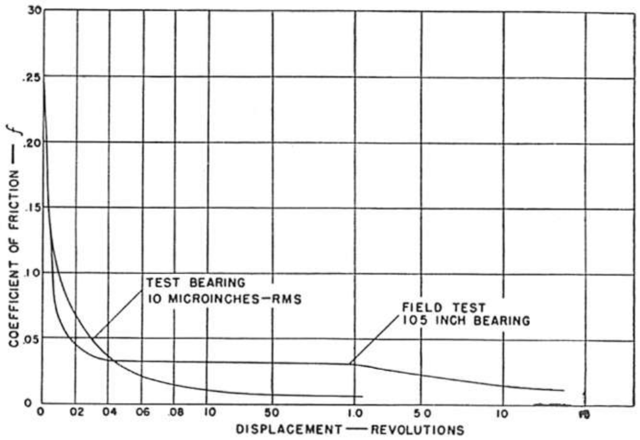

Fig. 20 Relation of Runner Displacement and Coefficient of Friction

characteristic number ZN/P, the values of the coefficient of friction obtained in the field correspond fairly well to the values obtained in laboratory tests.

For bearing runners having a surface roughness of 15 microinches rms or less, the oil film becomes partially established at a ZN/P value of 3. For bearing runners having a surface roughness of 45 microinches rms, the oil film becomes partially established at a ZN/P value of 6.

The speed N, at which a partial oil film is formed in function of the temperature, is shown in Fig. 21, for bearings tested in the field. Curves A and B correspond to bearing runners having surface roughnesses of 15 and 45 microinches, respectively, the same oil viscosity Z, and bearing pressure P. It will be seen that this speed increases rapidly with higher temperatures.

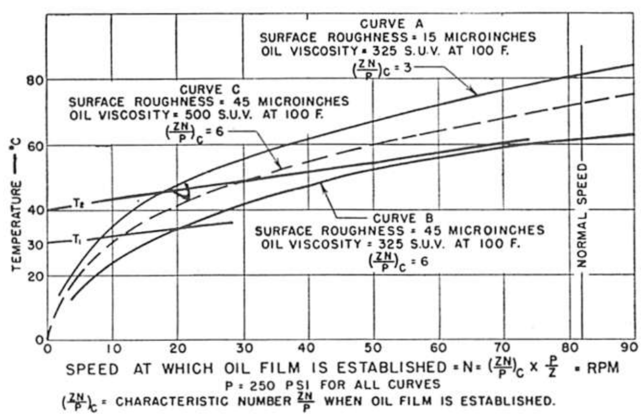

Fig. 21 Effect of Temperature and Bearing Finish on Reliability of Bearing Operation During Starting Period

During the starting period, there is an appreciable temperature rise of the bearing surfaces which results in a further increase of the speed at which the partial oil film is formed. The increase in bearing temperature is illustrated in Fig. 21, by curves T_1 and T_2 for starting temperatures of 30 C and 40 C, respectively.

The speed at which an oil film will form is then given by the intersection of the curves T_1 and T_2 with either of the curves A or B. For an inital starting temperature of 30 C, the partial oil film will be formed at a speed of 8 rpm for the 15-microinch rms runner and at 20 rpm for the 45-microinch rms runner. If the initial starting temperature is increased to 40 C, a partial oil film will not be established for the 45-microinch rms runner when normal speed is reached. If a heavier oil is used, as shown on curve C, corresponding also to a surface roughness of 45 microinches, a partial oil film will be established at the intersection of curves T_2 and C. This indicates that by using heavy oils and low starting temperatures, the part of the starting period where solid friction occurs will be kept to a minimum and the operating factor of safety of the bearing increased.

Conclusion

As a result of these laboratory and field tests, the minimum requirements for a satisfactory bearing surface have been established. With the profilometer and the microscope, it is now possible to control the machining of bearing surfaces so as to meet these conditions.

These tests have also shown the benefit of using oils containing oiliness agents and of establishing starting temperatures that will permit satisfactory operation of a thrust bearing during the starting period.

