Replacing the secondary cylinder of a Jatco CK2 to improve efficiency

Technische Universiteit Eindhoven, department of Mechanical Engineering

Bachelor final project

Replacing the secondary cylinder of a Jatco CK2 to improve efficiency.

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1. Introduction

The main advantage of a Continuously Variable Transmission over a conventional gearbox is that the engine can be operated near the so called sweet-spot. This results in an increased fuel-economy. However, due to friction and pumping losses in the transmission, the overall efficiency hasn't been significantly improved.
The hydraulic pump providing power to the system is directly connected to the engine shaft. The oil flow is therefore directly related to the engine speed. The oil flow the system demands is much lower than the available oil flow most of the time. The surplus oil is directed back to the reservoir, resulting in power loss. By replacing the oil pump with an electric driven one, the oil can be delivered “on demand”. And the efficiency can be increased.
The combustion engine, road impurities, braking etc. will produce torque peaks acting on the CVT. Therefore, in practice, the clamping force is multiplied by a safety factor of at least 1.3, reducing the risk off excessive slip and resulting damage. This means the clamping force is higher than required most of the time resulting in efficiency loss. Studies have shown that reducing the clamping force results in an improved efficiency. [1,2] If the slip in the system is controlled, the safety factor is no longer needed.
The hydraulic system of a Jatco CK2 CVT has an operating pressure between 8 and 40 bar. When minimum pressure is applied, the clamping force is still too high to allow slip in overdrive. Because of this a smaller pressure cylinder is designed by B. Bonsen. A Jatco CK2 test CVT is been fitted with this new cylinder. Measurements were taken using the modified CVT.

2. Working principle of a Continuously Variable Transmission

2.1 Schematic overview

The key elements of a CVT are a v-shaped drive belt and two pulley sets. On both pulley sets one sheave is fixed and the other can move in the axis direction. The moveable pulley is hydraulically actuated. By moving the pulley, the running radius of the drive belt and therefore the shaft speed can be varied and the continuously variable ratio is realized.

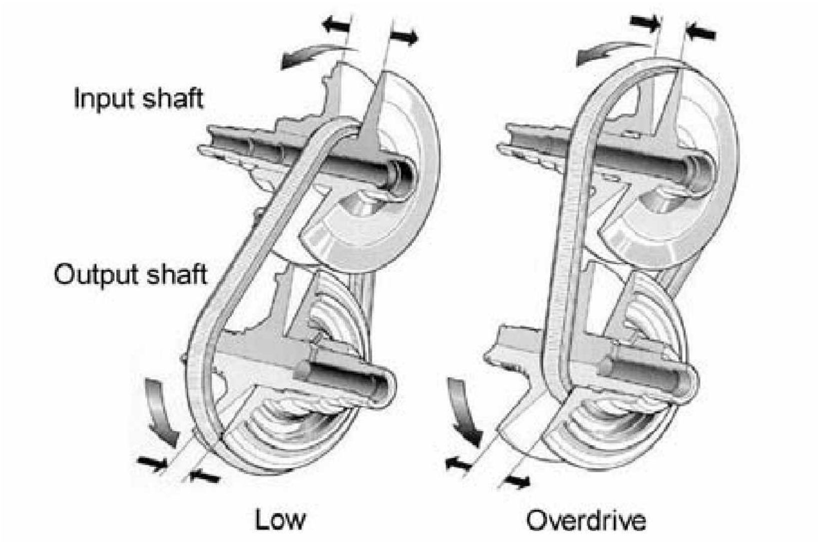


Figure 2.1: Schematic presentation of a CVT transmission [3]

The speed ratio is defined as follows:

$$r_s = \frac{R_p}{R_s} = \frac{w_s}{w_p} \qquad (2.1)$$

Where w_s is the secondary and w_p the primary shaft speed.

2.2 Slip

There's a maximum amount of torque the system can transmit. The maximum torque can be calculated with the following formula:

$$T_{cvt,p,s} = \frac{2F_{\min} R_{p,s} \mu}{\cos\alpha} \qquad (2.2)$$

Where T_{cvt} is the maximum torque on the primary or secondary cylinder, F_{min} the minimum clamping force $\min(F_s,F_p)$, $R_{p,s}$ the running radius of the minimum clamping force, μ is the maximum traction coefficient and α the wedge angle of the pulley.
Because the maximum torque in the systems depends on the traction coefficient, the efficiency will increase when the traction coefficient is increased. The traction coefficient itself depends on the amount of slip in the system. The slip in the system is denoted by:

$$\nu = 1 - \frac{r_s}{r_{s0}} \qquad (2.3)$$

Where r_s is the ratio and the r_{s0} is the no-load ratio. The no-load ratio is the ratio in the transmission when no torque is lead through the transmission and no slip occurs. The no-load ratio is determined by measuring the position of the secondary cylinder using a LVDT.

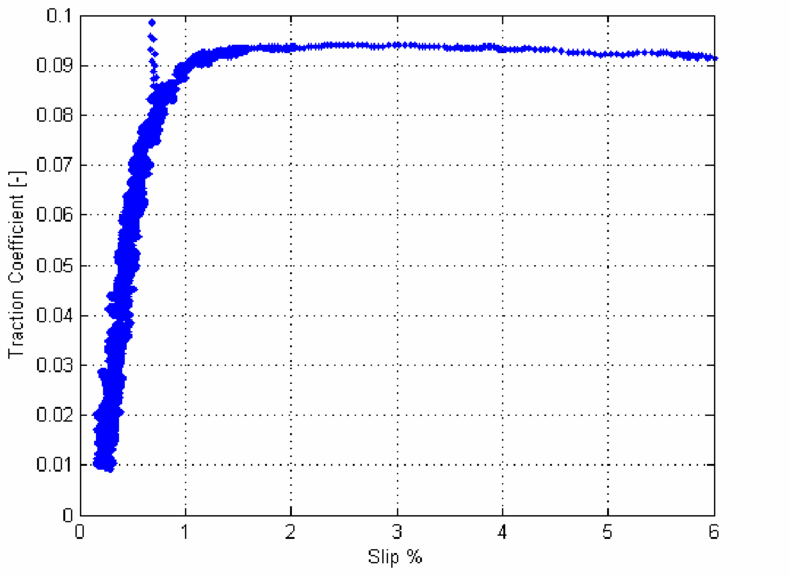


Figure 2.2: Traction coefficient plotted against slip of the drive belt

When the clamping force is reduced, the system is allowed to slip. When the slip increases the traction coefficient increases to a certain maximum occurring at a slip of about 2.5 percent (figure 2.2), after this the traction coefficient will slightly decline. The section where the traction coefficient increases with increasing slip is called the micro slip region. The other section is the macro slip region. Because the traction coefficient decreases with increasing slip, this section is unstable and the slip will increase even if the clamping force and the torque are kept constant**.** The ideal point to operate the CVT is the microslip region, near the maximum traction coefficient Because of torque peaks from the engine and road can cause excessive belt-slip, a safety factor is introduced. In the conventional clamping strategy in a Jatco CVT, the clamping force is multiplied by this safety factor of 1.3. When the transmission is operated in the micro slip region, close to the optimal point, it has to be operated without a safety factor. Therefore it has to be controlled to be stable, even in the unstable region.

