Applying a CVT in a two roller test machine

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Introduction

The CVT, known as the Continue Variable Transmission, is more and more used in cars to get a better performance of the car. In combination with a hybrid drive train, a decrease in fuel-consumption is achieved, which can be very large. Van Doorne's push-belt is a well-known device that makes a Continuous Variable Transmission possible. The CVT has an infinite number of ratios to shift within its limitations. But, are there not many other applications for which a push-belt is a useful component to transmit torque en speed?

At the university of Eindhoven there is an experimental setup called the "Power Loop Test Rig" (PLTR). This is a test machine that tests by means of a push-belt variator, the performance of gear-chain (also a variator belt, a pull-belt). The two CVT's are connected to eachother. A big advantage is that, in case of a test rig with a CVT-transmission, there can be used only one motor instead of two, to vary the working ratio of the test rig. A lot of torque can be transmitted between the two variators with a relative small electric motor, because only the losses of the test rig have to be compensated by the electrical motor. A lot of power can run through the test rig.

Additional research on tribology of the push belt, gear chain and pulleys has to be done on various types of materials and working conditions. A two-roller test machine is very suitable for this kind of research. Varies parameters can be prescribed and measured. Would it be possible to apply the same CVT-transmission from the PLTR onto a two roller test machine and what are the benefits.

The second reason why is looked at a two roller machine, is because the setup is qua layout very similar to the PLTR. There are two rotating parts that should be able to have different rotational speeds (a ratio) at the shaft ends. Existing configurations make use two electric motors or a fixed gear step, but could also be obtained by a CVT-transmission. But would it be possible to adapt the CVT-transmission of the PLTR and make it suitable for this purpose?

Problem formulation and assignments

A research on two roller test machines has been done [1]. In this report there is more information about:

- What kind of two roller test machines are there
- How they work
- Where they are used for
- What their working principle is and their working range
- Who develop and produce these machines
- Who are using these machines

The main question was: 'Is the application of a CVT in a two roller test machine feasible?' The answer is that it is feasible to apply a CVT onto a two-roller machine.

The goal of this research is:

- What are the benefits of applying a CVT in a two-roller test machine
- Is it possible to fix the conventional CVT of the PLTR onto an existing two-disc test machine and what should be adapted.
- What are the dimensions of the gears, shafts and bearings for this adaptation.
- There are also other test machines (testing couplings, gears, etc.) in which the CVT unit of the PLTR can be a useful device. Design the driving unit such that it can be implemented on a broad range of test machines.

Summarized the assignment is:

Redesign the drive of a two roller test machine by means of the application of a CVT transmission.

Because in the beginning of the research it was not completely clear what the assignment would be, some additional work has been done and is adapted in this report. It has a strong relation with the topic, but will not be included in the introduction and conclusion. It is also too important to be put in the appendix. If not, the information is lost, and has to be looked up by future students on this topic.

Chapter 1

The two roller test machine

A two roller test machine is a test machine that consists of two rollers that are pushed against each other and they create Hertzian- contact. This principle is used for several purposes:

- Oil can be characterized
- Gear-contact can be simulated. See the left picture in figure 1.1
- By using a train-wheel profile on one disc and on the other a track, train wheel contact can be simulated
- Cam/follower systems can be characterized
- With an adapter spin can be simulated. This is behavior, which occurs in a push-belt variator (on the right in figure 1.1). Originally designed for research of toroidal transmissions)

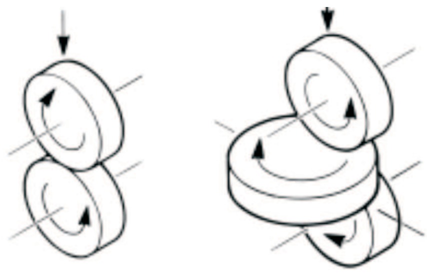

Figure 1.1: Contact types

Especially for gear and cam follower contact testing varying the ratio of slip is applied. This is because of when gears rotate, the contact between the teeth of the gears have a rolling and sliding contact with each other at the pitch circle to transmit torque and speed.

1.1 Types of two roller test machines

There are two main types of two roller test machines using each its own principles to achieve slip (a ratio between the discs):

- By using two motors on each shaft of each roller
- With fixed gear steps using one motor (gears and discs mounted back to back)

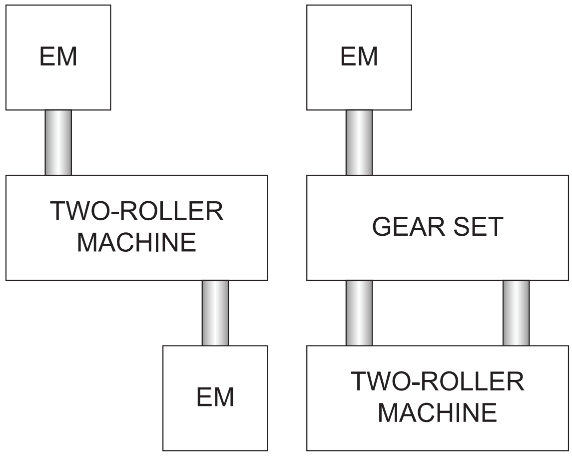

Figure 1.2: Layouts two roller test machines

On the left in figure 1.2, the layout in case of two motors is presented, at the right the layout with fixed gear steps. The advantage of the first possibility is that the slip ratio can be varied continuously and stepless. A disadvantage is there are two motors. One is supplying power and the other regenerating power. Because it is not possible to work in the best efficiency points of the two motors a lot of power is dissipated when supplying and regenerating power. The advantage of the fixed gear step is that you know the ratio and only only the efficiency losses of the test rig (gears, disc and bearing efficiencies) has to be compensated by the motor and a much smaller motor can be used. The disadvantage is that the ratio is fixed. Changing the ratio means that the gears have to be changed.

By applying a CVT in a back to back configuration it is possible to combine best of both possibilities. Only one power source is needed and the ratio can be varied continuously. So it is power saving and continuously variable in ratio. There is only one disadvantage in comparison with the setup with the fixed gear step. There are more efficiency losses in the test rig with the CVT because of the lower total efficiency of the test rig in comparison to the setup with a fixed gear set. This is efficiency is lower because of the extra gears, bearings and the CVT. A bit heavier electric motor is needed to maintain the power level that runs through the test rig, as in the other setups. The test rig with the adapted CVT drive is called from now on the Power Loop Test Machine (PLTM).

Chapter 2

Applying a CVT

2.1 Comparison of the technical specifications

To look whether it is possible or not to apply the CVT-transmission of the PLTR onto a two roller test machine, the technical specifications of both machines have to be compared. The technical specifications of the TE 73, a generally used two roller test machine of Plint, and of the CVT are presented in table 2.1 and 2.2:

By comparing these dimensions, it appears that the CVT-drive of the PLTR is capable of fulfilling the demands of the TE 73. It is quite over dimensioned; for instance the maximum torque is twice as high as necessary. This will result in a long lifetime of the CVT drive. There are smaller CVT-drives that might also be capable of fulfilling these demands. These are for example the dry band CVT (pull belt) that is used for driving the cooling fan of some DAF trucks or the one out of a smaller test rig out of the AES-lab.

There are two problems. The CVT is not capable of running at 7500 [rpm]. By some adjustment this will be possible. The configuration that is presented further on should be redesigned. At both output shafts a gear step with a ratio is needed of:

$$i = \left(\frac{\omega_{out}}{\omega_{in}}\right) = \left(\frac{7500}{6000}\right) = 1.25 \ [-]$$

The CVT has limited slip ratio of 39% that is sufficient for the TE 73, but not for the other Plint machines. If a broader range is desired the overall gear ratio of the second gear stage should be adapted so it will be sufficient. The design is optimized for 8% slip.

