Review and Failure Analysis of Three 4 Cylinder Engine Turbochargers

And Methods on How to Extend Turbocharger Life

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Abstract:

This paper concerns the review and failure analysis of three high speed turbochargers specifically designed for operation in 2 liter, four cylinder engines. In recent years, the use of turbocharging has become extremely popular, with applications ranging from large diesel engines, to various eight, six and four cylinder engines. The use of these turbochargers has greatly expanded, particularly for application to four-cylinder engines. The concept, then is to be able to replace an eight or six cylinder, heavier engine with a lighter weight, turbocharged four-cylinder engine with equivalent horsepower and performance.

One of the concerns with turbocharging is the problem of reducing turbo lag. This is the delay often experienced by turbochargers in coming up to speed, producing an irregular acceleration. This has often led to expensive, sophisticated configurations such as installing twin turbochargers, or bi-turbos, in which two turbochargers are installed with different capacities in order to minimize the turbo lag.

In an effort to reduce turbocharger lag, the modern turbocharger used in these four cylinder 2 L engines can operate at speeds of over 200,000 RPM. They are designed from an aerodynamic standpoint to have minimum impeller inertia, small shaft diameters, low bearing friction, and to operate at extremely high speeds. They are not designed from a rotor dynamics standpoint to be robust and operate stability.

The standard fluid film bearings used in these turbochargers, ranging from the very large diesel engines to the ultra high-speed turbochargers running at over 200,000 RPM, are called the floating bush bearings. These bearings are allowed to spin freely on the shaft. They were originally developed by GM in the 1930s to minimize bearing friction losses. These bearing are inexpensive to produce, but however, they are totally unsuitable for use in high-speed turbochargers operating at speeds in excess of 200,000 RPM.

The reason for this is that floating bush bearing turbochargers are inherently unstable at all speeds! This means that the rotors can become unstable with nonsynchronous whirling motion in either the first or second rotor critical speed modes. The first and predominate mode is a conical mode in which the compressor impeller is whirling out-of phase to the turbine wheel. In the higher second mode, the compressor impeller is whirling in-phase with the turbine impeller, but with a larger amplitude of motion. Since the steel turbine impeller is heavier then the aluminum compressor, the amplitudes of motion are higher at the compressor bearing, causing enhanced wearing of the bearing inner clearance. These modes, which can operate with high amplitudes of subsynchronous whirl motion, eventually cause significant bearing wearing damage.

A common misconception, for many years, is that turbochargers failed to due to ingestion into the turbocharger. This is a very rare occurrence. The principal reason for the failure of all turbochargers is due to the initial wear encountered with the compressor end bearing. As this bearing clearance opened up due to wear, the impeller may contact the volute causing blade damage. With a very severe rub, the impeller will whirl backwards, like a top causing the turbine wheel to snap at the shaft.

A secondary cause of turbocharger failure is with the occurrence of coking occurring with the turbine end bearing. The turbine end bearing encounters a considerable amount of heat. If the floating bush turbine bearing should seize to the shaft because of its tight clearance then the outer surface essentially becomes a large clearance plain journal bearing. A large clearance journal bearing is unstable at speeds of 5000 RPM. When this situation occurs with the turbine bearing seizing to the shaft, turbocharger failure is instantaneous. A high quality synthetic lubricant is required for these turbochargers to operate.

Resume

Edgar J. Gunter, PhD

Dr. Gunter is a retired Prof. 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 fatigue life of rolling element bearings. In his graduate program, he majored in applied mathematics, 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 in charge of the Gas Bearing Division. While at the Franklin Institute, he received a NASA Lewis Research 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 given 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 tenured Associate Prof 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.

He has been elected to the following honorary engineering societies of Pi Tau Sigma, Tau Beta Pi and Sigma Xi. He was elected as a fellow of ASME in 1996.

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

2 Liter Engine Turbocharger Failure Analysis

Edgar J . Gunter, PhD

1 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 auto industry to move to the smaller 4 cylinder engines that are turbocharged. For example, the popular 3 liter six cylinder engines in many models have often been replaced by a 2 L turbocharged four-cylinder engine. There are currently over 95 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 Consumer Reports. Reliability problems and performance response have been mentioned in a number of their articles.

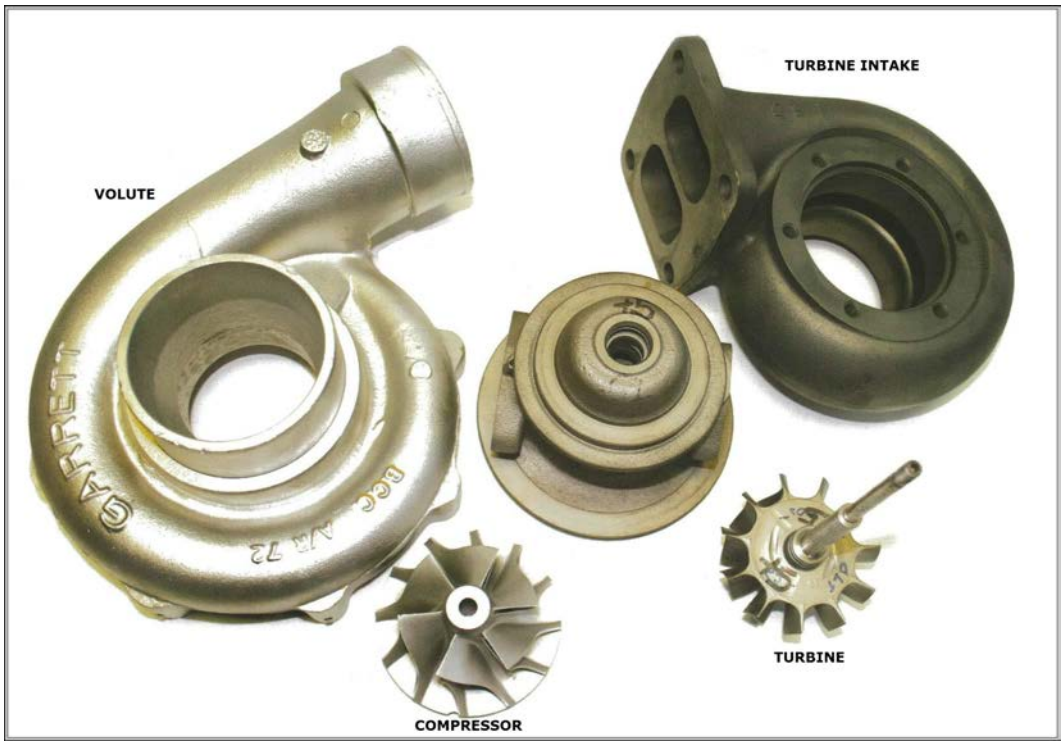


Fig. 1.1 Typical Turbocharger Components
(TURBO - *Turbocharger Systems J. K. Miller 2008*)

The basic components of the turbocharger are shown in the above figure by J. K. Miller in his practical applied book on turbocharger performance. Not shown in this figure is the wastegate that controls the volume of exhaust gas into the turbine section of the turbocharger. The main engineering effort of the turbocharger manufacturers however rests in the efficient design of the compressor and turbine wheels.

1.2 Background and Introduction - Turbocharger Bearings

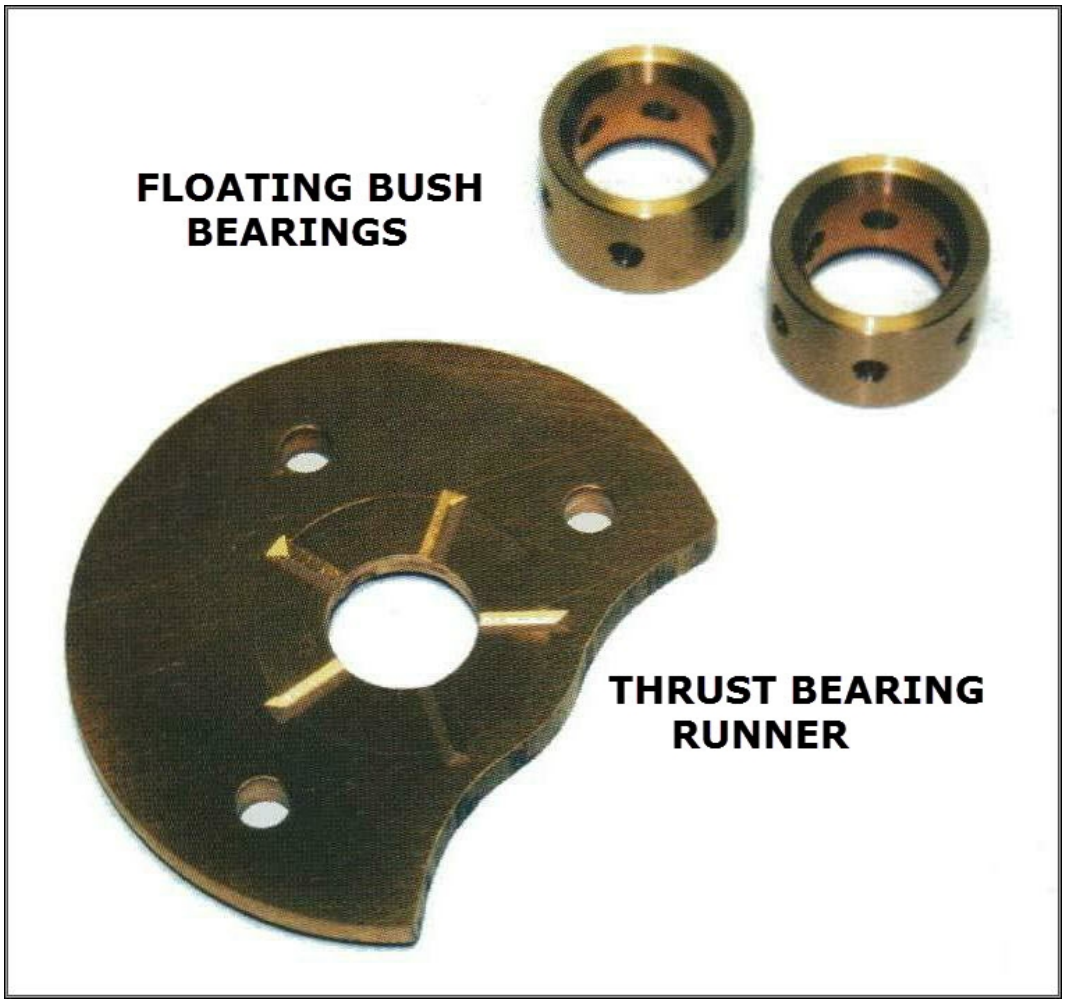


Fig. 1.2 Typical Turbocharger Floating Bush Bearings and Thrust Runner
(TURBO - *Turbocharger Systems J. K. Miller 2008*)

Figure 1.2 represents the typical floating bush bearings and thrust runner used in most automobile turbochargers. The bronze bushing bearings are the most common bearing type used in automobile turbochargers. These bearings are inexpensive and easy to fabricate. They are installed to freely rotate on the shaft.

The bearing design is such that the inner bearing to shaft clearance is very tight and the outer clearance between the bearing and the casing is more generous. The clearance ratio averages about three times the inner bearing clearance. The rotation of the high-speed shaft causes the ring to rotate about 20 to 25% of the shaft speed. This creates an external bearing which generates additional stiffness and damping coefficients acting on the bearing ring.

Very little effort has been given by the turbocharger manufacturers towards more advanced bearing designs such as the multi-lobed, pressure dam, tilt pad or foil bearings. Design of the turbocharger bearings are of a secondary consideration in relationship to the extensive design of the impellers to ensure maximum turbo efficiency.

As turbochargers have increased in speed, they have become smaller in size. The smaller floating bush bearings become more difficult to lubricate and protect from excessive wear.

