The application of a CVT in a two disc test machine

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Report number: DCT-2006.093

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Eindhoven, August 2006

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1 Introduction

Modern science allows for more extreme working conditions for machines. Workload increases while size and weight decreases. Additionally mass production puts bigger demands on reducing manufacturing costs. This increases the importance of studying the wear rates of contacts between bodies of different material combinations, with and without lubrication. Wear rates determine the life expectancy of many integral parts of modern machines. Consequently they also determine the life expectancy of the entire machine. By examining wear rates closely, safety factors can be decreased, reducing manufacturing costs.

Research in the field of tribology is mostly done in laboratories. Doing so as many variables as possible can be regulated or monitored. This increases the accuracy of the research results. The equipment necessary to perform sufficiently accurate experiments is very important to obtain good measurement results. An often used type of machine is the so-called two disc test machine. Such a machine contains two test discs which are tangentially positioned against each other. The discs rotate in opposite directions. By controlling both rotational speeds a combination of slip and roll can be created. This way working conditions of machine parts can be simulated.

Different setups of this type of machine exist. All of these setups have their strengths and weaknesses. In this study a machine will be designed that combines these strengths while eliminating the weaknesses. This will be done by applying a Continuous Variable Transmission (CVT) to control the slip-roll ratio between the test discs. In doing so a setup very similar to the "Power Loop Test Rig" (PLTR) needs to be created. This test rig has been created at the Laboratory of Automotive Engineering Science, Faculty of Mechanical Engineering at the University of Technology in Eindhoven. This source of knowledge can be seen as a big advantage for the further development of the new two disc test machine.

2 Problem formulation and assignments

This study is the first step in the development of a new type of two disc test machine. No previous research has been done on this type of tribology test equipment within the Laboratory of Automotive Engineering Science. Therefore the first goal should be to form an impression of the market for this type of machine. This goal can be broken down in to several study questions:

What is the theory behind the research that is supported by two disc machines?
What types of two disc machines are available?
What research is being done with two disc machines and by whom?

When an overview of the market has been formed the next step in the development process should be taken. During this step the design requirements should be formulated to which the new design should comply with. When this is successfully completed the main study question can be answered:

Is the application of a CVT in a two disc test machine feasible?

If the above question can be answered positively a concept design of the new machine should be made.

3 Theory

3.1 Introduction

Normally the need to design and develop a two disc test machine arises when tribologic research is needed. This study is an exception to this rule. A technology is available that could be very useful for application in a two disc machine, but that has not been developed for this purpose. Therefore this study is started by giving a summary of the basic theory behind the research that is being done with this type of test equipment.

3.2 Types of contact

Contacts between surfaces can be organized in three groups, namely conformal, non-conformal and flat contacts. Figure 3.1 shows the differences between these three groups. Non-conformal contacts provide for the most critical wear within a machine. These contacts can for instance be found in gear sets, friction discs, wheel-rail contact and roller bearings. Many contacts between machine components can be represented by cylinders which provide good geometrical agreement with the profile of the undeformed solids in the immediate vicinity of the contact. Two well-known examples are roller bearings and gears. For rolling bearings it’s obvious why its contacts can be simulated by cylinders because its solids are cylindrical and spherical. This is not the case for gears.

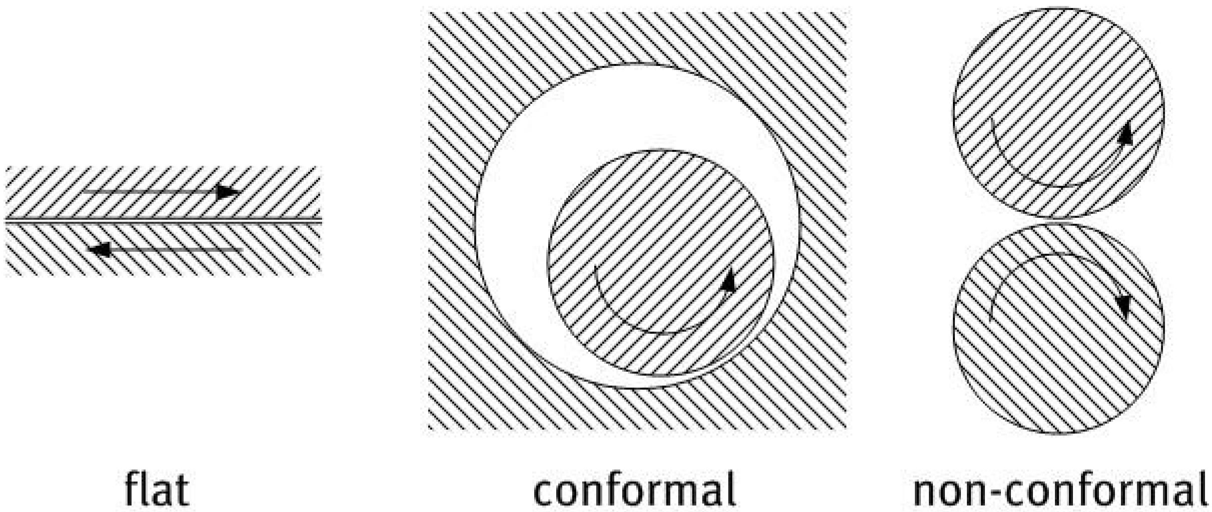

Figure 3.1: Flat, conformal and non-conformal contacts [3]

3.3 Analysis of tooth contact

In the case of gears the line of contact is parallel to the axes of the gears. A portion of tooth surface along the line of contact can be regarded as part of the surface of a cylinder. The conditions of tooth engagement can be represented by two cylinders making line contact with certain rolling and sliding motions. The theory that is needed to compute these cylinders with their matching rotational can be found in attachment A.1.

3.4 Slip-roll ratio

An important specification of two disc test machines is its range of the slip-roll ratio. This ratio can be computed using equation (3.1). Herein v_{rA} and v_{rB} represent the rolling velocities of the two test discs.

$$S = \frac{v_{rA} - v_{rB}}{v_{rA} + v_{rB}} = \frac{\Delta v_r}{2\overline{v}_r}, \quad 0 \leq S \leq \infty \tag{3.1}$$

- $S = 0$; $v_{rA} = v_{rB}$: pure rolling, no slip
- $S = \infty$; $v_{rA} = -v_{rB}$: pure slip, no rolling

3.5 Elastohydrodynamic lubrication

To reduce wear most contacts are lubricated. The goal of this is to prevent direct contact between the two surfaces. For non-conformal highly loaded contacts the theory that described lubrication is the principle of hydrodynamic lubrication. Because of high contact pressures both surfaces cannot be described as rigid. Elastic deformation exists, so the principle of elastohydrodynamic (EHD) lubrication can be applied. Because of this elastic deformation, the contact surface expands which increases the carrying capacity of the system.
Adhesion of oil to both elements increases pressure and creates a film between the bodies. Because the area of contact is very small, the force per unit area is very high. Under these pressures one would expect the oil to be entirely squeezed from between the surfaces. However, viscosity increases that occur under very high pressure prevent the oil from being entirely squeezed out. Consequently a thin film of oil is maintained. Within this film temperature changes take place. This also has its effect on the viscosity of the oil.

The main equation which describes the behaviour of lubrication films is known as the Reynolds equation. This equation is derived by applying the basic equations of motion and continuity to the lubricant. As this paragraph is intended to give a general description of the phenomenon EHD lubrication instead of thoroughly discussing the underlying theory, only the final derived equations will be given. Appendix A.2 gives a the derivation of equation(3.2).

$$\frac{dp}{dx} = 6\eta\left(v_{rA} + v_{rB}\right)\frac{h_0 - h_{(\sigma_{Hz})} + \dfrac{x^2}{2R} + h_d(x)}{\left[h_0 + \dfrac{x^2}{2R} + h_d(x)\right]^3} \tag{3.2}$$

3.6 Stribeck curve

Richard Stribeck was a German mechanical engineer at the TU Dresden. He observed that for low velocities, the friction force is decreasing continuously with increasing velocities. This phenomenon of a decreasing friction at low, increasing velocities is called the Stribeck friction or effect. The effect of contact speed on the friction coefficient of a lubricant can be described by a so-called Stribeck curve as shown in figure 3.2. Two disc test machines are often used to determine these curves for different lubricants by determining the traction forces that occur for rotational speeds at a certain normal load.

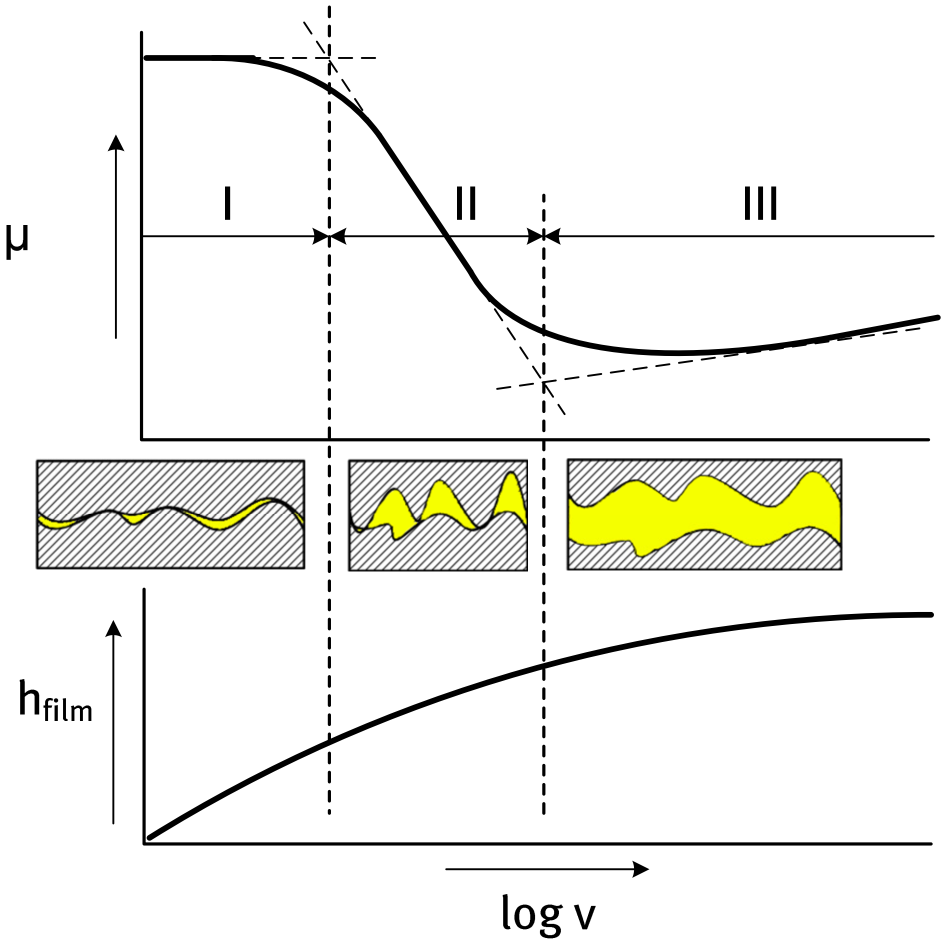

Figure 3.2: Stribeck curve [15]

