IMPROVING THE RELIABILITY OF LARGE DIESEL ENGINE TURBOCHARGERS

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Abstract:

This paper deals with the analysis of a class of turbocharger used on locomotives and marine diesel engines. These larger sized turbochargers operate at 30,000 RPM and use floating ring bearings. The floating ring bearing is very common for use in the smaller high-speed automotive engines. The use of floating ring bearings for the heavier turbochargers can encounter a number of difficulties which can lead to catastrophic failure of the impeller and shaft, depending upon operating conditions and clearance values of the bearings. This class of turbocharger mounted in floating ring bearings is inherently unstable with limit cycle whirling. The whirl orbits are controlled by the nonlinear bearing forces generated in the inner and outer fluid films of the bearing. Depending upon the bearing clearance values of the bearings, a sufficiently high whirl motion can occur, causing the compressor impeller to rub, leading to impeller and shaft failure. Dynamical models are presented to show the nonlinear behavior of the turbocharger and the magnitudes of the limit cycle motion. With many of the marine engines which lack turbocharger speed controls, it is shown under what conditions of overspeed, the third bending critical speed may be approached. Operation at these elevated speeds may lead to compressor failure due to rubbing of the inducer. Suggestions for reliability improvement are presented, including use of a synthetic lubricant, additional oil filtration, and proper bearing clearance values. Finally, an improved bearing design is presented using a multi-lobed bearing design which improves the stability problem and elevates the third critical speed from the operating speed range.

Keywords: Turbochargers, floating bush bearings, nonlinear transient response, rotor stability, self-excited whirling, three-lobed bearings, limit cycle motion

Introduction:

The turbochargers used for the large stationary and marine diesel engines are considerably larger than the typical turbochargers used in automotive engines. The diesel engine turbocharger under consideration is shown in Figure 1.

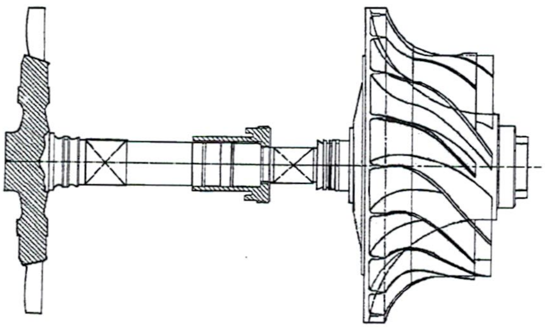


Fig. 1 Diesel Engine Turbocharger

These turbochargers operate at a speed of approximately 30,000 RPM, as compared to over 100,000 RPM for the automotive turbochargers. The typical turbocharger design is a double overhung turbo-rotor with a steel turbine wheel of high temperature alloy and an attached aluminum compressor wheel. The typical type of bearings used for the smaller high-speed automotive turbochargers is the floating ring bearing. The concept of the floating ring bearing was developed in the 1930s. The original concept was to reduce horsepower losses. A plain journal bearing, for example, will not operate satisfactorily in either the smaller turbochargers operating at 100,000 RPM or the much larger diesel engine turbochargers with an operational speed of only 30,000 RPM. The reason for this is due to the inherent self-excited whirl instability encountered with a fixed plain journal bearing. At higher speeds, the rotor will go into a subsynchronous whirl motion. Continued operation with a plain journal bearing at high speed will result in excessive bearing wear followed by total rotor failure.

With the floating ring bearing, for example, as shown in Figure 12, the journal bearing is not fixed but is a ring which is free to float on the shaft. The ring rotates at a fraction of the shaft running speed from a theoretical maximum of 50% to as low as 15% of running speed. The ring speed is controlled by the viscosity of the inner and outer oil films and the clearances of the inner and outer bearings. The inner ring clearance of the ring is kept at a much tighter clearance than the outer ring. This causes more friction losses in the inner clearance, resulting in higher temperatures and lower oil viscosity.

Although the floating ring bearing itself is unstable from a linearized standpoint, the rotor motion will form a limit cycle whirl motion due to the nonlinear bearing forces. This is referred to as subsynchronous whirling with double overhung turbochargers. The turbocharger will precess at a fraction of running speed. Normally, the whirl motion encountered is a conical whirling motion. Under properly designed clearance ratios for the floating ring bearing, the small turbochargers may run at high speed with controlled limit cycle whirling.

Although the concept of the floating ring bearing is quite simple, the analysis of the dynamical behavior of a turbocharger in a floating ring is exceedingly difficult. This is due to the nonlinear nature of the fluid film and the requirement of the numerical integration of the equations of motion. The advantage of the floating ring bearing is in its simplicity and cost of manufacture. Although the floating ring bearing is simple to manufacture and is low in cost, it is not recommended for the larger diesel engine turbochargers.

Description of Diesel Engine Turbocharger:

The left section as shown in Fig. 1 represents the steel turbine wheel. To the right is attached the aluminum compressor wheel with the inducer section. The inducer section is separate from the compressor wheel. Typical turbochargers are double overhung rotors. That is, the bearings are inboard of the turbocharger compressor and turbine wheels. The left bearing represents the turbine-end bearing, and the right end is the compressor-end bearing. Adjacent to the compressor bearing is the thrust bearing. It should be noted that the turbine-end bearing is larger than the compressor-end bearing. The reason for this is for ease of assembly and disassembly for field maintenance.

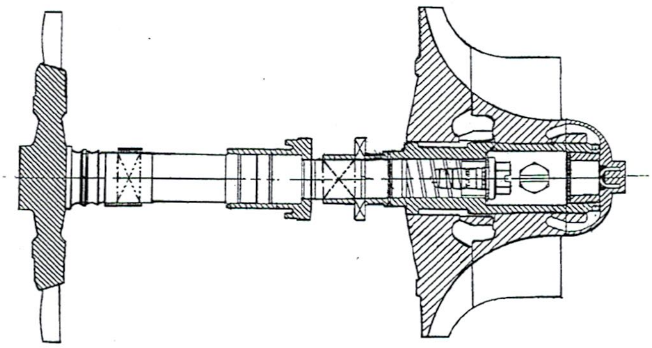


Fig 2 Turbocharger Cross Section

Figure 2 represents a cutaway of the diesel-end turbocharger. This figure is of interest as it shows the construction and method of attachment of the compressor and inducer wheels. The steel shaft extends under the compressor and inducer wheels. The compressor wheels are attached to the wheel shaft by means of a sleeve and bolt arrangement. There is an end cap, or bell, which maintains the compressor and inducer wheels. In Figure 1, the compressor wheel assembly appears to be fairly rigidly attached to the shaft. In reality, because of the extension of the steel shaft and the sleeve, the overhang of the compressor section is fairly flexible.

Design Considerations for the Turbocharger:

The principle design consideration for the turbocharger is primarily with the aerodynamic performance of the compressor wheel and the design of the turbine for the power required to drive the compressor. The aerodynamics of the compressor design are indeed quite elaborate, as the compressor design is quite sophisticated in order to meet the head requirements for the system. The turbocharger compressor wheel is always an open-bladed radial centrifugal compressor design with a backward sweep. The backward sweep of the blades is to generate a larger Q/N flow region. In order not to starve the inlet of the compressor wheel, there are 10 splitter vanes in addition to the 10 larger impeller vanes.

Small auto engine turbochargers operate at speeds exceeding 100,000 RPM. The design speed of this particular type of the turbocharger is 30,000 RPM. The primary consideration for the smaller turbochargers is cost, as the smaller ones may cost only several hundred dollars. These larger turbochargers may exceed $70,000 in cost. The cost of repair and replacement of these turbochargers therefore can represent a considerable outlay of funds. The second major design consideration for the turbocharger is the centrifugal stresses in the turbine, and particularly the compressor wheel. These compressors are designed such that they are well below the burst speed.

A third major consideration for the design of the turbocharger is ease of assembly and disassembly. The steel alloy turbine wheel is usually friction welded to the steel shaft. The aluminum impeller wheels are attached to the shaft, as previously mentioned, by the sleeve.

