ORSHANSKY TRANSMISSION CORPORATION

NEW YORK

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QUARTERLY PROGRESS REPORT

PHASE IV

DESIGN AND DEVELOPMENT OF
A CONTINUOUSLY VARIABLE RATIO TRANSMISSION
FOR AN AUTOMOTIVE VEHICLE

February 28, 1978

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I. INTRODUCTION

Under Modification A004 to Contract EY-76-C-03-1165 several tasks were actively pursued in the three-month reporting period December, 1977 through February, 1978. This report concentrates on the following tasks where the progress was significant.

- Task 6A Study of hydraulic control packaging for the second generation transmission.
- Task 17 Improve driveability of the demonstration car with the installed first generation transmission.
- Task 22 Development and testing of co-axial hydrostatic module for the second generation transmission.
- Task 31 Investigation of compliant materials for mounting the hydrostatic module.
- Task 38 Investigation of techniques for reducing the air entrainment in the hydrostatic working fluid.

Major emphasis has been placed on Task 22 where hardware development problems have resulted in serious schedule slippage. The co-axial module has been designed in accordance with industry practice but, for reasons not fully understood, fails to operate at design speed, pressure and displacement without serious damage to the port plates occurring. Investigations of the causes will continue to be the priority task.

Ford Motor Company, as subcontractor, has continued to review the progress of the second generation transmission design. A design review meeting was held with two of Ford's transmission specialists at the Orshansky facility on January 10, 1978 with a representative from the Jet Propulsion Laboratory Automotive Group in attendance.

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The meeting focused on automotive design practice as it applies to the hydromechanical transmission. The relative merits of alternate configurations for the second generation transmission was also discussed and the Ford engineers indicated a strong preference for the configurations with two simple-planetaries - designated at OTC0476 - on the basis of minimum cost.

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II. TASK 6A

The hydraulic system for the second generation transmission has been brought up to the concept layout stage. The packaging technique is similar to current automatic transmission practice where an open die cast valve body is sandwiched between a manifold casting and a cover casting. The entire valve block is removable from the bottom of the transmission. The valve block placement allows for all of the controls to be submerged so air ingestion is eliminated. The electrohydraulic proportional pilot valve and both electrical solenoids mount to the valve block, and are therefore easily accessible. All oil vents return to the charge pump inlet to minimize aeration. The new control system incorporates the neutral detection safety circuit developed in the concept demonstration transmission. The hydraulic components required to control the co-axial module have been manufactured and are on test in the co-axial module.

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III. TASK 17

"Test Driving the Demonstration Car"

Continued testing of the concept demonstration transmission in the vehicle has resulted in improved controls. The following improvements have been incorporated:

Neutral Sensing Circuit

- 1) Passage cross sections were increased in the sensing circuits that prevent the swashplate from going over-center in low ranges. The net result is to prohibit vehicle reverse motion when forward is selected and vice-versa. This problem manifests itself during hard downshifts when maximum flow is acting on the stroking cylinders from the 4-way valve. The neutral sensing circuits must have the capacity to vent the flow. (fig. 1)

Improved 4-Way Valve Response

- 2) System response has been improved to eliminate oscillations which occurred following a sudden change in throttle position. The oscillations were caused by a response problem in the pilot valve output circuit. The problem was eliminated by reducing the 4-way valve spool travel by 35% and by increasing the flow area in the notches on each spool land by 100%.

Trapped air was also a problem in the same circuit. A small relief valve was added to each signal end of the 4-way valve spool. The relief valve cracks at maximum pilot valve output pressure, beyond the operating range of the 4-way valve. When the relief valve cracks, new oil is introduced to the pilot valve circuit and trapped air is flushed out. (fig. 2)

Control Sequence Spool Valve Modifications

- 3) Inability to negotiate mode changing was observed at speed ratios of 0.34 and 0.85. At the mode change, swashplate control is switched from one hydraulic unit to the other. Timing modifications were made to the sequence control spool lands to eliminate the problem. (fig. 3)

NEUTRAL SENSING CIRCUIT

FIG. 1

4-WAY VALVE RESPONSE

FIG. 2

SEQUENCE CONTROL VALVE TIMING

FIG. 3

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Valve Plate Port Timing

- 4) Unconventional valve plate port timing had been incorporated in the hydraulic units to reduce noise. It was found that while improving noise, the valve plates produced very high swashplate moments. The moments were exceeding the capacity of the swashplate control system under certain high load conditions. The timing was retarded $13\frac{1}{2}^{\circ}$ to correct the problem. (fig. 4)

Clutch Fill Accumulators

- 5) Small accumulators were incorporated in parallel with the high and low range clutches to increase the duration of changeover. The duration was increased from 150 ms to 500 ms which is more in keeping with standard automotive practice. It may be possible to eliminate the accumulators following further shift developments utilizing the new microprocessor.

Relief Valves

- 6) A control problem was also discovered in the relief valve section. The control pressure relief valve had an excessive override at high engine speeds. Adequate pressure control in the auxiliary circuit is most important from the standpoint of operation of the range change solenoid. Prior to modification, system pressure could increase to the point where the range change solenoid could become unseated. In high range, unseating of the range change solenoid could possibly result in a violent and unsafe downshift. The relief valve and associated circuitry have been reworked to eliminate the high override.

