

FAILURE ANALYSIS OF 2 LITER ENGINE TURBOCHARGERS

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ABSTRACT

This paper presents a study of the failure of several 2 liter four-cylinder engine turbochargers. The failed turbocharger components are presented to show the damaged components. Detailed computer simulations are also presented to explain the nonlinear dynamical behavior of these turbochargers and to explain the causes of failures with these high speed turbochargers.

The standard bearing type employed for the majority of diesel and automotive turbochargers is the floating bronze bushing bearing. This bearing type has been well researched for its dynamical characteristics. This bearing type was first introduced in the 1930's by GM with the goal of reducing bearing friction losses.

From a linear stability standpoint, the floating bush bearing is inherently unstable at all speeds. The turbocharger first and second critical speed modes may be excited. The excitation of these modes creates distinct audible tones. All of the turbochargers utilizing floating bush bearings operate with subsynchronous whirling over the entire speed range.

The floating bush bearings are able to operate with bounded limit cycle motion due to the nonlinear forces generated by the floating bush inner and out ring pressure profiles.

Failure is essentially caused from the excitation of these unstable subsynchronous modes. It was determined, in particular, that the compressor end bearing was under lubricated and overloaded in all failed turbochargers. Some amounts of compressor inner bearing clearance wear will lead to impeller contact with the volute.

Keywords: Turbochargers, floating bush bearings, subsynchronous whirling, turbocharger acoustic signature

BACKGROUND AND INTRODUCTION

The application of turbochargers is well over 100 years in use and in development. For example, very few diesel engines operate today that are not turbocharged. The advantage of the turbocharger is that by using the exhaust engine gas to drive a turbine, a connected compressor can be used to increase the air pressure and intake volume into the engine. Increasing the amount of air into an engine allows for more fuel to be added and hence generates more horsepower with a smaller engine with increased fuel efficiency.

There is now an extensive trend in the automotive industry to move to the smaller four cylinder engines that are turbocharged. For example, the popular 3 liter six cylinder engines in many models have often been replaced by a 2 liter turbocharged four-cylinder engine. There are currently a wide variety of models offered on the market that have a turbocharged four-cylinder engine. However, the problems with these engines are of such significance that it has even come to the attention of several independent car reviewers. Reliability problems and performance response have been mentioned in a number of their articles.

The basic design of the majority of diesel and automotive engines is the floating bush bearing. As turbochargers have increased in speed, they become smaller in size. The smaller floating bush bearings become more difficult to lubricate and protect from excessive wear. In conventional turbochargers, the third bending critical speed is designed to be above running speed. With these high speed turbochargers that operate at speeds exceeding 200,00 RPM, not only is the third critical speed in the operating speed range, but also the first torsional critical speed as well. Excessive compressor bearing wear was observed in all failures.

TURBOCHARGER COMPONENTS

Figure 1 represents a typical double overhung turbocharger as shown below:

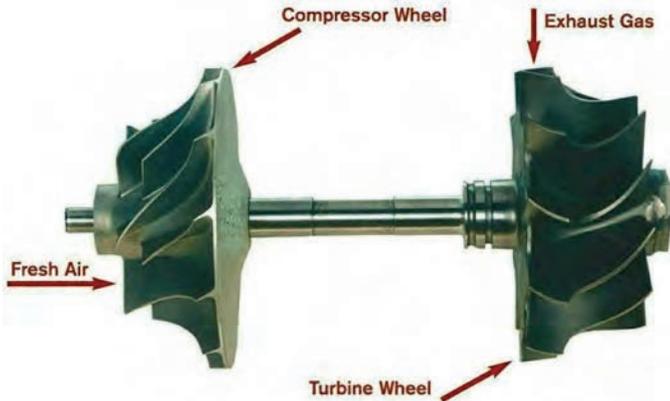


FIGURE 1 TYPICAL TURBOCHARGER DESIGN

In the standard turbocharger design, the compressor and turbine wheels are overhung. The turbine wheel is steel, while the compressor impeller is aluminum. This causes the system center of gravity to be closer to the turbine. It will be seen later that this will result in some unusual rotor dynamics which will cause more forces to be transmitted to the lightly loaded compressor bearing!

Figure 2 represents the turbocharger rotor components and turbine and compressor volute.



FIGURE 2 TYPICAL TURBOCHARGER COMPONENTS
(TURBO - Turbocharger Systems J. K. Miller 2008)

Figure 3 represents the typical float bush bearings and thrust runner used in the majority of most automotive turbochargers. The two bronze bearings are identical in design. The inner clearances of the bushings are kept tight, < 1mil (~ 0.02 mm) while the outer clearance are > 3 mils (~ 0.08 mm).

Figure 4 shows a comparison for three different types of floating bush bearings. The top figure is the floating bush bearing for the four cylinder 2 L engines. This small bearing only has a diameter of 0.484 in.(12.3 mm) with a length of 0.285 in.(7.2mm). Since these turbochargers operated at speeds of over 200,000 RPM, it is estimated that the ring rotates at a speed of around 50,000 RPM. Lubrication is very difficult for these bearings. The second figure represents the floating bush bearing for a 6 cylinder engine and the last figure represents the bearing for a 7 L diesel.

engine.