Compressor and inducer wheel clearances as such are required to be close to line to line contact. At the operating speed of 30,000 RPM, the centrifugal expansion of the compressor wheel assembly, therefore, does not provide a great deal of stiffening to the steel shaft and the extension sleeve. The major stiffness of the assembly is controlled by the steel shaft supporting the compressor wheel and inducer.

The final consideration for the design of these high-speed turbochargers is the bearing type to be employed with the turbocharger. With the smaller turbochargers, the floating ring bearing is commonly used. With the smaller automobile type turbochargers, where cost is a great consideration, the floating bush bearing has found to be satisfactory provided that proper clearances of the inner and outer ring are maintained. This type of bearing is also currently being used in this particular class of turbocharger.

The problem with the floating ring bushing design is that at high speeds, the turbocharger is inherently unstable. That is, the turbocharger exhibits self-excited whirl motion in which the turbocharger is precessing or whirling at a lower frequency. This is referred to as nonsynchronous precession. The predominate whirl mode is usually a conical motion of the rotor. By properly controlling the ring inner and outer clearances , a bounded finite limit cycle orbit or motion can be obtained. In order to keep this limit cycle motion within bounds, the inner clearance must be very tight, leading to high bearing temperatures and other operating problems associated with tight bearing clearances.

The floating bush bearing operates by spinning at a fraction of the rotor speed. The typical precession rate of the ring speed may be from 15-30% of shaft speed. The outer ring clearance is normally several times larger then the clearance of the outer ring. Theoretical ring speed can go as high as 50%, although this is never achieved in practice. The maximum ring speed is limited by the higher temperatures generated in the inner ring. By properly controlling the ring inner and outer clearances, a bounded finite limit cycle orbit or motion can be obtained.

The mechanism of the outer ring acts as a type of crude squeeze-film damper to generate nonlinear forces to cause a limit cycle whirl motion. This bearing type, which is well suited for the smaller turbochargers because of cost consideration, is not an ideal bearing for the larger, more expensive turbochargers. The larger turbochargers, which weigh approximately 44 lbs, may operate with large limit cycle motions, even with properly controlled clearances. In addition to the self-excited whirl motion inherent in the floating bushings themselves, there are additional aerodynamic forces that creates self-excited whirl motion in all compressors and turbines. This effect is referred to as the Alford effect. The Alford effect causes cross-coupled aerodynamics bearing forces to act on the compressor and turbine, which also promotes instability.

It will also be shown that the compressor bearing should be larger than the turbine bearing, since this carries the higher bearing loads. The smaller compressor bearing was designed from the standpoint of the ease of assembly and disassembly. It will be seen that damage and failure are predominantly associated with the compressor-end bearing.

Unfortunately, with most turbochargers, whether they be small or large, the bearing design is an approximation based on previous practice and experience. The engineering, until recently, has been limited because of the lack of rotor dynamics codes to compute the nonlinear fluid-film bearing forces and to also compute the nonlinear time transient motion of the turbocharger.

These bearings usually are designed and perfected primarily based on experimental testing. Since the large diesel engine turbochargers cost an order of magnitude greater than the smaller automotive turbochargers, one can afford to design a more effective bearing system for the turbochargers, such as the multi-lobed bearing. The theory of these multi-lobed bearings is well established and has also been employed in other large diesel engine turbochargers.

It is now possible to combine the nonlinear fluid-film characteristics of either the floating-ring or the multi-lobed bearing with a nonlinear time transient analysis of the complete turbocharger to generate the magnitude of the limit cycle motion and bearing forces transmitted. At the time this class of turbocharger was designed, this type of technology was not generally available for commercial use.

Operational Problems With Diesel Engine Turbochargers:

There are a number of situations and conditions that can occur with the large diesel engine turbochargers which can lead to turbocharger damage and even catastrophic failure. These problems and situations are as follows:

1. **Ingestion of Foreign Objects in the Inducer Section:** If a large particle or object should be induced into the inlet of the compressor, this may cause damage to the inducer and could lead to two other situations, such as inducer rubbing and rotor unbalance. The additional unbalance and impact of the foreign object could lead to immediate turbocharger failure.

2. **Turbine Blade Build Up:** A second condition is the observed buildup of deposits on the turbine blades under certain operating conditions of the diesel engine. This condition sometimes requires maintenance and cleaning of the turbine blades to ensure than excessive unbalance is not encountered in the turbine section.

3. **Insufficient Oil Supply to the Compressor Bearing:** The compressor bearing is the most heavily loaded, and it is imperative that the compressor bearing receive a proper lubricant supply. If the lubrication is somehow restricted to the compressor bearing, then overheating of the bearing will occur with glazing of the bearing surface and bearing ring seizure.

4. **Improper Maintenance of Compressor and Turbine Bearing Clearance Ratios:** In order for the turbodiesel to operate with a controlled limit cycle motion, the bearing clearance of the compressor and turbine bearing must be carefully controlled. The range often expressed in service manuals is unacceptable. The turbine bearing is seldom observed to fail. However, the paradox is that if the inner and outer clearances of the turbine bearing are not properly maintained, then large whirling orbits will occur on the compressor wheel leading to rubbing and seizure of the compressor bearing. These events leads to catastrophic failure with the shattering of the wheel and bending and shearing of the shaft. It will be shown that the reason for this is that the failure mode is primarily a conical whirling motion of the turbocharger, and the proper maintenance of the inner and outer ring clearances in the turbine bearing is essential to control the whirl motion at the compressor.

5. **Tight Tip Clearances With Compressor and Inducer Sections:** From the standpoint of efficiency, one wishes to keep the clearances between the volute and the open-bladed compressor and inducer section as tight as possible. This reduces leakage losses and adds to the efficiency of the compressor wheel. However, due to the subsynchronous whirling

motion of the compressor wheel and the centrifugal growth of the compressor, the clearance is usually too tight for reliable continuous operation. To be on the safe side for existing turbochargers of this design, the clearance values should be opened up about 5 mils to prevent rubbing.

6. Improper Lubricant Type and Lack of Advanced Oil Filtration System: The design conditions of lubrication for a diesel engine and high-speed turbochargers are quite dramatically different. All turbochargers, whether automotive or large diesel engine turbochargers, operate more reliably when a high grade synthetic lubricant is used. A preferred name for a synthetic lubricant would be a designer lubricant. Modern synthetic lubricants for automotive and diesel engine applications are usually based on a base stock of a polyalphaolefin. This type of long-chain molecule is similar to the earlier Pennsylvania paraffin long-chain oils that have superior lubrication properties. Back in the early days of lubrication, the Pennsylvania oil was rated as a viscosity of 100, whereas the Gulf oil with a benzene ring structure was rated with a viscosity index of zero. A well designed synthetic oil can have a viscosity index as high as 145. This is an important consideration for turbochargers, as they run extremely hot with very tight clearances.

The second important characteristic of the well-designed synthetic lubricant is its higher coking temperature. This is the temperature at which the oil begins to coke and turn to a tar-like substance. With conventional oils we have a variety of hydrocarbons varying from the light naphtha to the heavy paraffin. The lighter components may become volatile, whereas the heavier components can turn into a waxlike substance. Once this occurs with a turbocharger with a floating ring bearing, the ring can seize to the shaft and the ring will become like a plain journal bearing. The entire rotor system will fail shortly thereafter due to the large subsynchronous whirling. The third characteristic of the synthetic oil is the ability to take high loading or high film strength. This is due to the property of adding certain additives, such as diesters and high-pressure film additives. Current lubricants used in many unaspirated diesel engines are totally unsuitable for use with high-performance, high-speed turbochargers operating under tight bearing film conditions. In addition to the quality of lubrication, an additional consideration is the use of oil prefilters for the turbochargers. The turbochargers for diesel engine operation operate under more severe conditions than an automotive turbocharger. This is in part due to the higher operating temperatures of the diesel engine and also because of particle carryover of the diesel combustion products into the oil supply. This can cause clogging of the oil flow into the compressor bearing, which will then lead to failure. The compressor and turbine bearings must be assured of receiving a clean, filtered oil supply if bearing longevity is to achieved.

