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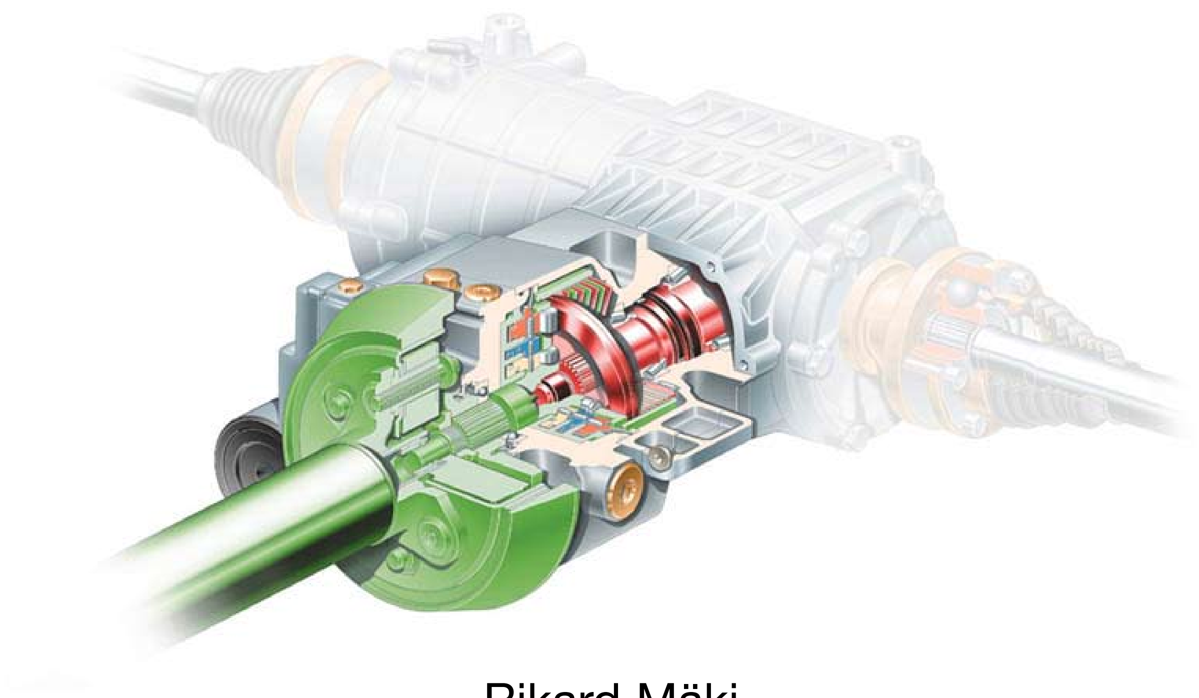
DOCTORAL THESIS

Wet Clutch Tribology

- Friction Characteristics in Limited Slip Differentials



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Preface

It has taken me almost five years to arrive at this point, but it has been great fun. During these years I have had the opportunity to work with many great colleagues at Luleå University of Technology who all deserve a big hug! Thanks are also due to Michael M. Khonsari and all colleagues at Louisiana State University for giving me a memorable time in Baton Rouge.

I also want to express my gratitude to my supervisor Erik Höglund and my corporate supervisors Bager Ganemi and Richard Olsson for invaluable support during the entire project. The contributions of Kent Ekholm, Rafel Triviño Flores and Pär Nyman to this thesis are highly valued and remembered.

This work has been dependent on the financial support of Haldex Traction Systems AB, Svenska Statoil AB, The Swedish Agency for Innovation Systems and The Swedish Foundation for Strategic Research. This support is much appreciated.

Two people deserve a special mention; Anders Pettersson who has been a valued friend in the office and on our various travels and adventures around the world on both business and "not so very much business" occasions. Mikaela Omark has always supported me in my endeavors and made it worth while to leave the office at the end of the day.

And of course I also want to express gratitude to all my family and friends for being there.

Luleå, August 2005.

Rikard Mäki

Summary of the Thesis

In recent years, electronically controlled automotive transmission systems, where wet clutches are used as intelligent differentials, have appeared in the market. These applications impose great demands on the transmission fluids and friction materials used as well as on controllability and vibration preventive (anti-shudder) properties of the clutch systems.

This thesis focuses on transmission fluids used in wet clutches in all-wheel drive systems. The investigated all-wheel drive system, featuring a wet multi-plate clutch with a sintered brass based friction material, is described.

A comprehensive literature review section outlines the state-of-art in this field and gives an insight into many of the problems commonly experienced in this type of application. Different methods used to investigate the function of wet clutch transmission fluids are also presented.

Test equipment designed during this thesis work in order to determine the frictional characteristics of transmission fluids is described. This equipment measures friction torque, applied load, oil temperatures and the actual temperature experienced by the fluid in the contact zone.

Base oil type and viscosity have been found not to significantly influence friction characteristics of a wet-plate clutch , indicating that the torque is primarily transmitted by asperity contacts rather than fluid films. Oil additives, on the other hand, have a considerable influence on friction, again leading to the conclusion that tribolayers on contacting asperities rather than fluid films govern friction. From these observations it can be concluded that the lubrication regime under the conditions studied are boundary lubrication, moving into mixed lubrication at high velocities and low temperatures.

Results show the influence of several operating parameters on the frictional behavior of the clutch.
Temperature is shown to have significant influence on the friction of transmission fluids which decrease with increasing temperature. It is therefore necessary to measure the true temperatures in the clutch contact in order to obtain realistic measurements of friction. A method which excludes the influence of temperature on measured friction data have been developed and verified.
The influence of clutch disc pressure on friction is quite moderate compared to the influence of temperature and sliding velocity.
The influence of velocity on friction is governed by the transmission fluid and friction material used in the clutch. The friction-velocity relationship is a good indicator of the fluid's ability to suppress friction induced vibrations. It is, however, important to measure the friction-velocity relationship at constant temperature, or to compensate the relationship for temperature effects.

A successful method to develop transmission fluids has been developed. Formulated fluids allow good anti-shudder properties to be combined with good lubrication performance for other machine elements present in the transmission. Interactions between different additives must be considered which can, in many cases, completely alter the friction characteristics since additives compete for the same adsorption surface. Extreme pressure additives have been found to be particularly troublesome when used in combination with other additives as far as their ability to maintain good anti-shudder properties is concerned.

Based on the knowledge of clutch performance obtained from the research presented in this thesis, a model to predict transmitted clutch torque has been developed. This accurately determines the transferred torque from the current operating conditions and the thermal history of the clutch.

It can be concluded that thermal effects have a significant influence on the torque transferred by the clutch, and it is therefore necessary to have a thermal model of the clutch combined with a temperature dependant boundary friction model based on empirical friction data for the friction material/transmission fluid combination of interest.

Sammanfattning

På senare tid har elektroniskt styrda drivlinor blivit allt vanligare i personbilar. I dessa transmissionssystem används ofta våta kopplingar för att överföra önskade vridmoment mellan olika axlar. För att uppnå lättkontrollerad och vibrationsfri momentöverföring ställs höga krav på de använda transmissionsoljorna och friktionsmaterialen.
Denna avhandling behandlar främst transmissionsoljor för våta kopplingar i fyrhjulsdriftsystem. En beskrivning av det undersökta fyrhjulsdriftsystemet och den ingående flerlamellkopplingen ges. Dessutom beskrivs de friktionslameller av sintermässing som används i denna tillämpning.

En omfattande litteraturstudie som syftar till att identifiera forskningsfronten inom området presenteras och ger en insikt i vilka problem som måste lösas i denna typ av applikationer. I anknytning till detta beskrivs även olika metoder som används för att undersöka funktionen hos våta kopplingar.

För att undersöka friktionsegenskaperna hos olika transmissionsoljor har en testutrustning konstruerats. Denna utrustning mäter överfört vridmoment, pålagd kraft, oljetemperatur och kontakttemperatur i kopplingen. Både konstruktionen och typiska testcykler beskrivs detaljerat.

Mätningar visar att basoljetyp och viskositet endast har en liten inverkan på friktionsegenskaperna (momentöverföringen), vilket tyder på att friktionen uppkommer i asperitetskontakter snarare än genom viskösa effekter i en oljefilm. Oljeadditiv har däremot kraftig inverkan på friktionen, även detta tyder på att friktionen genereras av additivskikt på yttoppar snarare än av en oljefilm. Sammantaget visar detta att den smörjregim som råder under undersökta förhållanden är gränsskiktssmörjning som går in i blandfilmssmörjning vid höga hastigheter och låga temperaturer.

Resultat visar även hur en mängd olika parametrar påverkar friktionsegenskaperna i kopplingen. Studerade parametrar inkluderar temperatur, glidhastighet, tryck och oljesammansättning.
Temperaturen har betydande inverkan på friktionsegenskaperna hos transmissionsoljor. Friktionen minskar med ökande temperatur. Därför är det nödvändigt att mäta den korrekta temperaturen i kopplingen för att få relevanta mätvärden på friktionen. En metod för att presentera mätdata på ett sätt där temperaturens inverkan elimineras har utvecklats och visat sig fungera väl.
Friktionspåverkan från den pålagda normalkraften är liten jämfört med inverkan från temperatur och glidhastighet.
Glidhastighetens påverkan på friktionen bestäms av den aktuella kombinationen av smörjolja och friktionsmaterial. Sambandet mellan friktion och glidhastighet ger en god bild av smörjmedlets vibrationsdämpande förmåga. Det är dock viktigt att detta samband mäts vid konstant temperatur eller att mätdata kompenseras på ett lämpligt sätt.

En framgångsrik metod för att utveckla transmissionsoljor har utvecklats. Formulerade oljor har visat att det är möjligt att kombinera vibrationsdämpande egenskaper med goda smörjegenskaper med avseende på andra maskinelement i transmissionen. Växelverkan mellan olika additiv måste beaktas och kan i många fall leda till kraftiga förändringar i friktionsbeteendet eftersom alla additiv konkurrerar om att fästa på samma metallyta. Speciellt extremkontaktstrycksadditiv (EP) har visat sig vara svåra att kombinera med andra additivtyper.

Baserat på den nyvunna kunskapen om kopplingars momentöverföring har en teoretisk modell utvecklats. Denna modell kan förutsäga vilket överfört moment kopplingen genererar under givna förhållanden.
Eftersom temperatureffekter har stor inverkan på friktionen är det nödvändigt att ha en termisk modell av kopplingen som kan kopplas till en temperaturberoende friktionsmodell. Friktionsmodellen är baserad på empiriska friktionsdata för aktuell kombination av friktionsmaterial och smörjmedel.

Contents

Introduction

History of Tribology

The word tribology was first used in 1966 in the "Jost Report" which was published by the UK's Department of Education and Science [1]. The word tribology originates from the Greek word tribos which means, "to rub". The emergence of this new interdisciplinary concept was based on the needs and demands of industry, rather than from general interest expressed by the scientific community.
The Jost Report was the first paper to investigate the potential, both economic and technological, of tribology. The report defined this new evolving science as:

TRIBOLOGY is the SCIENCE and TECHNOLOGY of interacting surfaces in relative motion and of the practices related thereto.

Although the name of the science is quite new, the practical applications of tribology date back much further. The first lubricant used to reduce friction was probably bitumen; indications of the use of bitumen date back to around 6000 B.C. [2]. In Egypt (around 2500 B.C.) wheels, primitive bearings and lubrication were used when building temples and pyramids.
Scientific studies of friction were first conducted by Leonardo da Vinci around 1500 A.D. He was the first to define the coefficient of friction as a ratio between the friction force and normal force. Unfortunately, the work of da Vinci remained unpublished for several centuries. The laws of friction were rediscovered by Guillaume Amontons in 1699 and later Leonard Euler (1750) who derived an analytical definition of friction using the symbol μ [3].
Even today, friction is commonly defined according to the findings of Charles Augustin Coulomb from 1785. Coulomb confirmed the results of the previous investigations, but in addition found that friction is independent of sliding velocity. He also made a clear distinction between static and dynamic friction.
This oversimplification actually persisted until the mid 20th century when researchers such as Dokos (1946) and Rabinowicz (1951) cast new light on the phenomenon of friction. These researchers showed that the static coefficient of friction is time dependant and that the dynamic coefficient of friction is dependant on the sliding velocity [4].
In most applications, reduction of friction is the primary concern. In some applications, however, friction is necessary and instead of reducing friction a high and well defined level of friction is required. One such application is in clutches.

Wet Clutches

A wet clutch is basically a clutch working under lubricated conditions. The configuration most commonly used is called a multiple disc wet clutch as shown in Figure 1. This consists of friction discs attached to one shaft by splines. Separator discs are similarly connected to the other shaft. When disengaged, the clutch transmits only a small drag torque due to viscous friction, and thus both shafts are free to rotate independently. The clutch is engaged by applying a normal force from the hydraulic cylinder shown in Figure 1 (mechanical or electromechanical actuators are also commonly used), clamping the friction and separator discs together and thereby allowing torque to be transmitted between the input and output shafts.

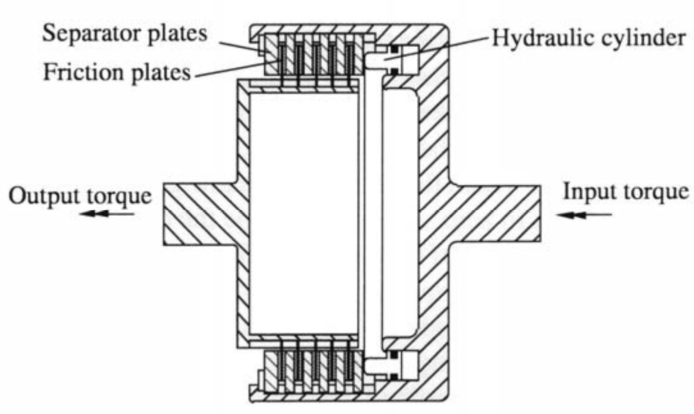

Figure 1: Multiple disc wet clutch [5].

The most common use of wet clutches is in automatic transmissions (AT) used in automobiles and trucks. As a result of this, most research on wet clutches related to this application. A good introduction to ATs has been written by Kato and Shibayama [6]. Investigations into the engagement characteristics of wet clutches in ATs have been performed by a number of authors, both experimentally [7-11] and theoretically [12-16].
Wet clutches are also used in applications where control of the torque transfer process is required. These applications include wet brakes, lock-up clutches, launch clutches in motor cycles and limited slip differentials.

Differentials using Wet Clutches

In recent years, several electronically controllable automotive transmission systems, where wet clutches are used as intelligent differentials, have appeared on the market. The work presented in this thesis is focused on transmissions for all-wheel drive systems, although there are similar applications where wet clutches are used as differential brakes on two-wheel drive vehicles [17].

All-Wheel Drive Systems

All-wheel drives (AWD) were originally used to improve the off-road capacity of military and other off-road vehicles [18]. In the early eighties, a new market for all-wheel drive performance vehicles emerged with the introduction of the Audi Quattro [19].
Since then a number of studies have shown the benefits to vehicle dynamics of all-wheel drive vehicles compared to two-wheel drive vehicles [20-22].
In traditional all-wheel drive systems, a viscous coupling is commonly installed on the propeller shaft in order to transmit torque while still allowing some difference in rotational speeds between front and rear axle. The function of the viscous coupling has been the subject of extensive research for quite some time [23]. The drawback with a viscous coupling is that it is not controllable during operation and therefore does not work well in combination with electronic driving aid systems such as electronic stability programs and traction control systems.
AWD-systems where wet clutches are used as controllable differentials include the Toyota Active Torque Control 4WD [24], the All-Mode 4WD [25] and the ATTESA E-TS [22, 26]. One of the most widely used systems of this type is the Haldex LSC (Limited Slip Coupling) system described below.

Haldex LSC All-Wheel Drive System

Haldex Traction Systems have developed an active-on-demand all-wheel drive system for automobiles with short system activation and deactivation times [27, 28]. Currently the Haldex LSC system is used in vehicles produced by several manufacturers including Volvo Cars Corporation, Ford Motor Company and the Volkswagen group.

The Haldex LSC AWD system features a multiple disc wet clutch with clutch plates covered with a sintered friction material. The clutch pack distributes drive torque to the rear axle of the vehicle. By using a wet clutch, torque transfer control is enhanced which makes it possible to electronically control the drive torque distribution between the front and rear axle in order to optimize vehicle dynamics.
A schematic of the coupling can be seen in Figure 2. The coupling is mounted on the propeller shaft. The shaft on the left in the figure is connected to the rear axle of the vehicle whilst the right hand shaft is connected to the front axle via the propeller shaft. When a speed difference occurs between the front and rear axle, a cam on the rear axle causes a pumping action on the hydraulic piston pump. The hydraulic pressure generated is applied to the clutch pack in order to reduce the speed difference between the shafts, thus engaging the all-wheel drive. The torque transmitted by the coupling is controlled by a throttle valve.

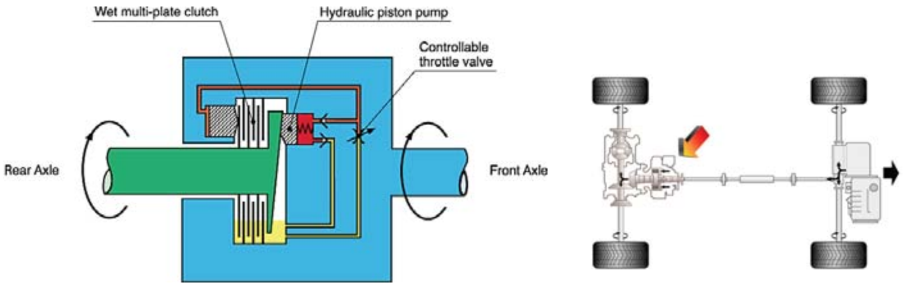

Figure 2: Schematic of the Haldex Limited Slip Coupling.

This type of AWD system has several advantages over traditional transmission systems. Activation and deactivation occur rapidly rapid and are electronically controllable via the throttle valve: The coupling therefore works well in conjunction with different electronic driving aid systems such as electronic stability programs, traction control systems and anti-lock brake systems which often require AWD deactivation. The possibility of system deactivation makes low speed maneuvering such as parking easier. It also makes it possible to enable run-flat-tire functions and vehicle towing.
In this type of application the clutch has to transmit high torque (~2000Nm) at low sliding velocities, while complying with strict demands for anti-shudder properties and torque control.

Friction Characteristics

In applications where the clutch is operating at low sliding velocities under long periods of time, such as limited slip differentials, it is important to control friction characteristics.
Typical friction characteristics for three different fluids, but with identical friction material, are presented in Figure 3 (similar differences can be seen for a single fluid with different friction materials). In order to avoid vibrations (commonly referred to as shudder in clutch applications and squeal in brake applications) the friction vs. velocity (μ-v) relationship should have a low static coefficient of friction (μ_s) and a dynamic coefficient of friction (μ_d) that increases as the sliding velocity increases. Oil A will suppress vibrations, whilst oils B and C may be susceptible to vibrations since they show a negative slope in some regions.
The influence of the friction vs. velocity curve on the anti-shudder performance of wet clutches is well established [11, 29-33]. The advantage of a positive μ-v slope can easily be shown mathematically from engineering vibration calculations [34-36], however, it should be noted that a positive slope is neither a necessary nor sufficient condition to guarantee stability [37, 38].
Additional information on friction characteristics can be found in *Paper A* of this thesis.

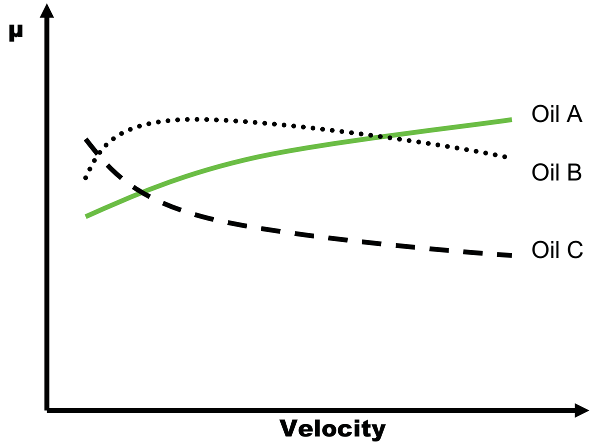

Figure 3: Schematic friction-velocity curves for different automatic transmission fluids (ATFs).

Clutch Temperature

The interface temperature in the clutch is known to have a significant effect on friction [11, 32, 33, 39]. This can be explained from the fact that temperature affects both fluid viscosity and the formation of so called tribolayers; surface active additives in the transmission fluid that are present at the sliding interfaces in the clutch and which govern friction. The rate of generation of the tribolayer is influenced by the temperature dependant surface activity. In addition, the surface temperature will also determine which type of additives will be dominant in the tribolayer [40, 41].

The clutch temperature is governed by power input (sliding velocity and transmitted torque) and heat dissipation. The clutch temperature can either be measured experimentally [7, 32, 42-44] or calculated from theoretical models [13, 16, 44-46].

Additional information on clutch temperature can be found in *Paper A* of this thesis.

Fluid Formulation

A transmission fluid consists of one or more base oils and a number of additives. A wet clutch transmission fluid must provide adequate friction performance and at the same time have good shear and oxidation stability, anti-wear performance and corrosion resistance.

The concentration of each additive, and the balance between them, is important when formulating transmission fluids [29, 36, 39, 47]. For surface active molecules, the type of friction materials must also be considered; different additives must be used for sintered bronze compared to paper-based materials [48, 49].

Performance requirements for transmission fluids are generally set by OEMs and include friction characteristics, friction durability, oxidation stability and wear. The first official ATF specification “Automatic Transmission Fluid, Type A” was published by GM in 1949. The Type A fluid, however, did not perform well in certain applications and in 1959 Ford Motor Company introduced a new standard called M2C33-A/B. With new applications, the demands placed on transmission fluids gradually increased which lead to further standards being developed. In 1967 GM published the Dexron standard and in 1987 Ford the Mercon standard, current versions of these standards (DX-III and Mercon V) are the most commonly used ATF classifications [50]. The friction requirements in current standards are, however, insufficient in applications such as differentials, especially if unconventional friction materials are used.

Additional information concerning fluid formulation and different additives can be found in *Paper A*.

Friction Materials

The friction material used has significant influence on the friction characteristics of a clutch. Friction materials commonly used include paper, sintered bronze, steel, carbon fiber, cork, asbestos and aramid fibers. Key material parameters include friction intensity, quality and stability, durability, heat resistance, heat adsorption and compatibility with oils and their additives.
Paper based friction materials, consisting of raw paper (cotton linters or cellulose fibers) in combination with a thermosetting resin [51], have been used since the late fifties. These offer low cost and good performance under low load conditions and remain by far the most commonly used friction material in wet clutch applications.
Sintered bronze and sintered brass friction materials are used where service conditions prevent the use of discs faced with paper, synthetic or organic linings. Sintered bronze friction materials are known to be sensitive to operating conditions but are able to cope with high temperatures and also decrease the clutch temperature due to their good thermal conductivity [52].
More recently, the use of carbon fiber as a friction material has increased. Mainly because of its very high heat/abuse resistance, combined with good, consistent friction behavior. However, widespread application of these materials is restricted by their high cost.

Experimental Equipment

Increasing effort is being put into developing experimental rigs for evaluating lubricants and friction material performance in wet clutches. The vast majority of these are designed for ATF applications or agricultural tractor oil applications.
Oil / friction material performance is generally studied with respect to three different criteria;

- Torque capacity – The amount of torque that can be transmitted before slipping occurs.
- Shudder – Frictionally induced torque variations resulting in noise.
- Engagement characteristics – The behavior of the torque transfer during clutch engagement.

The International Lubricant Standardization and Approval Committee (ILSAC) have published studies comparing different methods [9, 31].
Some standardized test rigs, such as the SAE #2 and LVFA, are described in *Paper A*. There is, however, no standard equipment designed to monitor clutch performance under limited slip conditions combined with high engagement forces, as prevalent in modern limited slip differentials.

Unanswered Questions

It is evident that a lot of work has been done investigating friction characteristics. There remain, however, areas that need to be further explored;
Current standard test equipment for the evaluation of wet clutches are designed for AT applications which have higher sliding velocities and lower normal loads compared to those found in the investigated differentials. It was therefore necessary to design a new test rig in order to simulate realistic operating conditions for this type of application.
Most experimental equipment does not measure the interface temperature even though it is known to greatly affect the friction. Much data concerning friction characteristics must therefore be considered incorrect. This is especially important at high engagement forces since these result in significant temperature increases.
Almost all research on wet clutches has been conducted on paper-based friction materials and it is therefore felt necessary to verify that the findings of other authors are valid with the much harder and more resilient sintered friction material and at the operating conditions of limited slip differentials.
In traditional AT-fluid formulation research many of the additives are surface active. This means that when changing the friction material from paper-based to brass-based, the effect of additives will be different resulting in the need to formulate and test specific transmission fluids for this type of applications.
Attempts to simulate clutches have so far addressed clutch engagements at high speeds, where hydrodynamic and squeeze film effects predominant. Existing simulations / models generally have a constant boundary friction model with respect to temperature and pressure, thus making the models useless in predicting torque in limited slip applications.

Contribution of this work

Research Questions/Tasks

The literature survey identified a number of areas where further research was needed, as presented in the previous section. In this work, the following questions were initially targeted;

- Which factors influence torque transmission in the clutch, and what lubrication regime is predominant in the limited slip differentials studied?
- Which methods can be used to measure and compare friction characteristics?
- How can fluids be formulated to give good friction characteristics for the clutch, whilst at the same time allow them to be able to lubricate other machine elements in the transmission?
- How can the torque transmission characteristics of the differential be predicted using simulation models?

Limitations

In order to limit the scope of work, the decision was made to investigate the sintered brass clutch material described below. This friction material is used in the commercial Haldex LSC which this research is aimed at. Investigations focused on effects related to the performance of the transmission fluids.
The torque transmission was primarily investigated at normal operating temperatures, i.e. 30-100ºC, since these temperatures are prevalent over the vast majority of the differentials life.
The investigations are also focused on steady-state conditions, i.e. not run-in or end-of-life.

Friction Material

The friction material investigated is a dispersion sintered brass lining applied to a hardened steel disc. This material is able to withstand higher stresses and temperatures compared to paper-based materials, and is fairly cheap to manufacture compared to, say, carbon-fiber materials. The friction material consists of a brass base (~70wt% Cu, ~20wt% Zn, ~2wt% Sn) containing a small amount of solid carbon based lubricants. In addition to this, the lining also contains solid friction increasing fractions such as silicon oxides. The separator disks are manufactured from hardened steel.
In order to investigate the validity of the results obtained during this study compared to other friction material, a few tests have been run on two different paper-based materials, another sinter material, one hybrid material and one carbon fiber material. From this it was found that the results are similar in most cases. The difference was most significant as far as interface temperature levels were concerned (where temperature increase almost doubled for friction materials with low thermal conductivity) and additive effects (most fluids developed for sintered materials worked quite well for all tested friction materials while fluids developed for paper-based materials did not work with the sintered materials).
The friction pair can be seen in Figure 4. Production discs have been used, and hence geometry and groove pattern are identical to the actual application. The outer diameter of the friction lining is 108mm, and the inner diameter 76mm. The area of contact when the oil grooves have been accounted for is approximately 2250mm^2. The grooves are intended to facilitate oil distribution to the area of contact and to help lower the temperature in the clutch by enhancing oil flow.

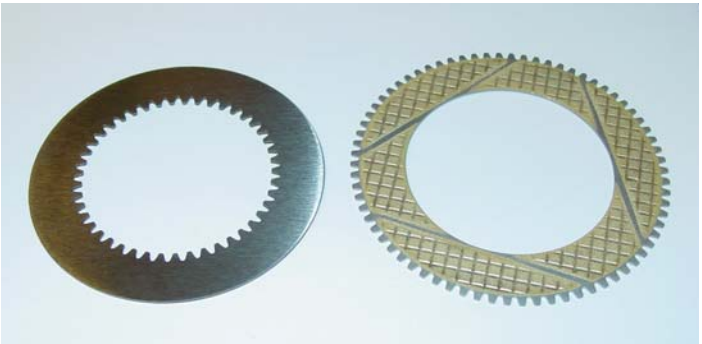

Figure 4: Investigated clutch discs. Hardened steel separator disk on the left, and friction disc with sintered brass friction material on the right.

Operating Conditions

Clutches in limited slip differentials, such as the Haldex LSC, work under conditions of continuous slip at low velocities while transmitting high torque. The investigated operating conditions reflect this.
The normal load used was in the range 10kN to 30kN, corresponding to a nominal surface pressure of 4.4MPa to 13.3MPa which are considered to be high pressures in clutch applications.
The rotational velocities studied were in the range of 0.5rpm to 125rpm, corresponding to a linear velocity of 2.4mm/s to 600mm/s at the mean radius of the friction disc (1rpm=4.82mm/s) which are considered to be low velocities in clutch applications.
The temperature was typically in the range of 30-100ºC, although some tests at low temperatures, -25 to 30ºC, have also been preformed. These temperatures are considered normal operating temperatures for transmission components.
For a typical transmission fluid with a viscosity of 35cSt at 40ºC the conditions described correspond to a $\frac{\eta V}{P}$ ratio in the order of 10^{-11} to 10^{-8}m.

Method

Experimental Equipment

The work in this project has been carried out using a number of different experimental test rigs; equipment to study friction, water compatibility and working life at Haldex [53], a Cameron-Plint TE77 reciprocating friction and wear tester for additive testing [54, 55] and the Wet Clutch Test Rig described in *Paper C* of this thesis.
The main part of the work was carried out using the test equipment described below which was designed to simulate the conditions prevalent in limited slip differentials while facilitating accurate measurement of interface temperature and friction.

The Limited Slip Clutch Test Rig

The Limited Slip Clutch Test Rig (LSCTR) was developed from ideas generated during the work on the Wet Clutch Test Rig presented in *Paper C*, combined with knowledge of the function of different test equipment in use at Haldex [53]. The main aims when developing this new test rig were to achieve high accuracy at low sliding velocities and to be able to apply higher normal loads on the clutch compared to the equipment described previously.

Function

An overview of the test rig can be seen in Figure 5 with a more detailed cross-sectional view in Figure 6. The rig is mounted on an aluminium stand (1). The base is a beam (2) of length 1600mm. On the left hand side, the motor and gearbox (3) is mounted. The driving force is transmitted by a shaft coupling (4), through a hollow piston hydraulic cylinder (5) to the clutch housing (6) containing ~200ml of transmission fluid. The torque transmitted by the coupling is transmitted through a torsion bar (7) to the torque measurement cell (8). The torque measurement cell (8) is connected to the beam (2) by a slider system (9) allowing for axial motion.

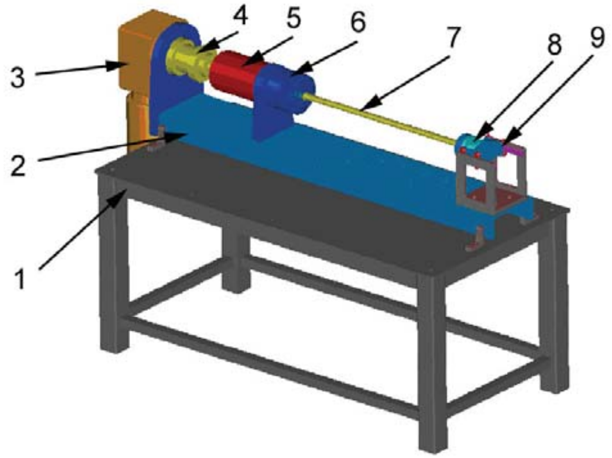

Figure 5: Overview of the Limited Slip Clutch Test Rig.

The maximum output torque of the motor (3) is 500Nm and its speed can be varied between 0.5 and 125rpm (2.4 to 600mm/s at the mean radius).
Torque is transmitted to the driveshaft (10) via a torsionally rigid coupling which is able to handle shaft misalignment (4).
The normal force on the test clutch is applied by a double acting hollow piston cylinder (5). The normal force is limited to 30,000N by a pressure limiting valve.

The friction disc (11) is connected to the drive shaft (10) and the separator discs (12) connected to the torsion bar (7). During operation, the parts shaded in Figure 6 rotate.
When a normal force is applied to the clutch by the hydraulic cylinder (5), torque is transmitted from the drive shaft (10) to the torsion bar (7). The transmitted torque can be measured by the torque measurement cell (8) and the applied normal force by the load cell (13); this is possible thanks to the slider system (9) which allows the torsion bar and torque cell to move freely in the axial direction. Both the force and torque transducers are full bridge, strain gauge type with built in amplifiers. The accuracy of the friction measurements from this rig is well within ±1%F.S. *Paper B* of this thesis provides additional details of this test rig.

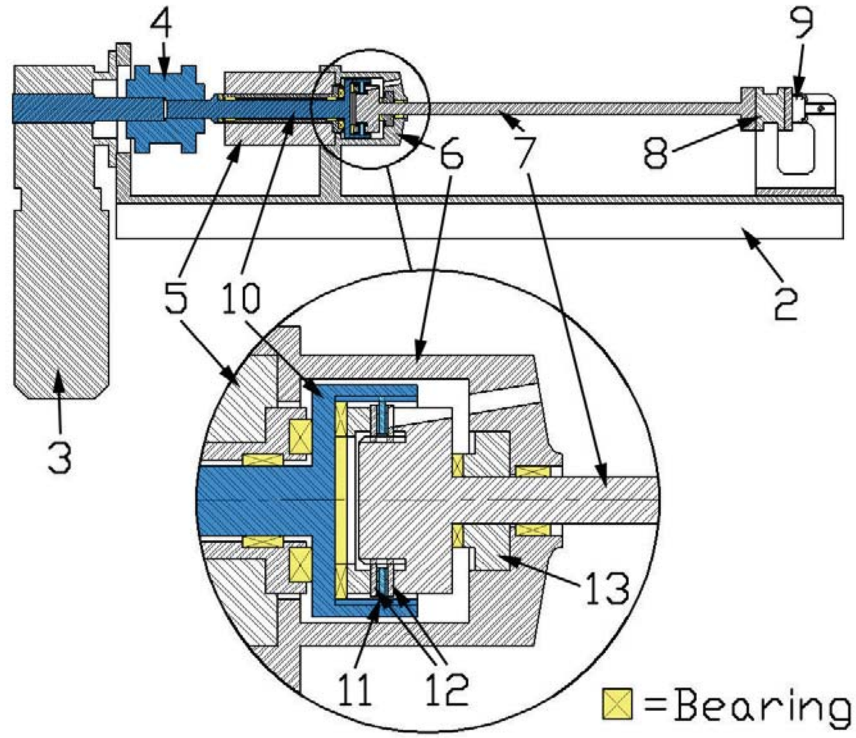

Figure 6: Simplified cross-section of the test apparatus. Shaded parts are rotating during operation.

Infrared temperature measurements of the friction disc can be made through a hole in one of the separator discs, the torsion bar and the housing. Thermocouples are installed in the oil sump to measure bulk oil temperature and in the separator disc to measure contact temperature. Values of contact temperatures obtained by the thermocouple and infrared temperature sensor did not differ significantly under the operating conditions typically examined in this study. For further information see *Paper C*.
By changing the length and diameter of the torsion bar it is possible to vary the natural frequency of the apparatus in the range from 100 to 500Hz. Measurements during this study have been performed using torsion bars yielding eigenfrequencies of 175Hz and 378Hz, which did not influence the friction values obtained nor act as a initiator of shudder. Shudder frequencies, however, will differ depending on the torsion bar length.

Data Analysis

In order to determine the friction-velocity characteristics of a transmission fluid using the rig, a test cycle with a linearly increasing sliding velocity is performed.

Measurement data obtained from a typical test in the test rig is presented in Figure 7. Initially a velocity of 1rpm and a normal force of 20kN were applied; the force was kept constant throughout the test. After 5 seconds the velocity began to increase, reaching 100rpm after an additional 10 seconds. During this time, the temperature of the clutch disc surface increases by more than 20ºC. Additional information regarding the test cycle and the influence of variations to the test procedure are presented in *Paper D*.

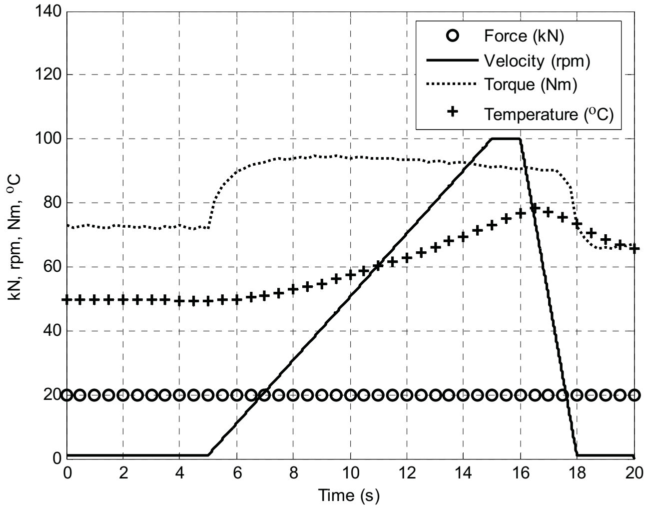

Figure 7: Typical measurement data obtained from the Limited Slip Clutch Test Rig.

Friction-Velocity

After run-in, the friction and separator disc surfaces are worn and the pressure distribution can be determined from the conditions of uniform wear. Knowing the pressure distribution and dimensions of the clutch disc it is possible to translate the measured torque and normal force into a coefficient of friction [56], (assuming constant pressure instead of constant wear would result in a decrease in the friction value by approximately 1%). The rotational speed can be converted into a mean sliding velocity (i.e. at the mean radius of the clutch disc) from which it is possible to plot the friction as a function of sliding velocity, Figure 8.

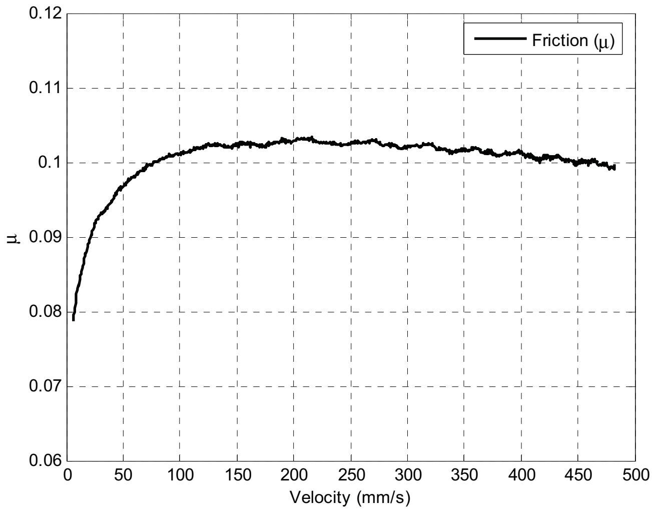

Figure 8: Friction-velocity characteristics calculated from data in Figure 7.

