ligger inom området från övre hälften av C3 till nedre hälften av C4.

SKF sfäriska rullager för vibrerande inbyggnader levereras med cylindriskt eller med koniskt hål för axeldiametrar från 40 t.o.m. 200 mm. De mindre lagren med håldiameter t.o.m 70 mm har tillägget A15 i efterbeteckningen, t.ex. 22314 CC/W33A15, medan de större lagren har JA och VA405 i efterbeteckningen, t.ex. 22320 CCJA/W33VA405.

För att undvika passningsrost kan lager med cylindriskt hål och håldiametern 75 mm eller större levereras med en ytbeläggning av PTFE i hålet. Dessa lager har samma mått som övriga lager för vibrerande inbyggnader, men håldiameterns toleranser skiljer sig från standard. Lagren har tillägget VA406 i efterbeteckningen, t.ex. 22324 CCJA/W33VA406.

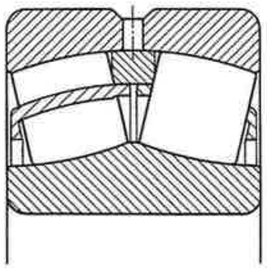
Närmare upplysningar om sfäriska rullager för vibrerande inbyggnader finns i särskilda trycksaker som lämnas på begäran.

Andra sfäriska rullager

SKF tillverkar även andra storlekar och serier av sfäriska rullager än de som visas i följande tabeller. Närmare upplysningar om dessa finns i SKF-katalogen "Large bearings".

Sfäriska rullager för vibrerande inbyggnader

Lager med håldiameter d < 75 mm



Lager med håldiameter d ≥ 75 mm

Mått

Huvudmåtten på de sfäriska rullager som visas i tabellerna överensstämmer med ISO 15-1981.

Snedställning

Ett sfäriskt rullager är självinställande, dvs. det medger snedställning av axeln relativt lagerhuset. Under normal belastning, normala driftsförhållanden och vid roterande innerring kan man tillåta snedställning enligt de riktvärden som anges i vidstående tabell. Om dessa värden kan utnyttjas fullt ut eller inte beror på lagringens konstruktion, typ av tätningar osv.

Toleranser

SKF sfäriska rullager med cylindriskt respektive koniskt hål levereras som standard med normaltoleranser. Värden för dessa toleranser återfinns i tabellen på sidan 74.

Lagerglapp

SKF sfäriska rullager levereras som standard med normalglapp. Nästan alla lager kan även fås med C3-glapp vilket är större än det normala. Vissa kan levereras med det ännu större C4-glappet eller med C2-glapp som är mindre än normalglapp. Leveransmöjligheterna för lager med annat glapp än normalt (även C5-glapp) bör kontrolleras före beställning. Värdena för de olika glappen anges i tabellerna på sidorna 464 och 465 och överensstämmer med ISO 5753-1981 för d ≦ 1 000 mm. De gäller för lager utan mätbelastning och före montering.

Lager	Tillåten snedställning
-	grader
Serie 213	1
Serie 222	1,5
Serie 223	2
Serie 230	1,5
Serie 231	1,5
Serie 232	2,5
Serie 239	1,5
Serie 240	2
Serie 241	2,5

Radialglapp i sfäriska rullager med cylindriskt hål

Håldiameter d över	t.o.m.	Radialglapp C2 min	max	Normalt min	max	C3 min	max	C4 min	max	C5 min	max
mm		µm									
18	**24**	10	20	20	35	35	45	45	60	60	75
24	**30**	15	25	25	40	40	55	55	75	75	95
30	**40**	15	30	30	45	45	60	60	80	80	100
40	**50**	20	35	35	55	55	75	75	100	100	125
50	**65**	20	40	40	65	65	90	90	120	120	150
65	**80**	30	50	50	80	80	110	110	145	145	180
80	**100**	35	60	60	100	100	135	135	180	180	225
100	**120**	40	75	75	120	120	160	160	210	210	260
120	**140**	50	95	95	145	145	190	190	240	240	300
140	**160**	60	110	110	170	170	220	220	280	280	350
160	**180**	65	120	120	180	180	240	240	310	310	390
180	**200**	70	130	130	200	200	260	260	340	340	430
200	**225**	80	140	140	220	220	290	290	380	380	470
225	**250**	90	150	150	240	240	320	320	420	420	520
250	**280**	100	170	170	260	260	350	350	460	460	570
280	**315**	110	190	190	280	280	370	370	500	500	630
315	**355**	120	200	200	310	310	410	410	550	550	690
355	**400**	130	220	220	340	340	450	450	600	600	750
400	**450**	140	240	240	370	370	500	500	660	660	820
450	**500**	140	260	260	410	410	550	550	720	720	900
500	**560**	150	280	280	440	440	600	600	780	780	1 000
560	**630**	170	310	310	480	480	650	650	850	850	1 100
630	**710**	190	350	350	530	530	700	700	920	920	1 190
710	**800**	210	390	390	580	580	770	770	1 010	1 010	1 300
800	**900**	230	430	430	650	650	860	860	1 120	1 120	1 440
900	**1 000**	260	480	480	710	710	930	930	1 220	1 220	1 570
1 000	**1 120**	290	530	530	780	780	1 020	1 020	1 330	1 330	1 720
1 120	**1 250**	320	580	580	860	860	1 120	1 120	1 460	1 460	1 870