7. Dynamic Maneuver Loads On Compressor Bearing: A seventh consideration for shipboard machinery, particularly with high-speed cutters, is the occurrence of dynamic loads generated on the compressor bearing due to maneuvering. The compressor wheel acts similar to a gyroscope. If a ship is rapidly plowing through high seas with large wave action, the pitching of the ship causes a change in the gyroscopic moment vector acting on the compressor wheel. This, in turn, creates additional dynamic loads, which are mainly transmitted to the compressor bearing. Thus maneuver loading, combined with inadequate oil supply and substandard bearing dimensions, can lead to bearing damage, compressor impeller rubbing and eventual system failure.

Critical Speed Analysis:

Although the bearings for the turbocharger are highly nonlinear, it is of considerable value to perform a critical speed analysis of the turbocharger. By means of a critical speed analysis, one can determine the relative mode shapes and the energy distribution in each bearing and shaft for that particular mode.

Figure 3 represents a computer model of the turbocharger used for the critical speed analysis. The model contains 48 degrees-of-freedom and represents three levels of attachment. It should be noted that a computation of a built up rotor or multi-level rotor construction is difficult to perform using the transfer matrix method.

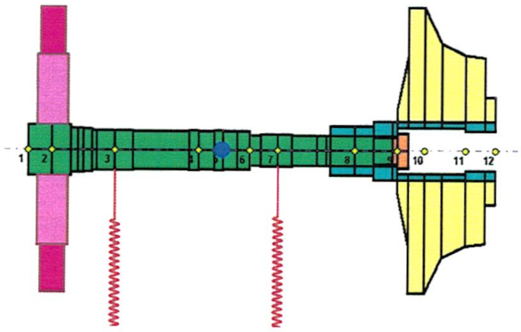


Fig 3 Critical Speed Model

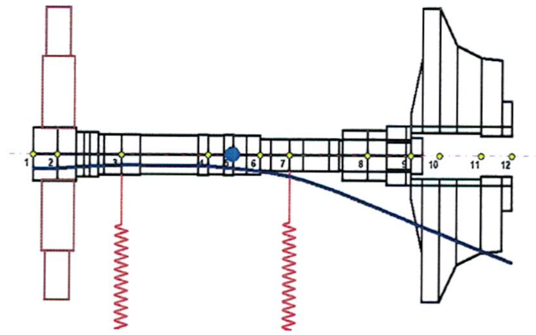


Fig. 4 Turbocharger Static Deflection

Figure 4 represents the static deflection of the rotor. The bearing loads on each bearing are also computed with this calculation. Note that since the center of gravity is closer to the compressor bearing, the compressor bearing carries a greater load than does the larger turbine bearing. The relative load on the turbine bearing is 15 lbs and it is approximately 30 lbs on the compressor bearing. Therefore, although the compressor bearing is smaller, it is carrying twice the static loading of the turbine bearing.

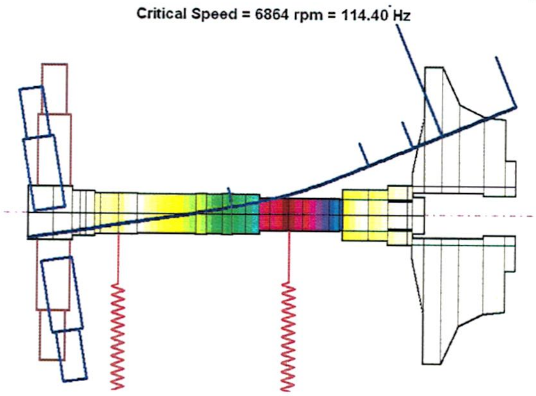


Fig. 5 1st Turbocharger Mode

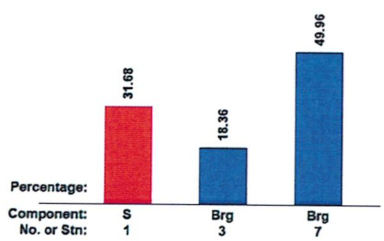


Fig. 6 1st Mode Shaft & Bearing Strain Energies

Fig. 5 represents the relative turbocharger 1st mode. This conical mode shape is typical with all double overhung rotors with inboard bearings. Note that the maximum motion is at the compressor inducer location.

Fig. 6 represents the relative strain energy in the shaft and bearings for the 1st mode. Note that 50% of the 1st mode strain energy is associated with the smaller compressor bearing.

The maximum shaft strain energy for this mode is concentrated under the compressor bearing. This is the location in which failure has been experienced. Since the disks are outboard of the bearings, the lowest mode is a conical whirl mode. It will be seen that this mode is the easiest mode to excite for self-excited instability. Also, when the turbocharger becomes unstable and whirls in this lower mode, large amplitudes occur at the compressor and inducer section, making it very susceptible to rubs.

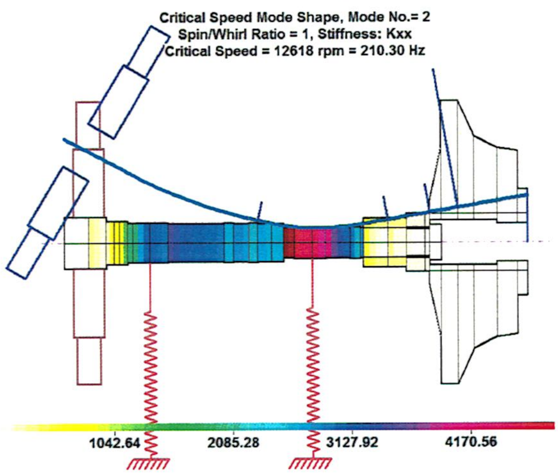


Fig. 7 2nd Turbocharger Mode

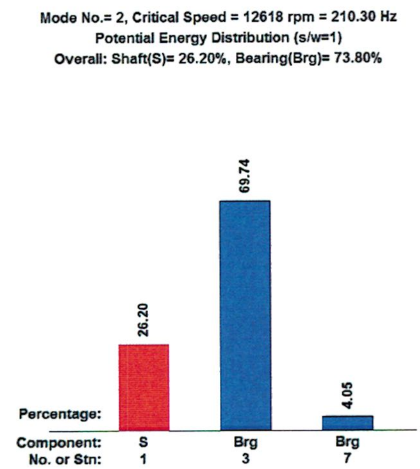


Fig. 8 2nd Mode Relative Shaft & Bearing Strain Energies

Figure 7 represents the second mode of the turbocharger. In this mode, the turbine and compressor are in phase. There is more bending strain energy in the shaft. The turbocharger may exhibit instability in the second mode, as well as the first. However, the higher, or cylindrical mode, is normally not as sensitive to unbalance or self-excited motion as the conical lower mode is. Figure 7 represents the strain energy distribution for the second mode. In this mode it is seen that there is a much higher strain energy distribution in the turbine-end bearing. The compressor end bearing has little influence on controlling this mode.

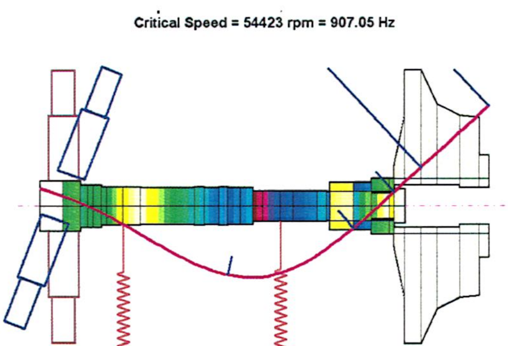


Fig. 9 Turbo 3rd Mode at 54,000 RPM

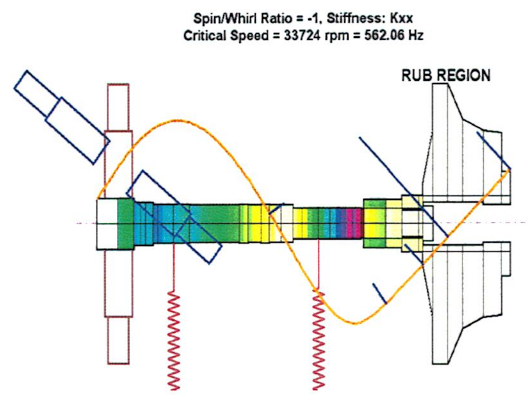


Fig. 10 Backward Mode at 33,700 RPM

Fig. 9 represents the turbocharger 3rd mode. Under normal conditions, this mode should be above the operating speed with well designed bearings. Fig. 10 represents the 4th backward mode at 33,724 RPM. If the turbocharger should overspeed and rub, then this mode could be excited causing a violent backward precession of the compressor wheel.