Acknowledgments

The authors wish to express their appreciation to Messrs. H. E. Criner, and E. H. Jernberg, formerly of the Westinghouse Electric Corporation, and to Mr. F. E. Kindig, for their contribution to the work described in this paper.

Discussion

S. J. Needs.[6] The starting of large thrust bearings in vertical hydroelectric machines is, perhaps, the most critical point in their operation. At the start, the high spots on the bearing surfaces presumably are in metal-to-metal contact. The load on the bearing is the dead weight of the rotating elements plus the hydraulic thrust necessary to start the rotor. Under certain conditions this hydraulic thrust may be maximum at breakaway and the bearing must be able to start under its heaviest load. Despite the heavy loading and the high friction of metallic contact, the initial rate of heat generation is not great because of the low speed during the first second or two of the start. Generally, a surface-separating oil film forms during a fraction of the first turn, the friction drops rapidly, and the bearing is off to a successful start. If, however, the oil film is incomplete and metallic contact persists as the speed increases, heat will be generated faster than it can be removed by the oil and the metals. Temperature distortion of the bearing surfaces will follow, aggravating metallic contact, and finally the melting point of the babbitt is reached and wiping occurs. Obviously, the success of the start depends upon the time required to establish complete separation of the bearing surfaces.

While it is not clear that the authors' experiments give any conclusive information on the time required for complete separation of the bearing surfaces, some striking measurements are given of the rapid decrease in friction from the instant of start. Fig. 22 of this discussion, drawn from the test-bearing curve in the authors' Fig. 20, shows friction coefficient as a percentage of the breakaway value as a point on the runner starts and moves 45 deg, the assumed angular length of the test shoe. It is seen that the friction has dropped to about one third of its initial value when the runner has moved only one tenth of a shoe length. When the runner has moved halfway across the shoe, the friction is only 8 per cent of its initial value. After the runner has traveled 45 deg, or one shoe length, the friction is but 4 per cent of the breakaway value. At the end of the first revolution, the friction has dropped to about 2.7 per cent, and from then on the drop is less rapid as normal operating conditions are approached.

[6] Service Manager, Kingsbury Machine Works, Inc., Philadelphia, Pa. Mem. A.S.M.E.

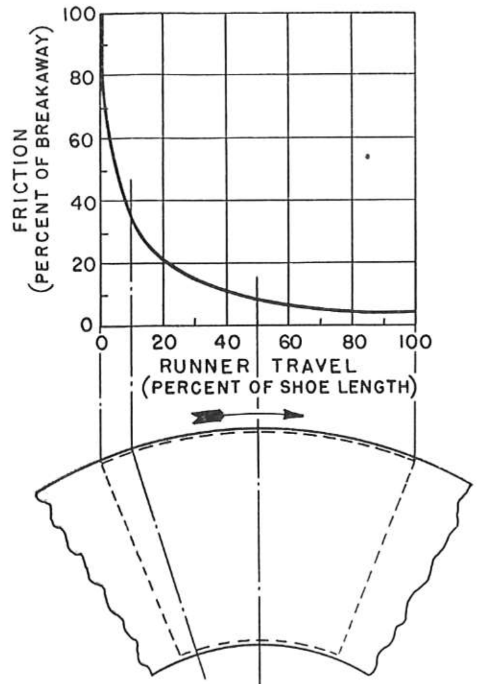

Fig. 22 Decrease in Friction After Breakaway
(Data from test bearing, authors' Fig. 20.)

Unfortunately, the foregoing gives us no clew as to just when complete separation of the bearing surfaces was established during the starting cycle. Laboratory tests of an air-lubricated thrust bearing with steel bearing surfaces would probably make it possible to measure the exact time at which the surfaces were definitely separated.

The fact that the friction drops to about one third of its initial value after the runner has traveled only one tenth of a shoe length is a most interesting basis for speculation. Assuming the runner to have carried in sufficient oil for at least partial lubrication of the first $^1/_{10}$ of shoe area, it is obvious that a great drop in friction has occurred in the remaining 90 per cent of shoe area, presumably still in at least partial metallic contact and certainly incapable of receiving lubricant from outside sources. Of course the bearing surfaces are not true planes and oil from the previous run will remain in the many depressions between contact areas. Also, adsorbed layers on the bearing surfaces will probably prevent true metallic contact of the same sort that would exist between perfectly clean dry plates.

