2001-01-0874

Performance Investigations of a Novel Rolling Traction CVT

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ABSTRACT

The Milner CVT is a patented rolling traction transmission offering advantages of high power density and simplicity of construction and operation.

A 90 mm diameter prototype variator is described which was sized for a maximum rated input power of 12 kW. Experimental data are presented demonstrating high efficiency and low shift forces. Resistance to overload torque is shown to be exceptional and preliminary durability trials indicate a highly viable concept for series production.

Based upon the measured data, characteristics of larger variators are predicted and prospects for automotive applications discussed.

INTRODUCTION

CVTs of various forms were invented a considerably long time ago **[1]**, but it is only in the last two decades with the constraints of emissions and fuel consumption, being thrust upon automotive engineers, that their use is being seriously considered. There are many different variants of CVT from rolling traction drives to variable sheave V-belt drives and split power, hydraulic motor/pump variator arrangements for use in high powered machinery. The Milner CVT (MCVT) is a new concept CVT that fits into the rolling traction drive sector of the CVT family.

The MCVT concept utilises three or more spherical planetary rollers to transfer drive from an inner race set to a carrier assembly. Altering the axial separation of a static outer race set varies the ratio. The variator is arranged with its output shaft co-axial with the input and is very compact and light for the torque and power transmitted.

These characteristics make the MCVT ideal for applications where space required for a conventional CVT is unavailable and for most applications where weight is an important factor. Torque dependent loading and ratio changing are both accomplished without the use of hydraulic systems, to the benefit of simplicity and overall efficiency. Materials and machining processes employed are adapted from conventional rolling bearing technology.

The prime objective of the work described here was to design, manufacture and test a relatively small prototype MCVT variator to determine the key performance characteristics of efficiency, torque capacity and life.

MCVT OPERATING PRINCIPLE

At the heart of the Milner CVT (MCVT) lies a rolling traction variator of high torque capacity and basically simple construction. It is essentially a variable geometry four point contact ball bearing with power transfer to or from the planet balls by roller followers located between the balls and mounted on a rotating carrier; see **Figure 1**.

The substantial contact forces necessary to support rolling traction are contained entirely within the races of the variator assembly. Precisely the correct amount of contact force for any input torque is automatically generated by a simple helical mechanism built into the inner race assembly. This mechanism is normally a ball screw but may be a cam and follower arrangement. The same type of mechanism is employed in the outer race assembly to provide low-load ratio changing.

When the ratio change actuator is rotated, typically by an electric actuator, the axial separation between the outer race halves changes and the planet balls automatically adjust their position and contact points with the raceways. At the same time, the inner race halves automatically adjust their axial separation and contact points with the planet balls. The overall function may be likened to that of a normal epicyclic (planetary) gearbox fitted with compound planets (one gear to mesh with the annulus and another with the sun) in which all four gears have variable radii; see **Figure 2**.

The MCVT variator may be made fully reversible with respect to both torque and rotation directions, or it may be equipped with a built-in freewheel for uni-directional applications. Input may be to the inner race assembly with output from the roller follower carrier, as shown in the figures, or vice versa, depending on the application. No high-pressure hydraulic control system is required and lubrication is by grease, splash or a low power hydraulic system using a special traction lubricant.

An additional design feature of great benefit in certain applications, especially automotive transmissions, is the availability of a simple, passive, direct drive top gear lock-up facility. This increases top gear efficiency by about 10% to approximately 99%. Since the aim of the work reported here was to investigate the rolling characteristics of the transmission, this feature was not included in the present design.

It should also be noted that, due to its epicyclic type layout, the MCVT variator combines well with epicyclic gearsets to match prime mover to load and/or create various split torque, dual mode and 'infinitely' variable transmission configurations.

GENERIC DATA FOR VARIATOR

Performance

- Capacity - generic design scaleable to cover at least the range from 1kW to 300kW
- Efficiency – typically 85% to 90% with 99% available in direct drive lock-up
- Ratio range – ideally up to 5:1 but up to 8:1 with reduced rating for additional ratio range
- Life – typically 1Grev (referred to input) but as long as required; failure modes same as rolling bearings (overload, contact surface fatigue, debris, poor lubrication)
- Static torque capacity safety factor – as large as required but typically 3:1

Package

- Torque density (mass) – Up to 30Nm/kg (rated input torque) equivalent to 22kW/kg at 7000rpm excluding external support systems (ratio change actuator and oil cooling circuit)
- Torque density (volume) – Up to 150Nm/litre (rated input torque) equivalent to 110kW/litre at 7000rpm excluding external support systems (eg variator 140mm diameter by 100mm long rated at up to 230Nm input torque)
- Parts count - 7 principal components (excluding ratio change actuator, oil cooling circuit, small parts and standard parts)

