Final Report

SRI Project No. ISC-1494

ANALYSIS AND REFINEMENT OF THE FULERTON AUTOMATIC TRANSMISSION – PHASE I

Prepared for:
Fullerton Design

Prepared by:

Steven H. Johnson
Research Engineer
Engineering Sciences Laboratory

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INTRODUCTION

This document contains the pertinent results and conclusions reached during SRI Project No. ISC-1494, “Analysis and Refinement of the Fullerton Automatic Transmission -- Phase I." The project, which dealt w1th an existing transmission design, was started on 15 September 1971 and was essentially completed on 4 November 1971.

Accompanying this report is one copy of a data package for the project. The data package contains the details of the mechanical analysis of the transmission, copies of relevant articles on the subject of traction drives, details of some fundamental tests, and manufacturer's literature on a candidate roller material. Although some comments are given in the data package, it should not be regarded as a separate document. Rather, it should be reviewed as a supplement to this report. Everything that SRI considers important is contained in this report; the data package gives the scientific basis for the statements made here.

Consistent with SRI's Proposal No. ISC 71-108, five general areas, were considered in the course of the project. Progress in each of these areas was reviewed with personnel of the Fullerton Transmission Company on 6 October 1971, midway through the project. Each area is treated below. Following these discussions, some general comments and recommendations for future work are presented.

II MECHANICAL ANALYSIS

The mechanical analysis in detail makes up the first section of the data package. Since it is quite technical, it will be most easily reviewed by those with a background in mechanical engineering. The analysis is broken into ten sections, labeled A through J, dealing with different aspects of the device. Transmission components are named in the first two drawings in the data package.

The purpose of the mechanical analysis was to gain detailed understanding of the operation of the transmission. The following points of interest were uncovered.

A. <u>Preload Force and Spring Selection</u>

To transmit the maximum input torque (35 ft-lbf, at a high gear ratio) without slip between the input or output disc and the rollers, a preload force of 80 lbf (assuming a static coefficient of friction of 0.6) between disc and roller at each point of contact will be required (Section E*). This force will act to keep the transmission in a ratio as far from 1:1 (either lower high) as possible. This situation can be corrected by incorporating two springs in opposition into the device, provided that the preload force is constant. The springs must balance the shifting moment due to the preload force and provide a squeezing force on the speed-governing balls. Note that the latter function is now accomplished by the main coil spring in the transmission. (A means for selecting the two springs to be used is given in Section A.)

The magnitude of the spring constants for two opposing springs for use in the prototype is quite large (~600 lbf/inch). The spring constant is a direct function of the roller preload force. Springs of this capacity should be available from commercial sources.

B. Transmission Speed Ratios

Since, for most materials, the static coefficient of friction is greater than the sliding (dynamic) coefficient of friction, the outward motion of the speed-governing balls will be stepwise rather than continuous. This means that a theoretical transmission will have a finite number of roller positions and hence speed ratios (Section B). Applying this analysis to the prototype transmission results in a predicted four-speed operation.

It is possible to increase the number of steps or transmission ratios significantly. This can be done by selecting material for use on the sliding disc and input disc such that the static coefficient of friction between that material and a steel ball is the same as (or approaches) the sliding coefficient of friction. Further, by intentionally making one of the ball contact surfaces steeper than the other, it would be possible to ensure that the ball will always roll on the steeper surface and slide on the other. Thus, frictional consideration would have to be given to only one contact surface.

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* The sections referred to in parenthesis are sections in the data package.

To realize static and sliding friction coefficients that are nearly the same, the critical surface or surfaces can be coated with one of several low-friction materials. Among these are molybdenum disulfide dry lubricant or a grease coating. Under these conditions, the transmission can approach an infinitely variable unit between its speed limits.

C. Output Speed

The output speed range available from the transmission is a function of both input speed range and geometry. Minimum output speed is dependent on the input speed required to shift the device out of neutral. Maximum output speed depends upon the extreme angular position of the rollers and upon maximum input speed. Once the unit is in its highest gear, output speed will continue to rise as long as input speed can be increased.