Friction-Temperature

In order to obtain more information from the measurement data it is good practice to measure friction-velocity characteristics at several different temperatures since these vary, as can be seen in Figure 9.

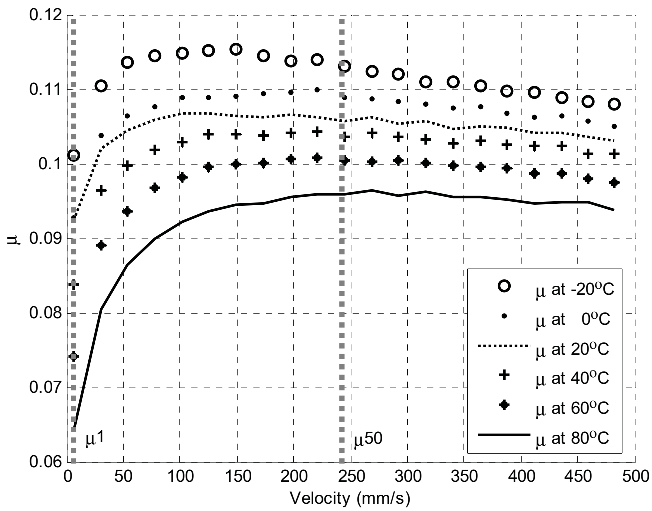

Figure 9: Friction-velocity characteristics measured at different initial temperatures, the friction at 1rpm (μ1) and 50rpm (μ50) are highlighted.

From the curves in Figure 9 it is possible to plot friction values at a given velocity as a function of temperature. In this thesis, the friction at 1rpm and 50rpm, denoted μ1 and μ50 respectively, are often referred to. These values and nomenclature where originally proposed by Ohtani et al. (with similar clutch plate sizes) and have been adopted by other authors as well [32, 57].

μ1 can be interpreted as the maximum torque capacity and μ50 as the dynamic friction. The ratio μ1/μ50 gives a good indication of the susceptibility of the friction material / oil combination to shudder, where a value lower than 1 is desirable in order to suppress vibrations. In Figure 10, μ1 and μ50 calculated from the data presented Figure 9 are shown.

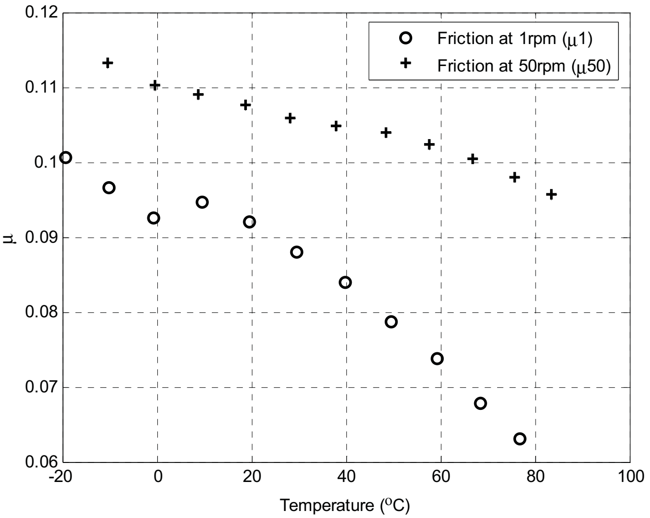

Figure 10: Friction at 1rpm and 50rpm as a function of temperature calculated from the data in Figure 9.

Temperature Compensation of Measurements

Temperature has a significant influence on friction. Since the temperature will change during measurements, as can be seen in Figure 7, it is desirable to remove the temperature effects from measured data. By doing this it is easier to distinguish other factors that influence friction, such as normal load, sliding velocity, transmission fluid used and friction material type. At the same time the influence of the chosen test cycle (which affects how temperatures will change) will be lessened and allow for easier comparison of measured data obtained under different conditions or in different test equipment.

In the context of examining anti-shudder properties, obtaining the friction at a constant temperature is also to be preferred. Since vibrations are initiated in time scales of tenths or hundreds of a second, the temperature will not have time to change during the initiation of shudder. In these cases it is the friction behavior at a constant temperature that is of interest.

When analyzing the measurement data, the friction-temperature relationship for a number of velocities have been evaluated, Figure 11. These results are based on data obtained from velocity ramps run at a range of different initial temperatures. By fitting a second degree polynomial to the data at every fifth rpm, using a least squares method, the continuous functions μ1(T), μ5(T), μ10(T) and so on could be determined.

From the data in Figure 11 it is possible to find the friction values at a specified temperature at any temperature within the interval, Figure 12. Thus the influence of the contact temperature change during the test has been eliminated since the friction can now be evaluated at a constant temperature.

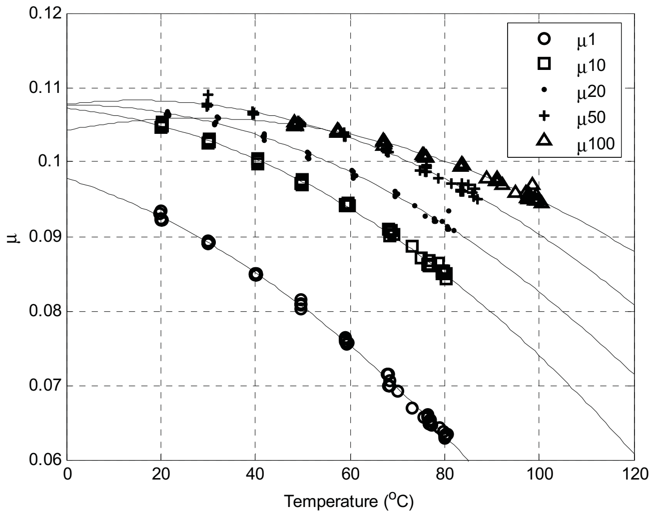

Figure 11: The measured friction-temperature relationship μ1, μ10, μ20, μ50 and μ100 with least squares fitted quadratic curves, plotted in that order where μ1 shows the lowest friction and temperatures.

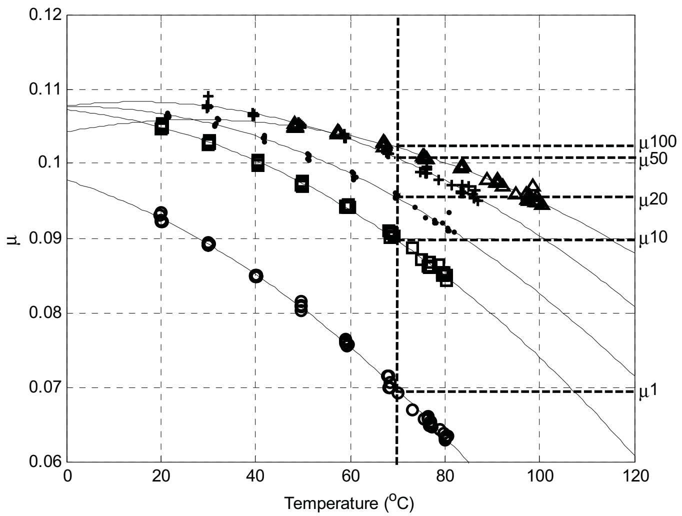

Figure 12: The friction values μ1, μ10, μ20, μ50 and μ100 for a constant temperature. In this case the values at 70°C are marked in the plot.

From the different friction-temperature curves shown in Figure 12 it is possible to extract data points for the friction at each velocity for a given temperature and to plot these values in a discrete friction-velocity plot, Figure 13.

In reality, measurement data is fitted to μ1, μ5, μ10, …, μ100, resulting in more data points than shown in Figure 13 and hence a smoother friction-velocity curve. This discrete friction-velocity plot is not sensitive to temperature changes in the contact zone during measurements and therefore less sensitive to varying test conditions. For more information on temperature compensation of measurements, refer to *Paper D*.

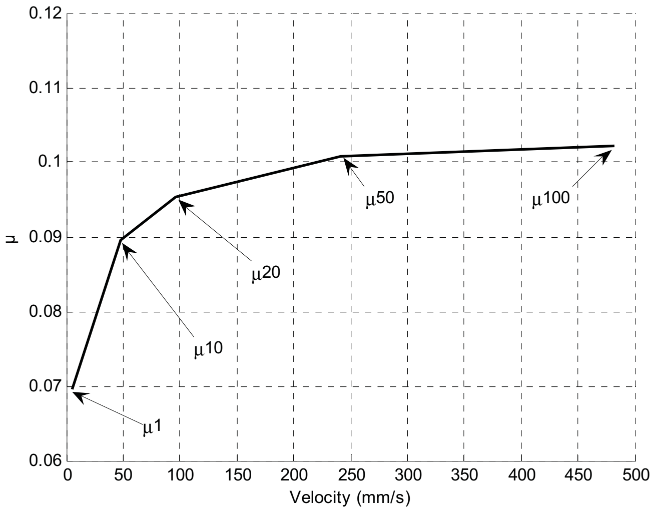

Figure 13: Discrete friction-velocity curve at a constant temperature of 70°C. Constructed from the values marked in Figure 12 with linear interpolation between μ1, μ10, μ20, μ50 and μ100.

Fluid Formulation

The process of designing transmission fluids was divided into a number of different steps;
An initial evaluation of the influence of base oils on friction characteristics was performed. A suitable base oil was then chosen based on thermal properties, long-time durability, lubricity requirements of other components and friction characteristics.
The next step was to evaluate the performance of single additives in the base oil. These tests were performed in a Cameron-Plint TE77 reciprocating friction and wear tester [54, 55]. This provides an efficient way to screen a large number of different fluids and is able to measure friction characteristics even with additive blends likely to cause shudder in a real clutch application.
A 'base package' was then combined using additives showing minimal influence on friction in combination with a known effective friction modifier. To this 'base package', other surface active additives were added one at a time to allow additive interactions to be studied. This lead to a completely formulated fluid being developed.

Results and Discussion

Influence of Operating Conditions

Different operating conditions such as velocity, normal load and temperature are known to influence friction. This section will briefly explain the influence of these different parameters for a sintered brass base friction material under low sliding velocities and high load; conditions commonly found in limited slip differentials.

Sliding Velocity

The sliding velocity has a significant influence on the observed friction. However, it is not possible to give any general answer as to how friction will change with respect to velocity since this will differ depending on the fluid formulation, the friction material and temperature.

In Figure 14 friction is plotted as a function of velocity with identical test conditions but with slightly different additive combinations; the addition of a surface active extreme pressure additive. From these results it is possible to conclude that friction-velocity characteristics are determined by the choice of friction material and transmission fluid.

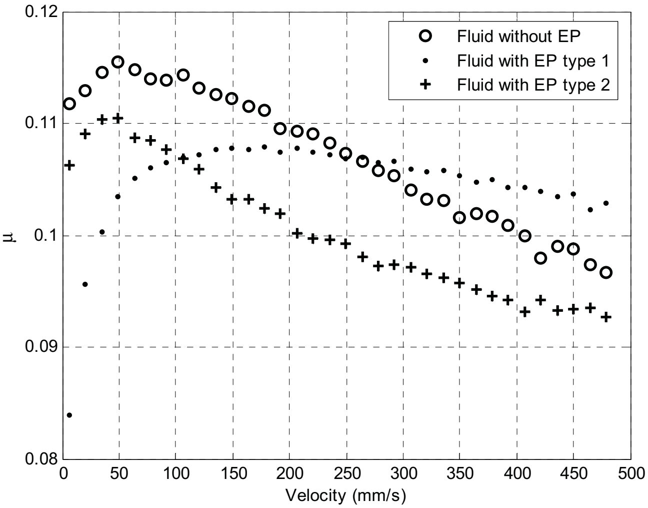

Figure 14: Influence of sliding velocity on friction for three slightly different fluids, 50ºC and 20kN.

Temperature

The contact temperature has a significant influence on friction. Figure 15 shows the result from a test with constant normal force and constant velocity applied. The test commenced at room temperature and the coefficient of friction was monitored at increasing temperatures. It can be seen that the friction decreases around 20% with a temperature rise from 40ºC to 100ºC. Similar behavior can also be seen in Figure 9 and Figure 10 for data obtained from velocity ramps performed at different temperatures.

This behavior is caused by the increasing ease with which tribolayers form as temperature increases. This is due to increased mobility of the additives in combination with higher flash temperatures (i.e. higher activation energy) in the contact. The decreasing friction at higher temperatures is thus the direct result of increasing additive performance, rather than viscous effects. This is supported by the fact that it is possible to alter the friction-temperature behavior by choosing different additives (*Paper F*), whist using different base oils has a much smaller effect (*Paper E*). It can be concluded that friction decreases as temperature increases, and that the rate of decrease in friction can be altered by using different additive combinations.

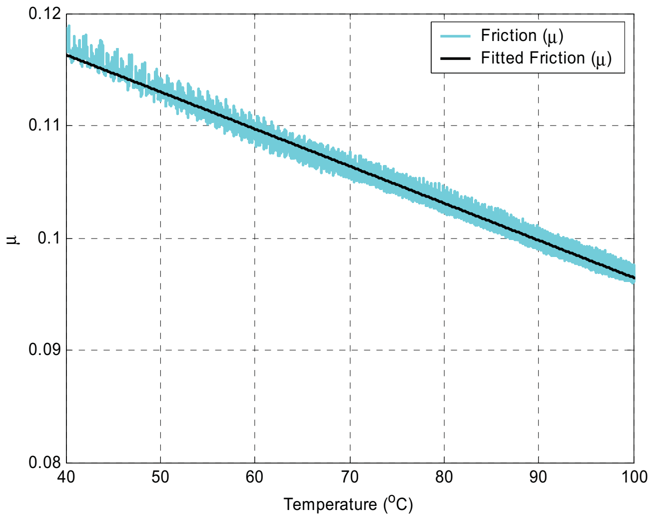

Figure 15: Friction as a function of temperature, obtained from a test with constant velocity and constant force.

Normal Load

The normal load will generally not influence the friction to any large extent (typically in the order of 5%), but in some cases, a slight increase in friction can be observed at higher loads. It is easy to think that the coefficient of friction decreases with increased load, yet the apparent decrease in friction is actually the result of an increase in temperature at the clutch interface. Figure 16 shows temperature compensated friction-normal force curves at different sliding velocities. A slight increase in friction can be observed which can be explained by an increasing real area of contact due to deformation of the friction material. It should also be mentioned that in this case, for all loads, friction increases with increasing velocity, i.e. a low $\mu 1$ and a high $\mu 50$. In other words showing good anti-shudder properties.

At low temperatures and low loads, however, friction might become dependant on normal load as can be seen in Figure 17. This is probably due to insufficient contact energy which inhibits activation of certain additives in combination with a transition between boundary and mixed lubrication. However, further studies are needed in this area. It is also apparent from Figure 17 that the anti-shudder properties at low loads and low temperatures are poor since friction decreases with increasing velocity. It can be concluded that normal load has a minor influence on friction at normal operating temperatures, but that this might not be true at low temperatures.

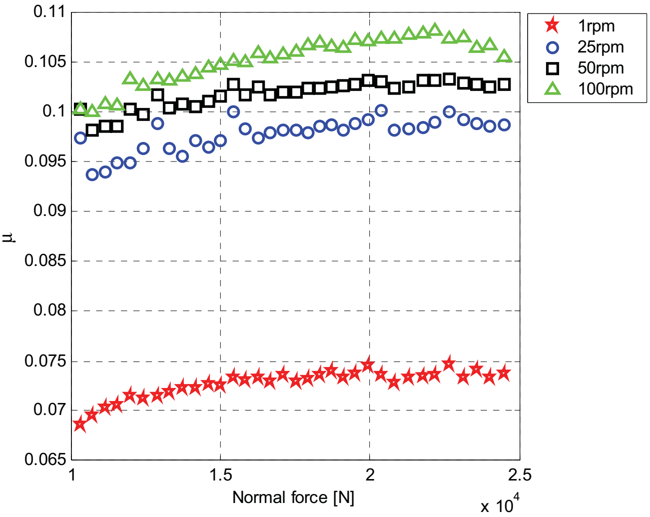

Figure 16: Friction at different normal loads, temperature 50ºC.

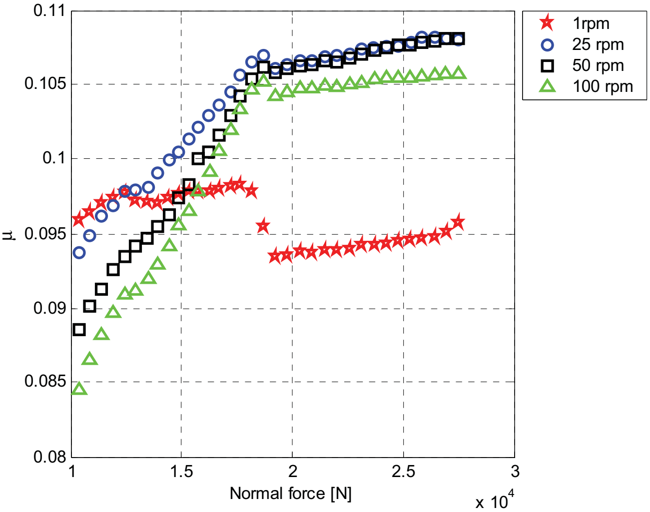

Figure 17: Friction at different normal loads, temperature 20°C.

Lubrication Regime

It is important to know the lubrication conditions in the contact in order to improve the performance of the clutch system. This section summarizes observations on the lubrication conditions in limited slip differentials.

By changing the lubricant in the clutch, friction will change. From these changes it is possible to obtain information indicating the prevalent lubrication regime.

Friction Characteristics

Different Base Oils

The influence of different base oils on friction characteristics can be seen in Figure 18. The friction characteristics are similar in shape for all fluids and only differ around 5% even though the viscosity differs by a factor of around 8 times between the high and low viscosity fluids.

The differences in torque capacity (μ1) are further illustrated in Figure 19. In this case the base oil shows no significant influence on the friction. The dynamic friction (μ50) is presented in Figure 20. Here it can be seen that the base oil has some influence on friction even though the difference is less than 10%.

These observations indicate that hydrodynamic effects are not present at low velocities (μ1). At higher velocities and low temperatures, however, the high viscosity fluids result in slightly lower friction indicating that hydrodynamic effects start to come into play and that a transition from boundary lubrication into mixed lubrication occurs.

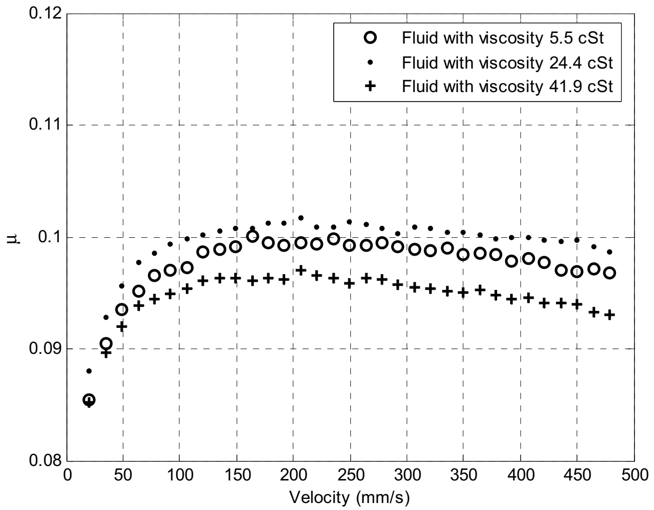

Figure 18: Friction characteristics for fluids with the same additive package but different viscosity. Friction measured at 50ºC and with an applied load of 20kN. Viscosity values are given for 50ºC.

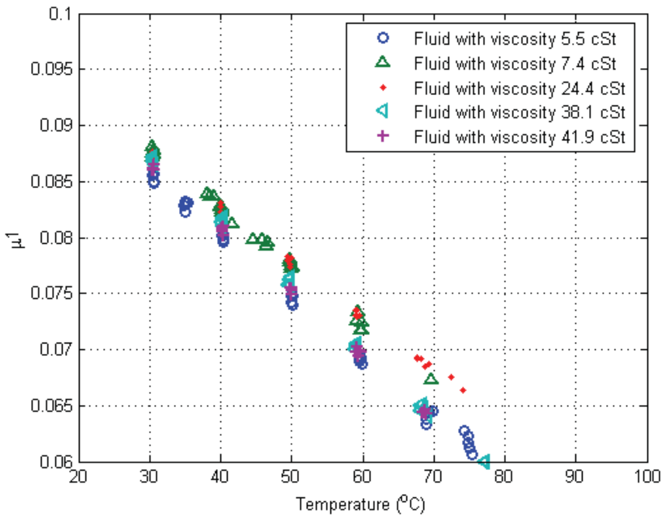

Figure 19: Torque capacity for fluids with different viscosities at 20kN load.

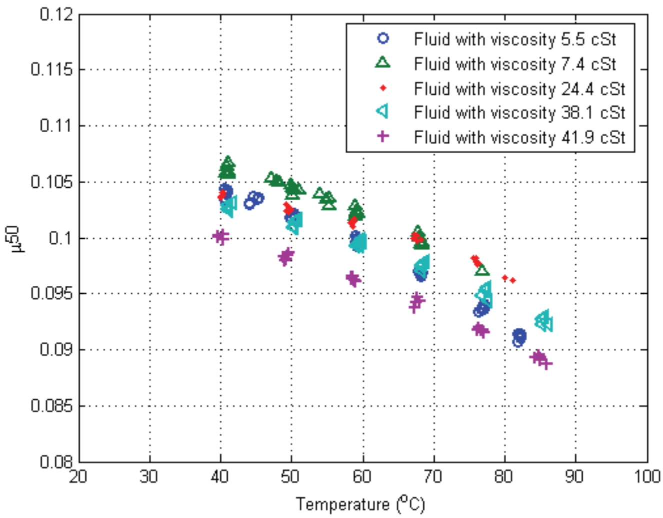

Figure 20: Dynamic friction for fluids with different viscosities at 20kN load.

Different Additives

The influence of different lubricant additives (e.g. corrosion inhibitors) on friction characteristics can be seen in Figure 21. Changes of up to 20% can result from even slight changes to additive composition. Similar behavior can be seen in Figure 14 for extreme pressure additives, where difference in friction of up to 30% can be observed.

The differences in torque capacity (µ1) is further illustrated in Figure 22. In this case different lubricant additives have a significant influence on the friction, especially at higher temperatures. In Figure 23 the dynamic friction (µ50) is presented. In this case the additives still have some influence on friction, especially at higher temperatures, although the differences are less than 10%.

These observations indicate that friction at low velocities is strongly dependant on tribolayers formed by lubricant additives. At higher velocities, however, the effects of additive are less pronounced indicating that hydrodynamic effects start to come into play and suggesting a transition from boundary lubrication into mixed lubrication.

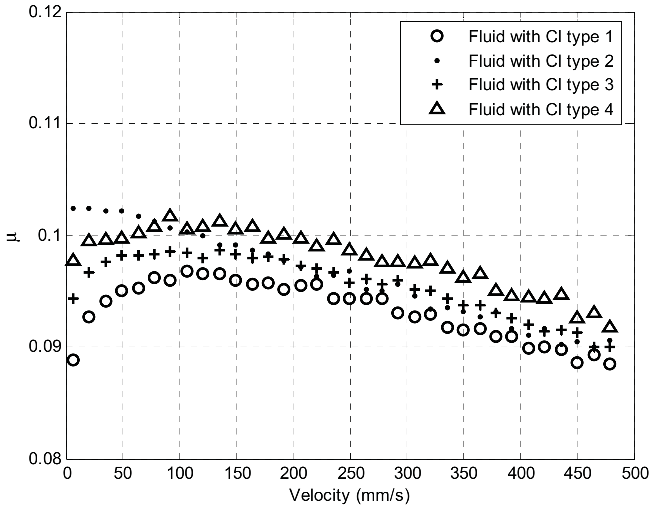

Figure 21: Friction characteristics for fluids with different additive package but the same base oil. Friction measured at 50ºC with an applied load of 20kN.

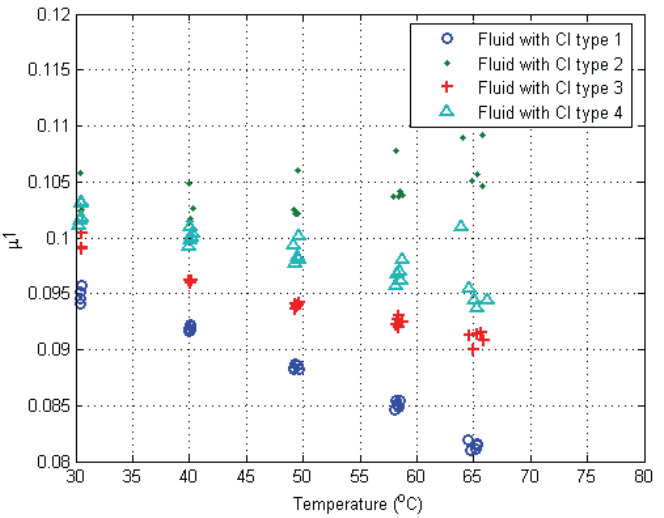

Figure 22: Torque capacity for fluids with different lubricant additives at 20kN load.

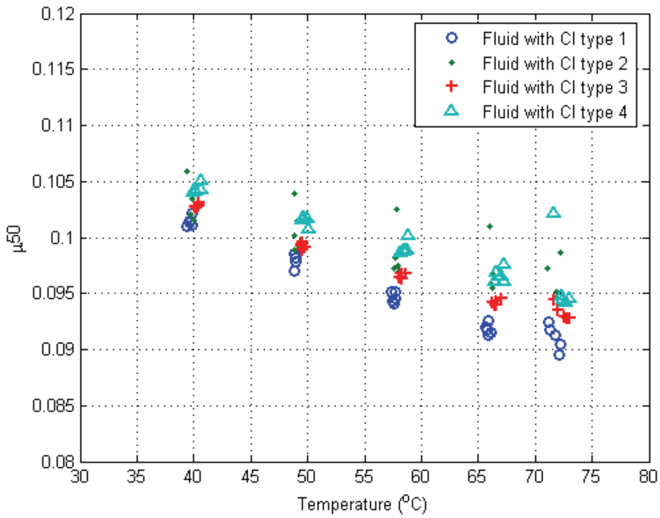

Figure 23: Dynamic friction for fluids with different lubricant additives at 20kN load.

Fluid Formulation

Choice of transmission fluid, as was seen in earlier sections, is vital in order to achieve satisfactory performance in a clutch system. This section will describe the development of a transmission fluid for limited slip differentials. The aim was to combine good anti-shudder properties with the ability to lubricate hypoid gears, which meant that the addition of extreme pressure additives was necessary. From practical experience it is known that extreme pressure additives cannot be combined with commercial wet-clutch lubricants without losing anti-shudder properties. This is especially true with sintered friction materials.

Base Oil

Choice of base oil has only a minor effect on anti-shudder properties as can be seen in Figure 18, Figure 19 and Figure 20. For this reason, a synthetic polyalphaolefin (PAO) base oil was chosen.
PAO has good low temperature performance, good oxidative stability and a high base oil viscosity index making it possible to use less polymer based viscosity index improvers. This will help increase fluid life since shearing of viscosity index improvers is known to be a limiting factor on the fluid life in clutch applications. In addition to this, PAOs are non polar in nature, limiting interaction problems between the base oil and boundary lubricating additives that may otherwise inhibit the surface active additives from reaching the intended adsorption surface.
The kinematic base oil viscosity was 25cSt at 40ºC, and 5cSt at 100ºC.

Single Additives

After the base oil had been chosen, single additive tests were carried out in a reciprocating friction and wear tester, see *Paper F*. Initially, only one additive at the time was added to the base oil and the friction characteristics measured. With only a single additive in the base oil it was not possible to use the Limited Slip Clutch Test Rig due to stick-slip problems.
While some of the additives studied such as antioxidants, dispersants and detergents were found to have only minor influence on friction other, more surface active additives, showed significant influence as can be seen in Figure 24, Figure 25 and Figure 26. However, some exceptions to this have been found. For instance one tested detergent affected friction and some anti-wear and friction modifiers had only minor influence on friction.

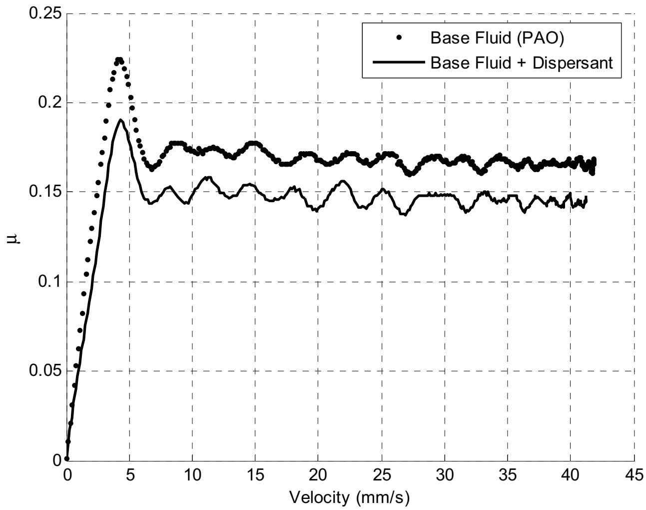

Figure 24: Comparison between base oil and base oil mixed with 1wt% dispersant. Temperature 70°C.

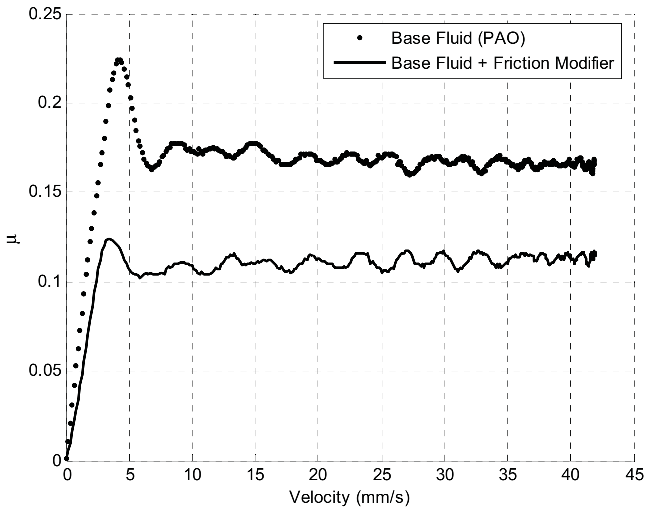

Figure 25: Comparison between base oil and base oil mixed with 1wt% friction modifier. Temperature 70°C.

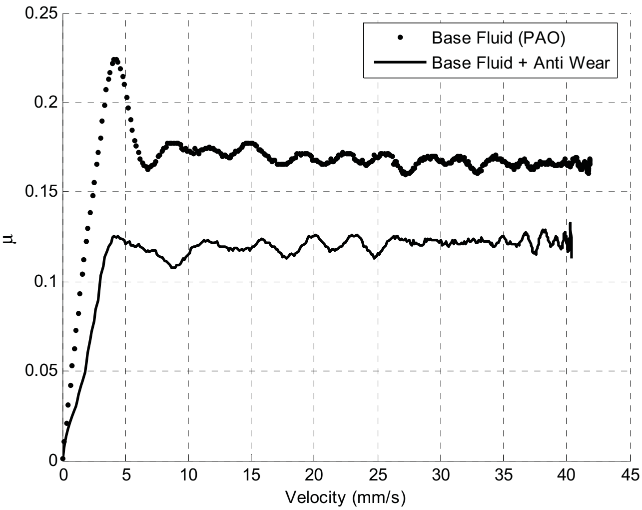

Figure 26: Comparison between base oil and base oil with 1wt% anti wear. Temperature 70°C.

The performance of the additives tested were ranked based on the ratio between the static friction (peak value) and the dynamic friction. A low value is desirable. In addition to this, a high dynamic friction was sought in order to increase the torque transmission capability of the clutch.
The most promising additives were selected for further evaluation in combination with other additives in the next stage of development.
The antioxidant and the detergent did not affect the anti-shudder performance during single additive testing, but are needed in a fully formulated fluid in order to maintain life-time performance.
The friction modifier improved the anti-shudder performance significantly and the best friction modifier from the single additive testing was selected as the basis for the new formulation.

Additive Combinations

The second stage of development evaluated interactions between additives. An initial formulation was defined, consisting of a synthetic base oil with antioxidant, friction modifier and detergent additives.
The evaluation of additive combinations were divided into three steps and at each step one type of additive introduced. Selection of a combination to be developed further was made by comparing the friction-velocity curves and temperature dependence of friction. At the first step, rust and corrosion inhibitors were added. At step two and three, anti wear and extreme pressure additives respectively were added. Further information regarding the combinations tested can be found in *Paper F*. At the end of step three a complete formulation had been obtained.

Complete Formulation

In the final stage of development, the new formulation was compared with a commercial Statoil LSC fluid. In Figure 27 the compensated friction characteristics for the new fluid and the commercial fluid are plotted for a temperature of 50ºC.

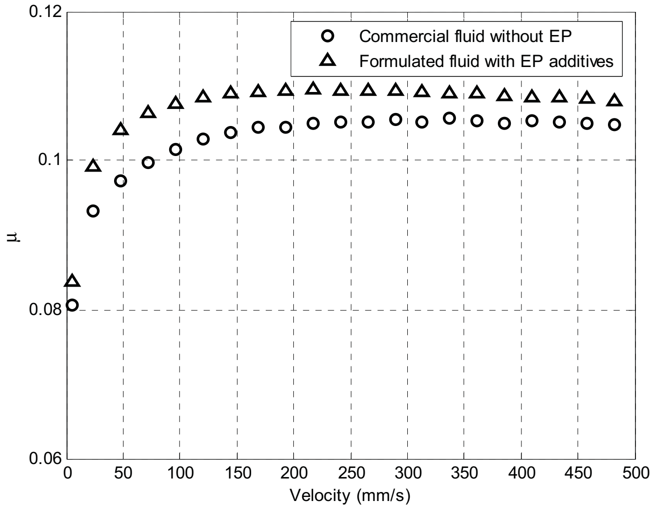

Figure 27: Comparison between the new fluid and a commercial fluid. Load 20kN and temperature 50ºC.

The new fluid generates higher friction compared to the commercial fluid. The friction characteristics (anti-shudder properties) are also good for both fluids even though the formulated fluid shows a slight negative slope at higher velocities.

Figure 28 and Figure 29 show friction as a function of temperature for 1rpm (torque capacity) and 50rpm (dynamic friction) respectively. Both fluids have low temperature dependency at high sliding speeds whilst friction at low velocities is more greatly influenced by temperature. This is to be expected since the additives, as shown earlier, have a strong influence on friction especially at low velocities and the effects of additives (i.e. formation of tribolayers) are strongly temperature dependant.

The new fluid had a greater temperature dependence than the commercial fluid especially at low velocities and at low temperatures. However, both fluids maintained acceptable anti-shudder properties even at temperatures as low as -20ºC.

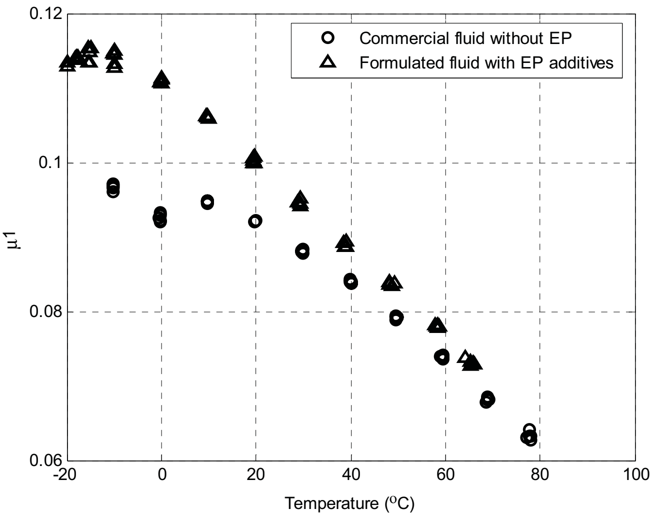

Figure 28: Friction at 1rpm for a commercial fluid and the new fluid. Applied load was 20kN.

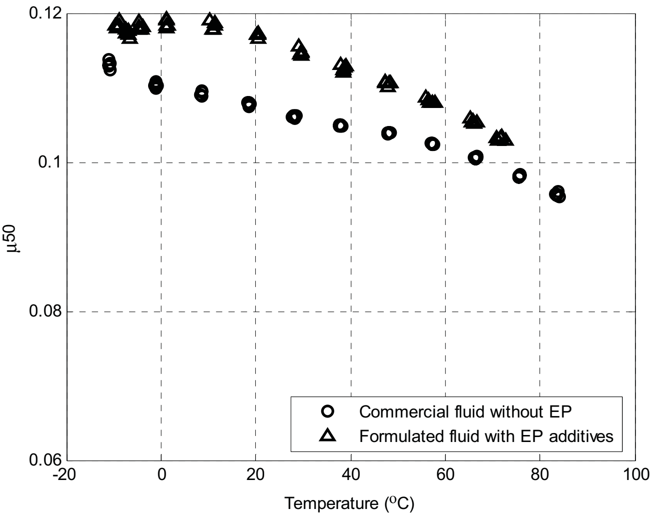

Figure 29: Friction at 50rpm for a commercial fluid and the new fluid. Applied load was 20kN.

While the commercial fluid is a 'fill-for-life' product of proven performance, the behaviour over time of the new fluid needs further improvement.

Temperature Compensation of Measurements

The performance of a transmission fluid is generally described by its friction characteristics, it is therefore important to obtain and present the friction characteristics in a proper and consistent way. This section shows how different conditions during measurements can influence the data obtained and describes a method to present the data which can diminish this influence.

Uncompensated Data

The influence of different measurement conditions on experimental data can be clearly seen in Figure 30. In this case the same friction material and transmission fluid were investigated and the oil sump temperature, normal load and oil flow were the same for both tests. The only difference is that one curve shows measurements made while the rotational speed was increased and the other while the rotational speed was decreased.

From this it is reasonable to believe that the direction of velocity change influences friction. However, it is possible to prove this assumption wrong from knowledge of the interface temperature and applying the temperature compensation method described earlier. This is illustrated in the next paragraph.

