There are three solutions to improve the performance of high speed couplings:

a. Cool the oil just before it enters the couplings.
b. Use a high viscosity oil in a separate lube circuit.
c. Use a very high viscosity lubricant in a lube packed coupling.

The first solution is the simplest and easiest to implement, but its advantages are marginal. The second solution is difficult to implement because it is not possible to prevent the oils from two different circuits from mixing. The best solution is evidently the third one. Not only does it provide the minimum wear rate and minimum friction coefficient, but a sealed coupling is not affected by adverse environments and does not accumulate sludge. Such a sealed coupling must have a large lubricant capacity, to insure trouble-free operation for a few years, and use a lubricant that is not adversely affected by the high centrifugal forces present in high-speed couplings.

The Influence of the Surface Hardness of the Teeth

This parameter was left intentionally last, because it is the only one that showed a strong interaction with others. To start with, the majority of high speed couplings are machined out of alloyed steels, through hardened and have the teeth surfaces nitrided. It is a known and accepted fact that case hardened teeth have a lower wear rate and lower friction coefficient than through hardened teeth. The work described in this paper was hence limited to analyzing whether an additional increase in the surface hardness of the teeth would further reduce the friction coefficient.

Usually high speed coupling teeth are nitrided, and the material used is AISI 4140 or equivalent. The surface hardness obtained through nitriding is about 50 Rockwell C. To obtain a high hardness, we had to use steels that respond better to nitriding, such as Nitralloy steels. We obtained surface hardnesses of 60 Rockwell C. As the graph from Fig. 8 illustrates, the additional surface hardness has practically no influence on the friction coefficient when the lubricant had a low viscosity, but it has a significant influence when a high viscosity lubricant is used. We assume that the changes in chemical composition did not contribute to the changes in friction coefficient.

Test Apparatus

The final configuration of the test rig was established using the experience we accumulated with rigs that we built and rebuilt over the years. The schematic of the test rig is shown in Fig. 9. A floating shaft arrangement with a relatively long span was used in order to facilitate the accurate measurement of the misalignment at each half-coupling. The coupling sleeves were attached at the shafts of two gearboxes in a "four-square" arrangement. The floating shaft was connected to the axial motion mechanism through a rod (inside the hollow shaft of the gear box), a small thrust bearing, and a load cell. The signal from the load cell was fed to one channel of a strip chart recorder. Another channel indicated the coupling travel, as given by a linear potentiometer.

For the curious researcher we would like to describe some details of the previous arrangements. One can notice that in the arrangement from Fig. 9, we are displacing *two hubs*. Is the thrust of two half-couplings *twice* the thrust of one half-coupling? To find this out we replaced one of the half-couplings with a metal diaphragm coupling for which we knew precisely the axial force versus displacement curve. Such arrangement proved difficult to use because the slightest error in measuring the displacement could result in large thrust variations, but we learned from it that the resistance to axial movement of a half-coupling is the same whether the power flows from the hub to the sleeve or from the sleeve to the hub. One can notice that in the test arrangement from Fig. 9, two half-couplings are moved axially at the same time. Thus, the axial thrust we were measuring was twice as large as the one seen in a normal coupling installation.

In most of the previous tests conducted by the author's company or by others, the motion was imposed to the floating shaft by a relatively large thrust bearing installed on the shaft. We observed that this large bearing influences the test results because it subjects the couplings to some unwanted radial loads, and because it consumes a sizeable amount of energy.

At very low axial velocities, direct readings are not practical, so that strip chart recorders were used.

Conclusions

Gear couplings, which were invented roughly 60 years ago, are still the most popular type of coupling. However, as they started to be used on high speed machinery, and use continuous oil flow lubrication, two of their weaknesses became apparent:

1. They tend to accumulate sludge from dirty lubricating oil through centrifugal action, and
2. They transmit an axial thrust, which in some instances taxes the capabilities of existing thrust bearings.

The results of the tests described in this paper indicate that: under the worst conditions, the friction coefficient can be as high as .16; but that with good lubrication, surface hardness, etc., the maximum friction coefficient (μ max) can be reduced to no more than .06.