Integration & Interfacing

- Layout - single shaft type variator (co-axial input/output shafts); input/output shafts on same or opposite sides
- System flexibility - integrates ideally with input/output gearing to match driver/driven requirements and/or produce IVT with integral reverse and launch capability; uni- or bi-directional drive according to requirement
- Mounting - via housing or supported on input shaft, depending on configuration
- Ratio control - rotating collar driven mechanically or by electric actuator
- Shift management - fully flexible (manual, automatic or combined)
- Ratio availability - continuously variable or stepped
- Shift speed - to choice; can be extremely fast (eg for stepped ratio emulation)
- Shift quality - excellent; power ('hot') shifting possible

DESIGN OF TEST TRANSMISSION

The test transmission, shown in Figure 3, was designed expressly for the test programme, without regard for any potential specific commercial application.

Two versions were tested, one with a ratio range of 4:1 and one with 5:1. The latter was achieved by using larger planet balls and a re-cut outer set.

Input drive was to the smaller shaft and output to the larger. An oil cooling circuit (not shown) was installed consisting of a low pressure, low flow rate, supply of traction oil to the inner race 'nut'.

A photograph of the test transmission, partly disassembled is shown in Figure 4.

THE TEST RIG

The test program was performed on a test rig developed previously at the University of Bath to measure accurately the no load and low load torque losses through automotive transmissions. A schematic drawing of the test rig is shown in Figure 5 and the transmission installation on the rig can be seen in Figure 6.

The test rig consists of a hydraulic motor to drive the test transmission and a hydraulic pump to load the transmission. Thus the input speed and output load on the transmission can be modulated by controlling the oil flow to the motor and the pressure at the pump outlet respectively. The rig is instrumented to measure with a high degree of accuracy both the input and output speeds and torques so that the efficiency, power loss or effective slip through the transmission may be calculated.

Accuracy of the torque transducers from previous work and the through shaft tests described below has been shown to be better than +/- 0.05 Nm. The accuracy of the speed measurement on both sides of the transmission is +/- 0.1%.

A stepper motor system was adopted as the method of ratio change for the test rig installation. The motor was installed in a rotating cradle such that the shift torque could be calculated from a strain gauged reaction arm.

TEST PROGRAMME

The test program performed on the MCVT was split into a number of different areas. Initial efficiency test were performed and these were followed by a number of static overload and endurance tests. Other tests performed on the transmission were through-shaft tests and lower μ traction fluid tests.

EFFICIENCY TESTS

The efficiency tests were performed at a range of speeds, ratios and input torques. These were

- Ratio
  - 0.2, 0.35, 0.65 (0.2 replaced by 0.16 for 5:1 ratio transmission)
- Torque
  - Input Torque set at 5, 12.5 and 20 Nm in addition to tests with output shaft disconnected
- Input Speed
  - For each test condition results were obtained at 1000, 3000 and 6000 rev/min

Therefore a complete test matrix totals 36 individual tests on each transmission.

THROUGH-SHAFT TESTS

These tests were performed to measure any inefficiency within the couplings on the rig, or any inaccuracy in the torque transducers. A rigid shaft was installed in a transmission body, allowing a 1:1 lockup ratio to be used. The results can be found in **Table 1**. In conclusion the tests showed that the losses in the couplings were negligible, and that the calibration of both the torque and speed transducers was highly accurate.

STATIC OVERLOAD TESTS

The static overload testing was performed on the same test rig facility used for the efficiency test work. To perform the tests the output shaft of the transmission was mechanically locked and a simple static torque load was applied using a lever on the input shaft of the transmission. The static overload input torque was applied in steps of around 10 Nm up to the expected failure torque. If the expected failure torque was exceeded then the input torque was incremented in further steps of 20 Nm. After each application of the overload torque the transmission was run at 1000 rev/min and minimum load to detect any signs of failure. The methods for detecting failure were excessive noise, a change in efficiency, a change in effective ratio, or failure to operate. The ratio change mechanism was also operated around the test condition to detect any defectiveness in its operation. Two transmissions were tested in this part of the work in high ratio (0.65), and low ratio (0.2) respectively.

LOWER μ TESTS

A number of tests were performed on the MCVT using lower coefficient traction fluids. This testing was again performed on the same test rig as used for the efficiency work. For each fluid the transmission was tested over the standard nine point matrix (3 ratios at 5, 12.5 & 20 Nm input torque) at 3000 rev/min input shaft speed. A baseline test was performed using a standard Santotrac 50 traction fluid. This was then compared to Findett 2080A low μ traction fluid. Two further tests were performed by diluting the 2080A with standard ATF (automatic transmission fluid), at 75% 2080A to 25% ATF and 50% 2080A to 50% ATF.
Small increases in micro slip occurred as the traction fluid was diluted. However, the MCVT always maintained traction even at the highest load conditions and never operated under gross slip conditions.

