

**AN INVESTIGATION INTO THE LOSS
MECHANISMS ASSOCIATED WITH AN
AUTOMOTIVE METAL
V-BELT CVT**

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Abstract

To date the reduced fuel consumption and emissions predicted for CVT vehicles have not been realised by production cars. A contributing factor is that CVT systems have a lower efficiency than their fixed ratio counterparts.

This paper introduces work being undertaken to investigate the low load loss mechanisms in a pushing metal V-belt CVT. A test rig has been developed to accurately measure steady state losses through a CVT at low and no load levels. This paper gives an overview and discussion of the existing work undertaken on belt drive CVT efficiency, and gives a description of some of the initial modelling work. The experimental work also aims to produce an understanding of the effects of operating temperature on the transmission loss mechanisms, in particular low temperature operation often seen during a cold start have been investigated.

This paper will present details of the test rig, preliminary findings from the experimental work and highlight areas in which improvements to CVT models can be made. It is planned to use the models as the basis for CVT drive cycle simulation work.

Notation

A	Clutch Plates Area (m^2)
b	Tooth Width (m)
C_f	Coulomb Friction Coefficient
C_v	Viscous Friction Coefficient
C_m	Drag Torque Coefficient
d_m	Bearing Mean Diameter (mm)
D	Pump Displacement (m^3/rad)
f_0	Bearing Torque Loss Factor
l	Tooth Depth (m)
M_0	Speed dependant Bearing Torque (Nmm)
n	Bearing Speed (rev/min)
n_i	Number of Clutch Plates
r_m	Mean Clutch Plate Radius (m)
R_o	Pitch Circle Radius (m)
P	Pump Pressure (N/m^2)
S	Clutch Plate Separation (m)
T_{pump}	Pump Torque Demand (Nm)
T_{clutch}	Clutch Drag Torque (Nm)
T_{churn}	Gear Churning Losses (Nm)
η	Viscosity of Transmission Fluid
η_m	Pump Mechanical Efficiency
η_v	Pump Volumetric Efficiency
ρ	Density of ATF (kg/m^3)
ν	Kinematic Viscosity (mm^2/s)
$\Delta\omega$	Relative Clutch Plate Speed (rad/s)
ω	Rotational Speed (rad/s)

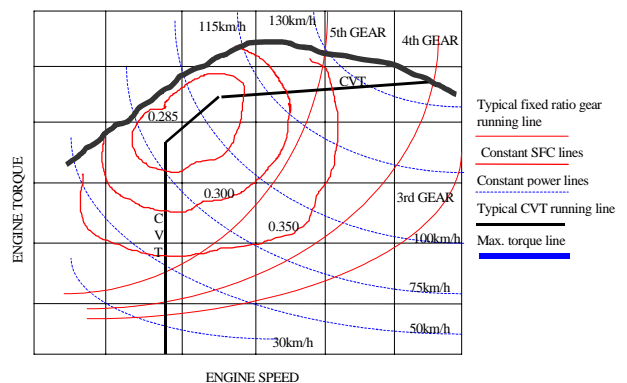
1 Introduction

The efficiency of a vehicle transmission system is an important factor in the overall efficiency of any vehicle. With the increased environmental constraints which todays vehicles must reach, in terms of emissions and fuel consumption, it is important to understand where the inefficiencies lie within a transmissions design. Continuously

variable transmissions (CVTs) have been on the market for a number of years now, and should in principle offer increased fuel efficiency over similarly sized fixed ratio transmissions.

By effectively having an infinite number of gear ratios the CVT should allow better matching of the engine operating conditions to the variable driving conditions that may be experienced. Figure 1 shows how a typical CVT system might be controlled so that the engine can be constrained to operate as near as possible to its maximum efficiency point.

Figure 1 Typical Operating Map for a CVT Vehicle



However, the reduced fuel consumption and emissions predicted for CVTs have not been realised by production cars. When fuel consumption figures have been compared to equivalent fixed ratio vehicles they have been at best equal and in most cases considerably lower.