As can be seen in the above figure there exist three lubrication conditions, indicated by I, II and III. They can be described as follows:

I **Boundary Lubrication**: The surface speed in the contact is low and no pressure will be build up in the lubricant. The load is completely carried by the asperities. The surfaces are protected by absorbed molecules of the lubricant or a thin oxide layer.

II **Mixed Lubrication**: The load is carried by a combination of the hydrodynamic pressure and the contact between the asperities of test discs surfaces.

III **Hydrodynamic Lubrication**: The load and the hydrodynamic pressure are in equilibrium. As a result both test discs are completely separated by a lubricant film.

3.7 Hertzian pressure

The general method of describing the stress conditions and deformation at the surfaces of elastic bodies making point or line contact is the theory by Hertz. The stress distribution on the contact between two round bodies is elliptic with a maximum value σ_{Hz}, the Hertzian stress, see figure 3.3 and 3.4. For two cylindrical bodies this value can be computed by equation (3.3). Herein E_1, υ_1 and E_2, υ_2 represent the elastic modulus and the Poisson ratio for both bodies. The width of the stress distribution, as shown in figure 3.4, follows from equation (3.4). A severe stress condition is an important origin of wear on contact surfaces.

$$\sigma_{Hz} = \sqrt{\frac{1}{\pi}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)^{-1}\cdot\left(\frac{1}{r_1}+\frac{1}{r_2}\right)\cdot\frac{F_0}{l_w}} \tag{3.3}$$

$$b = \sqrt{\frac{4}{\pi}\left(\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}\right)\cdot\left(\frac{1}{r_1}+\frac{1}{r_2}\right)^{-1}\cdot\frac{F_0}{l_w}} \tag{3.4}$$

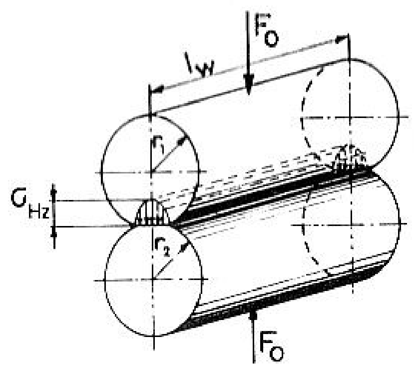

Figure 3.3: Hertzian pressure at the contact[3]

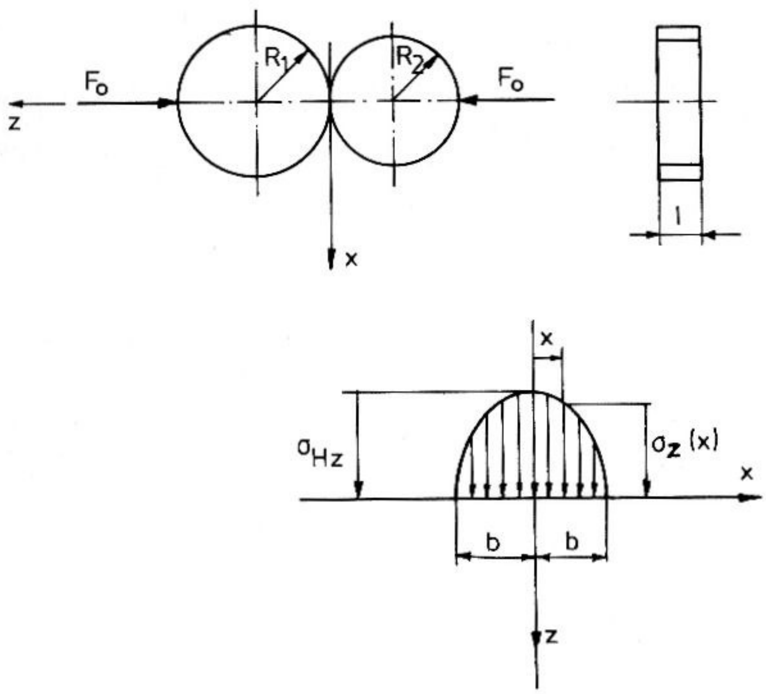

Figure 3.4: The stress distribution in a line contact[3]

3.8 Wear

3.8.1 Types of wear

There are two main types of wear that can occur in non-conformal contacts. These are surface welding and pitting.

3.8.2 Pitting

Pits in the surface of a test body are created in the following way. First a crack appears at the surface. Secondly it changes direction and proceeds roughly parallel to the surface. Finally it returns to the surface to form a closed area. This way a piece of material separates from the surface and a pit exists. Pitting only exists in the presence of a lubricant. The crack is propagated by pressure in the lubricant inside the crack.

3.8.3 Surface welding

A high stress and sliding velocity can result in a temperature flash of several hundred degrees centigrade. This combination of high temperature and pressure can result in the welding together of both surfaces which are in contact. When the surfaces are torn apart, material from one surface can stick at the other or both surfaces can become rough. This process is subdivided in three categories of increasing severity, namely ´picking up´, ´scuffing´ and ´scoring´. Flash temperatures are dependent on the amount of friction; welding failures indicate inadequate friction reducing properties of the lubricant.

3.9 The influence of elastohydrodynamic lubrication on friction, wear and the life span of gears

To test the effectiveness of a certain lubrication liquid for given operating conditions several quantities need to be measured. In an elastohydrodynamic contact the quantities: pressure gradient, film thickness, temperature gradient and coefficient of friction, depend strongly on the working parameters: material and lubricant properties, torque, rotational speed and temperature. How these parameters depend on each other and which consequences they may have on the test results is shown in Attachment B.

4 Field of application

4.1 Applications

4.1.1 Introduction

Twin disc test machines have many applications within the tribology research field. They are mainly used for fundamental research under controlled conditions on lubricants and materials. As mentioned earlier two well known examples of application are rolling bearings and gears. Next to these research applications there also exist several practically-oriented applications. The following chapter will briefly describe different applications of two disc test machines.

4.1.2 Fundamental research on lubricants

For the development of better lubricants a lot of research is required. Two disc test machines play a big role in this field of research.[3][10][12] By controlling temperature, load, slip and speed and measuring applied and carried torques the very important coefficient of friction can be determined. It is also simple to study the influence of additives or pollution. This way lubricants with different characteristics can be developed. This is desired because different applications require different lubricant characteristics.

4.1.3 Research on the influence of different lubricants

There are many types of lubricants available. To determine which one is best suited for certain applications, the contact that needs to be lubricated can be approximated by the two discs of a two disc test machine. By applying a load and giving the discs a rotational speed specific working conditions can be simulated. This way the performance with respect to wear or friction of different lubricants can be determined. Also, manufacturers of machines with lubricated contacts like to check the provided specifications of lubricants. Instead of making the reliability of their product depend on the correctness of the lubricants manufacturer.

Manufacturers of rolling bearings are well known users of two disc test machines. Their products performance obviously greatly depends on the performance of the applied lubricants. Another, less known, example of an application is research that is carried out on the so-called full toroidal Infinitely Variable Transmission (IVT).[9] This paper describes an experiment with a twin disc test machine. The objective of this experiment is to test a rheological model that describes the behaviour of elastohydrodynamic traction drive contacts over their operating range.

4.1.4 Practically-oriented research

So far all the covered applications relate to research on lubricants. Now a different kind, a more practically-oriented application will be discussed. A field of application is the research relating railway development. Much research is being done on rail-wheel contacts.[5][6][8][11] One disc represents a wheel and the other a piece of rail. Sometimes one disc is replaced by a complete wheel and on the second disc a stretch of real rail is fixed. By adding water, oil or pollution, like leafs or sand, different rail conditions can be simulated. This is very useful because of the possibility to simulate real life situations. The surface of rail is under constant change, because of temperature, rain, leaves or other pollution. This has a great influence on the coefficient of friction between wheel and rail.[11] Figure 4.1 shows a schematic overview of such a test carried out by The University of Sheffield in cooperation with Aegis Engineering Systems.[8] Additional research is being done on new rail materials and coatings. Here twin disc machines are also frequently used.