FIGURE 3 Typical Floating Bush and Thrust Runner
(TURBO J. K. Miller 2008)



Figure 4 Comparison of Floating Bush Bearings For 2 , 3 and 7 Liter Engines

FAILED TURBOCHARGER COMPONENTS

Figure 5 represents a turbocharger with a damaged compressor impeller. Extensive wear marks may be observed on both the compressor and turbine end bearings. In particular the turbine end bearing shows a distinct narrow wear band due to sub synchronous whirling in the conical mode.



FIGURE 5 VIEW OF 2 L TURBOCHARGER WITH DAMAGED COMPRESSOR WHEEL AND WORN BUSHINGS

Figure 6 shows the turbo compressor inlet and air discharge passages.



FIG. 6 COMPRESSOR INLET AND DISCHARGE

Figure 7 represents the compressor volute showing large rub marks from contact with the compressor impeller.



FIGURE 7 VOLUTE SHOWING HEAVY RUB MARKS

In order to maximize the efficiency of an open bladed centrifugal impeller, the clearance space between the impeller and volute is kept as close as possible to minimize leakage. As the compressor bearing wears over time due to inadequate lubrication, the compressor bearing clearance will open up. The predominant unstable mode of the turbocharger is a conical whirl mode. Hence the small increases in compressor bearing clearance allow the impeller wheel to contact the volute causing blade damage.. The drop in efficiency then requires the turbocharger to be replaced.

Failed Compressor and Turbine Impellers

Figure 8 represents a damaged compressor impeller due to contact with the volute.



FIGURE 8 DAMAGED COMPRESSOR

If the compressor impeller experiences a hard rub against the volute, the friction torque will cause the turbine to experience a torsional shaft failure at the turbine hub as shown in Figure 9.



FIGURE 9 TORSIONAL SHAFT FAILURE AT TURBINE HUB DUE TO COMPRESSOR RUB

Damaged Compressor End Bearings

In all of the failed turbochargers examined, the compressor end bearings showed extensive scoring and bearing clearance enlargement. As previously mentioned, the turbocharger floating bush bearing inner clearance is kept very tight in order to control rotor motion. Once the bearing inner clearance tends to wear and open up, the turbocharger is subjected to large subsynchronous conical whirl motion which can cause contacting of the compressor impeller on the volute surface. This results in either damage to the compressor vanes as shown in Figure 8, or to catastrophic turbine shaft failure as shown in Figure 9.

Figure 10 shows the compressor end bearing in which the internal bearing clearance has been expanded to over 10 mils (0.25 mm) from the original specified design clearance.

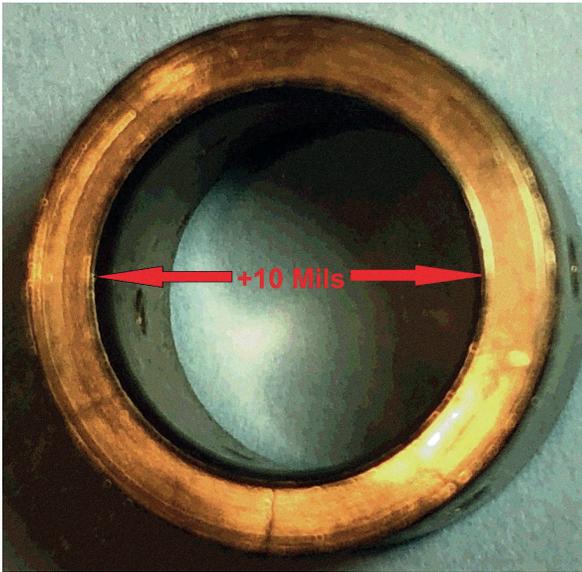


FIGURE 10 COMPRESSOR BEARING WITH ENLARGED BEARING CLEARANCE

Figure 11 represents a side view of one of the compressor end floating bush bearings from one of the failed turbochargers. All of the compressor end bearings of the failed turbochargers showed evidence of heavy rub marks.



FIG. 11 COMPRESSOR BEARING SIDE VIEW

CRITICAL SPEED ANALYSIS

In order to understand the dynamical behavior of the double overhung turbocharger in which the turbine wheel is substantially heavier than the compressor impeller, it is necessary to first start with the basic critical speed analysis of the system to understand the critical speed mode shapes and energy distribution of the bearings and shaft. Figure 12 represents a computer model of the 2 L engine turbocharger mounted in floating bush bearings. In the first phase of the analysis for the fundamental critical speeds of the system, the bearings are simply treated as linear springs. Later more advanced analysis is performed computing the bearing inner and outer film stiffness and damping coefficients and finally a nonlinear transient analysis with unbalance and Alford forces acting on the impeller wheels is computed.

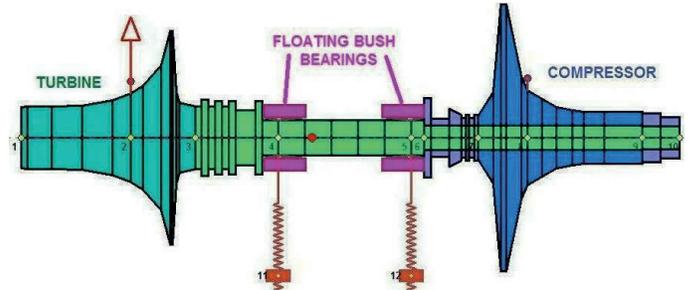


Figure 12 Turbocharger Computer Model with Floating Bush Bearings

Figure 13 represents the turbocharger first critical speed. With double overhung rotor designs, the first critical speed is a conical mode in which the turbine and compressor are moving out of phase. All the strain energy is in the bearings and there is very little shaft bending occurring in the first mode.