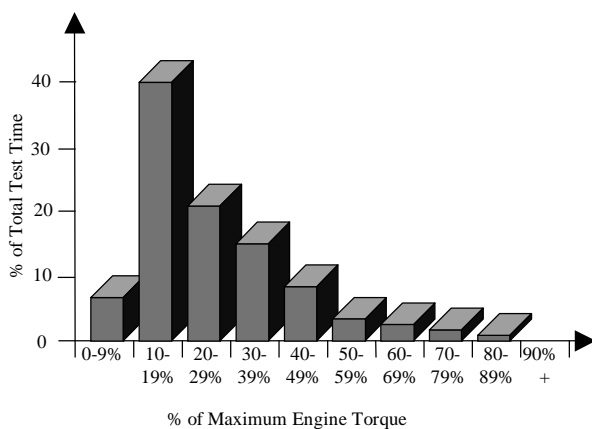
Since the control strategy for reduced fuel consumption is well founded, it can be concluded that existing CVT systems have a lower efficiency than their fixed ratio counterparts.

This inefficiency has been linked to a number of possible inherent parasitic losses

associated with pushing V-belt CVTs, namely torque losses within the belt mechanism itself, belt slip and hydraulic control system pumping losses. These losses exist alongside the normal losses associated with fixed ratio transmissions, namely gear meshing losses, bearing losses, oil churning and windage.

Work by Simner [1], which categorises transmission losses, gives a description and understanding of the importance of parasitic or no load losses. Figure 2 shows the proportion of time that a vehicle spends over a range of different engine torque levels while undertaking an FTP75 drive cycle. It can be seen that over 40% of the cycle time is spent at torque level of 10-19%, and some 90% of the drive cycle time is spent below 50% of maximum engine torque. At these loads the proportion of parasitic losses to power proportional losses becomes significant. This would therefore indicate that reductions made in parasitic losses could significantly reduce fuel consumption and emissions over a typical drive cycle test.

Figure 2 Percentage Time vs. Load Level during an FTP75 Drive Cycle



A major factor in the magnitude of parasitic losses is the effect of fluid viscosity. Since viscosity is temperature dependant, the effect of temperature on parasitic losses is going to be significant. This significance will increase as proposed legislation may include sub zero start condition for drive cycle legislative tests, where parasitic losses may increase considerably.

2 The Transmission

The transmission being investigated is currently used in Rover 200 automatic cars. It is based on the Van Doorne push V-belt variator system. A more detailed description

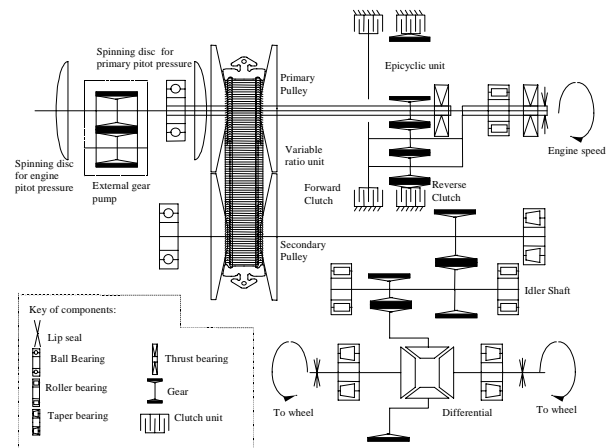
of the function of a metal V-belt CVT is given by Hendriks et al. [2].

3 Proposed Loss Mechanisms

This section describes the preliminary modelling undertaken on the transmission. These models will be validated using experimental data obtained from a test rig facility. Where comparison of model and experimental data indicates inaccuracies in the existing modelling, improved models will be developed.

Figure 3 below shows a schematic diagram of the possible loss components in the transmission.

Figure 3 Schematic View of Transmission Components



3.1 Clutch Losses

For this work it is assumed that the forward clutch is locked and that no slip occurs, and thus that no losses occur in the forward clutch. However, a clutch torque loss is associated with the drag on the disengaged reverse clutch. This is due to the shearing of the oil film between each adjacent clutch plate and the oil film separating the plates. A number of models exist such as the one proposed by Vahabzadeh [3], which is based on the assumption of Couette flow between the disengaged clutch plates. The torque loss is given by the expression.

$$T_{clutch} = v \Delta \omega n_i A r_m^2 \frac{1}{s}$$

A more complex analysis of clutch drag torque is given by Park et al [4], which includes a radial pressure distribution between the disengaged plates taking into effect the centrifugal force component acting on the fluid. This results in a clutch torque

loss characteristic, which decreases at higher speeds as the fluid begins to break away from the clutch plates. At present the simulation uses the Couette flow model. However, future experimental work may lead to the more detailed model being adopted.