TEST RESULTS

EFFICIENCY TESTS

A number of plots of power loss vs. input speed and torque at different ratios and for different MCVT configurations are shown in **Figure 7** through to Figure 15. Due to the small power rating of the MCVT designed for these tests it is better to describe results as power losses rather than efficiencies, since the efficiency can be skewed by the low power levels being transmitted. This can be illustrated by examining no load condition, where the efficiency will not be calculable.
Figure 7 to **Figure 9** show the power losses in a standard 4:1 configuration of the MCVT. The graphs show clearly that the power loss increases at the lower ratio conditions. Maximum power loss in low (0.2) is 2800 W, in medium (0.35) it is 2250 W, and in high (0.65) it is 1600 W. The general trends show an almost linear change in power loss with speed and a similar linearity with respect to input shaft load. Power loss at low loads and low speeds is negligible.
Figure 10 to **Figure 12** show similar results obtained in a 4:1 configuration with the standard steel planet balls replaced by ceramic balls. These results show a clear reduction in the power loss of approximately 400W in low ratio, 200 W in medium ratio, and 100 W in high ratio.
Figure 13 to Figure 15 show the results achieved with the transmission in a 5:1 ratio range configuration. In all the test conditions the power loss has increased slightly over the 4:1 configuration. The low ratio test condition in this configuration is performed at i=0.16 and this results in a power loss of over 3000 W.

The power losses described above equate to a maximum efficiency of approximately 90%. The efficiency is highest in high ratio (0.65), and reduces as the ratio decreases, such that efficiencies are around 83% in medium ratio (0.35), and 75% in low ratio (0.2). The replacement of the standard balls with ceramic balls increases the efficiency across the operating range by 1~2%. Some drain down tests that were performed showed that there was a small torque loss component associated with the churning of the transmission lubricant.
It should be noted that the almost linear relationship between speed and power loss, and torque and power loss, described above, results in an effectively flat efficiency profile for the transmission that varies only with ratio.

RELIABILITY TESTS

Both the transmission units tested far exceeded their design input failure torques. In high ratio failure was expected to occur at 55 Nm in the inner ball screw race, while in low ratio failure was expected to occur at 50 Nm. The high ratio test eventually failed at 180 Nm while the low ratio unit was deemed to have failed at 120 Nm. In both cases the failure mode was deformation of the follower carrier. In high ratio there were noticeable flat spots on all three of the followers. In low ratio there was also some damage to the front outer bearing race surface.

ENDURANCE TESTS

The endurance tests were performed at a mid point in the operating envelope of the transmission. The transmission was run at this condition for nearly 100 hours with no degradation in performance apparent. It was decided to abandon endurance testing of the MCVT at this stage, in favor of performing a more rigorous endurance test exercise on the next generation of MCVT.

RESULTS ANALYSIS & DISCUSSION

At the design stage, prior to the test programme, it was not known how large a coefficient of traction the design would support. The traction fluid used, Findett Corporation's Santotrac 50, has a quoted coefficient of traction of 0.1, but this was measured under different conditions from those prevailing in the MCVT. Since catastrophic slip had to be avoided at all costs, it was decided to design the MCVT very conservatively, and a maximum value for the coefficient of traction of 0.63 was accordingly adopted. This was achieved simply by an appropriate choice for the lead of the inner race ball screw.

At the conclusion of the test programme reported above, tests were run with a range of traction fluids of lower coefficient of traction without any sign of catastrophic slip. It is therefore concluded that the main test programme would still have been successfully completed if the inner race ball screw lead had been increased sufficiently to reduce contact forces by about 25%. This would be expected to increase efficiency by about 2% and torque capacity by about 25%. As a result of these findings, later designs have been specified to utilise higher coefficients of traction.

LARGER SIZES & AUTOMOTIVE APPLICATIONS

The key components determining the torque capacity of the MCVT are the rolling assembly and the ball screws. It is known from the international standards governing the ratings for rolling bearings and ball screws that both these systems posses static load capacities approximately in proportion to the volume of the transmission, or the cube of a linear dimension.

In the tests reported here, the 90mm diameter, 50mm long test variator did not fail until the input torque exceeded 100Nm. This represents a torque density of over 300Nm/litre to failure. According to the safety factor required, and any de-rating necessary to attain the desired fatigue life, rated torque is normally between 50 and 150Nm/litre.

The weight of the MCVT variator depends on the materials used, but is typically 5kg/litre. Torque density based on mass, therefore, is normally between 10 and 30Nm/kg. A complete automotive transmission also requires a reverse ratio, a launch device, a shift actuator and an oil cooling circuit. These are expected, typically, to double the mass of the system and reduce the torque density range to 5 to 15Nm/kg.