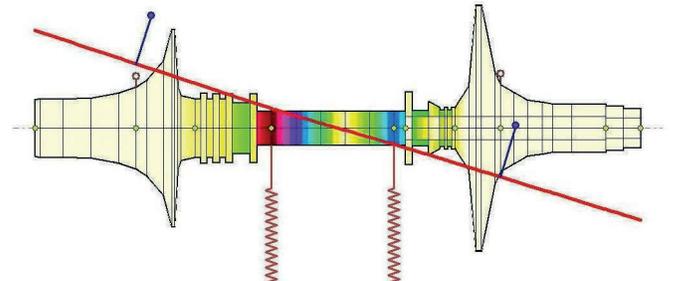


Figure 13 First Turbocharger Critical Speed

Figure 14 represents the second rotor critical speed. In this mode, the turbine and compressor are in phase. What is unusual with this second critical speed and that there is a considerable amount of bending at the shaft center.

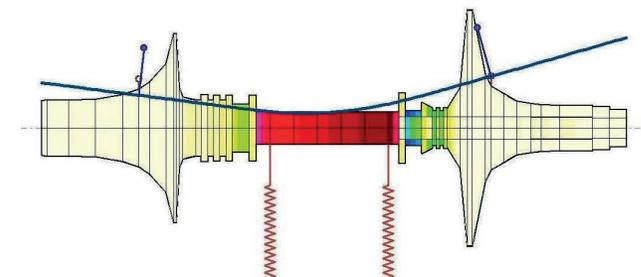


Figure 14 Second Turbocharger Critical Speed

As seen in Figure 12 for the turbocharger, the system mass center is very close to the turbine end bearing. As a result, during the second mode, the heavier turbine motion is reduced and larger amplitudes of motion are observed at the compressor end as shown in Figure 14. This explains in part why more damage is always observed on the compressor end bearings. It is the compressor end bearing that requires the larger bearing as compared to the turbine end. At speeds above the second critical speed, larger forces are transmitted to the compressor bearing. However, with all turbochargers with floating bush bearings, the bearings are always identical.

Figure 15 represents the third critical speed which is essentially a free-free bending mode. This frequency is predicted to occur at 129,000 RPM. In all turbochargers of previous generations, the third bending critical speed has always been above the running speed. The occurrence of the third bending critical speed for double overhung rotor, in the operating speed range, is highly undesirable. This third bending critical speed occurs in the operating speed range because of the extremely slender shaft used in this design in order to reduce friction losses and minimize turbocharger lag.

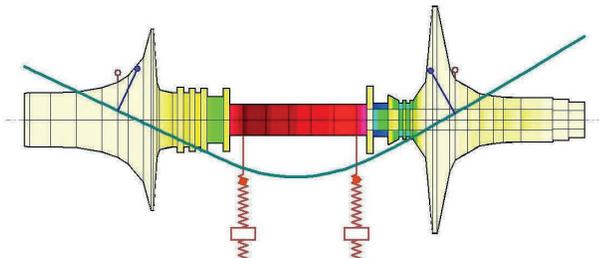


Figure 15 Third Turbocharger Bending Critical Speed at 129,000 RPM

The reason that it is undesirable to have the third bending mode occur within the operating speed range is that the bearings provide little damping to control this mode. The majority of the system strain energy resides in the shaft. Figure 16 shows the distribution of the strain energy of bending and of the bearings for this particular mode. As can be seen from Figure 16, over 90% of the system strain energy resides in shaft bending. This is a totally unacceptable situation.

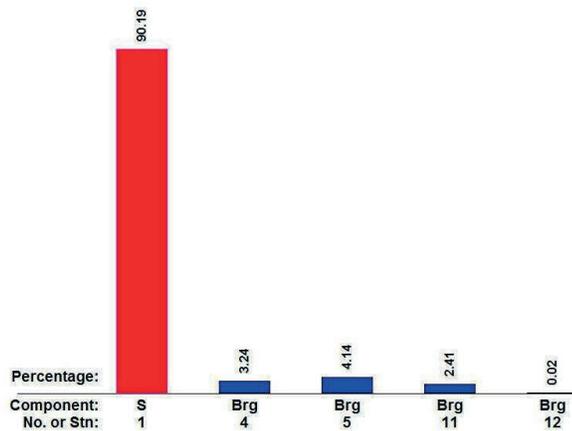


Figure 16 Shaft and Bearing Strain Energy Distribution For the Third Rotor Critical Speed- N = 129,000 RPM

TURBOCHARGER 1ST TORSIONAL MODE

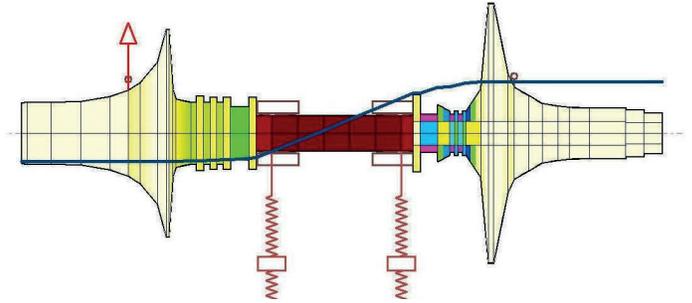


Figure 17 Turbocharger First Torsional Mode at 131,108 RPM

Figure 17 represents the turbocharger first torsional mode. The first torsional mode is computed to be at a speed of 131,108 RPM. This speed is well within the operating speed range of the turbocharger. Normally for turbochargers, the third bending critical speed and the torsional critical speed both are well above the operating speed range of the turbocharger. It is extremely unusual and highly undesirable to have both the 3rd bending critical speed and the turbocharger 1st torsional critical speed to be not only be in the operating speed range, but to closely coincide. This is a highly undesirable situation.