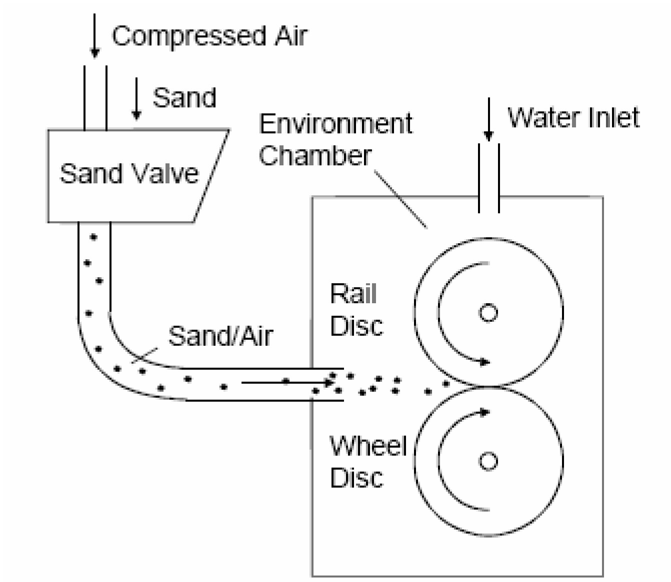

Figure 4.1: Schematic representation of a wheel-rail contact test with addition of pollution [8]

4.2 Specifications

The specifications of the most used two disc test machines have been collected in table 4.1, where they have been broken down into three categories. These categories are the lower and upper values of the specifications of the range of Plint[13] two disc machines and custom build test machines used in research found on the internet. This data can be used to formulate a set of requirements for the new test machine and its components.

Table 4.1: specifications of two disc test machines

	Power [kW]	Torque [Nm]	Speed [rpm]	Load [kN]	slip ratio [%]	Diameter test discs [mm]	Width test discs [mm]
Lower boundary Plint selection	5.5	10	0	0	0	25	10
Research	75	10	1200	29	-5 ←→ 28	30 ←→ 173	10
Upper boundary Plint selection	65	1363	7500	36	100	125	25

4.3 Market potential

The development of a new product is only useful if a healthy market exists. There should be sufficient demand for it. Otherwise the costs of development will never be fully recovered. The fact that this is a project for a University of Technology changes this view slightly, because it is an educational institute. The main goals of such an institute are to provide academic education and perform scientific research. But the fact remains that even an University should aim at returning its costs.

Consulting several scientists who are linked to the field of tribology[12][18][19] resulted in the view that the market for two disc test machines is not very large. Therefore it might be more interesting to develop a module which contains a CVT and can be implemented by connecting an ingoing and outgoing axle. An even more feasible method would be to develop the technology needed to integrated a CVT in a two disc test machine. This technology might be very interesting for an existing manufacturer like Plint. For the second idea it is sufficient to develop a prototype of a dynamic two disc test machine with an integrated CVT.

5 Basic setup of the test machine

5.1 Conventional two-roller setups

To understand the advantages of applying a CVT it is important to look at the currently available solutions. To drive the two discs of a two-roller test machine two main setups exist.

- Most widely used are test machines which consist of one electric motor and a gear transmission. The ratio of the gear pair determines the slip ratio of the test discs. A big disadvantage of this setup is the need to replace a gear pair to gain a different slip ratio. Testing a broad range of ratios will be labour-intensive and therefore time consuming. An advantage of this setup is its relative low cost because of the use of only one motor. This motor can be of relatively small size. The power applied on one test disc gets submitted by the second disc on the first axle. This way it forms a power loop similar as present in the PLTR[19]. As a result the motor only has to compensate for the energy losses.
- Furthermore there are test machines that utilize two electric motors to drive the discs independently. This is a very flexible but expensive solution. By controlling the speed difference between the two motors the slip ratio between the test discs can be adjusted continuously over a big range. This eliminates the need for different gear pairs. On the other hand the cost of its two high-power electric motors makes it rather expensive. In addition the need for an extra motor also increases the size of the total setup.

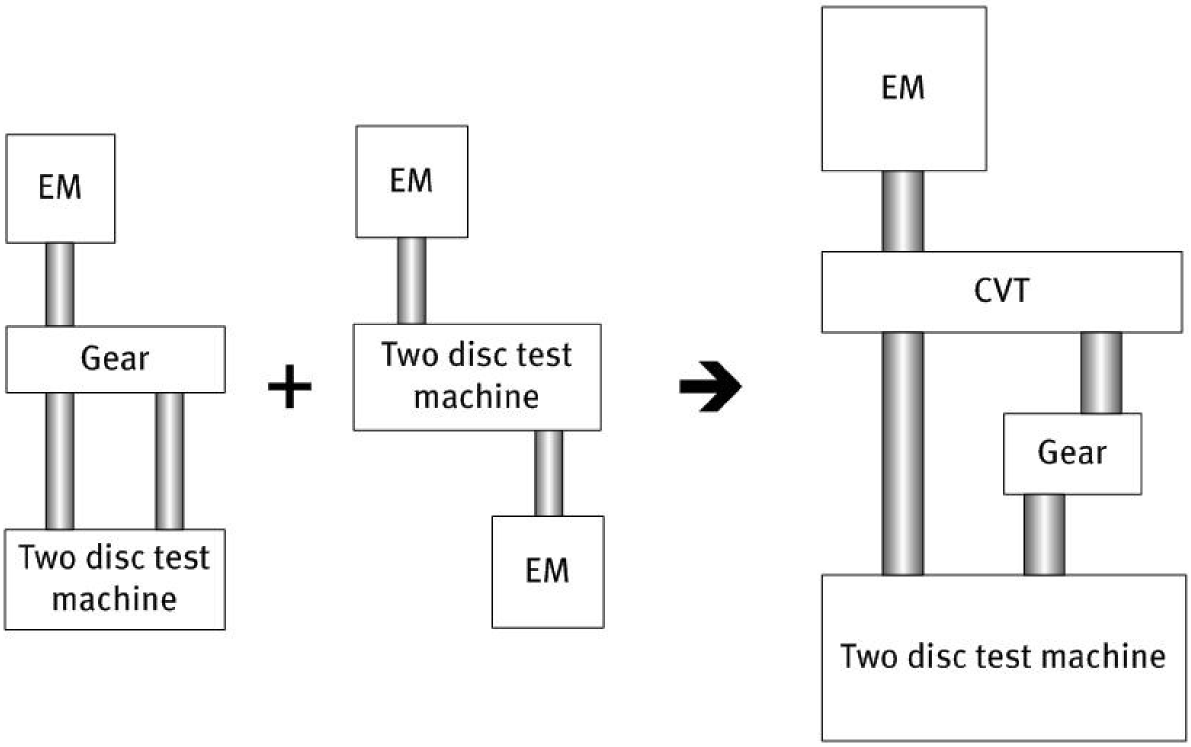

Figure 5.1: Schematic overview of the different setups of twin disc test machines

5.2 Application of a CVT in a two-roller test machine

Obviously the ideal test rig should combine the advantages of above-mentioned systems without including its disadvantages. Therefore it should be able to continuously vary the slip ratio over a broad range while utilizing only one electric motor. The setup with one engine, described above, should be modified so that it could adjust the slip continuously. By replacing the gear pair with a CVT this would be realised. An overview is shown in figure 5.1. This is a setup very similar to the PLTR, which is constructed in the AES-lab at the University of Technology in Eindhoven. A picture of this setup can be found in attachment C.

5.3 Power Loop Test Rig

The PLTR has been designed and built at the University of Technology in Eindhoven.[19] The main purpose of development was to make endurance testing of V-shaped elements for a CVT possible. To minimize the needed engine torque and so reduce the costs of the setup a power loop was created by installing two variators. The secondary pulley of the first variator is connected to the primary pulley of the second variator and vice versa. By setting both ratios to be different slip will occur between the belt and the pulleys. Due to this slip a power flow circulates through the test rig. The electric motor only needs to compensate for the energy loses in the system.
The same principle can be used in a two-roller test machine. The second variator can be replaced by the two test discs.

5.4 Comparison of the different setups

Now the new layout can be compared with the two common types of layout that current two disc test machines are based on. Table 5.1 gives a survey of the properties between the three layouts.

Table 5.1: Survey of the properties between the three layouts

	One motor with a gear set	**Two motors**	**One motor with a CVT**
Flexibility	-	++	++
Costs	++	--	+
Installation space	++	--	++

6 Feasibility

One of the main goals of this study is to check if the application of a CVT in a twin disc test machine is feasible. To check this, the specifications of the concept need to be compared with the data collected in table 4.1. If the new machine will have a chance of succeeding as a commercial product its achievable specifications should cover a sufficient portion of the markets requirements.

6.1 Extremes

As for most markets, the field relevant for this study has its extreme applications. For instance some tests on wheel rail contact were conducted with custom build test machines at the Canadian National Research Centre by the Argonne National Laboratory[5] shown in figure 6.1. To simulate very realistic test conditions two complete train wheels are mounted on one axle. These are loaded against large rims that have a diameter of 1600 mm and simulate the track.
Obviously it is not desirable to design a test machine that can accommodate test discs with diameters ranging from 40 to 1600 mm. For most of its applications the machines would be unnecessarily large. Instead of designing a machine that is applicable in all possible two roller tests, the most extreme applications will not be taken into account when comparing the specifications.

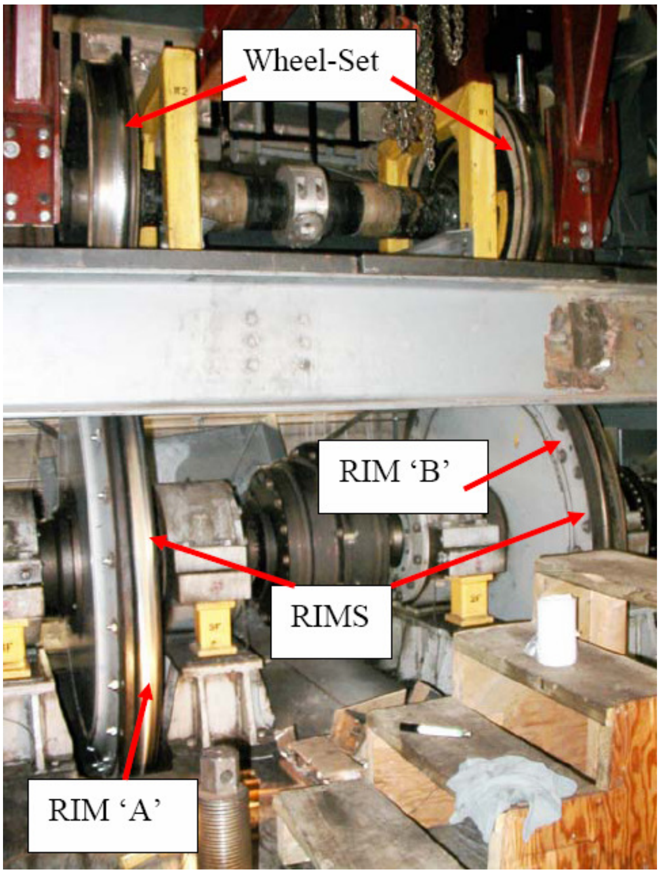

Figure 6.1: Test rig at the Canadian National Research Centre [5]

6.2 Specifications of the CVT

The main innovation of the new concept designed during this study is the application of the setup of the Power Loop Test Rig. The main components of this setup are the CVTs. Consequently the main component of the new concept is that same CVT. To check the setups feasibility the physical and operational constraints of the variator need to be known. The report by Gijs Commissaris[4] provides these constraints:

- The maximal absolute value of the torque in the test rig is 250 Nm on the primary shaft.
- The CVT ratio is bounded between r_{UD}= 0. 43 and r_{OD} = 2.33.
- The maximum ingoing speed is 6000 rpm.

