Shaft Whipping Due to Oil Action in Journal Bearings

By Dr. B. L. Newkirk and H. D. Taylor
Research Laboratory, General Electric Company

Shaft whipping is a form of vibration or whirling developed under certain conditions in shafts running above their critical speeds. The authors have produced this phenomenon with models, and have traced the cause to the action of the oil film in the bearings. They find that the phenomenon when produced in this way does not occur at running speeds lower than about double the critical speed of the rotor. To prevent trouble from this source the authors suggest that very low unit bearing pressures be avoided or that a friction-damped spring be used. A previous investigation reported in this magazine dealt with a similar phenomenon caused by internal friction of the rotor. The article by A. L. Kimball in this issue details the measurement of the internal friction of shafts, which also has been found to be a cause of whipping.—Editor.

Introduction

A previous article[1] described an investigation of a troublesome form of shaft vibration or whirling termed shaft whipping, which, it was found, was supported by the cramping action of long sleeves or hubs on the shafts of rotors running above critical speed. In the present article an account will be given of later tests which showed that whipping of much the same nature is caused, under certain conditions, by the action of oil in journal bearings.

Nature of Whipping

Whipping in general may be defined as resonant vibration or whirling[2] of the shaft of a machine when running at other than the resonant or critical speed. The vibrations encountered at the critical speed are synchronous with the revolutions, and are caused by the deflecting forces due to unbalance; also, under ordinary conditions, vibrations of this nature reach a maximum at the critical speed and outside of a narrow range of speed disappear almost entirely. Whipping, however, has been found to occur over wide ranges of speed, the vibration or whirling frequency bearing little if any relation to the number of revolutions per minute, and the phenomenon is practically independent of balance.

Whipping Due to Cramping Fits

In the case of the whipping caused by the cramping action of long fits on a shaft, the whipping did not occur at speeds below the critical speed, but it did develop at any higher speed. The motion was found to be an elliptical or circular whirl of the shaft in the same direction as the rotation, with a frequency about equal to the critical speed. This action did not usually develop of itself, but required a jar, or other stimulus which would deflect the shaft, to start it. Quantitative tests carried out with small but nicely built models running in ball bearings demonstrated that whipping of this nature was rendered much more difficult to produce—i.e., less likely to develop—by reducing the length of the fits or by making them tighter (as by increasing shrink or press fit allowances); but that it was possible to cause whipping even with very short, tight fits by extreme shock, causing a large deflection of the shaft. Without an extreme stimulus to start the whipping, however, these models would run at any speed, up to 10,000 r.p.m.—the highest speed studied—without appreciable vibration of any kind save that occurring at critical speed (about 1950 r.p.m.) when the models were thrown out of balance. A remedy for the commercial machines showing a tendency to whip was found in the use of spring-supported bearings with frictional damping imposed. An attempt to stop the whipping in one case by correcting the only recognized source of the trouble—i.e., by shortening the fits on the shaft—was unsuccessful. This in itself aroused a suspicion that there was another source of whipping; but in the absence of quantitative data on fits, such a conclusion was not considered established at that time. Furthermore, the spring bearings were entirely successful, not in concealing the whipping, but in stopping it altogether.

Indication of a New Source of Whipping

In the light of these results, two models were built, for study of shaft behavior in the critical speed range, with short and extremely tight fitting wheel hubs which we felt sure could cause no whipping unless the shaft was given a dangerous deflection while running; —and the possibility of this was prevented

(1) Shaft Whipping, by Dr. B. L. Newkirk, General Electric Review, March 1924, p. 169.
(2) Whirling is viewed as a composite of two vibrations in planes 90 deg. apart and with a phase difference of one-quarter period.

by the use of a guard ring, with limited clearance over the shaft. The rotors of these models weighed 400 lb. each, and ran in journal bearings. They came fully up to expectation as to behavior at critical speed; but they regularly developed pronounced whipping when the speed was carried higher than about twice critical speed.

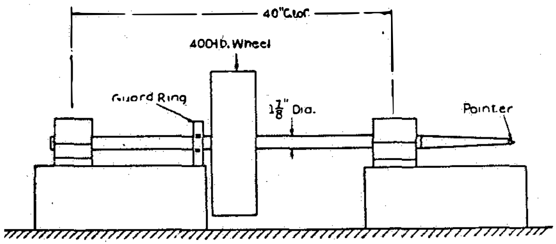


Fig. 1. Model Used in the Study of Shaft Whipping

The set-up of the models is indicated in Fig. 1. The rotor, known as Model IX, consisted of a 1⅞-in. shaft carrying a 400-lb. wheel, running in two journal bearings 40 in. apart, center to center. The bearings were of usual oil-cooled design, 1⅞ by 2½ in. with a 0.003 to 0.004-in. clearance (difference of diameters) over the journal. A pointer, shown in the figure as a tapered extension of the shaft, was attached at one end and the point ground to run truly, so that any vibration or whirling of the shaft produced a corresponding motion of the tip of the pointer which could be observed conveniently in strong light with a low power microscope. The critical speed of this model was about 1210 r.p.m., the pointer indicating an elliptical whirling at this speed, of amplitude approximately proportional to the degree of unbalance of the model.