As an example, consider the prototype transmission. At an input speed of 1320 rpm, the unit will shift out of neutral and provide an output speed of 700 rpm. At an input speed of 1700 rpm, the device will move to its highest gear position, with an output speed of 3610 rpm. From these outputs speeds, an output speed ratio of 5.16:1 results. However, if the engine that delivers power to the transmission can continue to do so at higher input speeds, output speed will rise proportionately. If input speed reached 3500 rpm, for example, output speed would rise to 6920 rpm, yielding an output speed ratio of 9:1

D. Roller Contour and Elastic Modulus

In the initial prototype design, the contour of the roller was made to match the contour of the input and output discs at the point of contact. This will lead to significant slip within the contact area (Section D). The slip, in turn, will result in a power loss and heating at the interface and may also be the cause of the observed roller wobble. The magnitude of the power loss will be on the order of 1.9 hp for a nominal operating point of the prototype, provided that the required preload on the rollers is applied.

This loss can be greatly reduced by two means. First, the curvature of the roller should be greater than that of the toroidal surface of the input and output discs, so that only "point" contact will result when the preload is light. Second, a roller material with a high modulus of elasticity should be used, so that, under the required preload force, the width of the roller "footprint" (area of contact) will remain small in comparison to the roller width.

E. Roller Deformation

On the basis of a stress analysis of the transmission roller, the deformation of the roller (due to the preload force) at the point of contact is substantial (Section E), The deformation figure was obtained by assuming use of a urethane roller with a modulus of elasticity of $E \approx 10^4$ psi. The roller deformation can lead to large amounts of relative slip, as reported above, plus hysteresis losses in the urethane. These problems can be minimized through use of a roller material with a high modulus of elasticity.

F. Contact Stress

The contact stress between the governing balls and the sliding and input discs is likely to exceed the yield point of aluminum, which would cause local yielding of the material until the contact area was sufficiently enlarged (Section F). Some evidence of this was observable on the prototype unit; shiny scratches in the ball paths appeared. This problem can be alleviated in several ways. One solution is to make both discs of steel, which can support the contact stress. A second possibility is to make the curvature of the ball groove (a radial path), match that of the ball, so that a greater area exists to support the contact force.

G. Disc Loading

Due to the preload force on the rollers and outward force on the speed-governing balls, there is a tendency for the three discs (input, output, and sliding) to bend (Section G). Measurements taken on the prototype unit show that excessive deflection (≈13 mils) will occur at the input disc. This deflection can be greatly reduced either by using steel for the input disc or by making it thicker. The equations in Section G reflect the sensitivity of deflection to thickness and material. The other two discs will not deform appreciably, given the present design. When the deflection of the input disc has been reduced, no bending stress levels are expected to be high enough for concern.

H. Roller Center Pin

The bearing and the center pin at the center of the rollers were considered (Section. H). The load on this bearing, which is also transmitted to the pin, is about 180 lbf. By treating the center pin as a cantilever beam and using the dimensions that resulted from a design change during the project, it was found that at worst the center pin will deflect about 7 mils. This deflection is undes1rable, because it will result in displacement of the rollers with respect to the input and output discs. Since room to strengthen the center shaft is available, this approach should be used.

I. Sliding Collar Bearing

A bearing surface exists between the sliding collar (which controls roller tilt angle and turns at input speed) and the pivoted arms that extend to the rollers. Because of the presence of the preload force on the rollers and the resulting moment, a thrust force must pass through this bearing surface to the sliding collar. From the sliding collar, the force will be balanced by the proposed dual-spring arrangement. The maximum magnitude of the thrust force is 450 lbf, and the force can be applied in either direction, depending on the angle of the rollers. Thus, at this location a thrust bearing is required that can support a bi-directional load up to 450 lbf.

J. Bearings

A maximum load and a maximum speed for all the transmission bearings were determined, including those just discussed (Section 1). These data were compared with data showing the support capacity of inexpensive oil-impregnated bearing material (Oilite). In all cases, the capacity of Oilite was exceeded, indicating that rolling element bearings should be used. None of the imposed loads or speeds is too great for readily available commercial bearings.