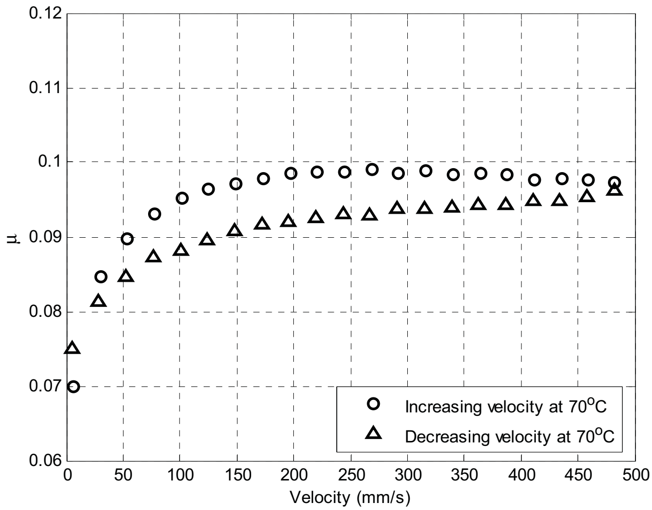

Figure 30: Friction characteristics from a positive and a negative velocity ramp at an oil sump temperature of 70ºC.

Compensated Data

In Figure 31 the measurement data presented in Figure 30 are shown together with the oil sump temperature and interface temperature measured during the test cycle. It is obvious that although the oil sump temperature is constant the interface temperature varies significantly (oil sump volume is 200ml).

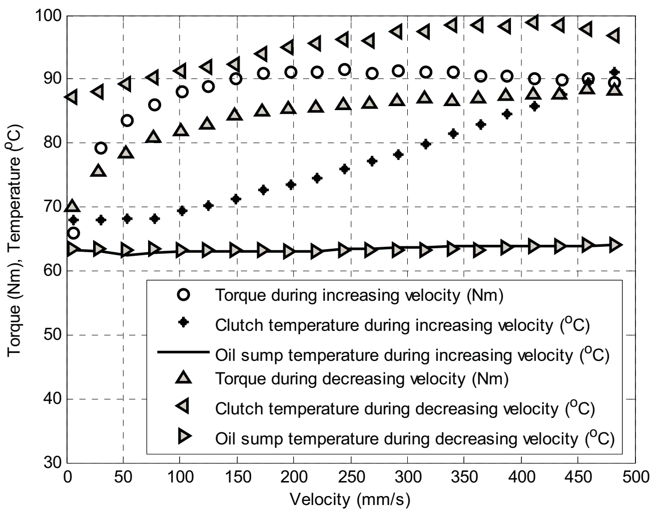

Figure 31: Measurement data from which Figure 30 where obtained.

However, running similar tests at different initial temperatures and by knowing the interface temperature it is possible to apply the temperature compensation method presented earlier in the section on data analysis.

Figure 32 shows compensated friction vs. velocity graphs for the same data as seen in Figure 30. The curves correspond well to each other, indicating that the method developed is indeed an efficient way

to facilitate comparison of friction characteristics obtained under different conditions. As mentioned earlier, the constant temperature curve governs the anti-shudder properties in an actual application since the timescale of shudder initiation is such that the temperature can be considered constant.
In *Paper D,* this method is applied to other data obtained under different operating conditions including differences in normal load, ramp time and initial velocities. In all cases, the method was successful in removing unwanted temperature influence from the results and at the same time emphasises the importance of monitoring the clutch interface temperature.

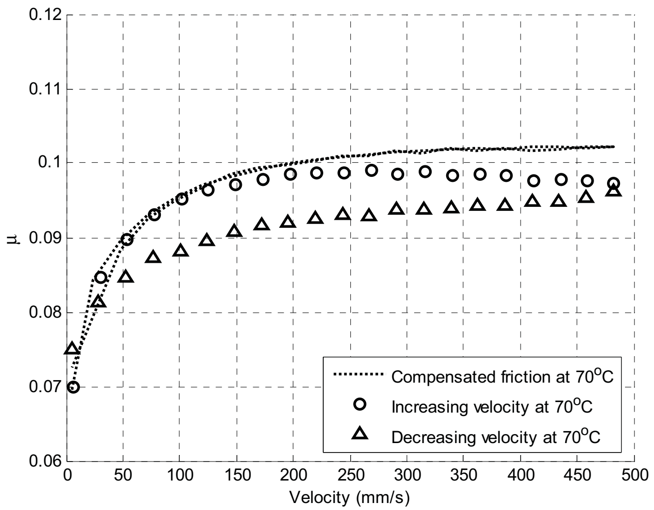

Figure 32: Friction characteristics measured during speed increase and speed decrease, as in Figure 30, and the same two curves compensated for temperature differences during measurements.

Clutch Simulations

Knowing the torque transmission behaviour of a clutch is vital in order to achieve a well functioning transmission system. This section describes how transmitted torque from a limited slip differential can be predicted using a semi-empirical friction model in combination with a temperature model of the clutch.

Friction Model

The boundary friction is modeled as a function of load, velocity and temperature based on an extensive amount of experimental data. The following expression has been curve fitted to this experimental data,

$$\mu = C_1 \tanh(C_2 \cdot v) + C_3 v^{0.1} + C_4 \qquad (1)$$

where μ is friction and v is sliding velocity. C_1 relates to the friction value at the point where the friction starts to level out. C_2 is used to adjust the curve with respect to the x-axis, i.e. velocity and C_3 governs the slope of the curve at higher velocities. C_4 is used in order to adjust the friction level, i.e. shift the friction curve up or down.
Figure 33 shows measured data points and corresponding fitted curve for one arbitrary case; normal load 20kN and temperature 70ºC.

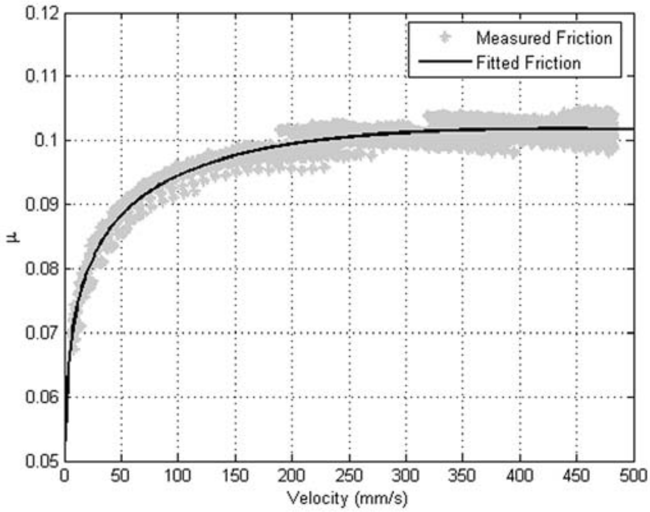

Figure 33: Observed values of friction and the fitted friction model at load 20kN and temperature 70ºC.

The fitting parameters from Equation 1, for different loads and temperatures, are stored in a matrix that spans loads from 10kN to 25kN and temperatures from -40ºC to +200ºC. Friction for any arbitrary set of required conditions can be obtained using linear interpolation between the friction values generated by the parameter sets surrounding the required operating condition. More information on the friction model can be found in *Paper G*.
Figure 34 and Figure 35 show friction values generated by the friction model at arbitrary conditions. It can be seen that at velocities higher than those that can be measured in the Limited Slip Clutch Test Rig, i.e. above 0.6m/s, friction is assumed to be constant. This means that the model should be used with caution at higher sliding velocities.

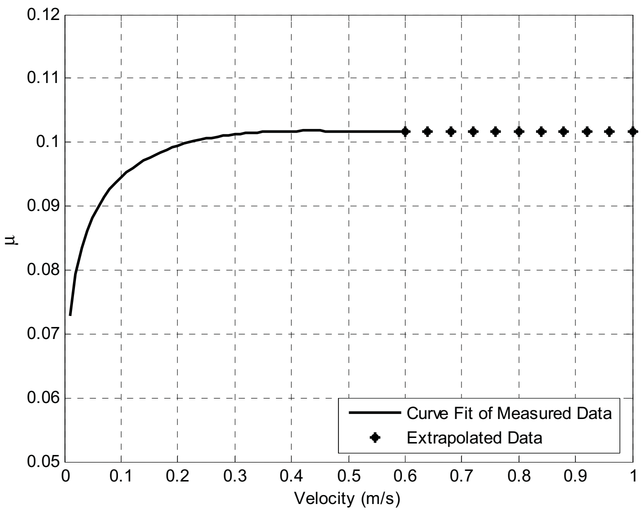

Figure 34: Friction characteristics generated by the friction model at load 20kN and temperature 70ºC.

In Figure 35, the friction values between -20ºC and 100ºC are based on a vast amount of speed ramp tests (several thousand ramps). From -40ºC to -20ºC and 100ºC to 200ºC too few speed ramp tests have been carried out (less than 50 ramps for each of these temperature intervals) and in these regions friction predictions are based on measurements such as those presented in Figure 15 where friction vs. temperature has been measured at different velocities. More information about this can be found in *Paper G*. At temperatures below -40ºC and above 200ºC no measurements have been made and the friction model will return the same value as at -40ºC and 200ºC respectively, which means that the model should not be used at such temperatures.

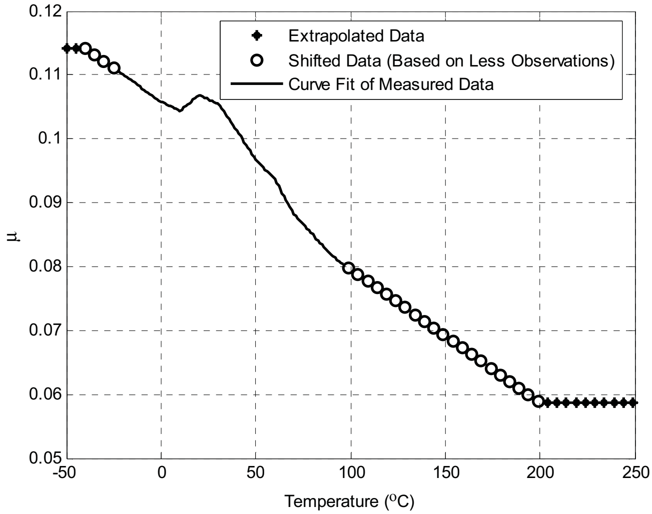

Figure 35: Friction at different temperatures generated by the friction model at a load of 20kN and velocity 0.05m/s.

Temperature Model

The thermal model considers heat dissipation in the fluid as well as heat conduction into the separator disc, friction material, and core disc. The computation domain is the axisymmetric cross section presented in Figure 36. In this figure, the separator and core disc are half the thickness of the actual discs used in the clutch. Since the simulated friction discs are assumed to be located in a clutch consisting of several similar discs, heat conduction over the edges in the axial direction can be neglected due to symmetry.

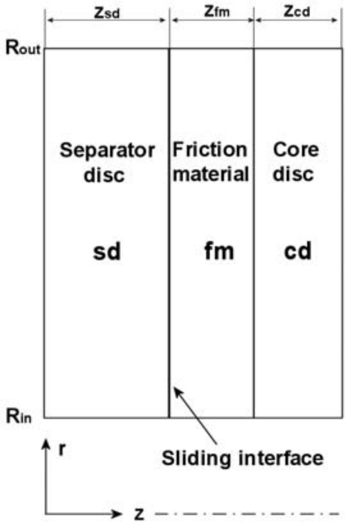

Figure 36: Schematic sketch of calculation domain.

In the separator disc, friction disc and core disc, the temperature is solved with the heat equation in polar coordinates,

$$\rho C_p \frac{\partial T}{\partial t} = k\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{\partial^2 T}{\partial z^2}\right] \qquad (2)$$

where density ρ, specific heat capacity C_p and thermal conductivity k have different values for the different parts of the clutch. The sintered brass friction material is considered impermeable, so there is no heat convection in the friction material.
For more details about the thermal model, see *Paper G*.

Combined Model

Since friction depends strongly on temperature and temperature depends on friction due to frictional heating it is necessary to simultaneously calculate both these quantities in order to accurately predict the torque transmitted by the clutch.

Solution Technique

Engagement force, sliding velocity, and initial temperature are assumed to be known parameters and are used as input to the model. These parameters can be easily measured in a test rig or in a real clutch on-board a vehicle.
The energy equation (2) is discretized on an axisymmetric grid. The friction in each sliding grid point is given by the boundary friction model at each time step based on velocity, pressure and temperature. The generated heat is calculated from the friction and velocity and is used in the thermal model to predict the temperature in the next time step. Based on this temperature, a new friction value is calculated. Finally, the transmitted torque is calculated by integrating the friction force over the clutch area.

Simulation Results

In Figure 37, results from a simulation are compared to measurement data at an arbitrary condition. As can be seen, the agreement between predicted and measured torque is good. The accuracy of the prediction is similar for other load cases. In *Paper G* more results for different cases are presented together with temperature distributions in the clutch.
The same torque curves as shown in Figure 37 are also plotted in Figure 38 together with predicted torque for a case where no temperature change is accounted for. Initially the agreement is good for a

constant temperature approach as well, but after only a few seconds the predicted torque starts to differ substantially from measured data. This error will continue to grow with time diminishing the value of the prediction.

The friction function works for the system it is adapted for, but for other friction materials or transmission fluids it will be necessary to run a quite extensive experimental investigation in order to obtain new and accurate fitting parameters throughout the friction parameter matrix.

One way to reduce this effort would be to assume that the friction-velocity curve is only influenced by temperature and not by normal load. This will significantly reduce the amount of testing necessary whilst still maintaining reasonable accuracy in the prediction.

From this it can be concluded that accurate friction and temperature models are needed in order to predict torque transmission over long engagement times. The developed model has both these elements and hence gives good torque predictions compared to experimental measurements. However, for longer engagement times (more than a minute) or several consecutive engagements the cooling rate of the system must be known in order to be able to maintain the accuracy of the temperature prediction.

The model and the simulation approach used are general and can be applied to any clutch system working under similar operating conditions. In order to do this, boundary friction measurements for the friction material/transmission fluid must be obtained as input to the friction model whilst the temperature model must be adjusted for the geometry and cooling rate to the environment.

The developed simulation model can simplify product development and be of great assistance when tuning the transmission system control software. In the not too distant future, the model, or parts of it, could even be implemented in the clutch control computer in production vehicles.

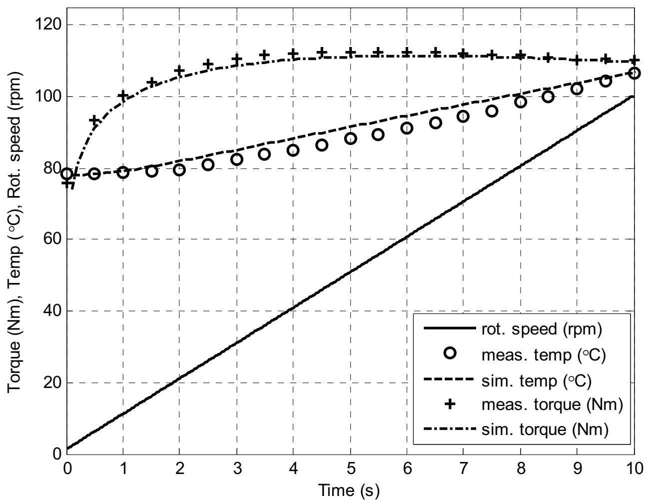

Figure 37: Comparison between measurements and simulation with an axial force of 25.3kN.

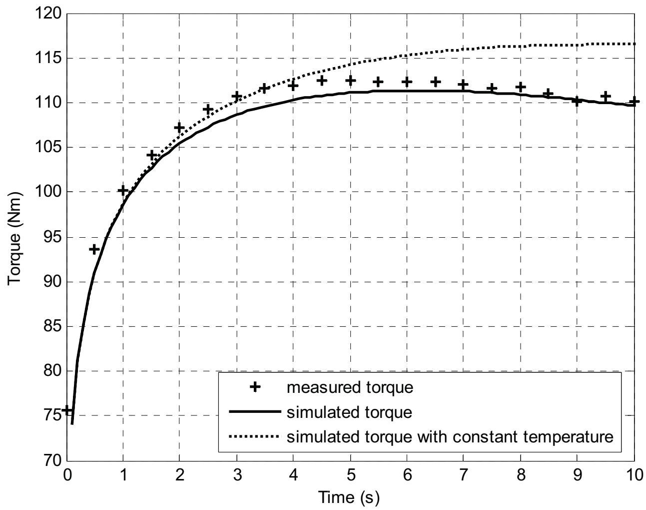

Figure 38: Comparison between torque for constant temperature vs. simulated temperature at a normal force of 25.3kN.

Conclusions

During this work four main questions were to be addressed and can now be answered.

1. Which factors influence torque transmission in the clutch, and what lubrication regime is predominant in the limited slip differentials studied?
   - Sliding velocity has a significant influence on transmitted torque (μ may differ by up to 50% in some cases), but the influence is different for different fluids and must be measured experimentally for each fluid/friction material combination.
   - Normal load has only a minor effect on friction at normal operating temperatures (μ differs by around 5%), but this is not necessarily true at temperatures below room temperature.
   - The lubrication regimes in the investigated applications are boundary lubrication at low velocities, moving into mixed lubrication as the speed increases. This means that it is lubricant additives that govern torque transmission rather than base oil properties.
   - Temperature has a significant influence on torque transmission. Increasing temperatures caused a decrease in friction for all tested fluid, but at differing rates. Friction characteristics (μ-v) are also influenced by temperature, typically worsening anti-shudder properties at lower temperatures.

2. Which methods can be used to measure and compare friction characteristics?
   - Small scale test equipment is useful for giving indications of the performance of different fluids but the results cannot be directly applied to complete clutch assemblies primarily because of thermal effects due to differences in geometries and movement pattern.
   - A special test rig has been designed to measure the performance of different fluids and friction materials under conditions found in limited slip differentials. Data obtained from this agrees favourably with data from test equipment used by other researchers and from test vehicles, which will enable accurate prediction of how well tested fluids will perform in the real world.
   - Monitoring the interface temperature in the clutch is vital in order to accurately asses the torque transmission performance of different samples.
   - A method to compensate measured data for differences in test conditions (primarily temperature) has been developed and shown to work very well.
   - The use of ‘standard’ friction values, i.e. torque capacity (μ1), dynamic friction (μ50) and the ratio μ1/μ50, has been found to be very useful when comparing the friction and anti-shudder properties of different fluids.

3. How can fluids be formulated to give good friction characteristics for the clutch, whilst at the same time allow them to be able to lubricate other machine elements in the transmission?
   - Since the base oil has been shown to have only minor effects on friction characteristics, a non polar base oil with high viscosity index and good thermal properties should be chosen.
   - A successful methodology for the development of formulated fluids has been created. This is a two stage process. Firstly, single additives are tested to asses their frictional performance from which a base package of necessary additives (anti-oxidants and detergents) with limited influence on friction is created in combination with the base oil. Finally, surface active additives (friction modifiers, corrosion inhibitors, anti-wear and extreme pressure) are added one at a time to this base formulation in order to study combination effects of the surface active additives.
   - From observation of the tested additives it can be stated that anti-oxidants, dispersants, detergents and viscosity index improvers generally have a minor influence on friction whilst more surface active constituents such as friction modifiers, anti-wear agents, corrosion inhibitors and extreme pressure additives have a significant affect on friction. However, some exceptions to this have been found. For instance one tested detergent affected friction and some anti-wear and friction modifiers had only minor influence on friction.
   - Interactions between different additives must be considered since these can often completely alter the friction characteristics. This is because additives compete for the same adsorption

surface, hence the addition of one additive may inhibit another from forming tribolayers. Extreme pressure additives have been found to be especially difficult to combine with other additives whilst maintaining good anti-shudder properties.

- The formulated fluids developed have demonstrated that it is possible to combine good anti-shudder properties in the clutch with good lubrication performance with other machine elements (i.e. hypoid gears).

4. How can the torque transmission characteristics of the differential be predicted using simulation models?

- Thermal effects have a significant influence on the torque transferred by a differential under limited slip conditions. It is therefore necessary to use a thermal model of the clutch in combination with a temperature dependant boundary friction model.
- The boundary friction model must be based on empirical friction data obtained for the friction material/transmission fluid combination of interest.
- The contribution of hydrodynamic effects to the transferred torque is small under limited slip conditions and can be neglected without influencing the accuracy of the torque prediction.
- The model developed has shown that it is possible to accurately determine the transferred torque knowing the current operating conditions and the thermal history of the clutch, and given that the boundary friction model takes clutch temperature into account.

Future Work

Although this thesis has answered many questions there remains much work to do in this field. Areas that deserve further investigation include:

- For the limited slip differential investigated, the friction characteristics are well understood at normal operating temperatures. However, at low temperatures there are several effects which remain unexplained; although they have been touched upon in this work (e.g. influence of normal load on friction).
- It has been shown that torque transmission is governed primarily by additive effects. How different additives perform is generally assessed by measuring the friction. Gaining a better understanding of the influence of chemical composition and structure of lubricants and additives on friction would open up new opportunities for enhancing fluid performance with less testing.
- The simulation model developed can simplify product development and will be of great assistance when tuning vehicle transmission system control software. The model, or parts of it, could even be implemented in the clutch control software in production vehicles. Further work in this area is strongly advised.
- The feasibility of using friction data obtained from small scale test equipment for the friction function in torque transmission simulations should be investigated since this will reduce the experimental effort necessary when changing fluid or friction material. The thermal model could also be further enhanced by developing improved models for the cooling oil flow and groove effects.

Enclosed Papers

Paper A; New demands driving new technology; A literature review of research into the behaviour and performance of wet clutches.

Mäki R., “New demands driving new technology; A literature review of research into the behaviour and performance of wet clutches”. To be submitted for publication.
This paper reviews published research concerning wet clutches in applications such as automatic transmissions and limited slip differentials.
The literature review and writing the paper was done by Rikard Mäki.

Paper B; Limited Slip Wet Clutch Transmission Fluid for AWD Differentials; Part 1: System Requirements and Evaluation Methods.

Mäki R., Ganemi B., Olsson R. and Lundström B., “Limited Slip Wet Clutch Transmission Fluid for AWD Differentials; Part 1: System Requirements and Evaluation Methods”. SAE Technical Papers, Paper number: SAE 2003-01-1980 / JSAE 20030119, Presented at SAE/JSAE Spring Fuel & Lubricants Meeting, Yokohama, May 2003.
This paper describes the design and function of the Limited Slip Clutch Test Rig. Typical measurement data from the equipment is also presented and discussed.
The design and building of the test equipment was done by Rikard Mäki, who also conducted the experiments and wrote most of the article. Other authors assessed performance requirements and contributed with feedback and ideas during the design stage.

Paper C; Wet Clutch Tribology - Friction Characteristics in All-Wheel Drive Differentials.

Mäki R., “Wet Clutch Tribology - Friction Characteristics in All-Wheel Drive Differentials”. Tribologia - Finnish Journal of Tribology, 2003.
This paper is a condensed version of Rikard Mäki’s Licentiate thesis. The paper describes test equipment used and results obtained; mainly on the influence of different operating conditions on friction. In addition, some material concerning the influence of lubricant additives on friction is also presented.
The experimental work was conducted by Rikard Mäki with the exception of the additive friction measurements which were performed by Kent Ekholm. The article was written by Rikard Mäki.

Paper D; Measurement and Characterization of Anti-Shudder Properties in Wet Clutch Applications.

Mäki R., Nyman P., Olsson R. and Ganemi B., “Measurement and Characterization of Anti-Shudder Properties in Wet Clutch Applications”. SAE Technical Papers, Paper number: SAE 2005-01-0878. Presented at SAE World Congress, Detroit, April 2005.
This paper describes how measured friction is influenced by the measurement conditions/test cycle used. A novel method to present measured friction data in such a way that the influence of test conditions is eliminated, is outlined.
The original ideas behind the study and the experiments to be conducted were formulated by Rikard Mäki who also did most of the writing of the paper. Experimental work and data analysis were performed jointly by Rikard Mäki and Pär Nyman.

Paper E; Wet Clutch Transmission Fluid for AWD Differentials; Base Fluid Influence on Friction Characteristics.

Mäki R., Ganemi B. and Olsson R., ”Wet Clutch Transmission Fluid for AWD Differentials; Base Fluid Influence on Friction Characteristics”. Accepted for publication in Tribotest.

In this paper the influence of different base oils on friction is presented. Several different base oil types and base oil viscosities were tested. It was found that the base oil has virtually no influence on torque capacity and only minor effects on the dynamic friction.
The fluids tested were formulated by Bager Ganemi. Experimental work, data analysis as well as writing of the paper was done by Rikard Mäki.

Paper F; Wet Clutch Transmission Fluid for AWD Differentials; Influence of Lubricant Additives on Friction Characteristics.

Mäki R., Ganemi B., Triviño Flores R. and Olsson R., ”Wet Clutch Transmission Fluid for AWD Differentials; Influence of Lubricant Additives on Friction Characteristics”. Accepted for presentation and publication in the proceedings of the 15th International Colloquium Tribology, Esslingen, January 2006.
In this paper, the influence of different additives on friction are investigated. Single additives as well as additive combinations were tested in order to arrive at a fully formulated transmission fluid with good anti-shudder properties and good lubricant properties with machine elements such as hypoid gears.
The fluids tested were formulated by Bager Ganemi. Experimental work was done by Kent Ekholm (single additive testing) and Rafel Triviño Flores (additive combination testing) under the supervision of Rikard Mäki. The article was written by Rikard Mäki.

Paper G; Thermal Influence on Torque Transfer of Wet Clutches in Limited Slip Differential Applications.

Marklund P., Mäki R., Larsson R., Höglund E., Khonsari M.M. and Jang J., ”Thermal Influence on Torque Transfer of Wet Clutches in Limited Slip Differential Applications”. Submitted for publication in Tribology International.
This paper presents a model for predicting the transmitted torque for wet clutches operating at low sliding velocities. A semi-empirical friction model was used in conjunction with a temperature model. The results of the simulations gave torque predictions in good agreement with observed values.
Engagement simulations were done by Rikard Mäki based on Fortran code written by Joonyoung Jang. Experimental work and development of the friction model were primarily done by Rikard Mäki, whilst Pär Marklund did most of the work on the temperature model. The paper was written jointly by Rikard Mäki and Pär Marklund. Other authors supervised the work and made valuable contributions to the model.

Other Papers Published During Work on the Thesis

Apparatus for Measurement of Friction Surface Temperature in a Wet Clutch

Mäki, R. “Apparatus for Measurement of Friction Surface Temperature in a Wet Clutch”. Proceedings of the 2nd World Tribology Congress, Vienna, September 2001.
This paper describes modification of the Wet Clutch Test Rig, designed by Mikael Holgerson, necessary to make it possible to study the application at hand. Initial results, mainly dealing with the influence of temperature on friction characteristics are also presented.
The work was performed by Rikard Mäki under the supervision of Richard Olsson and Jan Lundberg. The paper was written by Rikard Mäki.

Wet Clutch Transmission Fluid, Development Method

Mäki R., Ganemi B. and Olsson R., “Wet Clutch Transmission Fluid, Development Method”. Proceedings of the 10th Nordic Symposium on Tribology (NordTrib), Stockholm, June 2002.
This paper describes how different test equipment can be used in combination to evaluate a number of important performance aspects of a lubricant.

The experimental work on the Wet Clutch Test Rig was performed by Rikard Mäki. Experimental work on the other equipment was performed by Richard Olsson et al. at Haldex. The text was written by Rikard Mäki and Bager Ganemi who also formulated the lubricants evaluated.

Limited Slip Wet Clutch Transmission Fluid for AWD Differentials; Part 2: Fluid Development and Verification

Ganemi B., Mäki R., Ekholm K., Olsson R. and Lundström B., "Limited Slip Wet Clutch Transmission Fluid for AWD Differentials; Part 2: Fluid Development and Verification". SAE Technical Papers, Paper number: SAE 2003-01-1981 / JSAE 20030121, Presented at SAE/JSAE Spring Fuel & Lubricants Meeting, Yokohama, May 2003.
This paper describes the effects of the thermal properties of the base oil on clutch temperature and also screening of different additives with regard to their effect on friction using a reciprocating friction and wear tester.
Experimental work on the reciprocating friction and wear tester was performed by Kent Ekholm under the supervision of Rikard Mäki and Bager Ganemi. Experimental work on the clutch tester was performed by Rikard Mäki. Most of the text was written by Bager Ganemi, with some parts being written by Rikard Mäki.

Wet Clutch Transmission Fluid for AWD Differentials; Base Fluid Influence on Friction Characteristics

Mäki R., Ganemi B. and Olsson R., "Wet Clutch Transmission Fluid for AWD Differentials; Base Fluid Influence on Friction Characteristics". Proceedings of the 14th International Colloquium Tribology, Esslingen, January 2004.
In this paper, the influence of different base oils on friction are investigated. Different base oil types as well as different base oil viscosities were tested. It was found that the base oil has virtually no influence on torque capacity and only a minor effect on the dynamic friction.
Tested fluids were formulated by Bager Ganemi. Experimental work, data analysis as well as writing of the paper were done by Rikard Mäki.

Influence of Surface Topography and Composition of Sintered Friction Material on Friction Characteristics in Wet Clutch Applications

Mäki R., Nyman P., Ganemi B. and Olsson R., "Influence of Surface Topography and Composition of Sintered Friction Material on Friction Characteristics in Wet Clutch Applications". Proceedings of the 11th Nordic Symposium on Tribology (NordTrib), Tromsö-Harstad-Bodö, June 2004.
This paper deals with changes in friction during the lifetime of a clutch and correlates this change to changes in surface topography of the sintered friction material.
Experimental work and writing of the paper were done jointly by Rikard Mäki and Pär Nyman. Pär Nyman performed the surface topography measurements in the topometer.

Influence of Surface Topography on Friction Characteristics in Wet Clutch Applications

Nyman P., Mäki R., Ganemi B. and Olsson R., "Influence of Surface Topography on Friction Characteristics in Wet Clutch Applications", Accepted for publication in Wear.
This paper deals with changes in friction during the lifetime of a clutch and correlates this change to the change in the surface topography of the sintered friction material.
Experimental work and writing of the paper were done jointly by Rikard Mäki and Pär Nyman. Pär Nyman performed the surface topography measurements in the topometer.

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Paper A

New demands driving new technology; A literature review of research into the behaviour and performance of wet clutches

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Abstract

The aim of this review is to outline the state-of-the-art in the field of research into the behaviour and performance of wet clutches. The review is based on a comprehensive literature survey and deals with subjects such as clutch design and applications, clutch engagement and anti-shudder properties, temperature, durability, friction materials, transmission fluids and experimental test equipment. Both experimental and theoretical findings are reported.

Introduction

New transmissions and gearboxes must cope with increases in transmitted power and offer improved efficiency and NVH performance (noise, vibration and harshness), extended service intervals, superior shift quality and at the same time be less expensive than their predecessors.
Current trends to achieve these demands involve the increased use of computing power to control the transmission and the optimisation of power transmission for efficiency, improved handling and safety and fun to drive experiences for the driver. As a result of this the number of clutches in a modern passenger car have increased significantly largely due to the way clutches can be controlled electronically in order to distribute drive torque.

Function/Design

Basic Design

A wet clutch is basically a clutch working under lubricated conditions. The configuration most commonly used is the multiple disc wet clutch shown in Figure 1. The multiple disc wet clutch consists of friction discs attached to one shaft by splines. Separator discs are similarly connected to the other shaft. When disengaged, the clutch transmits only a small drag torque due to viscous friction and both shafts are free to rotate independently. The clutch is engaged by applying a normal force, from the hydraulic cylinder[1], clamping the friction and separator discs together and thereby allowing transmission of torque between the shafts.

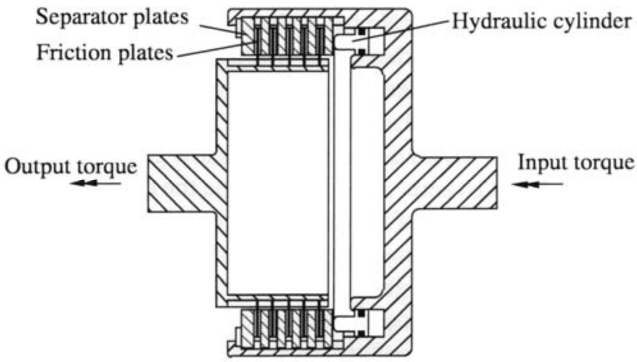

Figure 1: Multiple disc wet clutch [1].

[1] Although hydraulic actuation is most common, both mechanical and electro mechanical actuators are also used.

The clutch configuration shown in Figure 1 is the most common. This double-sided design has friction discs comprising of a steel core with friction material bonded to each side and separator discs made of steel only. In order to improve cooling and lubrication of the clutch, grooves on the surface of the friction material are generally introduced. An alternative to this design is the single-sided design where each disc in the clutch is composed of a steel core with friction material bonded to one side. The advantage of the single-side design is more efficient utilisation of the thermal capacity of the clutch, since all the steel in the clutch discs is heated. In the double-sided design the friction material acts as a thermal insulator preventing efficient heating of the core discs. This means that the single-sided clutch can be thinner or allow a higher torque capacity compared to double-sided designs of the same size. The main drawback of single-sided designs is potential failure due to disc coning [2].

Lubrication regimes

The mechanisms of torque transmission in wet clutches have been studied by many researchers both experimentally and theoretically.

The disengaged rotating clutch operates in full film lubrication, transmitting only a small torque due to viscous shearing of the fluid between the discs, referred to as the drag torque [3, 4].

When the clutch is engaged, the first stage is typically referred to as the hydrodynamic squeeze or squeeze film phase when the clutch discs approach each other, squeezing the oil out of the contact and creating a hydrodynamic squeeze pressure supporting the load and preventing asperity contacts [5-7]. The next stage is called the squash film or mixed asperity contact phase and begins when physical contact between asperities on the discs is made, deforming the porous friction material and squashing fluid out of the material [5-7] In this phase torque is transmitted by both asperity contact friction and viscous friction, the load being supported by both the asperity contacts and hydrodynamic film pressure. The third and final stage of engagement is referred to as the adhesive or consolidating contact phase where the hydrodynamic pressure no longer plays a significant role since the clutch has reached a boundary lubrication condition [5-7]. In this third stage the torque is transmitted by boundary layers formed by additives in the transmission fluid. During this stage, the influence of the base fluid is small compared to that of additives and the properties of the friction material [8, 9].

Another factor influencing torque transmission in the clutch is the presence of cavitation zones between the discs, resulting from fluid shortage or low pressures due to wedge effects [4, 10-12].

Applications

Automatic Transmissions

The most frequent application for wet clutches is without doubt in automatic transmissions (ATs) for passenger cars where wet clutches and band brakes are used to engage the different gears [13]. In recent years virtually all ATs have also been equipped with a lock-up clutch[2] that increases the efficiency of the transmission by locking the hydrodynamic torque converter when it is not needed. Much research has been focused on improving the performance of lock-up clutches [12, 14-18] and optimising the engagement of clutches during shifting [19-24]. New gearbox designs providing better efficiency and performance compared to traditional planetary type ATs have emerged on the market, typically offering performance comparable to manual transmission designs. Many variation on clutch based transmissions can be found including dual clutch transmissions where two wet clutches are employed to provide gear changes without any torque interruption [25]. Another concept is to replace the torque converter in an AT with a “starting clutch” in order to improve vehicle fuel economy. The main challenge with this concept is the control of the starting clutch to ensure acceptable vehicle drivability [26, 27].

Continuously Variable Transmissions

Continuously Variable Transmissions (CVTs) have started to replace traditional ATs, especially in the Asian market. Instead the 4 to 7 gears typically found in an AT, CVT’s offer continuously changing gear ratios and thereby allow the engine to remain at peak power or maximum efficiency at all times. Although CVT’s generally use a belt or toroidal drive, wet clutches are still needed, for example, as starting clutches. This has

[2] There are several different types of lock-up clutches. Traditional ‘lock-up’ torque converter clutches (TCC) rigidly lock the engine crank shaft and turbine shaft. More modern slip-controlled TCCs (or continuous slip TCCs) allow a controlled slip between the input and output axles, transmitting less vibrations and making earlier lock-up clutch engagement possible improving comfort and efficiency.

given rise to new demands on clutch systems as the fluid is required to provide high friction for the traction drive whilst at the same time lubricate the clutch where high static friction make good anti-shudder properties hard to achieve [28-31].

Limited Slip Differentials

In limited slip differentials (LSDs) wet clutches are used in order to lock a differential. These have been around for a long time. In the sixties LSDs were primarily used to provide traction on slippery surfaces and mud [32].
As more powerful vehicles emerged the demands placed on LSD's regarding traction, handling and controllability have increased. Advanced electronically controlled differentials providing optimum torque split between the wheels have been used in high performance cars since the early nineties [33].
In all-wheel drive cars wet clutch type LSDs are also used to provide controllable torque split between front and rear axles to improve traction and handling [34-36].

Construction Equipment

Wet clutches are commonly used in various types of construction equipment in gearboxes and differentials and as wet brakes [37, 38].

Torque Transmission; Clutch Engagement

Achieving good gear shift quality in ATs has been an important issue for transmission and vehicle designers and manufacturers for many decades. An idealised clutch engagement where a rotating inertia is stopped by engaging a clutch can be seen in Figure 2. The time scale of an engagement is typically in the order of one second.
Three different coefficient of friction are generally defined (Figure 2) to describe the engagement; First the initial friction, μ_i, at start of engagement which is influenced mainly by the ease with which fluid is squeezed from between the discs, i.e. geometry, friction material permeability and grooves. The dynamic friction, μ_d, at the mid-point of clutch engagement is a combination of dynamic asperity contact friction and viscous friction. The coefficient of friction at end of engagement, μ_0, is influenced mainly by the asperity friction generated by the boundary layers formed by lubricant additives absorbed on the friction material and separator disc surface[3].

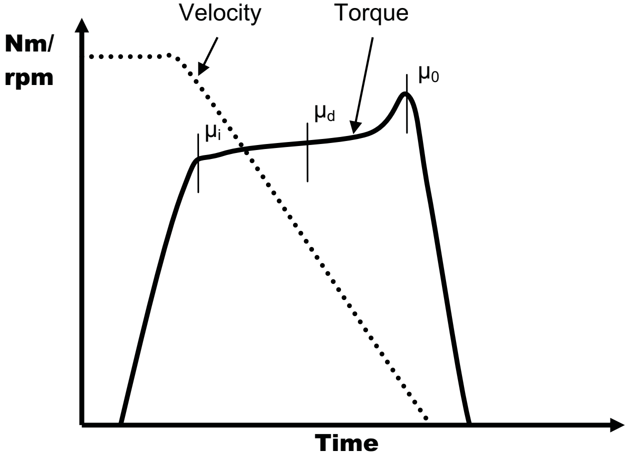

Figure 2: Schematic torque response during engagement.