The whipping showed itself as a similar whirling of the shaft when the speed rose above 2300 r.p.m. Motion pictures were taken of this pointer, the exposures being timed so as to catch a complete whirl on each picture. Some s mples from these records are shown in Fig. 2, from which it can be seen that the motion was an *open whirl*. By means of an oscillograph and magnetic "exploring coils" it was determined that the rate of whirling was 1205 to 1280 per minute (very nearly the critical speed frequency), at speeds from 2300 to 5000 r.p.m;—also that the whirling took the same direction as the rotation of the shaft.

Tests to Determine the Cause of the Whipping

From the start, the journal bearings were suspected of being responsible for the whipping. Indeed, there was little else in the model to suspect. The wheel fit was so tight that the whipping could hardly be due to "working" of the fit. Three factors apart from the journal bearings, however, were tested as possible sources of the trouble: (1) the belt drive; (2) the degree of unbalance; and (3) the thrust-collars on the shaft. The tests eliminated these possibilities. The model whipped when coasting with the belt removed just as it did when running steadily or accelerating with the belt on. The balance was refined until the model could be run at critical speed with less than half a mil vibration at the middle of the shaft, and still it whipped very much as it had when considerably out of balance. The thrust collars were backed off and the rotor held in position by blocks of soft carbon clamped at the ends of the shaft, but this had no noticeable effect on the whipping.

The first positive indication of the cause of this phenomenon was the accidental discovery that the whipping was stopped immediately *by shutting off the oil* supply to the bearings, and could be brought back to full amplitude promptly by turning the oil on again.

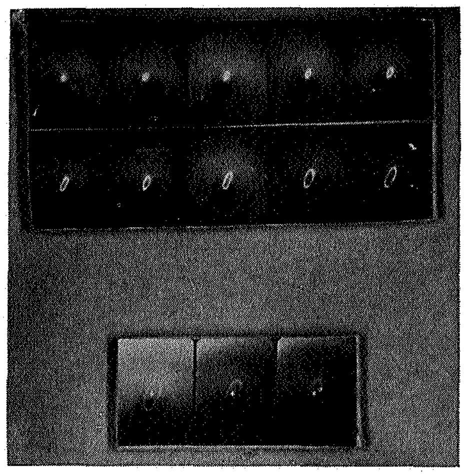
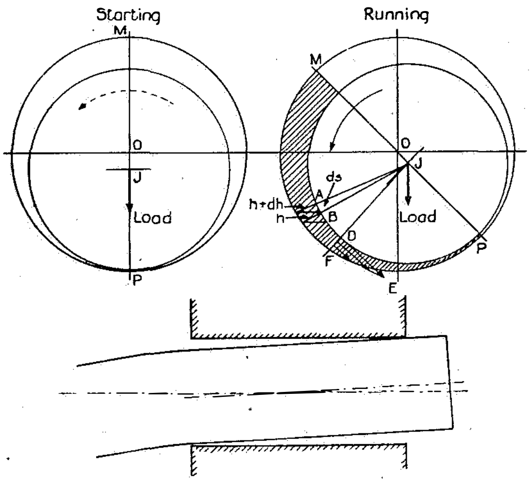


Fig. 2. Motion Picture Records of Whipping of Model IX, Due to Oil Action

Top: Building up of whip at 3000 r.p.m.—good balance
Bottom: Double exposure whirl centered about steady position

The use of one friction-damped spring bearing was found also to stop the whipping. The friction was essential, however, as it was found that a special spring bearing designed so as to avoid inherent damping permitted violent whipping.

The solid sleeve bearings first used with this model were rebabbitted several times in the course of tests on vibration at critical speed. In every set-up, however, the whipping developed, this indicating that the phenomenon was not due to accidental irregularities in the bearing lining. Another model rotor, No. XI, was also available, which ran in the same bearings. It had a stiffer shaft and a critical speed of about 1950 r.p.m., but was otherwise similar to Model IX. When run



Figs. 3 and 4. Displacement of Journal in Bearing by Oil Action

Fig. 5. Diagonal Position of Journal in Bearing in Well Developed Whipping

at 3600 r.p.m. or faster, this model developed whipping very similar in character to that of Model IX. This served to indicate that the whipping could not be considered due to accidental peculiarities of Model IX.

It had been thought that the whipping might be due in part to misalignment of bearings. This point was difficult to test with the older sleeve bearings, but was settled by the use of a new set of self-aligning bearings with spherical seats. It was found that intentional misalignment to any marked degree *prevented the whipping*, but that when the ball seated bearings were permitted to align themselves the models whipped quite as violently as they had when running in the sleeve bearings.