Engine Flywheel

- 7) Engine flywheel inertia has been reduced to improve driveability response. The flywheel was originally designed to duplicate the inertia of a standard torque converter/flywheel assembly. This proved to be more than necessary as the equivalent inertia of the HMT is greater than the transmission it replaces.

VALVE PLATE FOR INCREASED FLOW AREA AND MIN. SWASH MOMENT 'B' UNIT

FIG. 4

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The magnetic sensor utilized for transmission input speed has been relocated from the inside of the transmission to the engine flywheel where it is near the crankshaft torsional nodal point. This also makes the transducer a part of the vehicle electrical system. As such it is no longer necessary to remove the speed pick up and related wiring harness each time the transmission is removed from the vehicle. In addition, replacement and/or diagnosis of problems in the engine speed measuring circuit are simplified.

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IV. TASK 22

For first test of the new in-line module parts, the decision was made to run one half of the unit as a pump, to reduce cost and down-time in case of a catastrophic failure. To accomplish this a thick aluminum end cap was manufactured to replace the normal manifold. Figure 5 shows the schematic of the basic test set-up used for tests from late January thru late February, when complete module tests began.

Pump tests were to be made using the variable unit at full displacement, ($D_v = 3.025 \text{ in}^3/\text{rev.}$), at 1000 rpm starting at 500 psi and increasing in 500 psi increments up to 3500 psi, plus one run at maximum pressure obtainable of approximately 3850 psi. Each pressure setting was to be tested for thirty minutes, followed by a complete tear-down and inspection. This was to be followed by the same series of tests at 2000 rpm, 3000 rpm and finally 3500 rpm.

The inspection following the 1000 rpm, 2000 psi run, (Table 1), revealed a scuffed valve plate near the center line of the high pressure kidney centroid under the bearing pad area. The valve plate was re-lapped and tested several times with essentially the same results. Volumetric efficiency was also lower than predicted values. Since the aluminum end cap was not pressure balanced, it was reasoned that if it were deflecting it could cause both valve plate scuffing and a leak path, thus lower volumetric efficiency.

By reducing torques on bolts, leakage was reduced (yet still greater than predicted), and although valve plate scuffing did not seem so severe, damage was still occurring.

Oil trapped between the rotating cylinder block and the stationary end cap, immediately inside the inner sealing land, would create a centrifugal head, tending to separate the two parts. This force was not taken into account in the pressure balance calculations. Holes were

PREPARED BY <i>Sam Reddy</i>	ORSHANSKY TRANSMISSION CORPORATION ENGINEERING CALCULATION FORM	PAGE 1 OF
CHECKED BY	SUBJECT PUMP TEST SCHEMATIC	PROGRAM 48-22
		DATE 1-18-78

FIG. 5

SUBJECT

PAGE 2 OF

PROGRAM

48-22

DATE 1-30

1-25-98

PUMP TEST DATA

CHARGE PRESSURE = $P_{CHG.}$ = 150 p.s.i.
CASE PRESSURE = P_C = 5 p.s.i.
SYSTEM TEMPERATURE = T_S = 120°F. } Constant, all tests.

TABLE 1

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drilled in the cylinder barrel to communicate this cavity to the annulus between the outer sealing land and the bearing pad area. This not only reduced this separating force, but also supplied more oil to the bearing pad to help resist scuffing of the valve plate. This appeared to be beneficial, but did not entirely solve the problem.

Tufftriding and micro-sealing of the cylinder barrel did not appear to have any major effect on the scuffing problem. All testing to date was with case carburized valve plates and some authorities believe that re-lapping does not completely remove the damage. The damaged area was always in an area where a dowel pin hole existed on the opposite side of the valve plate, which locally removed half of the material thickness. Since spare hardened parts were not on the shelf, a bronze (CA670) valve plate was made, and the dowel locations were moved to another area, where they no longer should be a factor. The bronze valve plates also were damaged and repaired several times.

On the basis that this might still be a result of deflections, the entire module was assembled and installed for testing. After thirty minutes operation at 1000 rpm and pressures as high as 1800 psi, there was no sign of damage. The next thirty minute test (Table 2) at pressure under 500 psi and input speeds up to 2550 rpm resulted in no damage to the variable displacement unit operating as a pump (H_{VP}). The fixed displacement motor (H_{FM}) had worn very deeply into the brass valve plate, with brass particles embedded in the cylinder block. This damage was repaired, and both valve plates were modified by opening up all four timing ports in the H_F valve plate and two ports at the timing notches in the H_V valve plate by 5° . This should reduce "trapping" and possibly valve plate damage.

Another thirty minute test (Table 3) with pressures less than 250 psi and speeds up to 1500 rpm again resulted in damage to the H_{FM} valve plate. Investigation of all parts which might be restricting the cylinder barrel from seating squarely on the valve plate was initiated.