There is one location where a bushing might be used: the space behind the neutral ring. However, a bushing could be used here only if the surface of the neural ring were at a slightly greater radius than the toroidal surface. This would relieve some of the preload force on the roller, so that the thrust load that must be supported would be reduced. This might also be advantageous in that it would partially unload one of the two opposing preload forces on the roller when the device was in neutral. The other preload force, applied to the roller by the output disc can be reduced when the transmission is in neutral by increasing the radius of the toroidal cross section at the outer diameter of the output-disc.

K Power Loss and Efficiency

The transmission was also considered from the standpoint of power loss and efficiency (Section J). Since all the critical bearings are likely to be rolling element bearings, they can be treated in a straightforward manner. Such bearings are typically quite efficient, and the power they consume increases in proportion to the transmitted power. There are two losses associated with the roller/disc contact area, however, that are much more difficult to analyze. The first of these is the loss due to slipping within the contact area, discussed above. This loss is <u>not</u> a function of the transmitted power, but only of the input and output disc speeds and the roller materials. Thus, to derive efficiency for the roller that accounts for slipping loss, one must assume an input power, a roller material, and a transmission operating point. This was done, with an input of 30 hp assumed. If a lower input power were assumed, the efficiency would be less, since the magnitude of the slipping loss remains constant. The second roller-associated loss is that resulting from rolling resistance. This power loss is a function of transmitted power, which facilitates the efficiency calculation. However, accurate determination of the efficiency is dependent on accurate knowledge of the coefficient of rolling resistance under actual operating conditions. Only a crude estimate of this coefficient can be made; an actual value would have to be determined experimentally. A conversation with the chief engineer at AEROL, Inc., in Los Angeles revealed that they had never determined the value of such a coefficient for any of their urethanes. Nevertheless, a value of this coefficient was assumed, on the basis of a subjective comparison of a rigid urethane rolling on metal with other material combinations.

Given this background, an overall transmission efficiency of 80 percent is predicted. Efficiency will be lower if the power input is lower since slipping loss is not a function to input power, as discussed above. In the computation of this figure, it was assumed that many of the suggestions contained in this report will be incorporated. Thus, the predicted efficiency should be considered an approachable goal. Given careful development and construction of the device the 80-percent figure may be reached or even be exceeded by a few points. At the same time, the assumptions concerning roller losses must be kept in mind. In particular is the coefficient of rolling resistance is higher than that assumed, or if the modulus of elasticity cannot be made high enough the total efficiency will drop. Further, cumulative errors in machine work, insufficient bearing lubrication, and numerous other details can erode the overall efficiency. Thus, it should be possible to fall short of the predicted efficiency figure.

L. Thermal Problems

Accompanying the power loss calculations was the prediction of the heat that will be produced by the rollers turning on the discs. This figure, again admittedly crude, is 500 watts per roller. Since the heat will be generated at the point of contact between roller and disc, and because the thermal conductivity and thermal mass of the discs are greater than those of the rollers, most of the heat generated may go into the discs. Whether this will happen is dependent on the heat transfer across the contact area. Unfortunately, prediction of thermal behavior in such a case as this is extremely difficult.

The best approach to the thermal problem appears to be an empirical one. If the unit is tested in a dynamometer, the actual total power loss can be determined. From this, the power loss associated with the rollers can be estimated. Simultaneously, temperature rise measurements, as can be made with Tempilac indicating lacquer, should be conducted. In this manner, the true nature of any heat rise problem can be defined.

III TRACTIVE FRICTION INVESTIGATION

The concept of using the traction of rolling elements for power transmission is not new. Numerous schemes have been tried with varying degrees of success; some of these have resulted in commercial products. The reason for using traction drives rather than gears, belts, and so forth are the infinitely variable speed ratios (within a range), high efficiency power transmission, and quiet operation.

A literature search was conducted as part of this project to gain familiarity with what has been done in the field of traction drives for power transmission. Both manufacturers' literature and technical article were surveyed. The second part of the data package contains a sampling of these articles. The results of this search can be summarized quite briefly. Basically, all the traction arrives that have met with reasonable success use all-metal components for power transmission. The actual working components are usually hardened steel finish ground to close tolerance. High quality bearings are used throughout. Several units provide high efficiency and reliable operation over acceptable lifetimes.

