High Performance Coating For Limited Slip Differentials

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Preface

The work presented in this master thesis has been carried out at the Division of Machine Elements at Luleå University of Technology as a TRIBOS master student. I would like to express my gratitude to Dr. Pär Marklund and Dr. Kim Berglund for their discussions, support and guidance throughout the project. Acknowledgements should also be made to Quay Surface Engineering (QSE) Metalblast Ltd for their financial and material support.
Abstract

Today’s vehicles are consistent of many different mechanical components. One of these components is the clutch. In this work focus will be on the clutch for limited slip differentials, which are commonly used in recent years to overcome traction problems to increase the overall traction and handling when driving. This increased traction is crucial for racing applications.

One of the material used in clutch for limited slip differential is a molybdenum coating applied on steel-based disc. Behavior of this material for this typical application is still unknown, although many investigations with a similar material have been done in the past decades. Therefore investigation of molybdenum based friction material for clutch applications has been conducted. The focus of this master thesis is to show the friction behavior of a wet clutch working under similar operating conditions as in racing applications.

Results show high sensitivity on initial surface roughness of the molybdenum coating. In case of non-surface finished coating, friction performance shows negative slope of the friction coefficient with increasing sliding velocity, resulting in excessive wear, leading to system failure. However in case of surface finished coating, friction performance shows a stable friction coefficient with increasing sliding velocity, which results in stable system.

Moreover, two different contact pressures have been tested in order to investigate pressure dependence on friction coefficient. However it has not been found out any direct effect from pressure on coefficient of friction. In addition two different discs sizes have been tested in order to obtain higher pressure with the same contact load, which resulted in difficult comparison, as the system has slightly changed.

Comparison of the same material used with a different lubricant is presented. In addition comparison between molybdenum-based material and paper-based material is presented used with the same lubricant.

More focus should be given on how system changes, by changing surface roughness of molybdenum and steel plate and how this behavior effect the overall behavior of wet clutch.
**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ra</td>
<td>Average surface roughness</td>
<td>[µm]</td>
</tr>
<tr>
<td>µ</td>
<td>Coefficient of friction</td>
<td>[-]</td>
</tr>
<tr>
<td>v</td>
<td>Sliding velocity</td>
<td>[m/s]</td>
</tr>
<tr>
<td>n</td>
<td>Number of revolutions</td>
<td>[rpm]</td>
</tr>
<tr>
<td>M</td>
<td>Torque</td>
<td>[Nm]</td>
</tr>
<tr>
<td>n_i</td>
<td>Number of sliding interfaces</td>
<td>[-]</td>
</tr>
<tr>
<td>F_n</td>
<td>Normal load</td>
<td>[N]</td>
</tr>
<tr>
<td>p</td>
<td>Nominal contact pressure</td>
<td>[Pa]</td>
</tr>
<tr>
<td>R_m</td>
<td>Mean radius</td>
<td>[m]</td>
</tr>
<tr>
<td>R_i</td>
<td>Inner radius</td>
<td>[m]</td>
</tr>
<tr>
<td>R_o</td>
<td>Outer radius</td>
<td>[m]</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>[°C]</td>
</tr>
<tr>
<td>A</td>
<td>Surface area</td>
<td>[m²]</td>
</tr>
</tbody>
</table>
1. Introduction
A clutch is a mechanical device which transmits torque and rotational speed from the driving shaft to the driven shaft by means of friction. The main advantage of the clutch is to allow different rotational speed from input to output shaft, while transferring torque. A clutch can be of either dry or wet type. The main difference is that the dry clutch operates on air and the wet clutch is submerged into a lubricant. Due to the presence of lubricant the wet clutch is continuously cooled down, which enables longer engagement time. Moreover, lubricant fluid keeps clutch interfaces free of contaminants and provides a smoother performance, which extends service life. Wet clutch normally consists of a clutch pack see Fig. 1, where separator disc and friction disc are alternately positioned. Separator discs are connected to the input shaft and friction discs to the output shaft or vice versa. Both separator and friction discs are connected to the shaft by spline geometry. By applying axial force to the clutch pack, friction force is generated between separator and friction discs, which makes it possibility to transfer torque.

There exists many applications where wet clutches are used. For instance, wet clutches are frequently used in Automatic Transmissions (AT) to shift gears. Clutches are also used in Continuously Variable Transmissions (CVT), which allow “neutral” state. Further on control of torque transfer presents another function of clutch. This can be found in applications such as wet breaks, lock-up clutches, lanch clutches in motor cycles and Limited Slip Differentials (LSD).
Limited Slip Differentials (LSDs) see Fig. 2, are used to overcome traction problem with the drive wheels. Wet clutches in this applications can brake or lock the differential to give more traction.

There exists several types of limited slip differentials, such as fixed value, torque sensitive, speed sensitive or electronically controlled. As the name indicates, engagement force on the clutch pack is either a fixed value or a function of applied torque or speed or freely controlled. This type of clutch system is being used in All-Wheel drive cars to improve traction and handling by providing controllable torque split between front and rear axles.