For the calculations of thrust bearings in high speed machinery, API recommends the use of a friction coefficient in gear couplings of .25. This recommendation, hence, establishes, for couplings having μ max = .16, a safety factor of 1.5 for the calculations of thrust bearings.

Although a coupling can, under certain conditions, operate with friction coefficients lower than μ max, the author recommends that μ max be used for the calculation of thrust bearings. This recommendation is made because the torque at which μ max occurs is a function of misalignment, and one can never know how the misalignment changes with time.

The work described in this paper will hopefully help in obtaining a better understanding of the friction process in gear couplings, and provide the information needed to manufacture couplings which will transmit significantly lower axial thrusts, which in turn will permit the use of smaller thrust bearings.

References

1 Shipley, E. E., Report No. DF54TG601, Gear Engineering Division of General Electric, 1954.
2 Boylan, W., "Marine Application of Dental Couplings," Paper No. 26. *The Society of Naval Architects and Marine Engineers*, 1966.
3 Calistrat, M. M., "Wear and Lubrication of Gear Couplings," *Mechanical Engineering*, Oct. 1975.

Discussion

R. N. Brown.[1] The author should be congratulated on a clear and well written paper. He has done a fine job of explaining many of the apparent contradictions which appear in the literature to date. His explanation of Boylan's result is enlightening and removes apparent contradiction nicely, in fact adding a new ring of logic to the result presented by Boylan.

Several comments and questions do come to mind and hopefully the author will not consider these as other than constructive.

[1] Dow Chemical Co., Houston, Texas.

While quite a minor point, would it not be better in the introduction to refer to the couplings as flexible rather than flexible shaft couplings?

I would take exception to the statement in the paragraphs under *Influence of Misalignment;* coupling users are indeed interested in the range of the friction coefficient, not just the maximum value, for a number of reasons, but I believe the intent is recognized and accepted.

In the section *Influence of Axial Velocity* the sliding velocity is compared to the test axial velocity. The ratio is given as 60 times. Since hours are compared to seconds is not the ratio 3600?

Referring to the section *Influence of Teeth Geometry* the criteria cited is given, I would think, as a definition for relative comparison. The author does state the value is theoretical but is that enough would not a little more information on the assumptions for this type of value be in order so as to avoid misunderstanding - such as zero misalignment, no tooth crown, etc? The reason for concern is the pressure values given in *Influence of Contact Pressure*, are these ideal values as computed using equation (5). However, if the reader is not careful to keep the definition in mind, I believe in the way the paper is written a reader might be mislead to think actual operating pressures are at this value. Therefore, a reiteration or reference back to equation (5) might be helpful to recognize this as only an idealized relative value. It is recognized that determining the true contact pressure under actual operating conditions is difficult. The comment is made only in maintaining the proper perspective of such a low value.

In Fig. 3 the "A" shown is a torque and in the text is referenced as torque when referred to in *Influence of Misalignment* near the end of the section. Another letter would have been helpful since the nomenclature defines "A" as axial thrust.

The paper does not specifically state the range used for the test, but states that low viscosity causes the coupling to operate at a higher friction coefficient. As a solution to this the author suggests a high viscosity in a packed coupling. Does this not disregard the heat build up? If one accepts the heat build up which will surely be present, then is not the viscosity at the teeth truly a higher value? Does this not also ignore the many successful couplings operating on 150 SSU at 100 degrees F. oil in the field on a continuous flow system. Because of problems with friction it is not suggested that the continuous flow arrangement is anywhere near optimum, but the reader is somewhat frustrated in the best proposal to implement an improvement manifesting itself in a non-circulating packed coupling design. For reasons stated previous relative to heat build up, there are sealing problems if oil is used and separation problems if grease is used. Are there other alternaives available or must we consider other than gear couplings if three of more years of continuous reliable operation are desired?

Again, these comments are offered in the spirit of constructive discussions and should not deter from a well written paper on a good piece of research.

Author's Closure

The author appreciates Mr. Brown's kind comments on the paper. Mr. Brown raised a few valid points: the ratio between sliding velocity and axial velocity is 3,600 and not 60; the designation "A" in Fig. 3 is a poor choice.