Radialglapp i sfäriska rullager med koniskt hål

Håldiameter d över	t.o.m.	Radialglapp C2 min	max	Normalt min	max	C3 min	max	C4 min	max	C5 min	max
mm		µm									
24	**30**	20	30	30	40	40	55	55	75	–	–
30	**40**	25	35	35	50	50	65	65	85	85	105
40	**50**	30	45	45	60	60	80	80	100	100	130
50	**65**	40	55	55	75	75	95	95	120	120	160
65	**80**	50	70	70	95	95	120	120	150	150	200
80	**100**	55	80	80	110	110	140	140	180	180	230
100	**120**	65	100	100	135	135	170	170	220	220	280
120	**140**	80	120	120	160	160	200	200	260	260	330
140	**160**	90	130	130	180	180	230	230	300	300	380
160	**180**	100	140	140	200	200	260	260	340	340	430
180	**200**	110	160	160	220	220	290	290	370	370	470
200	**225**	120	180	180	250	250	320	320	410	410	520
225	**250**	140	200	200	270	270	350	350	450	450	570
250	**280**	150	220	220	300	300	390	390	490	490	620
280	**315**	170	240	240	330	330	430	430	540	540	680
315	**355**	190	270	270	360	360	470	470	590	590	740
355	**400**	210	300	300	400	400	520	520	650	650	820
400	**450**	230	330	330	440	440	570	570	720	720	910
450	**500**	260	370	370	490	490	630	630	790	790	1 000
500	**560**	290	410	410	540	540	680	680	870	870	1 100
560	**630**	320	460	460	600	600	760	760	980	980	1 230
630	**710**	350	510	510	670	670	850	850	1 090	1 090	1 360
710	**800**	390	570	570	750	750	960	960	1 220	1 220	1 500
800	**900**	440	640	640	840	840	1 070	1 070	1 370	1 370	1 690
900	**1 000**	490	710	710	930	930	1 190	1 190	1 520	1 520	1 860
1 000	**1 120**	530	770	770	1 030	1 030	1 300	1 300	1 670	1 670	2 050
1 120	**1 250**	570	830	830	1 120	1 120	1 420	1 420	1 830	1 830	2 250

Driftstemperaturens inverkan på lagermaterialet
SKF sfäriska rullager är som standard speciellt värmebehandlade för att kunna arbeta vid driftstemperaturer upp till +200 °C utan att några otillåtna måttförändringar uppstår; se även avsnittet "Material för rullningslager", sidan 89.

Massa
Den massa som anges i lagertabellerna avser lager med cylindriskt hål; massan hos lager med koniskt hål anges i tabellerna för lager med kläm- och avdragshylsa.

Passande lagerhus
Passande lagerhus för sfäriska rullager med cylindriskt hål och för lager med klämhylsa återfinns i avsnittet "Lagerhus". Det finns även andra lämpliga lagerhus, t.ex. stålagerhus för oljesmörjning, serie SOFN, lagerhus för bandtransportörer, lagerhus för stora lager osv. Information lämnas på begäran.

Driftstemperatur och axialbelastning
SKF sfäriska rullager har, tack vare sin speciella inre konstruktion, inte bara lägre friktion än andra sfäriska rullager, utan kan också överföra betydligt större axialbelastningar, även rent axiella belastningar. Om $F_a/F_r > e$ (se lagertabellerna) rekommenderas emellertid kortare eftersmörjningsintervall än som anges i diagrammet på sidan 155.