Many studies on shift quality have been reported, mostly using the SAE #2 test machine [23, 24, 39], but also with other test equipment [20, 21]. It is also possible to improve the smoothness of the gear shift by control of engagement force or by removing the drive torque during a part of the shifting [19].

[3] It is important to note that the friction at end of engagement, μ_0, is not the same as the static breakaway friction or torque capacity generally referred to as μ_s.

Simulation of Engagements

The common simulation approach is to study an engagement from a high velocity to a state of lock-up. Typically Reynolds equation is used to predict the torque and film thickness at the beginning of the engagement and then a measured friction characteristic is applied for the boundary friction during the later part of the engagement. In order to obtain good agreement with experimental data in the hydrodynamic film phase, it is necessary to include an appropriate thermal compensation for changes in fluid viscosity during the engagement. Boundary friction, however, is generally not compensated for temperature. The following section will highlight a few contributions in this area.

In 1973 Wu presented an engagement model for un-grooved but porous wet clutch plates. A modified Reynolds equation was used for the fluid film and the Laplace equation used for the flow in the porous region. A one-dimensional conduction based heat transfer model was used to adjust viscosity and boundary friction. No mixed lubrication was considered, only full film until a critical film thickness had been reached when a boundary model was used instead. The results presented were, however, in poor agreement with experimental observations [40].

In 1975 Ting presented a three stage model comprising a squeeze film phase, a mixed asperity contact phase and a consolidating contact phase. Results for the mixed contact phase were not presented and the results only dealt with oil film thickness and squeeze film pressure and not transmitted torque [6, 7].

In 1994 Natsumeda and Miyoshi presented an engagement model utilising Reynolds equation corrected for surface topography according to the method proposed by Patir and Cheng in 1978 [41]. Their model assumes Darcian fluid flow in the porous friction material, and viscosity was compensated for temperature using predictions from a heat conduction model. The results gave quite a good representation of an engagement and the influence of permeability on engagement characteristics were well explained [22, 42].

In 1996 Berger et al. presented a model similar to Natsumeda's but with the addition of the velocity slip condition suggested by Beavers and Joseph [43] as well as groove effects and asperity load sharing. The model did not, however, include any thermal effects and the boundary friction used varied only as a function of velocity. The results were not compared to experimental data [44].

Yang et al. presented an approximate solution to the clutch engagement problem in 1998. This model was developed from the model presented by Berger et al., but with a thermal model adjusting the fluid viscosity and correction of the flow factors [45]. The torque response predictions showed good agreement with experimental data.

The use of thermal models was further refined in 1999 by Jang and Khonsari who used the same approach as Berger and Yang but with a comprehensive thermal model for predicting the distribution of heat in the clutch. The temperature distribution obtained was used to both adjust the fluid viscosity and to predict the onset of thermo elastic instabilities [46, 47].

In 2002 Gao et al. proposed that the use of a Weibull density distribution would give a better representation of the topography of paper based friction materials compared to the Gaussian topography models used in previous engagement simulation attempts [48, 49].

Torque Transmission; Anti-Shudder Properties

A problem area in most wet clutches working at low velocity is the occurrence of shudder caused by friction induced vibrations or stick-slip in the clutch. The anti-shudder performance of automatic transmissions has been of primary interest in new designs featuring lock-up and continuous slip torque converter clutches [18, 50].

Shudder Modes

Generally stick-slip occurs at low velocities when the static coefficient of friction is higher than the dynamic coefficient of friction; the difference in friction makes the surfaces stick to one another thus causing an uneven sliding velocity. Friction induced vibrations are seen at higher sliding velocities, induced by a negative slope of the friction vs. velocity curve (refer to next section) [12, 51-53].

Shudder Caused by Friction Characteristics; Friction vs. Velocity

The influence of the friction vs. velocity (μ-v) curve on the anti-shudder performance of wet clutches is well established [12, 18, 24, 51-54]. The advantage of a positive μ-v relationship can easily be shown mathematically from engineering vibration calculations [16, 17, 55].

Typical μ-v curves for different automatic transmission fluids (ATF) are shown in Figure 3. In order to avoid vibrations the μ-v relationship should have a low static coefficient of friction (μ_s) and a dynamic coefficient of friction (μ_d) that increases as the sliding velocity increases. Oil A will suppress vibrations, while Oil B and Oil C may bring about self-induced vibrations since they present a negative slope in some regions.
More information on the various factors influencing friction characteristics is found later in this review in the section on friction.

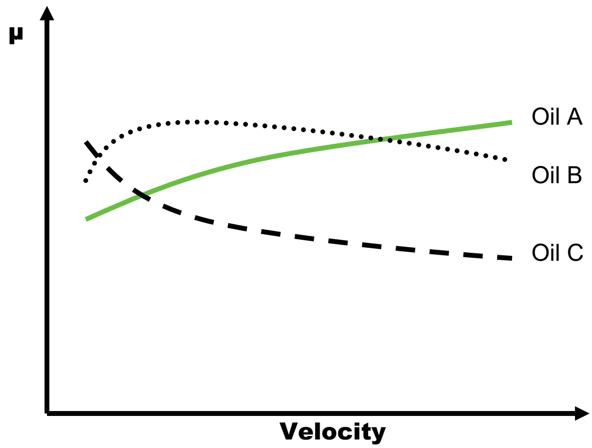

Figure 3: Schematic μ-v relationships for automatic transmission fluids.

Other Factors Influencing Shudder

Several theoretical studies of stick-slip and the influence of damping, sliding speed and excitations in the system have been presented.
Berger et al. concluded that the combination of velocity-dependant friction and a harmonically varying normal force (resulting from, for instance, a rough surface) might produce oscillations [56].
Cameron et al. modelled a vehicle transmission and showed that a μ-v curve with positive slope is indeed an advantage, but neither a necessary nor sufficient condition to guarantee stability. It was concluded that factors such as level of friction, engine speed and engine torque-speed slope also affect the dynamic behaviour of the system [50]. In a later study by the same authors it was found that fluid related factors such as level of friction, friction-velocity characteristics and friction-pressure characteristics must be optimised in order to improve clutch performance [14].
Gao et al. presented a comprehensive model with five equations describing stick-slip, taking into account stiffness, sliding speed, damping and most importantly the time dependent static friction. The predictions presented were consistent with a wide range of observations, and the equations suggest practical methods of reducing or eliminating stick-slip for a general system [57].
Van De Velde et al. investigated stick-slip originating from the slip phase. It was concluded that the friction-velocity gradient determines the occurrence of stick-slip, but also that the friction-velocity gradient must be derived from velocity changes with the same time-constant as the mechanical system in order to be useful when modelling stick-slip [58].

Torque Transmission; Friction

The friction of the wet clutch is the most important parameter to be controlled in order to achieve the desired characteristics for a clutch system. Generally, high torque capacity (high friction) and good anti-shudder properties are desirable, but these two requirements are not easily combined [59]. For clutch applications in general, friction is modelled as a function of velocity only, although a few attempts have been made to present data in Stribeck diagrams [60-62].

Static Coefficient of Friction

The static coefficient of friction (μ_s) should be kept low to give good anti-shudder properties (as described in the previous paragraph), however, a low static coefficient of friction will limit the maximum torque capacity of the fully engaged clutch. The value of μ_s is dependant on oil additives, surface pressure and groove type [63].

The static coefficient of friction will also depend on the stick-time, or the time the surfaces have been immobile. Generally the friction increases with increasing stick-time [64, 65]. Gao et al. showed that the increase of μ_s with stick-time has a crucial influence on the stick-slip behaviour of the system [57].

Dynamic Coefficient of Friction

The dynamic friction (μ_d) is dependant on the respective contribution of boundary and hydrodynamic lubrication between the friction surfaces [9]. The hydrodynamic contribution is influenced by several different parameters such as fluid viscosity, shearing of the fluid film and fluid temperature. The boundary contribution will depend on surface active additives, flash temperature and contact pressure. Additives, including detergents and dispersants can increase the dynamic friction [66].

Dynamic friction is also influenced by the apparent elastic modulus of the friction material as demonstrated by Ohkawa et al., a low modulus of elasticity results in high dynamic friction while factors such as material composition, porosity and lubricant additives have less effect at higher sliding speeds [67].

Friction Characteristics

The friction characteristics are dependant on the friction material, the mating surface and the lubricant present. In addition, operating conditions such as temperature, sliding velocity and applied load also influence the frictional behaviour [53].

Kato et al. showed that increasing the waviness of the friction surfaces leads to cavitation which has a direct effect on the dynamic friction, causing a negative slope in the μ-v relationship. Porosity of the frictional material was also investigated and increasing porosity found to moderate the negative slope of the μ-v curve [12].

Tohyama et al. showed that increasing contact area roughness and low boundary friction increased the slope of the μ-v curve, and therefore was desirable in order to prevent shudder [68].

Friction Coefficient Ratios

One commonly used method for quantifying the μ-v relationship is to define ratios between the coefficient of friction at different velocities. Unfortunately, there are no agreed standard ratios. Many authors present the ratio μ_0/μ_d (generally used in SAE #2 investigations) or μ_s/μ_d between static and dynamic friction, but the definition of the friction coefficients differs [24, 51, 53, 69].

Ohtani et al. defined two values called μ1/μ50 and μ100/μ300 where μ1 is the coefficient of friction at 1rpm and so on. In this case the first ratio is the anti-shudder performance at low sliding velocities, and the second is the anti-shudder performance at higher sliding velocities. Values of these ratios larger than 1 corresponds to observed vibrations in vehicles [18].

Yoshizawa et al. defined the ratio μ_1/μ_{20} where μ_1 and μ_{20} is the coefficient of friction at sliding velocities of 1 and 20cm/s respectively [24]. Murakami et al. defined two values denoted by dμ/dV(50) and dμ/dV(150) as the ratio between the friction at 1 and 50 or 1 and 150rpm divided by the difference in sliding velocity [54]. Miyazaki et al. used the ratio between the friction at 6mm/s and 180mm/s, denoted μ6/μ180, as an indicator of the potential for frictionally induced vibrations [62].

Temperature Influence on Friction

Numerous authors have shown that the coefficient of friction is temperature dependent. This influence of temperature on friction can be explained by the fact that temperature affects both fluid viscosity as well as the formation of so called tribolayers; surface active additives in the transmission fluid present at the sliding interfaces in the clutch which have a significant effect on friction. The rate of generation of the tribolayer is influenced by the temperature dependant surface activity. In addition, the surface temperature will also determine which types of additives will dominate in the tribolayer [70, 71].

Using a modified SAE #2 machine, Ohtani et al. showed that friction decreased with increasing temperature during clutch engagement [18].

Haviland et al. and Watts et al. both used a low velocity friction apparatus (LVFA) to show that static and dynamic coefficients of friction decrease with increasing temperature; static friction due to increased additive performance, and dynamic friction due to viscous effects. They also concluded that the rate of change differed between static and dynamic friction and may therefore alter the anti-shudder properties [53, 72].
Both Yoshizawa et al. and Bengtsson have shown that the μ-v relationship may change drastically when the temperature is changed. Thus a fluid with good anti-shudder properties at high temperatures may have very poor properties at low temperatures as seen in Figure 4 [24, 73].
Derevjanik concluded that the slope of the μ-v curve becomes more negative at lower temperatures [74].

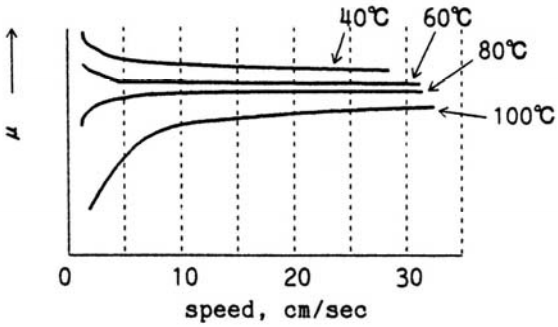

Figure 4: μ-v characteristics at different temperatures [24].

Load Influence on Friction

For ideal clutch control minimal load dependence is desirable. However, the influence of the normal load on friction characteristics have been found to vary depending on the fluid used [53].
Haviland and Rodgers claimed that neither the static coefficient of friction, nor the friction characteristics are load dependent but that the dynamic coefficient of friction seems to decrease with increasing load [72].

Drag Torque

Drag torque is unwanted friction generated in disengaged clutches which results in a loss of energy and thus lowers the efficiency of the transmission. Drag torque is generated by viscous effects in the clutch, therefore an increased distance between the disengaged discs is an advantage as is entrapment of air rather than fluid between the discs.
Kitabayashi et al. presented an experimental study of factors influencing drag torque. It was found that clutch facing area and clutch plate waviness had a major influence on drag torque while groove area had a lesser effect. The conclusion was that drag torque is governed by fluid viscosity at low velocities, but largely influenced by air entering the clutch at high velocities [3].
Yuan et al. used computational fluid dynamics (CFD) to simulate drag torque as well as the fraction of entrapped air in a clutch [4].

Clutch Temperature

The clutch temperature is the most important parameter as far as clutch and fluid durability are concerned and also affects friction coefficients and characteristics as indicated earlier [16, 17]. Working temperatures are governed by the power input (sliding velocity and transmitted torque) and how well the clutch dissipates heat.

Oil Flow

Increased oil flow through the clutch reduces wear and results in lower operating temperatures [75]. According to Payvar, in continuous slip mode, the temperature at various points in the clutch will reach a steady state value within a few seconds after which the generated heat is removed by the oil in the clutch [76].

Oil Grooves

The groove pattern also influences the temperature in the clutch and hence the durability of the friction material. Less porous materials (such as sintered bronze) are more sensitive to the presence of grooves and to groove geometry compared to more porous materials since any pores will help by supplying oil to the contact [75].

Measurement of Clutch Temperatures

The most common way to measure clutch interface temperature is to use clutch plates equipped with thermocouples running against paper-based friction plates [18, 77, 78].
Payvar measured the clutch temperature with thermocouples in the clutch disc and also in the oil inlet and outlet (see next section) [76].
Ost et al. equipped the clutch pack of an SAE #2 test rig with thermocouples. Results showed that the temperature approached 300ºC in the clutch plates furthest away from the oil inlet, oil being admitted near the axial piston [79].
Throop and McWatt equipped a lock-up clutch with several thermocouples as well as pressure and torque sensors in order to investigate the performance of continuously slipping clutches and showed significant temperature variations within the clutch [80].
Holgerson obtained clutch interface temperature by measuring infrared emissions from the friction material through an inspection hole in the separator disc [1, 81].
Osanai et al. constructed test equipment with a paper-based friction material sliding against a single crystal sapphire. With this arrangement it was possible to measure the temperature of the clutch interface through the sapphire using an infrared temperature monitoring system [82, 83].

Modelling of Clutch Temperature

A number of papers dealing with the estimation of clutch temperatures have been presented.
Tataiah formulated and solved the partial differential equations of heat conduction to predict interface temperatures during engagement. Experimental validation showed better agreement compared to earlier models [78].
Payvar developed a numerical model for determining the heat transfer coefficient (which governs the steady state temperature level) taking into account the oil film between the surface and oil in the grooves. Results agreed with experimental data to within 10% [76]. In a later study, an approximate model accounting for groove influence on heat transfer coefficient was proposed [84].
A comprehensive model was formulated by Yang et al. in 1995 which took account of heat balance, boundary conditions, material permeability, effect of splines, etc. Predictions using this model were in good agreement with experimental measurements [77]. An approximate solution to this model was later successfully applied to engagement simulations [45].
Mansouri et al. formulated a theoretical FE model that provides an accurate way to describe the dynamic and thermal behaviour of a wet clutch engagement. Results agreed well with experimental data [85].

Thermo Elastic Instabilities; Hot-Spots

Wet clutch systems are susceptible to a form of surface failure that manifests itself as macroscopic hot-spots or patches appearing on the surface of the separator disc. Hot-spots occur at locations of greater contact pressure (greater thermal input), generally around the outer radius of the separator discs and in the vicinity of the splines [86].
The hot-spots become smaller with increasing modulus of elasticity of the friction material [67]. It is also possible to categorise different hot-spotting types based on their appearance [87].
It has been shown that fluid additives influences hot-spot formation and thus careful selection of anti-wear additives can reduce the tendency of a clutch to form hot-spots [88].
Zagrodzki et al. showed that severe hot-spots can occur during short-term clutch operation with high initial sliding speed and that small geometric imperfections on the discs can trigger the appearance of hot spots [89].

Modelling of Thermo Elastic Instabilities

Zagrodzki published several FEM models of thermo mechanical phenomena in a multiple disc wet clutch. Results shows that large temperature differences occur in the clutch and that the differences are influenced

by the initial distribution of pressure on the particular friction surfaces and the Young's modulus of the material [90, 91]. Another study showed that severe hot-spots can be produced during short-term clutch operation with high initial sliding speed, but that the problem can be reduced by lowering the modulus of elasticity of the friction material and by changes in disk geometry [89]. Similar models have also been applied to single sided clutches [2].
Jang and Khonsari presented a comprehensive model for analysing the onset of thermo elastic instability (TEI) in a wet clutch. Their model could even predict the maximum number of hot-spots in a clutch [46]. Important factors mentioned were surface roughness and separator disc thickness [92, 93].
Yi et al. and Afferrante et al. have also studied the onset of TEI [94, 95].

Long Term Durability

The μ-v characteristics will change over the lifetime of the clutch due to both wear and degradation of the clutch plates and fluid [96]. Changes in the fluid during a lifetime test can either improve or deteriorate the anti-shudder performance depending on the fluid used [18]. Ohkawa et al. presented a good overview of the factors leading to the occurrence of shudder (Figure 5) [38].

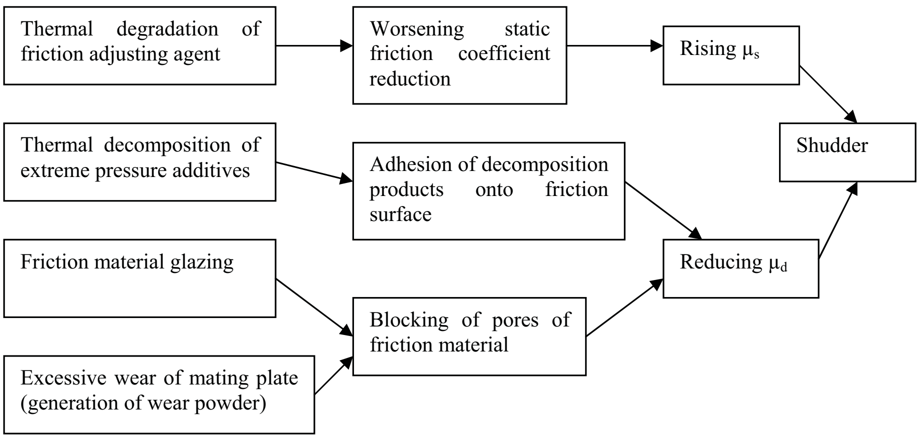

Figure 5: Shudder mechanism of a wet clutch [38].

A few attempts to predict durability and degradation have been made. Afferrante et al. introduced "damage parameters" and "damage maps" to predict TEI damage on the clutch [97]. Yang et al. developed a theoretical model based on experimentally obtained "degradation rate constants" to predict friction material degradation. It was concluded that the degradation of wet friction materials can be predicted using a theoretical model based on temperature history near the friction interface [98].

Friction Material Degradation

Ward et al. published a study of the change in μ-v characteristics during aging of ATF's and paper-based friction materials. Large differences between new and aged systems were observed, largely due to degradation of the friction material [99]. Li et al. on the other hand concluded that friction plate glazing does not affect boundary friction coefficients [100].
Guan et al. concluded that the interaction between ATFs and friction material may contribute significantly to the degradation of friction material systems and the frictional performance of clutches [101]. The ATF used can also affect friction material pore blockage and cause reduction in material strength (delamination life) as shown by Maeda and Murakami [102].
Osanai et al. found that high interface temperature causes degradation due to carbonisation of paper-based friction materials and that there is a critical value of carbonisation above which dynamic friction will suddenly decrease [83].

Fluid Degradation

Early work by Rodgers investigated the influence of oxidation on fluid life which was found to contribute greatly to fluid degradation. However, it has been demonstrated that the presence of oxygen may improve the performance of some additives [52]. Willermet et al. later presented results supporting the idea that bulk oxidative degradation of ATF's in-service is a key factor in friction durability [103].
Albertson and Okubo studied the influence of lubricant breakdown on different fiction materials and found that sintered copper linings accelerated lubricant degradation [104].
Slough et al. used a scanning force microscope and tested fresh and degraded ATF's on new as well as glazed friction material and found that the fluid degradation was the main reason for the loss of anti-shudder properties [105].
Li and Devlin et al. presented a study on the influence of both clutch plate and fluid on anti-shudder durability. It was confirmed that boundary and thin-film friction governed by the fluid are related to anti-shudder and torque capacity durability [8, 100].
Miyazaki and Hoshino attributed the deterioration of friction properties to the dissipation of friction modifier effects during prolonged use [62]. Kugimiya et al. investigated μ-v characteristics during aging and found that heating rather than shearing were responsible for the deterioration of friction modifiers [17].

Friction Materials

The friction material used has a major influence on the friction characteristics of a clutch. Commonly used friction materials include paper, sintered bronze, steel, carbon fibre, cork, asbestos and aramid fibre. Some of these are described in more detail below.
Important material properties are friction level, quality and stability, durability, heat resistance, heat adsorption and compatibility with oils and their additives [13, 106, 107].
Friction materials used in clutches should ideally be porous in order to distribute oil to (or retain in) the contact zone and to improve heat transfer from the bulk friction material into the fluid [61, 75, 107, 108]. Higher porosity generally increases the coefficient of friction [10, 61] and inhibits undesirable shudder [12]. However, porosity also promotes cavitation at the clutch interface [10].
Ohkawa et al. concluded that the lower the elastic modulus of the friction material, the higher the friction coefficient and energy/power absorbing capacity. According to Ohkawa the influence of porosity is mainly due to the fact that it lowers the modulus of elasticity [67].
The permeability of a friction material is an important factor, especially during clutch engagement. The effects of various processing parameters such as the basis weight of raw material, resin pick-up and the compression ratio on permeability, porosity, and pore size of friction materials were investigated by Chavdar using a specially developed permeameter [109].
Surface topography and chemical composition will influence friction and various surface treatments have been suggested in order to improve clutch performance [108, 110-112].

Paper Based Friction Materials

So called paper based friction materials have been used since the late fifties due to their low cost and good performance under low load conditions. These materials consist of raw paper (cotton lint or cellulose fibres) in combination with a thermosetting (organic or phenolic) resin. Kitahara and Matsumoto published a good review of the production and performance of paper-based friction materials [113].
A number of articles comparing different paper based friction materials have also been presented [39, 52, 54, 61, 75].
The coefficient of friction and loading capacity of paper-based materials is high, but due to their comparatively low thermal conductivity, their range of use becomes limited when large torque flows are involved [38]. However, improved paper based materials for high energy applications are being developed [108, 114].

Sintered Bronze Friction Materials

Sintered bronze friction materials are used where service conditions do not allow the use of steel discs or discs faced with paper, synthetic or organic linings. Sintered bronze friction materials have been shown to be sensitive to operating conditions such as sliding speed, load, groove pattern and energy compared to paper based materials, but their breakaway static friction is less influenced by operating conditions and therefore easier to predict [75].

The composition of the sintered material will influence its friction and durability. For example, a high graphite content will give a higher dynamic friction [67].

Carbon Fibre Friction Materials

Carbon fibre is preferred as a friction material due to its very high heat/abuse resistance, combined with a good, consistent, less oil dependent coefficient of friction. Carbon fibre is inert and therefore does not react with additives and does not char, melt or soften when exposed to high temperatures. However, the normally carbon filled, carbon fibre base structure is expensive, which restricts the applications for which it is used. More cost effective manufacturing methods are being investigated and may result in a breakthrough for carbon fibre as a friction material [106, 115, 116].

Hybrid materials

Hybrid materials are a more cost effective alternative to carbon fibre materials. Hybrid materials are typically manufactured using a process similar to that used for paper-based materials, but using carbon fibres in combination with organic or synthetic fibres [106, 115, 116].

Transmission Fluids

The transmission fluid is an important part of the clutch system. As seen earlier, the fluid must give the desired friction characteristics and maintain these properties throughout its working life. In addition to clutch performance, the lubricant must also protect other machine elements in the transmission such as bearings and gears.

Standards

Performance requirements for transmission fluids are generally established by transmission and clutch manufacturers. Fluid specifications typically cover friction characteristics, friction durability, oxidation stability and wear. The first official ATF specification "Automatic Transmission Fluid, Type A" was introduced by General Motors (GM) in 1949. The Type A fluid, however, did not perform well in certain applications and in 1959 Ford Motor Company introduced a new standard known as M2C33-A/B. With new applications, the demands placed on transmission fluids gradually increased leading to the introduction of modified standards. In 1967 GM introduced the Dexron standard and in 1987 Ford introduced the Mercon standard. Variants and developments from these standards are still used when classifying ATF's [117].

In the mid-nineties, JAMA (Japan Automobile Manufacturers Association), JASO (Japan Automobile Service Organisation) and Japan Petroleum Union started work on Japanese ATF standards, resulting in standards such as JASO M349-98, JASO M349-2001 and JASO M315 [51].

More recently, the International Lubricant Standardization and Approval Committee (ILSAC) ATF subcommittee has started work on comparing and merging the different standards [39, 54] since it has been demonstrated that it is possible to formulate fluids that comply with both the American and the Japanese standards [118].

Fluid Formulation

A transmission fluid consists of one or more base fluids and a number of different additives. In Table 1, typically used ATF additives and their influence on friction are presented [51, 69]. As can be seen it is the surface active additives that have the greatest influence on friction characteristics.

The concentration of each additive, and the balance between them, is important in order to achieve a working transmission fluid [17, 51, 72, 74]. In the case of surface active molecules, the type of friction materials must also be considered and different additives chosen for different friction materials such as sintered bronze, organic fibres, synthetic fibres or carbon fibres [69, 115, 119].

Current challenges are to formulate automatic transmission fluids that also work in continuously variable transmissions and in fill-for-life AT's [29, 30, 74, 120].

Kugimiya et al. have presented good articles describing the development of modern ATF's [16, 17].

Table 1: Additives for ATFs.

Additives	*Representative Compounds*	*Influence on friction*
Anti-oxidant	Alkyl phenol, aromatic amine, (ZnDTP)[4]	
Dispersant	Metal sulfonate, alkenyl succinic acid amide, organic boron compounds	Significant
Detergent	Phenate, sulfonate	Significant
Viscosity index improver	Poly-methacrylate, poly-isobutylene, poly-alkylstyrene	
Friction modifier	Fatty acid, amide, amine, polymerised phosphoric acid ester	Significant
Anti-wear agent	Phosphate, acid phosphate, sulfidised oil fat, organic sulphur or chlorine compounds, (ZnDTP)	Significant
Metallic deactivator	Organic sulphur compounds, organic nitrogen compounds, (ZnDTP)	
Rust preventer	Metal sulfonate, fatty acid, amine	Significant
Corrosion inhibitor	Perchlorinated metal sulfonate, (ZnDTP)	
Seal swelling agent	Phosphate, aromatic compounds, chlorinated hydrocarbon	
Anti-foaming agent	Silicone oil	
Colouring agent	Azo-compounds	

Base Fluids
Both mineral oils and synthetic fluids are used as base fluids; commonly paraffinic or PAOs [17, 23, 62]. According to Haviland et al. base oil viscosity is of lesser importance than the additives used and generally does not affect the additive dependent friction characteristics [32]. The choice of base fluid will, however, influence fluid durability [118, 121]. Yamamoto claims that the friction of paper-based wet friction materials increases with increasing viscosity of base fluid [107].
Shirahama found that that μ_d rises with increasing viscosity, but that the addition of succinimide gives a coefficient of friction independent on the viscosity [69]. Kitanaka showed that μ_d rises and μ_s decreases with increasing viscosity, but that viscosity dependence diminished with the addition of dispersants (alkenylsuccinimide) and detergents (metal sulfonate) [66].

Friction Modifiers
Friction modifiers are very important in order to obtain the desired friction characteristics. Friction modifiers are surface active and form molecular layers on the contacting surfaces. A typical friction modifier is a long straight-chain molecule with an active group which will strongly attach itself to the rubbing surfaces [32, 52].
The critical differences between anti-wear/extreme pressure (EP) additive films and friction modifier films are in their mechanical properties. Anti-wear/EP films are hard to shear off, while friction modifier films can easily be sheared off. A comprehensive review of the theory and application of friction modifiers has been published by Papay [122].
Friction modifiers have significant influence on friction, primarily at low sliding velocities [18, 51, 62, 69, 123]. A major drawback when adding friction modifiers in order to improve anti-shudder properties is that torque capacity decreases with increasing amounts of friction modifiers in the fluid [16-18, 24, 32, 59].
Kugimiya et al. investigated μ-v characteristics during aging and found that heating rather than shearing was responsible for the deterioration of friction modifiers. Friction modifiers should therefore be judged on their ability to resist high temperatures rather than their ability to resist shearing [17].

Dispersants
Dispersants seem to have an impact on anti-shudder performance. Shirahama concluded that succinimide raised μ_d, and that it seemed to adsorb to the paper-based material as do borated dispersants [69]. Ichihashi

[4] The use of ZnDTP is less common nowadays since it is known to cause pore blockage in friction materials [69].

noted that Ca-based cleaning dispersants increases the coefficient of friction over the entire range of sliding velocities [51].
Nakada et al. concluded that dispersant/friction modifier interactions govern low speed friction and proposed an additive mechanism model [59].

Detergents

Metallic cleaning agents are classified as surfactants composed of lipophilic and polar groups. Thus, in terms of structure they are similar to friction modifiers, but the polar groups are generally sulfonate (adsorb strongly to both steel and paper), phenate and other salts with metals. Detergents promote a rise in μ_d and a reduction in μ_s [69]. Kitanaka also concluded that metal sulfonate adsorbs to steel plates and plays an important role in the friction characteristics by raising μ_d [66]. Detergents may also alter friction characteristics due to interactions with friction modifiers. It is therefore necessary to consider the combined effect of the additives [74].

Anti-Wear Agents

Anti-wear agents are often phosphate based consisting of a lipophilic group (hydrocarbon radical) and a polar phosphoric acid. The structure of the lipophilic group has been found to exert significant influence on the μ_0/μ_d ratio [69].

Experimental Equipment

A number of different experimental rigs designed for evaluating the performance of lubricant and friction material in wet clutches are described in the literature. The vast majority of these are designed for ATF or agricultural tractor oil applications.
Clutch performance is generally studied in respect to three different criteria;

- Torque capacity – The amount of transmittable torque before slipping occurs.
- Shudder – Frictionally induced torque variations resulting in noise and or vibration.
- Engagement characteristics – The variation of torque transfer during engagement.

In recent years increased attention has been paid to the test methods used to evaluate wet clutch applications. The International Lubricant Standardization and Approval Committee (ILSAC) has published studies comparing different methods [39, 54].
In the following sections commonly used experimental equipment is presented.

SAE #2 Friction Tester

The SAE No. 2 machine is probably the most widely used equipment for friction performance evaluations and has been used for over thirty years [18, 23, 24, 39, 79]. This rig tests a full clutch pack, allowing assessment of the overall performance of lubricant/friction material combinations. A schematic of the SAE #2 machine is presented in Figure 6.
The SAE #2 machine can be used in two different ways. In one type of test, the large drive motor accelerates the flywheel to 3600rpm at which point the motor is turned off and the clutch engaged to stop the flywheel. This procedure is repeated several times a minute for 50 to 100hours in order to investigate the durability of the fluid.
To determine the torque capacity, the clutch is engaged and then the low-speed drive is run at 0.72 or 4.37rpm. Typical data that are obtained include the maximum breakaway friction and the friction coefficient after some seconds of continuous slip. The applied normal force in the SAE #2 machine is generally well below 10kN.

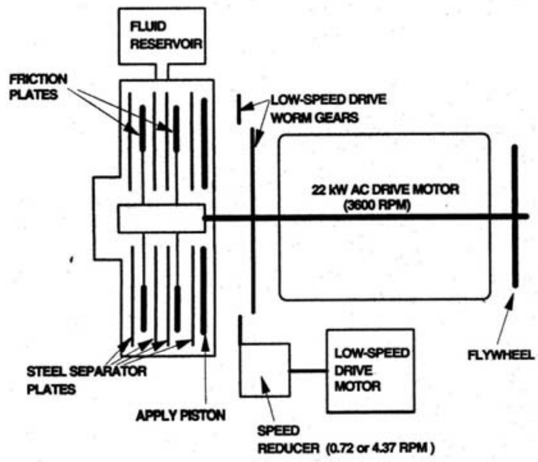

Figure 6: Schematic of the SAE #2 machine [39].

The JASO M394-95 is a Japanese standard for friction evaluations using the SAE #2 or similar apparatus [39]. The DKA machine is similar to the SAE #2 and is used by some European companies [15].
Researchers have also used modified SAE #2 machines with some success to evaluate friction characteristics at low speeds [18, 54].
According to Huron, it is possible to predict the SAE #2 performance of a transmission fluid based on the physical properties of the lubricant combined with friction data obtained from a Low Velocity Friction Apparatus (described below) [124].

Low Velocity Friction Apparatus (LVFA)

The General Motors Low Velocity Friction Apparatus (LVFA) has been around since the sixties and has proved useful in the evaluation of μ-v characteristics at low velocities [52, 53, 72, 99]. The LVFA uses a small scale annular part manufactured from the friction material running against a steel counterpart. The LVFA uses a flywheel coast-down in order to evaluate a full range of sliding speeds. The normal load is applied by deadweights through a lever at the bottom of the apparatus. A schematic of the LVFA machine is presented in Figure 7.

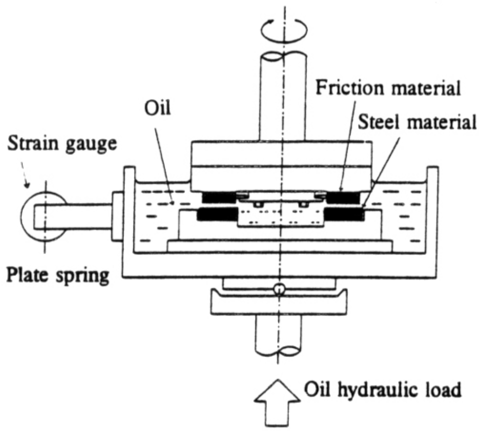

Figure 7: Schematic of the LVFA [51].

The Variable Speed Friction Tester (VSFT) [74, 99] and the μ-v TESTER [16, 17] are modified versions of the LVFA. The Tribometer test used in Europe is also similar to the LVFA [15].

The R-H Friction Apparatus

The R-H friction apparatus was developed in the early sixties by Rodgers and Haviland to investigate the μ-v characteristics of controlled-slip differentials [32]. The R-H friction apparatus is a larger version of the LVFA and can be used to investigate the frictional characteristics of full size clutch plates [32, 54].

JASO M349-98

The JASO M349-98 is often referred to as a modified LVFA test. The JASO M349-98 is driven by an electric motor instead of using the flywheel coast-down of the traditional LVFA. This means that it is possible to run many different test cycles controlled by a computer, such as endurance tests. The applied load is normally 1MPa, and the velocity is between 0.006 and 1.5m/s during μ-v testing and 0.9m/s when endurance testing [54, 100].

Other Clutch Test Machines

Gil et al. developed a test rig with a friction disc running against a sapphire disc in order to study cavitation in the clutch. This equipment was also equipped with sensors measuring the oil film pressure [10, 11]. Very similar equipment using quarts glass was described by Kato et al. [12]. The same pressure measurement method were adopted by Ito et al., but without the use of a transparent clutch disc [125].

Both Eguchi et al. and Sano et al. attempted to measure the oil film formation process and the friction contribution from boundary and hydrodynamic lubrication by measuring the electric conductivity of the oil film. The method was shown to work reasonably well with pure paraffinic base fluids, but with fully formulated ATFs conductivity is low and the results are not easily interpreted [9, 23].

The wet clutch test rig designed by Holgerson is able to apply a drive torque as well as an inertia torque during engagement, thus closely simulating the clutch engagement in an automatic transmission in use. In addition to friction measurements the equipment was also equipped with infrared temperature monitoring, enabling measurement of the clutch interface temperature [19, 81].

The low-speed slip testing machine used by Toyota was designed to investigate the anti-shudder performance of ATF's by measuring μ-v characteristics in the speed range 0.06-60rpm. This rig was equipped with a torsion bar to vary the rigidity of the tester [24, 59].

The ZF GK Test Bench is a multi purpose test bench for evaluation of wet clutch performance. The equipment is able to simulate the LVFA, DKA and other methods [15].

Small Scale Test Methods

Many authors have chosen to conduct their investigations using small scale parts in standard test equipment instead of full clutch discs in dedicated rigs.

Plint and Plint used a reciprocating friction and wear tester type TE77 to investigate the stick-slip behavior of lubricants [126].

Ost et al. utilised a pin-on-disc test rig to investigate the friction of a wet clutch material. Results were compared to the SAE #2 tester. The friction showed similar behaviour, but the wear rate differed [79].

Ward et al. used a Falex 6 tester together with a LVFA in order to evaluate anti-shudder durability. Friction was measured in the LVFA whilst fluid ageing was evaluated in the Falex 6 tester [99].

A standard Falex Block-on-Ring test apparatus was used by Derevjanik et al. in order to measure metal/metal friction coefficients for different ATF formulations [74].

Conclusions

This paper has summarised some of the research efforts associated with improving and understanding the performance and behaviour of wet clutches. New applications employing wet clutches continue to appear, and the increased power density and increased efficiency that these demand have necessitated improvements in clutch performance and will continue to do so for a long time.

The major challenges facing clutch designs today are increased demands on smooth controllable torque transfer, often under conditions of continuous slip. This, in combination with increased power density, necessitates improved performance of both friction materials and transmission fluids.

Novel friction materials have emerged, but at the same time paper-based materials have improved with respect to allowable power input and durability while still maintaining a cost advantage in many applications.

As regards transmission fluids, stricter demands on friction characteristics and friction durability are necessary in many applications where friction control is important since the transmission software is programmed based on measured friction characteristics for the friction material combined with the originally used transmission fluid. Other factors facing fluid designers are decreased oil volumes (to reduce churning losses and cost) in combination with increased drain intervals, or even demands on fill-for-life service.