Use of all-metal traction elements requires that the power transmission be run wet, i.e., partially filled with a special fluid. This is necessary to prevent trauma to the metal surfaces, which would result if metal-to-metal contact were ever made. An important point here is that in all of the so-called metal-to-metal drives, metal-to-metal contact is ideally never made. Instead, power transmission takes place through the development of shear forces in extremely thin fluid layers.

The fact that these power transmission units are run wet has an important implication. The coefficient of traction (analogous to a coefficient of friction) for two metal surfaces separated by a fluid layer is very small. Even with special "high-traction" fluids, developed for these devices, a coefficient of ≈ 0.06 is usually all that is obtainable. Thus requires, then, that extremely high loading forces be used between the adjacent metal power-transmission elements.

Another characteristic of most commercia1 units is the incorporation of a load-sensing feedback mechanism, which serves to increase the loading forces between adjacent elements in response to increased torque load on the transmission. Thus, the preload is increased as required. The load feedback mechanism usually consists of a number of balls on inclined planes near the output element. When output torque increases, the balls move up on the ramps, thereby increasing the contact force between the output and its adjacent element.

Thus, existing traction drives are characterized by metal power transmission elements, immersion in special tractive fluids, load sensitivity, and high normal loads between elements. Usually they are expensive, because of the close tolerances required. Although there are departures from this general description, there appear to be no successful units available in the multiple-horsepower range that run dry and/or with nonmetal elements.

The Fullerton Transmission design, then, is a departure from existing traction drives, in that it is to run dry and is to rely on metal-plastic (or plastic-plastic) contact for power transmission. This means that a fluid film no longer plays a role, so the normal forces between adjacent elements can be drastically reduced. Further, problems of wet operation are eliminated such as sealing the unit against package and keeping the fluid clean.

At the same time several new considerations arise. For example: the power losses and heat generation cue to the plastic-metal contact will probably be greater than in all-metal units. Further, the wear and lifetime of the plastic parts must be contended with. (Note that, in the all-metal units. as long as the power-transmitting fluid film is never broken, metal-to-metal contact does not occur hence no wear occurs in the ideal case). As discussed above, accurate prediction of power losses and heat generation is difficult. The crude estimates of these that have been made indicate that neither will present an intolerable burden on transmission design or performance, it is even more difficult to predict how well the plastic will wear and consequently what the roller lifetime will be. So many variables are involved that suggestions as to how the roller might fall first and under what conditions would be groundless. A more realistic approach is to choose several roller materials on the basis of their quantitative and qualitative characteristics and to run them in a test transmission. Then, the basis of actual operating conditions and observations, power loss and lifetime figures can be predicted with some accuracy.

IV MATERIAL RECOMMENDATIONS

On the basis of the mechanical analysis and a general knowledge of operating conditions, the critical transmission components were identified. By far the most critical component is the roller itself; effort went into identifying candidate materials that would be most likely to meet the roller design requirements. It is understood that the roller material will be mounted on a metallic hub. The second component that received attention with respect to material selection is the disc (both input and output) against which the roller will turn.

The approach taken was first to specify the roller design requirements. Given these, general categories of plastic materials were examined to identify those of poss1ble use. When these categories had been identified, manufacturers were contacted for specific physical data. These representative data are reflected in the accompanying table, which includes data on both unfilled and filled plastics. It should be understood, however, that physical data for plastics can vary, owing to small differences in formulation or manufacturing techniques, so the figures given should be considered mean rather than exact values. Data pertaining to one specific material are included in the data package,

The design requirements specified for the roller material were:

- High friction coefficient, greater than 0.6
- High elastic modulus, greater than 10.000 psi
- High compressive strength, greater than 2.000 psi
- High abrasion resistance
- High tear strength (elastomers)

- Minimum creep, minimum compression set
- Low elastic hysteresis (to minimize heat generation)
- Low water absorption
- Usability over temperature range of -50 to 150°F
- Low cost and ease of manufacture.