Using equation (3.1), the given limits of the CVT ratio result in a slip ratio range of -39% to +39%. The test machine can be constructed in such a way that the slip ratio range between the two test discs differs from the above calculated values. This can be done by choosing the gear ratio of the gear set that is needed to change the direction of rotation of the secondary variator axle different from 1.

6.3 Feasibility

The most important question to be answered during this study is: "Is the variator used in the Power Loop Test Rig suitable for application in a two disc test machine?" If it is not suitable, the concept could never result in a successful setup.

To test the feasibility these constraints can be compared to the specifications found during the literature study. These specifications are summarised in table 4.1. A comparison of the working limits between the used CVT and two of the most competing commercially available products is given in table 6.1. This shows that the basic idea of applying a CVT in a two disc test machine is feasible. The CVT specifications almost completely cover the specifications of the competing machines.

Table 6.1: Comparison between two commercially available products and the applied CVT

	Plint Te 72 High Power	**Plint Te 73**	**Limits of the CVT**
Torque [Nm]	207 (0 – 3000 rpm)	115	250
Speed [rpm]	0 – 3000 - 4500	100 ←→ 7500	Maximal 6000
Slip ratio [%]	0 – 100	0, 1, 2, 4, 6, 8, 9	-39 – 39

7 Test machine requirements

7.1 Recommendations

After discussions with ing. Kees Meesters[18] and ir. Harry van Leeuwen[17] both working within the Department of Mechanical Engineering at the University of Technology in Eindhoven and ing. Erik de Vries[12] working within the Department of Design, Production and Management at the University of Twente, some practical recommendations can be made regarding the design of the machine.

Because the machine need to be optimised to handle great loads on the test discs, bearings need to be used on both sides of the discs. Otherwise the axles would bent to much under the stress and prevent the contact between the discs to be straight. The possibility to dynamically apply a load by hydraulics would be a very useful quality.

It is important to understand how the potential users of the test machine would like to use it. Generally speaking scientists have very specific ideas regarding experiments. Research has shown that many institutes or companies build their own two disc test machines if they need one or they modify a commercially available product. This way researchers can develop a setup that is specifically designed for certain experiments. They can measure the variables, apply the loads, fix discs with specific radii and apply torques they need.
When designing the two disc test machine this must be kept in mind. The goal should be to create a product that can easily be modified by its users. The following design guidelines help achieving that goal:

- Different methods of applying a load should be possible. For instance by hydraulics, an electric coil or death weights.
- It should be possible to mount test discs with different radii on the axles.
- The area around the discs should be easily accessible to make measuring easy. This way research worker can use their favourite method of measuring.

7.2 Test machine requirements

In 1973 prof. dr. ir. M.J.W. Schouten successfully defended his doctoral thesis on “the influence of elastohydrodynamic lubrication on friction, wear and the lifespan of gears”.[3] During his study he designed a two disc test machine. This machine was also built within the Department of Mechanical Engineering at the University of Technology in Eindhoven. Unfortunately it has been disassembled since, so no physical examination is possible. Nevertheless the doctoral thesis provides for much practical and theoretical information that might come in handy while further developing the new test machine. Some of the information most useful for the next stage of the design process has been summarised below:

- There can be no friction forces which could influence the measurements. For instance bearing friction.
- Load force, rotational speed, lubricant temperature and friction moment should be measured with an accuracy of 0.5%.
- Every measuring instrument must be calibrated properly.
- The rotational speeds of both discs should be fully independent adjustable.
- The load should be steplessly applicable.
- The machine must be able to accommodate discs with different radii.
- Different types of lubricant must be applicable.

8 Concept study

8.1 Construction requirements

Because it is proved that the implementation of a CVT in a two disc test machine is feasible, a concept design can be made. This concept should comply to the requirements listed below:

- The test discs in a two-roller test machine rotate in opposite directions. This means that the axles connecting the two test discs also need to have these opposite rotation directions. The gear set used in conventional two-roller test machines changes the direction of rotation between in- and output. For a CVT this is not the case. In and outgoing axles rotate in the same direction. This means an extra gear set is needed to counteract this difference.
- The machine should be suitable for test discs of different radii. This can be accomplished by making one of the disc carrying axles movable.
- It should be possible to introduce a load on the test discs.

8.2 Concept design

A concept design has been made in which a movable disc carrying axle exists. This axle can be rotated around the secondary CVT axle. This is possible because of the gear reduction needed to change the direction of rotation of the second test disc. This design makes it possible to fix discs of different radii to the axles. This can be seen in attachment F. A load can be applied by a dead weight. Of course different load applying systems can be used. The choice of the most suitable system depends on the simulated operating conditions and the required test accuracy. This paragraph gives a first concept and not an in depth design of the two-roller test machine, therefore no further research has been done in this direction.

The dimensioning of the different components of the machine has been based on basic knowledge attained from studying literature and other two-roller test machines. No extensive calculations were made as only a concept sketch was to be designed.

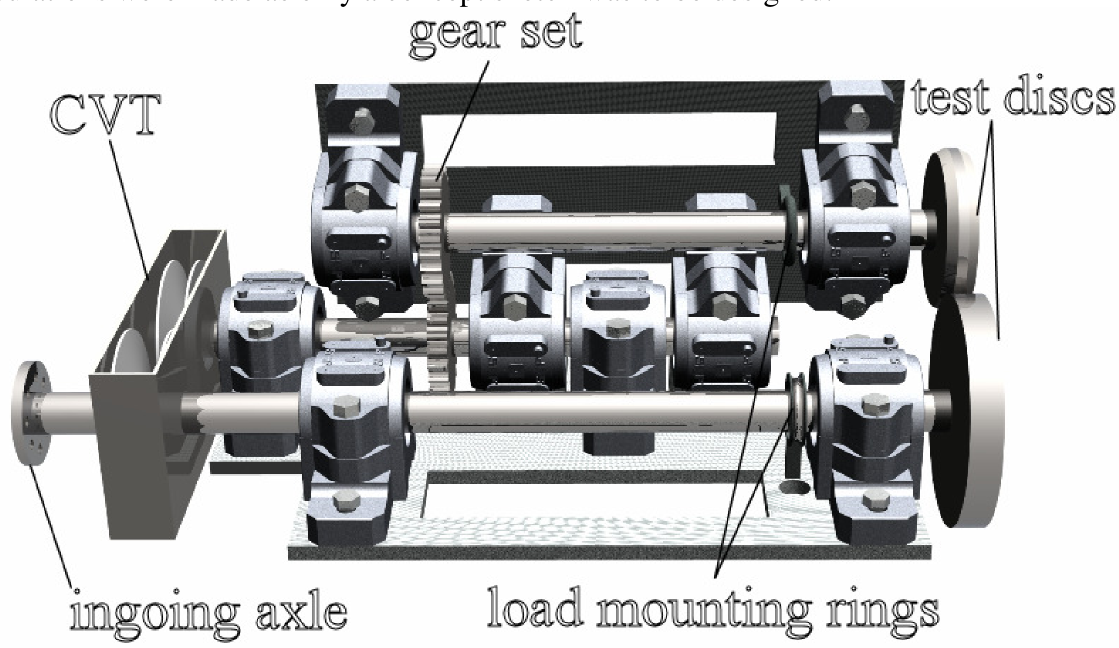

Figure 8.1: Overview of the concept design

9 Conclusions and recommendations

The study was started with a brief summary of the theory behind the research that is being done with two disc test machines. This summary introduces the variables that the machine has to be able to measure. Following this theory overview is a description of the different applications of two disc test machines. It can be concluded that a very versatile field of application exists. This means that several types of possible customers need to be satisfied by the new design. This implies that it is desired that a dynamic setup is developed. For this reason test discs with different radii should be applicable and it should be possible to dynamically load them against each other.
A study of the different commercially available test machines and the custom build test rigs showed there are two common setups available. The strong properties of both setups can be combined by the application of a CVT. This should be regarded as the main selling point of the new machine.
The market for this type of test equipment is rather limited, so it will be hard to make a commercially successful product. Therefore it might be more interesting to focus the design process on developing the technology that manufacturers need to implement a CVT in their products.
This entire project and any further research on this topic will be of no use if the specifications of the CVT don't meet the requirements formulated during this study. By testing the working constraints of the CVT to this requirements and comparing them with the specifications of popular commercially available products it can be concluded that the application of a CVT in a two disc test machine is feasible.