These results indicated that some action of the oil in the journal bearings was responsible for the whipping and in consequence we have called it the "Oil Whip," to distinguish it from whipping due to cramping fits.

Theory of Oil Action Causing Whipping

It is a familiar fact, established by the experiments of Tower, that in a well lubricated journal bearing carrying load the journal is displaced somewhat from the bottom of the bearing, as is indicated in Figs. 3 and 4. The action is as follows: Oil clings to the rotating journal, and is carried along with it; at points around the bearings where the clearance is diminishing, oil is crowded in and builds up a considerable pressure (indicated by shading on the diagram); on the other side of the bearing, where the clearance is increasing, the oil pressure is very low. This results in a net difference of pressure which moves the journal forward, so that its center J takes up a position relative to the bearing center O, somewhat as shown.

It seemed that the "Oil Whip" might be due to this kind of action in the oil film, the journal center not finding a position of rest, as at J, but moving around the bearing center O, propelled by a pressure zone behind the point of nearest approach, P. It was recognized, however, that the action of the fully developed whip was more complicated than this, because the amplitude was so great as to cause the journal to take a diagonal position along the length of the bearing, as indicated in Fig. 5. But it was thought that substantially the same action took place at the ends of the bearing, with the pressure zones diagonally opposite at the two ends.

This theory would explain how the oil film can produce a whirling motion of a journal, and also account for the fact that the observed whipping took the same direction as the rotation. As given, however, it does not explain why the whipping was confined to a speed range above about twice critical speed. Further tests were therefore made to obtain additional information on the nature of the oil action.

Confirmation and Extension of Theory by Tests with Special Model. Description of Model XIIA

A simpler model was built for further study of the oil whip. In order to avoid dealing with two journal bearings either or both of which might be involved in the whipping, the new model was equipped with only one journal bearing, and one ball bearing. For further simplification and symmetry, the model was arranged to run with the shaft vertical, so that there was no static load on the journal bearings. Finally, for ease of manipulation, the model was made quite small, and with a low critical speed (800 r.p.m.). The set-up was made as shown in Fig. 6. The weight of the rotor, about 50 lb., was carried at the top by the self-aligning ball bearing. Four 1½ in. by 2 in. interchangeable journal bearings were provided, which were duplicates except for the clearance over the journal. The ball seats were carefully fitted. The bearings were each made in one piece and bushed with brass. Oil was fed in at the middle under pressure, distributed by a single lengthwise groove, and allowed to escape from the ends of the bearing.

Behavior of Model with Shaft Vertical

Fig. 7 indicates in a descriptive way the behavior of the new model when set with the shaft vertical, representing a run with a large clearance in the journal bearing. The critical speed vibrations are shown as a small peak in the curve at about 800 r.p.m. A much larger peak of vibration was found at 1520 r.p.m., a little less than double the critical speed. Whipping started at 2250 r.p.m., growing more violent with further increase of speed. It was noted at this time that the quantity of oil discharged from the ends of the bearing was greatly increased when whipping developed. These results were obtained with the journal bearing properly aligned, but restrained by the pinch of the ball seat. When this restraint was removed by loosening the bearing cap bolts, the model would not whip, but developed the same two peaks of vibration—one at critical speed, and one at about twice critical speed. Moderate restraint of the bearing resulted in moderate whipping, accompanied by some motion of the bearing on its ball seat.

Close observation of the journal and journal bearing revealed two phenomena which confirmed the theory of the oil action and led to an extension of it which explained the peak of vibration at double critical speed, and made possible a more satisfactory explanation of the whipping. The first was found with the model running at low speed, 200 to 300 r.p.m., with a large clearance (0.008 in.) in the journal bearing and plentiful oil supply. The journal took up a whirling motion in its clearance, traveling slowly around in the same direction as the rotation—according to the simple theory of oil action in bearings. This was not whipping. The shaft did not bend, but tilted around like a conical pendulum, pivoting at the top on the self-aligning ball bearing. The motion was slow and easily followed by the eye. It occurred regardless of whether the ball seat of the journal bearing was restrained or not. But it died out or failed to develop when the oil supply was shut off. This "journal whirl"