1.3 Background and Introduction - Comparison of Turbocharger Bearings

Fig. 1.3 Comparison of 2 L 4 Cylinder Engine Bearing to 3 L Diesel Engine Bearing

Figure 1.3 shows a comparison of the small 2 L floating bush bearing in comparison to the larger 3 L diesel engine bearing. The diesel engine runs at a much slower speed and as can be seen has a much more generous bearing. Note also in the case of the diesel engine floating bush bearing, that there is a central groove in order to ensure that sufficient oil is fed to the inner bearing surface.

The operation of diesel trucks is considerably different than that of turbocharged automobiles. The diesel engines are operated at a constant speed and the engines are normally kept at idle for long periods of time when stopped to ensure proper turbocharger bearing lubrication.

Unless these poorly lubricated, unstable, unreliable 2L floating bush turbocharger bearings are replaced by a more robust design, these four-cylinder engine configurations will eventually be replaced by hybrid designs which offer better reliability and excellent mileage.

1.4 Background and Introduction - *Turbocharger Failure*

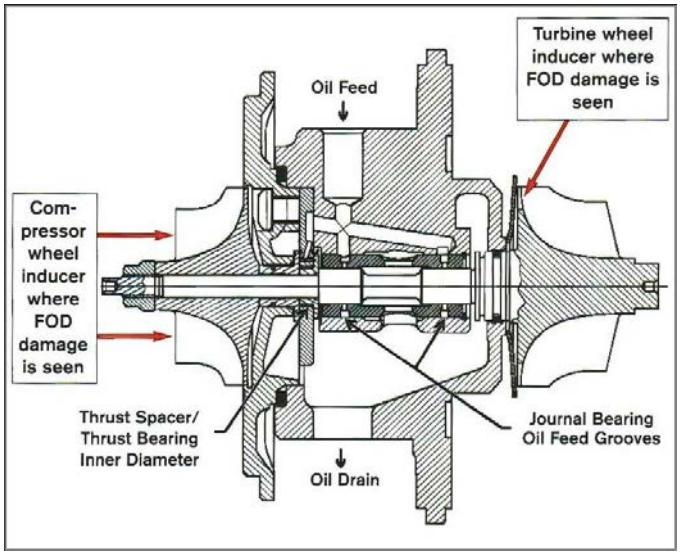


Fig. 1.4 Turbocharger Cross Section
(TURBO - *Turbocharger Systems J. K. Miller 2008*)

Figure 1.4 represents a schematic diagram of a turbocharger. The causes of turbocharger failures are not well understood in the industry. As an example in this figure, it assumes that foreign objects ingested into the turbine or compressor are one of the main causes of turbocharger failure. The main cause of turbocharger failure is the use of the inexpensive bronze floating bush bearings as shown in Figure 1.2.

These inexpensive bronze bearings are standard in most automotive turbochargers. The floating bush bearings, however, causes unstable whirling motion at all speeds, which cannot be corrected by the specification of the bearing inner or outer clearances. This instability is referred to as self-excited nonsynchronous precession in which either the rotor first or second critical speeds are excited during running speed.

The first unstable mode is a conical whirl mode corresponding to the rotor first critical speed. The second unstable mode is a cylindrical mode. Depending upon the rotor speed, either one of these modes, or both, may exist. The principal cause of failure with these small ultra high-speed turbochargers is associated particularly with the compressor end bearing. This bearing, in particular, is overloaded and under lubricated. Because of the fact that the turbine wheel is significantly heavier in weight than the aluminum compressor, the mode shapes are such that the largest amplitudes of motion and forces transmitted occur at the compressor bearing location.

The turbocharger itself may be considered as two high speed gyroscopes connected by a thin rod. Due to the unstable motion of the turbocharger and the lack of proper lubrication for the compressor bearing, internal bearing wear occurs. This wearing motion allows the impeller wheel to rub against the volute. When this occurs, the gyroscopic behavior of the impeller causes the turbo to go into a violent backward precession. The sudden change of torque on the impeller causes the shaft to snap at the turbine location.

Thus, with these turbochargers, it is not a matter of will they fail, but when this will occur. The failure of these turbochargers is dependent upon the quality of the synthetic lubricant and also on the operating conditions that the turbocharger is subjected to.

2 Review of 2 Liter Engine Turbocharger Failed Components

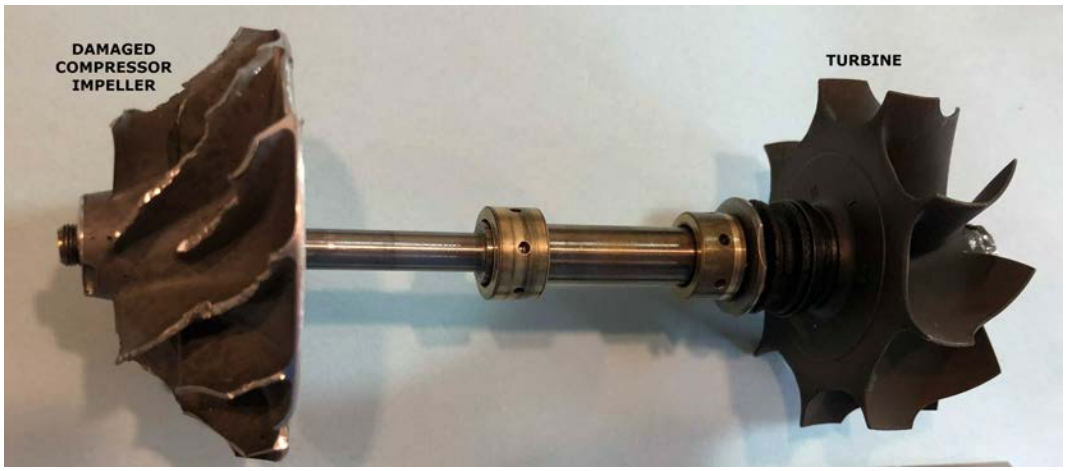


Fig. 2.1 View of 2 L Turbocharger With Damaged Compressor Wheel

Figure 2.1 shows the 2 L engine turbocharger with a badly damaged compressor impeller due to heavy volute rubs. Extensive wear marks may also be observed on the bearings.

Fig. 2.2 Closeup of Turbine, Floating Bush Bearings and Oil Seal Area

Figure 2.2 shows a closeup of the floating bush bearings and heavy coking in the turbine oil seal area. The heavy narrow band wear ring on the turbine side bearing is a strong indicator that the turbocharger was precessing in a conical whirl orbit. This conical whirling tendency of all turbochargers will be discussed in further detail in the next section on analysis. Measurements of the compressor end bearing clearance showed a large increase in internal bearing clearance due to wear. This large increase in clearance allowed the compressor wheel to rub on the volute.

2 Review of 2 Liter Engine Turbocharger Failed Components - Volute

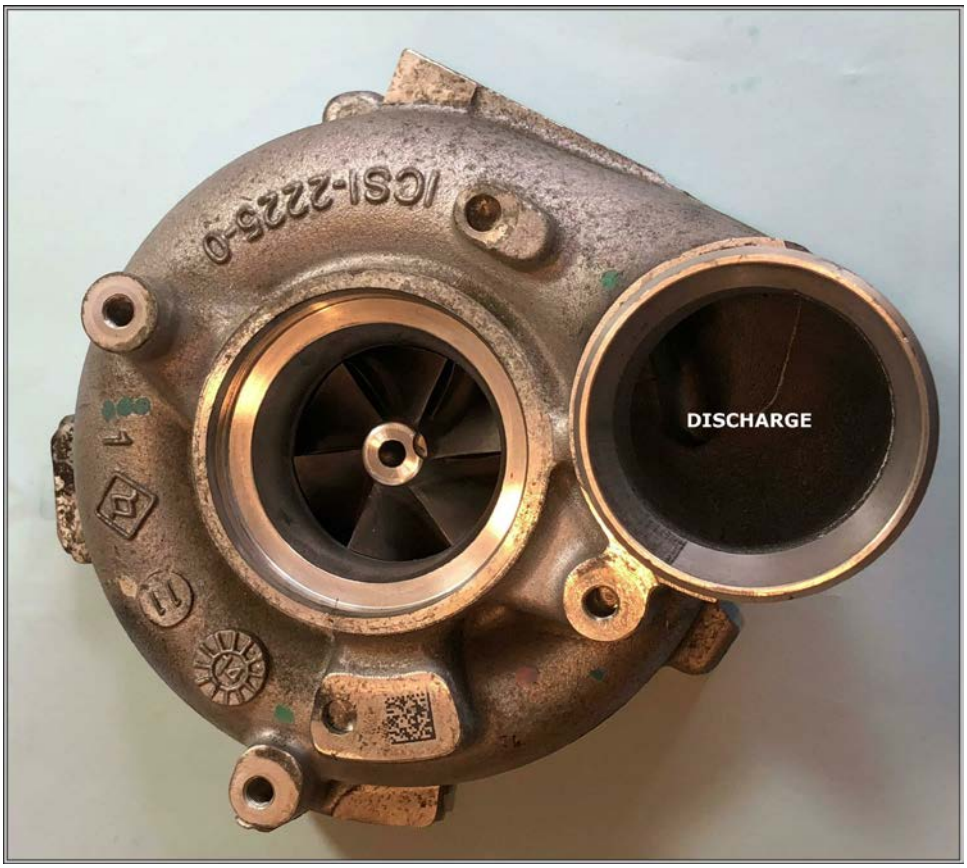


Fig. 2.3 Turbo Inlet Showing Compressor and Air Discharge Passage

Fig. 2.4 Compressor Volute Showing Heavy Impeller Wear Marks

2 Review of 2 Liter Engine Turbocharger Failed Components - Turbine

Figures 2.3 and 2.4 represent the front and back view of the compressor volute. In Figure 2.4, it is clearly seen that the impeller has caused heavy rub marks on the volute casing. When heavy rubbing occurs with the compressor wheel against the volute casing, the torque on the turbine power wheel may cause the turbine shaft to snap as clearly seen in Fig 2.5.