3.2 Pump Losses

One of the major losses within the transmission is the torque absorbed in driving the hydraulic pump. The pump supplies the ratio control pressures and lubrication flow for the transmission. In many operating situations the pump torque is the largest loss mechanism in the transmission. Initial modelling uses a simple mechanical and volumetric efficiency.

$$T_{pump} = \eta_m \eta_v DP$$

3.3 Belt Losses

The belt is possibly the least understood of all the components in the transmission and is fundamental to the modelling of the overall transmission losses.

A number of papers have been written in recent years related to the modelling of the metal pushing V-belt transmission, notably by Micklem et al [5,6&7], Fujii et al [8,9, &10], and Karam and Play [11].

Micklem produced an empirical model for the torque losses in the belt drive, and this was the model adopted as a starting point in this work.

Micklem proposed three specific torque loss mechanisms within the belt. The first and the largest component is a wedging force as the segments are forced into and pulled out of the pulley contact arc. On exit from the primary pulley the belt will be retained by the pulley to a radius smaller than the contact radius, and on entry into the secondary pulley the belt will be forced out to a radius greater than the contact radius. Experimental measurements by Kim and Lee [12] appear to support these assumptions. Secondly Micklem proposed a viscous shear film between any belt component having a relative motion to a neighbouring component. Hence, he proposed losses between the segments and the band packs and between individual bands assembled in the band packs.

Further to this Micklem also proposed a viscous shear film, based on elastohydrodynamic (EHD) theory, as the means of torque transfer between the belt

and pulleys, this too has an effect on the total belt torque loss, and belt slip.

More recently Fujii et al have written a number of papers describing experimental work in which a belt has been instrumented with strain gauges to measure segment compressive forces and ring tension variation throughout the belt pulley system. However, none of the work concentrates on the aspect of torque loss in the system. This work indicates that some improvement can be made to the Micklem modelling. A number of assumptions used in the work have now been shown to be incorrect, namely uniform band tension, and a band speed uniformly faster than the segment speed.

Both Fujii et al and Karam and Play describe the existence of an idle arc within both pulley wrap angles, in which the compressive load upon the segments does not change. The magnitude of this idle arc will have an influence on the force loading seen by individual segments as they pass through the loading and unloading arcs.

An aspect that has not been covered in previously published work is the effect of temperature on the magnitude of the belt torque loss. At present there appears to be no work undertaken on this subject, although the Micklem viscous shear models do have a viscosity term within them.

3.4 Belt Slip

A few papers have been produced discussing the measurement and possible mechanisms of belt slip in metal V-belt drives. Micklem's proposed model based on an EHD Lubrication regime, as the means of force transfer between the pulley and segment sides is effectively a model of belt slip. While Kim et Lee [12], have measured changes in radial belt positions around the pulley wrap angles, and produced ranges for coefficients of friction to fit slip data. More recently Kobayashi et al [13] have undertaken work measuring and modelling micro slip within the belt system. Accurate measurement of any belt slip is one of the key features of the test rig developed for this work.

3.5 Meshing Losses

The final drive of the CVT is similar to that found in any conventional transmission and can therefore be modelled in a similar way.

The losses due to gears can be treated, as they would be in a conventional transmission,

using conventional gear meshing loss theory as described by Buckingham [14]. The gear losses can be broken down into three distinct areas, namely meshing losses, churning losses and windage losses. It can be assumed that the windage losses are negligible and can thus be ignored. Churning losses in the final drive could become significant; a model for the torque loss due to gear churning is given by Park et al. [4].

$$T_{churn} = C_m \rho \omega^2 R_o^3 b l$$

In low ratio final drive speeds are up to 350 rev/min, however in high ratio this can increase to over 1600 rev/min. From the equation above a 5 fold increase in gear speed will increase the churning losses by a magnitude of 25. However, in high ratio the torque referred to the input shaft of the transmission is also increased by an order of 5, since the belt ratio has changed from a nominal 2.5 to 0.5. Hence the change in churning losses seen at the input shaft can vary by up to 125 times across the ratio range of the transmission.