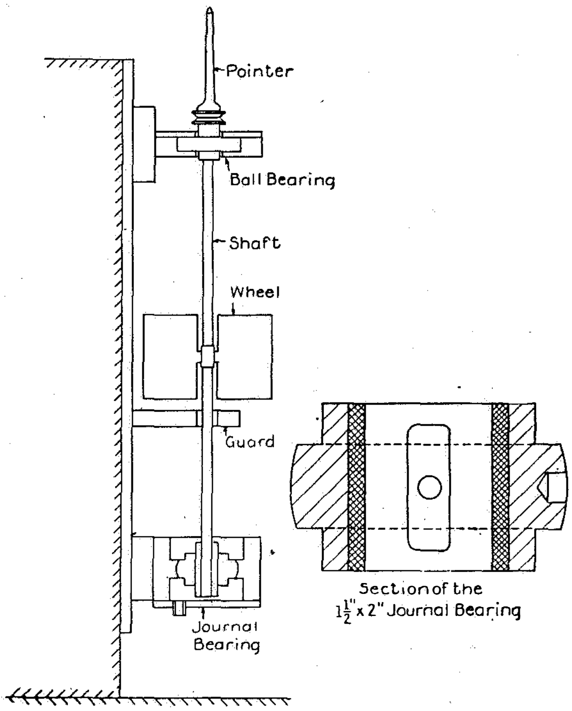


Fig. 6. Model XIIA Set with Shaft Vertica

was found to continue to higher speeds and then appeared to be faster, though not nearly so fast as the revolutions. It was found that the frequency of this journal whirl was very nearly one-half the shaft r.p.m.; i.e., the journal made one circuit of the bearing in approximately two revolutions. This was sufficient to explain the peak of vibration at about twice critical speed; for the frequency of the journal whirl would then correspond to the natural vibration frequency of the shaft, and the shaft would be expected to respond in resonance to this stimulus. This peak was therefore named the "Oil Resonance" peak.[3]

Another phenomenon was found when running at speeds higher than the oil reso-

(3) It is a fact, though perhaps not generally recognized, that a rotating shaft will respond in resonant vibration to a suitable stimulus in practically the same manner as though it were not rotating. Thus the rotation and the resonant vibration are superposed without mutual interference. Similar resonance peaks at other speeds have been found in operating very sensitive models, due to eccentric tachometer pulleys and even to resonance with the rotation of the ball cage in ball bearings.

nance speed with the journal bearing free to tilt. It will be remembered that no whipping developed under these conditions. The journal bearing, however, took up a wabbling motion which was found to be another case of conical whirling, but with the ball seat as pivot. The amplitude was small, and the frequency rather high, so that direct observation was more difficult than in the case of the journal whirl. A small post attached to the bearing and projecting up parallel with the shaft served to magnify the motion. It was now plainly seen that the direction of the whirl was forward—with the revolutions—and oscillograms were taken which showed that this frequency also was equal approximately to one-half the r.p.m. This "bearing whirl" stopped when the oil supply was shut off, and also if the bearing cap were drawn tight. With plenty of oil and a tight bearing cap, whipping developed if the speed exceeded 2250 r.p.m.; below this speed and above the oil resonance, both shaft and bearing were quiet. If, in this range, however, the bearing cap were drawn up very gradually, the bearing motion continued for a time and a critical adjustment was found in which the shaft responded in resonant vibration. It seemed that a slight restraint of the bearing imposed some friction on its tilting motion and retarded its frequency; and that in the critical adjustment the frequency had been reduced to the resonant value.

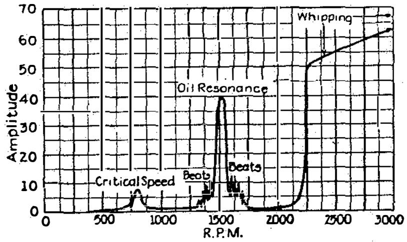


Fig. 7. Behavior of Model XIIA Set as Shown in Fig. 6

The effect of friction on the journal whirl was then investigated, and it was found that its frequency also was reduced quite appreciably by even such slight friction as that of a dial indicator bearing on the shaft.

Extension of the Theory of Oil Action

These discoveries led to an extension of the theory of oil action. Referring to Fig. 4, consider the distribution of velocity in the oil film. The layer next to the journal has the full journal velocity, as represented by the arrow DE. The outside layer, next to the stationary bearing surface, has zero velocity (point F.) In the intervening space, the oil velocity is somewhere between these extremes, and the simplest estimate would consider the distribution of velocity represented by a straight line, EF, through the tips of the arrows. The average oil velocity, then, across any radial section of the clearance, should be about half the surface speed of the journal. The rate of oil flow past any radial section should be proportional to the local clearance; much less oil, then, can pass the point of nearest approach, P, than is carried past the opposite side at M, where the oil space is widest. Assuming plentiful oil supply, this means that oil is pumped into the shaded space faster than it can escape circumferentially past the point P, and that a considerable pressure will be built up and the excess oil squeezed out axially, unless the journal or the bearing yields to this impulse and moves around, opening up space for the oil as fast as it is required. The rapidity of such motion may be estimated as follows:

Let

h = radial clearance at any section, say at B on the shaded side of the bearing;

$h+dh$ = radial clearance at section A, a small distance, ds, toward M from B.

