FORD/ERDA CONTINUOUSLY VARIABLE TRANSMISSION

PHASE I - TRANSMISSION DESIGN

Progress Report No. 9

For Period January 1-March 31, 1976

T. R. Stockton

Ford Motor Company ✓
New Concepts Research Department ✓
Scientific Laboratory and Research Building
20000 Rotunda Drive
Dearborn, Michigan 48121

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FORD/ERDA CONTINUOUSLY VARIABLE TRANSMISSION
PHASE I - TRANSMISSION DESIGN
FOR PERIOD JANUARY 1 - MARCH 31, 1976
CONTRACT E(11-1)-2674

A. Background

For many years, Ford Motor Company has been studying various types of Infinitely Variable (I.V.) Transmissions as one method for improving passenger car fuel economy. The Forster traction drive I.V. transmission appears attractive because it has potential for fuel economy improvement and would reduce the complexity and manufacturing costs associated with other traction drives by virtue of its unique ratio-changing mechanism. Although the Forster concept is yet to be proven in hardware, study of the concept justified a full engineering analysis. Contract E(11-1)-2674 was awarded with an effective date of June 15, 1975, to perform an evaluation of a continuously variable transmission.

B. Objectives

The primary objective of Phase I in Contract E(11-1)-2674 is to perform an analysis and design of the Forster concept, providing ERDA with an over-all transmission design layout, including controls, and test data on the traction drive components.

C. Progress during the Report Period

The revised program plan of Attachment I presents a general overview of the program status. As can be seen from this attachment, per ERDA request, the Phase I contract completion date was extended by two months, to June 30, 1976.

The Contractor's principal effort during this period, January 1 - March 31, 1976, was directed to the following four basic areas:

1. Design Analysis

Many geometric configurations of the disc were analyzed, using Structural Dynamics Research Corporation's finite element analysis computer program. The model used was an axi-symmetric disc with harmonic loading representing the rollers.

The initial design of the disc with 2.9" radius and .170" thickness was found to be very stiff to bend to give the required speed ratio of 3.5:1. Preliminary static tests on this disc verified this fact. The disc thickness was reduced to .105" in stages to reduce the stiffness but the bending stresses were high. The flat rims at the inside and outside diameters were eliminated to further reduce the stiffness. This helped in bringing down the maximum bending and principle stresses, but the magnitude of the stresses was still high.

1. Design Analysis (Continued)

Also, the disc would not give the required speed ratio coverage. The flat rims were designed back with slots in order to reduce the stiffness at the inner and outer rims. This was accomplished in the model by using a low modulus of elasticity in the circumferential direction. The improvement was not sufficient to get the most underdrive position. The stresses were found to be lower than the earlier models, with no slots at the rims. The stresses were still high at the contact point.

Tapering of the disc profile was tried, with a thicker center section and thinner section at the outside diameter. There was no significant improvement in obtaining the total speed ratio coverage. The bending stresses were found to be lower than those of the discs with constant thickness. A negative reaction force was observed at the outside diameter. Efforts to eliminate this negative reaction force were attempted in the following designs. The taper profile was changed to get the thinnest section at the most underdrive and overdrive contact points of the disc and thicker sections at the center and outside diameter of the disc.

A better angular displacement of the contact point was observed, but it was still short of reaching the required initial underdrive position. The negative reaction force was found at the outside diameter. The stresses were lower than the earlier models. Next, the disc profile was changed to keep the thinnest section (.050") between the most underdrive and overdrive position's contact points and the center of the disc, with larger sections (.100") at the center and O.D. of the disc. This attempt also fell short of reaching the extreme positions of the operation, with still higher bending and principle stresses.

A larger radius roller (6" Radius) with a flat disc is presently being analyzed. The speed ratio coverage was obtained but the stresses are still high at this time. A systemized computer analysis will be utilized to define the optimum profile geometry necessary to achieve the lowest possible bending and principle stress levels.