Generally, in any type of plastic, the harder material, the lower the coefficient of friction. Frictional properties are markedly dependent upon surface finish and to some extent upon temperature. Many plastic materials are noted for their low coefficient of friction; examples are fluorocarbons, polyethylene's, and nylons. The frictional properties of plastics can be altered by adding either lubricating materials to reduce friction or friction particles to increase friction. Usually, the highest coefficient of friction is obtained with the plastic against itself. Therefore, facing the input and output discs with the plastic used on the roller rim should be considered.

The specified temperature range necessitates use of materials having a high heat distortion temperature and a very low brittle point. It is well known that the mechanical properties of plastics are much more dependent upon temperature than those of metals. Furthermore, all plastic materials exhibit some creep and compression set, particularly at elevated temperatures.

Considering all these factors, some filled plastics and other materials were identified as good candidates for use in the rollers. It became evident that no unfilled plastic would be a highly desirable roller material, primarily because a high coefficient of friction does not accompany an unfilled plastic with high elastic modulus. Several materials were also identified as seeming to satisfy all the criteria except the coefficient of friction.

The polyphenylene oxides satisfy nearly all the requirements for the particular application, but they are somewhat deficient in the friction property, Their excellent creep resistance and dimensional stability over a wide temperature range, together with cost considerations, make these materials a possible candidate for the roller application. The mechanical properties of the phenylene oxide polymers can be improved by reinforcement. Data on increasing the friction coefficient were not available.

The hard-cast polyurethane's are a second possibility among the unfilled plastics. Their friction coefficient is somewhat low. They also exhibit some hysteresis, which can lead to local heating. However, under the amount of deflection anticipated, the heat buildup should not to excessive for rigid polyurethane. The wear and abrasion resistance of these materials are very good.

The acetyl copolymers appear to be a third possibility in the unfilled plastic category. The friction coefficient is again deficient. These polymers have good creep resistance and excellent low-temperature strength. Glass reinforcement can enhance some of the properties of the acetyls.

The polyamides are also deficient in the friction property creep resistance of these materials (unfilled) may be excessive reinforcement can improve the creep resistance as well as some of the perties.

The polyamides again are deficient in the friction property. Also, these materials sometimes fail at low temperatures when subjected to high mechanical stress or cyclic fatigue.

The materials often classified as filled plastics, because certain polymeric resins are used as bonding agents, are mainly composed of other materials. The resins usually represent only a small percentage of their total volume (15 to 30 percent). The fillers may be cork, asbestos, metals, inorganic oxides, hard rubbers, fibers and so forth. Brake linings and clutch facings are some typical uses of filled plastics.

Among the materials falling in this category are the filled phenolics, which can be formulated to possess excellent mechanical properties and good friction coefficients. An example is given in the table. It is very likely that such materials could meet all the requirements specified for the transmission rollers. Basic resin suppliers (such as Union Carbide) do not prepare these materials. Specific information would be available from large manufacturers of brakes and clutches.

Other materials belonging to the resin-bonded category are the proprietary friction materials manufactured by Johns-Manville. Representative properties are included in the Table. These materials have the best friction coefficient of the materials studied, and their mechanical properties are excellent. Although their wear resistance is somewhat lower than that of the other materials listed, and they may abrade the contact surface more than the other materials, these materials appear to be the best selection for the roller application, on the basis of their desirable properties and their general past performance.

Bonding of the plastics to the metallic hub can be accomplished with all the materials listed. However, in the case of the Johns-Manville friction materials, it is recommended that attachment to the hub be made by flange bolting (compression of the material between two flanges).

The input and output discs must be of metal, because of the preload force they must impart to the rollers. Either aluminum or steel can be used. With steel, a thinner disc could be used for comparable strength, since its elastic modulus is three times as great as that of aluminum. Aluminum has a tendency to abrade more readily than steel. If the transmission tests show abrasion of aluminum discs, the contact surface should be anodized. Steel is likely to be the better choice for the disc material, because of its greater strength, stiffness, and abrasion resistance. However, a corrosion-resistant alloy or some means of surface protection would have to be used, since moisture may enter the transmission case.