2.3 Hydraulics

The hydraulic power to drive the hydraulic actuators and accessories of the transmission is delivered by an internal gear pump. The hydraulic power generated by the oil pump is circuited to the main line. This line is connected to the secondary cylinder, which requires the highest pressure. The main line is also connected to control valves, reducing the pressure and the flow to the desired pressure level for the accessories and the primary cylinder.

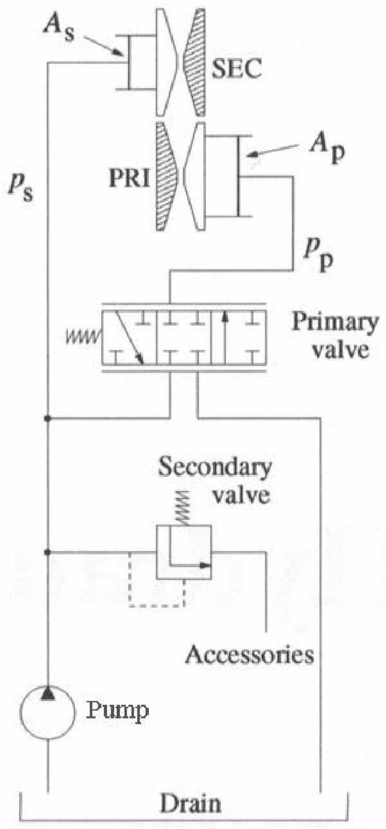


Figure 2.3: Simplified schematic overview of the hydraulic system in a Jatco Ck2 [4]

3. The pumping loss

3.1 Oil pressure and flow in the system

Part of the efficiency loss of a CVT is the pumping loss. The hydraulic internal gear pump is directly connected to the input shaft of the transmission. The rotational speed of the pump varies with the engine speed. Therefore the pump produces more oil flow than needed most of the time. With an electric pump the oil flow can be adapted to the systems needs at the moment, resulting in a much lower loss.

The maximum pressure and steady state oil flow off the system are 40 bar and 12 liters per minute, respectively. The oil flow is used for internal leakage in the system, lubrication and cooling.

The conventional pump has a displacement of 16.7 cm^3. Assume the efficiency of the pump is 70%. If the combustion engine revolves at 2500 RPM, the oil flow of the pump is about 30 liters a minute. The steady state oil flow loss is therefore 18 l/min. This results in a power loss of 240 up to 1200 watt loss, depending on the pressure.

3.2 Searching an electric motor

As mentioned above, an electric pumping system is searched to implement in the TR3 test rig. The power P needed is the product of the oil flow and pressure.

$$P = Q * p \qquad (3.1)$$

To deliver maximum pressure at steady state oil flow the rated power of the electric motor is about 800 Watts. Because the CVT will be build into a car in the future, it is convenient that the motor has a rated voltage of 12 volts. However, there is no conventional electric motor which can deliver such power at this low voltage. Eltromat, a company specializing in drive mechanisms, has been consulted whether such a motor can be produced. The results of the consult are in the appendix. The motor was expensive and was very inefficient due to the high rated current and therefore high losses. Therefore it was decided to search for a 42 volt motor. 42 volt is the new standard in automotive technology, because of lower losses when using power intensive electrical equipment.

R. v/d Boogaert was consulted for the problem. Other projects in which a combination of electric motor and pump were bought, failed repeatedly in the past. Therefore it is decided that there had to be performed tests first with the pump connected to a powerful motor. This motor is already available and works at a rated voltage of 220 and can deliver more power than needed. In that way, the power needed to drive the pump could be determined precisely and the exactly right motor and controller can be bought.

4. The new cylinder

A new cylinder for the Jatco CK2 is designed. The surface area of the new piston is about half the surface area of the conventional piston, reducing the clamping force by a factor two. The new cylinder has triangular grooves on the outer wall. These will be used to determine the position of the pulley using an inductive sensor. Because the sensor was not available, this feature is not yet implemented. The design of the new pressure cylinder is included in Appendix I

For the test of the new cylinder the CVT build in the TR3 Test rig, a Jatco CK2, is used. The transmission is taken apart so the secondary shaft could be replaced. Meanwhile a new shaft of the same type is dissembled, to place the new pressure cylinder. The shaft included a spring to ensure a minimum clamping force even when there was no hydraulic pressure. This spring is left out, to reduce the clamping force to a minimum. After completing the new secondary shaft, the gearbox is assembled including the new modified shaft. The Jatco ck2 was build back into the TR3 test rig.