3.6 Bearing Losses

The bearing losses can be based on conventional bearing theory as proposed in bearing manufacturers catalogues [15]. In particular for this work it is important to have an understanding of how the speed dependant losses in the bearings may vary. The torque moment for speed dependant losses is given below.

if $vn \geq 2000$, then

$$M_0 = 10^{-7} f_0 (vn)^{2/3} d_m^3$$

when $vn \leq 2000$

$$M_0 = 160 \times 10^{-7} f_0 d_m^3$$

3.7 Seal Friction

The torque loss associated with seals is due to the friction occurring between the seal and the sealing surface. This is likely to depend on a range of factors, including seal design, fluid pressure, rubbing speed and surface finish. A number of seal manufacturers produce readily available data on the friction losses due to their products.

4 Measuring the Losses

4.1 The Test Rig

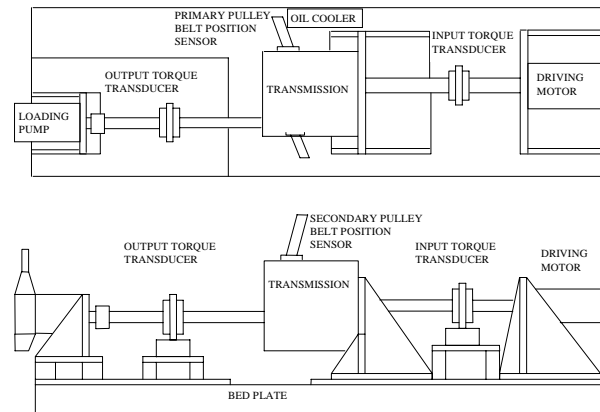
In developing the test rig, the main aim was to be able to accurately measure small changes in the torque losses through the

transmission. This was achieved by specifying highly accurate low torque capacity (100Nm) torque transducers.

So as to fully understand the speed dependency of any losses, the test rig was developed with an input speed range of 750 to 4500 rev/min. Similar to that existing in a vehicle application. These criteria further dictated the output speed range (60 to 1750 rev/min.) and the maximum input torque likely to be required (85Nm).

Figure 4 show a plan view and a side view of the rig. The transmission is driven by a hydraulic motor, and is loaded by a pressure controlled hydraulic load pump.

Figure 4 Schematic Plan and Side View of Test Rig



An oil to water heat exchanger is fitted in place of the oil to air unit used in 'in vehicle' applications. Radial probes are fitted to the transmission and measure the actual position of the V-belt running on the pulleys. Other variables measured are highlighted in Table 1 below. Up to 16 temperatures around the gearbox can also be measured, to collect data on temperature distributions within the transmission.

Table 1 Measured Test Rig Variables

Variable measured	Transducer/ Range
Input Torque (T_{in})	HBM T10f torque flange (0-100Nm)
Output Torque (T_{out})	As above
Input Speed (ω_{in})	Magnetic pick-up (0-4500 rev/min)
Output Speed (ω_{out})	Magnetic pick-up (0-1750 rev/min)
Primary Pressure (P_{prim})	0-40 bar
Secondary Pressure (P_{sec})	0-20 bar
Clutch Pressure (P_{CL})	0-10 bar

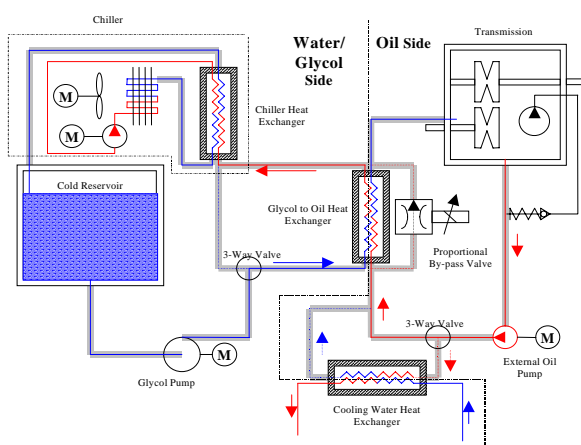
Primary Radius (R_{pri})	Pulley	Radial probe LVDT
Secondary Radius (R_{sec})	pulley	Radial probe LVDT
Belt Speed (V_{belt})		Magnetic pick-up Up to 20000 Hz
Primary Pulley Speed (ω_{pri})		Magnetic pick-up
Throttle Position (X)		Potentiometer
Sump temperature (T_{sump})		T-type thermocouple -10 to 120°C
Clutch temperature (T_{CL})		T-type thermocouple -10 to 120°C
Up to 12 other temperatures ($T_1, T_2, T_3, T_4, T_5, T_6, etc.$)		T-type thermocouple -10 to 120°C

4.2 Low Temperature Testing

An important factor in the work being undertaken is the effect of temperature on the loss mechanisms of the transmission. In particular the effects of cold temperatures.