Speed range:	$100 - 7,500$ $[rpm]$
Rolling velocity:	$0.75 - 60$ $[m/s]$
Maximum torque:	115 $[Nm]$
Load range:	$0.25 - 15$ $[kN]$
Maximum Hertz stress:	2.8 $[GPa]$
Roller diameter:	152 $[mm]$
Crown radius:	76 $[mm]$
Slip rates:	0%, 1%, 2%, 4%, 6% and 8%
Motor Power:	11 $[kW]$ dc

Table 2.1: Technical specifications of the TE 73

Speed range:	$0 - 6000$ $[rpm]$
Rolling velocity:	$0.75 - 60$ $[m/s]$
Maximum torque:	250 $[Nm]$
Minimum ratio:	0.43 $[-]$
Maximum ratio:	2.30 $[-]$
Possible slip ratio:	-39% - 39%
Shaft distance output of the CVT:	168 $[mm]$

Table 2.2: Technical specifications of the CVT

2.2 The layout with the PLTM

Combining the setups of figure 1.2 and the CVT, results in a first sketch of how the test rig will look like. A first impression is given in figure 2.1. The CVT replaces the two motors and the fixed gear step. In chapter 3 the design is explained in more detail. A description of what the functions of the different components are and why they are designed this way.

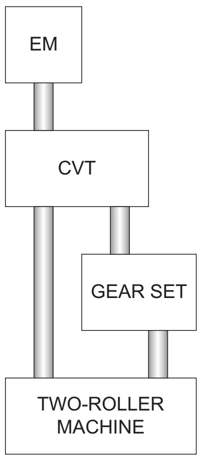

Figure 2.1: Layouts test machines

2.3 Test Rollers

The PLTM, as well as the test rig with the fixed gears have their limitations on the sizes of the test discs. The problem occurs with small disc sizes. The sizes of the gears, bearings and shafts become too large to be able to apply small discs. These components would then touch or intersect. Then the possibility of two motors, facing each other, is used. Small discs, means smaller powers and smaller bearings and thus smaller radial loads. A fixed gear pair results in a fixed center distance of the test discs. The extra gear pair of the PLTM makes it possible to apply test discs that can vary in diameters. Not only the drive side of the TE 73 should then be redesigned, but also the unit of the test rollers. This is not the goal of this research, but it is an extra possibility that is created by the application of the new drive of the PLTM.

Previous research [1] showed that the CVT of the PLTR is suitable as a transmission for the TE 73. This test machine is generally used and has a broad working range, and is covering most of the working conditions of all test machines of Plint, appendix A.2.

Chapter 3

Machine layout, design and basic calculations

3.1 The TE 73

Very little is known about the TE 73. A complete machine layout (sketch) is made of how the complete machine will probably look like. Probably, because there are of course no drawings of the TE 73 available. Only basic dimensions are given in the leaflet appendix A.1. Out of this leaflet a global sketch is made of how things probably look like. In figure 3.1 the configuration of the components is visible. In the table below the figure, the components are named and enumerated. At the right side in figure 3.1 two gears with different sizes are drawn which make a fixed ratio ($\neq$ 1 [$-$]) creating slip.

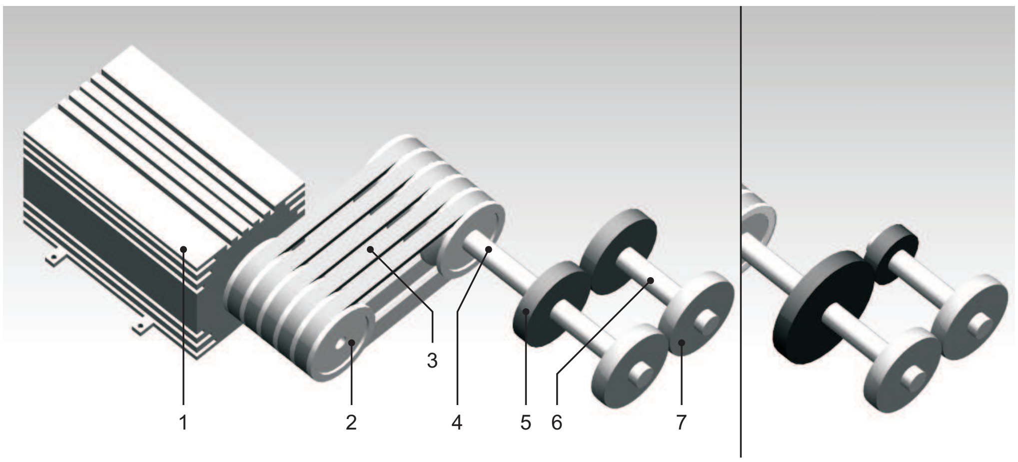

Figure 3.1: The TE 73

1	Electric motor
2	V-pulley
3	Rubber v-belt
4	Primary shaft
5	First gear
6	Secondary shaft
7	Test disc

3.2 Layout of the PLTM drive

Starting point of designing the new drive for the PLTM is the radius of the test discs. The discs prescribe the center distance of the shafts on which the CVT drive has to be adapted. Only a redesign of the drive has to be done so the accent is only on the drive side and a more detailed design of the drive side is given. The design is such that the output distance of the shafts can vary. This is important because the drive can easily be converted (other radii of the second gear set) for other test machines. The shaft distance range that is possible is limited by the size of the gears of the second gear stage and the thickness of the corresponding shafts. The discs are counter rotating. This is obtained with the extra gear at the center of the second gear set. A layout of how the new drive looks like is given below in figure 3.2.

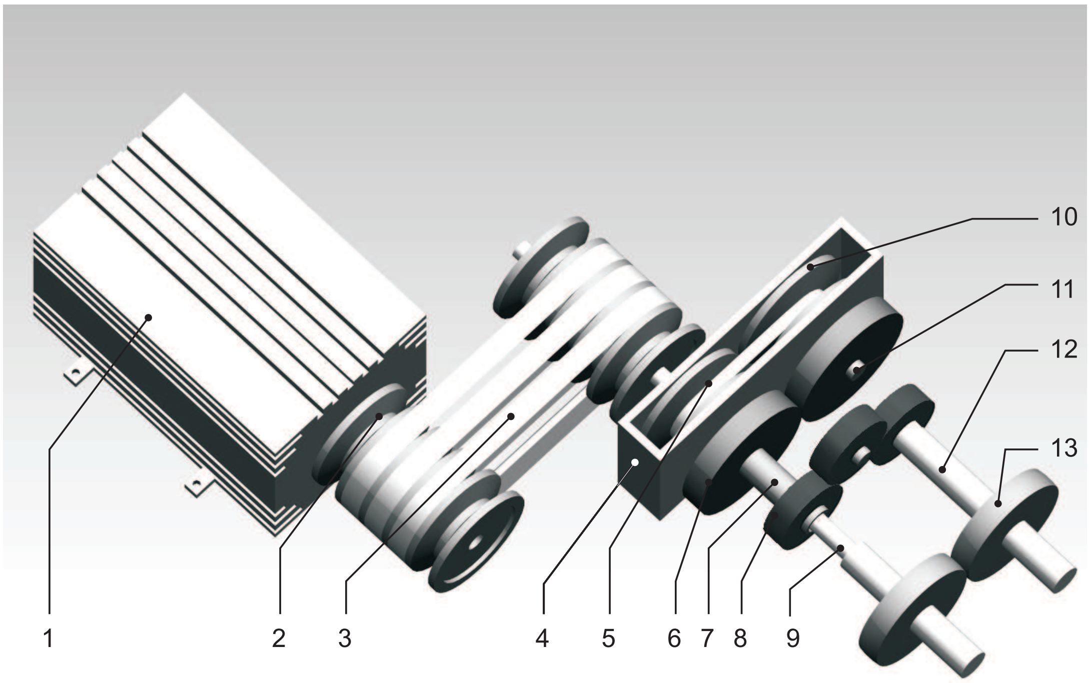

Figure 3.2: The PLTM

1	Electric motor	8	First gear of second gear set
2	V-pulley	9	Primary shaft
3	Rubber belt	10	CVT pulley
4	CVT housing	11	Output shaft of the CVT
5	CVT-belt	12	Secondary shaft
6	Second gear of first gear set	13	Test disc
7	Hollow intermediate shaft		

3.3 Design of the PLTM

The numbers in the text below refer to figure 3.2.