Bibliography

[1] Dowson, D., Higginson, G.R., “Elasto-hydrodynamic lubrication”.
Pergamon Press, UK, 1977

[2] Merritt, H.E., “Gears”.
Sir Isaac Pitman & Sons, UK, 1954

[3] Schouten, M.J.W., “Einfluss Elastohydrodynamischer Schmierung auf Reibung Verschleiss und Lebensdauer von Getrieben”.
Technische Universiteit Eindhoven, The Netherlands, 1973

[4] Commissaris, G.A., “Validation of Clamping Force Models using the Power Loop Test Rig”, Report number: DCT-2005.111.
Technische Universiteit Eindhoven, The Netherlands, 2005

[5] Aldajah, S., Fenske, G.R., Ajayi, L., Kumar, S., Reed, C., Xu, Z. and DiMelfi, R., “Laser Glazing of Rails WBB/IWS Tests (CNRC-Ottawa),”
Argonne National Laboratory, 2005

[6] Lewis, R., Braghin, F., Ward, A., Bruni, S., Dwyer-Joyce, R.S., Bel Knani, K. and Bologna, P., “Integrating Dynamics and Wear Modelling to Predict Railway Wheel Profile Evolution”.
The University of Sheffield, UK, Politecnico di Milano, Italy and Fiat Research Centre, Italy, 2003

[7] Frolish, M.F., Fletcher, D.I. and Beynon, J.H., “A quantitative model for predicting the morphology of surface initiated rolling contact fatigue cracks in back-up steels”.
Department of Mechanical Engineering, University of Sheffield, UK, 2002

[8] Lewis, R., Dwyer-Joyce, R.S. and Lewis, J., “Wheel/Rail Contact Isolation due to Track Contamination”.
The University of Sheffield, UK and Aegis Engineering Systems Ltd, UK, 2003

[9] Sharif, K.J., Evans, H.P., Snidle, R.W., “Modeling of Film Thickness and Traction in a Variable Ratio Traction Drive Rig”.
School of Engineering, Cardiff University, Cardiff, UK, 2004

[10] Wright, N.A. and Kukureka, S.N., “Wear Testing and Measurement Techniques for Polymer Composite Gears”.
University of Birmingham, UK, 2001

[11] Vasic, G., Franklin, F.J. and Kapoor, A., “New Rail Materials and Coatings”.
University of Sheffield, Report: RRUK/A2/1, UK, 2003

[12] de Vries, E.G., 2006, "Oral communication", 12 April 2006

[13] http://www.plint-tribology.fsnet.co.uk/

[14] http://www.ducom.com/g_tworoller.htm

[15] http://www.tribology-abc.com/abc/stribeck.htm

[16] Bryant, M.D., “ME 383S, Lubrication, Wear & Bearing Technology”,
http://www.me.utexas.edu/~bryant/courses/me383s/DownloadFiles/LectureNotes/Elastohydrodynamic.pdf
Mechanical Engineering at The University of Texas, Austin, USA, 2006

[17] van Leeuwen, H.J., 2006, "Oral communication"

[18] Meesters, C.J.M., 2006, "Oral communication"

[19] Nelissen, J.H., "Power Loop Test Rig, variator redesign and development of a power loop test rig”, Report number: DCT 2003/20n.
Technische Universiteit Eindhoven, The Netherlands, 2003.

A.1 Calculation of the equivalent cylinders with corresponding rotational speeds

To be able to determine the correct test conditions of the two cylinders figures A.1 and A.2 can be used. Figure A.1 shows a velocity diagram as used by Merritt [2] to compute the velocities occurring within a tooth contact between two external gears, O and Q, with pitch diameters OP (r) and QP (R) and angular velocities ω_p and ω_w. Point c1 is the point where tooth A is in contact with tooth B. The path c1 travels during the contact period is given by c_2Pc_3. To describe the position of c_1 on the contact curve, the angles α and β are used. Point a and b represent the position of c_1 on tooth A and B. Within the velocity diagram the pitch-line velocity is presented as v and the velocities of point a and b are presented as oa and ob. Since the velocity of the point of contact in space is represented by oe, the velocity of the point of contact relative to the point a will be represented by ae. This velocity is the "rolling velocity" with respect to tooth A. Similarly be represents the rolling velocity with respect to tooth B. With r and R being the radii of gear O and Q the pitch-line velocity can be computed with the following formula:

$$v = r \cdot \omega_p = R \cdot \omega_w \qquad \text{(A.1.1)}$$

The rolling velocities v_{rA} and v_{rB} can be determined by calculating ae and be:

$$v_{rB} = v \cdot \cos\psi \left[\tan\gamma + \tan\left(\alpha + \psi\right) \right] \qquad \text{(A.1.2)}$$

$$v_{rB} = v \cdot \cos\psi \left[\tan\gamma + \tan\left(\psi - \beta\right) \right] \qquad \text{(A.1.3)}$$

Herein γ represents the angle between the path of contact and the common normal. In the particular case of involute gears, γ = 0.

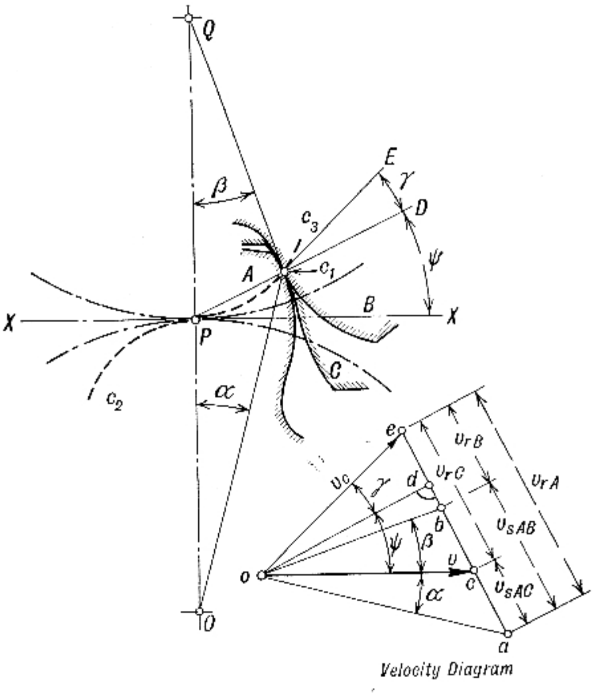

Figure A.1: Velocity diagram for rolling and sliding[2]

Merritt used figure A.2 to determine the centre of curvature of the tooth profile. At the point of contact c_1 the profile of tooth A can be regarded as a short length of circular arc having its centre at point D_a. The line D_aPc_1 can be imagined to be replaced by a link, of which the end

D_a moves in a circular path around O and the end c_1 moves along the path of contact, while at the same time it is constrained to pass through P. Now point F can be situated as the point of intersection between three lines. The first line is the normal at P, second is the normal c_1F to the path of contact and third is the line connecting O, the centre of gear A, with D_a. Given any two of these lines, the third can be found. Finally the intersection point between QF and c_1D_a locates D_b.

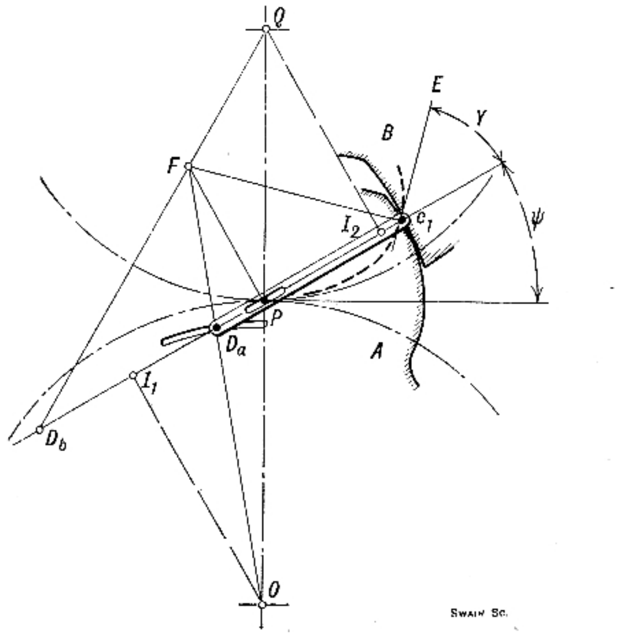

Figure A.2: Determination of centre of curvature of tooth profile[2]

By combining figure A.1 and A.2, the diameters and angular velocities of the equivalent cylinders can be determined. The cylinders may now be rearranged to work on fixed centres D_a' and D_b' with a fixed point of contact c_1'. If they are given peripheral velocities equal to their respective rolling velocities v_{rA} and v_{rB} they will simulate the exact same conditions of relative surface motion as the teeth of the actual gears at the contact point c_1.

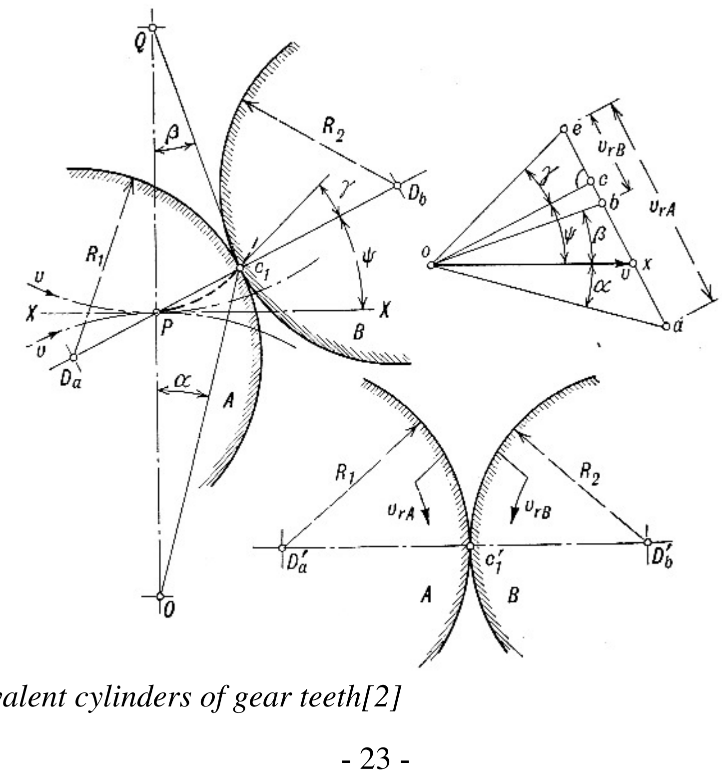

Figure A.3: Equivalent cylinders of gear teeth[2]