Now, while the journal revolves through the distance ds from A to B, a volume of oil equal to $\frac{h}{2}\,ds$ is carried past B (assuming unit width of oil film, perpendicular to the paper), while at the same time a volume equal to $\frac{h+dh}{2}\,ds$ moves past section A, bringing an excess of oil equal to $\frac{dh}{2}\,ds$ to section B. This excess oil can be accommodated if the clearance at B is increased by the amount $\frac{dh}{2}$. Now if the journal as a whole revolves around the bearing center O through an angle equal to AJB, the clearance at B will increase, becoming equal to $h+dh$, the clearance at A previous to this motion. Hence, if the clearance at A is to increase to $\frac{h+dh}{2}$, the journal must revolve about O about half as far, or through half the angle AJB. It will be seen that this or a similar analysis would hold at any point around the bearing; and that, in general, space will be adjusted to the local oil supply if the journal center J moves around O with about one-half the angular velocity of shaft rotation, or if the journal runs steadily, and the bearing yields to the oil at the same speed.

It may be concluded further that, if friction is present to resist the journal or bearing motion; i.e., if considerable work is required of the oil action, the oil pressure required will be higher, resulting in additional end leakage and a reduced frequency. The latter has been illustrated with the bearing whirl and journal whirl and an increased end leakage of oil noted when the model whipped violently.

In the horizontal setting, the oil resonance peak was not observed, for it apparently merged with the whipping. The journal whirl, however, developed very much as it did when the shaft was vertical. The bearing, no longer being free of restraint, failed to develop the whirling motion, but moved more or less irregularly on its spherical seat when the cap was loose.

Explanation of the Oil Whip

These considerations led to the conclusion that the oil whip was due to the type of oil

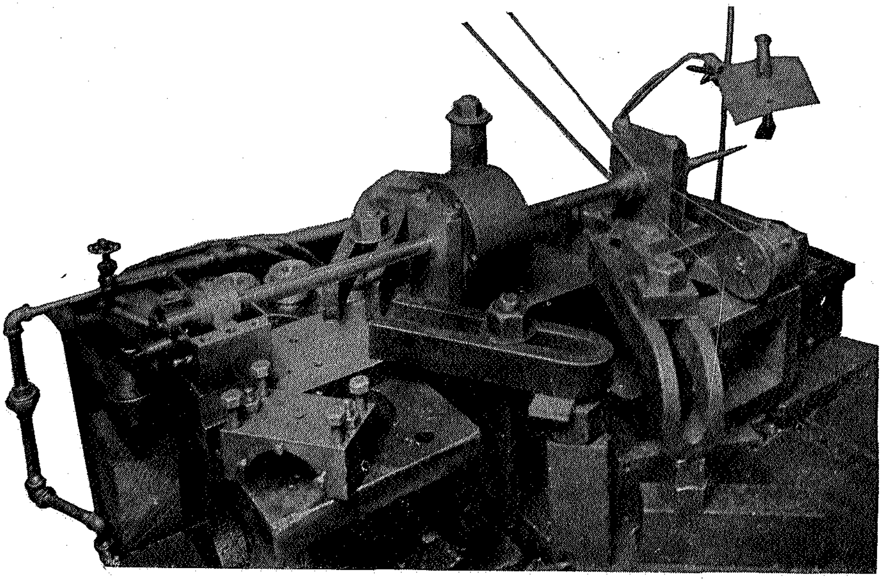


Fig. 8. Model XIIC Set With Shaft Horizontal

Behavior of Model XIIA with Shaft Horizontal

The model was set horizontally, with no other change, as in Fig. 8. Two differences in its whipping behavior were then noted: (1) whipping started not at 2250 r.p.m. but at the oil resonance speed of 1520, continuing to all higher speeds; and (2) whipping persisted even with the bearing cap loose, though it appeared less vigorous than with the cap tight. It had been noted when the shaft was vertical that partial restraint of the bearing had caused moderate whipping. Therefore, since the bearing carried load in the horizontal setting it was subject to partial restraint even with the cap loose and it was not surprising that the model whipped. The discrepancy between the starting points of whipping in the vertical and horizontal positions was thought to be due to the damping effect of the end thrust on the ball-bearing in the vertical position, as explained on page 565.

action that has been described. It seems that this oil action may occur at speeds below the oil resonance speed (about twice critical speed) but that its frequency is then too low to produce resonant vibration or whipping of the shaft. At the oil resonance, the shaft responds to its natural frequency and, as the speed is increased still further, one might expect the frequency of the oil action to increase, the resonance to be lost, and the shaft to run quietly at all higher speeds. This actually occurred with Model XIIA set vertically with the bearing cap loose. But with the model horizontal, the resonant vibration continued from the oil resonance to all higher speeds. That is, the shaft failed to "pull through" the oil resonance. The reason seems to be that the oil action is reduced in frequency when it meets with frictional resistance. The maintenance of whirling vibration requires the overcoming of frictional resistance, which probably increases sharply with the amplitude on account of the restraint of the bearing. It appears that this increased resistance holds down the oil action to the resonant frequency, so that it maintains the vibration even though the rotational speed is raised above that corresponding to the simple oil resonance speed. Also, since the oil is required to do more work, higher oil pressures are developed in the film, and the leakage of oil by the ends of the bearing is increased, as was observed. This explanation of the whipping makes it appear that if it were possible to quiet the shaft when whipping above the oil resonance speed, it would then run smoothly; that the oil action, if it occurred at all thereafter, would have a frequency too high to make the shaft whip. This seems entirely possible. In fact, in a few tests, Model XIIA after being quieted in this way would run smoothly unless disturbed; but a slight shock was sufficient to start it whipping violently again. In general, the oil whip would start itself when the speed and other conditions were right, with no special shock to initiate vibration.