Unlike diesel engines for trucking applications, automotive turbochargers operate over a wide speed range. The turbocharger run up and run down through the 120,000 to 130,000 RPM speed range could cause additional wear and degradation to the compressor end bearing.



Figure 18 Backside of the Turbine Wheel Showing Torsional Shaft Failure and Extreme Balancing Corrections

Figure 18 represents the backside of the failed turbine wheel. It is apparent that the shaft failure is a torsional failure. This type of failure is initiated by heavy rub the contact of the compressor impeller on the volute. Of particular concern with the turbine wheel is extreme balancing corrections that have been applied to

the turbine wheel. Turbochargers in general employ a two plane balancing procedure. With normal turbochargers, the first two critical speed modes are rigid body conical and cylindrical modes. The third bending critical speed is usually well above the running speed. Hence two plane balancing will properly a turbocharger for unbalance response throughout the speed range.

The situation is considerably different if the third bending critical speed lies within the speed range. Two plane balancing will not be able to correct the unbalance response if the rotor is operating in the third critical speed bending mode. With the heavy balance corrections shown on the turbine wheel, the turbocharger will be highly unbalanced and will excite the third bending mode at around 120,000 RPM. In addition, this mode will have a high amplification factor because 90% of the strain energy of the system is in shaft bending and very little energy is in the bearings to help dampen this particular critical speed response. Hence additional bearing damage may occur during run up and run down of these high-speed turbochargers when passing through the 3rd critical speed in the range of 120,000 to 130,000 RPM.

TURBOCHARGER STABILITY ANALYSIS

Floating Bush Bearing Coefficients

The calculation of a stability analysis for the turbocharger requires a knowledge of the floating bush bearing coefficients. By having the bearing stiffness and damping coefficients, the complex damped eigenvalue analysis of the turbocharger may be computed.

Figure 19 represents the characteristics of the floating bush bearing. To compute the fluid film bearing characteristics generated by the inner and outer films of the floating bush bearing a number of assumptions for the floating bush bearing must be made. These assumptions are concerned with the inner and outer bearing clearances film viscosities and ring rotational speed.

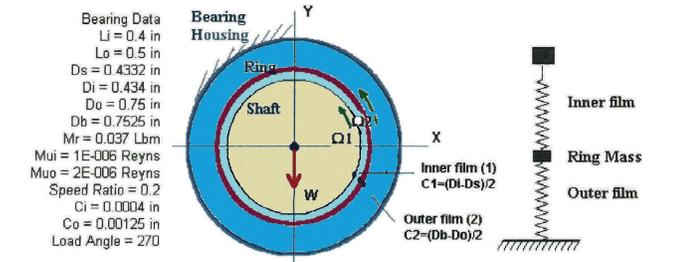
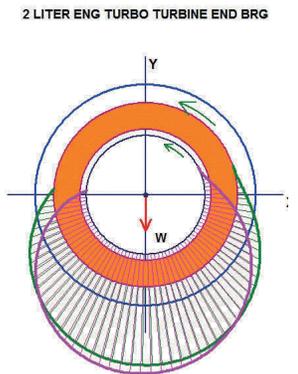


Figure 19 Floating Bush Bearing

Bearing Data
 $L_i, L_o = 0.2785, 0.2785$ in
 $D_s, D_i = 0.3125, 0.3135$ in
 $D_o, D_b = 0.4845, 0.4875$ in
 $C_i, C_o = 0.00050, 0.00150$ in
 $C_o/C_i = 3$
 $M_r = 0.007$ Lbm
 $\mu_{i1} = 9E-07$ Reyns
 $\mu_{o1} = 1E-06$ Reyns
 $Speed\ Ratio = 0.25$
 $Load = 0.2$ Lb
 Rotor RPM = 220000 rpm
 $e_j/(C_i + C_o) = 0.0164$
 $h_{min} = 0.50, 1.47$ mils
 $P_{max} = 4.26, 2.79$ psi
 $Loss = 0.5861$ hp
 $K_{ixx,xy} = 1.34E+04, 1.71E+05$ Lbf/in
 $K_{iyx,yy} = -2.40E+05, 1.22E+04$ Lbf/in
 $C_{ixx,xy} = 1.19E+01, -9.05E-01$ Lbf-s/in
 $C_{iyx,yy} = -9.05E-01, 1.67E+01$ Lbf-s/in
 $K_{oxx,xy} = 4.53E+02, 4.98E+03$ Lbf/in
 $K_{oyx,yy} = -6.29E+03, 1.43E+01$ Lbf/in
 $C_{oxx,xy} = 1.73E+00, -1.06E-01$ Lbf-s/in
 $C_{oyx,yy} = -1.06E-01, 2.19E+00$ Lbf-s/in

Figure 20 Floating Bush Bearing Stiffness and Damping Coefficients



Turbocharger Unstable Forward Modes

Figure 21 represents the first computed unstable turbocharger mode at a speed of 220,000 RPM. This mode is a conical whirl mode predicted to be at a frequency of 15,074 CPM. In this mode it is observed that the turbine and the compressor exhibit conical motion whirl in which the impellers are out of phase to each other.