Figure 4.1: The new cylinder mounted on the secondary shaft

5. Measurements

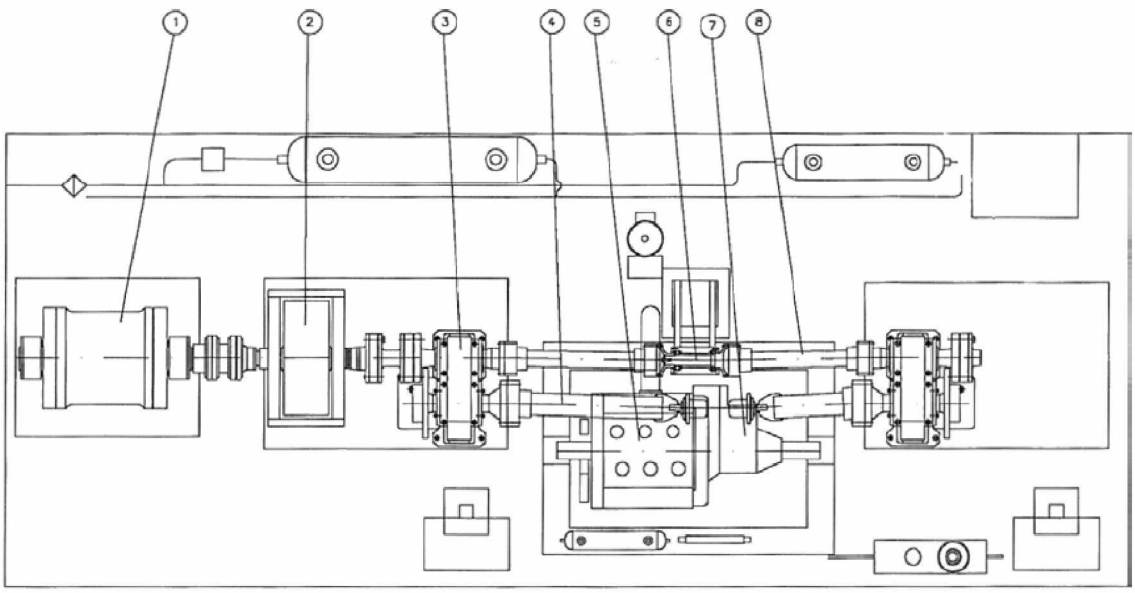


Figure (5.1): A schematic overview of the TR3 test-rig

For the measurements the TR3 test-rig was used. This test-rig can simulate realistic driving conditions using a combustion engine (5), eddy current brake (1) and flywheel (2). The Jatco CK2 (7) is connected to the combustion engine and with drive shafts to the flywheel and eddy current brake.
The measurements were made using D-space. Telemetry systems are used to determine the torque in the shafts. Hall sensors are used as speed sensors. With these, the power of the combustion engine, the power on the drive shaft, and therefore the efficiency, can be determined. Slip is calculated from the no-load ratio, which is determined using a LVDT. When the slip of the drive belt exceeds 10 percent, d-space automatically activates a safety condition, which overrides the system, turns off the load torque and cuts the accelerator, returning the engine to its stationary RPM.

Torque measurement shaft
The input torque measurements are done with strain gages, fixed on the shaft. The signal from these strain gages is transmitted wirelessly to a receiver, which is linked to D-space. The calibration data of this torque measurement shaft can be found in Appendix II.
The result of the calibration, after null-correction, is shown in figure 5.2. The standard deviation of the linear fit is +/- 0.0025 Nm. This is smaller than the resolution of the measurement shaft, which is +/- 0.62 Nm.

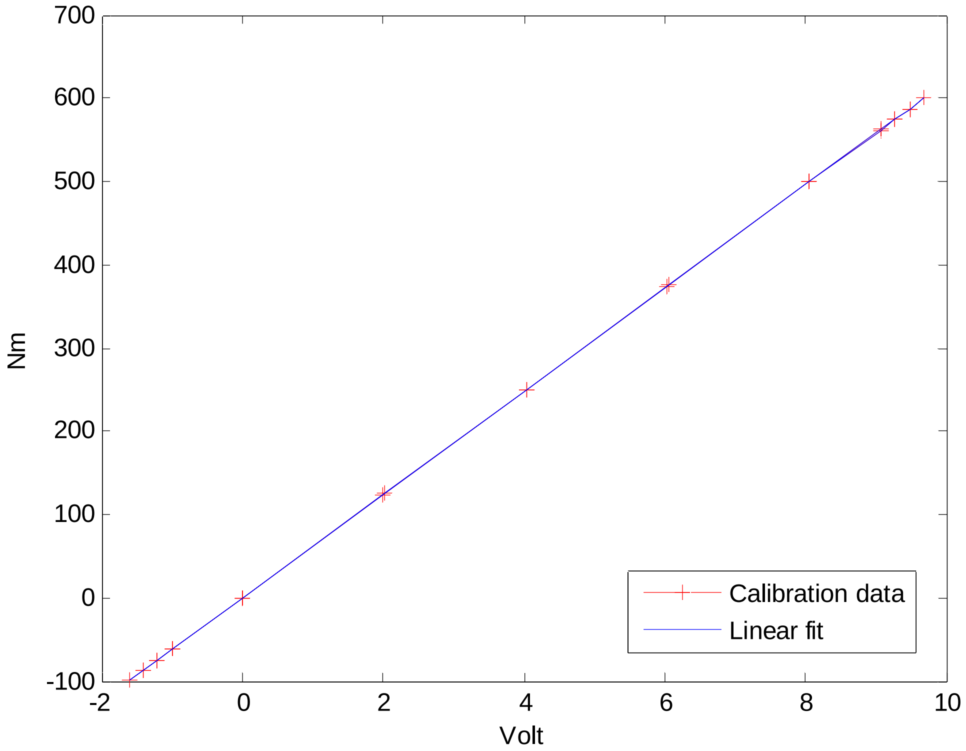


Figure (5.2): Calibration data

6. Testing the new cylinder

During measurements there are several key parameters which can be varied.

- The line pressure
- The engine speed
- The ratio
- The load torque on the drive shaft

The tests are performed with fixed pressure, engine speed and ratio. The load torque is then slowly increased until macro slip occurs, safety conditions are activated and the test is stopped. Several test are executed to determine the influence of the individual parameters.

6.1 Different line pressures

The line pressure determines the clamping force. Reducing the clamping force therefore reduces the friction loss and increases the efficiency. Measurements are done with a fixed engine speed of 1500 RPM and a ratio of 2.25. A ratio of 2.25 is chosen because a report of Pulles [3] claims a smaller clamping force is needed to increase efficiency in overdrive. The new cylinder should allow this smaller clamping force.