Greater contact friction might be achieved by facing the discs with the same material used in the rollers. However, this step would unjustifiable at this time, since successful operation appears to be achievable without it.

To summarize the results of the materials study, the candidate roller materials are, in the order of preference:

(1) Johns-Manville friction material No. 160, or an equivalent. Due to the proprietary composition, this may be just & filled plastic similar to one of, the following categories.

(2) Resin-bonded filled phenolics.

(3) Polyphenylene oxide, filled if Possible.

(4) Hard-cast polyurethane.

(5) Acetyl polymers.

(6) Reinforced polyamides, if the friction coefficient can be increased through additives.

Steel input and output discs are recommended as superior to aluminum, on the basis of the information available at this time.

It is recommended that several of the roller materials be evaluated in a comprehensive test program. This is the only satisfactory way in which a final material selection can be made. The manufacturers of the described plastics provide design assistance services, which should be used to arrive at the roller design for a production transmission.

They can offer specific information on physical properties, manufacturing and bonding techniques, cost estimates, and design constraints.

V DESIGN CRITIQUE

At the beginning of the research effort, aspects of the transmission design would come worthy of comment. Many of these points have already been treated. Comments on additional topics of interest are now presented.

A. Roller Preload

In the prototype transmission, preload of the rollers is achieved via jam nuts on both the input and the output discs. The Fullerton Transmission Company and SRI agree that this preload method should be improved in future models. Two methods of supplying the preload force stand out as being simple and better than the jam nut technique. First, the discs could be spring-loaded against the rollers, thus maintaining a constant preload force. This scheme is simple, practical, easily implemented, and would permit roller wear to be automatically compensated for. However, constant preload would mean that roller-associated power losses (due to slippage with1n the contact area) would be constant, regardless of the power transmitted. This would decrease the efficiency at low power levels. Second, the preload force could be applied in proportion to the power transmitted., as by the ball and inclined plane technique described earlier. This would have the advantage of keeping efficiency high when power transmitted is low. Unfortunately, this scheme implies that the moment tending to tilt the roller would vary with torque load. This, in turn, would rule out the use of springs to balance this moment, and a much more elaborate moment-balancing solution would be required. Therefore, it is recommended that a constant preload be applied to the rollers via springs. Belleville washers could be used for this purpose. Although this would cause some loss of efficiency at low power levels, this solution is superior from an overall design point of view.

B. Disc Wear

It has been mentioned earlier that smooth transmission operation is desirable and can be attained by ensuring that many (or an infinite number of) shifting steps are realized; ways to achieve this have been given. If the roller is hard, as it must be, preferential grooves could be worn in the input and output discs if discrete speed ratios exist. Smooth operation via an increased number of shifting steps should minimize the possibility of this happening.

C. Roller Centering

When this project began, the magnitude of the loads to be applied to the bearing and shaft at the roller center was not real1zed. After these loads had been estimated, two conclusions could be drawn. First, the center pin supporting the bear1ng must be strengthened. One step in this direction has already been taken. Second, the center of the rollers must not be permitted to "float," **as** had been suggested earlier. Instead, the rollers must be fixed in place at the center of the toroidal cross section.

D. Other

In the prototype shown to SRI at the start of this project, the speed-governing balls often left the radial paths they were to run in. Further, the neutral ring turned with the input disc and did not idle properly. Both of these situations were remedied.

VI PRETESTING ASSISTANCE

The final area considered under this research effort was testing. Which is obviously crucial to further development of the device. In this case, SRI's primary purpose was to describe a practical test setup for the transmission, utilizing available equipment. Whenever possible. Two simple tests of no-load transmission losses were also conducted.

A trip to South Lake Tahoe revealed that a Hartzell Industries, Inc. Mark II Dynamometer is available to Fullerton Transmission Company. This unit is capable of absorbing up to 40 hp and so can be used as the output load on the transmission. Curves are supplied with the dynamometer, indicating delivered horsepower as a function of hydraulic pressure and shaft speed. No accuracy figures are given for the unit, but it is probably sufficiently accurate to be very useful in most development tests. The manufacturer should be should consulted to determine more clearly the accuracy of the dynamometer. Several possibilities for power absorption were considered, but since the Hartzell unit is available, apparently at no cost, it should be used.