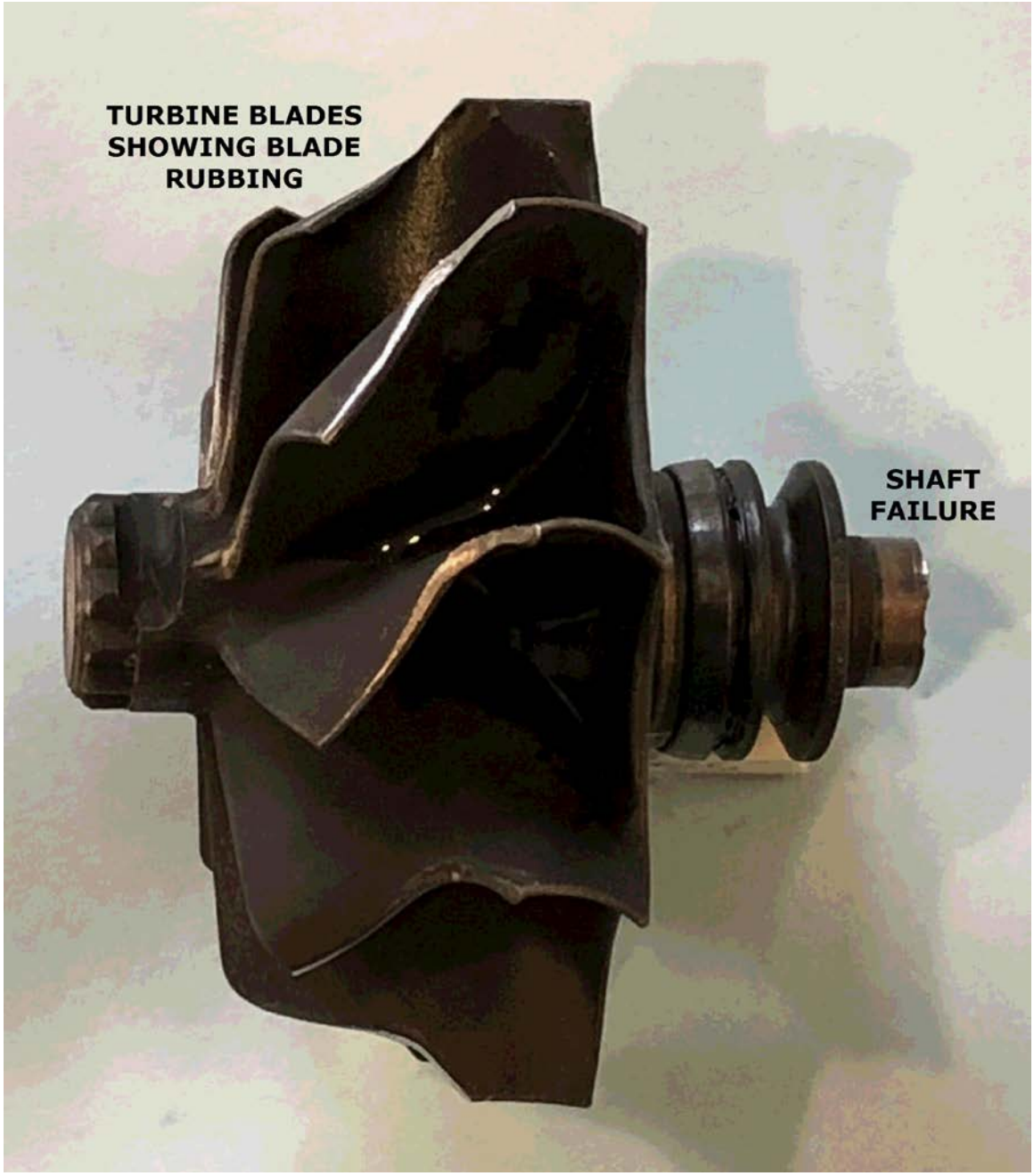


Fig. 2.5 Power Turbine Showing Blade Rubs and Shaft Failure

The high rub marks on the turbine blades may actually have occurred after the compressor wheel has had a heavy rub and the turbine shaft has sheared. The power turbine rubbing may have occurred after the shaft failure at the turbine location. This is very common with the occurrence of heavy compressor rubs. The shaft at the turbine location shears first and then the heavy rubbing on the turbine wheel is encountered.

2 Review of 2 Liter Engine Turbo Failed Components-Floating Bush Bearings

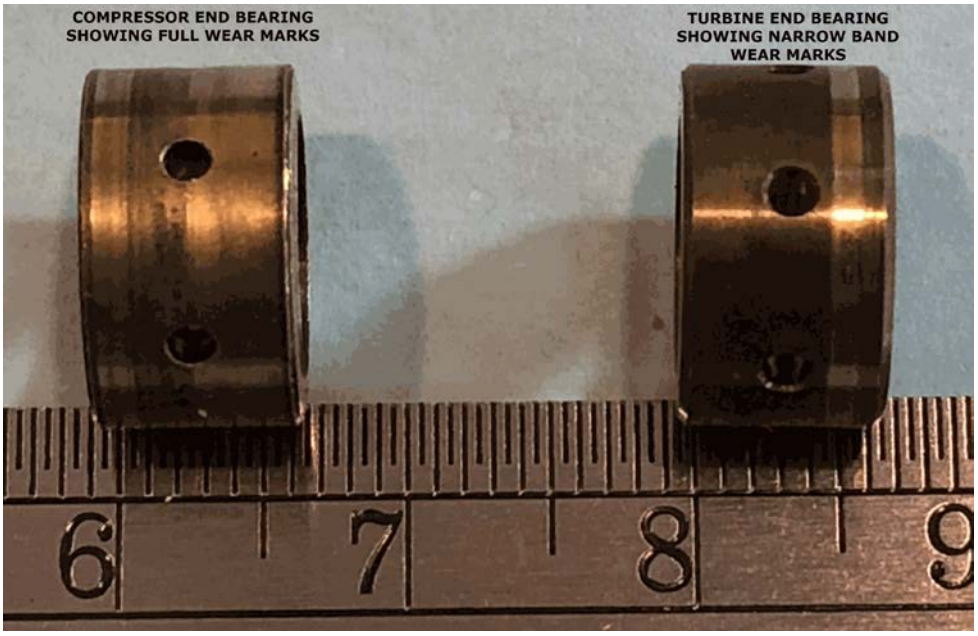


Fig. 2.6 Comparison of Compressor and Turbine Bush Bearings

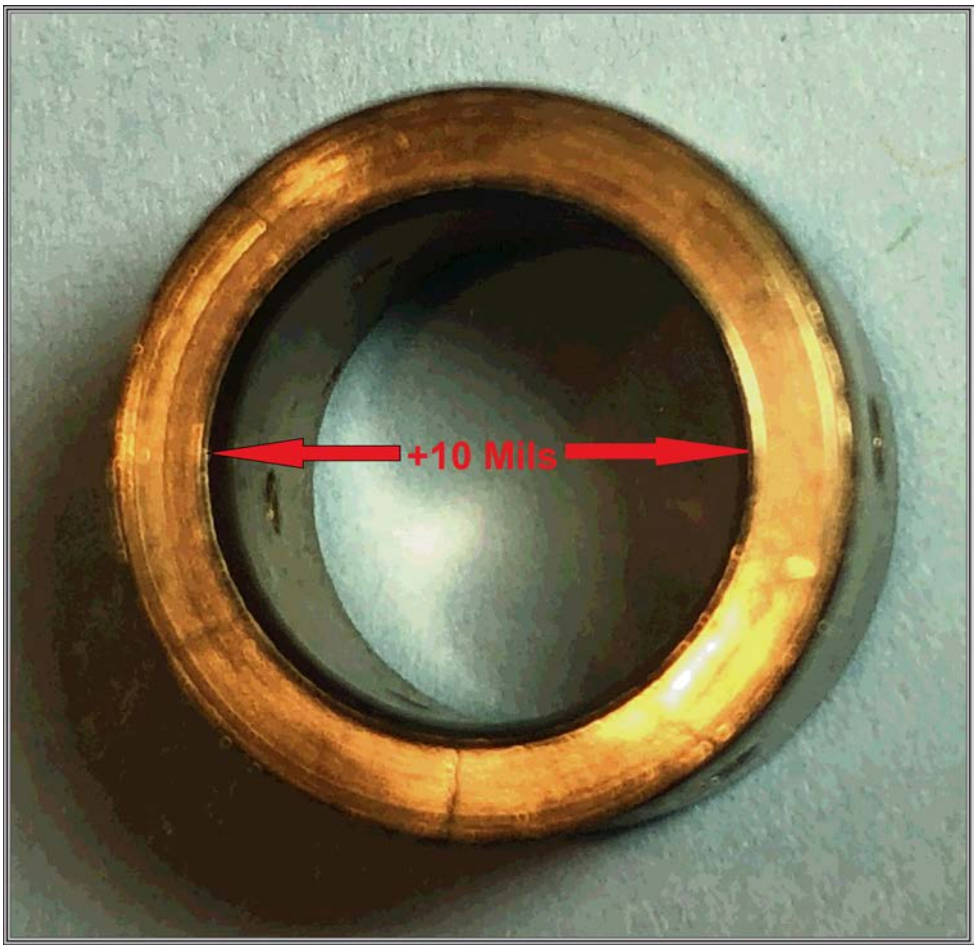


Fig. 2.7 Compressor Bushing Showing Enlarged Bearing by 10 Mils

The clearances on the floating bush bearings should normally be less than one mil. Measurement of the shaft and internal diameter of the compressor bearing has shown that the clearance has been enlarged to over 10 mils, allowing the impeller to impact the volute.

3 Analysis of a 2 Liter 4 Cylinder EngineTurbocharger Dynamics

3.1 Turbocharger Model

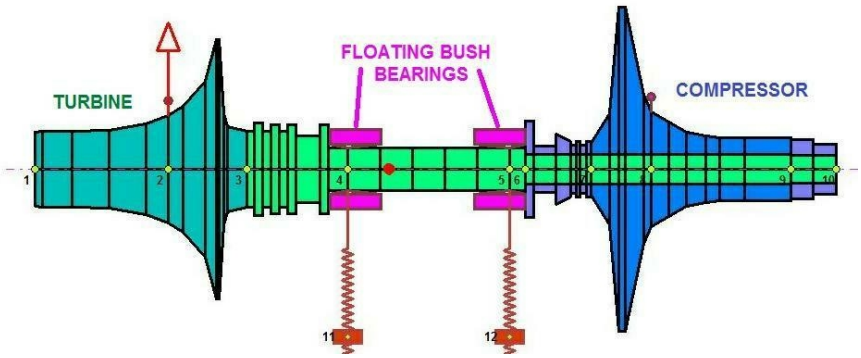


Fig. 3.1 2 L 4 Cylinder Turbocharger Model

Figure 3.1 represents a computer simulation model of the small turbochargers designed for two liter four-cylinder engines. The turbocharger is supported by two bronze floating bushing bearings which are slightly less than one half of an inch in diameter and slightly wider than one quarter of an inch. These turbochargers are designed to operate at speeds of over 200,000 RPM.

3.2 Critical Speed Analysis

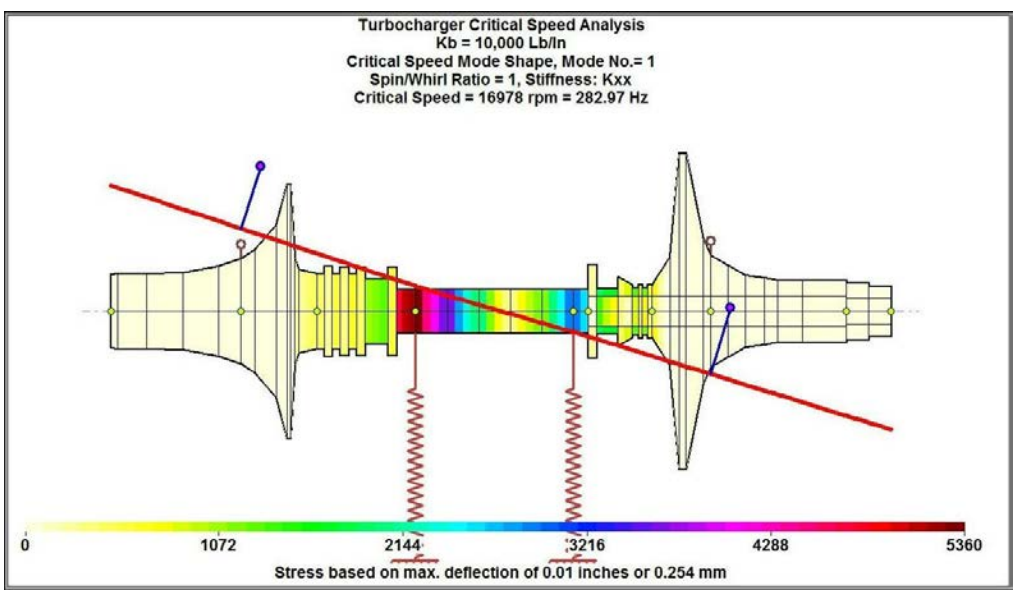


Fig. 3.2 1st Critical Speed at 283 Hz With Assumed Kb = 10,000Lb/In

Figure 3.2 represent the predicted turbocharger first critical speed based on an assumed bearing stiffness value of Kb = 10,000 Lb/In. This mode is essentially a rigid body mode in which the turbine and compressor wheels move out of phase. The exact frequency is a function of the actual bearing stiffnesses and may vary due to the actual bearing values of the floating bush bearings and rotor unbalance.