Working conditions of wet clutches are different for different applications. For instance in case of an AT, the sliding velocity can be high when engaging the clutch and between the engagement, clutch discs are separated by a lubricant film. When the clutch becomes fully engaged, sliding is no longer possible. In case of limited slip differentials, working conditions are different. Here sliding velocity is lower and surfaces of the clutch discs are not separated by a lubricant film, but are most of the time sliding against each other. Moreover, contact load in case of LSD is much higher than in AT application [1].
1.1 Friction characteristic

Performance of the clutch system is highly depended on the friction behavior of the clutch. There have been done many investigation of the friction characteristic and performance of the wet clutch [2,3]. Simple and inexpensive pin-on-disc method was design by Marklund et al. [4] to evaluate the clutch system. Moreover, Limited Slip Clutch Test-rig was developed, which was inspired from already developed test rig, designed by Maki [1] for further investigation, which presents full scale limited slip clutch system. This test-rig will be used to fully investigate the clutch system in this thesis. However, the friction behavior should be explain in details.

The clutch in a limited slip differential normally operates in the boundary regime, where asperity to asperity contact between two plates dominates. Therefore, chemical and physical interactions of the adsorbed thin film and surface topography play a significant roll. For example, three different friction behavior can be seen in Fig. 3. for different oils, where friction is the function of the sliding velocity. Both oils B and C see Fig. 3. have negative slope, which are prone to vibrations. However, oil A shows positive slope, where vibrations are suppressed.

![Figure 3 Schematic friction vs. velocity curves for 3 different oils](image)

Most common negative effects that can occur are so called stick slip and clutch shudder, also known by the name of judder. These two phenomena can occur, either of incorrect system design or due to clutch ageing [5].

Stick slip is described as unsteady intermittent sliding and sticking at the contact interface. This phenomena is “induced by discontinuous friction coefficient change in transition from static to dynamic” [6]. It shows dependence on system dynamics, friction characteristics, and external tangential and normal forcing [7]. While shudder is so called friction self-induced vibration, which is attributed to negative friction gradient known to reduce with system damping [6,8,9].
**1.2 Lubrication theory**

A wet clutch is designed to work in a wet environment, therefore lubrication properties play a crucial role when it comes to clutch performance. So the understanding of the fundamental properties of the lubricant is necessary. Although there exists variety of lubricant mixtures, all of them are normally consisted of base oil and chemical additives. Base oil provides the bulk properties of the oil, such as viscosity, while additives enhance performance and inhibit degradation of the oil.

Base oil can be divided into three types: vegetable oil, mineral oil and synthetic oil. Vegetable oils are known for their high biodegradability, but they usually age rapidly which is not beneficial, especially in applications where working conditions are severe. In case of mineral oils, in general they exhibit better performance compared to vegetable oils, however synthetic oils are usually predominant, when it comes to oil performance.

Base oil itself in general does not have sufficient properties to work as a lubricant. Therefore additives are added into base oil to improve the overall performance of the oil. There are different types of additives and each of them has their own purpose.

- **Friction modifiers (FM)** – Additives used to modify friction characteristics. Friction modifiers adsorb on the surface of a material, where they form a layer, which is easy to shear. Consequently friction and wear can be reduced. However, due to their physical bounding, they are used for lower temperatures and stresses.

- **Anti-wear additives (AW)** – Additives used to protect the metal surface, by forming a protective layer on the surface, which can be easily shearable. They can either physically adsorb onto the surface or can chemically react (mild reaction) with the surface. Normally used for higher temperatures and higher contact stresses. Apart from reducing wear they can also reduce friction, but in some cases friction can increase.

- **Extreme pressure additives (EP)** – These additives are also used to protect the metal surface. However, in this case additives chemically react (strong reaction) with the surface, where they form easily shearable layer, known as sacrificial layer. The purpose of this type of additive is to prevent seizure, scoring and welding. EP additives are used for elevated temperatures and stresses, where AW additives fail to work.

- **Anti-Oxidants** – Additives used to prolong lubricant service life. The function of this additive is to slow down the oxidation process, by acting as a radical traps to interrupt the oxidation chain reaction.

- **Corrosion Inhibitors** – Additives used to protect metal surface against acidic contaminants in the lubricant. Rust inhibitors are a special type of corrosion inhibitors, which are designed to protect ferrous materials. However, corrosion inhibitors form a tight passive film on the metal surface.
• **Viscosity-Index Improvers (VI)** – Additives used to improve viscosity-temperature relationship. These additives are high molecular weight polymers, which thicken the oil at high temperatures.

• **Detergent/Dispersant Additives** – Additives used to prevent deposits from forming on metal surface. Detergents act as surfactants that adsorb around surfaces of insoluble material formed by oil degradation at high temperatures. While dispersants surround solid and liquid particles e.g. oxidized oil-insoluble products, water, fuel or dust at relatively low temperatures.

• **Anti-Foam Agents** – Additives used to prevent foaming. They slow down the release of gas in the lubricant, by reducing surface tension

• **Pour point depressants** – Additives used to lower the pour point of an oil.
1.3 Wet clutch applications

Lubricants used in wet clutch applications has normally very complex structure, where additives are very important for the friction performance in boundary lubrication. Apart from that it needs to be compatible with other mechanical components e.g. different gears, bearings and seals. This makes lubricant formulation even harder to fulfill all demands that some system has. Lubricant in wet clutch applications also are expected to have high shear stability, due to constant shearing stress in the contact [10]. However, there exists different types of applications with different standards of wet clutch lubricants.