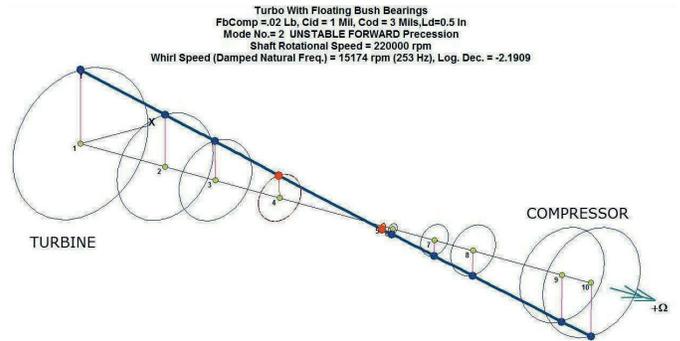


Figure 21 Turbocharger 1st Unstable Conical Whirl Mode At 220,000 RPM, Nf1 = 15,174 CPM, Log Dec = -2.2

Figure 22 shows the second unstable turbocharger mode. In this mode, the compressor and turbine are in phase. In the excitation of the second unstable mode, the heavier turbine wheel maintains a fairly small amplitude while large amplitudes of motion are generated at the compressor impeller. As previously mentioned, all examined compressor bearings for the various failed turbochargers showed considerable enlargement of the compressor end bearing inner clearance.

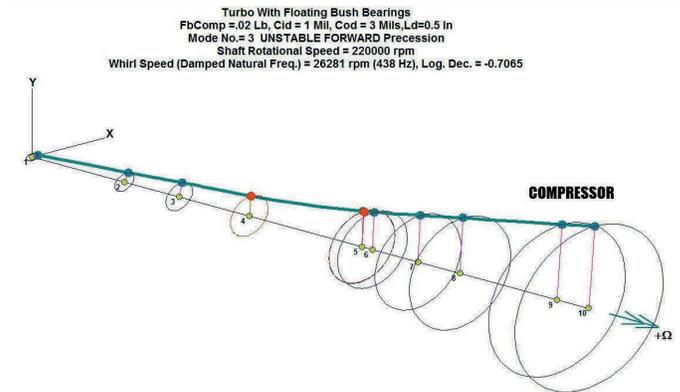


Figure 22 Turbocharger 2nd Unstable Whirl Mode At 220,000 RPM, Nf2 = 26,281 CPM, Log Dec = -0.71

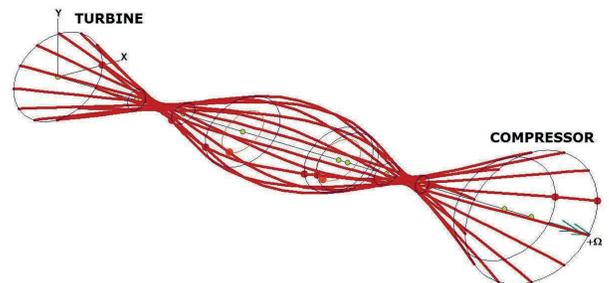


Figure 23 3rd Bending Forward Mode at 135,453 RPM

Figure 23 shows the third mode, which is a shaft bending mode, which is undesirable to have in the operating speed range.

NONLINEAR TIME TRANSIENT ANALYSIS

In order to determine the turbocharger actual amplitudes of motion, it is necessary to perform a nonlinear time transient analysis. In a time transient analysis, the equations of motion are integrated forward in time. The time transient analysis is necessary since the floating bush bearings are highly nonlinear. In the time transient analysis, several more excitation forces are assumed to be acting on the rotor system.

When the turbocharger is operating under high power levels, there are additional forces acting on the turbine and compressor referred to as the Alford forces. These forces are simulated by assuming a bearing location at the turbine and compressor locations and adding bearing cross coupling components. Unbalance components are also assumed to be acting on both the turbine and compressor locations. In addition, gravitational effects are also included.

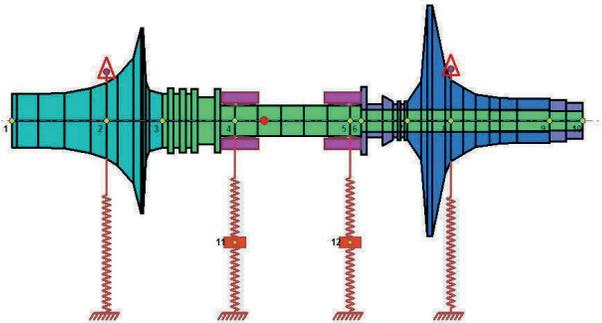


Figure 24 Turbocharger Unbalance and Alford Forces Acting on the Turbine and Compressor

The turbocharger as shown in Figure 24 has 12 master elements. The total number of degrees of freedom for the turbocharger is 44. In addition to the equations of motion for the rotor system, at each time step, the Reynolds equation for the pressure profiles of the inner and outer rings of the floating bush bearings must be computed. These pressure profiles are then numerically integrated to determine the forces acting on the floating bush and shaft. A Wilson Theta method of numerical analysis was chosen with 500 steps per cycle for 50 cycles of shaft motion.