Modeling of Turbocharger in Floating Bush Bearings:

The diesel engine turbocharger, including the floating bush fluid-film bearings, was modeled using the ***DyRoBeS*** rotor-bearing dynamics software. Figure 11 represents a cross-section of the turbocharger as modeled using the ***DyRoBeS*** software to include the floating bush bearings. The rotor is modeled with 11 finite element shaft elements which include 30 sub-elements. The rotor is actually modeled as a 3-bearing system. The bearings 1 and 2, at stations 3 and 7, represent the floating bush bearings.

The bearing station at station 10 is used to introduce the Alford type of aerodynamic cross-coupling forces acting on the compressor wheel. The complete system is represented by a total of 52 degrees-of-freedom for displacements and rotations. Note that the center of gravity of the turbo is closer to the compressor bearing. This causes the compressor bearing loads to be double the turbine bearing. Also shown on Figure 11 is a red arrow at station 10. This represents an applied unbalance. In addition to external forcing functions such as unbalance, one could also add shaft bow and disk skew as exciting forces. From a linearized stability standpoint the system would be characterized by 104 complex eigenvalues. Of these eigenvalues, only the lower modes are of interest.

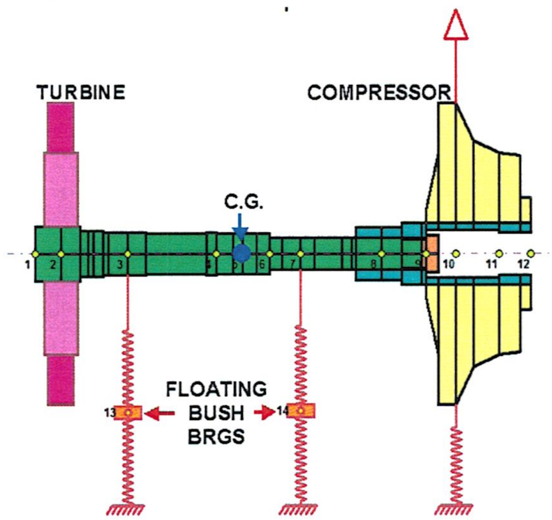


Fig. 11 *DYROBES* Turbo Model Including Floating Bush Bearings

Floating Bush Bearings:

A schematic of the floating bush bearings is shown in Fig. 12. The location of the floating bush bearings are at stations 13 and 14 as shown in Fig. 11. The masses of the rings are acting at stations 13 and 14.

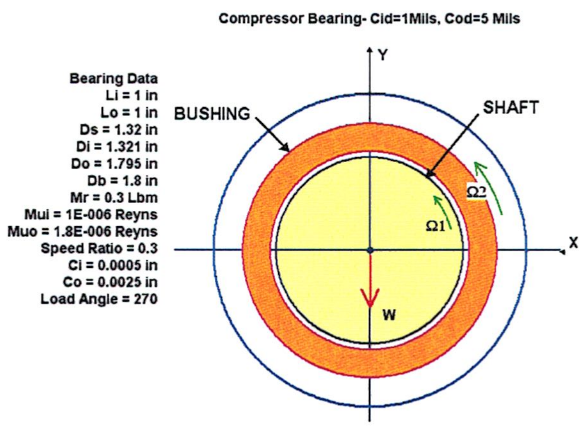


Fig. 12 Schematic of Floating Bush Bearing

In order to perform a time transient analysis, the equations of motion are integrated forward in time. It is required, therefore, to integrate 52 second order equations simultaneously while computing the nonlinear bearing fluid-film forces at each time increment. Some care has to be exhibited in generating the model in order to expedite the numerical time transient calculations and to ensure convergence of the solution. The numerical nonlinear time transient solution will be discussed in further detail in a later section.

With this model, in addition to the time transient response, one may calculate the nonlinear synchronous unbalance response of the turbocharger system due to assumed values of unbalance acting on the compressor or turbine. In addition to the response caused by radial unbalance, the influence of shaft bow or disk skew may also be included.

It is apparent from looking at the critical speed mode shapes and energy distribution that the conical whirl mode is of greatest concern. The larger turbine bearing will have little influence on controlling this mode as the predominant strain energy is in the compressor bearing. Although the turbine bearing has a reduced strain energy as compared to the compressor bearing, paradoxically an increase in the turbine floating ring clearances will cause excessive conical motion leading to turbocharger failure. Therefore, we shall see later that the clearance specifications of the turbine floating bush bearing cannot be ignored.

Floating Bush Bearing Coefficients:

Although the behavior of the floating bush bearings are highly nonlinear, it is desirable to determine the linearized bearing coefficients in order to perform a complex damped eigenvalue analysis on the system. This is usually referred to as a stability analysis. The critical speed analysis is an undamped analysis which assumed circular synchronous precession. In this analysis, only one nominal bearing coefficient is assumed. In order to perform a stability analysis, the computation of 8 bearing coefficients are required for both the inner and outer surfaces of the floating bush bearing.

Table 1 represents some typical dimensions of the turbocharger floating bush bearings. It is seen that the turbine bearing is larger than the compressor bearing even though it is carrying a much lower load. Using these dimensions the bearing stiffness and damping coefficients may be computed for each bearing. These dimensions are used for the computation of the bearing coefficients for each ring.

Table 1 - Floating Bush Bearing Characteristics

	D_s	L_i	C_d	D_b	L_o	C_{od}	W
Compressor	**1.32**	**1.0**	**.001**	**1.80**	**1.0**	**.005**	**33**
Turbine	**1.70**	**1.0**	**.002**	**2.165**	**1.0**	**.005**	**11**

In Table 1 is shown the inner and outer clearances of the compressor and turbine end bushing bearings. Note that the specified diametral internal clearance of the compressor bushing is 1 mil. This is about one-third the thickness of a human hair. With this tight a clearance, the compressor bearing must be well supplied with oil as it will run hot. The outer ring clearances are 5 mils. The outer ring clearances are normally specified as larger then the inner ring clearances. The specified ranges of clearances permitted for both bearings is rather large. For example, the turbine bearing clearance is specified to be as large as 3.2 mils for the inner clearance and 10.5 mils for the outer ring clearance. It will be seen that these large values of specified permissible clearances is excessive and will lead to turbocharger failure.

Figure 13 represents the bearing pressure profile and stiffness and damping coefficients for the compressor end bearing. The inner and outer surfaces of the ring develop a very similar pressure profile except that the bearing coefficients generated from each pressure field are considerably different.

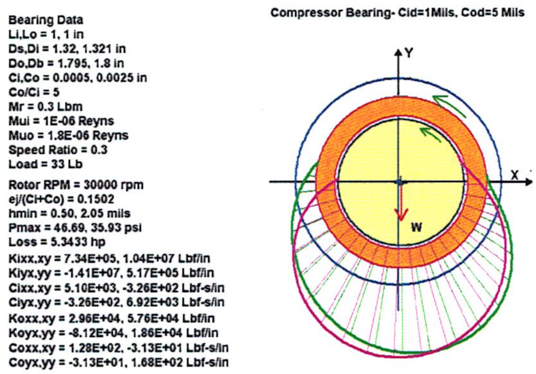


Fig. 13 Compressor Floating Bush Bearing Coefficients at 30,000 RPM

There are two assumptions that were made in the calculation of the floating bush bearing coefficients. The ring speed was computed to be around 0.3 and the oil viscosity of the inner ring is always assumed to be lower that the outer ring viscosity.