Automatic Transmission Fluid (ATF) – It is known to be one of the most complex structured fluid in the automotive industry. This is due to many different tasks that lubricant has in a gearbox. Lubricant not only has high shear and oxidation stability, but it also needs to reduce friction, where mechanical parts are moving, as well as it needs to reduce drag torque on the clutch.[10]

Motorcycle lubricant – Lubricant in most motorcycles has to work as lubricant for many components e.g. the engine, the transmission and the wet clutch. This forces the lubricant to have special properties.

Fluid for Limited Slip Differentials (LSDs) – As for all applications, the working conditions govern the fluid formulation properties. In case of wet clutches in LSDs, fluid is exposed under high surface pressure and low sliding speeds. Normally the fluid is not only designed to work with wet clutch, but also with differential gears e.g. hypoid gears, where normally the use of a transmission fluid of API (American Petroleum Institute) grade GL 5 is required [10].
1.4 Materials Theory
Material combinations in wet clutch applications has a crucial role apart from lubricant when it comes to the friction performance. The right combination of materials and lubricant may give the clutch optimal friction performance. However, the choice of the material and lubricant selection is related to the working conditions as well as the compatibility. Normally a wet clutch has two different types of discs, where one disc type is the friction disc and the other one serves as a separator disc. Both of them are normally made of harden carbon steel. However friction disc normally contains additional material, which acts as a friction material. This material is attached onto the steel surface.

Wet clutch usually consists of a clutch pack. There can be either single-sided friction discs or double-sided friction disc. In case of single-sided friction disc, there is no need to have separator discs, since the friction discs contain both separator disc on one side and friction material on another side. However, thermal problems can occur with this configuration, due to differences in heat conductivity and heat expansion of the different materials. This can lead to coning of the friction disc [10]. Therefore double-sided friction discs are preferable in wet clutch applications. In some cases friction material is not required, so only steel discs are used. For example in Limited Slip Differentials this configuration is used when low power density is required. However, high power density application demands additional friction materials.

There exists numerous kinds of friction lining material on today’s market for wet clutch technology. Paper based material is one of the most commonly used low-cost friction materials, especially in Automatic transmission application for vehicles. Normally it is composed of binder, filler or solid lubricant and reinforcing materials. For more details see [5,10,11]. However, a drawback for this material is that it offers good performance under relatively low load conditions. For tougher working conditions, like in LSD applications, sintered bronze is often used as a friction material. Good thermal conductivity apart from ability to withstanding high temperatures makes this material superior to paper based materials [1]. Apart from paper based and sinter bronze friction materials, there are also carbon based, Kevlar and hybrid materials, which can be used in wet clutch technology. For more information about these materials see [1,5,10,11]. All these friction materials have normally relatively thick friction lining attached on the surface of a core disc made of steel. However, apart from friction lining (relatively thick film), coating (relatively thin film) can be applied on the surface and act as a friction material.

Molybdenum coating technology can be applied as a friction material especially in LCD wet clutch applications. This coating is broadly used in automotive applications. It is known to be one of the most wear and heat resistant coating [12]. Moreover, Flame-sprayed molybdenum wire coating is known to be widely used, especially due to its high scuff resistance. The bond between steel-based material and molybdenum is metallurgical since steel material is molten by high-temperature particles. Therefore strong bonds between both of materials are established, which gives the material unique properties. Also these coatings show excellent hardness properties, attributed to the formulation of MoO₂. These oxides increase overall coating performance by increasing the hardness and improving the wear resistance of the coating [13].
Introduction
2. Objectives

The objectives of this thesis are:

- To investigate the performance of the molybdenum based friction coatings in wet clutches working under harsh operating conditions similar to the coated discs present in racing applications e.g. formula 1, rally cars etc.
- To investigate wet clutch degradation for molybdenum based friction coatings working under harsh operating conditions.
- To compare friction performance of the molybdenum based friction coatings in combination with different lubricants.
- To compare friction performance of two different friction materials used in wet clutches; molybdenum and paper based.
3. Materials and method

This section presents the main friction materials used in this thesis, with some specifications. Also the method used for evaluating the friction characteristics is presented and explained in details.

3.1 Materials

Friction material presents the molybdenum coating applied to the surface of the steel-based plate, provided by Quay Surface Engineering (QSE) Metalblast Ltd. The geometry of the friction plates were initially reduced for the investigation under severe contact pressure due to the hydraulic limitations of the test-rig. By doing so two different geometries of plates were manufactured, see Table 1. Further on, molybdenum coating was then applied on one side of the plate by flame wire spraying process. The coated surface presents the crucial interface of the sliding against steel interface, therefore it was additionally surface finished by mechanical process. However, both surface finished and non-finished molybdenum plates have been tested.