The transmission from the motor (1) to the primary shaft (9) remains the same. It consists of a rubber belt drive (2,3) from the motor to the primary shaft. It functions as a low bypass filter. Motor noise will be flattened and will affect the measurements less [6]. The primary shaft is also connected with the first pulley set (5) of the CVT (4) and runs through the hollow intermediate shaft (7) and at the end of the primary shaft the first test disc (13) is attached. The first test disc drives the test second disc. The second disc drives the secondary shaft (12). At the beginning of this shaft the last gear of the secondary gear set (8) is fixed. This gear drives the middle gear of this set, which can move up and down and back and forward with respect to the shafts primary and secondary shafts. These movements make it possible that the center distance of these shafts can vary in order to apply several disc sizes as well as for other applications in other machines. The middle gear drives the first gear of this set. This gear is positioned at the end of a hollow intermediate shaft (7). At the beginning the second gear of the first gear set (6) is attached. This gear drives drives the other gear, which is connected on the second output shaft of the CVT (11). This shaft is connected with the pulleys of the CVT and these are connected with the CVT belt (5).

3.4 Applying a flywheel

In the TE 73 a flywheel is not present. In Twente they did use a flywheel on their test rig. For measurements with a constant speed and torque a flywheel can be added at the input pulley of the CVT. The design with the flywheel is shown in appendix A.3. Some additional sheave grooves should be available a the pulley of the motor. When not using the flywheel, it can be used for extra belts for the input shaft, when needed. The flywheel has a certain inertia and when it is spinning this results in even more steady operation of the test rig and thus measurements. When dynamic measurements take place, for which the drive is very suitable, the flywheel should be disengaged, because then it acts as a load. A flywheel is not suitable for dynamic measurements [2].

3.5 Determination of the motor power for the PLTM

To determine the motor power of the new motor for the PLTM, an estimation of the losses of the original test rig has to be made. Two parallel shafts are connected with a gear at one side. At the end of these shafts the discs are attached. This configuration is shown in figure 3.1 and is called "back to back." The gears are form closed and determine the angular speeds of the shafts, thus the discs. The power from the input shaft runs through the primary shaft to first disc. This disc drives the second disc. The power runs back through the secondary shaft to the second gear and is returning power to the first shaft via the first gear. To make an estimation of the losses the worst case scenario should be considered. The efficiency losses in helical gears are typically 97%. The highest losses of the disc occur when the largest amount of slip is applied. This is 8% in this rig. 8% slip means that 92% of the power is transmitted to the secondary shaft. The exact number of bearings is not known. Usually they provide low losses, but when a lot of bearings are applied and they are highly loaded they do affect the efficiency. An estimation of 2.5% efficiency loss is considered. The total efficiency can then be calculated.

$$\eta_{total} = \eta_{gears} \cdot \eta_{discs} \cdot \eta_{bearings} \cdot \eta_{CVT} \tag{3.1}$$

Out of equation 3.1 the total efficiency can be derived and is:

$$\eta_{total} = 0.97 \cdot 0.92 \cdot, 9875 = 0.88,\ 88\%$$

This is used to calculate the total power running in the rig. The losses supplied by the motor are $11[kW]$, which is equal to 12%. So the power running in test rig is equal to $80[kW]$. With the maximum input power, torque and maximum angular velocity en and formula 3.2 and the specifications of table 2.1, the maximum power line of the electric motor can be calculated with use of Matlab the maximum motor power line is drawn in figure 3.3.

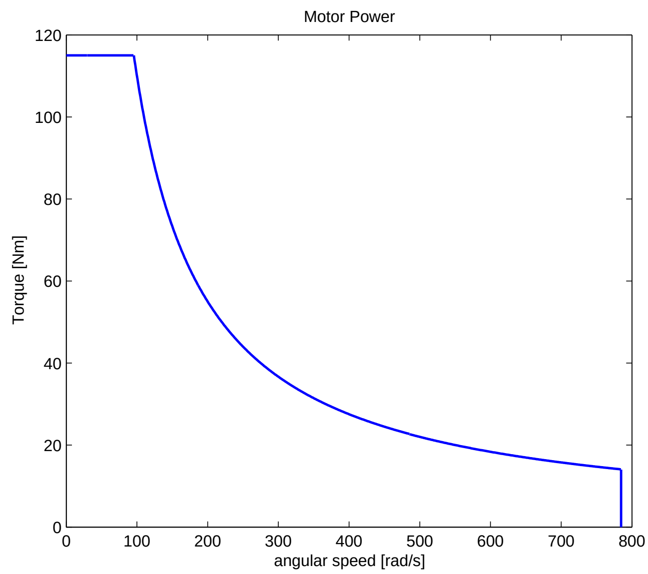

Figure 3.3: Motor characteristic of the electric motor

$$P = T \cdot \omega \tag{3.2}$$

3.6 The new drive for the PLTM

In the new test rig are more gears, bearings present and also a CVT added, so there will be more losses than in the original rig. An even amount of power is needed at the end of the primary and secondary shafts at the discs to be sure that the new drive of the PLTM is capable of the same working conditions as the TE 73. The slip ratio for which the drive is designed is 8%. A larger slip ratio is possible, but with less power.

3.7 Power calculations for the PLTM

An approach of total efficiency of the drive of the PLTM can now be made. With equation 3.1 the efficiency can be determined. There are three gear pairs, a CVT, the discs and the bearings. The losses in the bearings are assumed to be twice as high. The worst mechanical efficiency of the CVT is considered, which is 94%. Then the total efficiency is:

$$\eta_{total} = 0.97 \cdot 0.97 \cdot 0.97 \cdot 0.92 \cdot 0.975 \cdot 0.94 = 0.77,\ 77\%$$

77% is equal to 80[kW] power in the test rig. The motor should add the efficiency losses of the test rig, which are 23%. That is equal to 24[kW]. This is quite a large motor. The worst case is taken in to account and the efficiencies are probably not so poor. This is an assumption and the motor can be resized when the real efficiency losses are known. The size of the test rig is still smaller then in the case of applying two motors. And it is only a little larger then the original drive of the TE 73.

3.8 Gears

The type of gears that will be used are helical gears. This type of gear runs more smoothly than spur gears , because more teeth are in contact due to a skewed tooth profile (helical form). The mesh and release of the gears is more gradual. Helical gears have a higher load-carrying capacity and run more quietly (much smoother) than equivalent spur gears, especially for higher rotational speeds. Due to the smoother running gears, less noise is generated thus recorded by the several sensors. A disadvantage is that the skewed profile results in an axial force. The gears need active lubrications because the velocity at the pitch circle is higher than 15[m/s] for the higher speeds.

3.8.1 First gear set

The size of the gears of the first gear set have the same diameter as the shaft distance of the CVT, which is 168 [mm]. To make sure that the gears have a long lifetime the number of teeth of the gears have to be chosen with a difference of one tooth. Otherwise the same teeth always contact with each other and they wear in. The small ratio that is created can be eliminated through the CVT. The chosen pitch is 2.5 [mm]. This is a standardized pitch and makes sure that there are enough gear teeth at the pitch circle. A larger pitch means less teeth at the pitch circle, so less teeth in contact at the same time. Desired is that are about two teeth are in contact. With help from the online gear calculator [3] a first approach is made. In table 3.8.1 the dimensions that are presented.

Shaft distance	168 [mm]
Pitch	2.5 [mm]
Pitch diameter pinion/wheel	166.8/169.31 [mm]
Number of teeth pinion/wheel	66/67 [$-$]
Gear width	20 [mm]
Ratio	1.01 [$-$]
Contact ratio (number of teeth in contact)	2.01 [$-$]
Maximum driving torque:	335 [Nm]
Tooth angle (helical gear)	20^o
Maximum tangential force:	4 [kN]
Maximum axial force:	1.45 [kN]
Ratio:	1.01 [$-$]

The maximum allowable torque, tangential and axial forces will never be reached, the resulting forces of the gear teeth under the highest possible working conditions are much lower. The width of the gear can not be smaller, because there are less then two teeth in contact and the stability of the gear is lowered.