Driftstemperaturen kan vara en begränsande faktor för förmågan att överföra axialbelastningar. På begäran kan SKF, med en speciell datorberäkning, kontrollera om ett visst sfäriskt rullager är lämpligt för givna driftsvillkor i en bestämd lagerinbyggnad.

Axiell bärförmåga hos lager med klämhylsa
Den axiella bärförmågan hos ett sfäriskt rullager monterat på klämhylsa på slät axel bestäms av friktionen mellan hylsan och axeln. Om lagret är korrekt monterat kan den tillåtna axialbelastningen beräknas med formeln

$$F_{ap} = 3\,B\,d$$

där
F_{ap} = största tillåtna axialbelastning, N
B = lagrets bredd, mm
d = lagrets håldiameter, mm

Minsta belastning
Kul- och rullager måste alltid ha en given minsta belastning för att de skall fungera tillfredsställande. Detta gäller även för sfäriska rullager, i synnerhet om de arbetar vid höga varvtal där tröghetskrafterna hos rullar och hållare samt friktionen i smörjmedlet kan ha en menlig inverkan på rullningsförhållandena i lagret och ge upphov till skadliga glidrörelser mellan rullar och löpbanor.

Den erforderliga minsta radialbelastningen kan beräknas ur formeln

$$F_{rm} = 0{,}02\,C$$

där
F_{rm} = minsta radialbelastning, N
C = dynamiskt bärighetstal, N

Summan av egentyngden hos de lagrade delarna, tillsammans med de yttre krafterna, överstiger ofta den erforderliga minsta belastningen. Om så inte är fallet måste lagret belastas ytterligare, t.ex. genom ökad remspänning eller ökat tomgångsmoment.

Ekvivalent dynamisk lagerbelastning

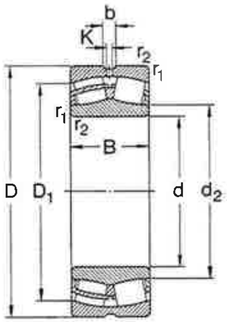
$$P = F_r + Y_1 F_a \qquad \text{om } F_a/F_r \leq e$$
$$P = 0{,}67\,F_r + Y_2 F_a \qquad \text{om } F_a/F_r > e$$

Värdena på faktorerna e, Y_1 och Y_2 för varje enskilt lager är angivna i lagertabellerna.

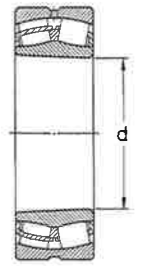
Ekvivalent statisk lagerbelastning

$$P_0 = F_r + Y_0 F_a$$

Värdet på faktorn Y_0 för varje enskilt lager anges i lagertabellerna.