It is not yet known where, within this range, automotive MCVTs will eventually lie, but the signs are encouraging. The lower end of the range corresponds approximately to 'lighter than a conventional automatic' and the upper end to 'lighter than a conventional manual' with the middle of the range corresponding approximately to where a well designed manual transmission lies today.

CONCLUSIONS

In general the performance of the MCVT has exceeded the results expected from initial design calculations. Although peak efficiencies are lower than those expected from a normal fixed ratio mechanical transmission, they are on a par with those measured for other traction drives and metal V-belt CVT arrangements. The clear advantage the MCVT has over most other CVT arrangements is that there is no requirement for a high pressure, hydraulic, ratio control system.

The MCVT appears to have particular benefits when compared to rubber V-belt CVT arrangements used in applications from 50 cc scooters, ATVs and Snow mobile applications. These existing transmissions have both a low efficiency and poor reliability, especially when operated in high levels of contamination.

Future iterations of the MCVT design will take into account the results of the testing with lower traction coefficient fluids and relax the safety factors that were used in the design calculation for these test pieces. This should result in a lower loading on the rolling element, less deformation of the rolling elements and thus a higher efficiency and prolonged fatigue life.

FURTHER WORK

A new test program to investigate the performance of a larger sized MCVT in a 60 kW motorcycle application is currently under development in conjunction between the University of Bath, Peter J Milner Consultant Engineers, and Yamaha Motor Europe. The work will concentrate on applying the MCVT as an electrically actuated step shift transmission, as well as a CVT.

CONTACT

Any questions concerning the MCVT may be directed to Mr. Peter Milner at: peter@pjmilner.demon.co.uk

Any questions regarding the test program performed at the University of Bath may be directed to Mr. Sam Akehurst at: enssa@bath.ac.uk

REFERENCES

1. Lubomyr O. Hewko: Automotive Traction Drive CVTs – An Overview. SAE Paper 861355, 1986
2. Ian Adcock: Ball-Bearing Gearbox is a World First – European Automotive Design, September 1998, pp. 26-27