3.8.2 Second gear set

The distance that has to be covered by the second gear pair is 152 [mm]. This is the center distance of the test discs. The diameter of the first gear of the second pair is chosen 100 [mm].

The three gears have the same size, because then there is no lowering of the gear train stiffness. The speed range is from 0 till 6000 $[rpm]$. Because the diameter of the gears are smaller, a smaller pitch is chosen out of [4] to make sure that there are enough teeth in contact with each other. A first assumption is made with online calculator [3] and the dimensions are listed in 3.8.2.

Shaft distance	152 $[mm]$
Pitch	1.5 $[mm]$
Pitch diameter pinion/wheel	100.5/99/100.5 $[mm]$
Number of teeth pinion/wheel	50/49/50 $[-]$
Gear width	30 $[mm]$
Contact ratio (number of teeth in contact)	2.03 $[-]$
Maximum driving torque:	235 $[Nm]$
Tooth angle (helical gear)	20^o
Maximum tangential force:	4.5 $[kN]$
Maximum axial force:	1.6 $[kN]$
Ratio:	1 $[-]$

3.9 Shafts

The diameters of the shaft have to be designed strong enough so they will not bend too much and will be able to transmit the prescribed torque and rotational speed. The maximum allowable bending is the major factor in the shaft design, which prescribes the thickness of the shafts. Standard ball bearings are used. The maximum degree of misalignment of this type of bearing is between 2 and 10 minute of arch ($1/6^o$) [6]. For the design 2 minutes of arch is considered.

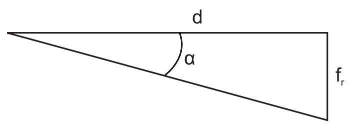

Figure 3.4: Minute of arch

In figure 3.4 the angle and lengths are shown. Once the maximal bending f_r is computed and the shaft thickness is determined by the resulting forces, moments and torque acting on the shaft. The load of the test discs are in the test disc unit and are not taken into account. In figure 3.5 an intersection of the shafts gears and bearings is visible. The lengths of the components are at the bottom. Number 1 is the output shaft of the CVT, number 2 the secondary shaft, number 3 the shaft of the middle gear of the secondary gear set, number 4 the primary shaft and number 5 the intermediate shaft.

1	Output shaft of the CVT	6	CVT housing
2	Secondary shaft	7	1st gear of 1st gear set
3	Shaft of the 2nd gear	8	2nd gear of 1st gear set
	of the 2nd gear set	9	1st gear of 2nd gear set
4	Primary shaft	10	2nd gear of 2nd gear set
5	Hollow intermediate shaft	11	3th gear of 2nd gear set

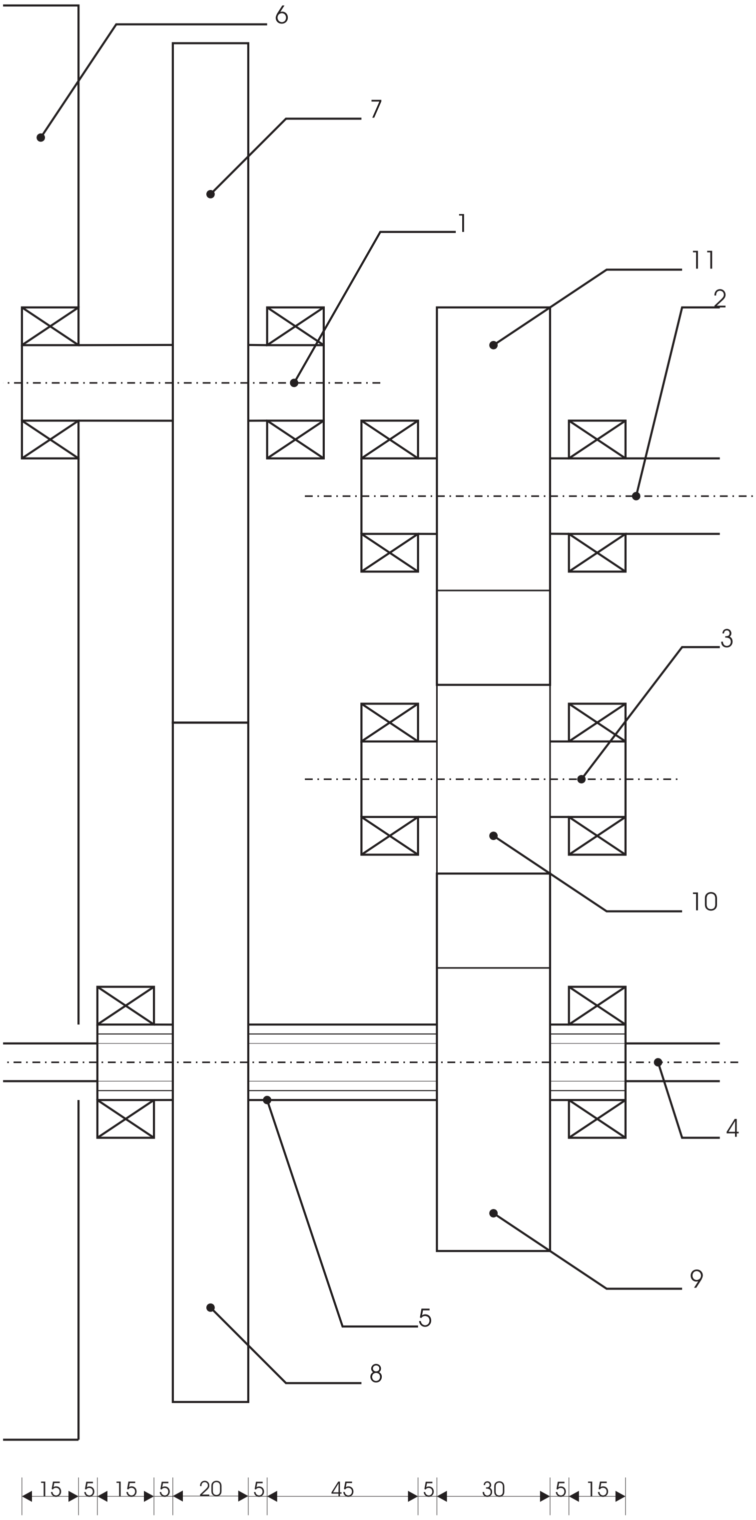

Figure 3.5: An intersection of the gears, shaft and bearings of the PLTM drive

3.10 Calculations on shaft design

The torque applied by the motor results in a torque in the test rig. This torque generates loads that acts on the shaft. The equations, which are needed for the shaft design are presented below.

With the first equation, equation 3.3, the resulting forces of the torque on the gear thus on the shaft can be calculated.

$$T = F \cdot r \tag{3.3}$$

With the second two equations, equation 3.4 and 3.5, the resulting forces at the centers of the bearings can be calculated.

$$\sum F = F_1 + F_2 + F_3 = 0 \tag{3.4}$$

$$\sum M_{tov\ A} = M_1 \cdot l_1 + M_2 \cdot l_2 + M_3 \cdot l_3 = 0 \tag{3.5}$$

When the forces are known they can be drawn in a figure against the length of the shaft. Out of this figure the bending moments can be determined and be putted in a figure against the length as well. In the shaft a certain torque is present for a certain length due to the motor torque or to the gears. This results in a figure where torque is set against the length of the shaft. Out of these three figures the most critical point will be determined and at this point the shaft has to resist the highest loads. The thickness of the shaft should be able to resist these loads.

A little beding of the shaft is allowed in order of the bearings and gears. With equation 3.6 this can be calculated. $\frac{1}{30}$ of a degree equals 2 minutes of arch.

$$f_r = l_1 \cdot \tan \frac{1}{30} \tag{3.6}$$

The sagging due to the forces can be determined with the following equation:

$$f_r = \frac{F \cdot l^3}{48 \cdot E \cdot I} \tag{3.7}$$

With I:

$$I = \frac{\pi}{64} \cdot d^4 \tag{3.8}$$

The smallest safe diameter follows out of the following equation 3.9

$$d = \left(\frac{32 \cdot S}{\pi \cdot \sigma_y} \cdot \sqrt{M^2 + T^2} \right)^{1/3} \tag{3.9}$$

3.10.1 The primary shaft

The primary shaft, denoted by number 4 in figure 3.5, is a shaft that only has to transmit torque. The thickness of this shaft is determined by the torque it has to transmit. This is 115 $[Nm]$. With equation 3.9 the diameter of this shaft can be determined and will be:

$$d = \left(\frac{32 \cdot 2}{\pi \cdot 130 \cdot 10^6} \cdot \sqrt{0^2 + (115)^2} \right)^{1/3} = 26.2\ [mm]$$

The diameter is standardized and will be 30 $[mm]$.