<table>
<thead>
<tr>
<th>Diameter (Outer/Inner) [mm]</th>
<th>100/90</th>
<th>95/90</th>
<th>118/90 (Original geometry for the Limited slip clutch test-rig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of grooves []</td>
<td>6</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Total plate area [mm²]</td>
<td>1449</td>
<td>684</td>
<td>4718</td>
</tr>
<tr>
<td>Plate area without grooves [mm²]</td>
<td>1274.4</td>
<td>599.4</td>
<td>4214</td>
</tr>
<tr>
<td>Hardness, Molybdenum, HV</td>
<td>600 - 800</td>
<td>600 - 800</td>
<td>600 - 800</td>
</tr>
<tr>
<td>Hardness, Molybdenum oxides, HV</td>
<td>900 - 1600</td>
<td>900 - 1600</td>
<td>900 - 1600</td>
</tr>
<tr>
<td>Hardness, Base material – Steel, HV</td>
<td>160 - 190</td>
<td>160 - 190</td>
<td>160 - 190</td>
</tr>
<tr>
<td>Average surface roughness Ra [µm], non-surface finished</td>
<td>7.7</td>
<td>7.7</td>
<td>7.7</td>
</tr>
<tr>
<td>Average surface roughness, Ra [µm], surface finished</td>
<td>1.8</td>
<td>1.8</td>
<td>1.8</td>
</tr>
</tbody>
</table>
Figure 4 presents the surface topography obtained with 3D optical profilometer for (a) a non-surface finished molybdenum coating (Ra = 7.7 µm) and (b) a surface finished molybdenum coating (Ra = 1.8 µm). Surface finishing was obtained by lapping the discs.

![Figure 4 Surface topography of: (a) Non-surface finished molybdenum coating, (b) Surface finished molybdenum coating](image)

Commercially available standard steel plate (≈0.35-0.45% C) are used for the separator discs, see table 2. Surface was not additionally finished.

| Table 2 Separator disc – Commercially available standard steel plate - Specifications |
|-----------------------------------|---------------------------------------------|
| Diameter (Inner/Outer)           | 90/119 [mm]                                |
| Material                         | Hot rolled mild steel                      |
| Thickness [mm]                   | 1.94                                       |
| Surface roughness Ra [µm]        | 0.2                                        |
| Hardness, HV                     | 230 - 250                                  |
The lubricant used for this investigation is Castrol Syntrax Limited Slip. Table 3 shows the lubricant specifications.

**Table 3  Lubricant - Castrol Syntrax Limited Slip 75W-140 - Specifications**

<table>
<thead>
<tr>
<th>Name</th>
<th>Method</th>
<th>Units</th>
<th>Syntrax Limited Slip 75W-140</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity, Kinematic 100°C</td>
<td>ASTM D445</td>
<td>mm²/s</td>
<td>24.7</td>
</tr>
<tr>
<td>Viscosity, Brookfield @ -40°C</td>
<td>ASTM D2983</td>
<td>mPa.s (cP)</td>
<td>120000</td>
</tr>
<tr>
<td>Appearance</td>
<td>Visual</td>
<td>-</td>
<td>Bright &amp; Clear (Yellow)</td>
</tr>
<tr>
<td>Flash Point, COC</td>
<td>ISO 2592</td>
<td>°C</td>
<td>228</td>
</tr>
<tr>
<td>Pour Point</td>
<td>ISG 3016</td>
<td>°C</td>
<td>-54</td>
</tr>
<tr>
<td>Density @ 15°C</td>
<td>DIN EN ISC 12185</td>
<td>g/ml</td>
<td>0.857</td>
</tr>
<tr>
<td>Viscosity, Kinematic 40°C</td>
<td>ASTM D 445</td>
<td>mm²/s</td>
<td>175</td>
</tr>
<tr>
<td>Viscosity index Grade</td>
<td>ISO 2909</td>
<td>None</td>
<td>174</td>
</tr>
<tr>
<td>Grade</td>
<td>API</td>
<td>-</td>
<td>GL – 5 (EP additives)</td>
</tr>
</tbody>
</table>
3.2 Method
The Limited Slip Clutch Test-rig used in order to investigate friction behavior of selected materials in this work was originally inspired by Maki [1] in order to achieve high accuracy at low sliding velocity with ability of applying high normal loads. Figure 5 presents the cross section area of Clutch test-rig with all of its components.

There are three main sections: Drive shaft, clutch pack and torsion bar. The drive shaft is driven by an electrical motor, the load applied on the clutch pack is generated with a hydraulic unit and is transferred through the driven shaft to the clutch pack. The clutch pack consists of only two discs; one separator disc which is connected to the drive shaft by outer splines and the friction disc which is connected to the torsion bar by inner splines. As the name indicates, the torsion bar has the possibility to measure torque with a torque measurement cell at the end. However, this bar is stationary, which means that the Clutch acts as a wet break. Only blue shaded parts from Fig. 4 are rotating, whilst the rest of the parts are stationary.

Furthermore, the lubricant is pumped to the sliding interface by an external oil pump through the oil inlet in the clutch pack. The external oil sump, from where the oil is circulating, has the possibility to be heated by additional heating device. To control and monitor the whole system thermocouples type K are located in the oil sump and in the separator disc 0,4 mm away from the sliding interface. For a more detailed description of the Clutch test rig see [1,5].