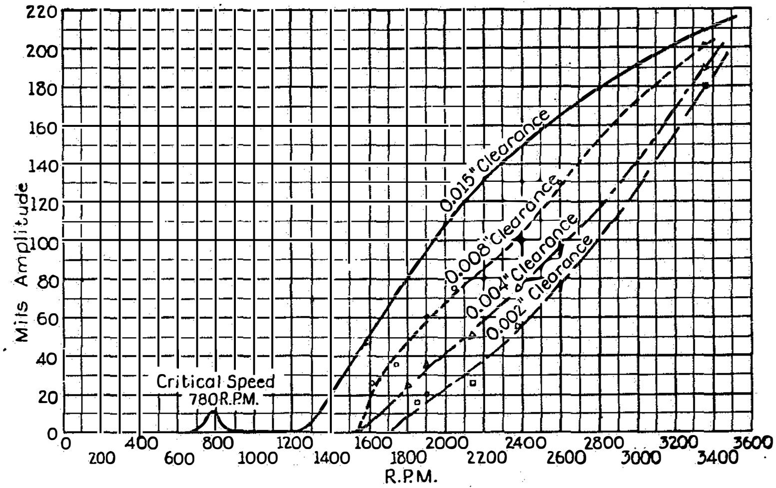


Fig. 9. Effect of Clearance on Whipping of Model XIIB
Shaft horizontal
Bearings 1 in. by 3 in.

Observations and theory agree that the tendency to whip is weakest at speeds just above the oil resonance, and that it increases with increase of speed. It seems reasonable, therefore, that if the damping resistance is large enough, whipping should not develop until the speed is raised considerably above the oil resonance. In the case of Model XIIA set vertically, the end-thrust on the ball bearing may have caused enough damping to prevent the whip building up below 2250 r.p.m. Setting the model horizontally relieved this end-thrust, and the model then whipped at any speed above the oil resonance. The application of a little extra damping in the horizontal position, however, by means of a carbon ring closely fitted at the middle of the shaft and held by spring pressure between two stationary plates, caused behavior corresponding to that observed in the vertical position. Whipping failed to develop except at speeds well above the oil resonance, and the heavier the damping pressure, the higher was the speed required to start whipping.

This explanation of the "oil whip" takes no account of the effect of gravity, which produces an unsymmetrical condition in the bearing when the shaft is horizontal. The minor differences in behavior of Model XIIA in the horizontal and vertical positions may be due partly to the effect of gravity in the horizontal position, though it seems likely that one cause of these differences was the damping effect of the ball bearing which served as thrust bearing in the vertical position, as previously suggested. The generally satisfactory correspondence between the theory and the observed behavior of this model in both positions, however, seems to indicate that the gravity effect was not large enough in this case to warrant special consideration. Later tests, to be described, indicated that in more heavily loaded bearings the effect of gravity may be so large as to suppress the whipping action entirely.

Tests with Various Special Journal Bearings

After the nature of the oil whip was recognized a number of tests were undertaken to investigate whether some modification of the journal bearing would not break up the oil action and prevent whipping. A preliminary change of this sort involved the oil feed arrangement: instead of feeding oil from a lengthwise groove along one side of the bearing, an annular groove was cut in the middle, so that the bearing surface was divided into two unbroken symmetrical parts

This was merely a step toward the next change and was not expected to have much effect on the whipping. Test with Model XIIA set vertically showed that the whipping was more pronounced than before. The same bearing was altered by the cutting of six lengthwise slots in each end of the bearing, which it was thought might serve to prevent the oil action responsible for whipping. This expectation was only partly realized; the whipping was not eliminated but decidedly weakened. It