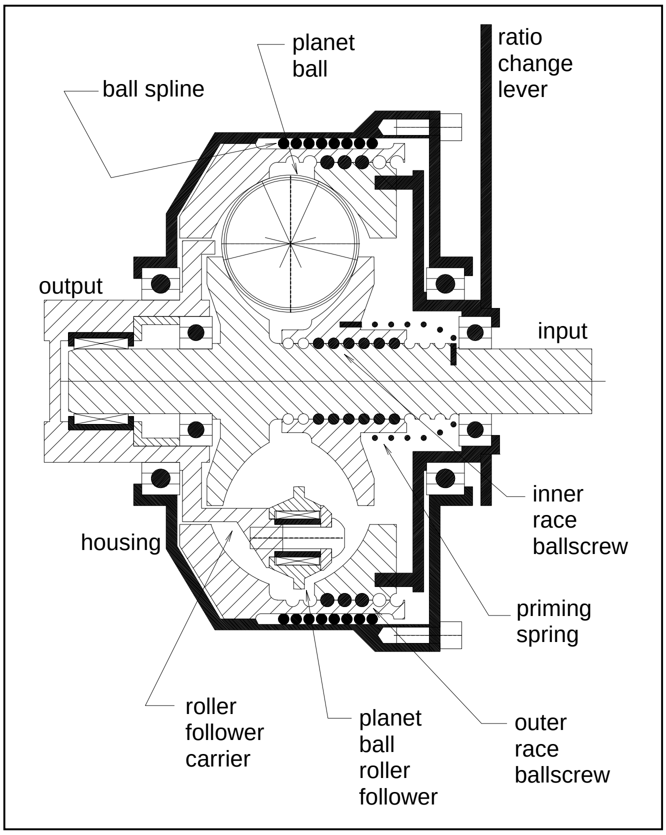

Figure 1 MCVT General Arrangement Schematic

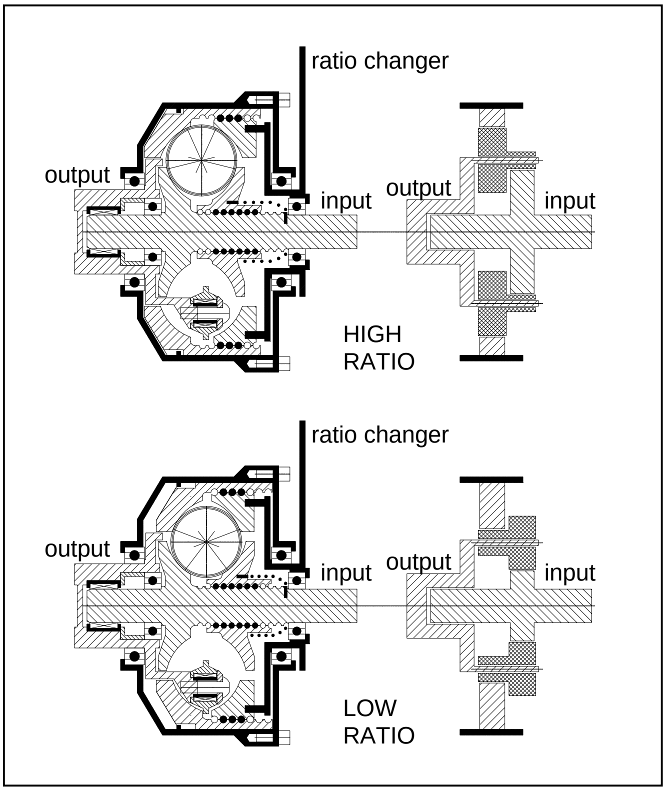

Figure 2 Principle of Operation

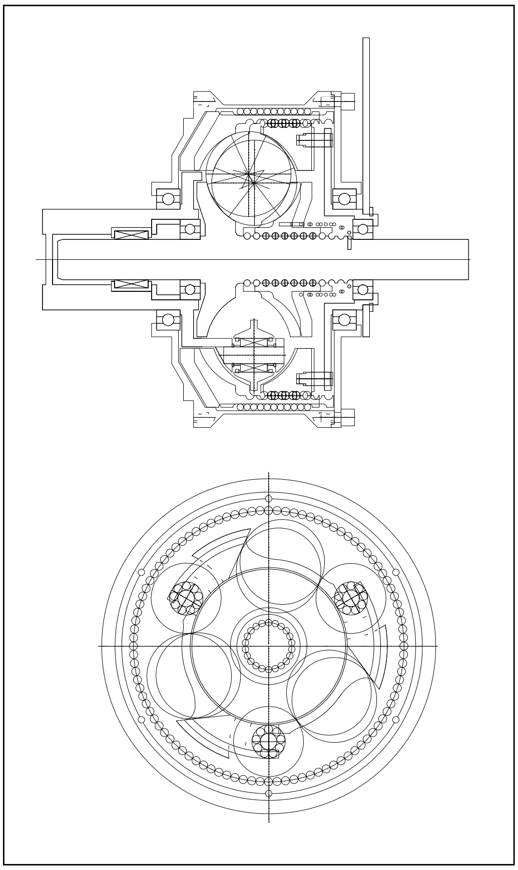

Figure 3 The Test Transmission

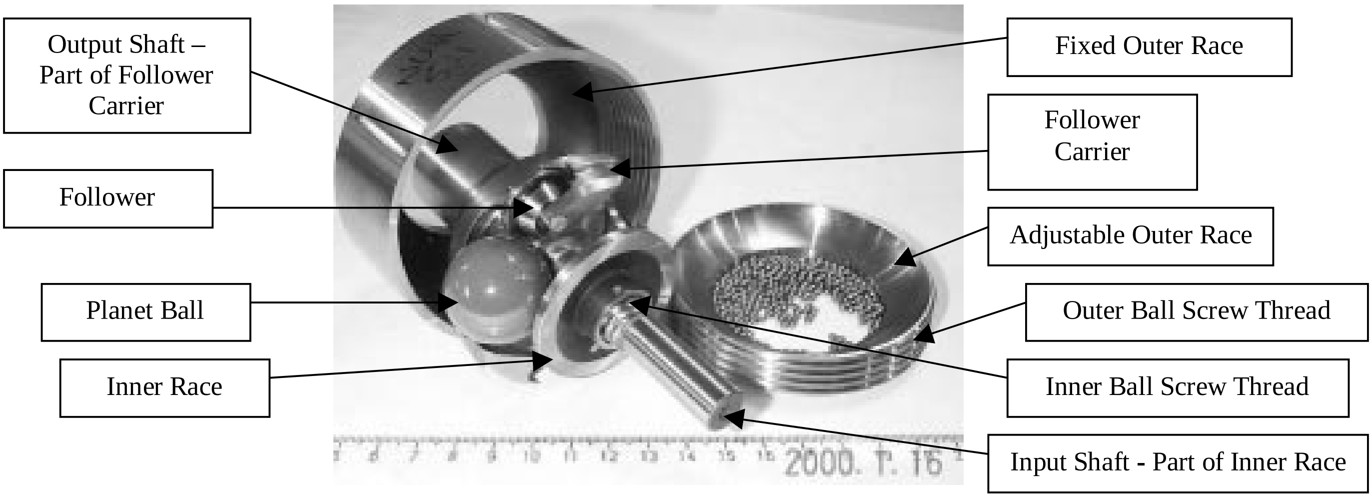

Figure 4 MCVT Assembly

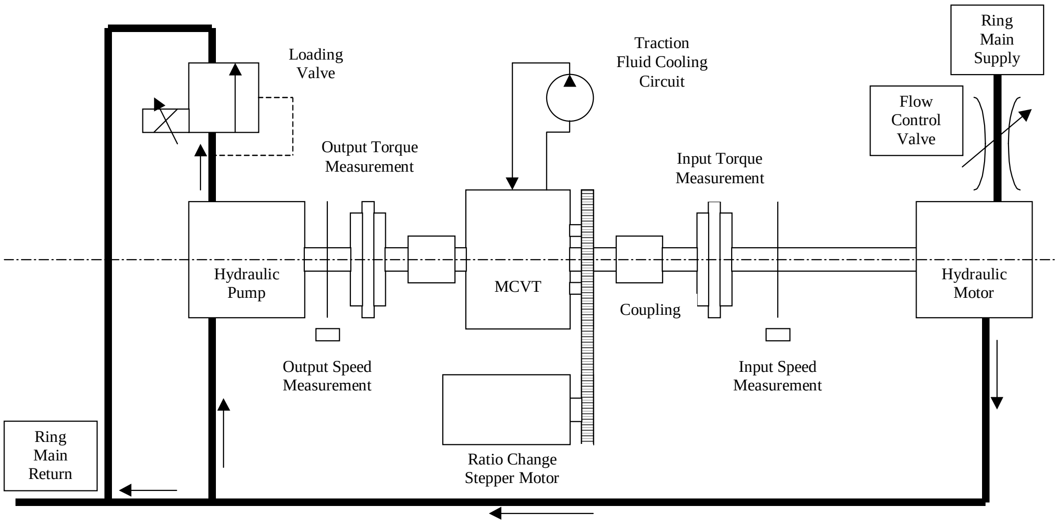

Figure 5 Schematic of Test Rig

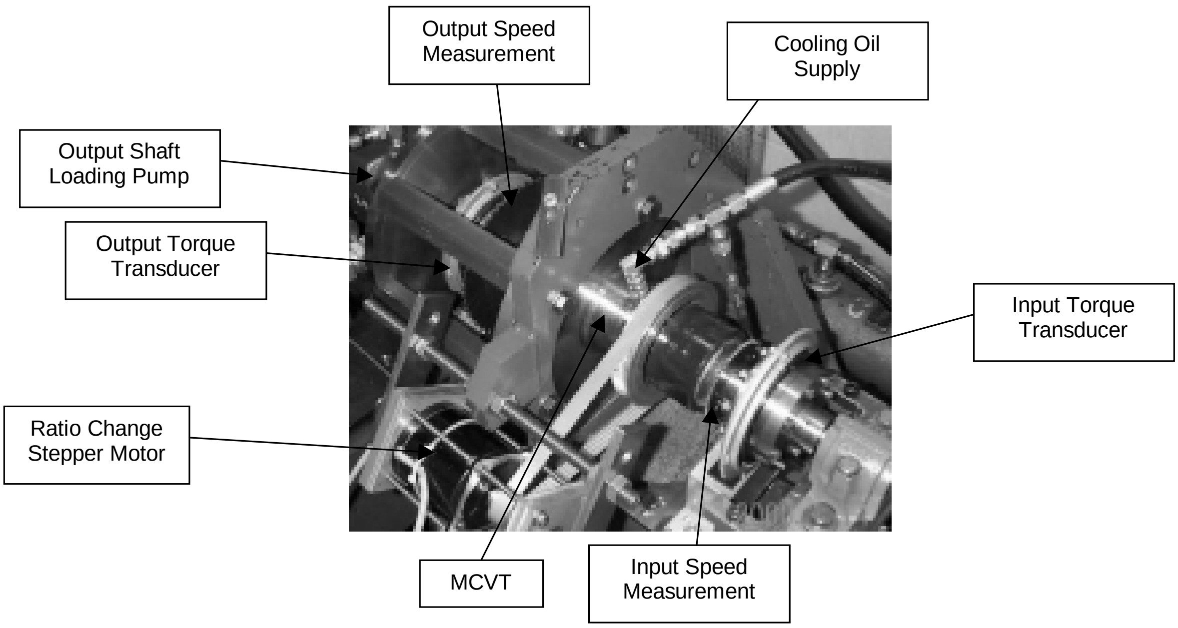

Figure 6 Photographs of the Test Rig

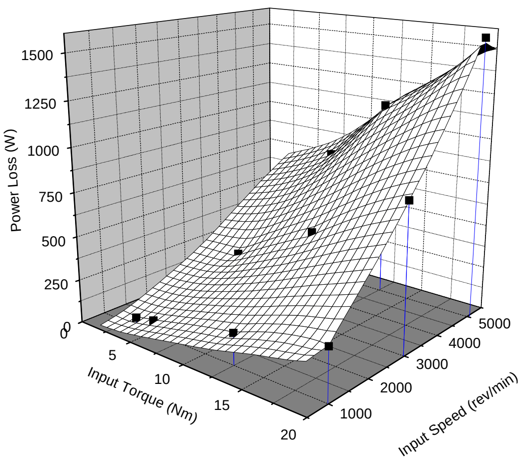

Figure 7 Power Loss 4:1 Transmission i=0.65

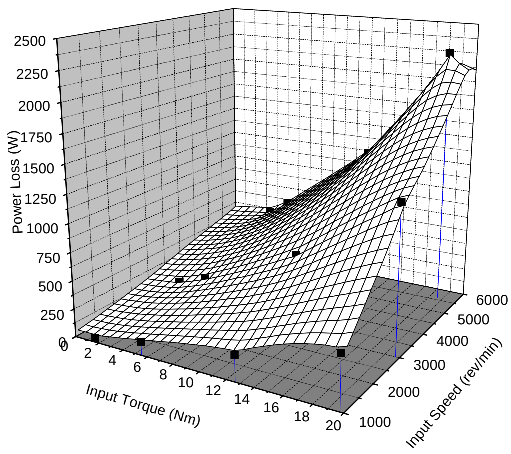

Figure 8 Power Loss 4:1 Transmission i=0.35

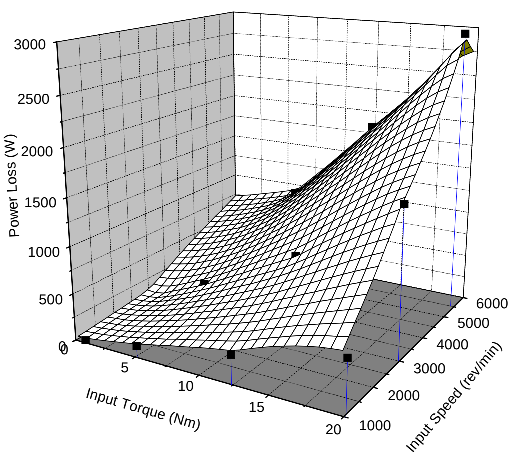