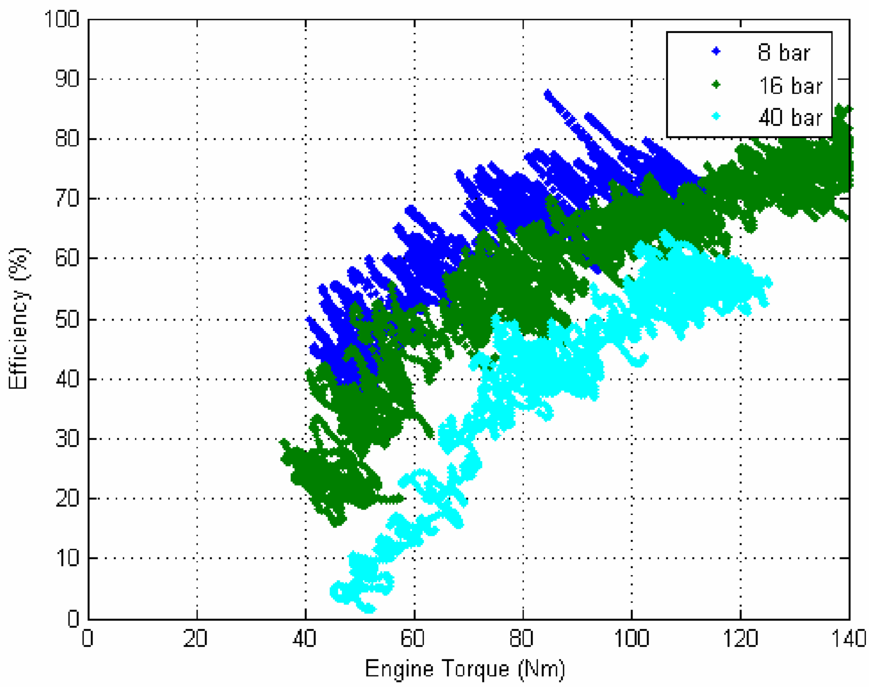


Figure 6.1: Efficiency at different pressures

As the figure above shows, the efficiency increases when the pressure on the secondary cylinder is reduced. A pressure of 16 bar corresponds to a pressure of 8 bar in the old configuration of the transmission. A major increase in efficiency can be observed

Due to a higher clamping force, a higher maximum torque is possible with higher pressure. This can be seen in the results of 8 and 16 bar pressure. The maximum torque reached with 40 bar is low because the test was halted too early. The engine stalled, furiously vibrating, with the measurement at 16 bar, because the clamping force was too high to allow macro slip and trigger the automatic safety conditions. To prevent damage to the test rig and the engine itself, the test was manually halted when the engine speed dropped below 1300 RPM. Afterwards it became clear the engine wasn’t loaded to its maximum, the throttle wasn’t completely opened yet and it would have been safe to continue the test. When this became clear there was no time left to perform the test again.

6.2 Different engine speed

A higher engine speed should result in a lower efficiency due to pump losses. To determine the influence of the engine speed, a test is done at a ratio of 1 and a pressure of 8 bar. The engine speed is varied to 2865 (300 rad/s) and 1500 RPM

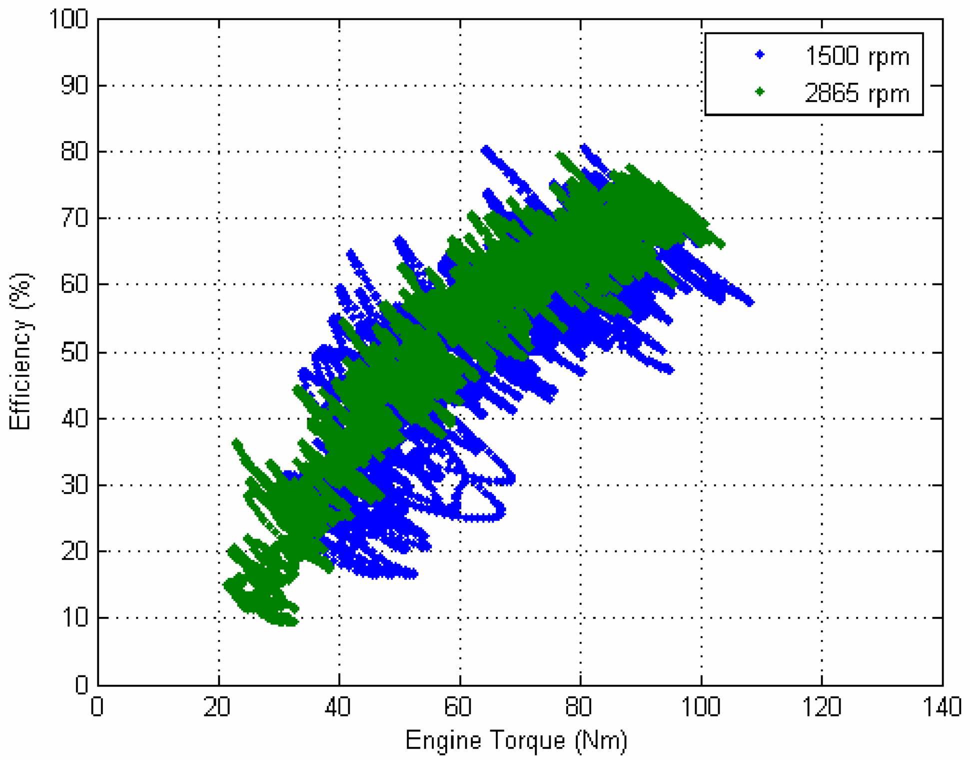


Figure 6.2: Efficiency at different engine speeds

The engine speed does not have a large effect on the efficiency. There's hardly any difference between the two measurements. This could point out the pumping loss is not an important efficiency factor.
A large deviation can be seen in the measurement at 1500 RPM. This is because of the speed controller used. At low ratio's and engine speed the controller could not be tuned precise enough. The engine speed still "vibrated" around the set engine speed. This causes the larger deviation in the measurement at low engine speed.

6.3 Different ratio

Selecting a different ratio on the CVT could also result in an efficiency gain. Tests at different ratios were performed. For the first comparison, the pressure was set at 8 bar and the engine speed at 2865 RPM. The second test was done at the same pressure, but the engine speed was lowered to 1500 RPM.