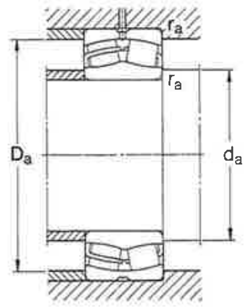
Cylindriskt hål



Koniskt hål

Huvudmått d	D	B	Bärighetstal dyn. C	stat. C_0	Utmattningsbelastning P_u	Basvarvtal Smörjning fett	olja	Massa	Beteckningar Lager med cylindriskt hål	koniskt hål
mm			N		N	r/min		kg	–	
60	110	28	122 000	146 000	16 300	4 000	5 000	1,10	**22212 CC**	**22212 CCK**
	110	28	140 000	173 000	19 000	4 300	5 300	1,15	**22212 E**	**22212 EK**
	130	31	161 000	200 000	23 200	3 000	3 800	1,95	**21312 CC**	**21312 CCK**
	130	46	235 000	280 000	30 000	3 000	3 800	2,95	**22312 CC**	**22312 CCK**
	130	46	271 000	335 000	36 500	2 800	3 600	2,90	**22312 E**	**22312 EK**
65	120	31	148 000	183 000	21 200	3 800	4 800	1,45	**22213 CC**	**22213 CCK**
	120	31	176 000	216 000	24 000	3 800	4 800	1,50	**22213 E**	**22213 EK**
	140	33	184 000	240 000	27 000	2 800	3 600	2,45	**21313 CC**	**21313 CCK**
	140	48	253 000	300 000	32 000	2 600	3 400	3,55	**22313 CC**	**22313 CCK**
	140	48	299 000	360 000	38 000	2 600	3 400	3,55	**22313 E**	**22313 EK**
70	125	31	148 000	186 000	21 200	3 600	4 500	1,55	**22214 CC**	**22214 CCK**
	125	31	179 000	228 000	25 500	3 600	4 500	1,55	**22214 E**	**22214 EK**
	150	35	207 000	260 000	29 000	2 600	3 400	3,00	**21314 CC**	**21314 CCK**
	150	51	311 000	380 000	40 000	2 400	3 200	4,30	**22314 CC/W33**	**22314 CCK/W33**
	150	51	345 000	430 000	45 000	2 200	3 000	4,30	**22314 E**	**22314 EK**
75	130	31	158 000	208 000	23 600	3 400	4 300	1,65	**22215 CC**	**22215 CCK**
	130	31	184 000	240 000	26 500	3 400	4 300	1,70	**22215 E**	**22215 EK**
	160	37	235 000	300 000	32 500	2 400	3 200	3,55	**21315 CC**	**21315 CCK**
	160	55	345 000	430 000	44 000	2 200	3 000	5,25	**22315 CC/W33**	**22315 CCK/W33**
	160	55	385 000	475 000	48 000	2 200	3 000	5,25	**22315 E**	**22315 EK**
80	140	33	176 000	228 000	26 000	3 200	4 000	2,05	**22216 CC**	**22216 CCK**
	140	33	207 000	270 000	29 000	3 200	4 000	2,10	**22216 E**	**22216 EK**
	170	39	258 000	335 000	36 000	2 200	3 000	4,20	**21316 CC**	**21316 CCK**
	170	58	374 000	455 000	46 500	2 000	2 800	6,20	**22316 CC/W33**	**22316 CCK/W33**
	170	58	431 000	540 000	54 000	2 000	2 800	6,20	**22316 E**	**22316 EK**
85	150	36	210 000	270 000	31 000	3 000	3 800	2,55	**22217 CC/W33**	**22217 CCK/W33**
	150	36	244 000	325 000	34 500	2 800	3 600	2,65	**22217 E**	**22217 EK**
	180	41	293 000	375 000	40 000	2 000	2 800	5,00	**21317 CC**	**21317 CCK**
	180	60	420 000	520 000	52 000	1 900	2 600	7,25	**22317 CC/W33**	**22317 CCK/W33**
	180	60	477 000	620 000	61 000	1 900	2 600	7,25	**22317 E**	**22317 EK**

Leveransmöjligheterna för lager i E(K)-utförande bör kontrolleras före beställning.