Apart from components’ description, specifications of the Clutch test-rig is detailed in table 4.
Table 4 Clutch test-rig specifications

**INPUT**

<table>
<thead>
<tr>
<th>Sliding velocity [RPM]</th>
<th>1 – 290</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal Force [kN]</td>
<td>0 – 20 (Maximum)</td>
</tr>
<tr>
<td>Oil circulation flow rate [l/min]</td>
<td>0 – 0.5 (max 4.5 Bar)</td>
</tr>
</tbody>
</table>

**OUTPUT**

<table>
<thead>
<tr>
<th>Max Torque [Nm]</th>
<th>500</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sampling rate [samples/sec]</td>
<td>16200</td>
</tr>
<tr>
<td>Data acquisition rate [Hz]</td>
<td>3000</td>
</tr>
</tbody>
</table>

The clutch test rig is fully controlled by a National Instrument system with the help of using the LABVIEW virtual instrumentation software for processing the data and calculating the coefficient of friction,

\[ \mu = \frac{M}{(n_i F_n R_m)} \]  

Eq. 1

where, \( M \) is the output Torque [Nm], \( n_i \) is the number of interfaces [], \( F_n \) is the axial normal load [N] and \( R_m \) is the effective radius [m], calculated as the average of both \( R_i \) inner and \( R_o \) outer radius of the friction plate.
3.2.2 Operating conditions
The purpose of this thesis is to investigate friction behavior under harsh operating conditions, which are similar to the ones present in tough application e.g. Formula 1. These conditions are known and are listed in the table 5.

Table 5 Operating conditions

<table>
<thead>
<tr>
<th>Operating conditions</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sliding velocity</td>
<td>50 RPM</td>
</tr>
<tr>
<td>Contact pressure</td>
<td>20-30 MPa</td>
</tr>
<tr>
<td>Temperature</td>
<td>120°C</td>
</tr>
</tbody>
</table>

3.2.3 Testing procedure
Test cycle Fig. 6 starts with engaging the steel plate against friction plate with initial load of 1.5 kN to avoid sudden impact, before engaging it with maximum load, which is varied during this investigation. Meanwhile the electrical motor is turn on to 1 RPM (easy motor start). After the maximum load is reached, sliding speed is ramped up to 50 RPM, hold there for 2 seconds and then ramped down to zero. After that the load is dropped to zero.

The test cycle ends with a fresh supply of lubricant which is carried into the interface of the clutch plates for 2 seconds (0.03 l/min). During testing cycle, torque is continuously measured with the torque cell, see Fig 5.
Materials and method

Testing is carried out for 1000 cycles according to Fig. 6 to analyze friction performance as well as how this changes throughout the testing. In addition degradation of material is analyzed after the testing. However different initial set-ups are conducted in order to investigate how this effects on friction performance and on degradation of the friction material. The test cases used in this investigation is presented in table 6.

Table 6 Testing setup

<table>
<thead>
<tr>
<th>Friction plate*</th>
<th>Nominal contact pressure [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter (Outer/Inner) [mm]</td>
<td>6,3</td>
</tr>
<tr>
<td>100/90</td>
<td>6,3</td>
</tr>
<tr>
<td>95/90</td>
<td>6,3</td>
</tr>
</tbody>
</table>

* Surface is additionally finished (Ra = 1.8 µm)

Testing is conducted for lower and higher pressure for two different plate sizes. For each test setup a fresh lubricant (0.8 l) and new steel disc were used. The lubricant sump was heated with external heating device to additionally heat the lubricant before entering into the clutch pack. For this testing setup pressure influence and disc size effects on overall system behavior are also investigated.

Table 7 shows the correlation between RPM (Revolutions per minute) and sliding velocity Eq. 2, while the table 8 is presenting the correlation between contact load and nominal pressure Eq. 3.

Table 7 Correlations for operating parameters - velocity

<table>
<thead>
<tr>
<th>Friction plate - Diameter (Outer/Inner) [mm]</th>
<th>RPM [rev/min]</th>
<th>Sliding velocity [m/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100/90</td>
<td>50</td>
<td>0,249</td>
</tr>
<tr>
<td>95/90</td>
<td>50</td>
<td>0,242</td>
</tr>
</tbody>
</table>

Table 8 Correlations for operating parameters - pressure

<table>
<thead>
<tr>
<th>Friction plate - Diameter (Outer/Inner) [mm]</th>
<th>Contact load [kN]</th>
<th>Nominal contact pressure [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>100/90</td>
<td>8</td>
<td>6,3</td>
</tr>
<tr>
<td>100/90</td>
<td>16</td>
<td>12,6</td>
</tr>
<tr>
<td>95/90</td>
<td>3,7</td>
<td>6,3</td>
</tr>
<tr>
<td>95/90</td>
<td>7,5</td>
<td>12,6</td>
</tr>
</tbody>
</table>

Correlations for mentioned parameters for sliding velocity can be calculated with

\[ v = 2\pi \frac{RPM}{60} r \]  

Eq. 2

And nominal pressure is calculated with

\[ p = \frac{F}{A} \]  

Eq. 3

where A is the surface area of the clutch interface [m²].
Materials and method


4. Results

The result section is divided into four subsections, from where first two sections are concerning actual performance of the wet clutch, including its degradation when using surface finish molybdenum discs only. Last two subsections are showing a comparison of the friction performance when using different lubricants and different initial surface roughness of the friction plates.

