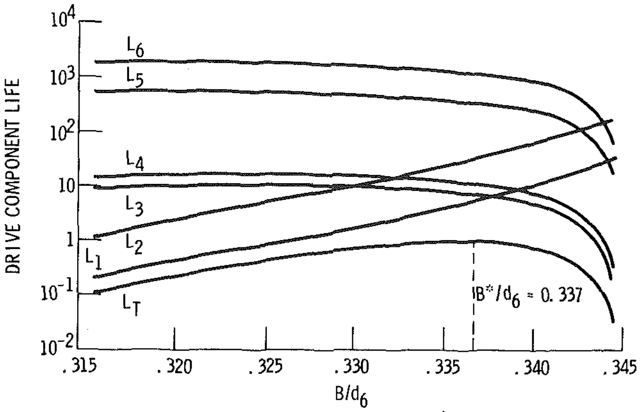

Fig. 8 Variation in drive component lives as a function of position on the ratio constraint boundary

in life as the geometry is changed from the optimum. Fig. 8 shows the variation in drive life and component lives with location on the ratio constraint boundary. The maximum life is achieved as a trade-off among the component lives. The sun and driven track of the first row rollers increase in life while the remaining components decrease in life as B and C increase. The loss of life for the smaller values of (B, C) was due to high stress on the high-speed sun element. The loss of life at the higher values of (B, C) was due to high stress caused by less effective toggle action and the resultant need for higher normal loads to preclude slip at the planet/planet and planet/ring contacts.

The multiroller traction drive reported in [5] had a $[B/d_6, C/d_6]$ = [0.322, 0.194]. The calculated drive life was 94 percent of the optimum.

Summary and Conclusions

A contact fatigue life analysis for multiroller traction drives has been presented. The life analysis takes account of stress, stressed volume, and depth of occurrence of the critical stress. The methodology is based on a modification of the Lundberg-Palmgren theory. It was assumed that the design coefficient of traction was 0.05. The analysis method was applied in a study of a Nasvytis multiroller traction drive having a nominal 15:1 drive ratio. The drive was approximately 25 centimeters (10 in.) in diameter with a main body length of approximately 11 centimeters (4.3 in.,). The following results were obtained:

1 The drive life was 2510 hours at a nominal continuous power and speed rating of 74.6 kW (100 hp) and 75,000 rpm. the maximum contact pressure was less than 1.4 GPa (200,000 psi) at all contacts for the above condition.

2 Life is dependent on geometry. There is an optimum geometry for longest life within certain design constraints among which are constant speed ratio and envelope size.

3 Life is proportinal to the inverse cube of torque.

4 Life is proportional to the 8.4 power of size.

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DISCUSSION

A. I. Tucker.[1] Although this paper is an in-depth analysis of a specific roller traction drive, the methodology used could also be usefully applied to other types of drive components such as gear trains, bearings, couplings, etc. A strong foundation was laid by the extensive use of known research and experimental data. This was brought together in a logical manner to determine both the limits of and the optimized size of a unique type of speed reducer using only roller traction with high surface loads.

Because surface compression is the critical stress in roller traction drives, this paper presents a thorough study of surface compressive stresses and the parameters which affect it. Particularly well presented are the proportionalities and effects of the parameters on each other. One example is the development of the probability of survival equation as a function of the magnitudes of critical stress, the stressed volume, number of cycles, and the depth of critical stress. In addition, the empirical values of the factors and exponents given are based on extensive tests of rolling element bearings carried out by previous investigators and which are well documented.

Another major strength of this paper is the extensive research and use of previous tests and investigations. The developed relationships between the important parameters are significant. The concepts and equations developed for rolling contact bearings and then applied to the multiroller drive are quite logical. It would also seem logical that the equations and relationships developed in this paper could be applied to other types of drive mechanisms where the critical stresses are

[1] Mem. ASME; San Diego, Calif.

surface compression. Additional applications could be explored, such as gear train lives and their probabilities of survival based on compressive stress levels.