It was therefore necessary to develop a system to maintain the transmission at a low temperature during testing. Figure 5 shows a schematic diagram of the system developed. The system comprises an insulated cold reservoir containing 70 litres of 50% antifreeze solution that is pre-chilled to -30°C. This acts as a heat sink for the heat energy generated by the transmission. Transmission fluid is pumped around a heat exchanger circuit fitted with a proportional bypass valve, which is controlled by a temperature control module, with PID control.

Figure 5 Transmission Chiller Circuit



4.3 Test Methods

The testing undertaken with the test rig can be defined into two distinct methods.

Namely, steady state testing, and temperature transient testing.

In steady state testing all test rig parameters were maintained continuously during the period which test data was acquired. In temperature transient testing the test rig was set to pre selected speed and load conditions, and then data was acquired throughout the transmission warm up period.

Initial testing was undertaken to ensure that the transmission under test was in a 'run in' condition.

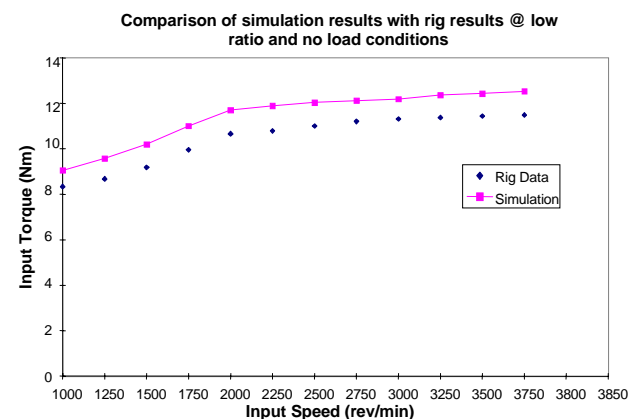
5 Preliminary Results

5.1 Steady State Simulation Results

Initial steady state test results have shown similar trends to those predicted by the modelling work. Figure 6 and Figure 7 show experimental data compared to the initial simulation work for low and high ratio situations respectively. The characteristic shape of the curve in low ratio is due predominantly to the pressure demand on the hydraulic pump, which is dependant on the Hydraulic controller within the transmission.

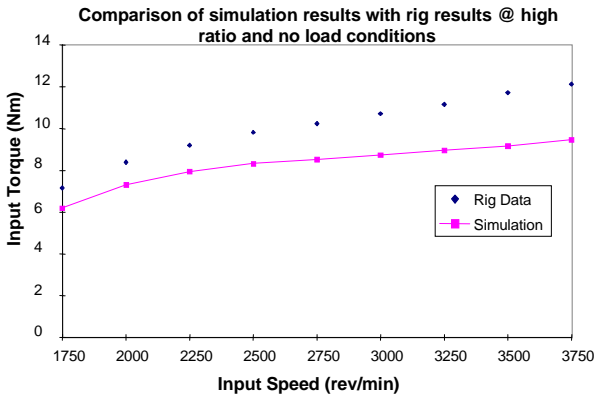
The losses in high ratio at low speeds are lower than the low ratio losses, due in part to the reduced pump losses. However as the speeds increase the high ratio losses exceed those in low ratio. This is due to the increased belt torque losses, and final drive losses.

Figure 6 Low Ratio Simulation vs. Experimental Results



The high ratio simulation appears to diverge from the experimental results, indicating that improvements need to be made to both the belt model and final drive model.

Figure 7 High Ratio Simulation vs. Experimental Results



5.2 Steady State Load dependant Losses

Figure 8 & Figure 9 below show the effect of load on the losses through the transmission. The data represents the losses relative to the no load losses. Dotted lines indicate that the transmission has changed ratio from the initial conditions.

It can be seen that in both ratios torque loss increases with output torque loading. However, in low ratio conditions the torque loss under no load conditions is higher than when the transmission is lightly loaded. This is shown by the negative values in Figure 8.