It is true that Formula 5 gives an "idealized" value for the contact pressure between the teeth, however, the actual value does not differ significantly from the idealized value. High speed couplings operate at very low misalignments and the teeth usually have straight flanks (even when the flank is curved, the radius of curvature is very large and can be neglected).

A gear coupling generates heat due to the loss of some of the energy transmitted. It is expected that the efficiency of a coupling will increase as the friction coefficient decreases, and the problem of "heat build-up" to become a minor concern. It is possible, however, that those couplings which operate within an oil-tight enclosure could require some external cooling.

It is true that sealing a high viscosity oil in a coupling is a serious problem, and grease lubrication is the answer. Most commercial greases will separate into oil and thickener under the large centrifugal forces present in high speed couplings. Recently, special greases that will not separate at any *G* forces, and which are blended with very high viscosity oils, were made available.

Finally, serious consideration was given to Mr. Brown's statement: "Does this not also ignore the many successful couplings operating on 150 SSU at 100°F oil in the field on a continuous flow system." Although thousands of such couplings are currently in service, we have to accept that they suffer from two deficiencies: sludge accumulation, and thrust transmission to the connected machinery bearings. Because of these deficiencies, some users decided to forego the inherent advantages of gear type couplings, (such as low overhung moment, low weight and small diameter) and switch to non-lubricated couplings. It seems evident that a low friction, grease packed gear coupling would eliminate the deficiencies mentioned.

Discussion

R. B. Bossler, Jr.[1] The author has produced a worthwhile paper with strong penetration into areas where conjecture, opinion and highly specific test results have generated considerable confusion He has also shown that a low-friction regime exists where a given gear coupling will operate very efficiently, and have a long life. It is to be hoped that the author will produce design methods to identify the low-friction operating regime for a given coupling.

I find the title does not indicate the specific area addressed, which is the axial thrust transmitted by gear couplings. The test apparatus, test data and all the conclusions are directed to axial thrust. Also, "High-Speed" is not defined or shown to apply. However, it is obvious that high-speed is a reasonable inference because only very small misalignments are used.

The paper does not mention the transverse moment due to misalignment. The transverse moment may be important as a source of alternating stress to the rotating components and a steady load to the nonrotating support structure. The transverse moment is easily found; it is the coefficient of friction times the torque ($M = \mu T$).

My experience in this area offers an interesting cross-check of results. As a member of the SAE Committee S-12, Helicopter Powerplant, I was asked to prepare a document giving guidelines to the designers of turbine engines for helicopters which would result in a satisfactory drive shaft connection. The designer of the turbine engine knows his output torque only. He does not know what couplings will be used or what the misalignment requirements will be. It was necessary to assume gear couplings might be used. Analysis of well designed helicopter gear couplings found the pitch radius ($PD/2$), inches, to be between $0.07T^{1/3}$ to $0.10T^{1/3}$, where T is maximum engine output torque in pound-inches. The variation of the constant from 0.07 to 0.10 was due to the design practice of the various organizations. The value of the

[1] Western Gear Corporation; City of Industry, Calif.

maximum coefficient of friction, 0.10, was established by test results. Using the author's symbols, A = axial thrust = $\pm 2\mu T/PD = \pm 0.10T/0.07T^{1/3} = \pm 1.42T^{2/3}$, use $\pm 1.5T^{2/3}$ and transverse moment $M = \mu T = 0.10T$. These equations can be used by helicopter turbine engine designers to select their output shaft support bearings. They need only know their maximum output torque. They must apply appropriate safety margins.

Turning to Fig. 10 of the paper, the design torque at the test coupling was given to me by the author as 11,600 pound-inches. The pitch radius is defined as 1.3125 inches. Therefore, the design practice implied for this coupling is pitch radius = $0.06T^{1/3}$, which is in reasonably good agreement with the helicopter technology cited above. The teeth are hardened to R_c60 as in helicopter practice, so μ = 0.1 is a reasonable first guess. If we use the equation cited above, $A = 1.5T^{2/3} = (1.5)(11{,}600)^{2/3}$, the thrust bearing must withstand a force of 769 pounds to make one coupling slide and 1,537 pounds to make two couplings slide. The maximum thrust force on Fig. 10 is slightly over 1,500 pounds to make two couplings slide. I find this to be remarkable good agreement. Note that the largest thrust on Fig. 10 does not correspond with the largest torque. Friction defies logic once again. However, driveshaft bearings selected by the equation should be satisfactory, assuming appropriate safety margins were used by the designer.