B. K. Daniels.[2] It is not clear to this discussor if the life adjustment factor of 6 to 8 includes any allowance for the effect of traction. Has the factor of 6 to 8 been included in the prediction of a 2510 hr life?

A life of 2510 hr corresponds to 137,500 miles at 55 mph which may be satisfactory for a family car but how good is it for the likely applications of these drives?

Broad ranges of many parameters have been covered but this discussor is curious about the sensitivity of life to the design traction coefficient. Has this been studied? Could a somewhat higher value, say 0.07, be used in low speed and low temperature applications?

"Toggle action" is not described in detail. Is it correct to conclude that torque increases the normal loads only because of the roller ramp and that with a 90 deg ramp angle the normal loads would be independent of torque? Some readers might gain the impression that planet rollers become wedged between their neighbors so that torque levers the normal load in the same way that sheet materials are crushed on passing through cylindrical rollers.

All the above points are minor and this discussor would like to be among those who congratulate the authors on a clear and interesting analysis of a novel and exciting transmission.

Authors' Closure

The authors express their gratitude to Mr. Tucker and Dr. Daniels for their discussions. With regard to Mr. Tucker's comments, it is indeed true that the methods of this paper are general and may be applied to other mechanical drive components and systems. The basic life analysis methodology was applied to the toroidal traction drive [9] and spur and helical gears [11]. The authors and their colleagues at the NASA Lewis Research Center are continuing their work in this subject.

The life adjustment factor depends on many parameters which are not constant over the operating range of the transmission. These were looked at in detail and applied to the life calculations in reference [15]. To answer Dr. Daniel's question, the drive life of 2510 hr resulted after multiplying the theoretical life by the life adjustment factors. To add more detail, for the nominal condition at which the drive was rated at 2510 hr, the life adjustment factors were 7.95, 7.80, and 7.95 at the three respective contacts from the sun roller to the ring roller. Material and processing factors of 6, traction factors of 0.5, and lubrication factors of 2.65, 2.60, and 2.65 were used at the three respective contacts. As discussed in reference [15] there are few conclusive data for the effect of traction on fatigue life.

The 2510 hr life at a 90-percent survival rate corresponds to an average life of approximately 13,000 hr at a continuous 74.6 kw (100 hp) and 75000 rpm. This is quite sufficient for most applications. Of course, if a longer life is needed then the drive could be simply scaled up. In most cases only a small increase in size would be needed since life is proportional to the 8.4 power of size.

In order to prevent gross slipping at the traction contacts the design traction coefficient μ^* at each contact is selected to be less than the available traction coefficient of the lubricated contact μ. This is accomplished by choosing the proper ramp angle on the loading mechanism. This provides a proportionality between torque and normal load on each contact. Without a roller ramp, or in other words, if the ramp angle were made to be 90 deg (or 0 deg) then the normal load on the contacts would be constant, independent of torque. "Toggle action" will not alter this.

Toggle action is a property of the relative roller sizes and center locations and is independent of the roller/ramp configuration. "Loss of toggle action" is the result of loosing the force balance on the second row planet roller. This would result in an impractical design which would let the second row rollers drop between the first row rollers, and hence, the constraint line of Fig. 6 is defined.

Regarding the sensitivity of life to design traction coefficient, the life being inversely proportional to the cube of normal load would therefore be directly proportional to the cube of the traction coefficient. This relation is not strictly correct since the change in traction coefficient may have secondary effects on life. Provided that a realistic design is being considered (i.e. within the shaded region of Fig. 6) then it is possible that for lower speed and/or temperature applications an available traction coefficient of $\mu = 0.07$ or higher may exist. In this case if the design traction coefficient is increased to say $\mu^* = 0.06$ from the value of 0.05 used in the calculation then the new life would be determined by

$$\text{Life} = (2510)\left(\frac{0.05}{0.06}\right)^{-3}$$
$$= 4337 \text{ hr}.$$

[2] Monsanto Co.