New transmission designs, such as continuously variable transmissions, whether based on toroidal elements, belts or chains, also introduce new demands on the lubricant. These include high friction under hydrodynamic and mixed lubrication as well as the ability to protect machine elements from scuffing and other damage during sliding.

Present models predicting clutch performance are reasonably accurate and are increasingly used during design stages as well as being implemented in clutch control software. Most models, however, still rely on experimentally obtained boundary friction data, an area deserving further attention.

In 1963 T. D. Newingham concluded that "In spite of the many excellent papers already published on friction properties of automatic transmission fluids, it remains a challenging and rewarding area for study." [119]. This statement is as true today as it was more than forty years ago.

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Paper B

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Limited Slip Wet Clutch Transmission Fluid for AWD Differentials; Part 1: System Requirements and Evaluation Methods

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ABSTRACT

Part 1 of this paper is focused on requirements and evaluation methods for transmission fluids used to lubricate wet clutches in all-wheel drive systems. Part 2: Fluid Development and Verification; presents more measurement data and shows how these results are used in the fluid development process.

The investigated all-wheel drive system, featuring a wet multi-plate clutch with a sintered brass base friction material, is described with emphasis on the demands placed on the transmission fluid.

A new test equipment to determine the frictional (μ-V) characteristics of the transmission fluid is described. The equipment can measure the actual temperature experienced by the fluid in the contact zone using an infrared temperature measurement method, as well as temperature in the oil sump and inside the clutch discs by use of thermocouples. Variable operation parameters include velocity, normal force and oil flow.

INTRODUCTION

During the last years several electronically controllable automotive transmission systems, where wet clutches are used as intelligent differentials, have emerged on the market [1, 2, 3].

These applications generally involve high stresses on the transmission fluids. The fluids must preserve the desired frictional characteristics at different torque levels during its lifetime in an environment where the ambient and system temperature vary in a wide temperature range. The fluid must also be able to function with both particles and water present in the system. Typically fill-for-life capability is also required, thus necessitating the use of shear stable additives.

Research on wet clutches has been conducted for a long time, but the vast majority of scientists have focused on wet clutches in automatic gearboxes. Wet clutches used in limited slip differentials generally operate under different conditions than the automatic transmission clutches. In this type of application it is common with long periods of continuous slip at low velocities; hence the engagement characteristics are not as important as the stick-slip counteraction. High surface pressures, over 10 MPa compared to less than 2 MPa [4] in the automatic gearbox, are also common, thus making the use of traditional organic or paper friction materials unfeasible.

The research on this type of wet clutches has increased recently, both on transmission fluids for these applications [5, 6] and on different test and evaluation methods [7, 8]. The aim of this paper is to present a newly developed tester for wet clutches used in limited slip differentials. The new equipment can be used to investigate a number of important parameters, at a fairly low cost.

HALDEX LIMITED SLIP COUPLING

An active-on-demand (AOD) all-wheel drive (AWD) system for automobiles has been developed by Haldex Traction Systems with the aim of meeting new demands on short system activation / deactivation time. Currently the Haldex LSC system is used in vehicles produced by Volvo Cars Corporation and the Volkswagen group.

SYSTEM FUNCTION

The Haldex LSC AWD system features a multiple disc wet clutch that consists of clutch plates covered with a sintered friction material. The clutch pack is used to distribute drive torque to the rear axle of the vehicle. By using a wet clutch, torque transfer control is enhanced. It is therefore possible to electronically control the drive torque distribution between the front and rear axle in order to optimize vehicle dynamics.

A schematic diagram of the coupling can be seen in Figure 1. The left shaft is connected to the rear axle of the vehicle, and the right hand side is connected to the front axle. When a speed difference occurs between the front and rear axle, a cam curve on the rear axle will perform a pumping action on the hydraulic piston pump. The generated hydraulic pressure can be applied to the clutch pack in order to reduce the speed difference between the shafts, thus engaging the all-wheel drive. The torque transmitted by the coupling is controlled by a throttle valve.

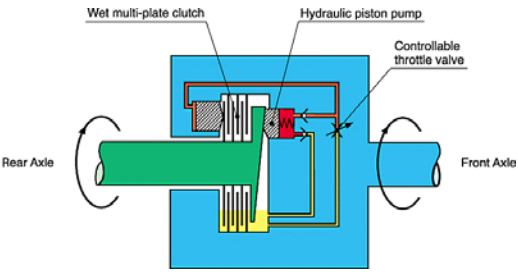

Figure 1. The Haldex Limited Slip Coupling in a schematic view.

This type of AWD system has several advantages over traditional transmission systems. The activation / deactivation are both rapid and electronically controllable via the throttle valve, therefore the coupling work well in cooperation with different electronic driving aid systems such as electronic stability programs (ESP) and anti-lock brake systems (ABS) which often require AWD deactivation [9]. The possibility of system deactivation also makes low speed maneuvering such as parking easier. It also makes it possible to use run-flat-tire functions and to enable vehicle towing.

FLUID REQUIREMENTS

Characteristic operating conditions for wet clutches in this type of application include low sliding velocities and high clutch disc pressure. Under these conditions it is common that stick-slip and shudder arise, resulting in noise and additional stresses on other parts of the transmission. This behavior has been investigated by a number of authors, both experimentally [10, 11, 12] and theoretically [13, 14, 15].

The results obtained indicate, as previously stated in literature [8, 16], that low static coefficient of friction (μ_s) and a dynamic coefficient of friction (μ_d) that increases with increasing sliding velocity will help in reducing stick-slip induced vibrations. The static friction coefficient depends on how long the surfaces have been immobile, while the dynamic friction coefficient depends on relative speed and pressure between the two surfaces [17].

Typical μ-V relationships from friction tests of different ATF fluids are presented in Figure 2 [16]. In order to avoid vibration problems, friction characteristics similar to that of Oil A are desirable. Parameters such as base oil composition, the additive composition including the relative concentration of the components, and the sintered friction material have significant influence on the friction characteristics of the system.

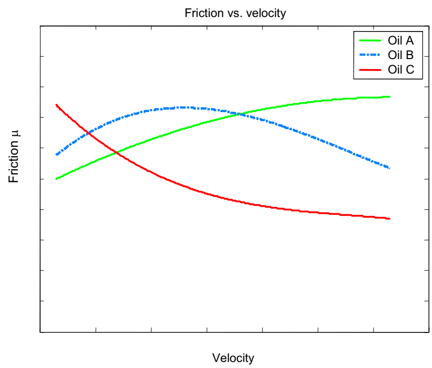

Figure 2. Schematic μ-V relationships for ATF fluids; Oil A will suppress vibrations, while Oil B and Oil C may bring about self-induced vibrations since they present a negative slope in some regions.

Traditional DEXRON / MERCON ATF fluids, however do not work in this application. This is partly due to poor adsorption and hence poor wetting ability of the friction-modifying additive on to the surface of used brass-based sintered friction material.

Fluids durability and its fill-for-life (FFL) capability is another important property, which has to be in consideration. High shear stability of viscosity improvers and low temperature properties of the finished fluid are additional important parameters.

Beside noise-free operation of the clutch, the additional demand in oil-water system is that the combination should not cause any stick-slip and damages in the clutch system.

CLUTCH PLATES

The used friction material is a dispersion sintered lining with a brass base applied to a hardened steel disc. This material is able to withstand high pressure, and is fairly cheap to manufacture. The separator disks are manufactured from hardened steel.

The friction pair can be seen in Figure 3. The outer diameter of the friction lining is 108 mm, and the inner diameter is 76 mm. The area of contact when the oil grooves have been accounted for is approximately 2250 mm^2. The grooves facilitate oil distribution to the area of contact, and help to lower the temperature in the clutch by enhanced oil flow [18].

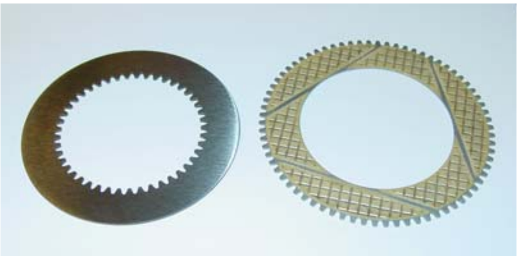

Figure 3. Used clutch discs. Separator disk from hardened steel to the left, and friction disc with sintered brass-base friction material to the right.

CLUTCH FRICTION TESTER

A clutch friction tester has been developed in order to evaluate frictional characteristics as a function of sliding velocity, normal force, temperature and oil flow.

The new apparatus is a disc-on-disc type equipment with two active friction surfaces. The equipment uses IR-temperature measurement equipment and thermocouples for temperature monitoring.

There are numerous examples of different test equipments described in literature [8, 10, 11, 12], but generally the obtainable normal force is to low to simulate these applications. The usable velocities are often to high to be of interest, resulting in poor accuracy at low sliding speeds. The IR-temperature measurement system is not commonly used, but has many advantages both in its rapid response and in the saved trouble of equipping the samples with thermocouples.

EARLIER EQUIPMENTS AND DESIGN CONSIDERATIONS

This equipment is the result of the combination of design ideas from two predecessors. The temperature measurement method using an IR-pyrometer was developed by Holgerson [4, 19] and further refined by Mäki [20]. Friction measurements have been carried out successfully in a test bench described by Ganemi et.al. [5, 6].

In the new test rig it is also possible to control both the natural frequency of the system, and the oil flow between the clutch discs.

GENERAL DESIGN

An overview of the equipment can be seen in Figure 4. The equipment is mounted on an aluminium stand (1). The base is a rigid beam (2) of length 1600 mm. On the left hand side, the motor with its gearbox (3) is mounted. The driving force is transmitted by a shaft coupling (4), through a hollow piston hydraulic cylinder (5) to the clutch housing (6). The torque transmitted by the coupling is transmitted through a torsion bar (7) to the torque measurement cell (8). The torque measurement cell (8) is connected to the beam (2) by a slide system (9). Important technical data for the equipment are summarized in Table 1.

Table 1. Summarized technical data for the test equipment.

Sliding velocity	m/sec rpm	0.005-0.6 0.5-125
Normal force	N	30000
Allowable torque	Nm	400
Oil circulation flow rate	ml/min	0-1000
Maximum sample rate	samples/sec	200000
IR-thermometer response time	msec	30
Detected IR-wavelengths	μm	8-14
Natural frequency	Hz	100-500

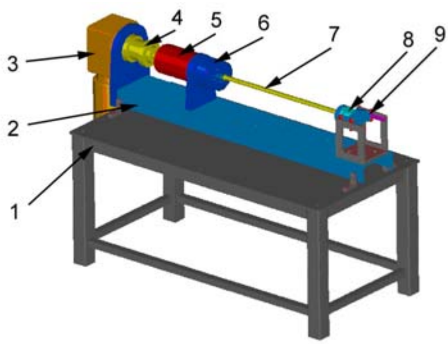

Figure 4. Overview of the test bench.

DETAILED DESIGN

A more detailed view is shown in Figure 5. The power is generated by an asynchronous helical bevel geared servo motor with forced cooling (3). The motor is controlled by a drive inverter with feedback control to obtain accurate drive speed. The maximum output torque is 500 Nm, and the speed can be varied between 0.5 and 125 rpm (2.5 to 600 mm/sec on the mean radius).

The torque is transmitted to the driveshaft (10) via a torsionally rigid all-steel flexible coupling (4). Axial, radial and angular shaft misalignments are compensated by flexible disc packs, which are torsionally rigid in circumferential direction.

The normal force on the test clutch is applied by a double acting hollow piston cylinder (5). In this case we have chosen to limit the normal force to 30,000 N by a pressure limiting valve. The force can be applied either manually, or by using a hydraulic power unit in combination with a feedback controlled pressure valve.

In the magnified view of the clutch housing in Figure 5, the friction disc (11) and the separator discs (12) can be seen. The friction disc is connected to the driving shaft (10), and the separator discs are connected to the torsion bar (7). During operation, the parts shaded in blue in Figure 5 are rotating.

When a normal force is applied to the clutch by the hydraulic cylinder (5), torque is transmitted from the driving shaft (10) to the torsion bar (7). The transmitted torque is then measured by the torque measurement cell (8). The applied normal force is measured by the load cell (13); this is possible thanks to the slider system (9) which allows the torsion bar and torque cell to move freely in the axial direction. Both the force and torque transducers are full bridge, strain gauge type with built in amplifiers.

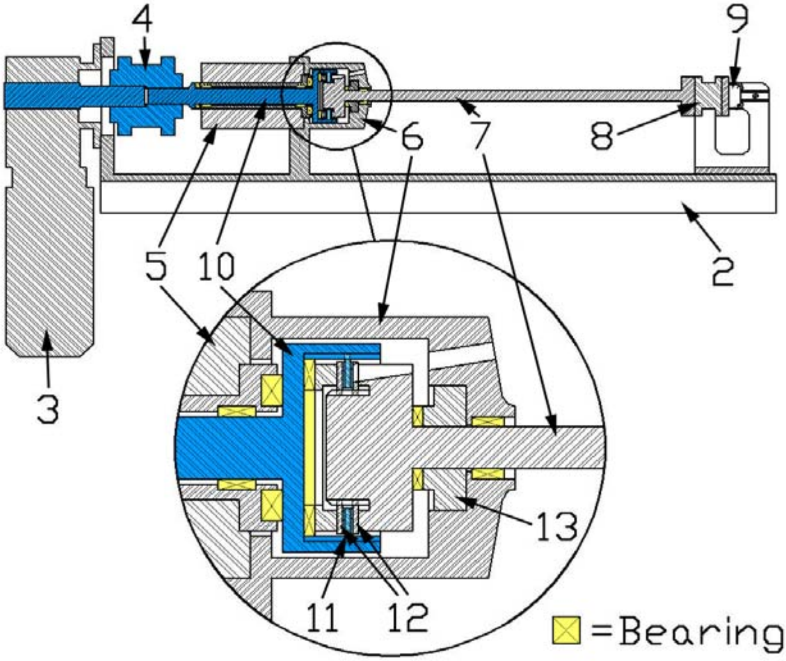

Figure 5. Simplified cross-section of the test apparatus. Blue shaded parts are rotating during operation.

Infrared temperature measurements are performed on the friction disc through a hole in one of the separator discs, the torsion bar and the housing. Thermocouples are installed in the oil sump to measure oil bulk temperature.

It is also possible to install thermocouples in the separator discs. In most cases, the temperature readings from a thermocouple drilled into the separator disc and the reading from the IR-thermometer do not differ. Under severe heat generation tough, the IR-thermometer returns a slightly higher temperature value than the thermocouple.

By changing the length of the torsion bar it is possible to vary the natural frequency of the apparatus from 100 to 500 Hz.

Accuracy

The test apparatus is equipped with a 16-bit data acquisition card and signal conditioning system from National Instruments. Maximum sample rate is 200000 samples/sec, but typically 1000 samples/sec is adequate.

The design of the equipment ensures that the friction torque (except for the bearing losses, μ=0.0050) is transmitted to the torque measurement cell. The combined error of the torque cell, including measuring bridge and amplifier, is less than 0.25% F.S.

The normal force is measured by a load cell with a combined error, including measuring bridge and amplifier, less than 0.25% F.S.

The IR-thermometer measures the mean temperature over a circle 4 mm in diameter. Detected wavelengths are 8-14 μm allowing good sensitivity in this application. The lowest response time of the thermometer is 30 msec. The accuracy is within ±1.5°C.

The thermocouples are generally more accurate with a typical error of ±0.67°C.

MEASUREMENTS

In order to determine the friction-velocity characteristics of a transmission fluid, a test cycle where the sliding velocity is linearly increased is run.

Measured data from one test is presented in Figure 6. At time=0 sec a velocity of 1 rpm is applied, the normal force 20 kN is kept constant throughout the test. At time=10 sec the velocity is starting to increase to reach 100 rpm at time=20 sec. During this time, the temperature of the clutch disc surface increases 15.5°C.

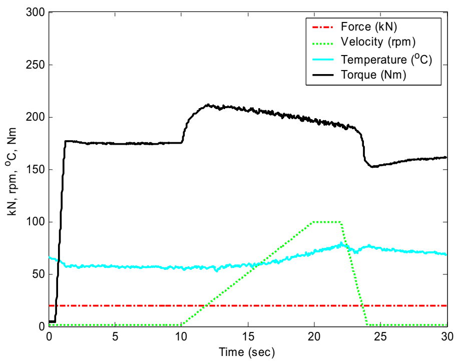

Figure 6. Measurement data from one test cycle used to determine friction-velocity characteristics.

From the transmitted torque, assuming Coulomb friction, it is a straight forward procedure to calculate the friction-velocity relationship presented in Figure 7.

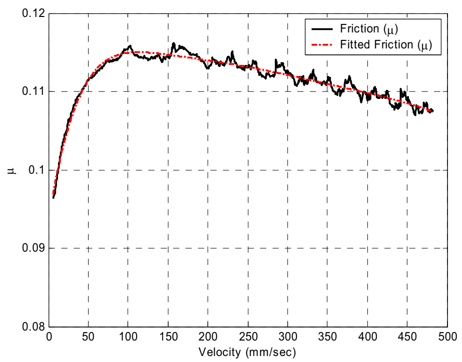

Figure 7. Friction-velocity relationship for a transmission fluid tested at 60°C.

To determine the friction-temperature behavior of the fluid, an additional test with constant normal force (20 kN) and constant velocity (25 rpm or 120 mm/sec) was performed. The test commenced at room temperature, and the coefficient of friction was monitored at increasing temperatures. In Figure 8 a typical friction-temperature graph is presented.

It is generally sufficient to fit the friction-temperature data with a straight line within the typical temperature range of operation, i.e.

$$\mu = \alpha * T + \beta$$

In the presented case $\alpha = -0.00033$ and $\beta = 0.130$

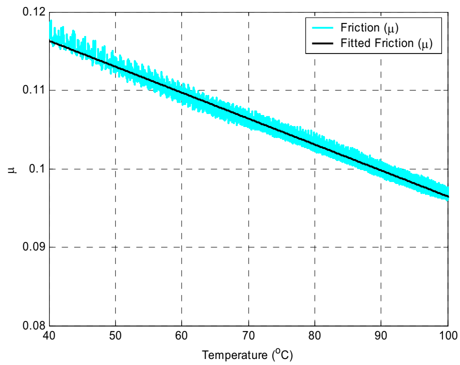

Figure 8. Friction-temperature behavior of a typical transmission fluid.

Temperature compensation

It is possible to use the known friction-temperature relationship and compensate the friction-velocity curve in Figure 7 with respect to the increase in temperature during the test. In this case the temperature at the low velocity is around 60°C, while the friction values at 450 mm/sec are measured at roughly 75°C.

In order to take this into account it is sufficient to keep track of the temperature rise since the start of the test, ΔT, and then for each velocity subtract the contribution from the change in temperature, i.e.

$$\mu_{Compensated} = \mu - \alpha * \Delta T$$

In Figure 9 both measured and temperature compensated friction from three different tests at similar conditions are shown.

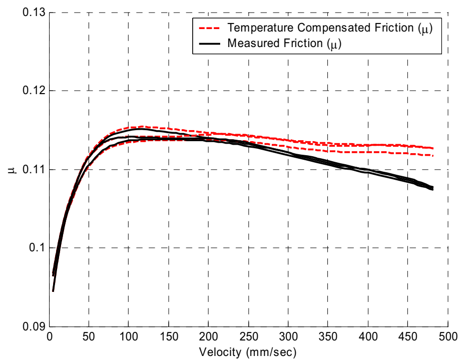

Figure 9. Friction-velocity curves from three different tests run at similar conditions. The red curves have been compensated for the change in temperature during the test.

The temperature compensated friction curve generally displays a more favorable appearance with respect to stick-slip due to less negative slope, especially at higher sliding velocities.

Stick-slip occurrence

The occurrence of stick-slip can be easily detected in the test equipment. In addition to the large fluctuations in transmitted torque visible in Figure 10, there is severe squeaking noise from the clutch.

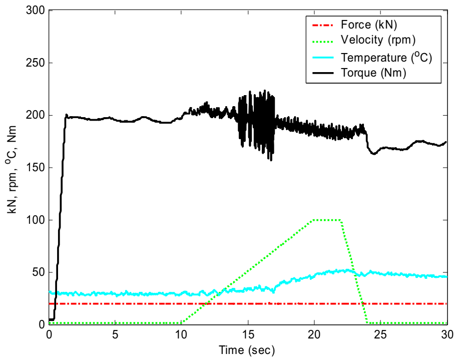

Figure 10. Measurement data from a fluid experiencing stick-slip phenomena.

MEASUREMENT FOR VALIDATION

Tests on a DEXRON –III ATF fluid in combination with organic friction material have been performed in the apparatus. The evaluated fluid-friction material combination is not suitable in the described application, but may serve as a validation of the test equipment in comparison with for instance the modified R-H friction test and the JASO M349-98 modified LVFA test [8].

Measurement data from 10 different tests on the ATF / organic lamella system are presented in Figure 11. In this case the tests are run with a load of 1.2 kN or 0.3 MPa at 30 °C. The system is very unstable with respect to stick-slip at low velocities as implied by the negative slope of the μ-V curve.

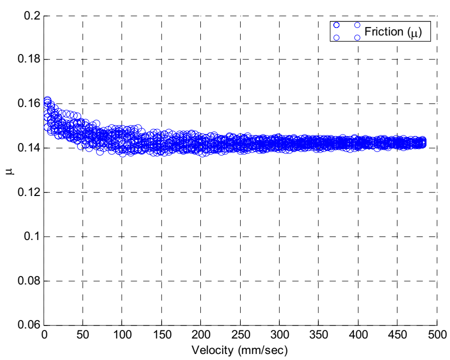

Figure 11. Friction-velocity data from 10 different tests on a ATF / organic lamella combination.

CONCLUSION

1. A new test rig has been developed to investigate the behavior of transmission fluid for wet clutches simulating the conditions prevalent in AWD differentials.

2. A non-contact IR temperature measurement technique has been employed and found suitable for measuring the temperature of clutch plates. This technique alleviates the need of mounting thermocouples in to the clutch plates.

3. Large differences between the temperature at the surface of friction material and the oil sump temperature have been detected. It is, therefore, necessary to take into account actual temperature experienced by the fluid during evaluation.

4. The occurrence of stick-slip is a problem with some fluids and can be detected by the newly developed test rig through friction-velocity measurements.

5. The temperature rise has a significant influence on μ-V curve, particularly at higher sliding velocities, and it can be compensated for temperature variation during fluid tests.

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CONTACT

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

ABS: Anti-lock Brake System. Brake system that prevents wheel locking during hard braking. Generally works better with deactivated AWD.

AOD: Active-On-Demand. The AWD system is only activated on signals from the electronic control units of the vehicle.

AWD: All-Wheel Drive. Often denoted 4WD or 4 wheel drive.

ESP: Electronic Stability Program. Electronic driving aid system preventing the vehicle from skidding out of control.

FFL: Fill-for-life. Typically a FFL fluid must have a life-time of at least 200000 km.

F.S.: Full Scale. Typically measurement accuracy is defined in per cent of the maximum reading.

LSC: Limited Slip Coupling.

LVFA: Low Velocity Friction Apparatus. Equipment used to investigate friction characteristics at low sliding velocities.

Paper C

WET CLUTCH TRIBOLOGY – FRICTION CHARACTERISTICS IN ALL-WHEEL DRIVE DIFFERENTIALS

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ABSTRACT

In recent past several electronically controlled automotive transmission systems, where wet clutches are used as intelligent differentials, have emerged on the market. These applications generally impose high stresses on the transmission fluids. The fluids must retain the desired frictional characteristics at different torque levels during its lifetime in an environment where the ambient and system temperatures vary in a wide temperature range.

This paper deals with friction characteristics of transmission fluids for wet clutches in all-wheel drive systems, and summarizes a recent licentiate thesis on the subject [1].

The all-wheel drive system, featuring a wet multi-plate clutch with a sintered bronze based friction material, is described. Test equipment designed in order to determine the frictional characteristics of transmission fluids are described. This equipment can also be used to measure the actual temperature experienced by the fluid in the contact zone using an infrared temperature measurement method.

Results show the influence of several operating parameters on the frictional behaviour of the clutch. Temperature is shown to have significant influence on the friction characteristics of transmission fluids. The friction decreases with increasing temperature, and it is therefore necessary to measure the correct temperature in the clutch to obtain realistic values of friction. The friction-velocity relationship is a good indicator of the fluid's ability to suppress friction induced vibrations. It is, however, important to measure the friction-velocity relationship at constant temperature, or compensate the relationship accordingly. The influence of clutch disc pressure on friction is quite moderate, as compared to the influence of temperature and sliding velocity.

The influence of different oil additives on friction has also been investigated. These additives have considerable influence on friction, and this must be considered when formulating new transmission fluids.

INTRODUCTION

The most important parameter to control in wet clutch applications such as wet brakes, limited slip differentials and lock-up clutches where the clutch is operating with low sliding velocities under long periods of time are the friction characteristics. In order to suppress vibrations it is important that the transmission fluid and friction material provide a low static coefficient of friction and a dynamic coefficient of friction that increases with the sliding velocity [2, 3, 4, 5, 6, 7, 8].

ALL-WHEEL DRIVE SYSTEMS

Originally, all-wheel drives (AWD) were used primarily in order to improve the off-road capacity of military and other off-road vehicles. In the early eighties, a new market for all-wheel drive performance vehicles emerged with the introduction of the Audi Quattro [9].

Since then a number of studies have shown significant improvements on vehicle dynamics of all-wheel drive vehicles compared to two-wheel drive vehicles [10, 11, 12].

In traditional all-wheel drive systems, a viscous coupling is commonly installed on the propeller shaft in order to transmit torque while still allowing some difference in rotational speeds between front and rear axle. The function of the viscous coupling has been a subject for extensive research for quite long time [13]. The drawback with the viscous coupling is that it is not possible to control it during operation. Therefore it does not work well in combination with electronic driving aid systems such as electronic stability programs (ESP) and traction control systems.

HALDEX LSC ALL-WHEEL DRIVE SYSTEM

An active-on-demand all-wheel drive system for automobiles has been developed by Haldex Traction Systems with the aim of meeting new demands on short system activation / deactivation time [14, 15]. Currently the Haldex LSC system is used in vehicles produced by Volvo Cars Corporation and the Volkswagen group.

SYSTEM FUNCTION

The Haldex LSC AWD system features a multiple disc wet clutch that consists of clutch plates covered with a sintered friction material. The clutch pack distributes drive torque to the rear axle of the vehicle. By using a wet clutch, torque transfer control is enhanced. It is therefore possible to electronically control the drive torque distribution between the front and rear axle in order to optimize vehicle dynamics.

A schematic diagram of the coupling can be seen in Figure 1. The coupling is mounted on the propeller shaft. The left shaft is connected to the rear axle of the vehicle, and the right hand side is connected to the front axle via the propeller shaft. When a speed difference occurs between the front and rear axle, a cam curve on the rear axle will perform a pumping action on the hydraulic piston pump. The generated hydraulic pressure can be applied to the clutch pack in order to reduce the speed difference between the shafts, thus engaging the all-wheel drive. The torque transmitted by the coupling is controlled by a throttle valve.

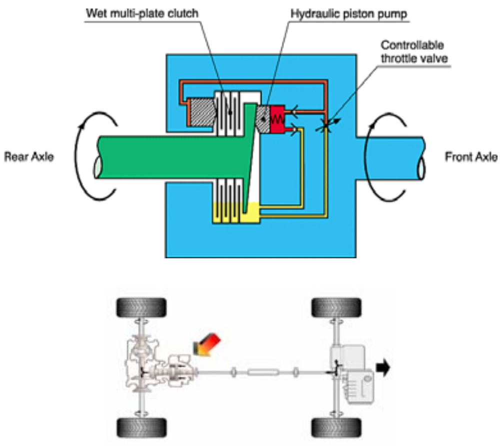

Figure 1: The Haldex Limited Slip Coupling in a schematic view.

This type of AWD system has several advantages over traditional transmission systems. The activation / deactivation are both rapid and electronically controllable via the throttle valve, therefore the coupling works well in cooperation with different electronic driving aid systems such as electronic stability programs (ESP), traction control systems and anti-lock brake systems (ABS) which often require AWD deactivation. The possibility of system deactivation also makes low speed manoeuvring such as parking easier. It also makes it possible to use run-flat-tire functions and to enable vehicle towing.

There are a number of other systems on the market that are working in a similar way as the Haldex LSC, such as Toyota Active Torque Control 4WD [16], All-Mode 4WD [17] and ATTESA E-TS [11].

SCOPE OF THIS STUDY

A lot of work has been done on the investigations of friction characteristics; there are however still areas that need to be further explored;

- *Design* - Current standardized test equipments for wet clutch evaluations are designed for AT applications; This means that most of the equipments on the market are designed for higher sliding velocities and lower normal loads compared to those present in the investigated differentials, therefore it is necessary to design a new test equipment in order to simulate realistic operating conditions for this type of application.

- *Verification / Investigation* - Almost all research has been conducted on paper-based friction materials and it is therefore necessary to verify that the findings of other authors are valid also with the much harder and more resilient sintered friction material at the operating conditions of limited slip differentials.

- *Fluid Formulation* - Many of the active additives in traditional ATF-fluids are surface active; This means that when changing the friction material from paper-based to bronze-based, the effect of some additives may be different. It is therefore necessary to formulate special transmission fluids for this type of applications.

The aim of this work is to gain an understanding of the friction / lubrication conditions prevalent in the Haldex LSC under varying and realistic operating conditions. Investigated fluids have been tailor made for this application by Statoil Lubricants R&D.

In order to limit the scope of work, a choice was made that the investigated clutch material should be the sintered bronze material described below.

WET CLUTCH FRICTION MATERIAL AND SEPARATOR DISCS

The investigated friction material is a dispersion sintered lining with a bronze base applied to a hardened steel disc. This material is able to withstand higher stresses and temperatures than paper-based materials, and is fairly cheap to manufacture contrary to carbon-fiber materials. The separator disks are manufactured from hardened steel.

The friction pair can be seen in Figure 2. The outer diameter of the friction lining is 108 mm, and the inner diameter is 76 mm. The area of contact when the oil grooves have been accounted for is approximately 2250 mm^2. The grooves facilitate oil distribution to the area of contact, and help to lower the temperature in the clutch by enhanced oil flow.

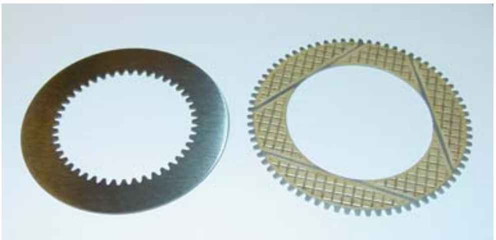

Figure 2: Investigated clutch discs. Separator disk from hardened steel to the left, and friction disc with sintered bronze friction material to the right.

EXPERIMENTAL EQUIPMENT

The main part of the work has been carried out on the two test equipments described below.

THE WET CLUTCH TEST RIG

The Wet Clutch Test Rig was originally developed by Holgerson for the investigation of engagement characteristics of paper-based wet clutches [18]. The original version of the Wet Clutch Test Rig was designed to work with very high velocities (up to 10000 rpm), and therefore required some modifications in order to be able to simulate the operating conditions prevalent in the Haldex LSC. The major modifications were the change of the hydraulic motor, re-dimensioning of the gearwheels and modifications of the control and measurement system.

FUNCTION

The apparatus is depicted in Figure 3. All shaded parts are rotating during operation. The apparatus is driven by a hydraulic motor (1), which can deliver 255 Nm between 5 and 940 rpm. The velocity is controlled by a flow limiting proportional hydraulic valve.

The motor drives the two gearwheels (2, 3) and the friction disc (4) is mounted on the second gearwheel (3). The gearwheel (3) is mounted on a hydraulic cylinder (5). The cylinder applies a normal force in the range 0-20 kN to the clutch.

The opposing steel lamella (6), with a laser cut inspection hole, is mounted on the non-rotating backing holder (7).

The force and torque applied to the backing holder are transmitted to the housing (8) by piezoelectric load and torque cells (9).

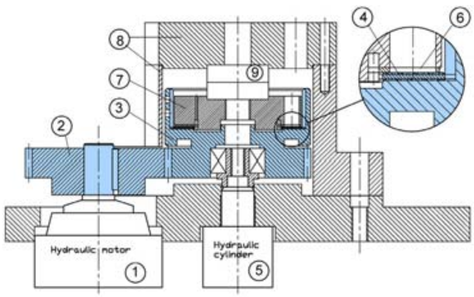

Figure 3. The Wet Clutch Test Rig. Shaded parts are rotating during operation.

Although this method was primarily designed for investigations of paper-based wet clutches used in automatic gearboxes, it has proven to work very well in this application. The accuracy of the measurements from this rig is well within ±5% [19].

TEMPERATURE MEASUREMENT METHOD

The temperature of the clutch surface is measured with an infrared thermometer through the hole in the housing, backing holder and steel disc. The thermometer measures the temperature over a spot 4 mm in diameter, and the measured temperature will be a mean surface temperature over the area.

In order to verify the temperature data obtained with this measurement method, the steel disc has also been equipped with a thermocouple radially drilled into the disc from the outside. During rapid engagements Holgerson has shown significant differences between the thermocouple and the infrared thermometer [18]. In the present case however, the time of the engagement is longer and therefore the differences in the measured temperature are generally negligible, see Figure 4. The accuracy of the temperature measurements are within ±1.5°C.

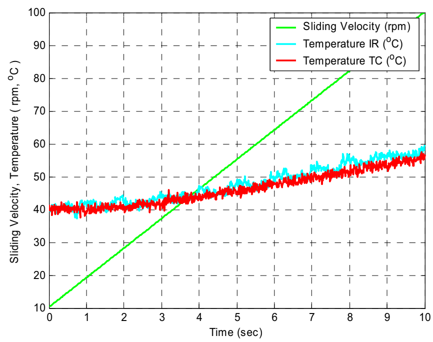

Figure 4. Temperature measurements from a thermocouple inside the clutch disc compared to the temperature on the surface measured from infrared emissions.

THE LIMITED SLIP CLUTCH TEST RIG

The Limited Slip Clutch Test Rig was developed based on ideas generated during the work on the Wet Clutch Test Rig in combination with knowledge of the function of different equipments at Haldex [20]. The main focus in the design process were improved accuracy at low sliding velocities, and to be able to further increase the normal load on the clutch, compared to the previously described equipment.

FUNCTION

An overview of the equipment can be seen in Figure 5. The equipment is mounted on an aluminum stand (1). The base is a rigid beam (2) of length 1600 mm. On the left hand side, the motor with its gearbox (3) is mounted. The driving force is transmitted by a shaft coupling (4), through a hollow piston hydraulic cylinder (5) to the clutch housing (6). The torque transmitted by the coupling is transmitted through a torsion bar (7) to the torque measurement cell (8). The torque measurement cell (8) is connected to the beam (2) by a slide system (9).

A more detailed view is shown in Figure 6. The power is generated by an electric motor (3). The maximum output torque is 500 Nm, and the speed can be varied between 0.5 and 125 rpm (2.5 to 600 mm/sec on the mean radius).

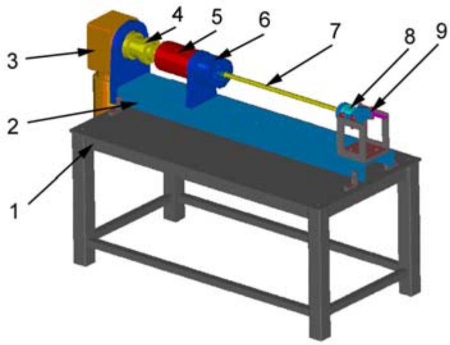

Figure 5. Overview of the Limited Slip Clutch Test Rig.

The torque is transmitted to the driveshaft (10) via a torsionally rigid flexible coupling (4).

The normal force on the test clutch is applied by a double acting hollow piston cylinder (5). In this case we have chosen to limit the normal force to 30,000 N by a pressure limiting valve.

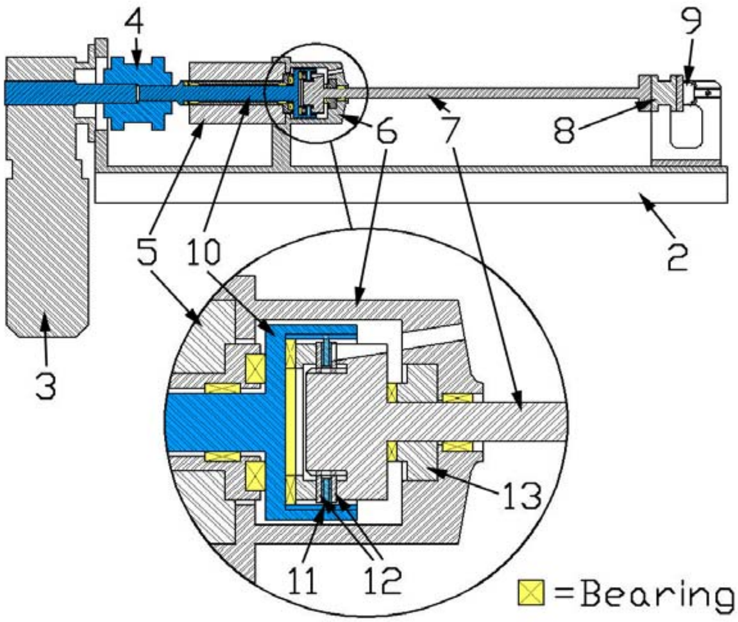

Figure 6. Simplified cross-section of the test apparatus. Shaded parts are rotating during operation.

In the magnified view of the clutch housing in Figure 6, the friction disc (11) and the separator discs (12) can be seen. The friction disc is connected to the driving shaft (10), and the separator discs are connected to the torsion bar (7). During operation, the parts shaded in Figure 6 are rotating.

When a normal force is applied to the clutch by the hydraulic cylinder (5), torque is transmitted from the driving shaft (10) to the torsion bar (7). The transmitted torque is then measured by the torque measurement cell (8). The applied normal force is measured by the load cell (13); this is possible thanks to the slider system (9) which allows the torsion bar and torque cell to move freely in the axial direction. Both the force and torque transducers are full bridge, strain gauge type with built in amplifiers. The accuracy of the measurements from this rig is well within ±1% [21].

Infrared temperature measurements are performed in the same manner as in the Wet Clutch Test Rig, on the friction disc through a hole in one of the separator discs, the torsion bar and the housing. Thermocouples are installed in the oil sump to measure oil bulk temperature.

By changing the length of the torsion bar it is possible to vary the natural frequency of the apparatus in the range from 100 to 500 Hz.