The paper provokes additional questions. For instance, is the coefficient of friction affected by periodic shock-type axial motions, such as result when an air-cushion vehicle encounters periodic bow-wave impacts? What is the long-term survival character of high viscosity lubricants in gear couplings? Do some lubricant components migrate with time and exposure to their environment? As to design, what is the importance of surface finish? What effect would surface treatments such as silver-plate or black oxide have? Let us encourage the author to continue his work.

Author's Closure

The author is pleased to see that his test results coincide with Mr. Bossler's experience with helicopter couplings. It is unfortunate that Figure 10 could not be better reproduced in print. The dual trace strip-chart recorder used red ink for the force trace, which makes it barely visible in a black and white photograph. Actually, the largest thrust does correspond with the largest torque. Mr. Bossler opens so many avenues with the many questions at the end of his comments. Some will have to be answered by future tests; some can be answered now. For instance, the author's experience with specially blended greases, having very high viscosity oils, showed that a coupling can operate up to five years without the need for relubrication. The problem of lubricants components migrating, particularly when subjected to high centrifugal forces, prevented users from lubricating couplings with greases for many years. Today a few specially blended "coupling" greases are available.

Discussion

M. J. Gustafson[1]. "Lock-up" the term described here and under "Influence of Axial Velocity", "Influence of Time" apply to friction static slip conditions which is not generally experienced on Bell's spherical couplings. "The Mechanical Lock-up" is where male teeth wears a pocket in female couplings. This has occurred on Bell's nitrided Nitralloy N female couplings with nitrided M-50 tool steel male couplings. (In this case pockets were wide enough to provide axial motion, although limited.) In general, one thinks of lock-up as occurring at high misalignment angles when all backlash has been used up.

The value of .16 friction coefficient for adverse conditions sounds low. This number should be more like .25 and the term **apparent coefficient** of friction should be used because of all the variables involved. Certainly lubrication is a big influence. Coupling temperature should also be included because there is a temperature threshold at approx. 250°F where friction increases as the physical properties of the lubricant changes. (This value will vary with the lubricant.)

[1] Bell Helicopter Co., Fort Worth, Texas.

Introduction

I tend to agree with (*a*), (*b*), (*c*) with minor exceptions since the paper primarily deals with gear-type flexible couplings. Certainly many of the early attempts to measure friction provided a wide range of results particularly where high power was involved; however, better instrumentation and techniques have helped to remove scatter, but even so, a coupling driveshaft is still difficult for high speed/high torque couplings.

It is agreeable that torque, misalignment angle, and lubrication affect the value of the *apparent coefficient* of friction and will include tooth geometry if used in context that a misaligned condition male and female teeth are not conjugate.

General Aspect of Friction

Influence of Misalignment. Although a good start, 70′ of misalignment (certainly is better than zero to ten minutes), is still low for aircraft and helicopter applications where misalignment angles can be as high as 6°. This is contrary to the authors statement, "High speed couplings usually accommodate only a small amount of misalignment".

Influence of Geometry. Properly crowned with adequate backlash, a wide tooth can be lubricated as easily as a narrow tooth in high speed applications.

Influence of Lubrication. The author is right in using lube packed couplings for high speed. However, the benefits of E.P. additives were not even discussed. Bell's plot of apparent coefficient versus torque and misalignment angle could not have been run without E.P. additives and hardened teeth.

Author's Closure

It appears that a definition of the term "high speed" is in order. In industrial applications, high speed couplings can operate at peripheral speeds up to 750 feet/second. Helicopter couplings operate at much lower velocities and this is why they can operate at large misalignments. The author is grateful to Mr. Gustafson for making him better define the title of the paper, and he recognizes that the results described in the paper can hardly be used in couplings that operate at 6° misalignment and at high temperatures.