Tables 2 and 3 represent the bearing coefficients for the compressor and turbine bearings

Table 2

COMPRESSOR BUSH BEARING COEFFICIENTS - 30,000 RPM

	K_{yy}	K_{xy}	K_{xx}	K_{yy}	C_{xy}	C_{xx}	C_{xx}	C_{yy}
Inner	**.733E6**	**10.4E6**	**-14.1E6**	**.52E6**	**5100**	**-326**	**-326**	**6920**
Outer	**30,000**	**58,000**	**-81,200**	**18,600**	**128**	**-31**	**-31**	**128**

In Tables 2 and 3, it is seen that the inner oil film stiffnesses are an order of magnitude higher then the bearing coefficients generated by the outer film.

TABLE 3

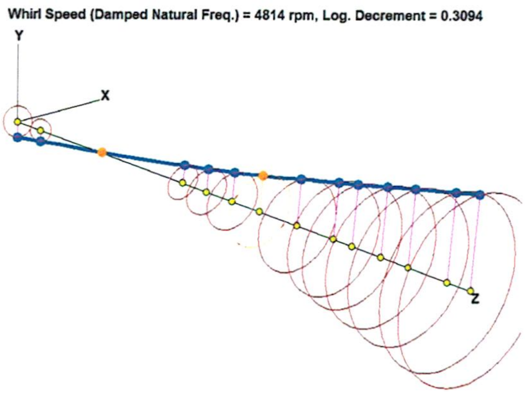
TURBINE BUSH BEARING COEFFICIENTS - 30,000 RPM

	K_{yy}	K_{xy}	K_{xx}	K_{yy}	C_{xy}	C_{xx}	C_{xx}	C_{yy}
Inner	**.63E6**	**15.0E6**	**-19.4E6**	**.57E6**	**7360**	**-300**	**-300**	**9500**
Outer	**11,100**	**69,000**	**-87,000**	**4810**	**150**	**-13**	**-13**	**184**

Stability of Turbocharger:

Using the floating bush bearing coefficients, based on nominal clearances, as shown in Tables 2 and 3, a damped eigenvalue analyses may be performed to determine the relative stability of the turbocharger. By varying the various clearances, the clearance effects may be observed to see which clearance combinations result in improved stability characteristics. In addition to the bearing coefficients, an additional aerodynamic excitation of 500 lb/in cross-coupling was assumed acting at the compressor location.

Based on the bearing characteristics, as shown in Table 2 and 3, the damped eigenvalues for the turbocharger were computed. Figure 14 represents the first forward mode of the turbocharger, and it is a conical motion with the maximum amplitude at the compressor. With the dimensions as specified in Table 1, the first conical mode is stable with a whirl frequency of 4814 CPM and a log decrement of 0.31.



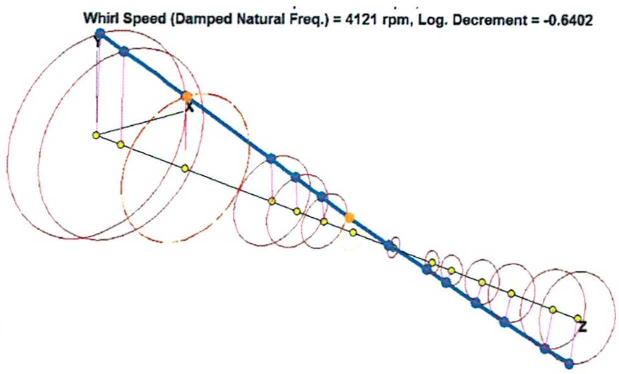


Fig. 14 Stable 1st Mode With Nominal Brg Clearances, Nf1 = 4814 CPM , LD=.31

Fig. 15 Unstable 1st Mode With Max Turbine Brg Clearances, Nf1=4121CPM , LD=-0.64

The 1st conical mode is stable with the coefficients as listed in Tables 2 & 3. If, however, the turbine bearing clearances are increased to the stated maximum values of 3.2 mils internal and 10.5 mils external, then we have an unstable situation as shown in Fig. 15.

The increase in turbine bearing clearances causes an unstable whirling situation. Strict limits on the turbine bearing must be maintained to minimize self excited instability to occur. Hence a sloppy turbine bushing bearing is an invitation to turbocharger failure, even though no failure of the turbine bearing itself is ever apparent. Therefore, it appears that the maximum turbine bearing clearance values as specified by the manufacturer are not permissible, as this would induce turbocharger instability in the first conical mode.

Although the turbine bearing never experiences failure, it is imperative that this bearing clearance be kept tight in order to maintain proper stability. Instability in the higher cylindrical mode is also indicated with excessive turbine bearing clearances.

Whirl Mode Near Running Speed:

On the marine diesel engines, there is no speed control on the turbocharger. This is a concern when the turbocharger is operated over 30,000 RPM. Figure 16 represents the eighth eigenvalue, which is the third forward critical speed. Note that this mode is at 33,000 RPM, which is 10% over design speed. There is a concern that if the turbocharger is over-sped, that this mode could be excited, leading to possible rubs.

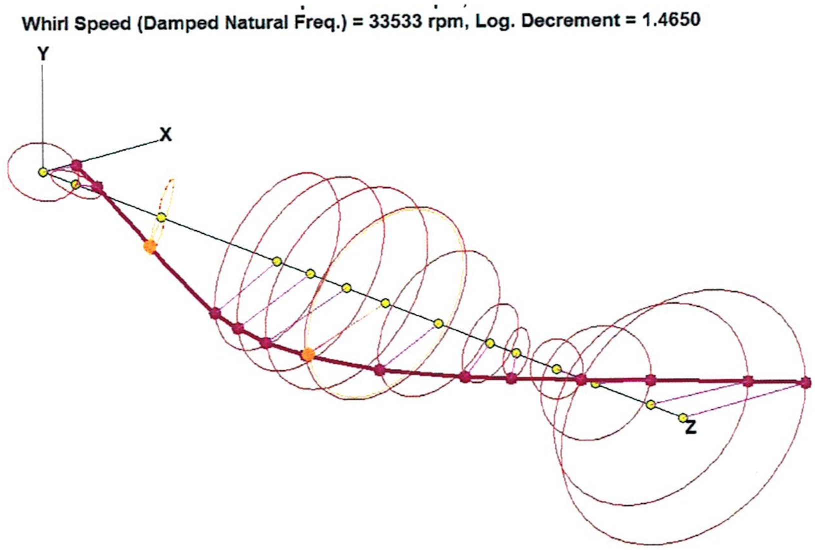


Fig 16 Turbocharger Bending Mode Above Running Speed at 33,500 RPM

In this mode, it is seen that the maximum amplitude occurs at the compressor and inducer locations.

Locked Compressor Bearing Leading to Failure:

Under marginal lubrication conditions and tight compressor bearing inner clearances, the compressor bearing may weld onto the shaft. When this occurs, the floating bush bearing essentially becomes a plain bearing with a large clearance. The result of a locked bearing causes a very strong subsynchronous conical whirling to occur with large motion at the compressor. This frequency is predicted to be about 8,000 RPM. To compound the problem, the third forward critical speed drops down further into the operating speed range. Thus, the frozen, or locked, compressor bushing bearing leads to immediate turbocharger failure, usually resulting in shearing of the shaft under the compressor bearing as shown in Fig.17.

Figure 17 shows the effect of a rub on the compressor wheel. The blades are stripped, and the compressor has sheared the shaft under the compressor bearing.

Once the inducer section or compressor wheel initiates a rub on the volute, the gyroscopic moments of the wheel reverse moment direction and tend to cause the wheel to precess backwards in the casing.

This leads to rapid and extensive damage that is difficult to catch in time to avoid a total catastrophic compressor, inducer and shaft failure.