4.1 Friction performance

Figure 7 shows the friction performance for the 100th, 300th and 600th cycle, after running-in. Coefficient of friction has slightly positive slope with increasing the sliding velocity. In addition the friction coefficient is very stable and repeatable, indicating stable working performance.

![Friction performance graph](image)

*Figure 7 Friction performance of molybdenum-based material - Coefficient of friction in relationship of sliding velocity (Corresponding to tested specimen in case of smaller friction plate, with contact pressure of 6.3 MPa, see Fig 10.)*
4.1.2 Average COF for larger friction plate

Figure 8 shows the change in mean coefficient of friction for sliding period on 50 RPM (≈ 0.25 m/s) for one thousand cycles. Running-in period is seen for both cases for first hundred cycles. However, in case of higher contact pressure this period is prolonged up to two hundred cycles. After that the coefficient of friction stabilizes, although it shows slight decrease in coefficient of friction.

In the case of the lower pressure, coefficient of friction suddenly drops from 0.11 to 0.085 in first hundred cycles. In addition the behavior of mean COF (Coefficient of friction) shows very stable decrease compare to the case for higher pressure, where the running-in phase is prolonged (up to 200 cycles). It shows highly unusual behavior for first 70 cycles, because testing was aborted twice, in the 20th cycle and 40th cycle, due to the unusual noise. Therefore the plate was visually inspected for each shut down in case of any indications of a failure, which was not detected. In addition the work of EP additives was detected on the surface of a steel plate as a corroded layer. Testing was continued after both shut downs. However, due to disassembling/assembling of the Test-rig, system was interfered. This can be seen with the sudden increase in mean COF (21st cycle and 41st cycle). Further on, mean COF decreases much slower as in the case of lower pressure. This could be attributed again to the work of EP additives, which form low-shear layer on the damaged ferrous surface, which can affect the running-in period.

![Figure 8 Mean COF for Large molybdenum friction plate](image)

The contact surface temperature is continuously rising from room temperature, T = 24°C to 80°C in case of higher surface pressure, where in case of lower surface pressure it only reached to 60°C. This could also explain the behavior of lower friction in case of high pressure.
Apart from running-in period, where the mean COF drops significantly, there is also a stable steady-state period. In case of lower pressure mean COF is slightly higher than in case of the higher pressure. It also shows lower fluctuations for mean COF. However, for both cases the mean COF drops for around 0.005 after running-in period until the end of one thousand cycles.

Friction performance is shown in Fig. 9 for three stages; the beginning of the experiment, after running-in period and the rest of the experiment.

The COF shows very stable behavior over the sliding velocity. After the running-in phase, COF drops from the initial value of around 0.11 to 0.09 in case of higher surface pressure and in case of lower surface pressure to around 0.085, but again it a shows stable behavior over sliding velocity. In case of the lower pressure, the friction performance does not show any significant change from 2\textsuperscript{nd} to 3\textsuperscript{rd} stage, which is also seen in Fig. 8. However in case of higher pressure, friction performance changes, from relatively stable slope of COF to slightly negative, which also explains the behavior of mean COF in Fig. 8.
4.1.3 Average COF for smaller friction plate

Figure 10 shows similar behavior for the mean coefficient of friction, see Fig. 8. First 100 cycles shows similar running-in stage for case with lower pressure, where in case of high pressure this is prolong to approximately 450 cycle. However testing starts with relatively high load impact, which could affect the later running-in phase. In case of a lower pressure the system becomes very stable after running phase up to approximately 700 cycle, after that the friction coefficient slightly drops for both cases.

![Figure 10 Mean COF for small molybdenum friction plate](image)

Contact surface temperature is continuously rising from room temperature, $T = 24^\circ$C, to $55^\circ$C in case of higher surface pressure, where in case of lower surface pressure it only reached $45^\circ$C.
Friction coefficient Fig. 11 has a similar stable behavior as in the case of using large friction plate, see Fig. 9. In addition the same pattern is seen for the case of using a smaller friction disc, where the friction coefficient drops with increasing the number of cycles. Furthermore a slope of the friction coefficient shows slightly positive behavior after running-in phase throughout the testing.

The system has relatively better friction behavior than the system with bigger plate size, which could be attributed to the effect of lubricant, since the flow in both cases of testing (large and small friction plate) is the same. Also better friction behavior could be the effect of applying lower load, due to the initial undesirable load impact. In addition by using bigger disc size, sliding velocity is slightly increased at outer radius, which could also attribute to the different behavior. Moreover in case of using smaller plates, the contact temperature is much lower, since heating from friction force is not sufficient.
4.2 Wear

Material degradation for previous testing procedure is presented in this section. Surface topography analyses are carried out before and during the experiment in order to monitor any surface changes. This is conducted with a 3D Optical profilometer. Surface is characterized by average surface roughness Ra, which is most commonly used to describe the surface topography. Furthermore, wear of material is measured by measuring the depth of the cross section area of the sample for steel disc only, since the degradation of material is clearly visible. However, surface changes for the friction plates are also measured. In addition, surface defects on the friction discs are presented in this section.