3.10.2 Output shaft of the CVT

In figure 3.6 the configuration of the shaft, bearings and gear is given. Due to the maximum torque, the maximum reaction force of the gear on the bearings can be calculated with equation 3.3, the gear diameter and the maximum torque.

$$\left.\begin{array}{l} T = F \cdot r \\ T = 115\ [Nm] \\ d = 0.168/2\ [m] \end{array}\right\} F_{max} = 1400\ [N]$$

This force generates two reaction forces at the centers of the bearings. With equations 3.4 and 3.5 and the lengths out of figure 3.6 they can be determined:

$$\left.\begin{array}{l} F_{T1} = 1400\ [N] \\ \sum F = F_{L11} + F_{L12} - F_{T1} = 0 \\ \sum M_{tov\ A} = F_{L12} \cdot l_{AC} - F_{T1} \cdot l_{AB} = 0 \end{array}\right\} \begin{array}{l} F_{L12} = 915\ [N] \\ F_{L11} = 485\ [N] \end{array}$$

The maximum torque, which is 115 $[Nm]$, enters the shaft before point A and is transmitted in point B to the next gear. Out of these loads the point where the highest loads are present occurs. This is point B.

The axial force that is generated through the skewed teeth profile of the gear is:

$$F_{A1} = 1400 \cdot \tan(20) = 510\ [N]$$

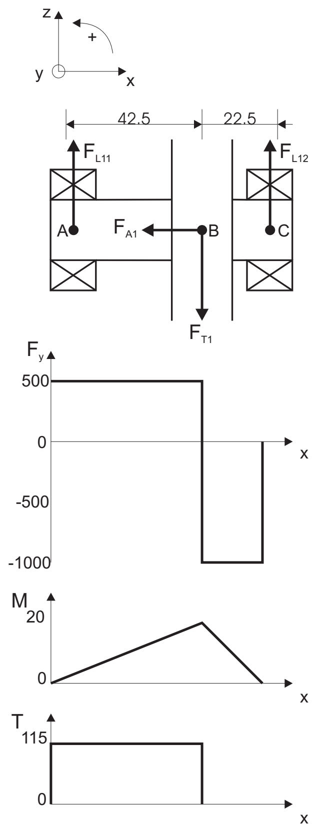

Figure 3.6: Dimensions, forces, bending moment and torque of the output shaft of the CVT

With all the loads and lengths the critical diameter of the shaft can be determined. A safety factor of 2 is considered. The yield strength of steel is 130 $[MPa]$. Then the shaft diameter becomes:

$$d = \left(\frac{32 \cdot 2}{\pi \cdot 130 \cdot 10^6} \cdot \sqrt{20^2 + 115^2} \right)^{1/3} = 26.4\ [mm]$$

This is not a standard thickness. The chosen thickness is 30 $[mm]$. The last thing at which the shaft should fulfill is the sagging. Equation 3.7 determines the maximum allowed sagging at the center of the shaft f_r:

$$f_r = 0.0225 \cdot \tan \frac{1}{30} = 1.3 \cdot 10^{-5}\ [m]$$

Under the given circumstances with equations 3.7 and 3.8 it is:

$$I = \frac{\pi}{64} \cdot 0.030^4 = 4 \cdot 10^{-7}\ [m]$$

$$f_r = \frac{1400 \cdot 0.0225^3}{48 \cdot 210 \cdot 10^9 \cdot 4 \cdot 10^{-7}} = 4 \cdot 10^{-9}\ [m]$$

This is much lower than it has to be. The shaft will be able to run under the given conditions without any trouble.

3.10.3 Intermediate shaft

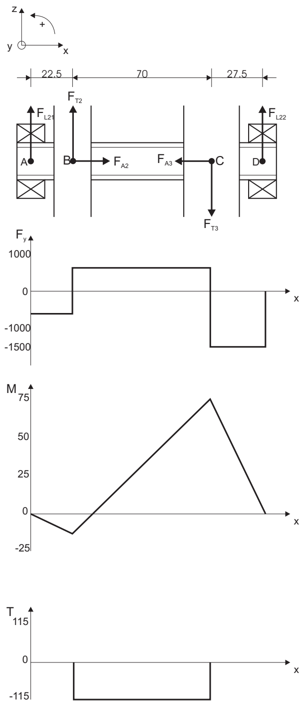

Figure 3.7: Dimensions, forces, bending moment and torque of the intermediate shaft

In figure 3.7 the shaft and lengths of the intermediate shaft are presented. The maximum reaction forces of the gears on the bearings are calculated with equation 3.3, the gear diameters and maximum torque.

$$\left.\begin{array}{l} T = F \cdot r \\ T = 115\ [Nm] \\ d = 0.168/2\ [m] \end{array}\right\} F_{max} = 1400\ [N]$$

$$\left.\begin{array}{l} T = F \cdot r \\ T = 115\ [Nm] \\ d = 0.100/2\ [m] \end{array}\right\} F_{max} = 2300\ [N]$$

These forces generates two reaction forces at the centers of the bearings. With equation 3.4, 3.5 and figure 3.6 they can be determined:

$$\left.\begin{array}{l} F_{T2} = 1400\ [N] \\ F_{T3} = 2300\ [N] \\ \sum F = F_{L21} + F_{L22} + F_{T2} - F_{T3} = 0 \\ \sum M_{tov\ A} = F_{T2} \cdot l_{AB} + F_{L22} \cdot l_{AD} \\ \qquad -F_{T3} \cdot l_{AC} = 0 \end{array}\right\} \begin{array}{l} F_{L22} = 1510\ [N] \\ F_{L21} = -610\ [N] \end{array}$$

Together with the maximum torque, which is 115 $[Nm]$, entering in point B and transmitted in point C to the next gear. The point where the highest loads occur is point C.

The axial forces that are generated through the skewed teeth profile of the gears are:

$$F_{A2} = 1400 \cdot \tan(20) = -510\ [N] F_{A3} = 2300 \cdot \tan(20) = 840\ [N]$$

A resulting axial force occurs, which is

$$F_r = 330\ [N]$$

With all the loads and lengths the critical diameter of the shaft can be determined. Again the safety factor is 2 and the yield strength of steel 130 $[MPa]$. An extra factor k for the ratio between the inner and outer diameter has to be added in equation 3.9. The equation for the minimum outer diameter becomes:

$$d_o > \left(\frac{32 \cdot S}{\pi \cdot \sigma_y \cdot (1 - k^4)} \cdot \sqrt{M^2 + T^2}\right)^{1/3} \tag{3.10}$$

$$k = \frac{d_i}{d_o} \tag{3.11}$$

The shaft diameter is determined iteratively:

$$k = \frac{35}{40} = 0.875$$

$$d_o > \left(\frac{32 \cdot 2}{\pi \cdot 130 \cdot 10^6 \cdot (1 - 0.875^4)} \cdot \sqrt{73^2 + (-115)^2} \right)^{1/3} = 37.2 \ [mm]$$

The only thing that remains at which the shaft should fulfill is the sagging. The allowable sagging at the center of the shaft f_r is:

$$f_r = 0.06 \cdot \tan \frac{1}{30} = 3.5 \cdot 10^{-5} \ [m]$$

In point C occur the highest loads. With equations 3.7 and 3.8 the sagging is calculated and will be:

$$I = \frac{\pi}{64} \cdot (0.040^4 - 0.035^4) = 2.67 \cdot 10^{-5} \ [m]$$

$$f_r = \frac{2300 \cdot 0.0275 \cdot \sqrt{(0.12^2 - 0.0275^2)^3}}{9 \cdot \sqrt{3} \cdot 210 \cdot 10^9 \cdot 2.67 \cdot 10^{-5} \cdot 0.12} = 1.53 \cdot 10^{-11} \ [m]$$

The demand on sagging is fulfilled. Chosen is for standardized shaft of 40 $[mm]$ with an inner diameter of 35 $[mm]$. The thickness of the primary shaft is 30$[mm]$. When the shafts are running a backlash of 2.5$[mm]$ is present, so the shaft will not touch each other.