RESULTS AND DISCUSSION

Typical measurement data from the different equipments have been published previously [19, 20, 21, 22], therefore this section will be focused on highlighting and summarizing obtained results. Presented results are obtained using a semi-synthetic transmission fluid with a viscosity of 35 mm^2/s at 40 °C.

TEMPERATURE INFLUENCE ON FRICTION

To determine the friction-temperature behavior of the fluid, a test with constant normal force and constant velocity was performed. The test commenced at room temperature, and the coefficient of friction was monitored at increasing temperatures.
In Figure 7 a typical friction-temperature graph is presented.

It is generally sufficient to fit the friction-temperature data with a straight line within the typical temperature range of operation, i.e.

$$\mu = \alpha \cdot T + \beta$$

In the presented case $\alpha = -0.00033$ and $\beta = 0.130$

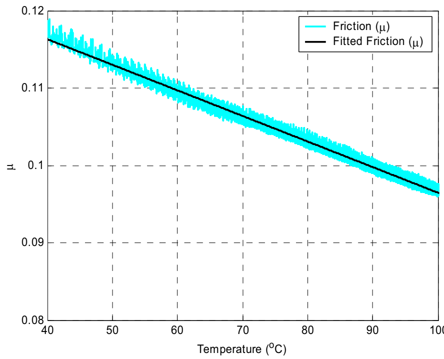

Figure 7: Friction-temperature behavior of a typical transmission fluid.

The general appearance of the graphs looks similar for all tested fluids, but the coefficients of the line, α and β, are dependant on normal load, sliding velocities and fluid type.

Since the sliding velocities are quite low, and the normal load is high, it can be assumed that the operation is in boundary to mixed lubrication regime. Therefore the friction will depend on additive actions. The decreasing friction at higher temperatures is probably a result of increasing additive performance, rather than viscous effects.

FRICTION VS. VELOCITY CHARACTERISTICS

The slope of the μ-v curve has shown to be a good measurement on the fluids ability to suppress vibrations. In Figure 8 a μ-v curve for a well performing fluid is presented, but in this case it must be noted that the temperature is not constant during the test, and will therefore influence the slope of the curve as explained in the next section. The friction variations that may be observed in the figure are a result of imbalance in the test equipment, and does not influence the obtained μ-v curve.

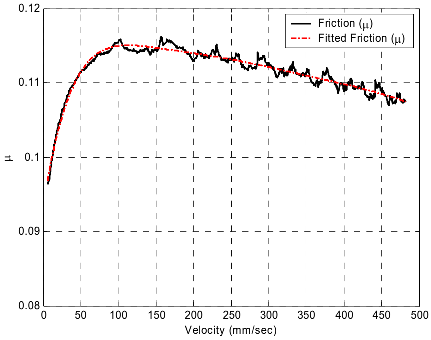

Figure 8: μ-v curve for a transmission fluid tested at 60 °C.

TEMPERATURE COMPENSATION

It is possible to use the known friction-temperature relationship and compensate the μ-v curve in Figure 8 with respect to the increase in temperature during the test. In this case the temperature at the low velocity is around 60°C, while the friction values at 450 mm/sec are measured at approximately 75°C.

In order to take this into account, it is sufficient to keep track of the temperature rise from the start of the test, ΔT, and then for each velocity subtract the contribution from the change in temperature, i.e.

$$\mu_{Compensated} = \mu - \alpha \cdot \Delta T$$

In Figure 9 both measured and temperature compensated friction from three different tests at similar conditions are shown. The temperature compensated friction curve generally displays a more favorable appearance with respect to anti-shudder performance due to less negative slope, especially at higher sliding velocities. The reason that the friction characteristics at a constant temperature are of primary interest is that induced vibrations are of high frequencies, and the temperature during a vibration can therefore be assumed to be constant.

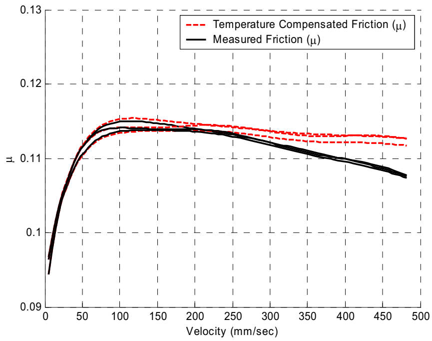

Figure 9: μ-v curves from three different tests run at similar conditions. The dashed curves have been compensated for the change in temperature during the test.

LOAD INFLUENCE ON FRICTION

The normal load will have an influence on the coefficient of friction. Increased normal load generally results in a small decrease in friction; this effect is quite small as compared to that of temperature and sliding velocity.

In many cases, although, it is easy to think that the coefficient of friction decreases with increased load, yet the decrease in friction actually is a result of the increased temperature at the clutch interface.

TEMPERATURE RISE DURING OPERATION

The clutch temperature during operation is increasing with increased power transmitted to the clutch as seen in Figure 10. The rate of temperature increase seems to be independent of normal load and sliding velocity as may be expected from the heat equations.

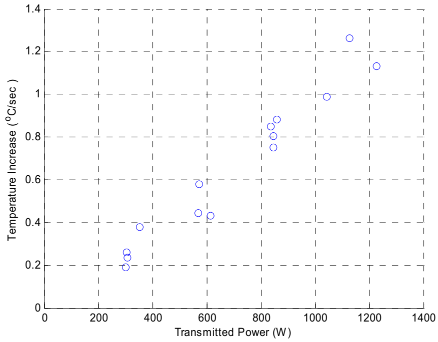

Figure 10: Temperature increase as a function of transmitted power.

ADDITIVES EFFECT ON FRICTION

The effect of various additives on the friction characteristics have been investigated using a Cameron-Plint TE77 reciprocating friction and wear tester [22, 23]. It has been found that the TE77 is useful in screening many different fluids in quite a short time. Some results from this study are presented below.

FRICTION MODIFIERS

Friction modifiers have a positive effect on the anti-shudder performance of the lubricant. Some friction modifiers have only small effects on the friction characteristics, while some friction modifiers have a pronounced effect as seen in Figure 11.

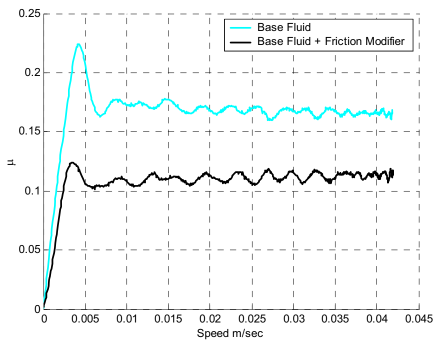

Figure 11: Influence on μ-v characteristics with the addition of a friction modifier to the base fluid.

In this case the addition of 1% by weight of a friction modifier to a mineral base fluid lowers the static coefficient of friction by a large amount (the first peak in the diagram), the drawback is the decreased torque capacity due to lower friction throughout the speed range.

DISPERSANTS

Dispersants are another investigated additive, which some authors claim to have an effect on frictional properties of ATF-fluids in combination with paper-based friction materials [2, 5, 24, 25].

In the presented study no effect of dispersants on the friction characteristics was observed. The dispersant with the largest influence are presented in Figure 12 (1% by weight in a mineral base fluid). In this case a small decrease in the coefficient of friction can be observed, but that should not influence the anti-shudder performance significantly.

The absence of effect from dispersants on friction characteristics can be explained by the use of metal-based friction material instead of paper-based materials.

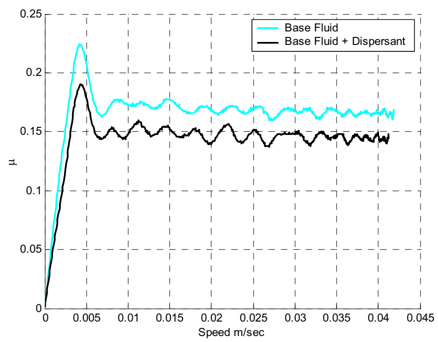

Figure 12: Influence on μ-v characteristics with the addition of a dispersant to the base fluid.

CONCLUSIONS

As stated earlier in this paper, the work has been aimed at carrying out three different tasks; Design of a suitable test equipment, Investigation and verification of different parameters influence on friction characteristics using sintered bronze friction materials and Formulation of suitable transmission fluids.

Obtained results are summarized below:

- The Limited Slip Clutch Test Rig was designed, and it is able to simulate all realistic operating conditions of wet clutches in limited slip differentials.

- Results from the Limited slip clutch test rig have shown good agreement with other equipments, and data published by other authors

- Measurements have shown that the friction decreases with increasing temperature. Similar results have been reported previously for paper-based friction materials

- The μ-v relationship have been shown to be a good measurement of anti-shudder properties, but the temperature increase during measurements influences the results

- By measuring the change in temperature during the test, temperature compensated μ-v curves, corresponding to tests at constant temperature, have successfully been calculated

- The coefficient of friction decreases slightly as the normal load is increased. But the effect is small compared to the influence of sliding velocity and temperature

- The temperature rise is shown to increase linearly with the transmitted power, as suggested by the heat equations

- The influence of a large number of different additives on friction has been established. This knowledge is used in formulating new transmission fluids

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Paper D

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Measurement and characterization of anti-shudder properties in wet clutch applications

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ABSTRACT

Wet clutches operating under low velocity and continuous sliding are increasingly important in vehicle transmissions. This presents new challenges in improving anti-shudder properties of the clutch system.

In order to determine anti-shudder properties the friction-velocity relationship is generally examined. The friction-velocity characteristics are however influenced by the way they are measured, and the measurement methods differ between different standards and OEM approvals, thus making direct comparisons of the anti-shudder requirements difficult.

This study investigates how different parameters influence anti-shudder properties. Results show how the obtained friction-velocity characteristics are influenced by the choice of different test procedures and parameters.

Methods for presenting the anti-shudder properties are compared for the different cases and a method for evaluation of the properties is suggested. Presenting data using the proposed method makes comparison of results obtained from different measurements easier or even possible.

INTRODUCTION

In recent times the use of wet clutches in low velocity and continuous sliding applications has increased. This applies to both automatic transmissions [1, 2, 3], limited slip differentials [4, 5] and all-wheel drive applications [6, 7]. These applications place high demands on the anti-shudder properties of the transmission fluid and the friction material. Studies comparing measurement data for friction characteristics from wet clutch tests have been published but further work in this area is still needed [8].

In limited slip applications featuring wet clutches, the friction characteristics and more especially the anti-shudder performance are of interest. The results from such tests are usually difficult to compare due to varying test conditions, different measurement methods and the way the data is presented.

The most common way of describing the anti-shudder performance is a friction-velocity plot. To obtain such data a test cycle containing a velocity ramp is typically designed. Although a temperature increase occurs during the ramp, in the contact between the surfaces, and even though friction in a wet clutch usually has a strong temperature dependency where the coefficient of friction tends to decrease with temperature [9, 10], this phenomenon is seldom taken into account when friction characteristics are presented. Therefore there is a need for a standardized method of describing the friction characteristics which also takes contact temperature into account. The aim of this paper is to present and verify such a method.

FRICTION REQUIREMENTS

For wet clutches operating under low sliding velocities and high clutch disc pressure it is common that stick-slip and shudder arise, resulting in noise and additional stresses on other parts of the transmission. This behavior has been investigated by a number of authors, both experimentally [11, 12, 13] and theoretically [14, 15, 16].

The results obtained indicate, as previously stated in literature [8, 17], that low static coefficient of friction (μ_s) and a dynamic coefficient of friction (μ_d) that increases with increasing sliding velocity will help in reducing stick-slip induced vibrations. A positive slope of the friction curve will act as a positive damping in the transmission system, and a negative slope will be a negative damping. Therefore less positive slope will present higher demands on damping elsewhere in the system [2]. The static friction coefficient depends on how long the surfaces have been immobile [18, 19], whilst the dynamic friction coefficient depends on relative speed, normal force and temperature between the two surfaces [7, 20].

Typical μ-V relationships from friction tests of different AT fluids are presented in Figure 1 [17]. In order to avoid vibration problems, friction characteristics similar to that of Oil A, with a positive slope in all regions, are desirable. Parameters such as base oil composition, additive composition (including the relative concentration of components) and the friction material have significant influence on the friction characteristics of the system.

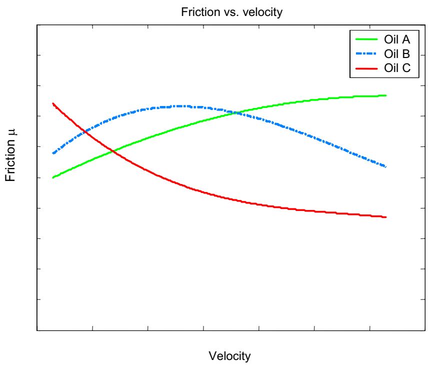

Figure 1: Schematic μ-V relationships for different ATF:s; Oil A will suppress vibrations, while Oil B and Oil C may bring about self-induced vibrations since they present a negative slope in some regions.

METHODS FOR MEASURING FRICTION CHARACTERISTICS

Mercon V

Ford´s ATF-classification, Mercon V, specifies that in order to avoid shudder in the lock-up clutch of an automatic transmission, torque ratios at 2rpm/20rpm and at 40rpm/120rpm >1.0. This means that the friction characteristics must show a positive slope in the friction velocity relationship in order to avoid shudder.

JASO 349:2001

JASO, the Japanese Automobile Standard Organisation, specifies in their anti-shudder test method for anti-shudder performance of ATF:s 349:2001 that: dμ/dv shall be defined for 40°C, 80°C and 120°C by performing a least squares fit of a fifth degree polynomial. dμ/dv at v=0.3m/s and v=0.9m/s shall be multiplied by 1000 and reported together with μ-v graphs. JASO 349:2001 does not quantify shudder. Instead the results test is compared to the test results of a reference fluid.

Other methods of anti-shudder characterisation

Ohtani et al and Devlin et al quantify anti-shudder properties using μ1/μ50 (where μ1 and μ50 are the coefficient of friction at 1rpm and 50rpm respectively) and state that a ratio greater than 1 indicates shudder [1, 21]. Since the friction disc in the paper published by Ohtani et al is about the same size as in this investigation, the sliding velocity is about the same and thus the friction ratio μ1/μ50 is comparable [1]. The μ1/μ50 ratio was also defined for quantifying of anti-shudder by Murakami et al as well as μ50/μ150 [8].

Yoshizawa et al defined the ratio μ1/μ20 for quantifying anti-shudder [22], and Miyazaki et al used the ratio between the friction at 6mm/s and 180mm/s which is equal to 1.2rpm and 37rpm as an indicator of possible shudder [13].

AIM OF THE STUDY

This study investigates different methods to quantify the anti-shudder properties and the influence of the chosen measurement method on the measured friction-velocity relationship. Studied parameters are: normal force; initial velocity; and the effect of chosen ramp time for speed ramp measurement. Also, the differences between positive and negative speed ramps are examined. During all measurements the temperature in the clutch interface is closely monitored in order to determine the temperature influence on the measurements.

The aim of the study is to gain an understanding of how different test conditions influence the obtained friction-velocity data, and based on that knowledge find a way to present the friction characteristics so that the data are not influenced by the chosen test conditions. Presenting data in such a way also makes comparison between results obtained by different test methods easier, or even possible.

EXPERIMENTAL

All data in this study have been obtained using the test equipment, fluid and friction material described in this section.

TEST EQUIPMENT

A clutch friction tester which is able to evaluate frictional characteristics as a function of sliding velocity, normal force, temperature and oil flow has been utilized during this study. The apparatus, called the Limited Slip Clutch Test Rig (LSCTR), is a disc-on-disc type arrangement with two active friction surfaces [7]. The equipment use thermocouples mounted at the back of the separator plate and in the oil sump for temperature monitoring.

General design

An overview of the equipment can be seen in Figure 2. The equipment is mounted on an aluminium stand (1). The base is a rigid beam (2) of length 1600 mm. The motor with its gearbox (3) is mounted on the left hand side. The driving force is transmitted by a shaft coupling (4), through a hollow piston hydraulic cylinder (5) to the clutch housing (6). The torque transmitted by the coupling is transmitted through a torsion bar (7) to the torque measurement cell (8). This in turn is connected to the beam (2) by a slide system (9). Important technical data for the equipment is summarized in Table 1.

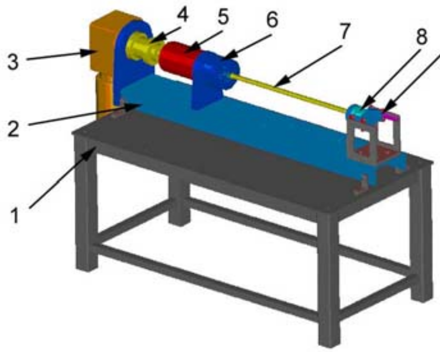

Figure 2: Overview of the test bench.

Table 1: Summarized technical data for the test equipment.

Sliding velocity	m/s rpm	0.005-0.6 0.5-125
Normal force	N	30,000
Allowable torque	Nm	500
Oil circulation flow rate	ml/min	0-1,000
Maximum sample rate	samples/s	200,000
Natural frequency	Hz	100-500

Detailed design

A more detailed view is shown in Figure 3. The power is generated by an asynchronous helical bevel geared servo motor with forced cooling (3). The motor is controlled by a drive inverter with feedback control to obtain accurate drive speed. The maximum output torque is 500 Nm, and the speed can be varied between 0.5 and 125 rpm (2.5 to 600 mm/sec on the mean radius).

The torque is transmitted to the driveshaft (10) via a torsionally rigid all-steel flexible coupling (4). Axial, radial and angular shaft misalignments are compensated for by means of flexible disc packs which are torsionally rigid in the circumferential direction.

The normal force on the test clutch is applied by a double acting hollow piston cylinder (5). In this case we have chosen to limit the normal force to 30,000 N by using a pressure limiting valve. The force can be applied either manually or by using a hydraulic power unit in combination with a feedback controlled pressure valve.

In the magnified view of the clutch housing in Figure 3, the friction disc (11) and the separator discs (12) can be seen. The friction disc is connected to the driving shaft (10), and the separator discs are connected to the torsion bar (7). During operation, the parts shaded in Figure 3 are rotating.

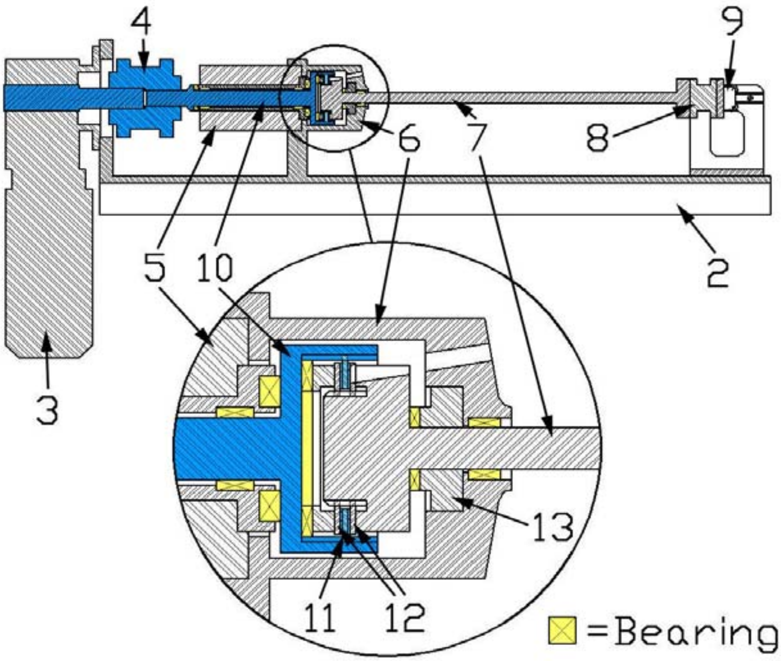

Figure 3: Simplified cross-section of the test apparatus. Shaded parts are rotating during operation.

When a normal force is applied to the clutch by the hydraulic cylinder (5), torque is transmitted from the driving shaft (10) to the torsion bar (7). The transmitted torque is then measured by the torque measurement cell (8). The applied normal force is measured by the load cell (13). This is possible thanks to the slider system (9) which allows the torsion bar and torque cell to move freely in the axial direction. Both the force and torque transducers are full-bridge, strain gauge type with built in amplifiers.

By changing the length of the torsion bar it is possible to vary the natural frequency of the apparatus from 100 to 500 Hz. These frequencies is where shudder is most likely to occur in the clutch system in this investigation.

<u>Accuracy</u>

The test apparatus is equipped with a 16-bit data acquisition card and signal conditioning system from National Instruments. Maximum total sample rate is 200,000 samples/s, but typically 100 samples/s for each channel is adequate.

The design of the equipment ensures that the friction torque (except for the bearing losses, $\mu=0.0050$) is transmitted to the torque measurement cell. The combined error of the torque cell, including measuring bridge and amplifier, is less than 0.25% F.S.

The normal force is measured by a load cell with a combined error, including measuring bridge and amplifier, less than 0.25% F.S.

The thermocouples on the back of the separator discs have a typical error of ±0.67°C.

TEST SPECIMEN

Friction discs and lubricant are of types commercially used in the Haldex LSC AWD-system [7, 23]. The lubricant is a semi-synthetic Statoil LSC fluid designed for this application [24, 25]. It has a viscosity of $35 mm^2/s$ at 40°C and $6.7 mm^2/s$ at 100°C and meets the requirements for fill-for-life service in a full scale application.

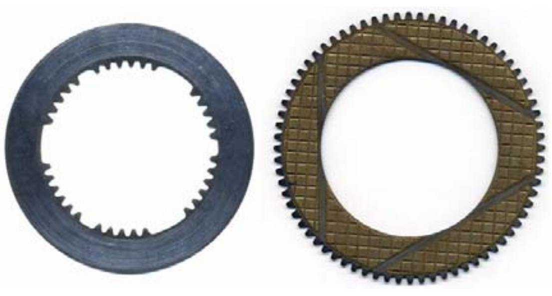

Figure 4: Steel separator disc to the left, and sintered friction disc to the right.

The friction disc pair, Figure 4, features a hardened steel separator plate and a friction disc from hardened steel covered with a 0.5 mm layer of a sintered brass based friction material on each side [26]. The outer diameter of the friction lining is 108 mm, and the inner diameter is 76 mm. The area of contact, when the oil grooves have been accounted for, is approximately 2250 mm^2. The grooves facilitate oil distribution to the area of contact, and help to lower the temperature in the clutch by enhanced oil flow [27].

TEST CYCLE

The test cycles used in this investigation in order to study different parameters are described below. Before every test the clutch discs and lubricant was run-in for 180 minutes.

During the test, the influence of normal force on the friction characteristics was investigated. Five velocity ramps were run at 34 different loads ranging from ~10kN up to 27kN i.e. clutch disc pressures from 3.3MPa to 8MPa. The velocity cycles used are presented in Figure 5. This test was run at a number of different initial temperatures. During the test cycle the sliding velocity was linearly increased from 1rpm to 100rpm over a period of 10s.

A test designed to investigate the influence of chosen ramp time (denoted t_{ramp} in Figure 5) and ramp direction (increasing or decreasing) was performed. In this test the ramp time was altered ranging from 1s to 20s. The friction characteristics were measured both during the velocity increase and the following decrease. During this test series the load was 20kN i.e. a clutch disc pressure of 6.6MPa.

Another test was performed to investigate the influence of initial sliding velocity, denoted N_0 in Figure 5, on friction characteristics. The ramp time was set to 10s and the load to 20kN. During this test series, N_0 was altered between 0rpm and 5rpm.

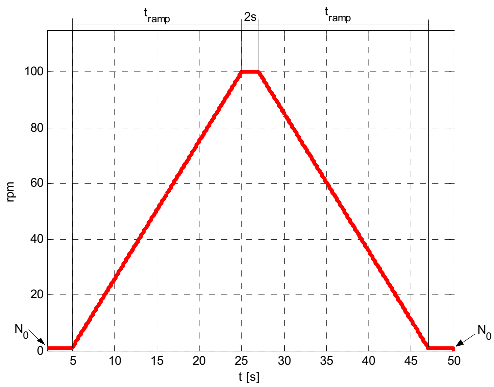

Figure 5: Test cycle; The initial velocity is set to N_0 rpm and held constant for 5s, then the sliding velocity is linearly increased to 100rpm in t_{ramp} s. After 2s at 100rpm, the sliding velocity is linearly decreased to N_0 rpm in t_{ramp} s, before finally being held constant at N_0 rpm for 3s. Each test is started at a specified temperature and the normal force is held constant throughout the test.

During the entire test series the oil flow through the clutch system was maintained constant at 200 ml/min. This is oil flow was recommended by the friction material supplier.

RESULTS

In order to describe the friction-velocity curve, the parameters µ1, µ25, µ50 and so on are used, where µ1 and µ50 are the friction at 1 and 50rpm respectively, refer to Figure 6. These types of parameters have been used previously by several authors [1, 21]. From Figure 6 it is obvious that the defined parameters are influenced by temperature. The temperature will therefore have to be considered.

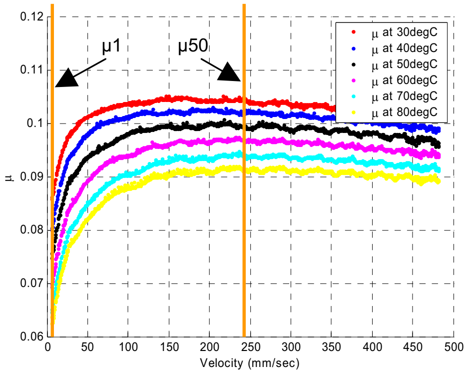

Figure 6: Parameters µ1 and µ50

NORMAL FORCE

As seen in Figure 7, the friction characteristics are quite independent of normal force over the entire range of sliding velocities at 80°C. This holds true for all temperatures down to and including 40°C. However, there is a load dependency at lower loads at lower initial temperatures, typically below 30°C, as shown in Figure 8 for 20°C.

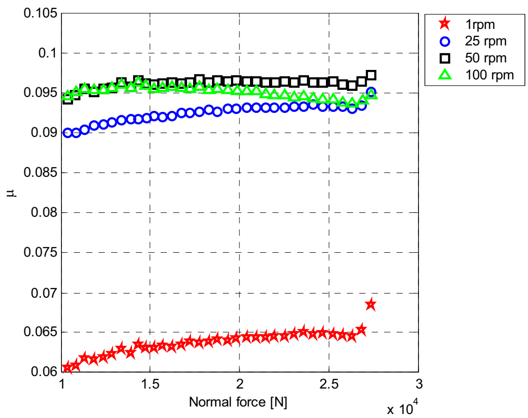

Figure 7: Friction at several different velocities (µ1, µ25, µ50 and µ100) as a function of normal force at 80°C. At the highest load, which also is the maximum load specified by the supplier of the friction material, the friction increase. This could be caused by seizure tendencies.

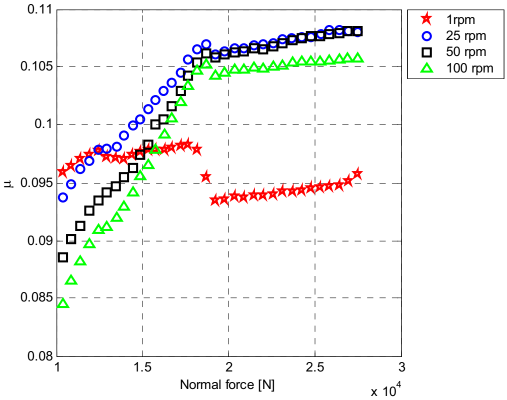

Figure 8: Friction at several different velocities (µ1, µ25, µ50 and µ100) as a function of normal force at 20°C. At ~1.8kN the friction-load relationship display an abrupt change. The cause of this behavior has yet to be explained.

The changing behaviour at temperatures below room temperature in combination with low normal force are probably caused by the fluid/friction material combination. Possibly some additives are not activated under these conditions. This low temperature/load behaviour will need further investigations. As a consequence the results in this paper should be used with caution at temperatures below 30°C.

INITIAL VELOCITY

As results in Figure 9 show, the initial velocity affects the initial torque transfer and thus the coefficient of friction at low sliding velocities, since the normal force is held at a constant level. At higher velocities the friction-velocity relationship is quite similar for different N_0 values.

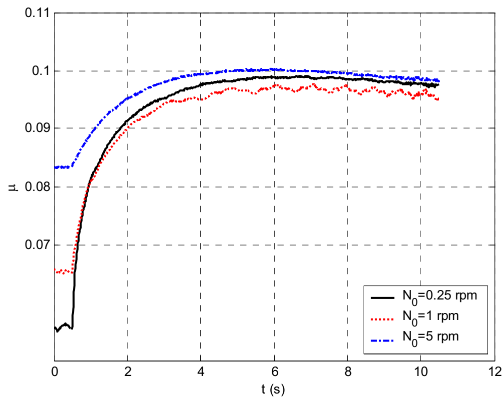

Figure 9: Friction at different initial velocities.

RAMP TIME

Tests were run to investigate the influence of ramp time. Results, seen in Figure 10, show that ramp time affects the friction characteristics. A short ramp time tends to give a smaller slope at velocities over 150 mm/s.

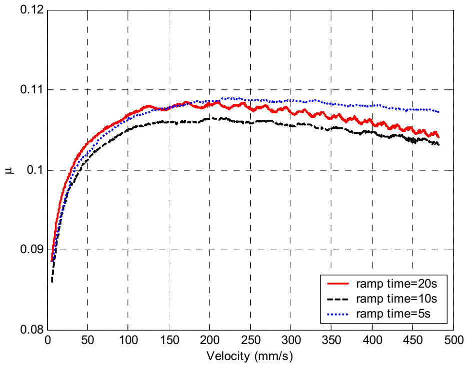

Figure 10: Friction characteristics at different ramp times.

The probable cause of this is that there is a smaller temperature increase in the contact during the short ramp compared to the long ramp. This will affect friction characteristics especially at high sliding velocities since this is where the temperature difference is largest.

POSITIVE/NEGATIVE RAMP

Since the test cycle contains both an increase and a decrease in velocity having the same elapsed time, it is possible to investigate if there is any difference between the two. Results, seen in Figure 11, show that, in general, friction is higher during the increase than during the decrease in velocity. The temperature quoted in Figure 11 is the temperature in the contact at t=0 for the test cycle (i.e. oil sump temperature).

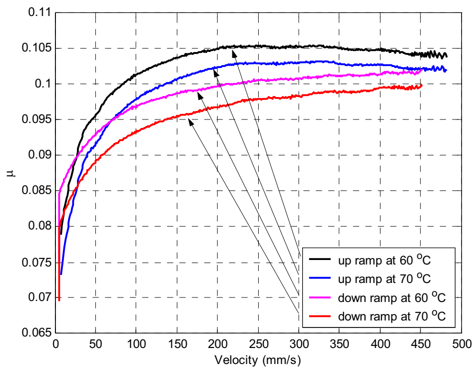

Figure 11: Friction characteristics at different initial temperatures showing positive and negative ramps. The reason that the curves for the same temperatures do not coincide at 482mm/s is the temperature increase during the 2s velocity plateau which can be seen in Figure 5.

Since the actual temperature in the contact is higher during the velocity decrease compared to that during the velocity increase, the friction is lower during the down ramp.

EVALUATION

THEORY

Since the friction characteristics of a wet clutch usually have a strong temperature dependency, the temperature rise during the test cycle has a large impact on friction characteristics. If the results of friction characteristics tests could be made more independent of the chosen test cycle, it would be much easier to quantify and compare tests from different clutch test equipment.

When measuring the friction characteristics the temperature must also be measured, preferably as close to the contact as possible. In this study, thermocouples were attached to the back of the separator plate. The method has been further described and verified previously [28].

When analyzing the measurement data, the friction-temperature relationship (i.e. μ1(T), μ10(T) and so on) are evaluated for a number of velocities, Figure 12. This is based on data obtained from ramps run at a range of different initial temperatures.

By fitting a second degree polynomial to the data at every fifth rpm, using a least squares method, the continuous functions μ1(T), μ10(T) and so on are determined, Figure 13.

From the data in Figure 13 it is possible to, at any temperature within the interval, find the discrete friction velocity relationship, Figure 14. Thus the influence of the contact temperature change during the test has been eliminated since the friction characteristics are now evaluated at a constant temperature.

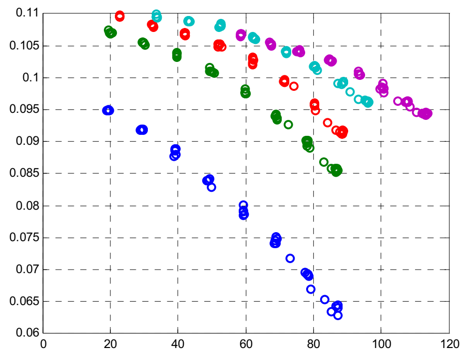

Figure 12: The measured friction-temperature relationship μ1, μ10, μ20, μ50 and μ100, plotted in that order where μ1 display the lowest friction and temperatures, refer to Figure 14.

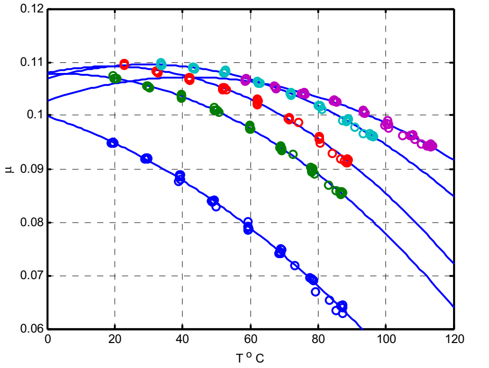

Figure 13: The measured friction-temperature relationships, μ1, μ10, μ20, μ50 and μ100, with least squares fitted quadratic curves.

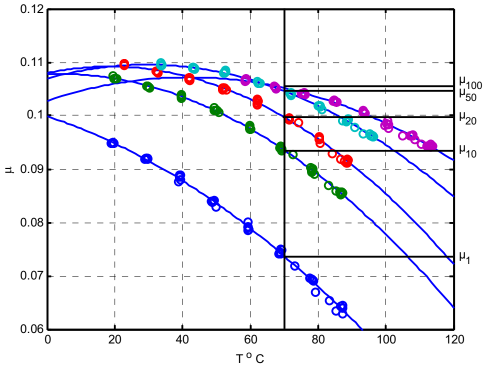

Figure 14: The friction characteristics μ1, μ10, μ20, μ50 and μ100 for a constant temperature are obtained. In this case the values at 70°C are marked in the plot.

From the different friction-temperature curves in Figure 14 it is possible to extract data points for the friction at each velocity for a given temperature and plot these values in a discrete friction-velocity plot, Figure 15.

In reality measurement data is fitted to μ1, μ5, μ10, …, μ100, resulting in more data points compared to Figure 15, yielding in a smoother friction-velocity curve, Figure 18. This discrete friction-velocity plot is not sensitive to temperature changes in the contact zone during the measurements, and therefore less sensitive to varying test conditions.

Which temperature that should be chosen for evaluation depends on the temperature range of the test series. In this investigation the evaluation is made at 70°C since this is in the middle of the evaluated temperature interval. In order to obtain short evaluation time and adequate accuracy of the least squares fit, a residual analysis is in place.

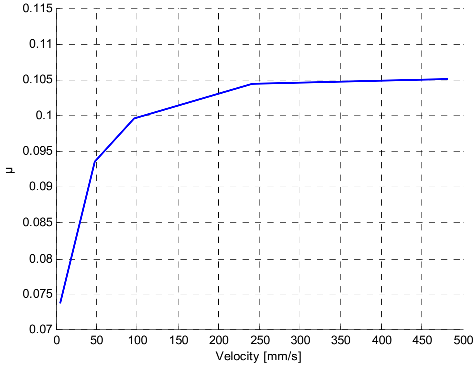

Figure 15. Discrete friction-velocity curve at a constant temperature of 70°C. Constructed from the values marked in Figure 14 with linear interpolation between μ1, μ10, μ20, μ50 and μ100.

NORMAL FORCE

As shown earlier, the coefficient of friction is quite independent of normal load. However there is a difference in temperature rise during the velocity ramp between different loads since the energy input differs.

In Figure 16 the friction at four different velocities is presented as a function of normal force for tests run at an initial temperature of 50°C (i.e. oil sump temperature). In this case the friction at 100rpm (μ100) is lower than the friction at 50rpm (μ50) indicating a negative slope of the friction-velocity curve and therefore possible shudder.

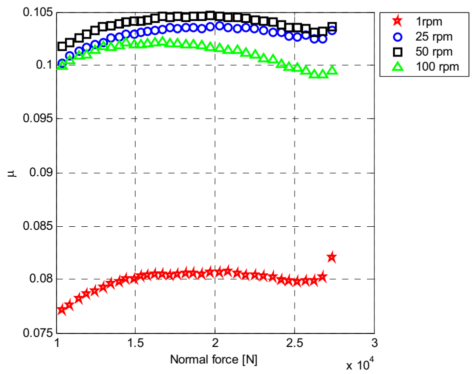

Figure 16: Normal force influence at initial temperature 50°C.

Figure 17 contains the same data as Figure 16, but in this case the data have been normalized to a constant temperature according to the method described in the previous section. For this temperature compensated data the friction characteristics have no negative slope for any velocities at any loads, i.e. the clutch system always contributes with a positive damping to the transmission system.

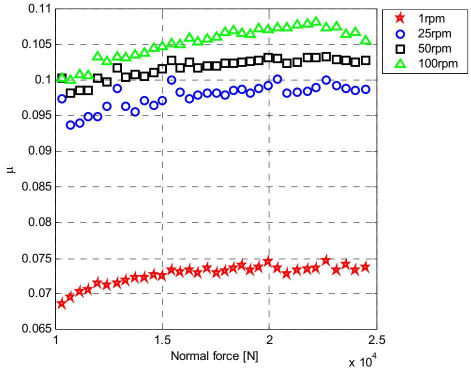

Figure 17: Normal force influence at constant temperature 60°C.

INITIAL VELOCITY

Since the differences between the tested initial velocities were quite low, there is not much difference in temperature increase in the tested range. Therefore there is no need for temperature compensation.

RAMP TIME

The only measurable difference corresponds to the temperature rise during the test cycle. However it must be mentioned that since the sampling frequency in this test is 100Hz, a shorter ramp time presents fewer measured data points. Therefore it can be wise, depending on the test equipment in use, to analyze measurement accuracy before choosing ramp time.

The temperature rise causes a reduction in friction which is ramp time dependent. But as can be seen in Figure 18 (compare with Figure 10), the friction is not influenced by ramp time when the curves have been compensated for differences caused by different temperature increases during the test. The small difference in the fitted curves is believed to be caused by transient responses due to the temperature being measured at the back of the separator plate and not directly in the contact zone.