With years of service the high spots on the shoes, due principally to initial hand scraping, will gradually wear down. Contact area thus will increase until practically the entire surface of the shoe becomes run-in and polished. There will then be no pools of oil between high spots to assist starting, and under these conditions initial lubrication evidently comes from adsorbed films on the bearing surfaces. The ability of boundary films to lubricate and to prevent metallic contact has been shown.[7]

Apparently the same or similar phenomena play a most important role in the starting of heavily loaded bearings.

Assuming the term "partial oil film" to mean complete separation of the bearing surfaces, the authors' estimate of from 8 to 20 rpm as the speed at which this film is formed agrees approximately with our belief. If the starting acceleration is 1 radian per sec per sec, a speed of 8 rpm will be reached by the time a point on the runner has moved about 20 deg from rest, and 20 rpm will be reached when the runner has turned about 125 deg. It is difficult to visualize complete separation of the surfaces until the runner has moved at least one shoe length, but the writer is definitely of the opinion that in the large vertical thrust bearings manufactured by his company, complete separation of bearing surfaces is accomplished during the first $^1/_4$ revolution.

An interesting example in point was recently encountered in a vertical machine with a 40-in. thrust bearing that had been in service for some 20 years. Weight of the rotating parts was about 75 tons, exclusive of hydraulic thrust. The machine had been shut down for 7 weeks, during air-housing repairs, and, at breakaway for the first start, the gates were shut and brakes applied. The rotor came to rest after $2^1/_2$ revolutions. Air was kept in the brake cylinders for more than 2 min. Total upward thrust of the brakes was not in excess of 10 tons. When the brakes were released the rotor started to turn slowly, driven by leakage through the closed gates. Since the turbine had been rebuilt the previous year, the leakage was not excessive. Apparently complete oil films of sufficient thickness to survive a stop of at least 2 min had been formed during the first $2^1/_2$ revolutions of the thrust-bearing runner.

It has been observed that when runner finish is not up to some undetermined standard, the finish will be improved by running-in and the starting torque will drop. When the runner surface is initially smooth, however, there is a gradual increase in starting torque as the surfaces become run in. The authors' Fig. 7 brings this out. Data in Fig. 23 of this discussion, obtained from a newly installed 93-in. bearing, show an increase in starting torque with succeeding starts. Lacking a method of measuring starting torque, it is assumed that for small openings of the turbine gates the torque will be proportional to the opening. The gate opening for each start is plotted and shows a marked increase with the number of starts. The bearing was initially assembled with grease on the shoes and the machine started practically as soon as the gates were cracked. The runs lasted from a few minutes to 24 hr or longer. Length of the stop seemed to have no effect on the starting torque after about 15 min of rest.

The first run lasted more than $1^1/_2$ hr, and it is felt that practi-

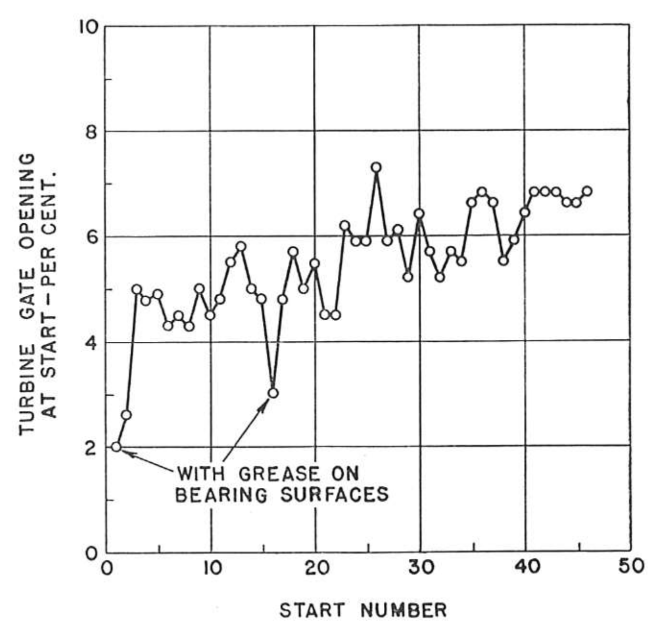

Fig. 23 Increase in Starting Friction With Number of Starts
(New 93-in-diam vertical thrust bearing.)