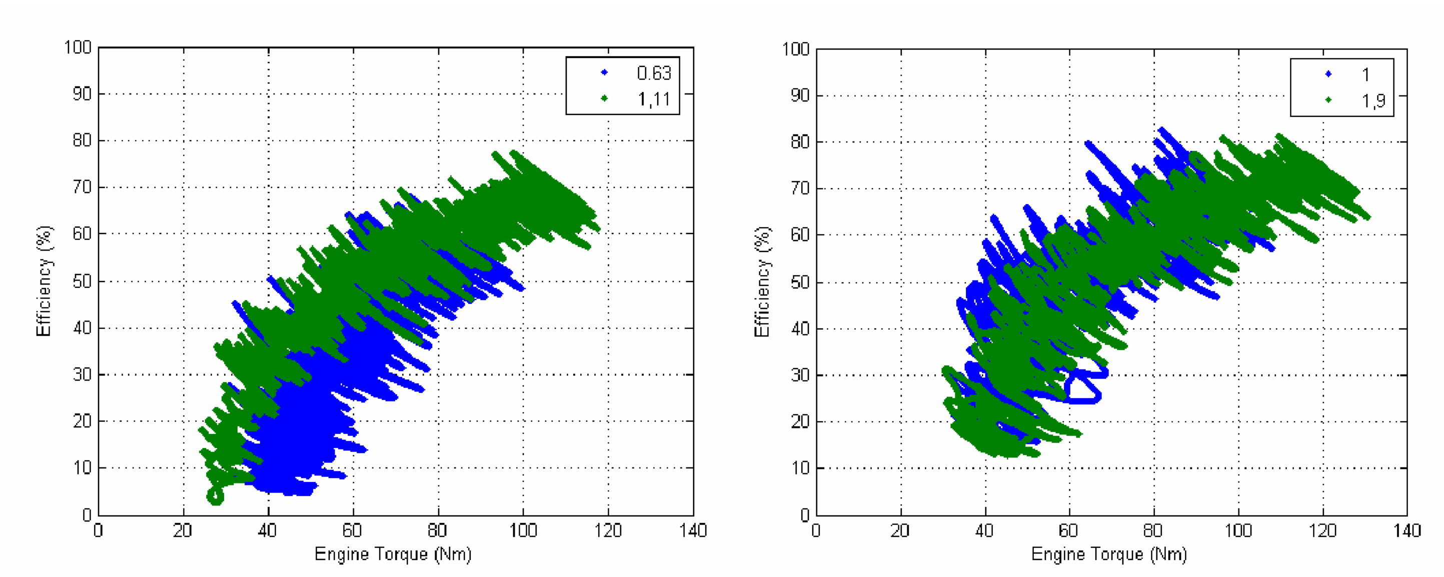


Figure 6.3: Efficiency at different ratios

The increase in ratio from 0.63 to 1.11 results in an increase in efficiency. This is mainly due to the links of the drive belt. When the ratio approximates 1, the shaft speeds and the running radius of the pulleys are equal. This situation requires minimum movement of the links of the drive belt and therefore less friction loss.
There is no obvious difference between a ratio of 1 and 1.9. A high ratio results in an increased torque on the primary shaft. The ratio between the friction loss and the torque on the primary shaft is higher. This results in an increased efficiency, although it does not exceed the efficiency gain of the link movement at a ratio of 1.

6.4 Comparison with the original cylinder

To see if the implementation of the new cylinder is an improvement, a comparative test is done. It is attempted to recreate the same test conditions. The engine speed is set at 2865 RPM (300 rad/s), the ratio at 1.11 and the pressure with the new cylinder is 8 bar.

However the test performed by Pulles [3] were done at two different clamping strategies. TCM is the original clamping strategy of the Jatco CK2. Slip control is a strategy were the traction coefficient is kept at its maximum. At low engine torque the controlled clamping force, and therefore the pressure, will be kept minimal. This is why the two tests can be compared to each other in the low engine torque range.

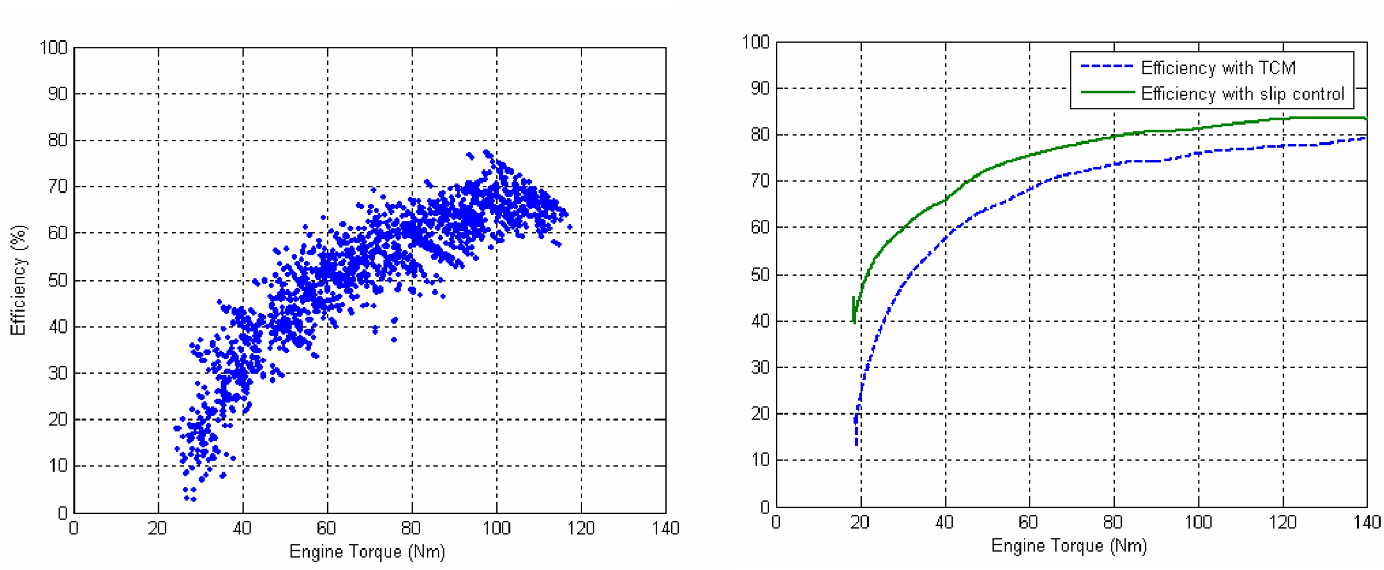


Figure 6.4: Efficiency comparison with the original cylinder

At first a large deviation in the new test results arrests the attention. This is mainly because the conditioners used for the engine and driveshaft speed were absent, and therefore there was more noise in the measurements.
The efficiency at 60 Nm is about 50 percent for the new cylinder and 68 to 75 percent for the conventional one. Unexpectedly, the new cylinder lowers the efficiency instead of increasing it. This can have a variety of reasons.

- The spring in the secondary shaft is left out. The new cylinder is smaller than the original one, providing less support to the sheave. If the stiffness of the sheave depends on the support of the pressure cylinder, both factors can be the cause of a reduced stiffness of the system and therefore power loss.
- A faulty calibration of the sensors or hysteresis in the torque measurement shaft can lead to a deviation of a couple of Newton meters. This can have a substantial effect on the calculation of the efficiency.
- Although the transmission was carefully assembled, errors in the assembly, such as a small misalignment in the drive belt, may result in higher friction loss.

7. Conclusion and recommendations

Conclusion

The transmission works fine with the new pressure cylinder, although the results are not as expected. The lower efficiency can be the result of several factors.