Figure 9 Power Loss 4:1 Transmission i=0.2

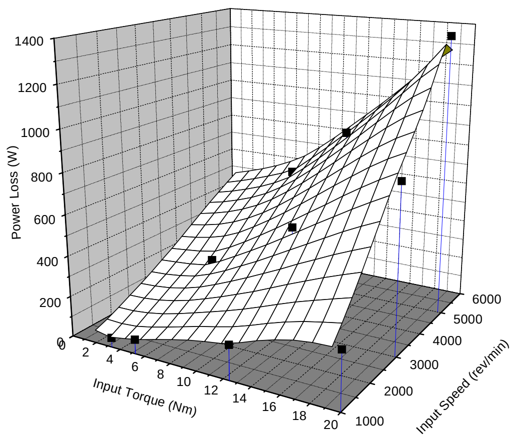

Figure 10 Power Loss 4:1 Transmission Ceramic Bearings i=0.65

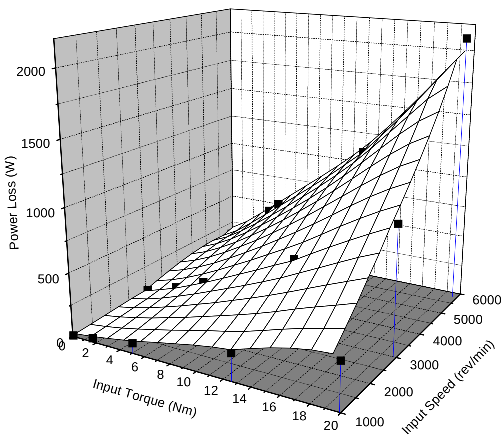

Figure 11 Power Loss 4:1 Transmission Ceramic Bearings i=0.35

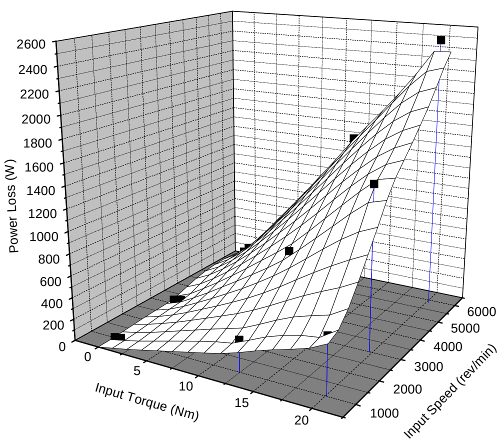

Figure 12 Power Loss 4:1 Transmission Ceramic Bearings i=0.2

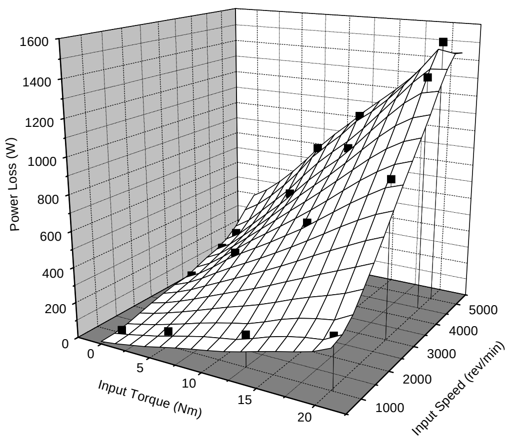

Figure 13 Power Loss in 5:1 MCVT at i=0.65

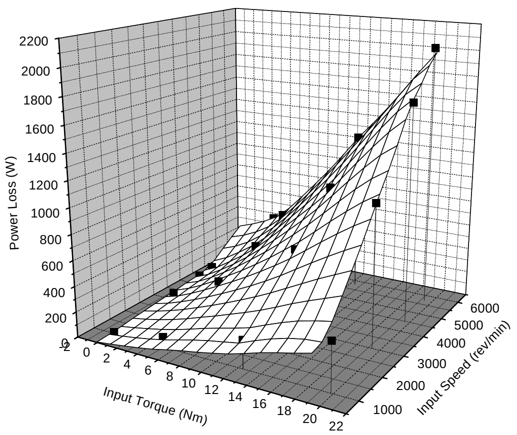

Figure 14 Power Loss in 5:1 MCVT at i=0.35

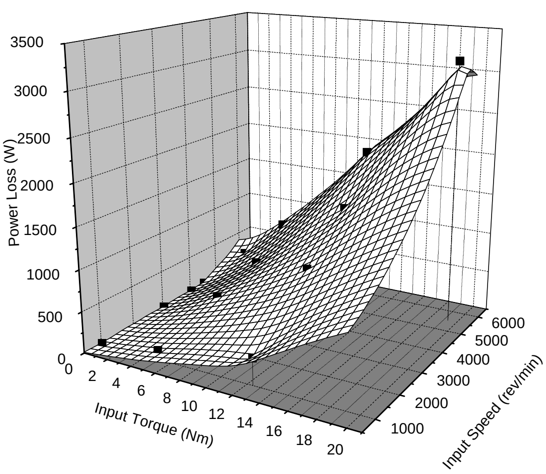

Figure 15 Power Loss in 5:1 MCVT at i=0.16