2. Performance and Fuel Economy Analysis

. W.O.T. Acceleration Performance Projection

Computer program has been modified to incorporate a ten-speed automatic transmission performance projection capability. It is hypothesized that an I.V. transmission W.O.T. performance can be simulated by a ten-speed automatic transmission which shifts at or near peak horsepower points. Tractive effort diagram (Attachment II) developed from computer results shows a comparison between the I.V. transmission with 4.25 axle ratio and a production three-speed automatic transmission for the

2. Performance and Fuel Economy Analysis (Continued)

. W.O.T. Acceleration Performance Projection (Continued)

Pinto with 2.8L engine. This chart was developed by assuming I.V. transmission efficiency to be the same as a three-speed automatic transmission. The I.V. transmission with 4.25 axle ratio has approximately the same tractive effort as the three-speed automatic transmission with 3.4 axle ratio.

. Fuel Economy

Computer program to calculate the optimum fuel economy for the desired routes has been completed. Required input consists of vehicle description, I.V. unit efficiency table, road loads, I.V. ratio range, engine fuel flow map, axle efficiency table, torque converter characteristics. Output consists of cycle fuel economy, modal fuel consumption, powertrain losses, second-by-second description of powertrain operating points.

The program breaks down the I.V. ratio range in finite gear steps. A comparison between ten steps and thirty steps showed that the thirty steps resulted in only a slight fuel economy improvement (.5 to 1%) over ten steps, with a considerable increase (150%) in computer cost.

Computer output would be utilized to design the control system, i.e., define engine operating line for maximum efficiency, and the effect of component size, i.e., a fuel economy comparison of the proposed 13" torque converter and a 13" torque converter scaled from the production 12" torque converter showed that the scaled-up version of the 12" torque converter would provide better fuel economy than the proposed torque converter. We intend to further investigate this aspect of our design.

We are attempting to determine the I.V. transmission efficiency in order that we may accurately predict the fuel economy potential of this unit.

Although our computer program uses a 2.8 liter engine map, a 2.3 liter or smaller engine is under consideration for use in this application.

3. Transmission Layout

A transmission layout has been completed, as illustrated in Attachment III. Several areas are undergoing re-design for package reduction and simplification. It is anticipated the torque converter will be reduced in size and the traction mechanism simplified.

The control system is presently being studied.

4. Dynamic Test Rig Evaluation of Traction Mechanism

The test rig components were inspected, assembled, and modified with necessary minor changes. Attachment IV illustrates the traction mechanism components, e. g., rollers, roller housing drive and driven disc, etc. The rig has been mounted to the dynamometer and is presently undergoing shakedown testing.

Attachment V illustrates the test rig installation and the rig separated at the traction mechanism. The actual test rig hydraulic circuit is illustrated in Attachment VI.

The format of all preliminary testing will include varying flex disc axial loads, torque, and rpm. The discs to be evaluated will consist of a 0.170" and .090" thick flex disc, several flat discs, and a combination of solid or constant ratio discs.

D. Meetings

Met with R. Kost and L. Forster on March 8, 1976, for general review.

E. Communications

The following communications were issued in this period:

Quarterly Progress Report No. 6 for Reporting Period
October 1 - December 31, 1975

Monthly Progress Report No. 7 for Reporting Period
January 1 - January 31, 1976

Monthly Progress Report No. 8 for Reporting Period
February 1 - February 29, 1976.

F. Financial Status

Contract Cost, Excluding Fee	\$370,920.00
Incurred Cost, through March 31, 1976	<u>287,130.00</u>
Balance as of March 31, 1976	\$ 83,790.00

FORD/ERDA CONTINUOUSLY VARIABLE TRANSMISSION STUDY
CONTRACT NO. E(11-1)-2674

SCRAM PLAN (REVISED)

1975 | 1976

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TRACTION EFFORT DIAGRAM
I.V. TRANSMISSION VS. 3-SPEED AUTOMATIC TRANSMISSION
(VEHICLE - 1975 PINTO, 2.8L ENGINE, 3090 LB. TEST WT.)

I. V. TRANSMISSION LAYOUT

A

DYNAMIC TEST RIG TRACTION MECHANISM

DYNAMIC TEST RIG INSTALLATION