The behaviour of gear-tooth surfaces at any selected point can be isolated for the purposes of experimental investigation by running the equivalent cylinders together under the corresponding load with the appropriate angular velocities. The path of contact for involute gears will be the straight line Pc_1 ($\gamma = 0$). If the distance Pc_1 for any chosen point of contact is called s, the pitch diameters are D_A and D_B and the respective speeds are ω_A and ω_B. Then the rolling and sliding velocities are:

$$v_{rA} = \left(\frac{1}{2} D_A \bullet \sin\left(\psi\right) + s \right) \omega_A \tag{A.1.4}$$

$$v_{rB} = \left(\frac{1}{2} D_B \bullet \sin\left(\psi\right) - s \right) \omega_B \tag{A.1.5}$$

$$v_s = v_{rA} - v_{rB} = s\left(\omega_A + \omega_B\right) \tag{A.1.6}$$

$$v = \omega \bullet R \tag{A.1.7}$$

From relations (A.1.4), (A.1.5) and (A.1.7) it can be shown that the contact at a distance s from the pitch point can be represented by two cylinders of radii:

$$R_{A,B} = \frac{1}{2} \cdot D_{A,B} \cdot \sin\left(\psi\right) \pm s \tag{A.1.8}$$

By combining equations (A.1.1), (A.1.2), (A.1.3), (A.1.4) and (A.1.5) the for the chosen point of contact corresponding angular velocities ω_A and ω_B for the two test discs can be computed.

A.2 The theory behind Elastohydrodynamic lubrication

For the purpose of calculations on elastohydrodynamic lubrication two cylinders can be adequately described by an equivalent cylinder near a plane as shown in figure A.2.1(1). The requirement for this simplification is that for a value of x the distance between the two cylinders is the same as the distance between the equivalent cylinder and the plane. This requirement can be adequately satisfied for a small value of x. If x approaches the radii of the cylinders the simplification is less accurate. This is not a big problem because a small x represents the most important region for the theory of elastohydrodynamic lubrication. The radius of the equivalent cylinder can be calculated using equation (A.2.2).

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \tag{A.2.1}$$

$$R = \frac{R_1 R_2}{R_1 + R_2} \tag{A.2.2}$$

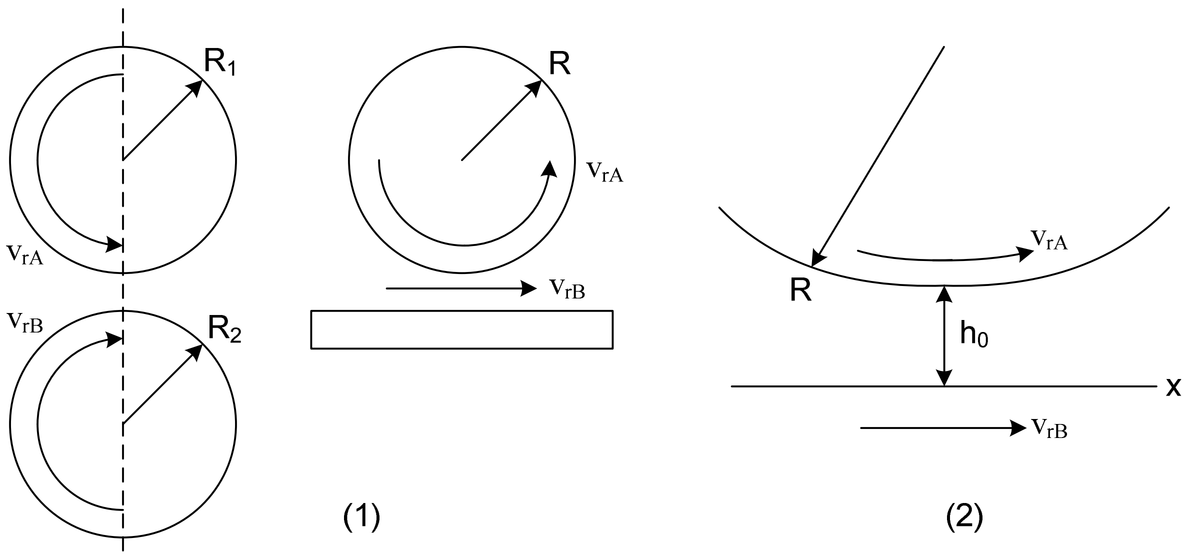

Figure A.2.1: The original and equivalent situation[16] (1); Film thickness (2)

The following assumptions lead to the simplified Reynolds Equation (A.2.3).

- For lubricating films, the inertia and body forces are negligible compared with the viscous and pressure forces.
- In a lubrication film the derivates of the velocity components u and w with respect to y are large compared with all other velocity gradients.
- If l is the length of the contact and h the film thickness, the pressure gradient across the film (pressure variation in y-direction), can be shown to be of the order of h/l times the pressure gradients along the film (x-direction). Since h << l the variation of the pressure across the film can be neglected. This means that the second equation of motion can be neglected.
- The displacements are calculated for a semi-infinite solid in a condition of plane strain.
- Side-leakage is neglected. This is justified because the zone width is very small compared with the length of the cylinders.
- The boundary conditions for pressure are: at inlet p = 0 at a large distance from the high pressure zone and at the outlet $p = \partial p/\partial x = 0$.

$$\frac{d}{dx}\left(\frac{h^3}{\eta}\frac{dp}{dx}\right)=12u\frac{dh}{dx} \tag{A.2.3}$$

In which the following equations need to be solved:

$$\eta = f(p,T) \tag{A.2.4}$$

$$h(x)=h_g(x)+h_d(x) \tag{A.2.5}$$

$$u=\frac{v_{rA}+v_{rB}}{2} \tag{A.2.6}$$

The film thickness can be divided in a geometric part which depends on the rigid form of the equivalent roller and a part that depends on the relative squash of the contact bodies. The geometric part can be calculated from:

$$h_g(x)=h_0+\frac{x^2}{2R} \tag{A.2.7}$$

and the part that is due to the limited stiffness of the bodies from:

$$h_d(x)=-\frac{2}{\pi E'}\int_{s_1}^{s_2} p(s)\ln\left(x-s\right)^2 ds+cons \tag{A.2.8}$$

with:

$$\frac{1}{E'}=\frac{1}{2}\left[\frac{1-\upsilon_1^2}{E_1}+\frac{1-\upsilon_2^2}{E_2}\right] \tag{A.2.9}$$

Viscosity is pressure and temperature dependent:

$$\eta=\eta_0 e^{\alpha\left(p(x)-p_r\right)}e^{\beta\left(T-T_r\right)} \tag{A.2.10}$$

Equation (A.2.3) can be rewritten in the following form:

$$\frac{d}{dx}\left(\frac{h^3}{\eta}\frac{dp}{dx}\right)=6\left(v_{rA}+v_{rB}\right)\frac{dh}{dx} \tag{A.2.11}$$

After integrating over x equation (A.2.11) transforms to:

$$\frac{dp}{dx}=6\eta\left(v_{rA}+v_{rB}\right)\frac{h(x)-h_{(\sigma_{Hz})}}{h(x)^3} \tag{A.2.12}$$

Here $h_{(\sigma_{Hz})}$ represents the film thickness at the point where the highest pressure, the Hertzian pressure, exists. After substituting (A.2.5) in (A.2.12) the final equation representative for an elastohydrodynamic lubrication film between two rollers becomes:

$$\frac{dp}{dx}=6\eta\left(v_{rA}+v_{rB}\right)\frac{h_0-h_{(\sigma_{Hz})}+\dfrac{x^2}{2R}+h_d(x)}{\left[h_0+\dfrac{x^2}{2R}+h_d(x)\right]^3} \tag{A.2.13}$$

B The influence of elastohydrodynamic lubrication on friction, wear and the life span of gears