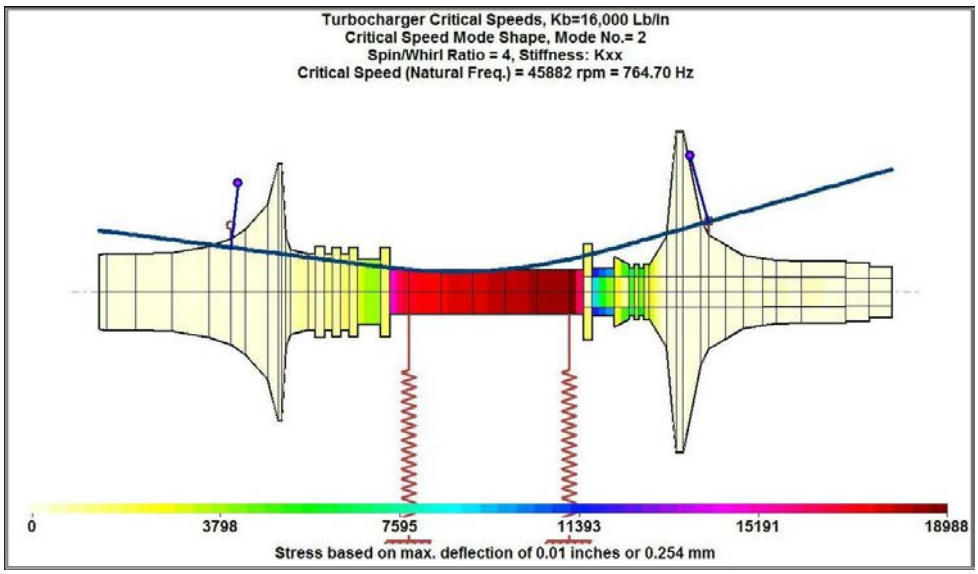


Fig. 3.3 2ND Turbocharger Critical Speed at 765 Hz

Figure 3.3 represents the turbocharger second critical speed under the assumed bearing stiffness values of Kb = 1e5 Lb/In. The exact value of this mode is determined by the bearing coefficients at the operating speed of 220,000 RPM. With floating bushing bearings, both the first and second modes may be excited at running speed. The instability in the second mode is of concern as the turbo amplitude is much higher at the compressor location leading to increased bearing wear and rubbing of the compressor on the volute.

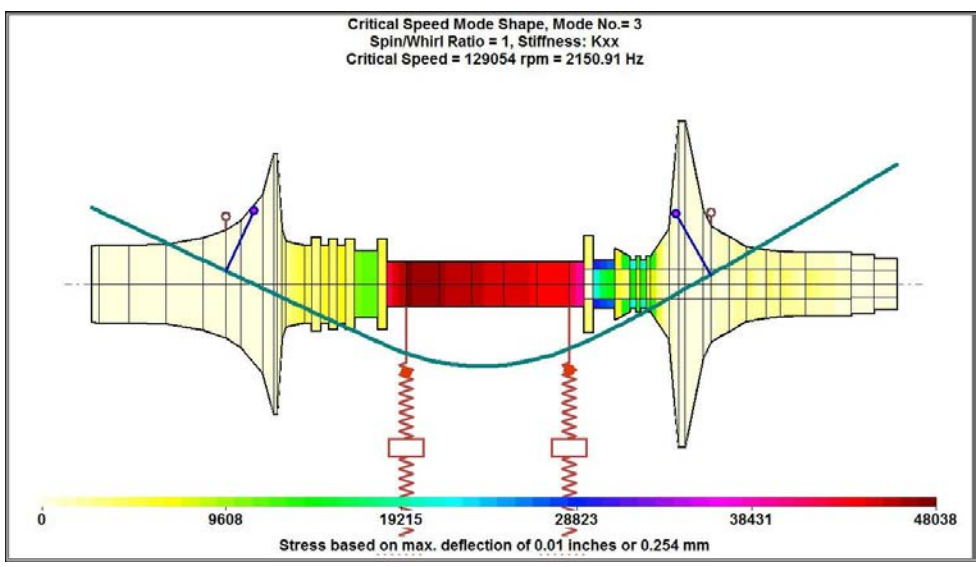


Fig. 3.4 3rd Critical Speed Bending Mode at 129,00 RPM (2150 Hz)

Figure 3.4 represents the third turbo critical speed which is a bending mode. In proper turbocharger design, the bending mode should be above the operating speed range, and should not be encountered in the speed range. Floating bush bearing damping is very ineffective in controlling the third bending critical speed.

3.3 Floating Bush Bearing Stiffness and Damping Coefficients

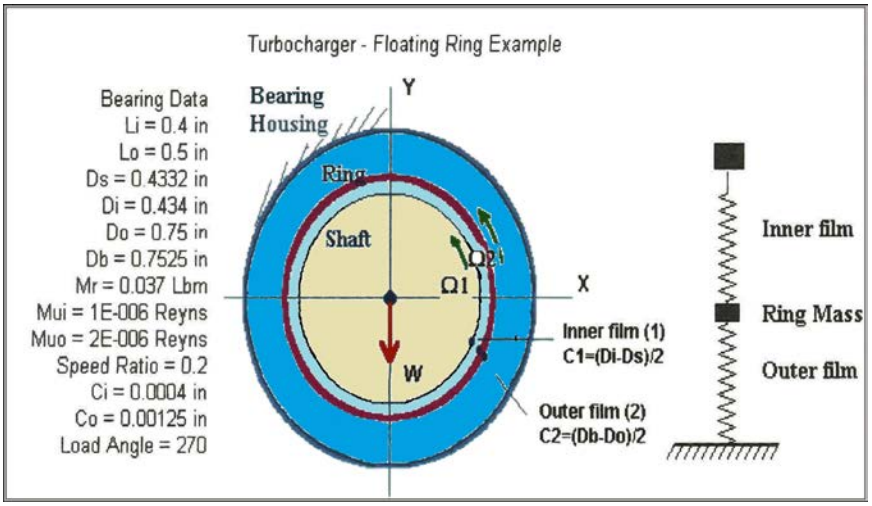


Fig. 3.5 Schematic Diagram of the Floating Bush Bearing

Figure 3.5 represents a schematic diagram of a floating bush bearing. In the design of the floating bush bearing, the center inner clearance is kept at a much tighter value than the outer clearance. The operation of a floating ring bearing is such that the ring spends at about a rate of 20% of shaft speed due to the internal friction. The original concept of the floating ring bearing developed by General Motors in the 1930s was to reduce frictional losses. The plain journal bearing is inherently unstable even at very low speeds. With the floating ring bearing, the system is still unstable but the motion of the ring produces nonlinear stiffness and damping components which allow for controlled limit cycle motion.

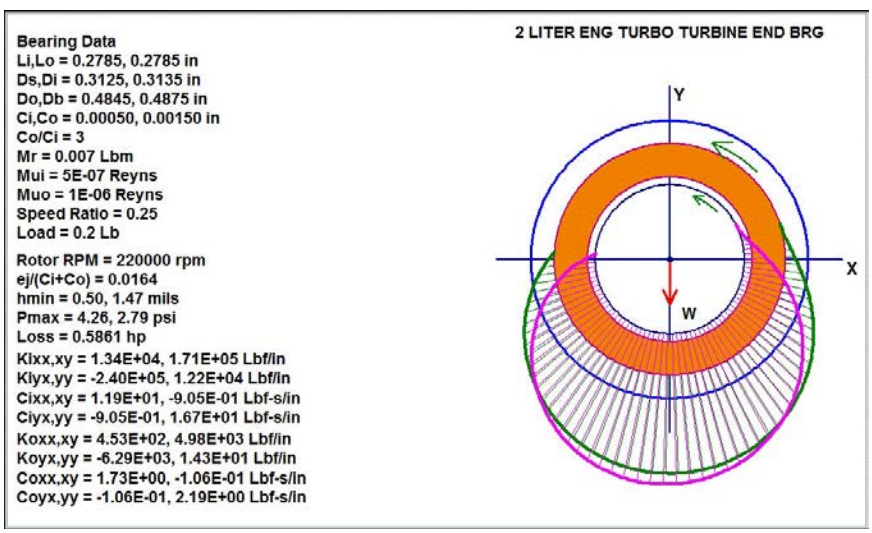


Fig. 3.6 Stiffness And Damping Characteristics of a Floating Ring Bearing at 220,000 RPM

Figure 3.6 represent the stiffness and damping characteristics of the compressor side floating ring bearing at 220,000 RPM. Because the compressor side bearing is so lightly loaded, the bearing has low direct stiffness but very high cross coupling stiffness which adds to the instability characteristics.

3.4 Stability Analysis of Turbocharger in Floating Bush Bearings

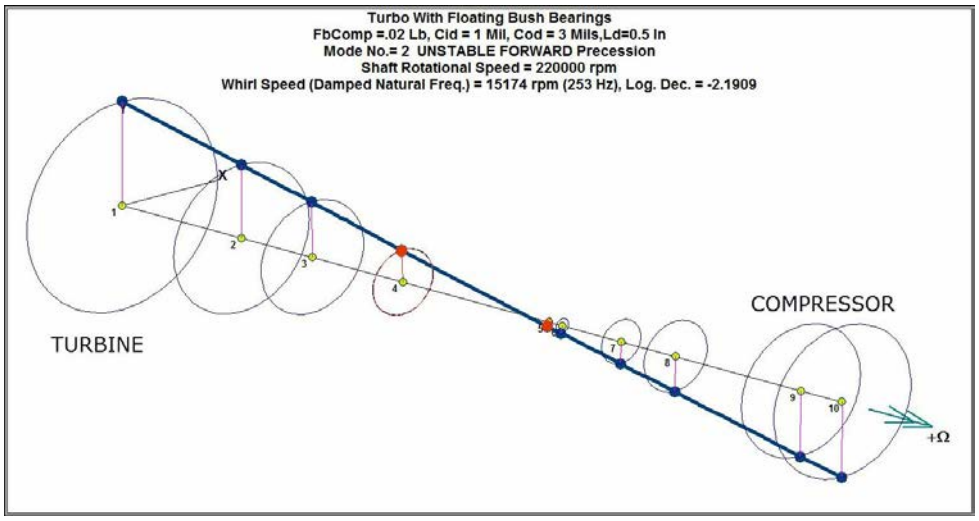


Fig. 3.7 Unstable Conical Whirl Mode at a Frequency of 253 Hz

Figure 3.7 represents the first conical whirl mode of the turbocharger at a frequency of 253 Hz. This is basically a rigid body mode in which the turbine and compressor motion are moved out of phase. This conical whirl motion can cause the impeller to rub against the volute causing turbocharger failure. All turbochargers with floating bush bearings exhibit this type of whirl motion. This distinct frequency may be detected on an engine using a frequency analyzer. It is a very low frequency compared to the frequency of running speed.