- The spring in the secondary shaft is left out. The new cylinder is smaller than the original one, providing less support to the sheave. If the stiffness of the sheave depends on the support of the pressure cylinder, both factors can be the cause of a reduced stiffness of the system and therefore power loss.
- A faulty calibration of the sensors or hysteresis in the torque measurement shaft can lead to a deviation of a couple of Newton meters. This can have a substantial effect on the calculation of the efficiency.
- Although the transmission was carefully assembled, errors in the assembly, such as a small misalignment in the drive belt, may result in higher friction loss.

The greater noise and large efficiency deviation lead to badly comparable data. At the moment, there's therefore no reason to deviate from the original hypotheses, the smaller cylinder may still increase the efficiency.

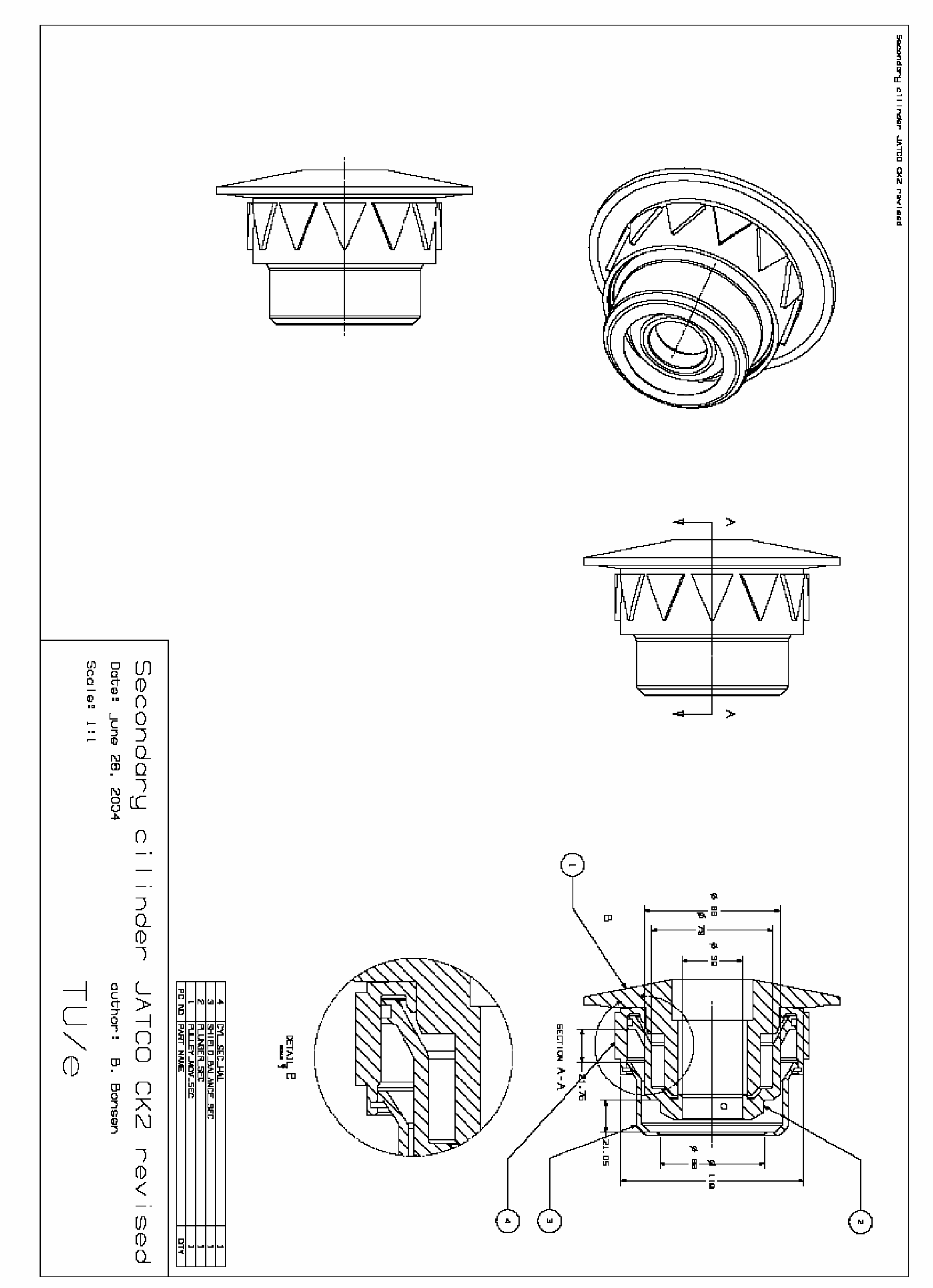
Recommendations
Due to badly comparable results, new test should be done to pinpoint the reason of the unexpected lower efficiency.
The conditioners of the engine and driveshaft speed should again be installed to improve quality of these signals.

To determine which electric motor should be bought to drive the hydraulic circuit, first a pump has to be bought. With this pump measurements can be done using a powerful 220 volt motor already available. With these measurements it can be determined what mechanical power is needed to drive the hydraulic system.

8. References

[1] J. D. Micklem, D. K. Longmore and C. R. Burrows, “The magnitude of the losses in the steel pushing V-belt continuously variable transmission,” Proc Instn Mech Engrs Vol. 210, Part D: Journal of Automobile Engineering, ImechE, 1996
[2] B. Bonsen, T. W. G. L. Klaassen, K. G. O. van de Meerakker, M. Steinbuch and P. A. Veenhuizen, “Analysis of slip in a continuously variable transmission,” Proceedings of IMECE’03, ASME International Mechanical Engineering Congress, Washington, D.C., November 15-21, 2003
[3] R.J.Pulles, “Slip controller design and implementation in a Continuously Variable Transmission” Report number: DCT-2004/102, TU/e, Eindhoven, 2004
[4] B. G. Vroemen, “Component control for the zero inertia powertrain,” TU/e, Eindhoven, 2001

Appendix I



Secondary cilinder JATCO CK2 revised
Date: june 28, 2004
Scale: 1:1
author: B. Bonsen
TU/e
SECTION A-A
DETAIL B

Appendix II

Calibratiesheet Koppelmeetas TR-3

Gecalibreerd op 3 mei 2005 door R. Pulles en W.J. Loor

Zender
Ceasar Microdac B1A
Rgain = 3.3 kE
Rctrl = n.c.