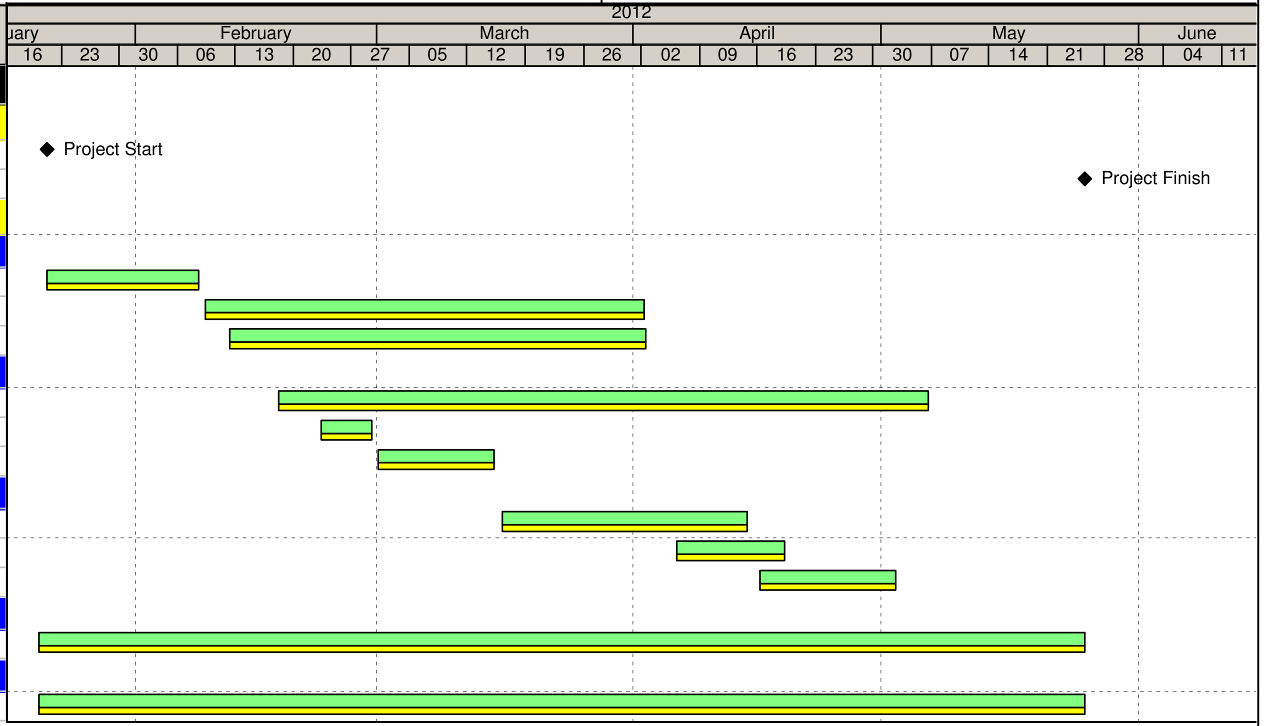
Mått d	d_2 ≈	D_1 ≈	$r_{1,2}$ min	b	K	Inbyggnadsmått d_a min	D_a max	r_a max	Beräkningsfaktorer e	Y_1	Y_2	Y_0
mm						mm			–			
60	72,7	96,6	1,5	–	–	69	101	1,5	0,24	2,8	4,2	2,8
	71,6	98	1,5	5,5	3	69	101	1,5	0,24	2,8	4,2	2,8
	79,7	113	2,1	–	–	72	118	2	0,24	2,8	4,2	2,8
	74,9	109	2,1	–	–	72	118	2	0,35	1,9	2,9	1,8
	77,9	112	2,1	5,5	3	72	118	2	0,35	1,9	2,9	1,8
65	79,4	106	1,5	–	–	74	111	1,5	0,24	2,8	4,2	2,8
	77,5	107	1,5	5,5	3	74	111	1,5	0,25	2,7	4	2,5
	86	119	2,1	–	–	77	128	2	0,24	2,8	4,2	2,8
	82	118	2,1	–	–	77	128	2	0,35	1,9	2,9	1,8
	81,7	120	2,1	8,3	4,5	77	128	2	0,35	1,9	2,9	1,8
70	84,6	111	1,5	–	–	79	116	1,5	0,23	2,9	4,4	2,8
	83	112	1,5	5,5	3	79	116	1,5	0,23	2,9	4,4	2,8
	92,6	127	2,1	–	–	82	138	2	0,24	2,8	4,2	2,8
	88	127	2,1	8,3	4,5	82	138	2	0,35	1,9	2,9	1,8
	90,3	130	2,1	8,3	4,5	82	138	2	0,33	2	3	2
75	89,7	116	1,5	–	–	84	121	1,5	0,22	3	4,6	2,8
	87,8	117	1,5	5,5	3	84	121	1,5	0,22	3	4,6	2,8
	99,1	135	2,1	–	–	87	148	2	0,23	2,9	4,4	2,8
	94,2	134	2,1	8,3	4,5	87	148	2	0,35	1,9	2,9	1,8
	92,7	136	2,1	8,3	4,5	87	148	2	0,35	1,9	2,9	1,8
80	95,1	124	2	–	–	90	130	2	0,22	3	4,6	2,8
	94,2	127	2	5,5	3	90	130	2	0,22	3	4,6	2,8
	105	145	2,1	–	–	92	158	2	0,23	2,9	4,4	2,8
	100	144	2,1	8,3	4,5	92	158	2	0,35	1,9	2,9	1,8
	98,2	144	2,1	8,3	4,5	92	158	2	0,35	1,9	2,9	1,8
85	100	132	2	5,5	3	95	140	2	0,22	3	4,6	2,8
	101	135	2	5,5	3	95	140	2	0,22	3	4,6	2,8
	111	153	3	–	–	99	166	2,5	0,23	2,9	4,4	2,8
	106	154	3	8,3	4,5	99	166	2,5	0,33	2	3	2
	108	155	3	8,3	4,5	99	166	2,5	0,33	2	3	2