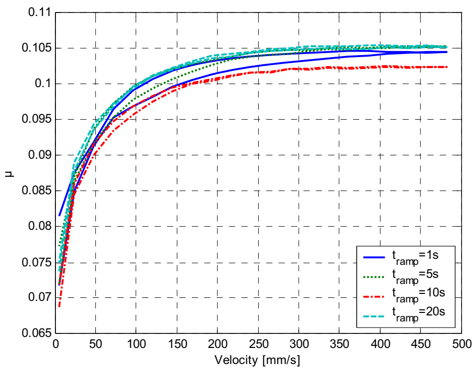

Figure 18: Friction characteristics at constant temperature 70°C at different ramp times.

POSITIVE/NEGATIVE RAMP

This is where the proposed method really comes into play. The friction velocity graph in Figure 19 (as in Figure 11) shows a distinct difference depending on whether the velocity is increased or decreased.

But, as for the ramp time, the temperature rise during the test, rather than the direction of the velocity ramp causes the difference. As Figure 19 shows, the temperature compensated friction-velocity relationship really is direction independent.

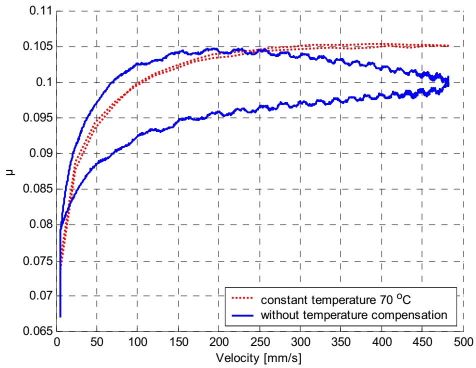

Figure 19: Friction characteristics measured during speed increase and speed decrease, the friction is lower during decreasing velocity than during increasing velocity. Also plotted: The same two curves compensated for temperature differences.

CONCLUSION

Different methods to measure anti-shudder properties have been investigated. Studied parameters are: normal force; initial velocity; and the effect of chosen ramp time for speed ramp measurement. Also, the differences between positive and negative speed ramps are examined.

Methods for presenting the anti-shudder properties are compared for the different cases and a method for evaluation of the properties is suggested. Presenting data using the proposed method makes comparison of results obtained from different measurements easier or even possible.

The conducted study has shown that:

- The choice of test conditions influences the obtained friction-velocity data. This is primarily caused by differences in the contact temperature between the clutch discs. It is therefore important to monitor the contact temperature.

- The friction at constant normal load and constant velocity is a function of temperature. The influence of temperature on friction is significant and should not be overlooked.

- The friction at constant velocity is only to a small extent influenced by the choice of normal load at all temperatures above 30°C.

- The friction-velocity relationship obtained from measuring the friction during an increasing sliding velocity ramp is very different from the friction-velocity relationship obtained during a decreasing sliding velocity ramp.

- A way to present the friction as a discrete function of velocity at a constant temperature so that the data are not influenced by the chosen test conditions has been developed.

- Presenting data using the proposed method makes comparison between results obtained from different test conditions possible.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

AT: Automatic Transmission.

ATF: Automatic Transmission Fluid.

AWD: All-Wheel Drive, or 4-Wheel Drive.

F.S.: Full Scale. Typically measurement accuracy is defined in percent of the maximum reading.

LSC: Limited Slip Coupling.

LSCTR: Limited Slip Clutch Test Rig.

μ1: Coefficient of friction at 1rpm.

μ25: Coefficient of friction at 25rpm.

Paper E

Wet Clutch Transmission Fluid for AWD Differentials; Base Fluid Influence on Friction Characteristics

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SUMMARY

In recent years, several electronically controllable automotive transmission systems, where wet clutches are used as intelligent differentials, have emerged on the market. In this type of application the anti-shudder properties of the lubricants are of vital importance. This paper investigates the influence of base fluids on the anti-shudder properties of transmission fluids for wet clutches in all-wheel drive systems. The investigated all-wheel drive system, featuring a wet multi-plate clutch with a sintered brass base friction material, is described. Test equipment used to determine the frictional characteristics of the transmission fluid is described. Studied parameters include both base fluid type and base fluid viscosity. It is shown that the choice of base fluid has no impact on torque capacity, but that the base fluid influences the temperature dependence of the dynamic friction and the anti-shudder properties. It is also shown that the major effect on the friction characteristics is caused by additive effects rather than base fluid effects.

1 INTRODUCTION

The use of wet clutches working as electronically controlled differentials, as a part of intelligent transmission systems in passenger cars, has increased in recent years [1, 2, 3].

1.1 Haldex Limited Slip Coupling

The Haldex Limited Slip Coupling (LSC) is a limited slip differential that distributes drive torque between front and rear axles of all-wheel drive passenger cars. Under normal conditions (good traction) the car is driven only by the front wheels. When a front wheel loses traction, a speed difference occurs between the front and rear axle of the car. When this occurs the wet clutch on the propeller shaft is engaged in order to transmit drive torque to the rear axle. In Figure 1 a typical drive line equipped with the Haldex LSC is shown. The Haldex LSC is currently used in passenger cars manufactured by Volvo Cars Coorporation and the Volkswagen group. More detailed descriptions of the AWD system can be found in the literature [4, 5].

The low continuous sliding velocities during engagement place significant demands on the anti-shudder properties and durability of the transmission fluid/friction material combination [3, 6].

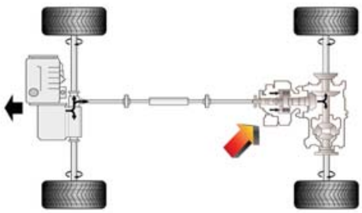

Figure 1: Drive train equipped with the Haldex Limited Slip Coupling.

The friction material used in the Haldex LSC is sintered brass which, in combination with the fill-for-life demand on the product, necessitates the use of a transmission fluid specifically designed for this application. Fluid development has been conducted for a number of years by Statoil Lubricants [3, 7, 8].

In order to achieve the required anti-shudder properties it is important to control the friction characteristics. To prevent shudder the static friction must be lower than the dynamic friction and the dynamic friction should increase with increasing velocity [4, 5, 6].

Both mineral oils and synthetic fluids are commonly used transmission oil base fluids [9, 10, 11].

According to Haviland et al., the base oil viscosity is not that important since it generally does not affect any additive dependent friction characteristics [12]. However, Yamamoto reports that the friction of paper-based wet friction materials increases with increasing viscosity [13].

Shirahama showed that dynamic friction rises with increasing viscosity, but that the addition of succinimide gave a coefficient of friction independent of viscosity [14]. Kitanaka showed that dynamic friction rises and static friction decreases with increasing viscosity, and that the viscosity dependence diminished with the addition of dispersants (alkenylsuccinimide) and detergents (metal sulfonate) [15].

2 AIMS AND METHOD

The aim of this investigation was to study the influence of the base fluid on the friction characteristics in the investigated application. This was felt to be necessary since there is some disagreement between the studies presented in the literature. It was also of interest to verify the findings for sintered friction materials.

In this study, five different fluids were investigated. The additives used were the same in all fluids. Fluid C is a commercially available semi-synthetic LSC fluid developed by Statoil. Fluid A and E are synthetic based fluids combined with the additives used in fluid C. Fluid A is a low viscosity fluid and fluid E is a high viscosity fluid. Fluid B and D are mineral based fluids, again combined with the additives used in fluid C. Fluid B is a low viscosity fluid and fluid D is a high viscosity fluid. Important fluid properties are summarized in Table 1.

Table 1: Important fluid properties.

Fluid	**Base fluid**	**Kinematic viscosity @40°C (cSt)**	**Kinematic viscosity @100°C (cSt)**
A	**Synthetic**	**7.05**	**2.18**
B	**Mineral**	**9.74**	**2.71**
C	**Semi-synthetic**	**35.0**	**6.70**
D	**Mineral**	**58.9**	**8.26**
E	**Synthetic**	**63.1**	**9.71**

All friction tests were performed using friction discs with a sintered brass lining running against separator discs made from hardened steel (hardness ~42HRC). The surface finish of the steel separator discs is much smoother than that of the sintered friction material; typically R_t of the sinter material was more than $10*R_t$ of the steel [16].

2.1 Test Equipment

The friction measurements were carried out using the limited slip clutch test rig shown in Figure 2.

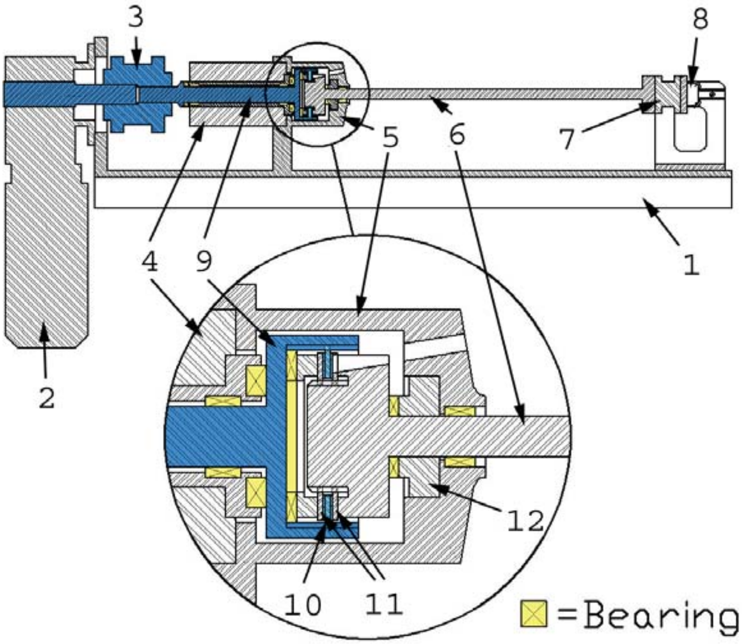

Figure 2: Simplified cross-section of the limited slip clutch test rig.

The base of the rig is a rigid beam {1} of length 1600mm. On the left hand side, the motor with its gearbox {2} is mounted. The driving force is transmitted by a torsionally rigid shaft coupling {3}, through a hollow piston hydraulic cylinder {4} to the clutch housing {5}. Torque is transmitted through a torsion bar {6} to the torque measurement cell {7} which is in turn connected to the beam {1} by a slide system {8}.

The speed can be varied between 0.5 and 125rpm (2.5 to 600mm/sec at mean radius). The normal force on the test clutch is applied by a double acting hollow piston cylinder {4} and was limited to 30,000N by a pressure limiting valve.

In the magnified view of the clutch housing in Figure 2, the friction disc {10} and the separator discs {11} can be seen. The friction disc is connected to the drive shaft {9} and the separator discs are connected to the torsion bar {6}. During operation, the shaded parts in Figure 2 are rotating.

When a normal force is applied to the clutch by the hydraulic cylinder {4} torque is transmitted from the drive shaft {9} to the torsion bar {6}. The transmitted torque is measured by the torque measurement cell {7}. The applied normal force is measured by the load cell {12}; this is possible thanks to the slider system {8} which allows the torsion bar and torque cell to move freely in the axial direction. Both the force and torque transducers are full bridge, strain gauge type with built in amplifiers. The accuracy of the measurements from this rig are well within ±1% [4].

For this investigation, thermocouples were installed in the oil sump to measure oil bulk temperature and in the separator disc to measure friction surface temperature.

More detailed descriptions of this equipment can be found in the literature [4].

2.2 Test Procedure

Before testing each fluid the test equipment was cleaned and new friction plates mounted. After that 200ml of the test fluid was poured into the oil sump. An oil flow of 150ml/min between the clutch discs was maintained during the entire test procedure.

The new friction discs were run in for a period of 15min with varying normal loads. After running in, the system was allowed to cool down to room temperature.

Each test cycle began at a velocity of 25rpm with a normal force of 20kN. The clutch was then run under these conditions until the clutch temperature reaches a specified value (typically 90-100°C) while simultaneously monitoring the friction torque, oil sump temperature and clutch disc temperature. After this temperature was reached, the normal force was removed and the equipment stopped until the temperature fell to that required for the first measurement of the friction characteristics.

When the temperature for the first measurement of the friction characteristics was reached, the test equipment was started at a velocity of 1rpm and an applied normal force of 20kN. After 3sec the velocity was ramped up, reaching 100rpm 10sec later. During this time, friction, velocity and clutch temperature were monitored and logged and used to determine the friction-velocity characteristics. This cycle was repeated five times for each start temperature.

When the friction characteristics had been measured at the first temperature, the system was allowed to cool down a further 10°C and the same procedure repeated. The friction-velocity characteristics were measured at 10°C intervals down to and including 30°C.

3 RESULTS

The measured friction-velocity characteristics for fluid A are presented in Figure 3. The given temperature corresponds to the temperature at which the speed ramp started. It can be clearly seen that friction decreases with increasing temperature. During the velocity ramp, the temperature will also increase and it is this that is responsible for the negative slope of the friction-velocity curves seen at higher velocities.

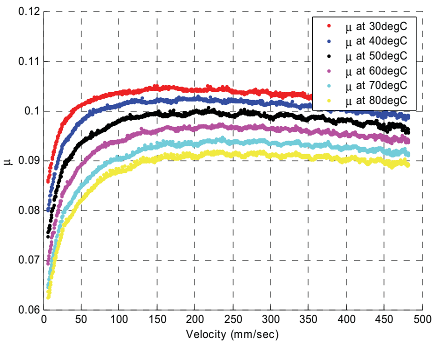

Figure 3: Friction-velocity curves for fluid A at different temperatures. Observe that the temperature value given is only true at low velocities since the fluid temperature will increase during the test.

The influence of temperature can also be seen in Figure 4 where friction as a function of temperature is plotted for constant velocity and normal load (20kN and 25rpm). This phenomena does not influence the results presented here since all data is given as a function of temperature.

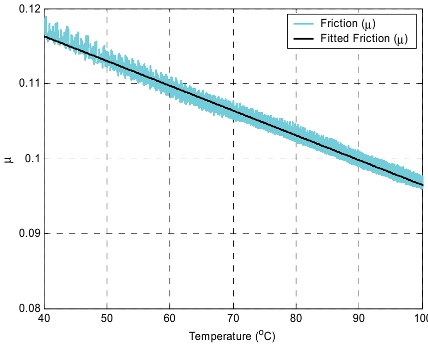

Figure 4: Friction-temperature curve for fluid C obtained under sliding with constant velocity and normal load.

To quantitatively characterise the friction-velocity relationship, the ratio between the friction at 1rpm (4.82mm/sec at the mean radius) and at 50rpm (241mm/sec at the mean radius), μ1/μ50, as defined by Ohtani et al., was used [17]. A value of μ1/μ50 greater than 1 implies that shudder is likely to occur. Ohtani et al. used slightly bigger discs than those used in the present work and therefore had slightly different sliding speeds, however, the principle is the same; the dynamic friction should be less than the static friction for shudder not to occur.

The values of μ1 and μ50 are also useful for describing the torque capacity and dynamic friction respectively.

In order to be able to determine the effects of temperature on the measured friction parameters, μ1 and μ50 are plotted as a function of temperature. In Figure 5 42 observations of μ1 and μ50 from different speed ramps for fluid A are plotted against clutch temperature.

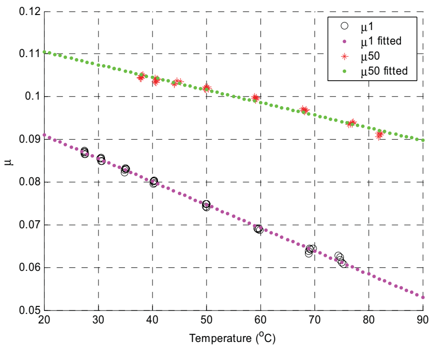

Figure 5: μ1 and μ50 as a function of clutch temperature for fluid A

The data in Figure 5 can be fitted with a linear equation of the form μ=α*temperature+β. The coefficients α and β for all the tested fluids are given in Table 2 for μ1 and Table 3 for μ50.

Table 2: Values of μ1 fitting parameters

Fluid	Temperature dependence, α	Value @0°C, β
A	**-5.42e-4**	**0.1018**
B	**-5.22e-4**	**0.1036**
C	**-4.45e-4**	**0.1004**
D	**-5.78e-4**	**0.1046**
E	**-5.37e-4**	**0.1021**

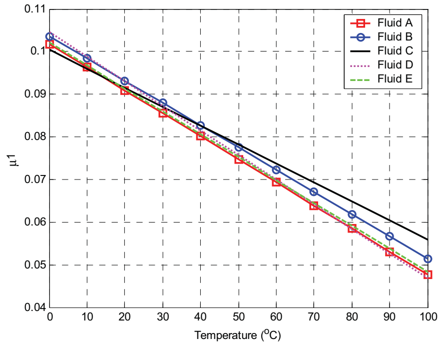

Figure 6: Friction-temperature curves for the tested fluids at 1rpm, from Table 2.

From data in Table 2 and Figure 6 it can be seen that the differences in μ1 are negligible for the fluids tested. No influence of the base fluid viscosity or type can be detected with respect to μ1.

The only fluid that shows a slightly different behaviour in Table 2 is the semi-synthetic fluid C that displays a smaller temperature dependence compared to the other fluids. This is likely to be due to the fact that the additive formulation used has been optimized for this particular semi-synthetic base fluid.

Table 3: Values of μ50 fitting parameters

Fluid	Temperature dependence, α	Value @0°C, β
A	**-2.96e-4**	**0.1164**
B	**-2.63e-4**	**0.1175**
C	**-1.72e-4**	**0.1108**
D	**-2.10e-4**	**0.1118**
E	**-2.06e-4**	**0.1083**

From data in Table 3, and Figure 7 it can be seen that the temperature dependence of μ50 is strongly influenced by the base fluid viscosity. The low viscosity fluids A and B have larger temperature dependence, i.e. higher dynamic friction at lower temperatures.

The mineral oil base fluids seems to give a slightly higher value of dynamic friction compared to the synthetic fluids, but this observation must be investigated further.

Again, fluid C displays slightly different behaviour with respect to μ50, showing the lowest temperature dependence of all tested fluids. This implies that the commercial package gains performance from the optimization of the additive/base fluid combination.

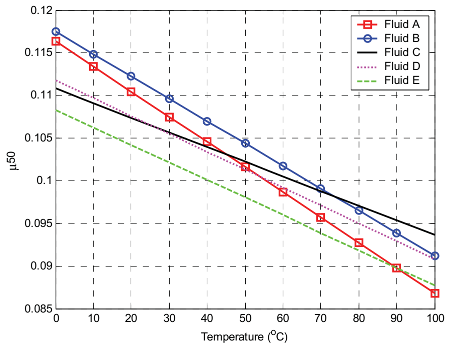

Figure 7: Friction-temperature curves for the tested fluids at 50rpm, from Table 3.

Table 4 shows friction data and viscosity for the fluids at 30°C. This again shows that µ1 is not influenced by the choice of base fluid, and that the low viscosity fluids have a slightly higher dynamic friction, µ50. This could be due to the low viscosity fluids supporting a thinner film because of their lower film forming capacity in combination with increased fluid flow through the friction material.

Table 4: Fluid properties at 30°C.

Fluid	Kinematic viscosity @30°C (cSt)	µ1	µ50	µ1/µ50
A	9.35	0.0861	0.1078	0.7985
B	13.3	0.0885	0.1099	0.8051
C	53.0	0.0875	0.1058	0.8269
D	97.2	0.0878	0.1057	0.8309
E	101	0.0865	0.1023	0.8455

From the data in Table 4 it is also possible to conclude that the low viscosity fluids have better anti-shudder properties, µ1/µ50, compared to the high viscosity fluids. This is primarily the result of the difference in temperature dependence of µ50 compared to µ1.

Table 5 shows friction data and viscosity for the fluids at temperatures corresponding to a kinematic viscosity of 10cSt. This table reveals that all friction parameters are significantly influenced by changes in temperature. Both µ1 and µ50 are lowered, and the anti-shudder properties are improved, as the temperature is increased. Since the viscosity at the chosen temperature is 10cSt for all fluids, the change in friction observed should be the result of changes in additive behaviour.

Table 5: Fluid properties at the temperature corresponding to a kinematic viscosity of 10cSt.

Fluid	Temperature @10cSt (°C)	µ1	µ50	µ1/µ50
A	27	0.0877	0.1087	0.8069
B	39	0.0838	0.1075	0.7791
C	81	0.0648	0.0971	0.6680
D	92	0.0520	0.0927	0.5611
E	98	0.0500	0.0883	0.5660

4 CONCLUSIONS

Five different transmission fluids, with different base fluids and the same additive package, have been tested in order to determine the influence of base fluid on friction characteristics.

The coefficient of friction at 1rpm, µ1, has been used as a measurement of the torque capacity. The coefficient of friction at 50rpm, µ50, has been used as a measurement of the dynamic friction. The ratio between these coefficients, µ1/µ50, has been used to quantitatively characterise the anti-shudder properties of the fluids.

The results obtained show that:

- Both the torque capacity and dynamic friction are largely affected by the clutch temperature.
- Neither base fluid type, nor base fluid viscosity, has any significant impact on the torque capacity µ1.
- The tested low viscosity base fluids show a greater temperature dependence on dynamic friction, µ50, compared to the high viscosity fluids.
- The tested low viscosity fluids have better anti-shudder properties compared to the high viscosity fluids, especially at lower temperatures.
- The friction characteristics are mainly additive dependant, since large differences in the friction parameters can be observed from tests run at temperatures corresponding to a given viscosity.
- Synthetic and mineral base fluids show similar behaviour while the semi-synthetic commercial fluid had better performance in some areas, implying that the additive formulation used does not work as well in a different base fluid than originally intended.

5 ACKNOWLEDGEMENTS

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Paper F

Wet Clutch Transmission Fluid for AWD Differentials; Influence of Lubricant Additives on Friction Characteristics.

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SUMMARY

In recent years, several electronically controllable automotive transmission systems using wet clutches as intelligent differentials have emerged on the market. These applications place great demands on the anti-shudder properties of the transmission fluids used.
The aim of this study was to investigate the influence of different additives on the friction characteristics of a transmission fluid for all-wheel drive systems featuring wet multi-plate clutch with a sintered brass base friction material and, based on this knowledge, to formulate a new transmission fluid with the desired frictional properties. In addition to excellent anti-shudder properties, the new fluid was required to lubricate hypoid gears under high load thus making it possible to run the limited slip clutch and the differential in the same oil reservoir. To meet this requirement it is necessary to add significant amounts of extreme pressure additives to the base oil, which are known to have an unfavourable influence on anti-shudder properties, necessitating the adoption of novel additive technologies.
The additives studied include anti-wear additives, friction modifiers, corrosion inhibitors, detergents, antioxidants and extreme pressure additives. The paper shows how different additives affect friction in different ways and that the interactions between different additives are important to consider. It was concluded that it is feasible to combine good anti-shudder properties for wet clutches with good lubrication of hypoid gears.

1 INTRODUCTION

The use of electronically controlled wet clutch differentials in intelligent passenger vehicle transmission systems has increased in recent years [1-3].

Due to low continuous sliding velocities during engagement, this kind of application places significant demands on the anti-shudder properties and durability of the transmission fluid/friction material combination [3, 4]. In this study, the development of a new transmission fluid for the Haldex Limited Slip Coupling (LSC) is described. The new fluid was required not only to lubricate the clutch but also the hypoid gears in the rear axle which require good lubrication under extreme pressure conditions.

The Haldex Limited Slip Coupling is a limited slip differential that distributes drive torque between the front and rear axles of all-wheel drive passenger cars. Under normal conditions (ie. good traction) the car is only driven by the front wheels. When a front wheel looses traction, a speed difference occurs between the front and rear axles of the car. When this occurs the wet clutch on the propeller shaft is engaged in order to distribute drive torque to the rear axle. Figure 1 shows a typical drive line equipped with the Haldex LSC.

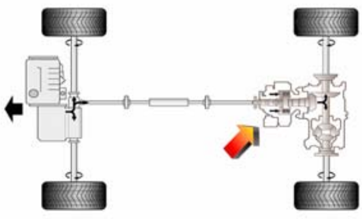

Figure 1: Drive train equipped with the Haldex Limited Slip Coupling.

The Haldex LSC is currently used in passenger cars manufactured by the Volkswagen group, Volvo Cars Corporation and Ford Motor Company amongst others. More detailed descriptions of the AWD system can be found in the literature [5, 6].

The friction material used in the Haldex LSC is sintered brass which, in combination with the fill-for-life demands of the product, necessitates the use of transmission fluids designed specifically for the application in hand. Fluid development has been conducted for a number of years by Statoil [3, 7, 8].

In order to achieve the desired anti-shudder properties it is important to control the friction characteristics of the clutch. To prevent shudder, the static friction must be lower than the dynamic friction. The dynamic friction should also increase with increasing velocity since this gives positive damping in the system. Limited regions with an inverse relationship between dynamic friction and velocity can be tolerated assuming adequate damping is present elsewhere in the transmission [4-6, 9, 10].

Base fluids are known to influence the expected life of the fluid, due primarily to their resistance to oxidation [11]. Choice of base fluid will also influence the thermal properties of the lubricant [8].

An earlier study of base fluids showed that low viscosity fluids gave better anti-shudder properties than high viscosity fluids, especially at lower temperatures. However, neither base fluid type, nor base fluid viscosity had any significant impact on torque capacity. It was also concluded that the frictional characteristics were mainly additive dependant, since large differences in the friction parameters were seen in tests run at temperatures giving comparable viscosities [12].

In limited slip differentials the lubrication regime is boundary and mixed lubrication, meaning that the friction torque is transmitted by boundary layers formed by additives and in some cases the base fluid. The concentration of each additive, and the balance between them, is thus very important when formulating a high performance transmission fluid [9, 13, 14].

2 AIMS AND METHOD

The aim of this investigation was to develop a transmission fluid for the Haldex Limited Slip Coupling which was able to lubricate both the clutch system and the hypoid gears in the rear axle. The fluid was required to give good anti-shudder properties and maintain those properties in “fill-for-life” service.

As already mentioned, the choice of base fluid has only a minor effect on anti-shudder properties. A synthetic PAO base fluid was used which had good low temperature performance. The base fluid also had a high viscosity index which made it possible to use less polymer based VII (viscosity index improvers) additives. This increased the fluid life since shearing of VII additives is known to limit the working life of lubricants in clutch applications. In addition, PAOs are non polar in nature, making the interaction between the base fluid and boundary lubricating additives less troublesome. The kinematic base fluid viscosity was 25cSt at 40ºC, and 5cSt at 100ºC.

After the base fluid had been chosen, single additive tests in a reciprocating friction and wear tester were carried out and the friction characteristics measured. With only a single additive in the base fluid it was not possible to do tests in the clutch test rig due to stick-slip problems.

More complex, multi-additive formulations were then tested in a clutch test rig. These combinations used additives found to have either a minor or a positive influence on the friction characteristics of the clutch. Different additives were added until a fully formulated fluid was obtained.

All friction tests were performed using a sintered brass friction material running against hardened steel (hardness ~42 HRC). The surface of the steel was much smoother than that of the sintered friction material; the R_t of sinter material was typically $10*R_t$ of the steel [15].

2.1 Test Equipment

Single additive tests were carried out using a reciprocating friction and wear tester, type TE77, from Plint and Partner [16]. A schematic view of this is shown in Figure 2. The upper specimen moves with a reciprocating motion with a stroke of 2.7 mm. The applied load was varied between 100 N and 200 N and the frequency between 2.5 Hz and 10 Hz.

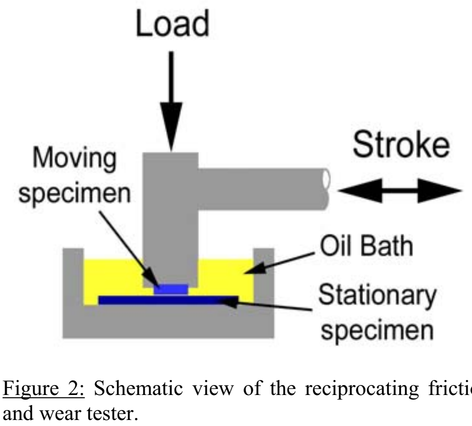

Figure 2: Schematic view of the reciprocating friction and wear tester.

The temperature of the fluid bath was controlled with tests being performed at temperatures between 25 and 115°C. The friction force was measured on the stationary fluid holder using a piezoelectric load cell.

The friction measurements using more complex additive combinations were carried out using the limited slip clutch test rig shown in Figure 3 [6]. The base of the rig is a rigid beam (1) of length 1600 mm. On the left hand side, the motor with its gearbox (2) is mounted. The driving force is transmitted by a torsionally rigid shaft coupling (3), through a hollow piston hydraulic cylinder (4) to the clutch housing (5). Torque is transmitted through a torsion bar (6) to the torque measurement cell (7) which is connected to the beam (1) by a slide system (8).

The speed can be varied between 0.5 and 125 rpm (2.5 to 600 mm/sec at mean radius). The normal force on the test clutch is applied by a double acting hollow piston cylinder (4) and was limited to 30,000 N by a pressure limiting valve.

In the magnified view of the clutch housing seen in Figure 3, the friction disc (10) and the separator discs (11) can be seen. The friction disc is connected to the drive shaft (9), and the separator discs connected to the torsion bar (6). During operation, the shaded parts in Figure 3 are rotating.

When a normal force is applied to the clutch by the hydraulic cylinder (4), torque is transmitted from the drive shaft (9) to the torsion bar (6). The transmitted torque is then measured by the torque measurement cell (7). The applied normal force is measured by the load cell (12); this is possible thanks to the slider system (8) which allows the torsion bar and torque cell to move freely in the axial direction. Both the force and torque transducers are full bridge, strain gauge type with built in amplifiers. The accuracy of the measurements from this rig are well within ±1% [6].

During this investigation, thermocouples were installed in the oil sump to measure bulk oil temperature and in the separator disc to measure friction surface temperature.

To measure the friction characteristics, the test equipment starts at a velocity of 1rpm and applies a normal force of 20kN. After 3sec the velocity starts to increase, reaching 100rpm 10sec later. During this speed ramp, friction, velocity and clutch temperature are monitored and logged. Using this data the friction-velocity characteristics can be determined.

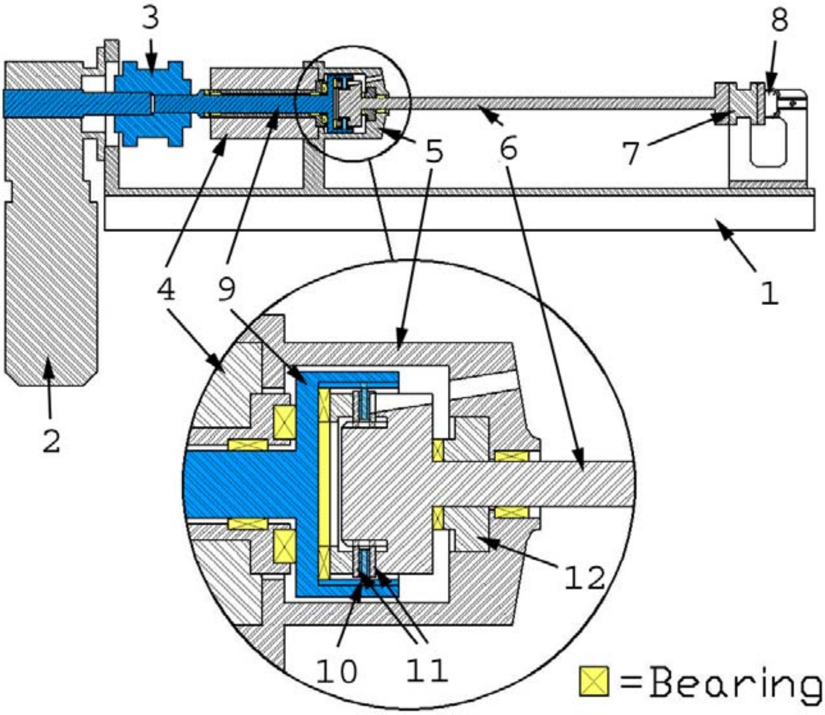

Figure 3: Simplified cross-section of the Limited Slip Clutch Test Rig.

The measured friction-velocity characteristics for a typical transmission fluid are shown in Figure 4. The temperature given corresponds to the temperature at which the speed ramp is started. It can be seen that the friction decreases with increasing start temperature. During the increase in velocity, the temperature will also increase causing the negative slope of the friction-velocity curves at higher velocities. It is therefore very important to keep the clutch temperature in mind when analysing measurement data [17].

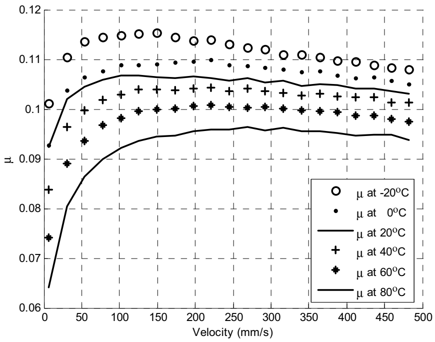

Figure 4: Friction-velocity curves for a typical transmission fluid at different starting temperatures. Note that the temperature given is valid only at low velocities since the fluid temperature will increase during the test.

3 RESULTS

3.1 Single Additives

Single additive tests were carried out in the reciprocating test machine in order to study their friction characteristics.

While some of the additives used, such as antioxidants, dispersants and detergents, were found to have only minor influence on friction other more surface active additives showed significant influence as showed in Figure 5, Figure 6 and Figure 7.

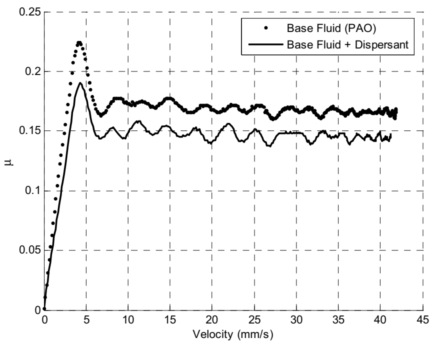

Figure 5: Comparison between base oil and base oil mixed with 1wt% dispersant. Experimental conditions: 70°C 150N 6Hz.

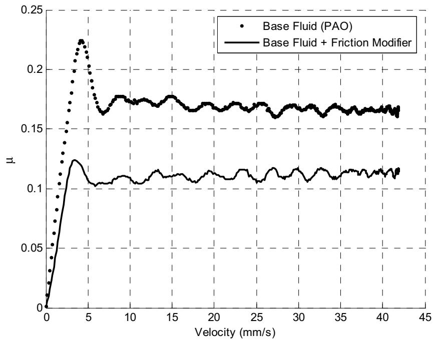

Figure 6: Comparison between base oil and base oil mixed with 1wt% friction modifier. Experimental conditions: 70°C 150N 6Hz.

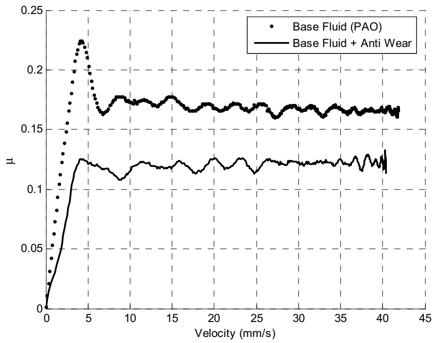

Figure 7: Comparison between base oil and base oil with 1wt% anti wear. Experimental conditions: 70°C 150N 6Hz.

The performance of the additives tested were ranked based on the ratio between the static friction (peak value) and the dynamic friction; with a lower value being desirable. In addition to this, a high dynamic friction was sought in order to increase the torque capacity of the clutch.

The most promising additives were selected for further evaluation in combination with other additives in the next stage of development.

3.2 Additive Combinations

The second stage of the development work evaluated interactions between additives. This stage was divided into three steps, Table 1. At each step one type of additive was introduced; step one rust and corrosion inhibitors, step two anti wear additives and step three, extreme pressure additives. Selection at each step was made by comparing the friction-velocity curves as well as the temperature dependence of the friction.

The initial formulation was a synthetic base fluid with antioxidant, friction modifier and detergent additives. The antioxidant and the detergent did not affect the anti-shudder performance significantly during single additive testing, but are needed to increase the working life of the fluid.

The friction modifier improved the anti-shudder performance significantly and the best friction modifier from the single additive testing was selected as the basis for the new formulation.

Table 1: Composition of tested additive combinations.

Designation	Antioxidant	Friction Modifier	Detergent	EP	Rust & Corrosion	AW	Step
Fluid 1A	✓	✓	✓		✓ Type CI A		1
Fluid 1B	✓	✓	✓		✓ Type CI B		
Fluid 1C	✓	✓	✓		✓ Type CI C		
Fluid 1D	✓	✓	✓		✓Type CI D		
Fluid 2A	✓	✓	✓		✓ Type CI A	✓Type AW A	2
Fluid 2B	✓	✓	✓		✓ Type CI D	✓Type AW A	
Fluid 3A	✓	✓	✓	✓Type EP A	✓ Type CI D	✓Type AW A	3
Fluid 3B	✓	✓	✓	✓Type EP B	✓ Type CI D	✓Type AW A	
Fluid 3C	✓	✓	✓	✓Type EP C	✓ Type CI D	✓Type AW A	
Fluid 3D	✓	✓	✓	✓Type EP D	✓ Type CI D	✓Type AW A	
Fluid 3E	✓	✓	✓	✓Type EP E	✓ Type CI D	✓Type AW A	

The results from step 1 can be seen in Figure 8. Fluid 1A and fluid 1D were chosen as the starting formulations for step 2. Fluid 1A showed the best friction characteristics and Fluid 1D had an attractively high friction which, in addition, was less sensitive to changes in temperature compared to the other fluids.

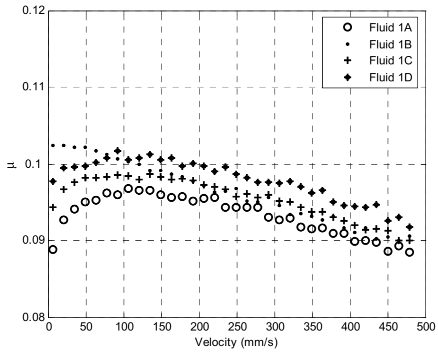

Figure 8: Friction characteristics for fluids in step 1. Load 20kN and initial temperature 50ºC.

The results from step 2 can be seen in Figure 9. For the next step fluid 2B was chosen since Fluid 2A produced stick-slip in some cases. Fluid 2B gave high friction and did not show any sign of shudder, even though the negative slope of the friction-velocity curve was quite high.