Table 1 Results of testing with straight through shaft

Test conditions Thru Shaft Ratio: 1 Notes:
Date 09/12/99

Measured Parameters				Calculated Parameters					
INPUT TORQUE	INPUT SPEED	OUTPUT TORQUE	OUTPUT SPEED	POWER IN	POWER OUT	EFFICIENCY	POWER LOSS	SPEED RATIO	TORQUE RATIO
Nm	rev/min	Nm	rev/min	w	w	%	w	#	#
5.22	996.48	5.17	993.54	544.66	537.68	98.72	6.97	0.997	0.990
12.59	1007.12	12.52	1004.25	1328.27	1316.40	99.11	11.86	0.997	0.994
20.04	1009.18	19.95	1006.53	2117.89	2103.02	99.30	14.88	0.997	0.996
7.48	2024.33	7.41	2023.10	1586.27	1569.60	98.95	16.67	0.999	0.990
12.62	2035.64	12.53	2034.72	2689.41	2668.96	99.24	20.45	1.000	0.993
20.26	1990.69	20.16	1989.68	4223.70	4199.56	99.43	24.14	0.999	0.995
12.43	3009.47	12.30	3010.57	3917.37	3878.42	99.01	38.95	1.000	0.990
20.14	2997.53	19.98	2998.71	6323.00	6274.53	99.23	48.47	1.000	0.992

Test conditions Thru Shaft Ratio: 1 Notes:
Date 13/12/99

Measured Parameters				Calculated Parameters					
INPUT TORQUE	INPUT SPEED	OUTPUT TORQUE	OUTPUT SPEED	POWER IN	POWER OUT	EFFICIENCY	POWER LOSS	SPEED RATIO	TORQUE RATIO
Nm	rev/min	Nm	rev/min	w	w	%	w	#	#
0.02	990.81	0.00	990.81	2.08	0.00	N/A	2.08	1.000	N/A
0.05	2009.53	0.00	2009.53	10.84	0.00	N/A	10.84	1.000	N/A
0.09	2994.66	0.00	2994.66	28.85	0.00	N/A	28.85	1.000	N/A
0.15	4510.26	0.00	4510.26	68.71	0.00	N/A	68.71	1.000	N/A