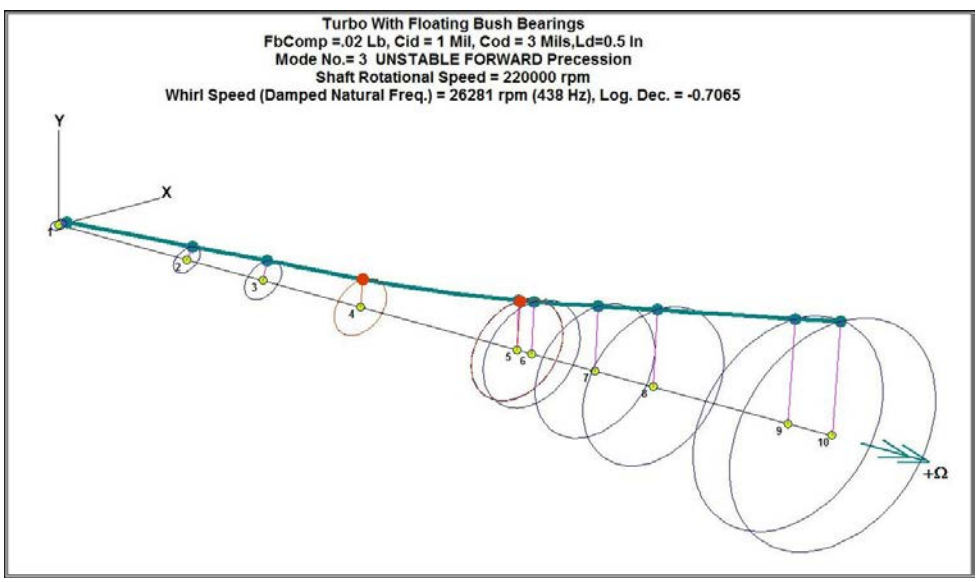


Fig. 3.8 Second Whirl Mode with Large Compressor Motion At a Whirl Frequency of 453 Hz

Figure 3.8 represents the second unstable whirl mode that may be encountered in the turbocharger. In this particular whirl mode, there is large amplitude of motion at the compressor location. The turbocharger vibration spectrum, on increase in speed, may exhibit one or both of these frequencies. This higher frequency whirl mode may also induce further wearing of the compressor side bushing bearing. These fundamental self-excited unstable whirl frequencies are inherent with all floating bush bearing turbocharger designs and can not be avoided.

3.5 Turbocharger Time Transient Analysis - Alford Forces and Unbalance

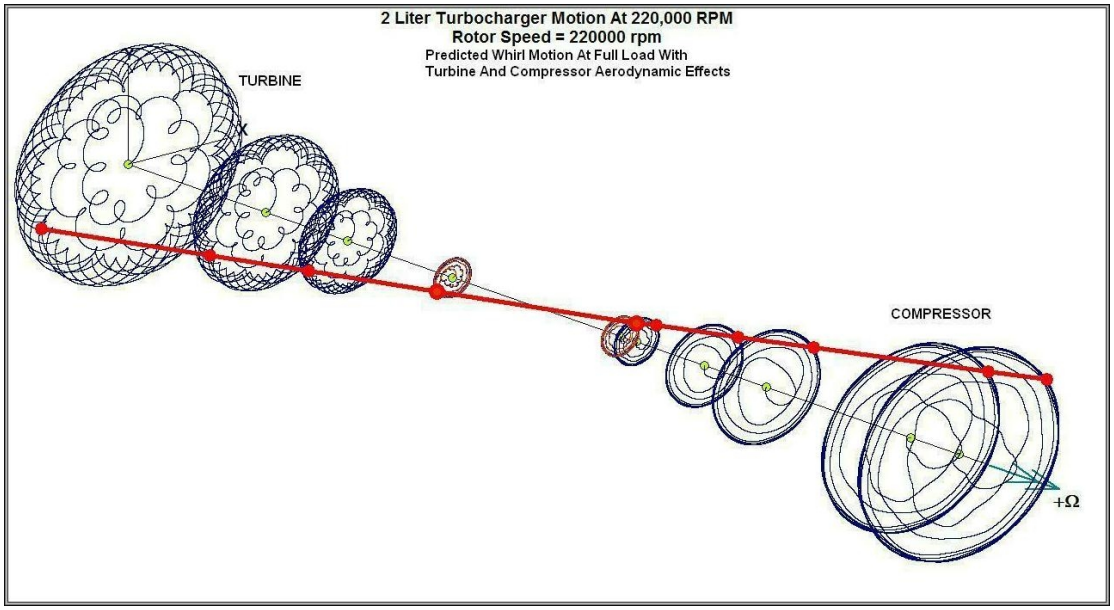


Fig. 3.9 Turbocharger Time Transient Analysis With Added Compressor and Turbine Alford Forces and Turbine Unbalance

Figure 3.9 represents the time transient analysis of the turbocharger at 220,000 RPM with the addition of the aerodynamic Alford forces acting at both the compressor and turbine locations. A small amount of unbalance has been added to the turbine location. Instability experienced in turbochargers is not solely do to the floating bush bearings. Even turbochargers operating with rolling element bearings may ask experience unstable self excited whirl motion due to the added effect of the Alford forces acting on the turbine and compressor locations.

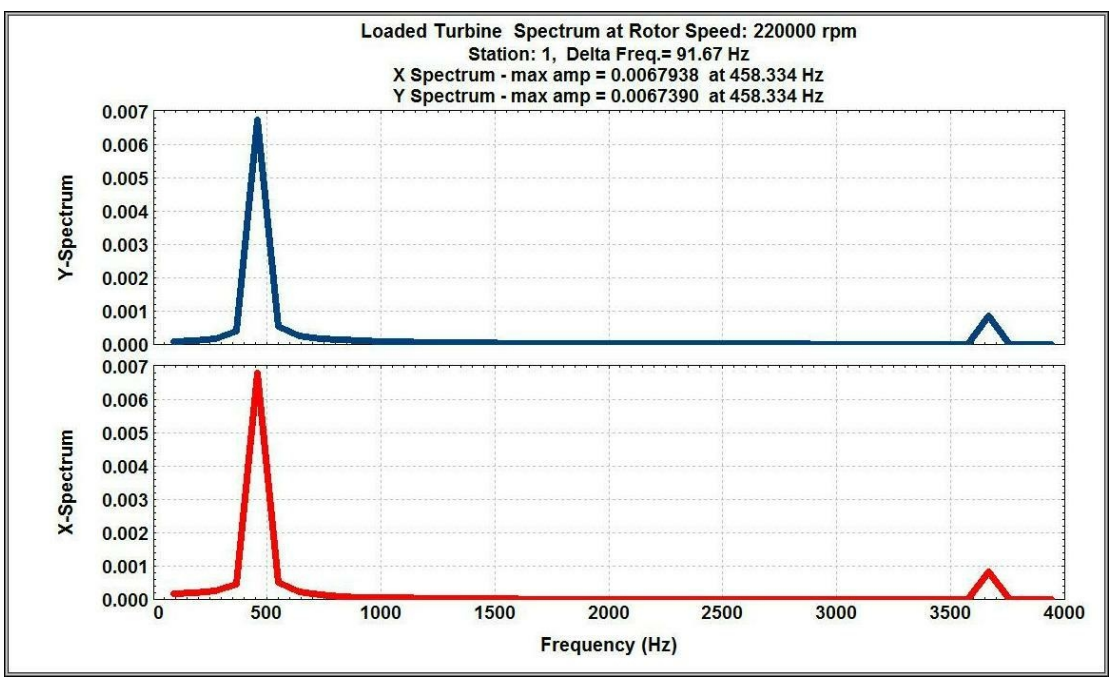


Fig. 3.10 FFT Frequency Spectrum Analysis of Turbocharger Vibration

These aerodynamic forces were first described by Joel Alford of GE Gas Turbine Division and are now referred to as Alford forces. He showed that when turbines and compressors operate in an eccentric fashion, the change in efficiency around the turbine or compressor wheel will cause cross coupling forces to act on the system.

The violent turbocharger motion as depicted in Figure 3.9 may be generated by operating the turbocharger at full power. When this occurs, there is a distinct tone or howl that may be observed admitted from the turbocharger. In this case the tone is similar to the pitch of A4# of the keyboard. Each turbocharger has its own distinct signature and frequency content which can be recorded with an iPhone 8 or 10 using the appropriate APPLE app for a frequency analyzer or for a decibel meter.

These apps, which market for only a few dollars, have the same effectiveness as over $50,000 of previous used lab equipment. By recording the turbocharger signature over various intervals, it would be possible to detect the probability of an early turbocharger failure by the increase of the subsynchronous whirl component emitted from the turbocharger. Bearings, seal and the thrust bearing replacement could then be performed before total turbocharger failure has occurred.

The time transient motion as depicted by Figure 3.9 was obtained by integrating the system of equations of motion forward in time for 40 cycles of shaft motion. The turbocharger system, as depicted in Figure 39 has 40 degrees of freedom. In order to obtain an accurate calculation of the time transient motion, the equations of motion must be integrated for approximately 300 time steps for each cycle of motion. At each time step, the nonlinear forces generated by the floating ring bearing inner and outer films must be determined to compute the nonlinear motion. The equations of motion are numerically integrated in time by either the Newmark Beta method or Wilson Theta. There are a number of more complex integration procedures that also may be used for more complicated cases which evaluate rubs.

If this numerical computation were attempted in the early 1970s, it would require the supercomputers available at NASA Goddard Space Center or NASA Lewis Research Center. The computations are now performed in minutes using a PC. This now gives us the ability to investigate more complex options and understand more fully the behavior of these high-speed turbochargers.

The computations were performed using the ***DyRoBeS*** software developed by Dr. Wen Jeng Chen of ***Eigen Technology***. The major turbocharger companies are using the ***DyRoBeS*** software to help develop their turbochargers. The results of their studies are considered proprietary and thus we have not been in a situation in which we can help them improve their turbochargers. Turbocharger development is a highly competitive field and few papers are presented by the turbocharger companies describing turbocharger behavior.

By using the time transient numerical approach, the rotor response can be determined including nonlinear bearings and rotor unbalance. Although the turbocharger may initially be well balance, over a period of time, unbalance can occur, particularly on the turbine impeller due to erosion and particle buildup from the exhaust gases deposits on the wheel. A response was computed for the turbocharger at running speed with a very small amount of unbalance on the turbine wheel. Figures 3.11 and 3.12 show the forces transmitted to the turbine and compressor bearings respectively.

Figure 3.11 shows the turbine bearing forces transmitted over 40 cycles of the shaft motion. The turbine bearing forces show an initial peak response of 100 pounds. This peak response attenuates to a value of approximately 60 pounds of force transmitted after 30 cycles of shaft motion. The undulation of the vibration spectrum is due to the low-frequency whirling of the turbocharger. Although the turbocharger is unstable with subsynchronous whirling, the motion is bounded due to the nonlinear bearing forces. This effect is referred to as limit cycle whirling.

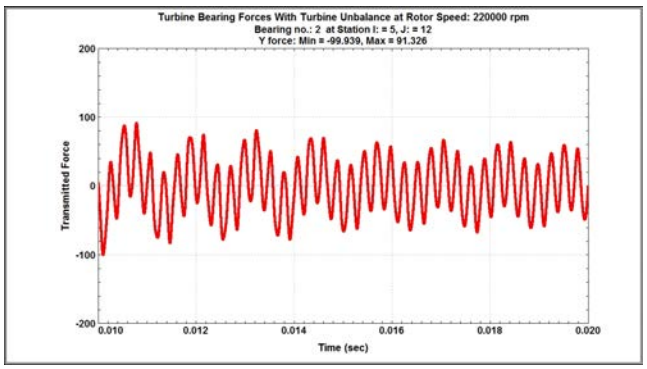


Fig. 3.11 Turbine Bearing Forces Transmitted With 0.01 Oz-in Turbine Unbalance

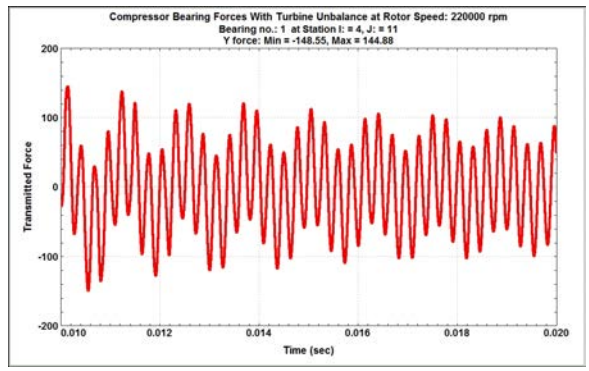


Fig. 3.12 Compressor Bearing Forces With 0.01 Oz-in Turbine Unbalance

Figure 3.12 represents the compressor bearing forces transmitted due to the turbine unbalance. The initial transient force due to the suddenly applied unbalance is 140 Lb. After 30 cycles of shaft motion the maximum compressor bearing load attenuates to 100 pounds. Thus we see that unbalance at the turbine impeller actually causes higher forces to be transmitted to the compressor floating bush bearing.