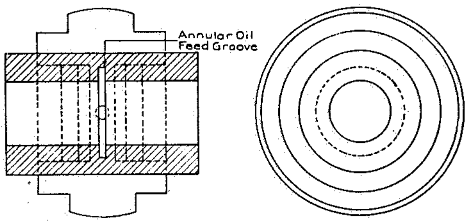


Fig. 10. Proportions of Journal Bearings Used With Model XIIC

was necessary to run to higher speeds before whipping developed at all, and then its violence was considerably reduced. Trial was next made of a very short bearing with annular oil feed. This change was more effective than the slots, but also failed to eliminate the whipping entirely. Later, with this model set horizontal, two other modifications were tried. A plain bearing with annular feed was relieved in the middle over most of its length, so that there remained only a narrow band of bearing surface at each end. Pronounced whipping developed in operation with this bearing, particularly at high speed, though with rather less violence than with the standard bearing. As compared with the short bearing test, this seemed to indicate some advantage in reducing the length of bearings—even independently of the loading pressure. Another test was made using a bearing, with helical grooves machined in the bearing surface, with right- and left-hand pitch on opposite sides of the annular feed groove at the center. Rotation in one direction pumped oil rapidly out at the ends of the bearing, and permitted some whipping which became rather vigorous at high speed. The model would not whip when running in the opposite direction, but this was not considered of importance because the oil flow was practically stopped due to the pumping action opposing the supply pressure. These qualitative tests indicated several means for weakening whipping, but none of these was satisfactory for completely preventing it in the model.

Further Tests on the Effect of Bearing Proportions, Etc.

Effect of Bearing Clearance

Model XIIA had been at first equipped with a journal so large as to be out of all proportion to the size of the shaft; this was turned down and reground to a 1-in. diameter, and the bearings bushed accordingly, 3-in. long and with clearances of 2, 4, 8 and 15 mils. The oil was fed through an annular groove. With these changes, the model was designated as Model XIIB. The curves in Fig. 9 show the effect of varying the amount of bearing clearance on the behavior of the model. As the clearance was reduced, the whipping started at higher and higher speeds, but in all cases grew rapidly in violence with further increase of speed.

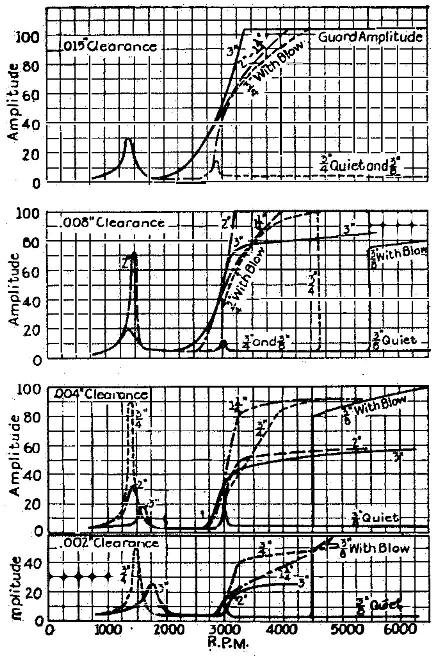


Fig. 11. "Oil Whip" Tests With Model XIIC
Bearing 1 in. dia., length given by figures on curves

Effect of Reducing Bearing Length

It had been intended to test with the same model the effect of using different lengths of bearing by cutting down the original 3-in. lengths to 2 in., $1\frac{1}{4}$ in., etc. Only the first series of these tests was completed (with the 3-in. lengths), because the severe vibration encountered at high speeds fatigued the model shaft, and in the first run with a 1-in. by 2-in. bearing, the shaft broke.

It was decided not to duplicate the old ⅝-in. shaft which had been made of soft steel, but to use a 1-in. chrome-nickel steel shaft which was already available. The same wheel, bearings, etc., were used, the only other changes being the attachment of a new pointer, and a further separation of the bearings. The new combination was designated as Model XIIC, and the set-up is shown in the photograph, Fig. 8. The stiffer shaft caused an increase in the critical speed from 800 to 1400–1770 r.p.m., depending on the clearance.