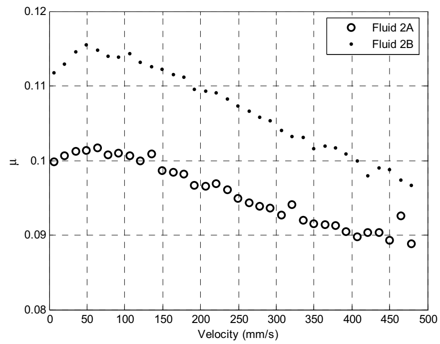

Figure 9: Friction characteristics for fluids in step 2. Load 20kN and initial temperature 50ºC.

The results from step 3 can be seen in Figure 10. Fluid 3B clearly showed the best friction characteristics. Fluid 3A, 3D and 3E showed severe stick-slip at all tested temperature and fluid 3C showed stick-slip in some cases and had less favourable friction characteristics.

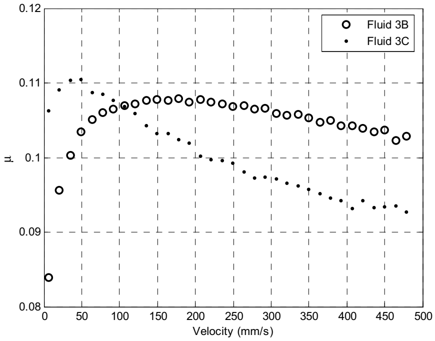

Figure 10: Friction characteristics for fluids in step 3. Load 20kN and initial temperature 50ºC.

3.3 Performance of the fully formulated fluid

The results of a comparison between fluid 3B with a commercial Statoil LSC fluid at an initial temperature of 50ºC are shown in Figure 11.

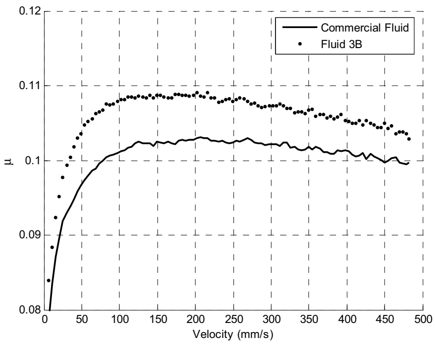

Figure 11: Comparison between fluid 3B and commercial fluid. Load 20kN and initial temperature 50ºC.

It can be seen that Fluid 3B generates higher friction than the commercial fluid, however, the commercial fluid shows a slightly less negative slope at higher velocities.

Figure 12 shows the friction as a function of temperature at 1rpm (torque capacity) and in Figure 13 friction as a function of temperature is shown for 50rpm (dynamic friction). Both fluids can be seen to have low temperature dependency at high sliding speeds while the friction at low velocities is more significantly influenced. Fluid 3B has a higher temperature dependence compared to the commercial fluid especially at low velocities and at low temperatures.

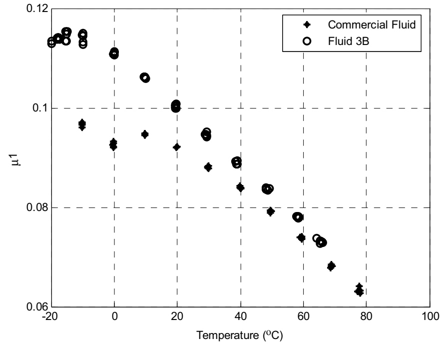

Figure 12: Friction at 1rpm for commercial fluid and fluid 3B. Applied load was 20kN.

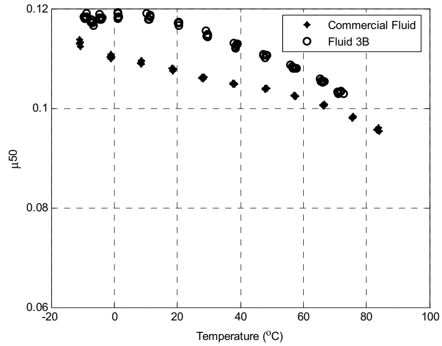

Figure 13: Friction at 50rpm for commercial fluid and fluid 3B. Applied load was 20kN.

While the commercial fluid is an off-the-shelf fill-for-life product, the working life characteristics for fluid 3B need further investigation. Life-time tests have started and initial results show that the fluid needs further improvements in order to maintain its frictional properties in a fill-for-life application. This work is an ongoing but, naturally, a time consuming task. It is, however, believed that commercial fill-for-life requirement can be achieved while still maintaining good frictional properties of the fluid.

4 CONCLUSIONS

The aim of this study was to investigate the feasibility of formulating a transmission fluid combining anti-shudder properties for use in limited slip clutches (with a sintered brass friction material) with hypoid gear lubrication ability.

The results obtained have shown that:

- Both the torque capacity and dynamic friction are largely affected by the choice of additives.
- Additives investigated, such as anti oxidants, dispersants and detergents, generally had a small influence on friction, with some occasional exceptions.

- Surface active additives such as friction modifiers, anti wear and extreme pressure had a large influence on friction.

- The effects of additive interactions must be considered but are hard to predict. This was most notable with extreme pressure additives which were hard to combine with other additives without compromising the anti-shudder properties of the clutch.

- The developed fluid shows that it is feasible to combine anti-shudder properties and hypoid gear lubrication abilities in one fluid. Good anti-shudder properties were achieved despite high concentrations of extreme pressure additives

- Further optimization of the fluid with regard to fluid life is needed.

5 ACKNOWLEDGEMENTS

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Paper G

THERMAL INFLUENCE ON TORQUE TRANSFER OF WET CLUTCHES IN LIMITED SLIP DIFFERENTIAL APPLICATIONS

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ABSTRACT

Wet clutches operating under low velocity and high load are studied with the aim of obtaining reliable models for the torque transfer during boundary lubrication conditions. A friction model which takes temperature, speed and nominal pressure into account is developed and used with temperature calculations to be able to simulate behavior of a wet clutch working in boundary lubrication regime. Predicted torque and temperatures from the model agree well with experimental data.

INTRODUCTION

Disk-type wet clutches are exceedingly important for controlling torque transfer in modern drive trains for both two- and four-wheel drive passenger cars. Modern drive trains are generally equipped with on-board computers that make it possible to predict the vehicle dynamics in real time and control vehicle handling by changes in the differential settings. Implementation of this system requires good theoretical or semi empirical models that are capable of accurately predicting torque transfer as a function of the actuator signal at any given operating condition.

The common simulation approach is to study an engagement from a high velocity to a state of lock-up. Typically the Reynolds equation is used to predict the torque and film thickness at the beginning of the engagement and then a measured friction characteristic is applied to the boundary friction at the latter part of the engagement. In order to obtain good agreement with experimental data, it is necessary to include an appropriate thermal model to compensate the change of fluid viscosity during the engagement. The boundary friction, however, is generally not compensated for temperature [1-4].

Traditional engagement models are generally not suitable for application to limited slip differential (LSD) applications. An LSD-clutch is generally engaged at a rather low velocity, making the contribution of the full-film Reynolds equation less important. The common approach of using flow factors in the model can not be applied since the oil film thickness is so small that the flow factors are not valid [5]. In addition, it is not always desired to reach a state of lock-up, but rather to allow a controlled limited slip, hence transferring only a certain given torque. Since this period of limited slip can be long, a significant amount of heat might be generated, making it important to use a temperature dependent model for the boundary, or asperity, friction.

The aim of this work is to develop a technique suitable for modeling thermal behavior and torque transfer for wet clutches in LSD-applications, in particular the Haldex LSC (Limited Slip Coupling) all-wheel drive system [6, 7]. The Haldex LSC features an electronically controlled disc-type wet clutch mounted between the propeller shaft and the rear differential. The clutch controls the torque transfer to the rear axle of the vehicle, and is actuated by hydraulic pressure generated by a hydraulic pump driven by the speed difference between the front and rear axles of the vehicle. In order to obtain a high power density while still keeping the cost down, a sintered bronze friction material is used in the friction discs in combination with separator discs made of hardened steel, see Figure 1. The lubricant used in this application is a tailor made semi-synthetic fluid with a special additive formulation.

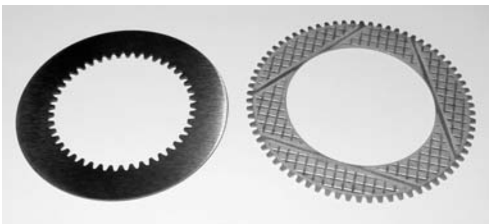

Figure 1. Separator- and friction disc.

Thermal data and geometries for the discs can be found in Table 1 as well as data for the lubricant. Typical operating conditions are sliding velocity less than 0.5 m/s and mean surface pressure exceeding 5 MPa.

APPROACH

Initially the wet clutch model of Jang and Khonsari [2], with a somewhat modified boundary friction model, was applied to investigate an LSD clutch. The boundary friction model was,

$$\mu = 0.15 + 0.011 \cdot \log\left(\frac{0.038 + 0.056}{2}\Delta\omega\right) \quad (1)$$

where $\Delta\omega$ is the difference in rotational speed between friction disc, ω_{fl}, and separator disc, ω_{sd}, respectively. The four constants in Equation 1 were obtained by curve-fitting experimental data from the Limited Slip Clutch Test Rig [7]. Input data were modified to represent the investigated clutch system, including the oil which was a Statoil LSC fluid. A drawback with this model is the values at very low velocities where the friction value will approach $-\infty$. However, at velocities of interest during engagement simulations (above ~10^{-5} rad/s), this factor does not influence the results.

Figure 2 shows resulting torque transmission obtained from a clutch engagement simulation, the respective contribution from full film (Reynolds) and asperity friction is also displayed.

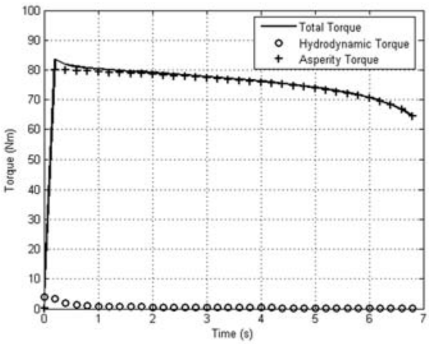

Figure 2. Simulated torque response during engagement from 3000 rpm at 1.25 MPa surface pressure.

It was found that the contribution from the Reynolds equation to the transmitted torque is negligibly small at the end of the engagement. In limited slip applications the system primarily operates in boundary lubrication regime with a corresponding time of 2-3 seconds and higher after the start of engagement in Figure 2. Therefore the torque contribution from Reynolds equation can be neglected.

Figure 3 shows measured friction data obtained at several different initial clutch temperatures obtained for the same fluid tested in the Limited Slip Clutch Test Rig [7]. As evident from Figure 3, for the working conditions in an LSD application, a boundary friction model only depending on velocity is not sufficient because friction is significantly influenced by the heat generated in the clutch during operation. Therefore a better boundary friction model is needed.

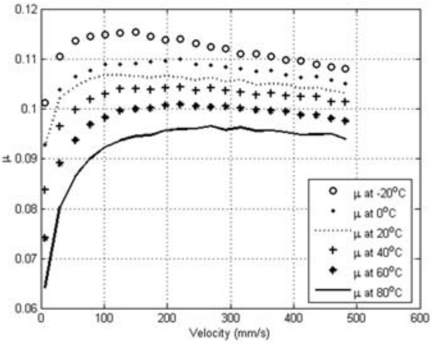

Figure 3. Measured asperity friction at different contact temperatures.

Based on these findings a new approach was adopted. The suggested model features a semi-empirical boundary friction model that calculates the friction as a function of sliding velocity, temperature, and applied pressure. This friction model is connected to a thermal model that predicts the temperature in an LSD clutch based on current operating conditions and the temperature history of the clutch.

Thermal Model

The thermal model considers heat dissipation in the fluid as well as heat conduction into the separator disc, friction lining, and core disc. The computation domain is the axisymmetric cross section presented in Figure 4. In this figure, the separator and core disc are half the thickness of the discs used in the clutch. Since the simulated friction discs are considered to be located in a clutch consisting of several similar discs, the heat conduction over the edges in axial direction is neglected due to symmetry [8, 9]. This means that the outer edges in z-direction on the core and separator disc at $z = 0$ and $z = Z_{sd} + Z_{fl} + Z_{cd}$ are considered as insulated and Neuman boundary conditions, can be applied:

$$\frac{\partial T}{\partial z} = 0 . \quad (2)$$

The boundaries in radial direction at R_{in} and R_{out} will have Dirichlet boundary conditions with a constant temperature at R_{in} and an oil sump temperature, T_{sump}, at R_{out}. T_{sump} will change during engagement according to Equation (7).

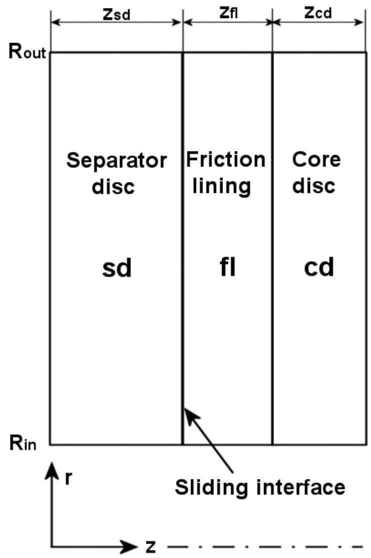

Figure 4. Schematic sketch of calculation domain.

A model was developed to simulate a wet clutch under limited slip conditions, i.e. when the surfaces of the friction lining and the separator disc are in contact with each other. The gap between the friction lining and the separator disc is, therefore, very small during the simulations, and there will not be a large oil flow through the gap. However, even though the oil flow is small, it is not insignificant. Much of the generated heat in the interface between friction lining and separator disc is not absorbed in the materials but transferred out of the clutch by the oil flow. In the computations there is also a certain amount of oil along the outer circumference (see Figure 4), which represents the oil sump. This absorbs the heat transferred by the oil and gives a better boundary condition on the outer edge of the clutch discs than a more simple constant temperature boundary condition. The oil in the oil sump is assumed to be perfectly stirred.

Temperature is solved in the clutch by taking advantage of the axisymmetric condition of the problem. In the separator disc, friction disc and core disc, respectively, the temperature is solved with the heat equation in polar coordinates,

$$\rho C_p \frac{\partial T}{\partial t} = k\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{\partial^2 T}{\partial z^2}\right], \quad (3)$$

where density ρ, specific heat capacity C_p and thermal conductivity k have different values for different parts of the clutch according to Table 1. The friction lining, which is made of bronze, is considered as impermeable so there is no heat convection in the friction material.

Table 1. Input data. Material constants, and geometry parameters. Subscripts according to Figure 4.

Variable	Value	Unit
ρ_{sd}	7600	kg/m^3
ρ_{fl}	5000	kg/m^3
ρ_{cd}	7600	kg/m^3
ρ_{oil} (40°C)	868	kg/m^3
$C_{p\text{-}sd}$	449	J/kgK
$C_{p\text{-}fl}$	5000	J/kgK
$C_{p\text{-}cd}$	449	J/kgK
$C_{p\text{-}oil}$	2040	J/kgK
k_{sd}	46	W/mK
k_{fl}	15.7	W/mK
k_{cd}	46	W/mK
k_{oil}	0.131	W/mK
z_{sd}	$0.75\cdot10^{-3}$	m
z_{fl}	$0.56\cdot10^{-3}$	m
z_{cd}	$0.55\cdot10^{-3}$	m
R_{in}	0.0381	m
R_{out}	0.0564	m
h_{mean}	$1\cdot10^{-5}$	m
V_{sump}	$500\cdot10^{-6}$	m^3

For each time step, the temperature in the friction discs changes. The heat flux, Q_{int} (W/m^2), generated in the interface between the friction lining and the separator disc at a given radius is

$$Q_{\text{int}} = r\Delta\omega p\mu \quad (4)$$

where r is radius to the contact, Δω is difference in rotational speed between the two discs, p is nominal pressure in the interface and μ is friction coefficient in the contact. Q_{int} is not constant for different values of the radius r. This is the case not only because of the difference in radius, but also the different friction coefficient because the temperature is not constant in radial direction. The total energy in interface for each time step is a summation of heat flux over the total contact area as

$$W_{gen} = \Delta t \int_{R_{in}}^{R_{out}} Q_{\text{int}}\, dA\,, \quad (5)$$

where Δt is the time for one time step. The absorbed energy in clutch discs for the same time is described by

$$W_{abs} = \Delta T \cdot \rho V C_p \, , \tag{6}$$

where ΔT is difference in temperature over the total calculated domain in clutch discs for one time step and V is volumes associated with calculation nodes in the domain. The difference between the energies described in Equation 5 and 6 is assumed to warm up the oil in the oil sump as

$$T_{sump} = T_{sump-old} + \frac{\left(W_{gen} - W_{abs}\right)}{\rho_{oil} V_{sump} C_{p-oil}} \, , \tag{7}$$

which gives the boundary temperature at R_{out}.

The boundary conditions between the friction lining and core disc are:

$$T_{fl} = T_{cd} \, ; k_{fl} \frac{\partial T_{fl}}{\partial z} = k_{cd} \frac{\partial T_{cd}}{\partial z} \tag{8}$$

which implies an energy equilibrium over the boundary.

The heat balance in the interface-layer between the friction lining and the separator disc is given by the following equation.

$$rp\omega\mu - k_{sd} \frac{\partial T}{\partial z_-} + k_{cd} \frac{\partial T}{\partial z_+} - \rho c_p u_{oil} r \frac{\partial T}{\partial r} = 0 \tag{9}$$

which is a combination of heat generation, Equation 4, conduction in materials, similar to Equation 8, and forced convection in fluid film,

$$\rho_{oil} C_{p-oil} u_{oil} r \frac{\partial T}{dr} \tag{10}$$

The convection in the fluid film gives a cooling effect to the interface. Heat conduction in the fluid is much smaller than the conduction in the friction discs and can thus be neglected. The velocity of the oil, u_{oil}, is computed with a mean film thickness, h_{mean}, and an oil flow, Q_{oil}, in the interface as

$$u_{oil} = \frac{Q_{oil}}{h_{mean} 2\pi r} \, . \tag{11}$$

The energy equation (3) is solved with an explicit finite difference method of the first order. The transient frictional heat source is computed with a friction coefficient according to the boundary friction model described below. This means that the friction is varying during the whole engagement process since the temperature and rotational speed is changing. It is also possible to let the friction coefficient vary with radius r because temperature in radial direction in the interface will be computed. When comparing the simulations with the tests from the test rig, it was obvious that the cooling oil flow in the interface had a great significance. However, only the total oil flow in wet clutch can be measured in the test rig. It is not possible to measure the oil flow in the interface, and the friction discs have a groove pattern, see Figure 1, which is not axisymetric. Between the friction lining and the separator disc, a forced oil flow, Q_{oiltot}, of 200 ml/min is pumped. Most of this oil is flowing through the grooves, but it is the oil flow in the interface where the disc surfaces are in contact that is of real interest. This flow stands for most of the total convection and has to be described for the simulation model to give a good correlation with measured results. In the simulations it was found that the oil flow in the interface was dependent on $\Delta\omega$ and temperature and that a small fraction of the total flow, Q_{oiltot}, were flowing between the rough friction disc surfaces even with zero rotation. To get an oil flow in the interface which gives a good correlation with simulations and test results, an empirical flow model depending on velocity and temperature is developed:

$$Q_{oil} = Q_{oiltot} \left(a \cdot \left(\Delta\omega + 1 \right)^b + c \cdot T_{mean}^d + e \right) \tag{12}$$

This model is valid only for the geometry used in this application and has to be recalibrated for other geometries of the friction discs, meaning the values of a-e is system dependent.

Boundary Friction Model

The boundary friction is modeled as a function of load, velocity and temperature based on an extensive amount of experimental data. The data is analyzed according to the method described in an earlier publication [10], generating friction-velocity profiles for different loads and temperatures. The following expression has been curve fitted to experimental data,

$$\mu = C_1 \tanh\left(C_2 \cdot v\right) + C_3 v^{0.1} + C_4 \tag{13}$$

where μ is friction and v is sliding velocity, similar models are commonly used as friction models for boundary and mixed lubrication [11]. C_1 is connected to the friction value at the point where the friction starts to level out. C_2 is used to adjust the curve with respect to the x-axis, i.e. velocity. C_3 governs the slope of the curve at higher velocities and C_4 is used in order to adjust the friction level, i.e. shift the friction curve up or down.

Figure 5 shows measured data points and the resulting fitted curve for one load case, load 20 kN and temperature 70ºC.

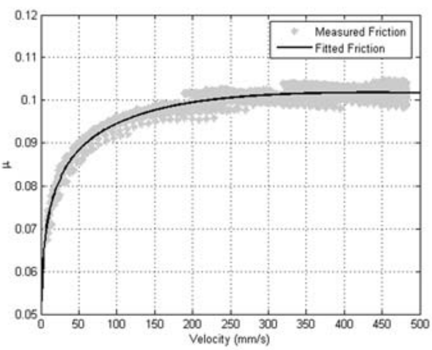

Figure 5. Observed values of friction and the fitted friction model at load 20kN and temperature 70ºC.

The fitting parameters from Equation 13, for different loads and temperatures, are stored in a matrix, Table 2. The matrix spans loads from 10 kN to 25 kN and temperatures from -40ºC to +200ºC. The friction is calculated using a linear interpolation between the friction values generated by the parameter sets surrounding the current operating condition.

Table 2. Example of values from the data matrix containing information on boundary friction fitting parameters.

	10 kN	15 kN	20 kN	25 kN
-40ºC	C_1=-0.031 C_2=0.0014 C_3=0.057 C_4=0.035	C_1=-0.026 C_2=0.0037 C_3=0.035 C_4=0.070	C_1=-0.0025 C_2=0.013 C_3=0.012 C_4=0.098	C_1=-0.0024 C_2=0.0098 C_3=0.012 C_4=0.101
…	…	…	…	…
70ºC	C_1=-0.024 C_2=0.0024 C_3=0.070 C_4=-0.010	C_1=-0.338 C_2=0.0001 C_3=0.077 C_4=-0.025	C_1=-0.039 C_2=0.001 C_3=0.077 C_4=-0.024	C_1=-0.038 C_2=0.002 C_3=0.089 C_4=-0.036
…	…	…	…	…
200ºC	C_1=-0.024 C_2=0.0024 C_3=0.070 C_4=-0.038	C_1=-0.027 C_2=0.0021 C_3=0.087 C_4=-0.067	C_1=-0.092 C_2=0.0006 C_3=0.097 C_4=-0.082	C_1=-0.038 C_2=0.002 C_3=0.089 C_4=-0.063

The different parameter sets are primarily based on a total of 1,750 friction measurements where the velocity is linearly increased from 1 rpm to 100 rpm (0.006 m/s to 0.59 m/s) in 10 s using the Limited Slip Clutch Test Rig [7]. This type of data is available from -20ºC up to 90ºC for all loads.

At higher temperatures (above 90ºC) the fitting parameters C_1, C_2 and C_3 are constant (the shape of the curve is constant and same as at 90ºC), the parameter C_4 is however modified to adjust the friction level. This adjustment is based on friction measurements conducted at a constant velocity of 100 rpm where the temperature is increased up to 200ºC under a constant load, yielding a friction-temperature dependence (the decrease in friction is $2.1\cdot10^{-4}$ per degree Celsius). These measurements were also conducted using the Limited Slip Clutch Test Rig.

At low temperatures (below -20ºC) the fitting parameters C_1, C_2 and C_3 are also constant (the shape of the curve is constant and same as at -20ºC), the parameter C_4 is however modified to adjust the friction level. This adjustment is based on friction measurements conducted on an actual Haldex Coupling.

Figure 6 illustrates the friction-velocity output from the friction model for a case with load 20 kN (3.8 MPa) and temperature 70ºC.

The friction is assumed constant at velocities in excess of the measured 0.59 m/s and for velocities less than 0.006 m/s. The friction coefficient for a velocity of 0.59 m/s is used for velocities higher than 0.59 m/s and for velocities less than 0.006 m/s the friction coefficient for the velocity 0.006 m/s is used.

Figure 7 shows an example of the friction as a function of temperature for an arbitrary case, load 20 kN (3.8 MPa) and velocity 0.05 m/s (~8 rpm). Around 20ºC an alteration in the curve can be observed, this is in agreement with experimental observations and are believed to be caused by additive effects in the fluid. At temperatures below and in excess of the observed -40ºC and 200ºC friction is assumed constant and at the same value as at -40ºC and 200ºC respectively. Between -40ºC and -20ºC and between 90ºC and 200ºC the friction characteristics (friction-velocity) have the same shape but the friction curve is shifted compared to the friction at -20ºC and 90ºC, respectively, as described above ($-2.1\cdot10^{-4}$ 1/ºC).

At normal loads under 10 kN and above 25 kN the friction function will return to the same value as at 10 kN and 25 kN respectively.

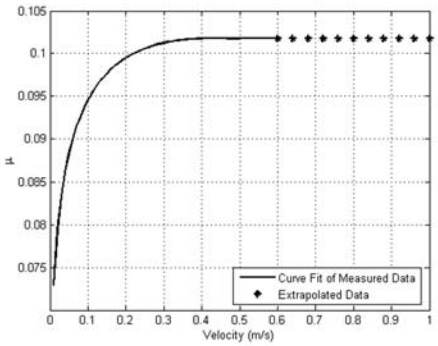

Figure 6. Friction characteristics generated by the friction model at load 20 kN and temperature 70ºC.

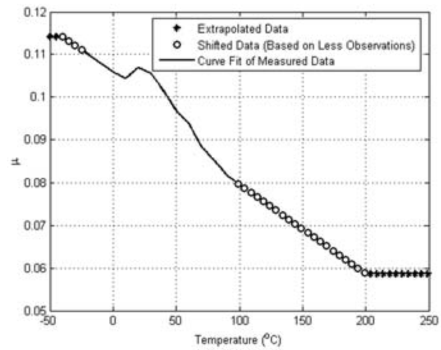

Figure 7. Friction at different temperatures generated by the friction model at load 20 kN and velocity 0.05 m/s.

Solution Technique

Engagement force, sliding velocity, and initial temperature are assumed to be known parameters and are given as input to the model. These parameters can be easily measured on-board the vehicle.

The energy equation (3) is discretized on an axisymmetric grid.

The friction in each sliding grid point is given by the boundary friction model at each time step based on velocity, pressure, and temperature. The generated heat is calculated based on the friction and velocity and is used in the thermal model to predict the temperature in the next time step. Based on this temperature, a new friction value is calculated. Finally, the transmitted torque is calculated by integration of the friction force over the clutch area.

RESULTS

A wet clutch used in an LSD application is working with high applied force and low difference in rotational speed between separator disc and friction lining, $\Delta\omega$. The behavior of a wet clutch in an LSD is tested in the test rig described in earlier publications [7]. Common tests for wet clutches used in LSD applications are carried out with a constant axial force during the whole experiment. During the tests, $\Delta\omega$ is often linearly increased from a rather low value during a couple of seconds or linearly decreased from a higher value. A combination of these tests can also be carried out, where $\Delta\omega$ first is linearly increasing from a low value, held constant a couple of seconds, followed by being linearly decreased to the lower start value. In this paper two cases with different loads are investigated.

- Case 1: $\Delta\omega$ is linearly increased from 1 to 100 rpm in 10 seconds.
- Case 2: $\Delta\omega$ is constant 1 rpm during 5 seconds, linearly increased from 1 to 100 rpm in 10 seconds, held constant at 100 rpm during 2 seconds, linearly decreased from 100 to 1 rpm in 10 seconds, and finally held constant at 1 rpm during 3 seconds.

The maximum difference in rotational speed in these cases, 100 rpm, gives a maximum surface velocity of 0.59 m/s. Temperature can increase up to 40°C in some cases. Start temperature in the clutch can vary from -25°C to 100°C, depending on the working temperature investigated. The investigations in Figure 8-13 describe the normal working conditions with temperatures during the tests between about 75°C to 110°C.

Figure 8 describes Case 1 with a relative small axial force, 10.4 kN. The start temperature in this test is 77°C and the small axial force will lead to a quite low torque transfer, about 45 Nm, which gives a small heat generation. This gives a temperature of about 90 degrees in the warmest parts of the clutch after 10 seconds. In this figure the difference between the measured and simulated torque and temperature is very small.

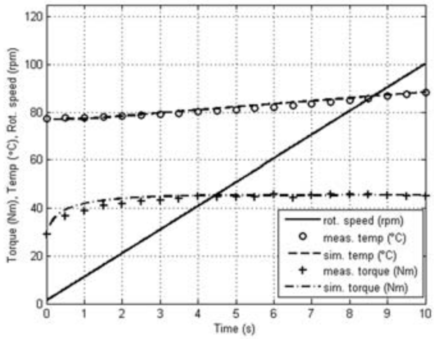

Figure 8. Comparison between measurements and simulation of Case 1 with an axial force of 10.3 kN.

The temperature in the computational domain described in Figure 4 during engagement is shown as contour plots in Figure 9 to Figure 11. The temperature distribution is shown for 3, 6 and 9 seconds of engagement in Figure 9-Figure 11, respectively. As expected, the temperature is rising during the whole engagement process. It is also shown that the maximum temperature is located in almost the same position in all figures and just slightly moves towards the outer radius when the oil sump is slowly heated. One more interesting observation in these figures is that the maximum temperature is not located where the temperature is measured in the test rig. The thermocouple is located in about $z = 0.5$ mm and $r = 48$ mm. This location is marked with a black point in the contour plots. The temperature, shown in Figure 8, Figure 12 and Figure 13 is the temperature in this location. As the contour plots show, the measured temperature in the test rig is not the maximum temperature in the clutch and the temperature in the sliding interface is far from constant. This combined gives a small error for the temperature dependency in the boundary friction model. Since the measured temperature only can be regarded as a mean temperature in the wet clutch, also the friction coefficient can be regarded as a mean friction coefficient. However, in the temperature model, the friction coefficient is computed individually at each discrete node. Therefore, the knowledge of more local friction behavior would have been of great interest to achieve more accurate solutions.

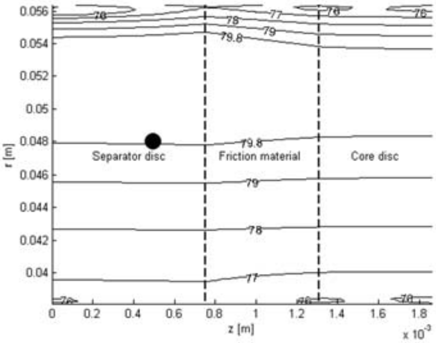

Figure 9. Simulated temperature for Case 1 with an axial force of 10.3 kN after 3 seconds.

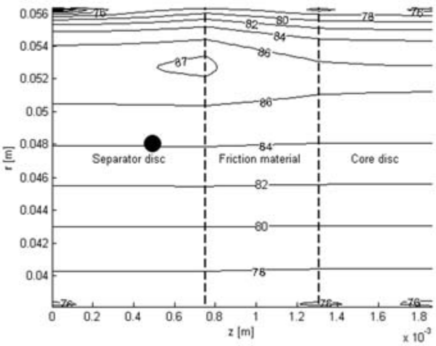

Figure 10. Simulated temperature for Case 1 with an axial force of 10.3 kN after 6 seconds.

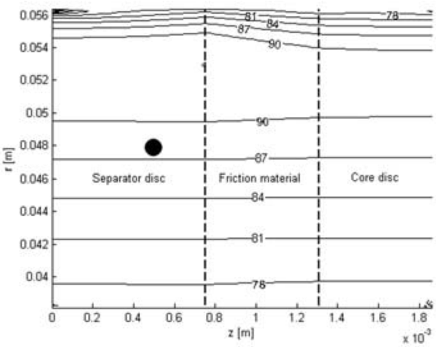

Figure 11. Simulated temperature for Case 1 with an axial force of 10.3 kN after 9 seconds.

Figure 12 shows a comparison between simulated and measured data for Case 1 with an axial load of 25.3 kN. When comparing Figure 12 with Figure 8, it is obvious how the applied force affects the torque and therefore the heat generation in the sliding interface. In this case the transferred torque is over 110 Nm and the temperature will rise to almost 110°C. Also in this case with much higher load, the difference in temperature and torque between simulations and measurements is not very large.

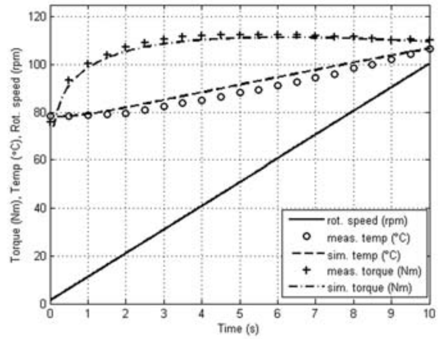

Figure 12. Comparison between measurements and simulation of Case 1 with an axial force of 25.3 kN.

Figure 13 shows a comparison between simulations and measurements for Case 2 with an axial load of 14.8 kN. In this case the difference between simulated and measured temperature and torque is larger than the simulations of Case 1. This gives a simulated torque which is lower than the measured torque in the part with increasing $\Delta\omega$ and higher torque than the measured torque in some parts with decreasing $\Delta\omega$.

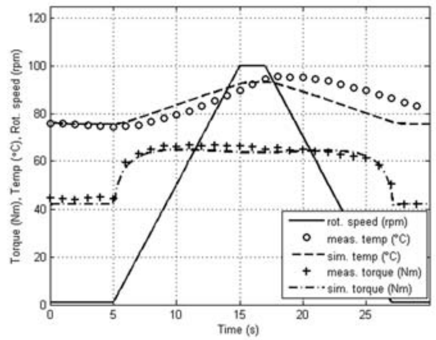

Figure 13. Comparison between measurements and simulation of Case 2 with an axial force of 14.8 kN.

All figures show a torque behavior which is rapidly changing with $\Delta\omega$ when $\Delta\omega$ is low. On the other hand, when $\Delta\omega$ is higher than about 40 rpm, the torque is not very sensitive for changes in $\Delta\omega$. Change in torque transfer during these conditions with $\Delta\omega > 40$ rpm is very much depended on the difference in temperature, and this makes it important to take temperature changes into account when simulating the torque transfer in wet clutches working under boundary lubrication condition. Figure 14 shows a comparison between the torque for a constant temperature and that for simulated temperature. Also shown is the measured torque for Case 1 with a axial load of 25.3 kN. This is the same case as in Figure 12. The difference in torque transfer in this clutch, consisting of just one sliding interface, is quite large. The torque for the case with constant temperature also shows a rather different behavior than the torque for the simulated temperature, which is decreasing when the temperature is rising. This can be very important to take into account when simulating vehicle dynamics, since it can change the behavior of the LSD and the whole vehicle drive train, when working under limited slip conditions.

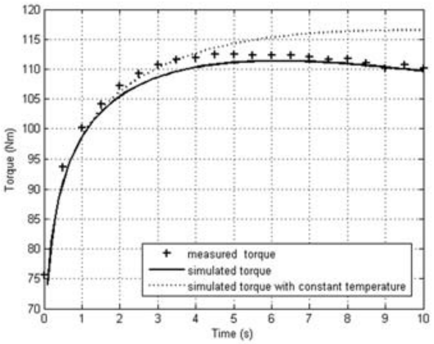

Figure 14. Comparison between torque for constant temperature vs. simulated temperature.

DISCUSSION

The presented model is valid at low velocities, i.e. the boundary lubrication regime. For applications where the velocity can be expected to be in excess of somewhere around 1 m/s, it should be combined with a model including hydrodynamic effects.

Temperature in the clutch is depending on the heat generation and the heat transfer as well as the boundary conditions. When working with the simulations it was clear that the cooling oil flow in the interface was of great significance. For instance, the temperature in the case described in Figure 12 will rise to over 160°C without any oil flow in the interface. This gives a torque about 10% lower than the measured torque.

In this paper an empirical formula for the oil flow was adopted. The empirical formula was estimated for experiments according to Case 1. The assumption that there is no heat conduction over the edges in z-direction may be correct in a clutch with many friction discs. Nevertheless, in the test rig, which is testing just one friction disc, we will have a small heat transfer over the boundaries in z-direction. This heat will be transferred to other parts in the test rig, such as the hydraulic piston which gives the axial engagement force. The formula for the interface oil flow is adopted with the assumptions that the z-boundaries are insulated. To keep the right temperature in the simulations, the adopted oil flow therefore will not be the same as in the test rig. This can explain the temperature difference in the parts with decreasing $\Delta\omega$ in Figure 13. When $\Delta\omega$ is decreasing, the temperature will also decrease. In the test rig, not only the friction discs have to decrease in temperature, but also the other parts of the test rig. This will give a slower decrease in temperature for the measurements in the test rig than in the simulation model.

To get better simulations in the future, the oil flow in the interface also has to be estimated by simulations. This will give better temperature prediction and will be more general and applicable on different geometries than the model used today.

This friction function is working for the system it is adopted to, but for other friction materials or transmission fluids it is still necessary to run a quite extensive experimental investigation in order to obtain new and accurate fitting parameters throughout the friction parameter matrix. A possible method to reduce this effort would be to assume that the friction-velocity curve is only influenced by temperature and not by normal load; this will significantly reduce the amount of testing necessary while still maintaining a decent accuracy in the prediction.

Torque transfer in wet clutches, working in boundary lubrication regime, is quite complicated to compute. The friction is depending on lubricant, friction material and geometries of friction discs as well as the environment the clutch is working in. The complicated models needed to describe these clutches have implied that most of the development of wet clutches for limited slip differentials have been carried out in test rigs in laboratories and full scale testing in working vehicles. These testes are expensive and time consuming and the product development process would benefit significantly if some of these testes could be carried out as computer simulations instead. Since the clutches work in boundary lubrication, the friction is depending much on different additives in the lubricant. These additive reactions are difficult to simulate, and today the only way to get a reliable friction coefficient as function of temperature and sliding velocity is to measure the friction coefficient under working conditions. However, a simplified friction measurement in combinations with temperature and flow simulations can give good indications on how future wet clutches would perform. These simulation models can simplify the product development and lower the manufacturing costs, Full scale tests could primarily concentrate on fine calibration of products already tested in the simulation models.

CONCLUSIONS

Thermal effects have a significant influence on the torque transferred by the differential under limited slip conditions. It is therefore necessary to have a temperature dependant boundary friction model.

The contribution to the transferred torque from hydrodynamic effects are small under limited slip conditions and can be neglected without influencing the accuracy of the torque prediction.

It is possible to accurately determine the transferred torque knowing the current operating conditions and the thermal history of the clutch, given that the boundary friction model is taking clutch temperature into account.

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