This characteristic of turbine unbalance causing higher forces transmitted to the compressor bearing is characteristic of most turbochargers. This is due to the fact that the turbine wheel is considerably heavier than the compressor impeller. This causes the turbine disk to act as a node point and the unbalance force on the turbine is magnified at the compressor bearing location. This effect adds to the wearing experienced by the compressor bearing. An increase in the compressor bearing clearance makes the system even more unstable.

What is of concern with the use of floating bush bearings in extremely high speed turbochargers is that floating bush bearings are always unstable at all speeds and exhibit sub- synchronous whirling. The turbocharger may exhibit whirling in either the first or second modes. Regardless of the claims made by the turbocharger manufacturers, these bearings can never be stabilized, regardless of the clearance ratios of the inner and outer bearings.

What keeps the bearings, shaft and casing from contacting is the generation of nonlinear bearing forces developed by the hydrodynamic films of the inner and outer bearing clearances. If, however, full lubrication is not achieved, then the nonlinear forces breakdown and rubbing of the floating bush bearings with the casing and shaft will occur. Examination of the wear patterns on three failed turbochargers clearly showed heavy rubbing occurring on the outer rings of both the compressor and turbine bearings. These small bronze floating bush bearings are totally unacceptable for use in high speed turbochargers operating at over 200,000 RPM.

3.6 Turbocharger Analysis Summary

The floating bush bearing is the standard bearing design used in both diesel and automotive turbochargers. This inexpensive bearing design has been used in a number of larger, slower speed turbochargers over the years. However, this bearing is inappropriate for use in these ultra high-speed turbochargers. The floating bush bearing, to begin with, has an inherent flaw in that it causes unstable sub-synchronous whirling to occur at all speeds. In fact, there are two unstable modes which can occur. The lowest is a conical whirl mode associated with the first critical speed. At higher speeds, the second in-phase whirl mode can occur. To keep the floating bush bearing from contacting the casing, full lubrication of the bearing is required. This is extremely difficult to achieve with these high-speed turbochargers.

From the analysis, it is seen that there is an additional problem in the design of these ultra high-speed turbochargers. The first two modes of the turbocharger are essentially rigid body modes of conical and cylindrical motion. In normal turbocharger design, the bending critical speed (3rd system critical speed) is designed to be above the running speed. It appears that in this design, the third bending critical speed is in the operating speed range. In the design of double overhung rotors, it is difficult to balance for third mode bending critical speed. Also, the bearings are very ineffective in damping out the third bending critical speed.

What is of interest to observe is that in the excitation of the second whirl mode, the largest amplitude of motion occurs at the compressor end. This is due to the heavier weight of the turbine. Examination of the wear on the compressor bearings showed excessive wear that was not apparent in the turbine bearings. The design of the compressor bearing actually needs to be larger in order to carry the increased loading.

One of the other problems encountered with these turbochargers is a change of balance during operation. This in particular occurs with the turbine which can become unbalanced due to exhaust deposits. Unbalance of the turbine actually causes larger forces to be exerted on the compressor bearing as shown in Figure 3.12

It is recommended that more stable bearings be designed to replace these simple floating bush bearings which are unsuited for this high-speed application.

4 Dynamic Behavior of Turbochargers in Floating Ring Bearings and How They Fail

4.1 Background of Plain And Floating Bush Bearings

The dynamical behavior of turbochargers supported in floating ring bearings is quite complex. The concept of the floating ring bearing was actually developed in the 1930s by General Motors in order to reduce bearing friction losses. The study of the dynamics of fluid-film bearings is concerned with the problem of rotor stability. The occurrence of self-excited journal bearing instability in large compressors has been estimated to have cost hundreds of millions of dollars of losses in the petrochemical industry alone.

The most unstable bearing to place in a large compressor is the plain journal bearing. These bearings have an operating speed of at most of 7,000 RPM before they become unstable. The instability of the journal bearing is manifested as a self-excited whirling in which the rotor centerline precesses at approximately a rate of 40 to 48% of rotational speed. This phenomena is also referred to as oil whirl or with flexible rotors as oil whip. This condition cannot be alleviated by balancing of the shaft. Careful balancing of a large compressor rotor may actually make the oil whirl phenomena worse since it removes some of the dynamic rotating load exerted by the unbalances.

The floating ring bearing is designed to be free-floating on the shaft. The friction of the shaft on the floating ring bearing causes the ring to precess or spin at a rate of approximately 20 to 40% of the shaft rotational speed. The original concept of using a floating ring was to reduce bearing friction losses.

From a stability standpoint based on linear bearing coefficients, all turbos are inherently unstable with self-excited rotor whirling induced by the floating rings.

However, in actual operation, the motion of the precessing rings in the bearing housing creates non-linear bearing forces between the rings and housing. Although the system is still unstable from a linear standpoint, the generation of the nonlinear ring forces causes a bounded limit cycle motion to occur.

System analysis of the resulting complex orbital motion shows that the motion is composed of vibrations at running speed due to unbalance and the lower frequency whirl motion. As long as the bearings are properly lubricated, and no sudden speed changes occur, the shaft, floating ring, and casing do not come into contact with each other.

4.2 Experimental Behavior of a 3.7 L Diesel Engine in Floating Bush Bearings

This bearing type has been adopted by many turbocharger manufacturers as it is cheap and inexpensive to manufacture. Test results of the turbocharger vibration characteristics by the turbocharger manufacturers are never presented.

A breakthrough paper was presented by Dr. R. Gordon Kirk of Virginia Tech in 2007 on the experimental behavior of a 3.9 L diesel engine turbocharger over a range of loads and speed profiles. Turbochargers of various sizes with floating ring bearings exhibit similar dynamical behavior as shown by the experimental data of Dr. Kirk. The frequency spectrum of the 3.9 L turbocharger is also referred to as a waterfall diagram.

Figure 4.1 shows the turbo vibration spectrum during a 4-minute run up and run down of the 3.9 L diesel turbo. The frequency is presented in cycles per minute (CPM) rather then the normal use of cycles per sec (Hz).

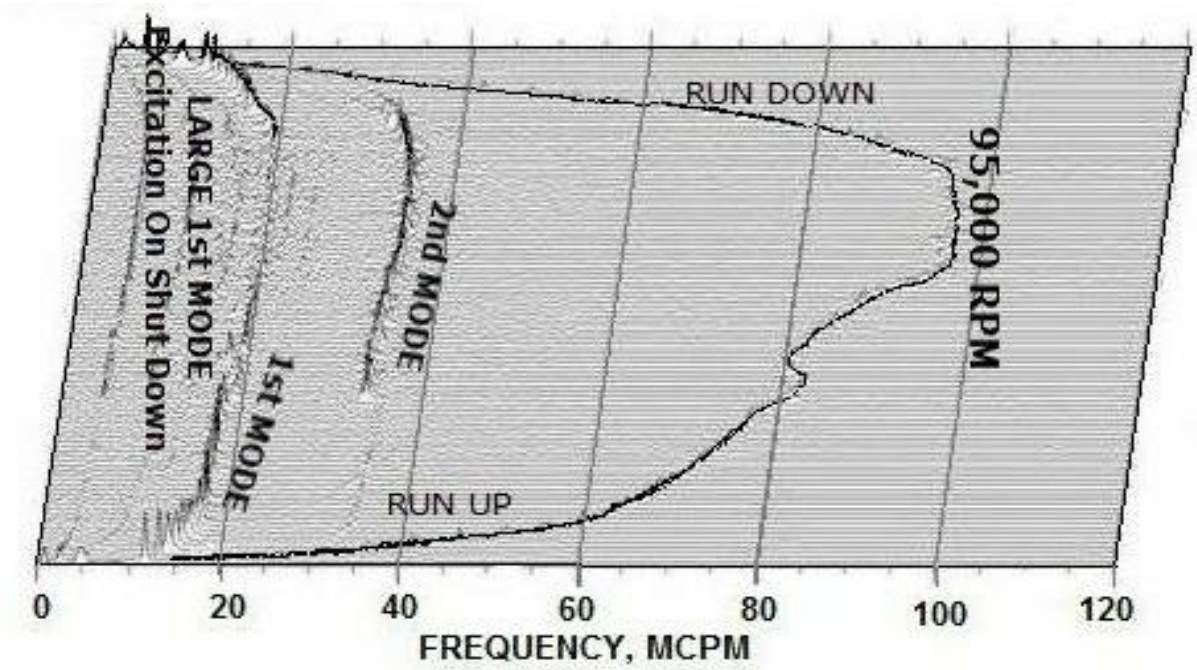


Fig. 4.1 Experimental Vibration Spectrum Of A Diesel Engine Turbocharger On Run Up And Run Down Showing Unstable Whirl Modes With Floating Bush Bearings

As the diesel engine starts up, a peak amplitude occurs at approximately 12,000 RPM This peak represents the first critical speed of the turbocharger, which is a conical rotor mode with the compressor and turbine disks whirling out of phase. As the rotor speed is increased to approximately 60,000 RPM, the first whirl mode dies out.

Once the speed of 70,000 RPM is reached, the unstable first conical mode is excited with a whirl mode of about 18,000 CPM which corresponds to a frequency of 300 Hz. This would create an audible sound from the turbocharger corresponding to a tone corresponding to D4 of a piano keyboard.

When the turbocharger reaches 80,000 RPM, the first whirl mode dies out and the second whirl mode is encountered with a frequency of approximately 40,000 CPM. In this mode, the turbine and compressor rotor motion are in phase with the largest motion occurring at the compressor location. This particular mode exerts additional loading on the compressor side bearing.

As the turbocharger speed is reduced to 80,000 RPM, it is seen that the 1st conical whirl mode is excited in addition to the 2nd mode. At a speed of approximately 75,000 RPM, the second mode completely drops out and only the first conical mode is excited. What is particularly disturbing about the behavior of the turbo diesel on shutdown is that the conical whirl mode continues to persist until the turbocharger is at rest.

This appears to be one logical reason why truckers do not shut down their engines in a parking lot. Considerable bearing wear can occur on shutdown, at both compressor and turbine bearing locations. For example, examination of the wear on the 2 L turbocharger bearings showed distinct wear bands on the turbine side bearings due to the conical motion of the rotor.

This characteristic of the persistence of an unstable whirl mode is observed in the experimental data of plain journal bearings during shutdown. With the high energy of the orbital motion of the shaft in the bearings, the unstable whirl motion does not cease until the rotor is at rest.

Regardless of the size of the turbocharger or its speed, all turbochargers with floating ring bearings are ***inherently unstable***! That is, after a speed that exceeds approximately twice the rotor first critical speed, the rotor exhibits self-excited whirling in which the first critical resonance frequency is excited.

It is seen from the experimental data, as shown in Fig. 4.1, that this unstable conical whirl mode can exist over a speed range until at a higher speed range, the second in-phase turbocharger 2nd critical speed mode is excited. This mode then becomes predominant mode of motion. In this higher unstable whirl mode, the turbine, which is considerably heavier than the compressor, becomes a stable node point and the maximum amplitude occurs at the compressor location. This increases the loading on the compressor end bearing.