Appendix J

Appendix J – Gantt Chart

Project Spinner Wrench

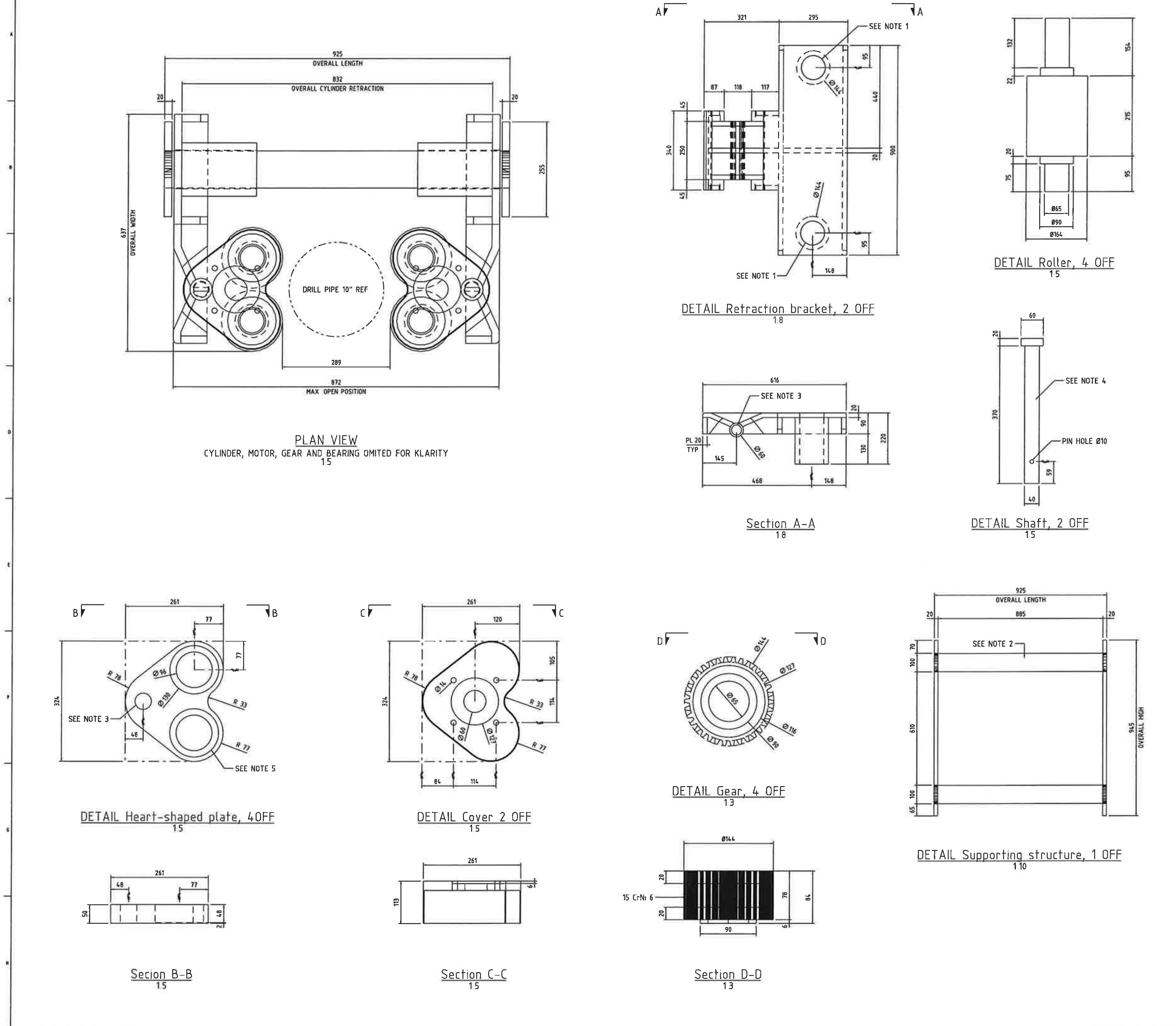
Activity ID	Activity Name	Start	Finish
Project Spinner Wrench-Sundal		20-01-12	25-05-12
MILESTONES		21-01-12	25-05-12
M1	Project Start	21-01-12	
M2	Project Finish		25-05-12
Project Plan		20-01-12	25-05-12
Spinner Wrench		21-01-12	02-04-12
SW10	Deciding Concept	21-01-12*	08-02-12
SW20	Modelling	09-02-12*	02-04-12
SW30	Calculation	12-02-12*	02-04-12
Hydraulic System		18-02-12	06-05-12
HYD10	Hydraulic system Components	18-02-12*	06-05-12
HYD20	Static Analysis	23-02-12*	29-02-12
HYD30	Dynamic Analysis	01-03-12*	15-03-12
Control System		16-03-12	02-05-12
CS10	Control System	16-03-12*	14-04-12
CS20	Building a Prototype	06-04-12*	19-04-12
CS30	Solid Works Simulation FEM Analysis	16-04-12*	02-05-12
Documentation		20-01-12	25-05-12
DOC10	Documentation	20-01-12*	25-05-12
Report		20-01-12	25-05-12
REP10	Report	20-01-12*	25-05-12