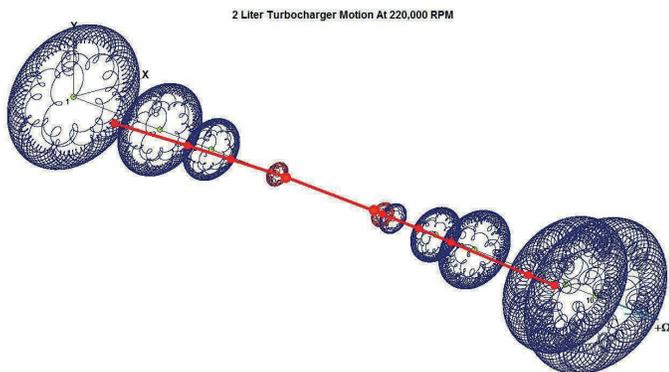


Figure 25 Time Transient Analysis of Turbocharger at 220,000 RPM With Impeller Unbalance and the Alford forces

FFT Analysis of Transient Motion

Figure 26 represents an FFT spectrum analysis of the nonlinear time transient motion of the turbine at station 1 at 220,000 RPM. The complex motion is composed of synchronous motion due to the effect of unbalance and the excitation of the lower system modes. It is also apparent that both the first and second turbocharger critical speeds may be excited by the Alford forces acting on the turbine and compressor.

Under high power levels, the turbocharger will emit a very high audible sound which is generated due to the excitation of the rotor first or second critical speeds. These audible frequencies cannot be removed by simply improving rotor unbalance. These are subsynchronous whirl modes which are due to the fundamental instability of the turbocharger employing floating bush bearings.

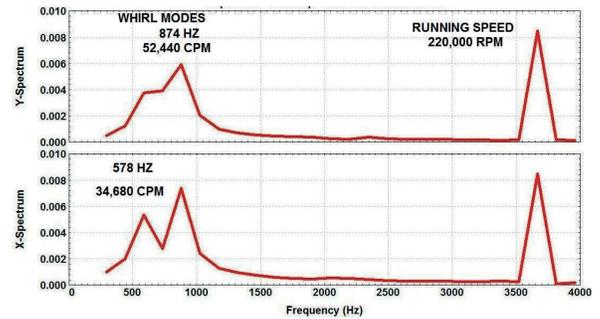


Figure 26 FFT Spectrum Analysis of Turbine Motion

Bearing Forces Transmitted

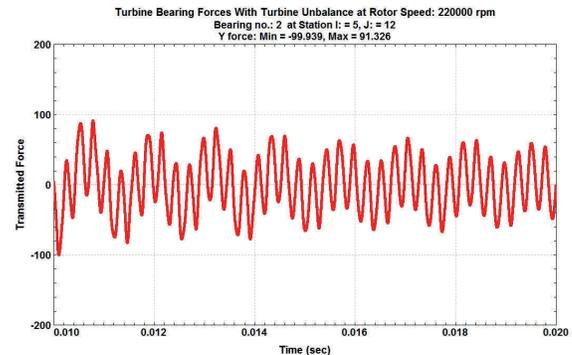


Figure 27 Turbine Bearing Forces Transmitted With Turbine Unbalance

Fig 27 represents the turbine bearing force transmitted with 0.001 oz-in of the turbine wheel unbalance. The maximum forces transmitted is shown to be approximately 90 pounds. Observation of the various failed turbochargers showed that the turbine wheels often had heavy unbalance corrections applied to the wheels.

Figure 28 represents the forces transmitted to the compressor end bearing with unbalance applied at the turbine wheel. It is of interest to note that the maximum force transmitted to the compressor bearing is 150 pounds, which is over 50% higher than the bearing forces transmitted to the turbine bearing. This seems to defy intuition that unbalance place near the turbine bearing

would cause higher forces to be transmitted to the compressor bearing. The reason for this is due to the dynamical characteristics of the turbocharger in which the turbine wheel is substantially heavier than the aluminum compressor impeller. After the rotor pass through the first critical speed which is a conical mode, the heavier turbine wheel forms a very stationary node point and causes higher amplitudes of motion to occur at the compressor bearing.

This helps to explain why the compressor bearing wear is always substantially greater than what has been observed with the turbine bearings. After passing through the first critical speed, the turbocharger amplitudes of motion are always higher at the compressor location as shown in Fig. 22.