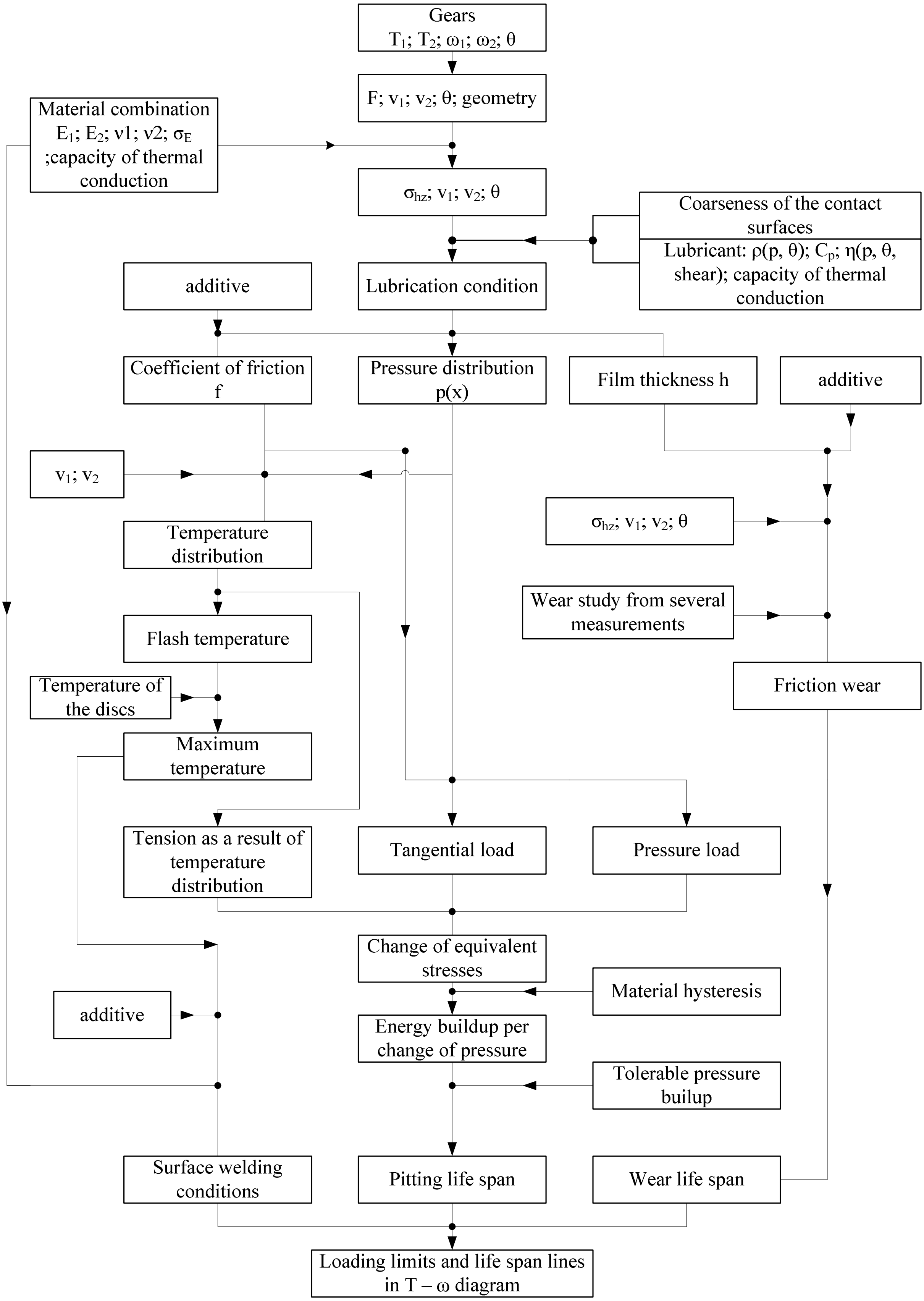

Figure B.1: Schematic overview of the influence of elastohydrodynamic lubrication on friction, wear and the life span of gears.[3]

C The Power Loop Test Rig

The design of the Power Loop Test Rig was the main source of the initial inspiration for using a CVT to create slip in a two disc test machine. The figure below shows a picture of the Power Loop Test Rig as constructed at the Laboratory of Automotive Engineering Science, Faculty of Mechanical Engineering at the University of Technology in Eindhoven.

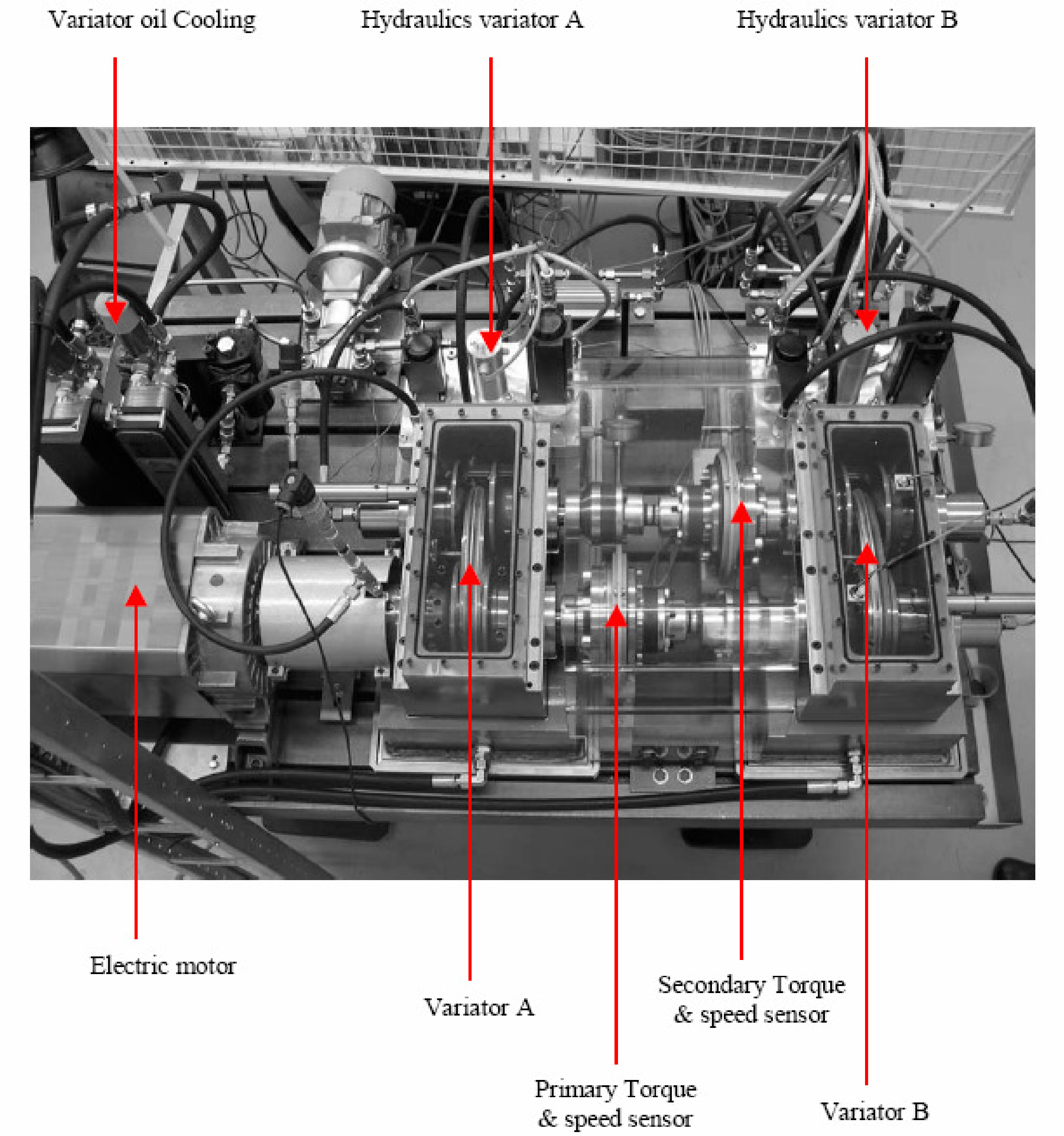

Figure C.1: The Power Loop Test Rig[4]

D Price indication

An offer of several tribology test machines is shown in figure D.1. These prices are currently not representative for they descent from 1981. To achieve a better estimate, inflation numbers for the Netherlands since 1981 have been looked up. These numbers taken into account, the current value of the Plint TE 73, which is most similar with the designed machine, is €160,182.

Table D.1: Numbers of inflation for the Netherlands since 1981

Year	Number of inflation [%]
1981	7.62
1982	8.72
1983	7.68
1984	6.36
1985	4.87
1986	1.30
1987	1.55
1988	1.16
1989	3.10
1990	3.45
1991	3.22
1992	2.43
1993	2.75
1994	2.38
1995	1.47
1996	2.06
1997	1.63
1998	0.96
1999	1.12
2000	2.55
2001	2.47
2002	1.65
2003	1.59
2004	2.10
2005	2.78
average	**3.0788 %**

$$1.030788\text{\^{}}^{25} \cdot \text{fl } 165{,}400 / 2.20371 = \text{€ } 160{,}182$$

1981

E / experimenteel

LANDRÉ-MIJNSSEN B.V.

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Telefoon 020 - 90 49 11
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Prof.Dr.Ir.M.J.W.Schouten
Den Dolech 2
5612 AZ EINDHOVEN

JG/LL

Diemen, datum postmerk
(november 1981)

Mijne heren,

Hiermede vragen wij uw aandacht voor beproevingsapparatuur voor smeerolie, die door onze Engelse fabriek
PLINT & PARTNERS Ltd., Wokingham
wordt vervaardigd.

Gezien de aard van uw bedrijf veronderstellen wij, dat ook bij u hiervoor belangstelling bestaat.
De voornaamste modellen zijn :

Model TE 73
Het testen van smeerolie, die zich bevindt tussen twee draaiende schijven met verschillende omtreksnelheid :
Richtprijs exkl. B.T.W. ƒ 165.400,-
Model TE 77
Een machine, die de kondities benadert van de cylinder van een verbrandingsmotor, in het bijzonder ter plaatse van het bovenste deel van de slag.
Richtprijs exkl. B.T.W. ƒ 70.000,-
Model TE 82
De vier-kogelproef. Het testen van smeerolie tussen de oppervlakken van met verschillende snelheid draaiende kogels.
Richtprijs exkl. B.T.W. ƒ 85.000,-
Model TE 97
Een machine voor het meten van slijtage voor verschillende materialen in droge of in gesmeerde toestand.
Richtprijs exkl. B.T.W. ƒ 50.000,-

Verdere gegevens gelieve u in bijgaande dokumentatie aan te treffen.

Wij vertrouwen, dat de toegezonden gegevens u een eerste indruk geven van de mogelijkheden van het programma van PLINT & PARTNERS Ltd., en zijn gaarne bereid e.e.a. nader toe te lichten.

Hoogachtend,
LANDRE-MIJNSSEN B.V.