MODULE TEST DATA

DATE

2-24-78

TIME	INPUT		OUTPUT		PRESSURES			TEMPERATURE			FLOW		REMARKS				
	SPEED	TORQUE	SPEED	TORQUE	IN	MANIFOLD		CASE	IN	MANIFOLD		CASE	IN	OUT	N ₁	N ₂	N ₃
						LO	H ₁			LO	H ₁						
					50								.20		N _{IN} = 0		
1524	920	134	870	20	50	46	224	1.9					.25		89.25	15.8	14.1
1533	950	142	905	10	165	160	350	2.1	77	122	122	111	.58		89.9	7.5	67
	664	116	630	-15?	165	160	310	2.1					.58		89.5	0	0
1541	560	106	474	-10?	170	170	300	2.1	76	100	100	95	.55		89.5		
1545	1000	152	951	-10?	165	160	360	2	76	119	120	104	.56		89.8		
1548	1512	186	1445	15	165	150	395	2	76	143	144	123	.66		90.2	8.5	7.7
1553	2000	224	1718	15	160	145	439	2	77	159	160	141	.85		90.5	7.1	6.4
1557	2544	336	2431	15	150	137	475	3.8	77	162	161	155	1.16		90.2	4.7	4.3
5.3 min. * cooled to 132° between runs																	
Stopped to Engage dyno -																	
Weseloh suggested parting change - unit torn down for modifications																	
HV looks normal - HV valve plate had been worn very																	
deeply (draw) with brass particles embedded in cyl. block.																	
Valve plate will require grinding and lapping, & cyl. block will																	
require lapping prior to further testing.																	
{ HV Valve plate - all 4 timing parts opened up 5° (-0689 X 02)																	
{ HV Valve plate - two ports @ timing notches opened up 5° (-0687 X 02)																	
to prevent 'trapping' (i.e. insufficient area in notches to carry displaced Volume)																	

TABLE 2

MODULE TEST DATA

DATE

JUL 2-25-98

TIME	INPUT		OUTPUT		PRESSURES			TEMPERATURE			FLOW		REMARKS		
	SPEED	TORQUE	SPEED	TORQUE	IN	MANIFOLD LO HI	CASE	IN	MANIFOLD LO HI	CASE	IN	OUT	η_L	η_M	η_{HP}
1040	497	106	466	-10	55	72 220	1.5	70	91 91	88	.20				
1050	493	100	469	-10	54	63 217	1.5	71	95 95	91	.20				
1055	508	98	475	-15	55	63 205	1.5	72	97 96	93	.20				
1100	1002	136	941	+15	53	42 224	1.6	72	112 109	101	.23		88.6	11.7	10.4
1105	1037	124	974	-10	53	40 210	1.6	72	119 117	110	.25				
1110	1514	158	1433	+15	50	38 250	1.6	73	131 131	130	.33		89.3	10.1	9.0
30 min.															
Gearbox oil is 5 in. above gage blank over (not) restricting actuator motion - Swashplate as per drawing in actuator - Piston sticks in - 769 (SV ~ 2.770)															
$D_1/D_2 \sim .9669$ Ratio used on all previous complete module tests.															
print	η_L	η_M	η_{HP}												
1	0														
2				Re-lapped HF valve plate & manifold (w/ kgs installed)											
3				'Blued' shaft splines to observe wear patterns											
4	97.27	10.65	10.36	Installed new spring washers - (-0747-14-2)											
5				to ensure cyl. barrel motion is unrestricted											
6	98.03	9.17	8.99	Modified swash plate to provide full stroke (~844)											
				(with $S_F = .796+$)											

TABLE 3

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Cylinder barrel springs and washer seats were re-worked to provide additional clearance, and shaft splines were 'blued' so that wear patterns could be observed.

FUTURE TEST PLANS

Determine and correct cause of valve plate damage and excessive leakage rates. Run performance and base line noise data, then start noise reduction program.

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V. TASK 31

Compliant members have been manufactured and are being tested in three locations within the new co-axial module:

1. In the input and output couplings. Full power is transmitted through the compliant members.
2. The four mounting points for the module itself are isolated. The compliant members support the housing and also carry torque reactions.
3. Both rotating groups utilize compliant liners under the thrust plate against which the piston slippers run.

DuPont "Hytrel" is used to manufacture all of the compliant members. The Hytrel physical characteristics in general fall midway between those for rubber and plastic. The material has demonstrated good tolerance to the hot transmission fluid.

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VI. TASK 38

Oil aeration has been investigated and the following improvements have been incorporated.

1. Oil Passages. All passages in the auxiliary pump discharge circuits, including those for the connecting relief valves, have been increased to reduce oil velocities.
2. Gear Shrouds. Two gears run partially submerged in the transmission oil in the sump. Shrouds have been installed to prohibit churning.
3. Dual Inlet. A second inlet port has been added to the gerotor auxiliary pump to provide for filling the tooth mesh from both sides. Incomplete filling results in cavitation.
4. Lube Oil Pressure. Lube oil fed to the needle and roller bearings has been reduced from 30 to 5 psi. Reduced oil flow is significant since the oil is churned as it passes through each bearing.
5. Module Oil Shields. Shields have been added to the hydrostatic module so it will run 100% flooded. This allows for lower sump oil level in the transmission itself.