3.10.4 Secondary shaft and the shaft of the second gear of the secondary gear set

Because the secondary shaft and the shaft of the second gear of the second gear set have the same working conditions they are presented together. The only difference is the directions of the forces. They are the opposite for the secondary shaft in order of each other.

With the maximum torque and distance between the gear and bearings, the maximum reaction force of the gear on the bearings can be calculated with equation 3.3.

$$\left.\begin{array}{l} T = F \cdot r \\ T = 115\ [Nm] \\ d = 0.100/2\ [m] \end{array}\right\} F_{max} = 2300\ [N]$$

This force generates two reaction forces at the centers of the bearings. With equation 3.4, 3.5 and figure 3.6 they can be determined:

$$\left.\begin{array}{l} F_{T3} = 2300\ [N] \\ \sum F = F_{L41} + F_{L42} + -F_{T4} = 0 \\ \sum M_{tov\ A} = F_{L42} \cdot l_{AC} - F_{T4} \cdot l_{AC} = 0 \end{array}\right\} \begin{array}{l} F_{L42} = 1150\ [N] \\ F_{L41} = 1150\ [N] \end{array}$$

Together with the maximum torque entering in point B and transmitted in point B to the next gear the point where the highest loads occur is point B for both shafts.

The axial force that is generated through the skewed teeth profile of the gear is:

$$F_{A4} = 2300 \tan(20) = 840\ [N]$$

With all the loads and lengths the critical diameter of the shaft can be determined. Again the safety factor is 2 and the yield strength of steel 130 $[MPa]$.

The shaft diameter becomes:

$$d = \left(\frac{32 \cdot 2}{\pi \cdot 130 \cdot 10^6} \cdot \sqrt{31^2 + (115)^2}\right)^{1/3} = 26.5\ [mm]$$

The thickness of the shaft will be standardized on 30 $[mm]$.

The only thing that remains at which the shaft should fulfill is the sagging. The allowable sagging at the center of the shaft f_r is:

$$f_r = 0.0225 \cdot \tan\frac{1}{30} = 1.3 \cdot 10^{-5}\ [m]$$

Under the given load the sagging is calculated with equations 3.7 and 3.8 and will be:

$$I = \frac{\pi}{64} \cdot 0.030^4 = 4 \cdot 10^{-7}\ [m]$$

$$f_r = \frac{2300 \cdot 0.0275^3}{48 \cdot 210 \cdot 10^9 \cdot 4 \cdot 10^{-7}} = 4 \cdot 10^{-9}\ [m]$$

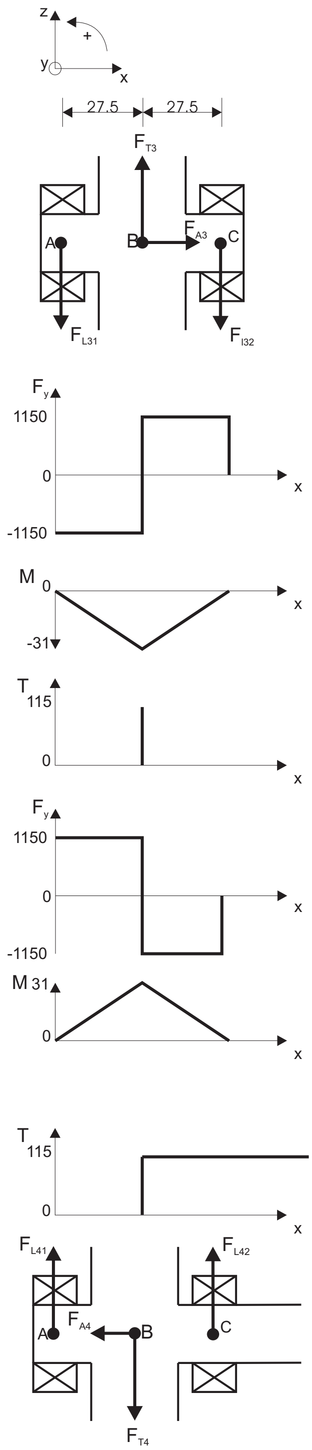

Figure 3.8: Dimensions, forces, bending moment and torque of the second gear shaft

The demand on sagging is fulfilled. A diameter of 30 [mm] is possible.
The shaft will be able to run under these loads without any problems

3.11 Bearings

The kind of bearings that are used in the design are standard single row deep groove ball bearings. They are suitable for the following working conditions:

- radial loads
- axial loads in both axial directions
- combined loads in both axial directions
- high speeds
- high running accuracy
- high stiffness
- run quietly
- low friction
- locating bearing locations in both directions
- non-locating bearing locations

But not so suitable for:

- moment loads
- compensation for misalignment
- combined loads in both directions
- no axial displacement possible in the bearing

Looking at the working conditions this kind of bearing can be used. Ball bearings are cheap and they are easy to apply in constructions. At one end the bearing has a fixed position. This is the side where the resulting axial force is pointing to. At the other side the bearing has a little axial backlash, so it can slide a little bit due to thermal expansions. These kind of bearings can not withstand axial displacements. It is very important that the bearings of each shaft are aligned very well in order to each other. Choosing the roller bearings is done with help of the site of SKF [6]. In table 3.11 the specifications of the roller bearings that have to be fulfilled are summarized.

Shaft:	thickness $[mm]$	F_{radial} $[N]$	F_{axial} $[N]$
Primary shaft	30	0	0
Output shaft CVT	30	915	510
Intermediate shaft	40	1510	330
Secondary shaft:	30	1150	840
Shaft of sec. gear: sec. gear set	30	1150	840

3.11.1 Bearings for the 30 [mm] shafts

The first bearing that has been selected is the bearing for the 30 [mm] shafts, with designation 6306 from the SKF catalogue. It is the left bearing in figure 3.9. The dimensions are in millimeters. Under the heaviest working conditions, which will never appear (running at 7500[rpm] with an radial load of 1510 [N] and axial load of 840[N]), the bearing has a lifetime calculated with the online calculator of SKF, of 214.000 [hrs].

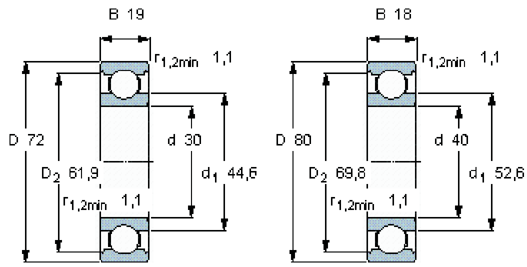

Figure 3.9: Dimensions of the selected bearings

3.11.2 Bearings for the 40 [mm] shafts

The second bearing for the hollow intermediate shaft has an outer diameter of 40 [mm]. The selected bearing has similar quantities as the bearing for the shaft of 30 [mm], table 3.11.2. The lifetime of this bearing under the same working conditions is 302.000 [hrs].

inner diameter	dynamic load	static load	limiting speed	designation of SKF
30	29.6 [kN]	16 [kN]	13.000 [rpm]	6306
40	32.5 [kN]	19 [kN]	11.000 [rpm]	6208

Conclusions and recommendations

After a study of applying a CVT in a two-roller machine it appears to be possible to apply the CVT-drive of PLTR onto the machine of Plint, the TE 73. The CVT-drive of the PLTR is a component that is capable of driving the machine. It is only limiting the rotational speed. The speed is lowered by 1500 [rpm]. In a new design the gear steps should be applied at both output shafts of the CVT and have a fixed ratio of 1.25 [$-$] or higher. Then this can be overcome

The demand of combining best of both common used setups for two roller test machines is also achieved. Less energy will be consumed by the test-rig in comparison with the test rig with two motors. And the back to back configuration using one motor as in the $TE73$ is also achieved. Only with the CVT the slip is continuous variable.