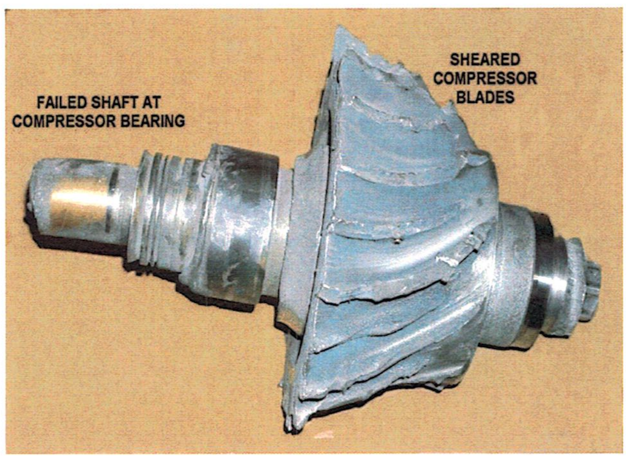


Fig.17 Failed Compressor and Shaft After Heavy Rub

Nonlinear Time Transient Analysis:

In order to determine the dynamical motion of the turbocharger in the nonlinear floating bush bearings, a nonlinear time transient numerical analysis must be performed. Figure 18 represents the transient motion of the rotor for 50 cycles of shaft motion. The turbocharger is modeled by 14 major mass stations each with 4 degrees of freedom. The masses of the floating bush bearings represent another 4 degrees of freedom.

Thus for each time step, it is required to integrate 60 equations of motion and solve the finite element fluid film bearing equations for the forces generated on the inner and outer ring surfaces of the two floating bush bearings. In order to maintain accuracy of the orbits, 2000 time steps were used for each cycle of shaft motion. The numerical computations were performed using the Runge-Kutta procedure.

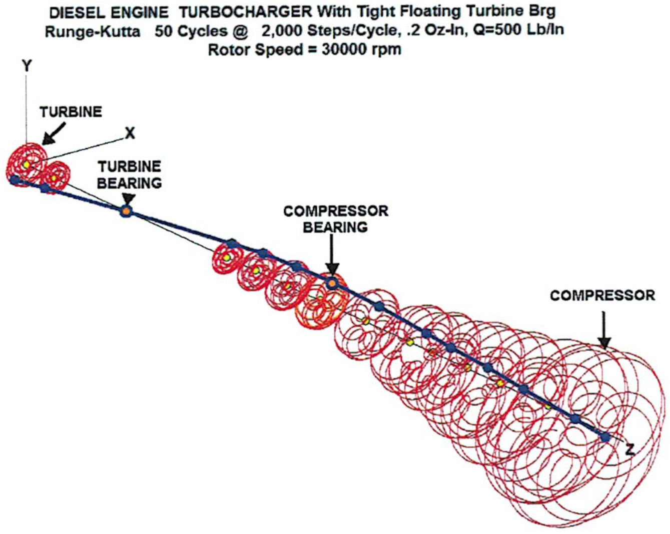


Fig. 18 Time Transient Motion of Turbocharger with Tight Turbine Brg, Q=500 Lb/In & 0.2 Oz-In Ub

In addition to the gravitational loads, an aerodynamic cross coupling force of 500 Lb/In is assumed to be acting at the compressor. Also in the model is a radial unbalance of 0.2 oz-in acting on the compressor wheel. The orbital motion shown nonsynchronous limit cycle whirl motion which is bounded due to the nonlinear effects of the bearings.

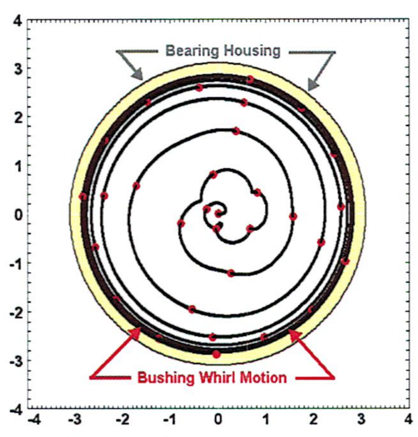


Fig. 19 Bushing Motion

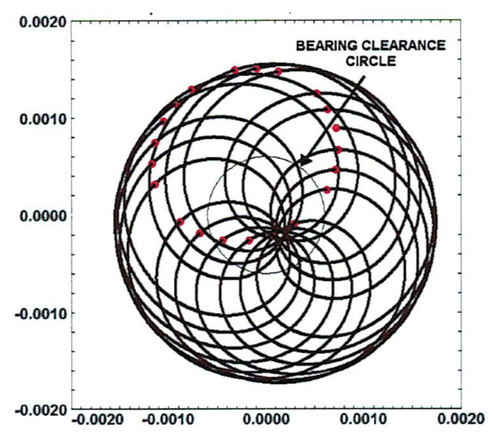


Fig. 20 Absolute Motion At Comp. Brg

Figure 19 represents the motion of the compressor bushing in the housing. Notice that the whirl orbit fills up over 80 % of the bearing clearance. Figure 20 is the absolute motion of the shaft at the compressor bearing location. The resulting bounded limit cycle orbit is a combination of synchronous precession due to unbalance and the lower frequency subsynchronous whirl motion. Note the size of the compressor inner ring clearance as compared to the total orbit observed at the compressor bearing.

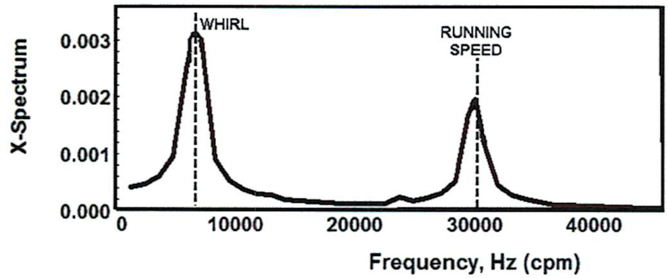


Fig. 21 FFT of Absolute Compressor Bearing Motion at 30,000 RPM Showing Subsynchronous Whirling

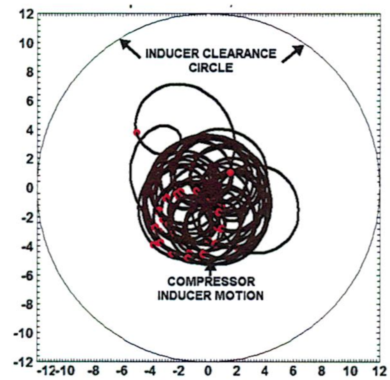


Fig. 22 Inducer Motion

Figure 21 represents the FFT analysis of the motion of the compressor bearing station as shown in Fig. 16. It is seen that the motion is composed of synchronous precession caused by the rotor unbalance and the lower self-excited whirl motion that is inherent with the floating bush bearings. If one filters the frequency data below 500 Hz, as is common practice, then it is apparent that the important whirling motion will not be observed!

Figure 22 represents the motion of the compressor inducer. Drawn on this figure is the inducer clearance circ of the stator vanes. Rubbing will occur when thus orbit exceeds the operating clearance causing blade contact. With the tight turbine bearing, the clearance is adequate.

Compressor Bearing Forces Transmitted:

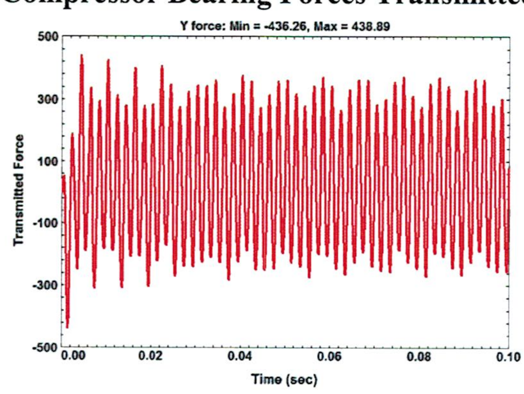


Fig. 23 Compressor Bearing Forces With 0.2 Oz-In Unbalance on Compressor

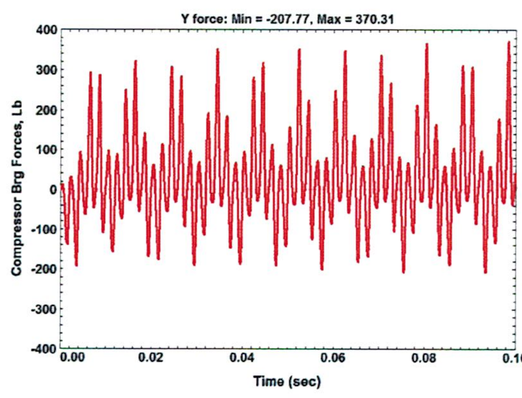


Fig. 24 Compressor Bearing Forces With Balanced Rotor

Fig. 23 shows the large bearing forces of over +-430 Lb with the 0.2 oz-in of unbalance. These large transmitted compressor bearing forces are not due to unbalance. Fig. 24 shows the compressor bearing forces for a balanced rotor with a maximum of 370 Lb.