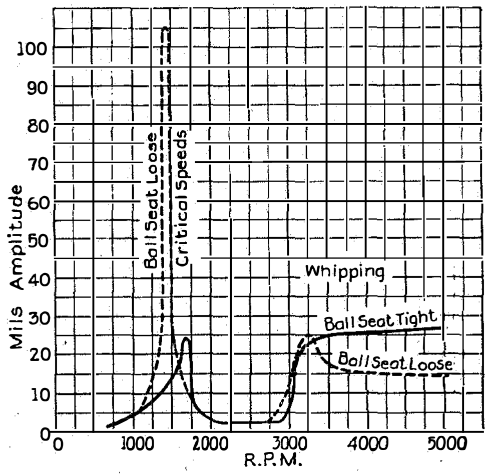


Fig. 12. Effect of Ball Seat Looseness on Vibration Characteristics

An extended series of tests was made with this model, the length of the bearings being cut successively from 3 in. to 2 in., 1¼ in., ¾ in. and ⅜ in., as indicated in Fig. 10. The results of these tests are shown in Fig. 11. The most interesting result is the fact that whipping developed in every run but one: the shortest bearing, ⅜-in. long, with the largest clearance, 0.015 in., failing to support whipping. All the other combinations whipped to some degree. It was noted that the shorter bearings produced less vigorous whipping, though in many cases whipping of greater amplitude. The distinction is that a "vigorous" whip builds up quickly to a definite amplitude when the rotor, running at a whipping speed, is quieted so that it runs smoothly, and is then released. Whipping which builds up vigorously may fail to reach large amplitude because of the restraint of a long bearing having small clearance. A shorter bearing, however, under like circumstances, builds up the whip more slowly but to larger amplitude. The amplitudes reached by the whipping, therefore, are not comparative indications of the intensity of the stimulus. It will be noticed that with bearings of 1¼-in. to 3-in. lengths, which included all of the combinations which produced the most vigorous whipping, the whipping started at almost exactly twice critical speed (about 3000 r.p.m.); and further, that the amplitude of whipping never decreased as the speed rose, although the speed was carried in most of the tests to 6500 r.p.m. In every test, also, the shaft was quieted at various speeds, but it almost invariably broke out quickly into violent whipping again when released.

Check on Frequency of Whipping

During the above series of tests, there were some indications that the frequency of the whipping increased with the speed. It was decided to check this point by taking oscillograms. A bearing was rebushed 1 in. by 2 in. with 0.004-in. clearance, and some films taken with the results shown in Table I.

TABLE I

R.P.M.	Whip Frequency	Amplitude
3070	1440	0.040 in.
3200	1523	0.060 in.
3420	1590	0.070 in.
3960	1640	0.085 in.

The lowest frequency, 1440, corresponds very closely to the observed critical speed for this set-up, 1430 r.p.m., at which the amplitude reach 0.030 in. The increase of frequency with the speed is thought to be due to the increase of amplitude, on account of the stiffening of the shaft by the bearing which tends to increase the critical speed. This restraint apparently accounts for the "high criticals" noted with long, tight bearings, which reached as high as 1770 r.p.m. (see Fig. 11). This would also account for the fact that in some cases the whipping was observed to start somewhat below twice critical speed, the critical being observed with larger amplitude and higher frequency than the start of the whipping.

Effect of Looseness of Ball Seats

In general, the whipping with the bearing cap loose was somewhat weaker than that experienced with the cap tight. Loosening of the bearing caps is not a procedure to be recommended, however, for the reason that it permits a much greater degree of vibration (due to unbalance at critical speed) than occurs when the ball seat is tight. An example from test is shown in Fig. 12. This point was also tested out with the heavier model, No. IX. With the same degree of unbalance, the amplitude reached at critical speed was 2 to 3 times as great with the bearing caps loose as with them tight.

Effect of Varying Bearing Loading Independently of Length

In order to determine the effect of increased loading, two additional 200-lb. wheels were made to shrink on the shaft of Model XI. Without these, the model weighed about 400 lb., and the bearing loading was about 43 lb. per sq. in. of projected area; violent whipping had developed at speeds of 3600 r.p.m. and up; the critical speed was then 1970. With the shaft diameter reduced slightly and one extra wheel shrunk on, the critical speed was reduced to 1700 r.p.m., the average loading was increased to 64 lb. per sq. in., and the model whipped just as violently as before from about 3100 r.p.m. upward. Two differences, however, were noted: the whipping seemed to depend principally upon the more lightly loaded of the two bearings; and at speeds above 4000 r.p.m., the whip diminished somewhat in amplitude unless a large increase in oil supply was given to the bearings. The loading was increased to 85 lb. per sq. in. by the addition of the other wheel, and the critical speed thereby reduced to 1425 r.p.m., the model now would not whip at all at any speed up to 4000 r.p.m. The indication is that the oil whip may be avoided by the use of sufficiently heavy bearing loading pressures.

Summary and Conclusions

A new cause of shaft whipping has been found which seems to account for the resonant whirling of shafts running in well lubricated journal bearings at speeds of about two times critical speed or higher. The source of the stimulus supporting the whipping has been found in the oil film in the bearings. A theory is proposed to account for the action of the oil in producing the resonant vibration of the shaft. This theory was checked and extended by test results on models. It appears that the oil film tends to produce a forward movement of the journal around in its clearance, at a frequency equal to about half running speed, or less if much frictional resistance to the whirling motion must be overcome; and that this oil stimulus comes into resonance with the shaft when the speed reaches about twice critical speed, producing resonant whirling, or whipping, of the shaft. This continues with increasing violence at all higher speeds, with sufficient oil supply, the frictional resistance to whirling serving to hold down the oil stimulus to the resonant frequency. In tests with models it was found that this form of whipping could be stopped or prevented in a number of ways:

(1) By shutting off the oil supply to the bearings
(2) By misalignment of bearings
(3) By steadying the shaft
(4) By confining operation to a speed range below twice critical speed
(5) By the use of a friction-damped bearing
(6) By avoiding very light bearing loading pressures

Of these remedies (1) and (2) are obviously impracticable for commercial application, and (3) involves making contact with the shaft, which also is undesirable. The other three are practical means to avoid the trouble.