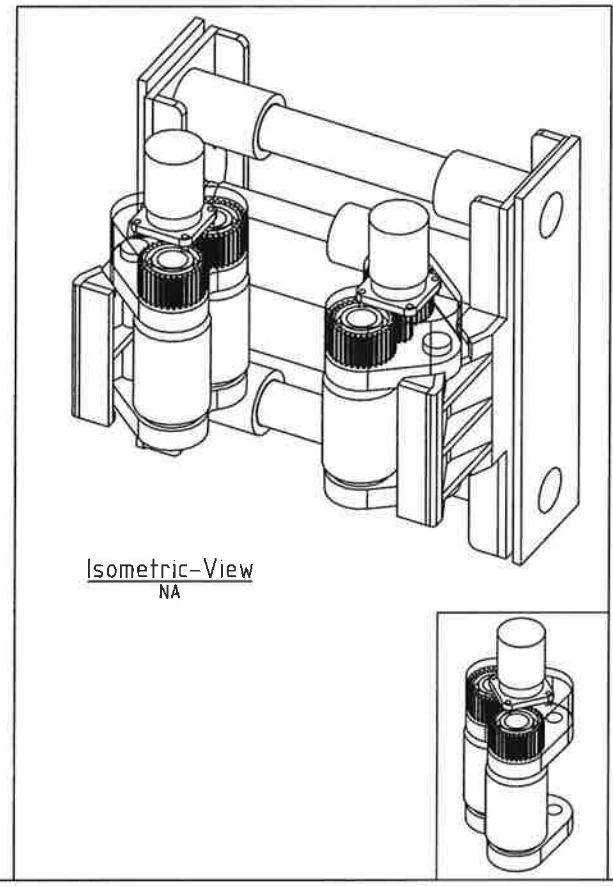


Project Baseline Bar
Critical Remaining W...
Actual Work
Milestone
Remaining Work

Appendix K

Appendix K – DRAWING





NOTES

1- HOLE TOLERANCE 104F6 THE MAXIMUM HOLE DIAMETER IS 104 058 AND THE MINIMUM HOLE DIAMETER IS 104 036 [14]

2- SHAFT TOLERANCE 100f6 THE MAXIMUM HOLE DIAMETER IS 99 946 AND THE MINIMUM HOLE DIAMETER IS 99 942 [14]

3- HOLE TOLERANCE 44F6 THE MAXIMUM HOLE DIAMETER IS 44 041 AND THE MINIMUM HOLE DIAMETER IS 44 025 [14]

4- SHAFT TOLERANCE 44F6 THE MAXIMUM HOLE DIAMETER IS 39 975 AND THE MINIMUM HOLE DIAMETER IS 39 959 [14]

5- BEARING MUST HAVE A PRESS FIT TO THE ROLLER SHAFT IN ORDER TO ELIMINATE WEAR

6- TOOTH WHEEL MUST HAVE A PRESS FIT TO THE ROLLER SHAFT IN ORDER TO ELIMINATE WEAR

7- ALL WELDS THIS DRAWING 6mm FILET OR 6mm PART PEN U N O

8- WELD GRINDED FLUSH BETWEEN Heart-shaped plate AND Cover plate

MATERIAL FOR THIS DRAWING U N O

DESIGN CATEGORY	SECONDERY
MATERIAL QUALITY	NVE 36
INSPECTION CATEGORY	II

WIGHT INFORMATION	WIGHT KG
	870

Rev	Date	Description	Made By	Checked	Approved
0	03 05 12	ISSUED FOR CONSTRUCTION	LFF	-	KjR

UNIVERSITETET I AGDER — NOV NATIONAL OILWELL VARCO

MAS 500 MASTER THESIS	Drawing Title EQUIPMENT SPINNER WRENCH	Scale 1 VAR Format A1
Project Spinner Wrench	Drawing No MANIPULATOR ARM - 1112 - N - 1000 - 01	Rev 0