Maybe it is useful to look at a smaller, cheaper CVT-drive for the application in this machine. It is quite over dimensioned for this purpose. The costs can be reduced and demands for torque and speed can be maintained. If the original drive of the PLTR is used, the construction should be redesigned to make it cheaper, because now a lot of extras are present which will not be used in this kind of setup.

For the exact machine layout and dimensions the manufacturer should be contacted. The construction drawings are not freely available, which is comprehensible.

The concept design is such that it can easily be converted to different shaft sizes and distances. So there will not appear new difficulties for a change in design for other test machines.

Further research has to be done on this topic. Things that should be done are:

- How well is the test rig controllable
- Work out the CVT drive into detail
- Work out an analysis of what this drive will cost
- Work out a patent-study on applying a variator (in a two roller machine)
- Make a product presentation for potential customers, for example Plint
- Make contact with potential customers and check their interests in this design
- If they are interested, discuss the possible adjustments for this particular costumer
- The lifetime of the drive should be designed equal to the test discs side
- Look if its commercially attainable (can we make profit out of it!!)

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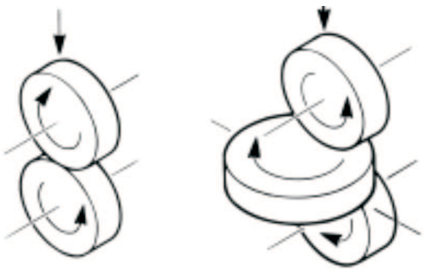
Appendix A

Technical specifications of two roller test machines

A.1 Leaflet of the TE 73 of Plint

TE 73 HIGH SPEED TWO ROLLER MACHINE



Keywords:

- aerospace materials
- automatic transmission fluids
- contact resistance measurement
- delamination wear
- dry contact conditions
- elastohydrodynamic lubrication
- fatigue life
- ferrous materials
- gear materials
- gear surface treatments
- gearbox lubricants
- hertzian contact

- hypoid gear lubricants
- oxidative wear
- rolling
- rolling contact fatigue
- rolling mill
- scuffing
- slide/roll
- spin
- spur gears
- traction coefficient
- variable ratio transmission

Background:

The High Speed Two Roller Machine is a research machine for the study of highly loaded lubricated power transmitting contacts and rolling contact fatigue of materials.The successful operation of machine elements such as gears, rolling bearings, cam/follower systems and traction drives is vital to industry. The common feature to all these is that fact that power or motion is transmitted through highly loaded lubricated contacts with rolling or combined rolling and sliding motion.The high loads (or contact pressures) result in elastic deformation of the surfaces in contact and the components operate under elastohydrodynamic lubrication. The performance of the gear, cam or bearing is dependent on both the material and the lubricant withstanding the highly stressed conditions in the contact. In the case of gears, the motion of rolling and sliding at a given instant in the meshing cycle can be reproduced by two circular rollers of radii equal to the pitch circle rotating with equal and opposite angular velocity about fixed centres. This is the basis of the two roller machine first developed by Merritt in 1935 to simulate conditions in the gear contact. Since the radii of curvature of the teeth at the contact point are the same as the rollers the contact stresses under a given contact load are also simulated by the roller machine.

For gear contacts, the obvious departure from complete similarity arises from replacing the cyclic behaviour of tooth meshing by a steady motion which reproduces the conditions at only one instant in the meshing cycle. However this does at least mean that one condition can be studied at a time and only transient effects are ignored. A similar picture exists in the simulation of the cam/follower contact. In a cam cycle there is a range of sliding and rolling velocities of the contact point. In traction drives even more attention is focused on the lubricant since the power transmitted by the drive is limited by the coefficient of traction of the fluid. Petroleum based oils possess neither the lubricating power at high pressures nor the chemical stability for the high temperature/stress conditions in the contact. Therefore new lubricants with piezo-viscous properties and good molecular stability are being developed.

The traditional method of studying the performance of such fluids under controlled conditions is by using a two roller machine. In the TE 73 the two rollers are driven at different speeds through a back-to-back gear arrangement to give varying amounts of slip in the contact and therefore to transmit increasing amounts of traction. The maximum traction coefficient is obtained at a particular value of slip and this is a characteristic of an individual fluid.

In modern variable ratio gear systems there is an additional complication. It is an inherent property of this kind of mechanism that there is a velocity gradient across the width of the contact zone in a direction perpendicular to the rolling direction. In other words there is spin in the contact zone.

The effect of spin is to modify the traction-slip characteristic. The distinction between the high and low slip regions is lost and instead there is a more gradual growth in traction as slip increases.

This phenomenon can be studied in the TE 73 by introducing a third disc between the two rollers that is free to rotate about a vertical axis. This is the TE 73/S Contact Spin Adapter.

Description:

The TE 73 is a closed-loop device. The two roller specimens are mounted on the ends of parallel test shafts which are connected to a helical gear pair at the rear of the machine. The lower shaft is driven by a thyristor controlled variable speed dc motor through a belt drive. The motor supplies only the losses in the loop, not the full tractive power which is locked into the loop.

Slip percentages from 0% (pure rolling) to 8% are selected by installing an appropriate gear pair. The test rollers have a crowned profile giving a well defined circular contact zone and high Hertzian contact pressures.

The lower shaft housing is fixed while the upper housing is hinged to allow motion in the transverse and vertical directions. A static compressive load is applied manually via a lever to the roller contact and this value is measured by an in-line strain gauge load cell. A separate load cell mounted in the transverse direction measures the traction forces in the contact. Both shaft speeds are measured by inductive pick-ups.

The upper shaft and casing are insulated electrically from the rest of the machine. This allows a small potential to be applied across the roller contact. Variations in this voltage give a clear indication of the frequency of inter-metallic contact between the rollers. These variations in voltage can be displayed on an oscilloscope and the rms potential may be fed to a chart recorder.

The test rollers run inside an aluminium housing with a glass front cover. At low speeds, dip lubrication of the lower roller is acceptable. At higher speeds, pumped lubrication by jet into the exit side of the contact is provided. The roller temperatures may be increased above the steady-state test temperature with infra-red heating through the front window. Bulk temperature is measured with a sliding contact thermocouple.

Instrumentation:

motor speed controller and mains isolator
digital tachometers for upper and lower rollers
digital read-out of applied load and traction force
temperature controller with indicator
Lunn-Furey Electrical Contact Resistance Circuit

TE 73/S Contact Spin Adapter:

One major application of rolling contact power transmission is in continuously variable transmissions. In such devices, of whatever configuration, the geometry of contact is such that there is a degree of rotation of one surface relative to the other and not pure rolling/sliding. This relative rotation or spin has a profound effect on the performance of the transmission:

1. substantial losses occur, even when no power is being transmitted
2. flash temperature effects become significant
3. the degree of slip for maximum traction is greatly increased
4. the overall efficiency is strongly dependent on the degree of spin

In the TE 73 relative motion of the two rollers is limited to combinations of pure rolling with varying rates of sliding in the rolling direction. Spin is introduced into the contact by running the machine with a third disc, whose axis is vertical, between the two smaller diameter test rollers. Two contacts are formed, one on the upper surface of the third disc and one on the lower surface and the contact shape is elliptical.

The degree of spin in the contacts may be varied over a range encountered in transmissions by changing the radius at which the two rollers contact the third disc. The TE 73/S is interchangeable with the standard two-roller configuration.