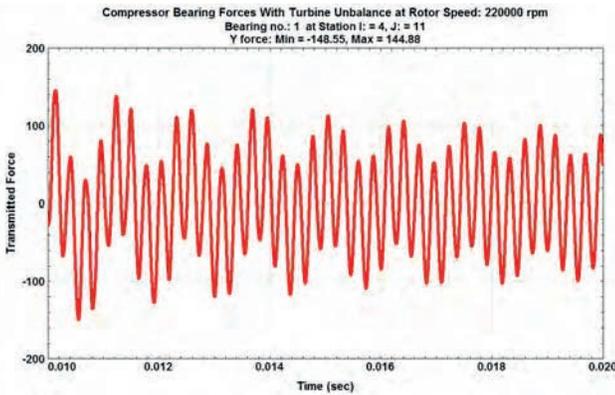


Figure 28 Compressor Bearing Forces Transmitted With Turbine Unbalance

Compressor Bearing Shaft Motion with Bearing Wear

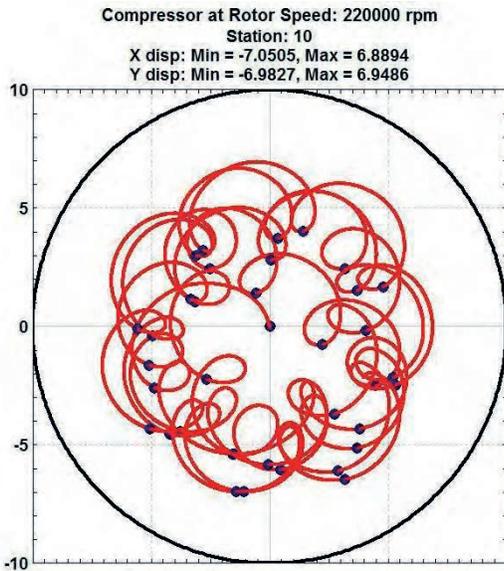


Figure 29 Shaft Motion with One Mil Cd

Figure 29 represents the compressor bearing shaft motion with one

mil (0.25 mm) diametrical clearance. The orbital motion combines the non-synchronous whirling in the synchronous motion due to the small amount of unbalance at the turbine wheel. Note that the total orbit fills up 70% of the bearing clearance.

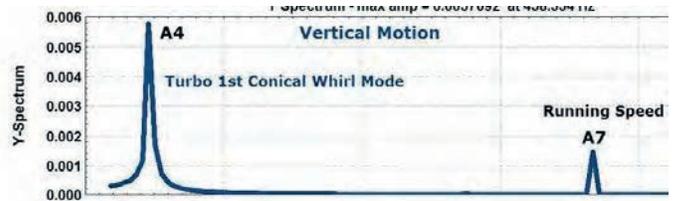


Figure 30 Spectrum of Compressor Bearing Motion Showing Strong Excitation of the Turbocharger First Critical Speed

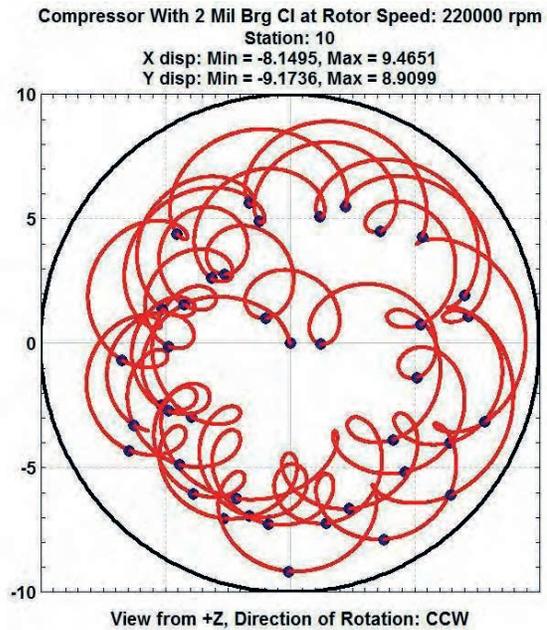


Figure 31 Shaft Motion with Two Mils Cd

Figure 31 represents the shaft motion in the compressor bearing with the clearance increase from 1 to 2 mils (0.05 mm). In order to maintain control of the turbocharger, the compressor inner bearing clearance must be kept extremely tight. As previously mentioned, all turbochargers with floating bush bearings are unstable and will excite the lower system modes. In order to operate, the turbocharger must rely on the nonlinear bearing forces in order to generate controlled limit cycle motion.

It is seen in Figure 31, that an increase of clearance from 1 to 2 mils now causes the rotor to operate in almost 90% of the bearing clearance. It was previously mentioned and shown in Figure 10 that one failed turbocharger compressor bearing had the internal clearance enlarged to over 10 mils.

Since the predominate turbocharger whirl mode is a conical excitation of the first critical speed, any increase in the compressor bearing clearance will allow the compressor impeller to contact the volute resulting in impeller damage

TURBOCHARGER FAILURE MODES

There are a number of ways in which the turbocharger may fail and will require replacement. The first mode of failure is due to the degradation of the compressor vanes caused by rubbing of the compressor on the volute.

In order for the turbocharger to be able to operate with stable limit cycle whirl motion, the internal bearing clearances must be kept very tight. As has been previously discussed, the compressor bearing clearance over time, continues to wear and enlarge. As the internal compressor end bearing clearance enlarges, the subsynchronous conical whirl amplitude of motion continues to increase, causing the impeller to contact the volute. This causes damage to the compressor vanes and reduces performance. The reduced turbocharger output then requires that the turbocharger be replaced.

The second class of turbocharger failure is more dramatic, as it is sudden, without warning, and fatal. This is the "*heart attack mode of failure*." The first type of sudden death failure is associated with excessive wear in the compressor end bearing internal clearance. As has been previously mentioned earlier, it was observed that one of the turbochargers that had a fatal heart attack failure (failed turbine shaft), had its compressor end bearing clearance enlarged to over 10 mils (0.25 mm)! When the compressor bearing clearance becomes larger than two mils (0.05 mm), it is possible for the compressor impeller to contact and rub against the volute. The compressor impeller spinning at 220,000 RPM has properties very similar to a gyroscope.

If a hard rub on the casing is encountered from the conical motion of the impeller, then the gyroscopic moment will act to cause the impeller to precess backwards. When this occurs, the drive torque on the turbine causes the shaft to shear off at the hub turbine, as seen in Figures 9 and 18. This type of failure often occurs when the turbocharger is operating under high power levels, such as accelerating up an incline. The additional Alford forces acting on the impellers causes a high amplitude subsynchronous conical whirl motion as shown in Figure 25. This motion also creates an audible sound.