All turbochargers with floating ring bearings use identical bearings for both the compressor and turbine ends. However, due to the mode shape of the second mode, it is the compressor bearing with the lighter wheel that actually experiences the highest loads and the largest amount of bearing wear. It is surprising that the bearing at the lighter compressor impeller should actually have larger bearing loads transmitted then the heavier turbine impeller at running speed.

(This was similar to the situation that we encountered with the bearing design of the oxygen pump for the NASA space shuttle engines. The heavier main impeller had a 55 mm bearing and the lighter preburner impeller only had a 45 mm size bearing. It required a larger bearing. As a result of the smaller bearing size and inadequate seal damping, the pump design life was reduced from 8 hours to only 7 minutes!)

A review of the wear pattern on the turbine end bearings for the 2 L turbochargers showed at distinct narrow band wear mark caused by the conical motion of the unstable 1st mode.

This distinct wear mark on the turbine end bearing is another indicator that the current lubricant had insufficient boundary layer lubricant properties to prevent this type of wearing.

4.3 Failure of Turbochargers in Floating Ring Bearings

There are a number of modes which can lead to the failure of turbochargers with floating ring bearings. In his book, **TURBO** ***Real World High–Performance Turbocharger Systems*** by Jay K. Miller, he presents a figure, courtesy of Honeywell Turbo–Technologies, that indicates that most turbocharger failures are due to foreign object ingestion into the compressor or the turbine areas. This mode of failure probably accounts for only 5% of the actual failures encountered with turbochargers employing floating ring bearings.

There are a number of ways that turbochargers with floating ring bearings may fail. They may fail over an extended period of time (***death from a thousand cuts***) or have an abrupt unexpected failure (***sudden death heart attack*!**).

4.3.1 Turbocharger Failure over an Extended Period of Time

This type of failure is due mainly to the wear of the turbocharger bearings over an extended period of time, particularly with the compressor end bearing. This type of turbocharger failure is associated with engine cold starts, in which of the lubricant has not properly reached operating temperature and has not fully lubricated the floating bush bearings. The other significant bearing wear mode occurs during hot shutdowns in which the engine is abruptly shut off depriving the turbocharger with lubricant and cooling fluid.

The other major contributing factor is the use of a synthetic lubricant which does not have the proper extreme pressure additives required to prevent bearing wear due to shaft contact. In a cold engine startup, before the engine has reached proper operating temperatures, the turbocharger bearings may not be fully lubricated. This leads to wear on both the compressor and turbine side bushing internal bearing clearances.

Unlike the operators of the diesel engines, auto drivers of the smaller 2 L gasoline engines with turbochargers, rarely allow their engines to warm up before starting off. This is most undesirable, particularly in winter conditions. It is highly recommended that the engine be allowed to warm up for at least one minute before operating. This is very seldom done by most owners of these automobiles and is rarely mentioned in the owners manual.

The second occurrence of bearing wear occurs during hot shutdowns. Under these conditions, the engine has been operating at an extended period of time and has reached high operating temperatures. When the engine is abruptly shut off with the turbocharger operating at over 200,000 RPM, the oil has been shut off to the turbocharger as well as the cooling water to the turbocharger.

As is seen from the diesel engine experimental data of Dr. Kirk as shown in Fig. 4.1, on shutdown the turbocharger excites the unstable conical whirl mode when the turbo speed drops to approximately twice the first critical resonance frequency.

As the turbocharger decreases further in speed the unstable conical whirl mode persists. This motion causing wear on both the compressor and turbine bearings. This wearing action is clearly seen on the narrow wear band that developed on the turbine side bearing.

In order to have controlled limit cycle motion of the shaft in the bearing, the inner bearing clearance is maintained to a very small clearance, usually less than one mil. As the compressor bearing clearance enlarges, in particular, the whirl motion becomes larger and more violent. This increase in turbo whirl motion due to compressor bearing clearance increase, generates an audible whine, which is clearly discernible. One of the damaged compressor bearings had the inner clearance opened up from less than one mil to over 10 mils of internal clearance, leading to massive compressor impeller rubbing on the volute.

4.3.2 Sudden Turbocharger Failure

There are a number of ways in which the turbocharger may experience a sudden failure without warning (***the heart attack mode***).

1. **Acceleration to full power**

When a turbocharger is accelerated to full power, there are additional cross-coupling forces acting on both the compressor and the turbine wheels which increase the unstable whirl motion. These forces are referred to as the Alford effect after Joel Alford of GE who first identified these forces acting on gas turbines. The same effect also occurs with centrifugal impellers.

As the turbine or compressor impeller moves eccentrically in the volute, there is a change in efficiency around the turbine or compressor which leads to forces acting perpendicular to the shaft centerline of motion. These cross-coupling forces increase the effect and severity of the whirl motion. Since the subsynchronous whirl motion increases in magnitude, there is an audible increase in the acoustic sound from the turbo. The audible frequencies correspond to the excitation of the turbocharger first and second critical speeds.

The increase in the turbo whirl motion may cause a hard rub of the compressor on the volute. A very heavy rub may cause an abrupt shaft failure due to the sudden friction of the compressor wheel resulting in the shaft shearing at the turbine location. These type of failures are sudden and without warning. This effect is exacerbated if there is some additional wearing that has previously occurred in the compressor bearing clearance. As the compressor bearing opens up in clearance, it becomes more unstable and more susceptible to unstable Alford forces. These Alford forces are also present in turbochargers with rolling element bearings

2. **Surging of the Compressor**

The compressor blades of a turbocharger are designed with a backward leaning blade. The reason for this is to improve its range of flow. It will not generate the pressure profile that a pure radial centrifugal compressor will generate. However it will have greater surge capacity.

A compressor wheel will go into surge if one is operating at a high pressure but with a reduction of flow from the impeller. Under these circumstances, the flow will recirculate around the impeller causing very high radial and axial forces.

Surging of the turbocharger compressor can occur when there is a sudden deceleration of the vehicle with rapid breaking. The high forces generated during a surge condition can cause the impeller to rub against the volute causing a sudden failure. Therefore, sudden deceleration of the engine due to hard breaking at high speed needs to be avoided. A heavy surge of a turbo that already has some compressor bearing wear can lead to a *fatal heart attack*.

3. **High Temperature Coking At The Turbine Bearing**

Hot exhaust gases may penetrate the seal area around the turbine end bearing and cause coking in the turbine bearing. In addition to the coking, the hot temperature experienced at the turbine location may cause the turbine bearing to seize to the shaft. When this occurs, we basically have a plain journal bearing with a large clearance whose stability threshold is only several thousand RPM. With the turbocharger operating at speeds of over 200,000 RPM, the failure that occurs when a turbine bearing seizes to the shaft is almost instantaneous. The subsynchronous whirl motion that occurs is so violent that the turbine experiences a massive rub with shearing of the shaft at the turbine location. This results in an instant fatal turbo *heart attack* with no previous warning.

4. **Contaminated Lubricant to the Turbo**

One of the major problems with all turbochargers with floating ring bearings is to ensure that an adequate amount of lubricant is fed to the journal bearings. The lubricant is normally supplied through radial openings through the bearings. This presents a serious problem with these extremely high speed turbochargers in which the bronze bushing bearings are slightly less than one half an inch in diameter and a little longer than one quarter of an inch.

The situation is exacerbated if there is any significant contamination particularly from carbonized oil being fed into the turbo's bearings. A restricted oil supply to the turbo bearings greatly reduces the life of the turbo. It appears that bearings of the to 2 L turbocharger class may suffer from a deficiency of oil supplied to them in addition to being too narrow to carry the radial loading exerted, particularly to the compressor bearing.

Since the turbine bearing runs considerably hotter than the compressor bearing, it is particularly susceptible to any oil contamination which could lead to a fatal heart attack. In many large diesel engines, an oil prefilter is often used to condition the oil before entering the turbocharger. The current turbocharger floating bush bearings are in adequate from the standpoint of receiving a sufficient oil supply.

In addition, the bearings are undersized to handle the dynamical loading that can be encountered At full speed due to the stability problems and additional turbine unbalance loads due to turbine carbon buildup and blade erosion.

5. Turbine Unbalance Due to Carbon Build Up And Blade Erosion

The exhaust gas turbine operates in a very harsh environment in comparison to the centrifugal impeller. The turbine, therefore is subject to changes in the unbalance due to the operating conditions. The unbalance of the turbine can change due to carbon and particle buildup on the turbine blades and also due to blade erosion. Any contacting of the turbine blades on the casing will change the balance quality of the turbine wheel.

When the unbalance is present, synchronous rotating forces are generated at the turbine wheel. Because of the extremely high operating speed of these turbochargers, a small amount of radial unbalance on the turbine wheel can translate into a significant rotating unbalanced load.

In review of the design of turbochargers, it is seen that the turbine wheel is considerably heavier than the compressor impeller. This places the center of gravity of the turbo very close to the turbine bearing. Thus, at full operating speed, unbalance in the turbine causes a pivoting around the system center of gravity which actually creates greater forces to be transmitted to the compressor bearing rather than the turbine bearing! The paradox of bearing design for turbochargers is that the lighter compressor bearing actually requires a longer and stronger bearing than does the turbine bearing. This concept seems to defy intuition, but it is a result of the heavier weight of the turbine as compared to the compressor.

All of the damaged bearings for the 2L turbocharger showed that there was considerable greater wearing of the compressor bearing clearance as compared to the turbine by a substantial amount. This is another indication of much higher loads being exerted on the compressor bearing under operating conditions.

6. Improper Speed Control Due To Poor Wastegate Operation

The speed of the turbocharger is controlled by a device called the wastegate. This device regulates the amount of engine exhaust gas that is fed to the turbine of the turbocharger. The design of the wastegate may be an internal design or an external mechanical design. These designs may be very simple mechanical designs regulating a flapper valve or can be extremely elaborate with electronic controls.

MIDSIZED SUVs

Hyundai Santa Fe

The Right Size at the Right Price

78

OVERALL SCORE

ROAD-TEST SCORE 80

HIGHS Handling, controls, rear-seat room and access, standard safety features

LOWS Uneven power delivery, ride is a touch firm

POWERTRAIN 235-hp, 2.0-liter four-cylinder engine; 8-speed automatic transmission; all-wheel drive

FUEL 21 mpg on regular fuel

PRICE AS TESTED $37,200

THE NEW SANTA FE is a compelling midsized SUV. Its price is close to that of some top-trim compact SUVs, which can make it a bargain. We like its size, safety features, and handling, but buyers should skip the more expensive turbo engine because of its uneven acceleration.

The Santa Fe's composed handling, with its responsive steering, gave our drivers confidence in rounding corners.

The controls are clear and easy to master, particularly the quick-to-respond infotainment system, though they're a reach for taller drivers who sit farther back from the steering wheel. The Limited trim we tested had rich cabin materials; well-padded surfaces where hands, elbows, and forearms might fall; and plenty of storage space.

It's easy to enter and exit this SUV. Once inside, the driver and passengers, particularly those in the back, find plenty of head- and legroom, and comfortable, supportive seats.

There are some blemishes. Our Santa Fe came with the optional turbo engine that delivers plenty of power, but it comes in bursts, unexpectedly hesitating or launching the SUV forward. We tried the smoother but less powerful 2.4-liter four-cylinder engine, and we think it's the better choice.

By skipping the turbo, buyers also avoid the 19-inch wheels and tires that make the ride overly firm.

Forward collision warning and automatic emergency braking are standard.