Motion with Large Clearance Turbine Bearing:

There is a range of allowable clearances specified for both the compressor and turbine floating bush bearings. The maximum specified values for the floating bush turbine bearing is approximately 3 mils for the inner clearance and 10 mils for the outer clearance. Figure 25 represents the nonlinear transient motion for 50 cycles of shaft motion.

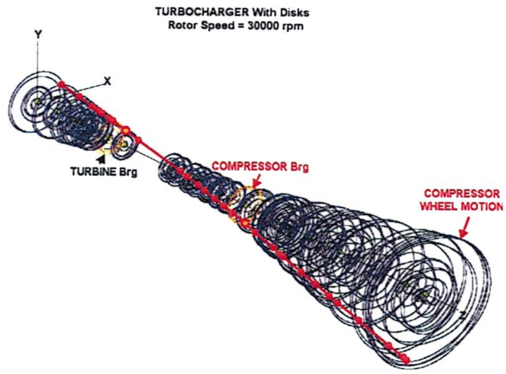


Fig. 25 Transient Motion With Large Clearance Turbine Bearing

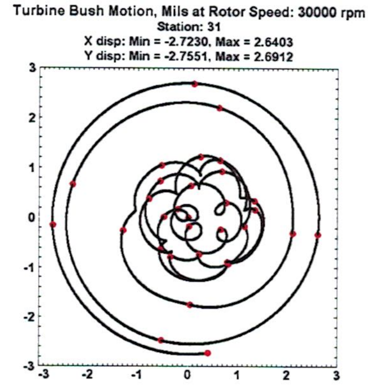


Fig. 26 Turbine Bushing Motion With Large Clearances

Figure 26 represents the turbine bushing motion with the large inner and outer bushing clearances. The initial transient motion starts out in a controlled limit cycle motion influenced by the rotating unbalance forces acting on the compressor wheel. The motion then changes into a large limit cycle motion with a conical motion as shown in Fig. 25.

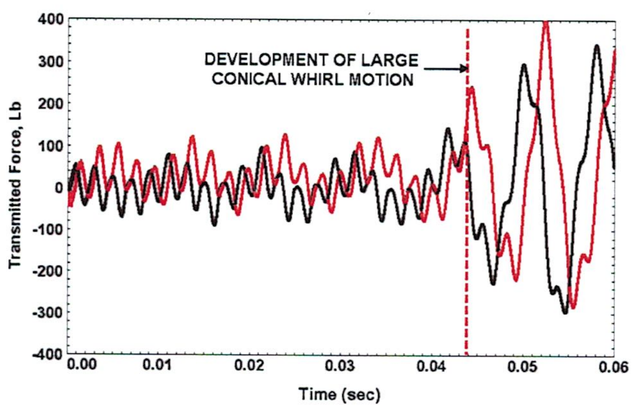


Fig. 27 Compressor Bearing Forces Transmitted With Large Clearance Floating Bush Turbine Bearing

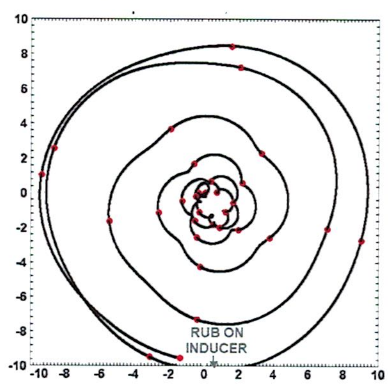


Fig. 28 Excessive Inducer Motion Caused By Large Turbine Bearing Clearances

Figure 27 represents the compressor bearing forces transmitted with the large prescribed turbine bearing clearances. The initial transient motion starts out with a small controlled limit cycle motion caused by the applied unbalance at the compressor wheel. After about 22 cycles of shaft motion, the motion of the system jumps into a large conical whirling mode with a rapid increase in the bearing forces transmitted. Figure 28 shows the resulting motion at the compressor inducer section. This large motion will lead to compressor rubbing and failure.

Turbocharger Motion with Three-Lobed Compressor Bearing:

The floating bush compressor bearing is not an ideal design for the large diesel engine turbochargers. This type of bearing is always inherently unstable. A more suitable design is the use of a three-lobed bearing. Figure 29 represents the pressure profile of a three-lobed bearing with a 75% converging film thickness.

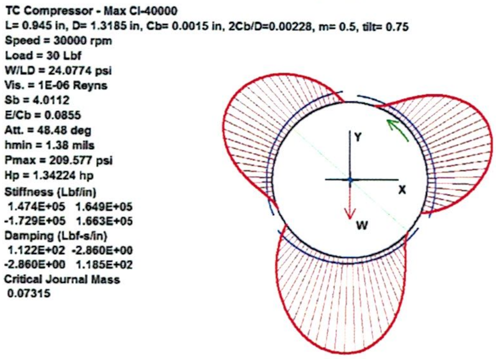


Fig. 29 3 Lobed Compressor Bearing With 50% Preload

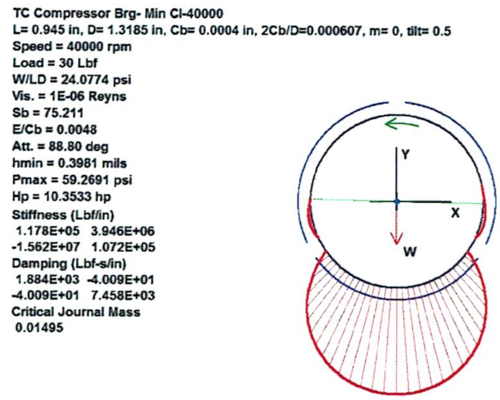


Fig. 30 3 Axial Groove Bearing With Zero Preload

Each lobe is developing a positive load capacity to center the shaft. The radial stiffness of this bearing has been increased, and also there is a reduction in the bearing cross-coupling forces. Often there is an attempt to make a floating bush bearing into a three-lobed bearing by placing axial grooves in it. The axial grooves are thought to increase the stability and lubrication oil supply to the bearing.

Figure 30 represents an axial groove bearing in which the lobes have no preload. Instead of each lobe generating a positive pressure, it is see that only the bottom lobe under the load vector generating a positive pressure profile. As a result, the bearing cross-coupling coefficients are extremely high with the axial bearing. Attempts to add axial grooves to either the inner or outer ring surfaces to improve its stability characteristics is futile. In fact, it even leads to larger limit cycle motion.

Figure 31 represents the nonlinear transient motion of the three-lobed bearing at the compressor station. Notice that the orbital motion has a lobed effect due to the bearing profile.

An FFT analysis of the compressor motion was performed. There is present small components of both the conical and cylindrical whirl motions which are well controlled. The reason for the small amounts of whirl is that the turbine bearing has not been replaced by a three-lobed bearing design. As long as the turbine floating bush bearing clearances are kept tight, the turbocharger motion is well controlled with the three-lobed bearing compressor bearing. Stable motion of the turbocharger is obtained.

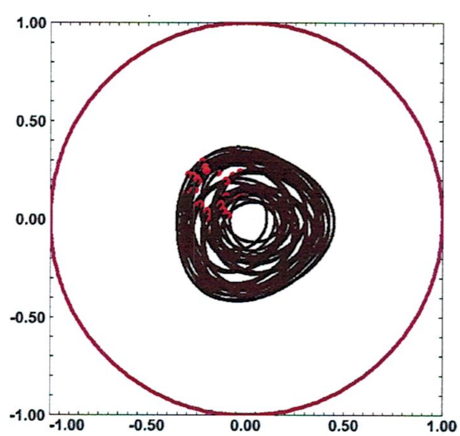


Fig. 31 Compressor Bearing Motion in 3 Lobed Bearing at 30,000 RPM

Discussion and Conclusions:

There are a number of steps that may be taken to improve the reliability of this class of large diesel engine turbochargers. Ideally, the most effective change that can be made would be to install a larger multi-lobed compressor bearing. This is not feasible, as it would require a total redesign of the turbocharger. The following are some of the steps, or procedures, that will improve the reliability of this class of turbocharger.