J. Gantzert

Figure D.1: Offer of several lubricator test equipment from 1981[17]

E Specifications of several commercially available two disc test machines

Table E.1: Specifications of the two disc test machines of Plint Tribology Products[13]

		TE 72 Standard	TE 72 High Power	TE 73	TE 74 Standard	TE 74 High Power	TE 103
Technical Specifications:							
Speed Range:	Master	0 - 3000 - 5000 rpm	0 - 3000 - 4500 rpm	100 to 7,500 rpm	0 - 3000 rpm	0 – 3000 – 3500 rpm	0 - 3000 - 6000 rpm
Motor Power:		0 - 10 -10 kW	0 - 65 - 65 kW	11 kW	0 - 5.5 kW	0 – 30 – 30 kW	0 - 65 - 65 kW
Speed Range:	Slave	0 - 3000 - 5000 rpm	0 - 3000 - 4500 rpm	N/A	0 - 3000 rpm	0 – 209 rpm & 0 to 800 rpm	0 - 3000 - 6000 rpm
Motor Power:		0 - 10 -10 kW ac	0 - 65 - 65 kW ac	N/A	0 - 5.5 kW	0 – 30 – 30 kW	0 - 65 - 65 kW
Maximum Torque:		32 Nm 0 - 3000 rpm	207 Nm 0 - 3000 rpm	115 Nm (nominal)	17.5 Nm	95 Nm @ 3000 rpm, 356 Nm @ 800 rpm, 1363 Nm @ 209 rpm	207 Nm 0 - 3000 rpm
Maximum Surface Velocity:		31 m/s (120 mm roller)	28 m/s (120 mm roller)	60 m/s	5.5 m/s	5 m/s (nominal)	48 m/s
Load Range:		0.25 to 15 kN	0.25 to 15 kN	0.25 to 15 kN	0.25 to 10 kN	0.25 to 36 kN	0.25 to 15 kN
Contact Geometry:	Point	Yes	Yes	Yes	Yes	Yes	Yes
	Diameter	25 mm to 120 mm	25 mm to 120 mm	152 mm	35 mm	25 mm on 125 mm dia	152 mm
	Radius	Variable	Variable	76 mm	Variable	Variable	76 mm
	Line	Yes	Yes	No	Yes	Yes	No
	Diameter	60 mm to 120 mm	60 mm to 120 mm	N/A	35 mm dia	25 mm on 125 mm dia	N/A
	Width	10 mm	10 mm	N/A	10 mm	25 mm	N/A
Slip:		Cont. variable	Cont. variable	Manual with gears	Cont. variable	Cont. variable	Cont. variable
Slip Rates:		0 to 100%	0 to 100%	0, 1, 2, 4, 6 & 8%	0 to 100%	0 to 100%	0 to 20%
Spin Roller Diameter:		N/A	N/A	120 mm	N/A	N/A	120 mm
Crown Radius:		N/A	N/A	20 mm	N/A	N/A	20 mm
Contact Elipticity:		N/A	N/A	2.36	N/A	N/A	2.36

Controlled Parameters:						
Speed	Automatic	Automatic	Automatic	Automatic	Automatic	Automatic
Slip Ratio	Automatic	Automatic	Manual	Automatic	Automatic	Automatic
Load	Automatic	Automatic	Manual	Automatic	Automatic	Manual
Fluid Temperature	Automatic	Automatic	Automatic	Automatic	Automatic	Automatic
Test Duration	Automatic	Automatic	Automatic	Automatic	Automatic	Automatic
Recorded Parameters:						
Speed of Rollers	Yes	Yes	Yes	Yes	Yes	Yes
Load	Yes	Yes	Yes	Yes	Yes	Yes
Slip Ratio	Yes	Yes	Yes	Yes	Yes	Yes
Traction Force	Yes	Yes	Yes	Yes	Yes	Yes
Temperatures	Yes	Yes	Yes	Yes	Yes	Yes
Test Duration	Yes	Yes	Yes	Yes	Yes	Yes
Vibration Level	Yes	Yes	Yes	Yes	Yes	Yes
Contact Resistance	No	No	Yes	No	No	Yes

Table E.2: Specifications of the Ducom Two Roller Rolling Sliding Tester[14]

Parameter	Unit	Min	Max	Remarks
Load	[N]	0	2000	
Speed	[rpm]	0	500	Variable
Temperature	[ºC]	Ambient		Heating option
Wear	[mm]	0	2	
Slip	[%]	0	20	
Fraction Force	[N]	0	2000	PID controlled
Roller Diameter	[mm]	40	80	Other sizes on request

F Visualisation of the working principle of the movable secondary test disc

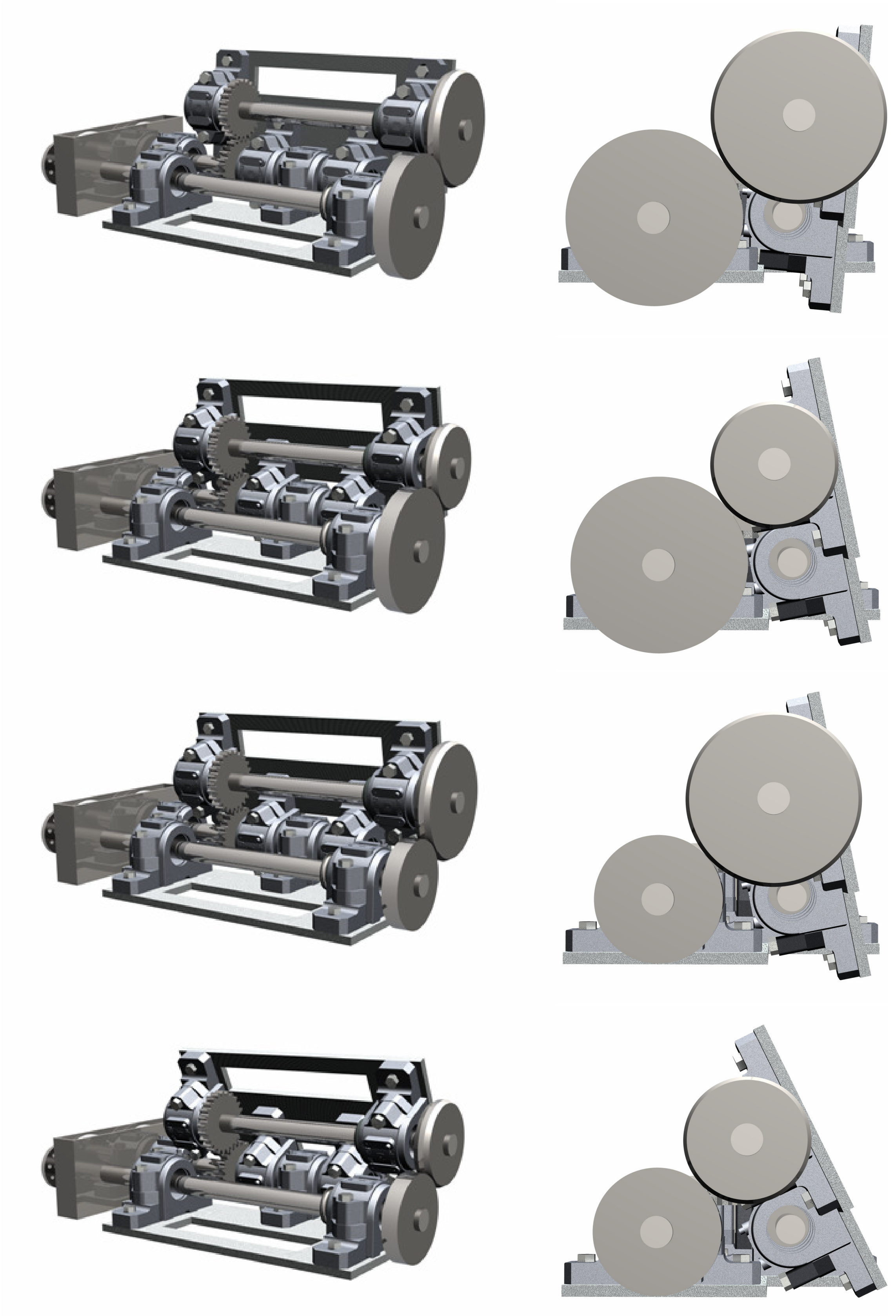

Figure F.1: Test discs with different radii can be installed on the test machine

List of symbols

Symbol	Description	[Unit]
v_{rA}	*Rolling velocity of test disc A*	*[m/s]*
v_{rB}	*Rolling velocity of test disc B*	*[m/s]*
v	*Surface velocity*	*[m/s]*
S	*Slip-roll ratio between the two test discs*	*[-]*
p	*Pressure*	*[Pa]*
x	*Position in x-direction*	*[m]*
y	*Position in y-direction*	*[m]*
z	*Position in z-direction*	*[m]*
η	*Viscosity*	*[N·s/m²]*
h	*Film thickness*	*[m]*
h_0	*Minimal film thickness*	*[m]*
$h_{(\sigma_{Hz})}$	*Film thickness at the point where the highest pressure exists*	*[m]*
h_d	*Part of the film thickness that is due to the limited stiffnes of the contact bodies*	*[m]*
h_h	*Geometric part of the film thickness*	*[m]*
R	*Radius of the equivalent cylinder for application in Reynolds equation*	*[m]*
R_1	*Radius of test disc 1*	*[m]*
R_2	*Radius of test disc 2*	*[m]*
r_1	*Radius of test disc 1*	*[m]*
r_2	*Radius of test disc 2*	*[m]*
μ	*Coëfficiënt of friction*	*[-]*
σ_{Hz}	*Hertzian stress*	*[Pa]*
ν	*Poisson ratio*	*[-]*
E	*Elastic modulus*	*[Pa]*
F_0	*Load force*	*[N]*
l_w	*Width of the test discs*	*[m]*
b	*Half of the width of the pressure distribution*	*[m]*
r_{UD}	*'Under Drive' ratio of the CVT*	*[-]*
r_{OD}	*'Over Drive' ratio of the CVT*	*[-]*
R	*Pitch diameter for computation on gear contacts*	*[m]*
r	*Pitch diameter for computation on gear contacts*	*[m]*
ω_p	*Rotational speed at the pitch line of gear 1*	*[rad/sec]*
ω_w	*Rotational speed at the pitch line of gear 2*	*[rad/sec]*
γ	*Angle between the path of contact and the common normal for gear contact*	*[°]*
ψ	*Angle that determines the fase of gear contact*	*[°]*
α	*Angle that determines the fase of gear contact*	*[°]*
β	*Angle that determines the fase of gear contact*	*[°]*

Acronyms

Symbol	Description
CVT	Continuously Variable Transmission
PLTR	Power Loop Test Rig