Figure 8 Effect of Load on Low Ratio Transmission Losses

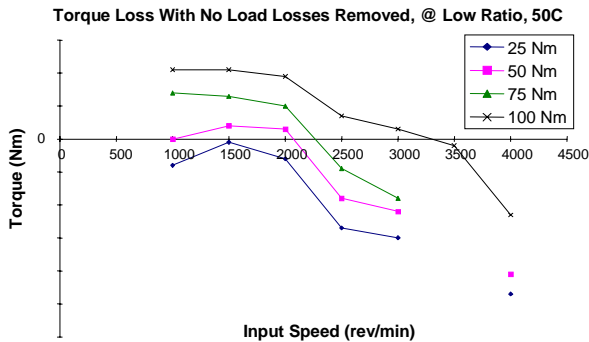
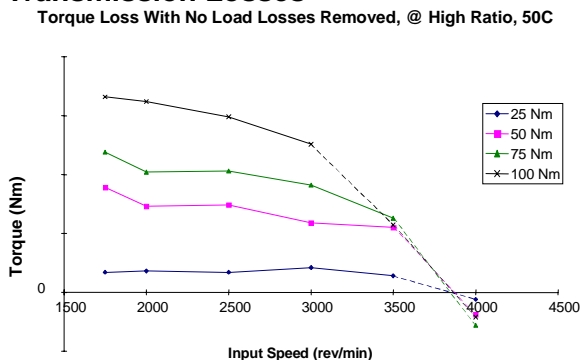


Figure 9 Effect of Load on High Ratio Transmission Losses



5.3 Temperature Transient Torque Loss

Shown below in Figure 10, Figure 11 and Figure 12 are some typical temperature transient results for a range of speeds, ratios and loads. It can be seen that in all cases the input torque at first decreases as the transmission begins to warm up, then the losses begin to increase as the temperature increases further. When a predetermined temperature is reached the cooling to the transmission is turned on and the torque losses recorded as the transmission cools.

Figure 10 Low ratio 1500 rev/min

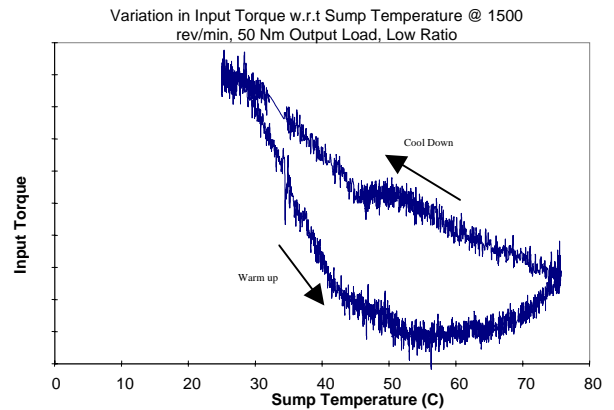


Figure 11 Low ratio 2500 rev/min

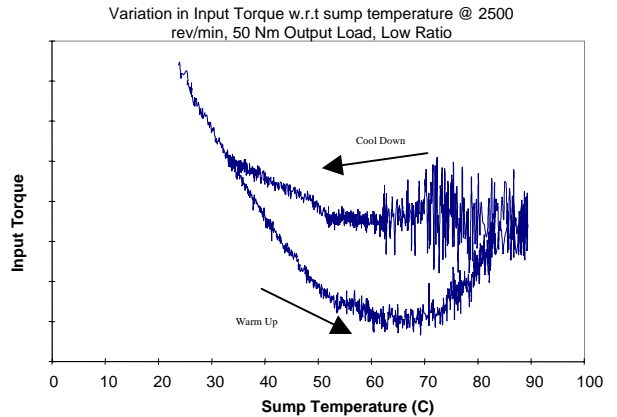
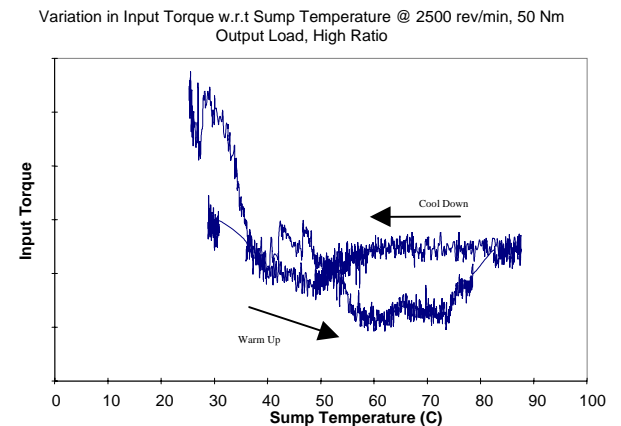