1. **Employment of a synthetic lubricant**. The turbochargers are lubricated with the same oil used in the diesel engine. The operating conditions in diesel engines are much more severe than in automotive engines. As a general rule in all high-speed turbochargers, whether diesel or automotive, the employment of a high-quality PAO (polyalphaolefin) lubricant is advisable. Conventional lubricants do not meet the requirements necessary for the adequate and reliable performance of high-speed turbochargers. For example, for the turbocharger to operate successfully with limit cycle motion orbital motion, the compressor bearing clearances must be tight. This leads to high heating in the compressor bearing. Conventional lubricants, for example, are a mixture of low and high paraffin type chains and also benzine ring type compounds. The lower paraffinic molecular compounds are volatile under high temperature, and the larger and higher molecular weight paraffinic oils tend to tar up and form varnish on the surface. The PAO lubricant is one specific molecular length and can withstand a higher operating temperature.

It is of interest to note that in the original classification of lubricant oils the Pennsylvania paraffinic type of oil was rated with a viscosity index of 100, and the Gulf oil, with a benzene type ring structure, was rated with a viscosity index of zero. A PAO lubricant may have a viscosity index as high as 145. This means that it maintains its viscosity at higher temperatures.

If varnish buildup should occur on the inner surface of the compressor bearing, the compressor ring may freeze to the shaft. When this occurs, conical whirling occurs with subsequent rubbing of the impeller and turbocharger failure. In addition to the higher viscosity index, the synthetic PAO should preferably be called a designer lubricant as it has higher film strength.

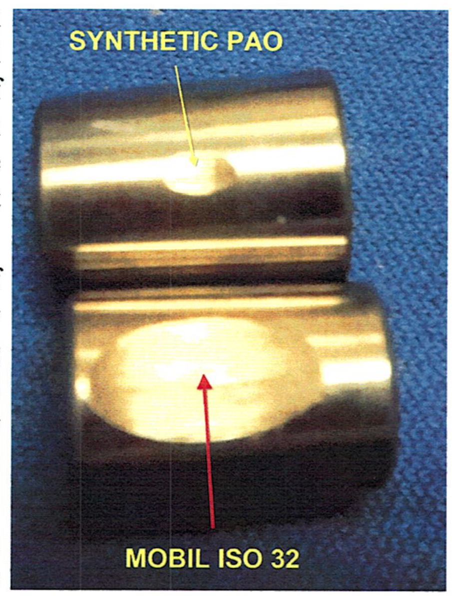


Fig. 32 Wear Comparison Between ISO 32 and PAO

Figure 32 represents a load test comparison between a conventional ISO-32 oil and a high quality PAO. The conventional motor oils do not have the load carrying capacity or boundary layer strength of a good PAO. This quality prevents scuffing of the bearing and shaft surfaces under high loads. A major objection to the use of a PAO is the higher cost of the lubricant. This is a small price to pay in comparison to the repair and replacement of $100,000 for a typical turbocharger. This does not include the losses due to the downtime of the equipment.

2. **Secondary oil filtration**. In addition to the use of a high quality PAO for these large diesel engines, it is also recommended that additional secondary oil filtration filters be used for each turbocharger. Due to the more extreme operating conditions of a diesel engine, as compared to an automotive engine, it is imperative that any particle debris or carbon deposits be filtered before entering into the lubrication system of the turbocharger. Carbon particle buildup in the turbocharger may result in limited oil flow to the compressor bearing or enhance the condition of varnishing of the bearing surfaces. It is imperative to have a clean oil supply to the turbocharger bearings.

3. **Control of the turbine bearing clearances**. Although failure of the turbine-end bearing rarely occurs, its clearance values are critical for the long-term reliable operation of the turbocharger. The range of stated permissible clearance values for both the compressor and turbine bearings is unacceptable. It should be remembered that the bearing stiffness and damping of a fluid-film bearing varies inversely as the bearing clearance cubed. An increase of the inner and outer bearing ring clearances by even small amounts may have a dramatic and drastic effect on the performance of the turbocharger. The inner clearance of the turbine bearing must be maintained to less than 2 mils and the outer ring clearance to less than 6 mils in order to restrict the conical whirl motion of the turbocharger. Excessive inner and outer ring clearances of the turbine bearing will lead to rubbing and failure of the turbocharger.

4. **Increased clearances in the compressor and inducer section**. The compressor design of all turbochargers is an open bladed design. From an aerodynamic standpoint, the closer the clearance of the blades to the casing, the greater reduction of leakage and greater efficiency of the compressor wheel. The current clearance values of the turbocharger, particularly those in marine operation, are too tight for reliable operation. The original clearance gaps between the compressor and inlet inducer blades and the casing are reduced by the centrifugal growth of the wheel at high speed. This causes an addition reduction in the operating clearances.

A more serious operating situation occurs with the marine diesel engines. These engines operate without speed control on the turbocharger. Therefore the turbocharger may exceed its design speed of 30,000 RPM. As we exceed speeds of 30,000 RPM, we start to encounter the beginning of the third critical speed with bending of the shaft under the compressor bearing and increased deflection of the compressor wheel.

Another serious dynamic loading that occurs with marine engines is the action of the ship over high sea waves. The pitching motion of the vessel causes a change of the gyroscopic moment. This creates an additional loading, referred to as a maneuver load, on the compressor wheel. This effect is quite common with high-speed aircraft engines in fighter planes. This additional moment loading on the compressor wheel, in addition to the higher operating speeds, can also induce rubs on the compressor wheel.

This particular design of turbocharger is particularly susceptible to rubs, as it has a backward whirl mode near running speed. The backward mode causes the compressor to be locked against the volute whirling backward, resulting in impeller damage or shaft failure. To produce an increased safety margin of operation, the impeller inducer section and compressor wheel should have approximately 5 mils of material removed to increase the running speed clearances. This is a small price to pay in efficiency for the improvement in reliability.

5. **Replacement of compressor bearing with a 3-lobed bearing**. The current compressor floating bush bearing design is marginal under the best of circumstances. Any attempts to add axial grooves to the inner and outer surface to make it into an axial bearing does not improve the stability, but actually somewhat degrades it. In order to generate a significant improvement in performance, a precisely ground 3-lobed bearing is preferred. In a well-designed 3-lobed bearing with 100% converging film thickness each, the film generates an active pressure profile on each lobe.

For example, with a floating bush bearing, whether it is a full design or has axial grooves, if the shaft is centered in the bearing, no pressure is generated. A film pressure is only generated when the shaft is eccentric to the ring. However, with the 3-lobed bearing, positive pressure at each lobe is generated with the shaft operating under zero eccentricity. This produces a very stable rotor and further minimizes the occurrence of rubbing of the compressor wheel.

While the cost of this type of bearing may not be warranted for automotive turbochargers, it is justified for this class of turbocharger with its much higher initial cost and improvement in reliability.

6. **Replacement of turbocharger**. If one does not wish to follow the above recommendations for turbocharger improvement, then the only course left at one's disposal is to completely replace the turbocharger with one of a new design with a more robust compressor bearing.

In conclusion, it is felt that the turbocharger reliability may be greatly improved by the above recommended changes. It is of interest to note that very little reliable experimental data has been obtained on these turbochargers. Most of the data is usually taken corresponding to running speed with the lower frequencies filtered out. However, it is the lower frequencies which represent the subsynchronous precession of the turbocharger which are of importance. It is recommended that vibration monitoring of the turbochargers be conducted to determine the magnitude and frequencies of the subsynchronous whirl motion.

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