There is another mode of instant failure of the turbocharger which is associated with the turbine end bearing. In all of the failed turbochargers, evidence of heavy coking was observed at the turbine bearing locations. Because of the high temperatures of the exhaust gas in the turbine area, heat shielding is required for the turbocharger. It is possible for the high temperature to cause oil coking, resulting in the turbine floating bush bearing to seize to the shaft. When this occurs, we have a large clearance cylindrical bearing which is unstable at 6,000 RPM. If the turbine bush bearing should seize to the shaft, total shaft failure occurs within a few cycles of shaft motion. This is another reason why only the highest grade of synthetic lubricant should be employed in these turbochargers.

The common assumption in the field is that turbocharger failure is due to such things as contaminated oil or ingestion of foreign objects into the intake of the turbocharger. These reasons account for a minority of turbocharger failures.

DISCUSSION AND CONCLUSIONS

This paper presents an analysis of the failure of several 2 liter engine turbochargers that are currently employed in a number of four-cylinder engines. A detailed computer model for the turbocharger was generated to explain the dynamical behavior of these turbochargers and to explain the process in which they may fail.

The 2 L turbocharged four-cylinder engine design has become very popular in many of today's modern automobiles. The basic bearing used in the majority of diesel and automotive engines is the floating bush bearing. The floating bush bearing is free to rotate and allows limit cycle motion for the turbocharger.

All turbochargers with floating bush bearings are inherently unstable and exhibit subsynchronous whirling. The high noise level or sounds emitted by these turbochargers reflect the subsynchronous tones generated under high amplitude whirling.

There are significant problems with these high-speed 2 L engine turbochargers which operate at speeds up to 220,000 RPM. At these high-speed, it is difficult to maintain proper lubrication, particular to the compressor end floating bush bearing

One of the concerns with the design of these turbochargers is to minimize turbocharger lag. It is therefore desirable to keep the bearing friction losses to a minimum. In the examination of all failed 2 L engine turbochargers, significant wear was observed in the compressor end bearing clearance. Basically the compressor end bearing is overloaded and under lubricated. To improve turbocharger reliability, the compressor end bearing, in particular, must be redesigned for improved lubrication and stability.

There is another significant concern and that is the very thin shaft connecting the turbine and compressor is such that it allows the third bending critical speed to be in the operating speed range. The torsional critical speed is also at a similar speed to the third bending critical speed. This is an undesirable feature as passing through the third bending critical speed causes additional loading and wear, particularly on the compressor end bearing.

In summary, considerable redesign of the compressor bearing and shaft is required to promote a level of reliability similar to that of aspirated three liter six cylinder engines.

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Resume

Edgar J. Gunter, PhD

Dr. Gunter is a Prof. Emeritus of Mechanical and Aerospace Engineering at the University of Virginia. He received his Mechanical Engineering degree from Duke University and Masters and PhD degrees from the University of Pennsylvania in Engineering Mechanics.

He was employed as a centrifugal compressor design engineer for four years at Clark Brothers, Olean, New York, now a division of Dresser-Rand. Based on his compressor design projects, he was awarded a National Defense Fellowship to pursue the PhD degree in Engineering Mechanics.

During his graduate studies, he received an internship with the SKF Ball Bearing Research Center to study the fatigue life of rolling element bearings. In his graduate program, he majored in applied mechanics, vibration and dynamics, fluid mechanics and lubrication theory.

After completing his formal training at the University of Pennsylvania, he assumed the position of Senior Research Scientist at the Franklin Institute Friction and Lubrication Laboratories and was in charge of research in the Gas Bearing Division. While at the Franklin Institute, he received a NASA Lewis Research Center Grant to study rotor-bearing stability. The study was initiated since at that time the Franklin Institute had some of the world's largest digital and analog computers at the Institute. The report on Rotor Bearing Stability was published by NASA as a special CR report and received national distribution. This report formed the basis of his PhD dissertation.

Upon receiving his formal PhD degree, Dr. Gunter was then offered the position of Associate Professor of Mechanical and Aerospace Engineering at the University of Virginia. At the University of Virginia, he developed the Rotor-Bearing Dynamics Laboratory to assist industry in the development of reliable high-speed rotating equipment.

In 1999, Dr. Gunter formed the software and consulting company, **RODYN Vibration Analysis Inc.**, for consulting services in the field of rotor-bearing dynamics. **RODYN** is the principal distributor of the rotor-bearing dynamics software *Dyrobex*, developed by Dr. Wen Jeng Chen of *Eigen Technologies*. This software is in use by the major turbomachinery manufactures in this country and abroad.

Dr. Gunter has been researching the dynamics of high speed turbochargers for the last 15 years for applications in the field of automotive and large marine diesel engines. He has written numerous papers on the subject of the dynamics of high speed turbochargers and squeeze film dampers for aerospace applications. Additional papers on the subject of rotor-bearing dynamics by Dr. Gunter and other researchers may be found on the company website, www.RODYN.com

He has been elected to the following honorary engineering societies of Pi Tau Sigma, Tau Beta Pi and Sigma Xi. He has been elected as a Fellow of ASME.

In 2007, Dr. Wen Jeng Chen and Dr. Gunter published the 409 page textbook entitled "*Introduction To Dynamics of Rotor Bearing Systems*". This reference book is currently being used as a textbook in both undergraduate and graduate courses in rotor-bearing dynamics, both in this country and abroad.

In 2008, Dr. Gunter was awarded the first Jack Frarey Memorial Metal by the Vibration Institute for contributions to the field of rotor-bearing dynamics.