Technical Specifications:

Speed Range:	$100\ to\ 7,500\ [rpm]$
Rolling Velocity:	$0.75\ to\ 60\ [m/s]$
Load Range:	$0.25\ to\ 15\ [kN]$
Maximum Hertz Stress:	$2.8\ [GPa]$ (based on steel rollers)
Roller Diameter:	$152\ [mm]$
Crown Radius:	$76\ [mm]$
Slip Rates:	0% supplied with machine
Optional Slip Rates:	$1\%, 2\%, 4\%, 6\%$ and 8%
Motor Power:	$11\ [kW]\ dc$

TE 73/S Contact Spin Adapter:

Roller Diameter:	$120\ [mm]$
Crown Radius:	$20\ [mm]$
Contact Elipticity:	$2.36\ [-]$
Spin Ratio:	$0.8\ to\ 1.33\ (4 steps)$
Maximum Hertz Stress:	$2.5\ [GPa]$

Services:

Electricity:	$480\ [V]$, three phase, $50\ [Hz]$, $12\ [kW]$ with neutral and earth mains water and drain

Installation:

Floor Standing Machine:	$1,450\ [mm]\ x\ 1,100\ [mm]\ x\ 1,400\ [mm]$ high, $540\ [kg]$
Bench Mounting Control Unit:	$520\ [mm]\ x\ 520\ [mm]\ x\ 400\ [mm]$ high, $80\ [kg]$
Packing Specification:	$2.2m^3$, GW $760\ [kg]$, NW $630\ [kg]$

Order As:

- ☐ $TE\ 73$ High Speed Two Roller Machine
- ☐ $TE\ 73/G$ Matched Gear Pairs (specify slip rate): for slip ratio $1,\ 2,\ 4,\ 6\ and\ 8\%$
- ☐ $TE\ 73/S$ Contact Spin Adapter

Consumables:

- ☐ TE 73/1 Three Pen y-t recorder
- ☐ TE 73/2 Five Packs of Paper and Three Pens
- ☐ TE 73/3 Two hardened NSOH $B-01$ Test Rollers

A.2 Technical specifications of all the two roller machines of Plint

In figure A.1 the specifications of all two roller test machines are summarized.

		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 - 7500 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 kW	0 - 209 rpm & 0 - 800 rpm	0 - 3000 - 6000 rpm
Motor Power:		0 - 10 - 10 kW ac	0 - 65 - 65 kW	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 (nominaal)	17.5 Nm	95 Nm@3000rpm, 356Nm@	207 Nm 0- 3000 rpm
						800rpm 1363Nm@209 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:		Continuously variable	Continuously variable	Manual with gears	Continuously variable	Continuous variable	Continuous variable
Slip Rates:		0 to 100%	0 to 100%	0, 1, 2, 4, 6 & 8%	0 to 100%	0 to 100%	0 to 20%
Contact Spin Adapter:		No	No	Yes	No	No	Yes
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
Spin Ratio:		N/A	N/A	0.8 to 1.33 (4 steps)	N/A	N/A	0.8 to 1.33 (4 steps)
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

Figure A.1: Technical specifications of the two roller test machines of Plint

A.3 Layout of the PLTM with flywheel

In figure A.2 the layout of the PLTM is given. The flywheel is connected at the motor pulley with a few rubber v-belts.

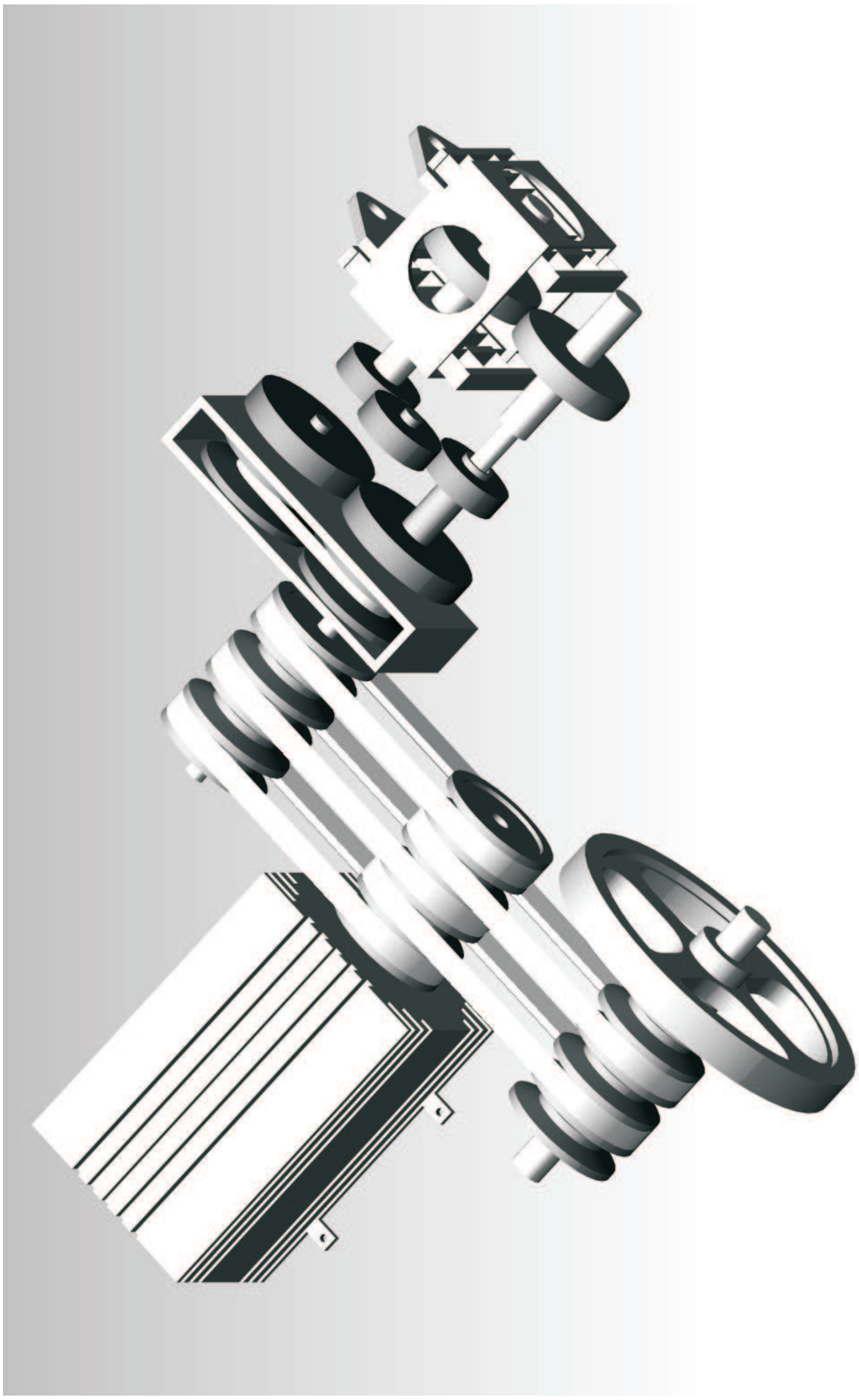


Figure A.2: Layout of PLTM including flywheel and measuring block

Appendix B

Material properties

Material properties stainless steel 85% Fe, 0.2% C, 13% Cr (at 293 K):

Density ρ $[kg/m^3]$	$7,8 \times 10^3$
Young 's modulus of elasticity E [Pa]	210×10^9
Yield strength σ_y [Pa]	500×10^6
Coëfficiënt of linear thermal expansion α $[K^{-1}]$	10×10^{-6}
Specific heat c $[J/(kg \cdot K)]$	460
Coëfficiënt of heat conduction λ $[W/(m \cdot K)]$	27
Melting point [K] $(p = p_0)$	1730
Specific resistance ρ $[\Omega \cdot m]$	720×10^{-9}

Acronyms and nomenclature

Acronyms

Acronym	Description
CVT	Continuously Variable Transmission
PLTR	Power Loop Test Rig
PLTM	Power Loop Test Machine

Nomenclature

Symbol	Description	Unit
i	ratio	[-]
η	efficiency	[-]
P	power	[W]
T	torque	[Nm]
ω	angular velocity	[rad/s]
F	force	[N]
r	radius	[m]
M	moment	[Nm]
f_r	sagging	[m]
d	diameter	[m]
l	length	[m]
E	Youngs modulus of elasticity	[Pa]
I	moment of area	[m^4]
S	safety factor	[-]
σ_y	yield stress	[Pa]
k	ratio between two diameters	[-]