[7] "Boundary Film Investigations," by S. J. Needs, Trans. A.S.M.E. vol. 62, 1940, p. 331.

cally all traces of grease were washed from the bearing surfaces during the first run. After 15 starts, the bearing was inspected and given a second coating of grease. Gate opening for each normal start continued to increase until a final value between 7 and 8 per cent was reached. This has remained constant during the life of the machine which, at this writing, is about 3 years.

During the first 20 starts of this bearing, the breakaway was inaudible. When the required gate opening for the start reached about 6 per cent, however, a faint grunt, due to metallic vibration at breakaway, could be detected. This noise gradually increased with the starting-gate opening and a short but plainly audible groan now accompanies each start.

With reference to starting torque, our experiênce with the use of so called "oiliness agents" in lubricating oil has not been as successful as shown in the authors' Fig. 15. Recent laboratory measurements of friction coefficient at breakaway, range from 0.172 with pure oleic acid to 0.214 with a general-utility turbine oil, such as used in the thrust-bearing housings of some of the largest hydroelectric machines. Addition of 2 per cent of sperm oil to the latter reduced the coefficient to 0.204. Greater sperm-oil content caused the coefficient to rise. Apparently there is little to be gained with oiliness agents on babbitt-bearing surfaces.

The starting behavior of the 105-in. bearing, Fig. 20 of the paper, is appreciably different from that of the test bearing. After a sharp drop in friction during the first $^1/_{10}$ revolution here is little if any drop during the remainder of the first turn. If the assumed acceleration persists, the speed will be about 34 rpm at the end of the first turn. When 10 revolutions have been made the friction coefficient is given as 0.015. Under normal operating conditions, the final friction coefficient of this bearing should be of the order of $^1/_{10}$ to $^1/_{15}$ of this value; hence it is not clear that the surfaces are definitely separated even after 10 revolutions. The authors have stated that partial oil films are formed at from 8 to 20 rpm. With this in mind, their comments on the apparently erratic behavior of this bearing will help clarify the general picture.

In so far as the writer knows, this is the first paper dealing with the important and most interesting subject of the starting of thrust bearings under load. For a long time the question has been in the mind of practically everyone concerned with hydroelectric machines, and the authors are indeed to be congratulated on bringing some really useful information to light.

Authors' Closure

Mr. Needs must be complimented for his very interesting discussion. The presentation of his experience with the performance of thrust bearings during the starting period is a very valuable contribution.

The authors are aware that the mechanics involved in the formation of the oil film during the starting period has not been satisfactorily explained. The rapid decrease in the coefficient of friction seems to indicate that each of the many contact areas between the pad and the runner acts as a small land-type or fixed-wedge thrust bearing which operates with the oil trapped between the two surfaces, and establishes immediately a partial oil film. Thus, as suggested by Mr. Needs, after the high spots have been worn out, the oil film would be established less rapidly as appears to be the case with the 84 in. bearing of Fig. 16 which had been operating more than three years before the test was made. The 64 in. and 105 in. bearings of the same figure had been operating two weeks and eight months respectively.

The authors used the term partial oil film to describe the period after which the coefficient of friction has been reduced to a fraction of the breakaway value but is still much higher than the final coefficient of friction corresponding to a perfect oil film. The apparent erratic behavior of the 105 in. bearing is believed to be due to the method of applying the strain gages which were intended only to be used for determining the coefficient of friction at or immediately after the breakaway. In order to measure the coefficient of friction with a perfect oil film, a more elaborate mounting of the gages would have been required, but was not believed to be necessary since the principal object of these laboratory and field tests was to establish the quality of finish required for the satisfactory performance of a thrust bearing. It was found that there is a definite relation between the breakaway coefficient of friction and the surface roughness as measured with the profilometer. With the smoother finishes, lower and more uniform values of the breakaway coefficient of friction are obtained.