The only required modification to the Mark II Dynamometer will be to put a chain sprocket on one of the shafts, which now carry rubber tires. Initially, this sprocket should have the same number of teeth as the output sprocket on the transmission. Sprockets with more teeth on the dynamometer shaft may be required if the input disc speed is much greater than 2000 rpm.

It is understood that engines are available that could supply power to the transmission. An engine that could supply up to 30 hp to the transmission would be useful. The engine and transmission input could be connected either via chain and sprockets or directly through the transmission shaft. The measure of transmission efficiency is not dependent upon the input or output connection.

The fourth part of the data package contains the detailed basis for making transmission efficiency measurements. Four measurements must be taken at any given operating point to permit an efficiency figure too be calculated: output power, 1nput and output shaft speeds, and reaction torque on the transmission case. Output power can be determined from the dynamometer. Input and output shaft speeds can be measured by several means, including a stroboscope, mechanical indicator, or electric generator. The data package contains product information on three possible tachometers. The Metron Model 29 is the least expensive ($50) but requires access to the end of each shaft whose speed is to be measured. The Metron Model 27 ($100) uses remote sensing of shaft speed, which is desirable if the shaft ends cannot be reached. This may be the ease with the output shaft. The most versatile unit is the Strobotac, but it is also the most expensive ($275). In addition to measuring shaft speeds accurately, it could also be used for stroboscopic observation of the rollers and speed-governing balls, which might be quite useful. If budget permits, this unit is the most desirable. These three possibilities are only a few of the commercial tachometers available.

Reaction torque is determined by measuring reaction force at a known distance from the transmission axis. At a radius of 1 foot from the axis the reaction force should not exceed 100 lbf. A search of available force gages turned up one candidate instrument, the Dillon Model X ($200). This unit is sufficiently accurate. To eliminate most tendencies toward jerky motion of the gage indicator, a block of firm rubber could be put between the transmission arm and the gage. This would damp out high-frequency force variations. Product literature on this is included in the data package.

Measurement of no-load losses in the prototype transmission was conducted on two different occasions. This was done simply by driving the input shaft, measuring the react1on torque, and letting the output sprocket turn free. On the first occasion, the rollers were 1n a minimum-ratio position; in the second, a 1:1 ratio was used. In. the second test, the prototype transmission had been upgraded by replacing several bushings with ball bearings; some parts had been strengthened; and new roller materials was used. Details and numerical results are presented in the data package.

Basically, the tests showed that no-load losses were relat1vely low for the given condition of the prototype. The maximum loss, for an input speed of 1370 rpm was 1.1 hp in the first test and 0.65 hp in the second. Losses increased in an expected fashion with increasing speed. It should be realized that losses will also increase with output load, which was zero in the tests. In both tests, the preload on the rollers was considerably less than that required; us1ng the recommended preload will increase the losses. However, no evidence was found to alter the overall effic1ency pred1ctions resulting from the mechanical analysis.

In each test, a local hot spot was found, but in both cases this was a readily correctable bearing problem. The roller surfaces became slightly warm to the touch.

One observation worthy of mention was that in both tests the rollers wobbled somewhat. This should be prevented by minimizing the bearing clearance at the roller center. Also, in both cases, the reaction force varied widely and rapidly with time at a given operating point. This variation in the reaction force is now thought to be associated with the wobble of the roller.

When the second test was conducted, the transmission was run once to demonstrate shifting via the governing balls. When the input shaft reached a speed between 775 and 1370 rpm, the rollers moved away from the neutral r1ng and brought the unit into "gear." The rollers seemed to shift quickly to a ratio greater than 1:1, and then slowly increase the ratio. It 1s believed that this slow shifting to higher ratios is a result of the unbalanced moment caused by even the light preload force. This is consistent with predictions. If a pair of compensating springs had been included, as they must be in future models, the

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