Figure 12 High ratio 2500 rev/min



5.4 Steady State Temperature Results

Initial steady state testing has shown good repeatability and some interesting temperature dependant results. Initial results from a second transmission being tested for back to back comparison show reliable repeatability between transmissions. Figure 13 and Figure 14 show no-load steady state results, where the losses decrease as the temperature increases. However, Figure 15 shows similar steady state results with the transmission loaded, in this case the losses begin to increase again once the temperature exceeds a certain value. These results compare well with the transient temperature results described previously.

Figure 13 Unloaded Low Ratio

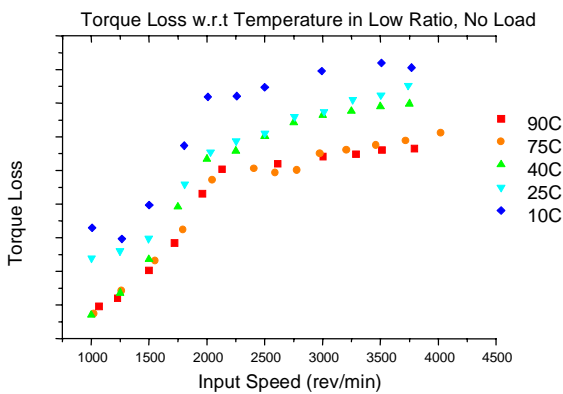


Figure 14 Unloaded High Ratio

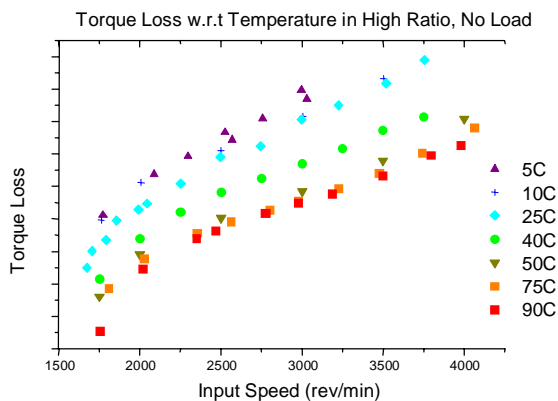
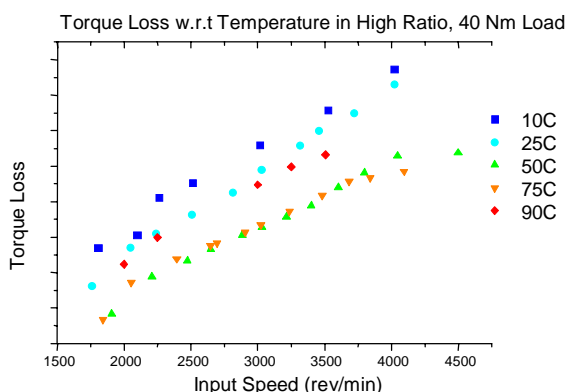


Figure 15 Loaded High Ratio



6 Future Work

To date the whole transmission has been tested across the complete operating envelope available on the test rig. A second transmission is currently being tested to offer a back to back comparison with the first transmission. Following this the transmission will undergo breakdown testing in which individual components will be removed sequentially, so that the individual loss mechanisms may be isolated.

Once the individual components have been isolated experimentally the individual models will be validated and improved upon where it is considered necessary.

7 Conclusions

A test rig has been developed that measures accurately low power level losses through a belt drive CVT, at a range of temperature conditions.

Initial steady state and transient temperature results have shown good repeatability. Trends shown in transient temperature testing have been shown to exist in steady state testing. It has been shown that torque loss through the transmission varies significantly with respect to temperature, and with bulk fluid temperatures in excess of 65°C the torque loss through the transmission can actually increase.

Initial modelling work has produced results that show comparable trends to the experimental results, while analysis of previous work has provided an insight into new aspects of the belt mechanism, which can be developed to improve the belt model.

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