Ontvanger

Type:	B1A-CU24-SD
Frequency:	24 MHz
Part No:	2002-4776
Serial No.:	0317047

Instellingen

Output Mode:	10V
Output Filter:	100 Hz
Offset:	een maal geregeld naar nul, bij begin van de meting
Gain:	Fixed

Gemeten op ‘Analog out’ met Fluke 79 serienummer 68310530

Calibratiebank
Hydraulieklab
Gebruikte schijven: 4 x 125, 1 x 62.5, 3 x 12.5

De as is gecalibreerd voor +600 Nm en – 100 Nm

+ zijn de schijven links aangebracht, - rechts, voor de calibratie bank gezien.

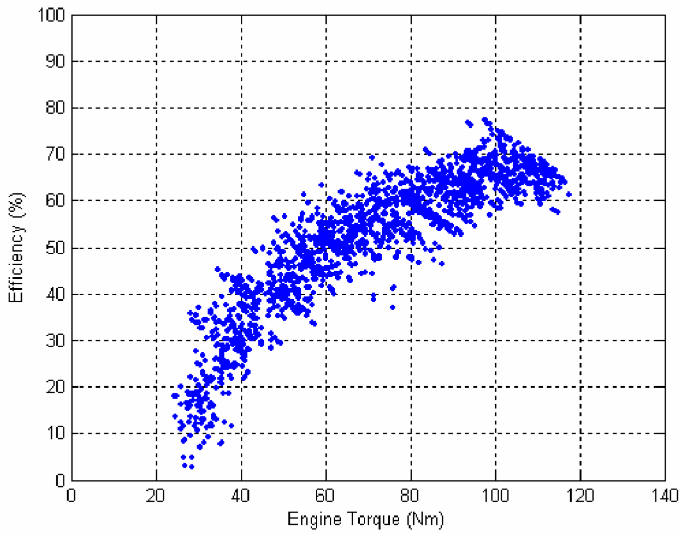
Metingen

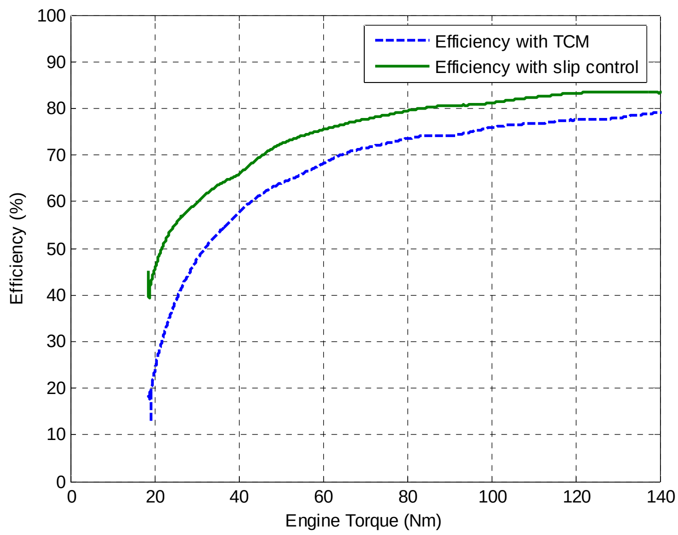
Moment aangebracht	Analog out	Null correctie	Berekende momenten (*61,9835)
0,00	0,00		0,00
125,00	2,00		123,97
250,00	4,03		249,79
375,00	6,02		373,14
500,00	8,06		499,59
562,50	9,06		561,57
575,70	9,28		575,21
587,50	9,48		587,60
600,00	9,68		600,00
587,50	9,48		587,60
575,70	9,28		575,21
562,50	9,07		562,19
500,00	8,06		499,59
375,00	6,06		375,62
250,00	4,04		250,41
125,00	2,02		125,21
0,00	0,00		0,00
0,00	-0,25	0,00	0,00
-62,50	-1,25	-1,00	-61,98
-75,70	-1,45	-1,20	-74,38
-87,50	-1,65	-1,40	-86,78
-100,00	-1,85	-1,60	-99,17
-87,50	-1,65	-1,40	-86,78
-75,70	-1,45	-1,20	-74,38
-62,50	-1,25	-1,00	-61,98
0,00	-0,25	0,00	0,00

Appendix III

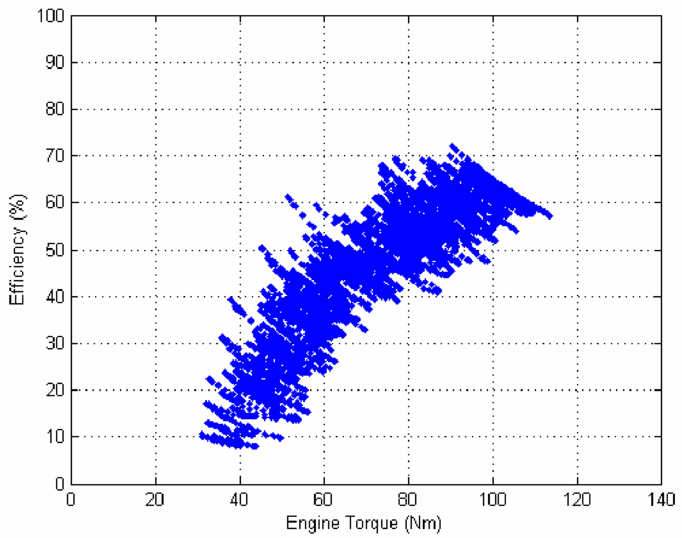
Tests were performed at different ratios to compare the results with the tests of Rob Pulles. As explained in the test results chapter, the efficiency with the new cylinder is lower than the efficiency of the old cylinder. The next figures show the efficiency is lower at all ratios. All tests are performed at 300 rad/s and a pressure of 8 bar. On the left side the results of the new cylinder is depicted, on the right side the results of the old cylinder.