Material degradation showed to be high in case of using non-surface finished molybdenum plates, which led to seizure of both discs; a molybdenum based friction coating on the friction disc and a counter steel plate. Therefore further investigations with non-surface finished discs have not been conducted.

4.5.1 Degradation of steel disc

For the investigated combination of materials, steel presents the softer material compared to harder molybdenum coating. Therefore wear of material generally occurs on the softer steel discs, when operating in boundary regime.

Figure 12 shows the depth of wear on the steel plates after 1000 cycles. In case of the larger friction plates the wear depth is higher compared for the case with smaller friction plates, although same contact pressures are used respectively. However, the same trend of higher wear for higher pressure is seen in both cases, but the ratio of it is different. In case of the larger friction plates for 100 percentage higher pressure the wear increases for approximately 20 percent, where in case of the smaller friction plates the wear increases for approximately 300 percent.
Results

In case of smaller friction plate operating under higher surface pressure initial load impact was present, resulting in high coefficient of friction (high torque). It seems that the coating is instantly locally damaged, as coating removal, see Fig. 13 (b), which has been detected after the experiment.

4.5.2 Roughness
Surface roughness measurements are conducted before and after the testing. The friction plate surface roughness (surface finished discs) drops from initial Ra = 1.8 µm to around Ra = 0.95 µm. This trend is seen for all four setups. For the steel plate the roughness value stays approximately the same as the initial value about Ra = 0.25 µm. However, the orientation of surface topography changes from parallel to circular.

4.5.3 Surface defects
Friction plate surface defects are detected after the experiment. Two main type of changes are seen in Fig. 13. Case (a) presents the effect of plowing of the harder material into the softer, where steel material is physically accumulating on the surface of Molybdenum coating. Case (b) presents the removal of the coating material, which could be attributed to initial load impact. These defects are present for all cases of testing, but the proportion of it changes from plate to plate, where some have more locations of these effects compare to others. However fully inspection and characterization of this defects have not been done.

![Figure 13 Surface defect for Friction plate](image)
4.6 Comparison of different lubricants

Friction performance for the selected materials in combination with different lubricants has been investigated. All discs in this investigation are run-in before testing. Running-in phase was carried out with about 20 cycles under the same operating conditions as applicable to testing. The purpose of this setup is to compare friction performance while using different oil only. Figure 14 shows friction behavior for two cases; case 1 presents friction performance with the lubricant mainly used in this work, case 2 presents friction performance for similar API grade lubricant, however this lubricant is without FM (Friction modifiers) additives.

From Fig. 14 can be seen distinguishable comparison of the friction performance for both cases. Case 1 shows vary stable friction coefficient when increasing the sliding velocity during ramp up phase. However in case 2 a negative slope of friction coefficient is seen, which indicates the lack of FM modifiers. Therefore it can be clearly seen the effect of the FM in case 1 compared to the case 2. In addition FM are used in common applications to modify the friction behavior – change the slope of friction coefficient over sliding velocity.

![Figure 14 Friction performance for two different oil (large plate)](image)

Negative slope of friction coefficient commonly leads to system vibrations, commonly known as shudder effect, which leads to unstable system. This behavior was detected with further experimental testing.
Figure 15 shows the typical effect of shudder, caused by negative slope of friction performance for case 2 with further testing under same operating conditions as for the case in Fig. 14. However, this behavior is undesirable for the overall system and can lead to system failure.

Figure 15 Shudder effect (large plate)
4.7 Comparison of friction for non-surface and surface finished Molybdenum friction discs

Figure 16 shows the friction performance for a non-surface finished and surface finished friction plate obtained after running-in period. Running-in phase was carried out with about 20 cycles under the same operating conditions as applicable to testing. Friction coefficient in case of non-surface finished friction plate with average surface roughness $Ra = 8 \, \mu m$ starts with relatively high value of around $0.10 - 0.11$ at low sliding velocity then drastically drops when increasing the sliding velocity. This trend is repeating itself regardless of contact pressure and discs size and leads to system failure, which occur as seizure of both discs.

![Friction performance](image)

Figure 16 Friction performance for surface finished molybdenum and non-surface finished molybdenum coating

Friction coefficient in case of the surface finished friction plates with average roughness $Ra = 1.8$ starts below $0.09$ at low sliding velocity and has a relatively steady friction value up to a sliding velocity of $0.25 \, m/s$. Therefore the system degradation is stable, without any unwanted failures, compared to the case with higher surface roughness.

Note that initial nominal contact pressure ($p = 6.3 \, MPa$) and contact temperature ($T = 60^\circ C$) are the same for both cases.
4.8 Comparison of friction performance by using different friction material

Figure 17 shows friction performance for a molybdenum and a paper based friction material. Operating conditions differ, therefore only a rough comparison can be made. The contact pressure in case of the paper based plate is three times lower as compared to the molybdenum based disc ($p = 6$ MPa), whereas the initial contact temperature of paper based plate ($120^\circ$C) is three times higher.