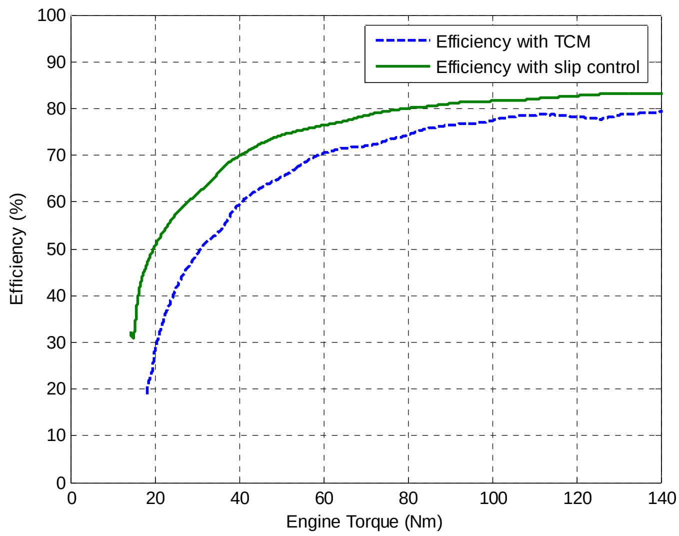
Ratio: 1.11



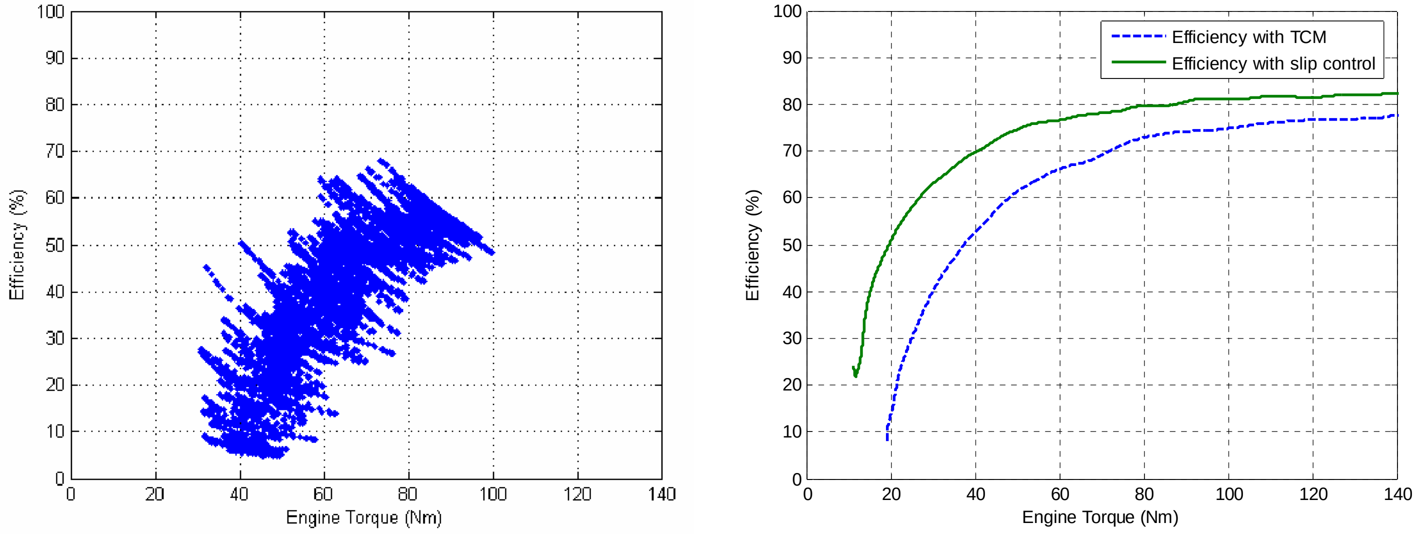


Ratio: 0.86

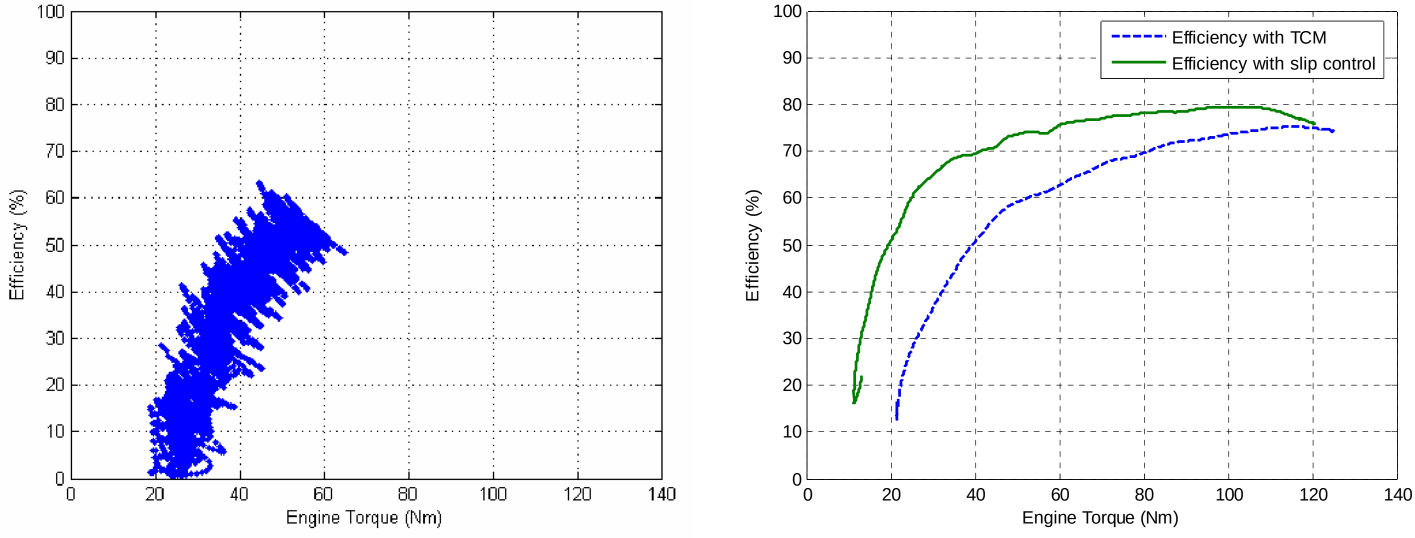




Ratio: 0.64



Ratio: 0.43



Nomenclature

Symbol	**Description**	**Value [unit]**
α	Wedge angle of the pulley	11[°]
μ	Traction coefficient	[-]
ν	Slip of the drive belt	[%]
p	Hydraulic pressure	[Nm/m^2]
r_s	Speed ratio	[-]
r_{s0}	No load ratio	[-]
$w_{p,\ s}$	Shaft speed (primary or secondary)	[rad/s]
F_s	Secondary clamping force	[Nm]
P	Hydraulic power	[W]
Q	Hydraulic flow	[m^3/s]
$R_{p,s}$	Running radius of the driver belt (prim. or sec.)	[m]
$T_{cvt,p,s}$	Maximum torque on the drive shaft (prim. or sec.)	[Nm]

9. Acknowledgement

I would like to thank my coach Bram Bonsen for his support during the project. Also I would like to thank Wietse de Loor, Toon van Gils, Erwin Meinders and Ruud v/d Boogaert for helping me with the TR3 test rig and implementing the new cylinder.