Friction coefficient in both cases is increasing with increasing the sliding velocity. However, coefficient of friction shows much stable behavior when using molybdenum based friction plate. In addition the coefficient of friction for paper-based disc stabilizes at $\mu = 0.1$, which is slightly higher than for molybdenum based disc ($\mu = 0.08$). However paper based disc has lower static coefficient of friction ($\mu = 0.06$) compared to molybdenum based disc ($\mu = 0.07$).

![Figure 17 Friction performance for two different friction material](image-url)
5. Discussion

The testing cycle, used in the measurements, has been designed in order to simulate the real operating conditions for a specific wet clutch application (LSD) with molybdenum coating as friction surface on the friction discs. Values of sliding velocity and contact pressure are in range with those in the applications. However, the sliding time this has been prolonged, which could potentially result in enhanced ageing of the clutch system. Further on, the condition of clutch discs are needed to be addressed, when it comes to the clutch behavior.

Surface roughness of the molybdenum coating was shown to be an important parameter, which affects the behavior and durability of the wet clutch to a large extent. Usage of non-surface finished molybdenum coated friction discs results in unstable system, due to its high wear rate, which eventually leads to system failure. However by reducing the surface roughness of the molybdenum coatings, the system becomes stable, which leads to a stable system degradation. Apart from reducing surface roughness, friction plate size has also been reduced from standard sizes, in order to obtain higher contact pressure, which are comparable to the ones used in the real system. The effect of reducing sliding area of the friction plates can be seen from Fig. 8 and Fig. 10. Under the same operating conditions a smaller clutch plates have a more stable average friction.

Surface finishing has not been performed on steel plates due to the limited amount of time. However, it seems reasonable that the system can become even more stable if steel plate discs are additionally surface finished. Overall, the plates’ condition present a crucial factor to obtain a relatively stable clutch behavior. Further on, this friction behavior can be divided into two main phases; running-in phase and operating phase, see Fig. 8.

The running in phase shows great dependence upon system design and operating conditions. This can be attributed to initial surface topography of both plates as well as to initial contact pressure, where in case of lower initial pressure the running-in phase is shorter. This indicates stable ongoing friction behavior and also results in lower wear rate. On the other hand, prolonged running-in phase could be the result of the work of EP additives, which protect the surface of the steel disc and indirectly effect the running-in phase. It seems that initial running-in phase drastically effect later friction behavior of the system. Moreover surface defects located on the coating are assumed to appear in the running-in phase, since system becomes relatively unstable after that. After the running-in phase, a relatively stable phase is establish, where friction stabilizes up to some extent, although a slightly decrease in mean coefficient of friction can be seen, which could be attributed to temperature effects.

Friction performance, where coefficient of friction is the function of sliding velocity, is presented during ramp up period only see test sequence in Fig. 6. Three types of friction slopes can be seen: Negative slope, zero slope and positive slope. In general slightly positive slope is desirable to have good clutch behavior and to avoid undesirable system vibrations.
Discussion

Wear of the material is shown to increase with increasing contact pressure. Apart from plates’ conditions, no consideration has been given to lubricant conditions, which drastically changes the appearance of color from yellow to completely black. Further investigations should be done in order to investigate possible compounds formed during testing.

Effect of temperature has not been fully investigated, since heating is not sufficient in the test rig used in this work. Temperature range is from 24°C to around 80°C, depending on the disc sizes and contact pressures used. Furthermore, temperature rises approximately 2°C after each testing cycle, see Fig. 6. In addition external heating device has been used to additionally heat the lubricant. However, the power of the heating system was insufficient to heat the lubricant above 80°C, since the lubricant was cooling rapidly between the test cycles in the test rig. Therefore, bigger discs should be used in order to obtain more heat, generated from friction torque.
6. Conclusion

- Molybdenum coating shows stable friction characteristics for wet clutch applications in combination with a suitable lubricant, when the surface of the coating is additionally surface finished to reduce surface roughness.
- Surface roughness affects the overall system behavior to a large extent. Rougher surface of the molybdenum coating (non-surface finished) shows higher coefficient of friction and also much higher system degradation which was leading to a fast failure of the system.
- Coefficient of friction shows relatively stable behavior in relationship of sliding velocity when smoother molybdenum coating is used. In addition, FM additives are shown to improve friction performance of the system.
- Abrasive wear of the steel plate presents the main mechanism of system degradation.
7. **Future work**

Although the scope of this work was originally focused to investigate the behavior of the system, much effort had been given to firstly establish the stable system. However some questions have been answered, more experimental work needs to be done in order to verify existing speculations. Some issues of interests are listed below:

- Investigate the system behavior, when steel plates are additionally surface finished. In addition the friction plate sizes should be as close to the ones used in real applications, since easier comparison can be made. By doing so, the used Test-rig’s capabilities of measuring higher load should be improved.
- Investigate lubricant condition and its degradation after testing, with a lubricant analyze.
- Investigate how elevated temperature (over 80°C) effects the friction performance of the molybdenum based friction plates in a wet clutch application.
Future work
8. References