Fig. 4.2 CR Review of Hyundal 2.0 Liter Santa Fe With Turbocharger

The problem with obtaining smooth speed control of automobile turbochargers is that they may operate over a wide speed range. These speed ranges can vary from under 1000 RPM to up to 6,000 RPM. The problem of speed control is not so much of a problem with diesel engines since they operate at relatively low speed ranges from about 1000 to 1500 RPM. The turbochargers for diesel engines are much larger and operate at a lower speed usually below 100,000 RPM. Although they also have floating ring bearings and exhibit self-excited whirl instability, the motion is a controlled limit cycle motion and can operate for long periods of time without apparent problems.

The occurrence of erratic speed control of an automotive turbocharger presents a number of serious problems. The operating characteristics of a turbocharged engine with poor speed control causes irregular operation on acceleration. This leads to a very poor driving experience. As an example, Figure 4.2 from the consumer reports December 2018 issue, specifically does not recommend the Hyundai Santa Fe because of the irregular operation of its 2-liter turbocharged engine.

7. Confluence of the Third Bending Critical With the 1st Torsional Mode

There is an even more serious problem with irregular speed control of these small high-speed turbochargers. One of the desired design characteristics of a turbocharger is that the third bending critical speed should be above the operating speed range. With these high-speed turbochargers with very thin shafts, it appears that the bending third critical speed is below the operating speed range. Figure 4.3 represents the mode shape of the third bending critical speed, which is predicted to occur around 130,000 RPM.

Figure 4.4 represents the relative shaft and bearing energy distributions for this mode. The occurrence of the bending mode for these overhung turbochargers is particularly dangerous since over 90% of the relative bending strain energy is in the shaft and very little is in the bearings. That implies that this mode is poorly damped and may be easily excited. Figure 4.5 represents the three dimensional bending mode including the floating ring bearing stiffness and damping characteristics. This places the bending mode at a frequency of 135,453 CPM.

It should be noted that the first two critical speeds of the turbocharger are essentially rigid body modes of conical and cylindrical motion. In general these modes may be well balanced out by careful balancing of the turbine and compressor wheels.

However, as a general rule with double overhung rotors, it is highly undesirable for the rotor to enter into its third bending critical speed. This is because the rotor may be balanced well for the two rigid body modes but is never in proper balance when it encounters the 3rd bending mode.

Encountering the third bending mode in the operating speed range can result in significant high synchronous vibration amplitudes at both the turbine and compressor wheels.

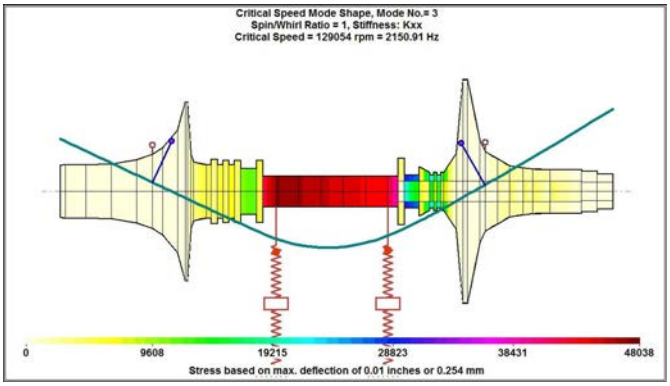


Fig.4.3 Turbocharger Undamped Bending Critical Speed (3rd mode)

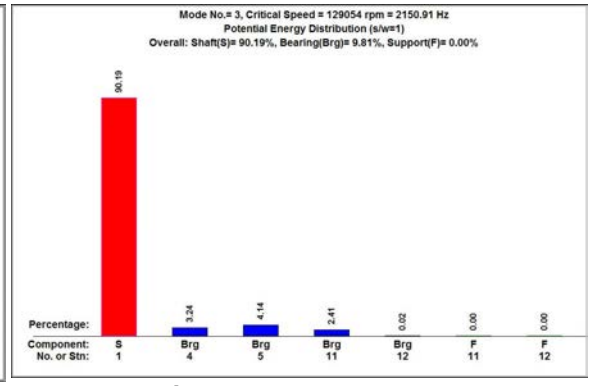


Fig. 4.4 3rd Mode Strain Energy Showing 90% Energy in Bending

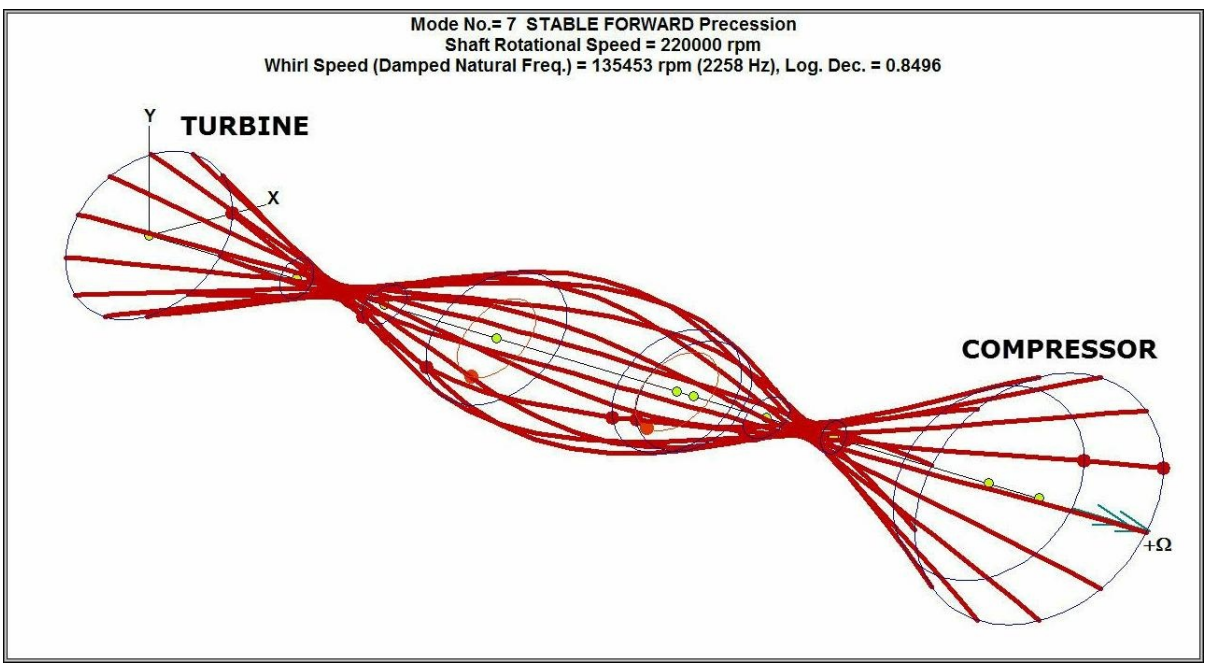


Fig. 4.5 3-Dimensional Bending Mode Including Bearing Stiffness and Damping Coefficients At 135,453 CPM

When damping is included, the third bending mode is slightly increased to 135,453 CPM. This low frequency is due to the slender turbocharger shaft. This bending mode lies well within the operating speed range of the turbo. This is not a desirable feature to have and is not recommended for good design. The problem is that turbomachinery designers are usually more concerned with turbocharger efficiency and performance rather than the rotor dynamics of the system. As a result we have a turbo design with too small a shaft diameter and inadequate fluid film bearings.

Figure 4.6 represents the first torsional mode of the turbocharger which is at a frequency of 131,108 CPM. It is of great concern to have the confluence of the turbocharger bending mode correspond with the first torsional mode. In the first torsional mode, the wheels are twisting out of phase to each other. In the bending mode, the turbine and compressor wheels are tilted and hence can rub and come in contact with the volute or casing. When this occurs, the torsional mode can also be excited, causing significant turbocharger damage.

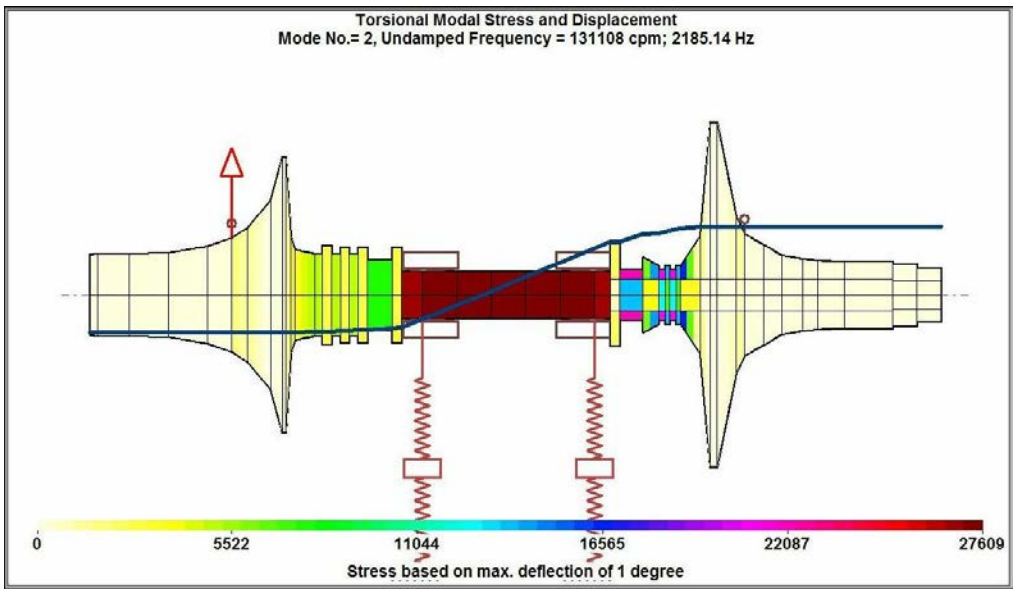


Fig. 4.6 1st Torsional Mode at 131,108 CPM

8. Heavy Turbine or Compressor Rub Causing Backward Whirl

A turbocharger may be considered as two spinning tops connected by a thin rod. At speeds of over 200,000 RPM, the turbine and compressor discs develop substantial gyroscopic moments. The disk gyroscopic moments raises the critical speeds for forward whirl motion and reduces the critical speeds in backward whirl.

Normally, the backward rotor critical speeds are seldom excited except in unusual circumstances. The situation is considerably more complex with overhung turbomachinery. If a situation occurs in which the overhung wheel comes into contact with the volute or casing, then the friction of the wheel on the casing can momentarily cause the disk center to precess or whirl in a backward motion. When this occurs, the gyroscopic moment reverses direction, keeping the wheel or impeller clamped against the casing wall. The high transient torque developed causes an instantaneous shaft fracture or ***fatal heart attack***.

Figure 4.7 shows the back side of the failed turbine. Figure 4.8 is a close-up showing the failed shaft. The shaft failure is similar to shaft torsional failures caused by suddenly applied torques.

Fig. 4.7 Rear View of Failed Turbocharger Turbine

Fig. 4.8 Close Up Turbine Shaft Showing Shear Failure

5 Summary, Conclusions and Recommendations

5.1 Summary

The four cylinder turbocharged engine has become extremely popular with over 95 models using the turbocharged 2 L engine. The use of the four-cylinder turbocharged engine over a six cylinder appears to be attractive as it reduces engine weight, produces good horsepower and mileage.

The standard bearing that is used in most turbochargers ranging from diesel engines to the four-cylinder engine is the floating bush bearing. This is a bronze bearing that is free to float and is inexpensive to produce.

The main emphasis in the design of turbochargers is for the design of the aerodynamic efficiency of the turbine and compressors. The design of the bearings for conventional turbochargers is of a secondary consideration in comparison to the efficiency of the turbocharger turbine, compressor and design of the wastegate to control the turbo speed.

These small turbochargers must operate at an extremely high RPM exceeding over 200,000 RPM. The failure rate of these turbochargers has been so excessive that it has even attracted the attention of the Consumer Reports discussing the reliability of these turbochargers.

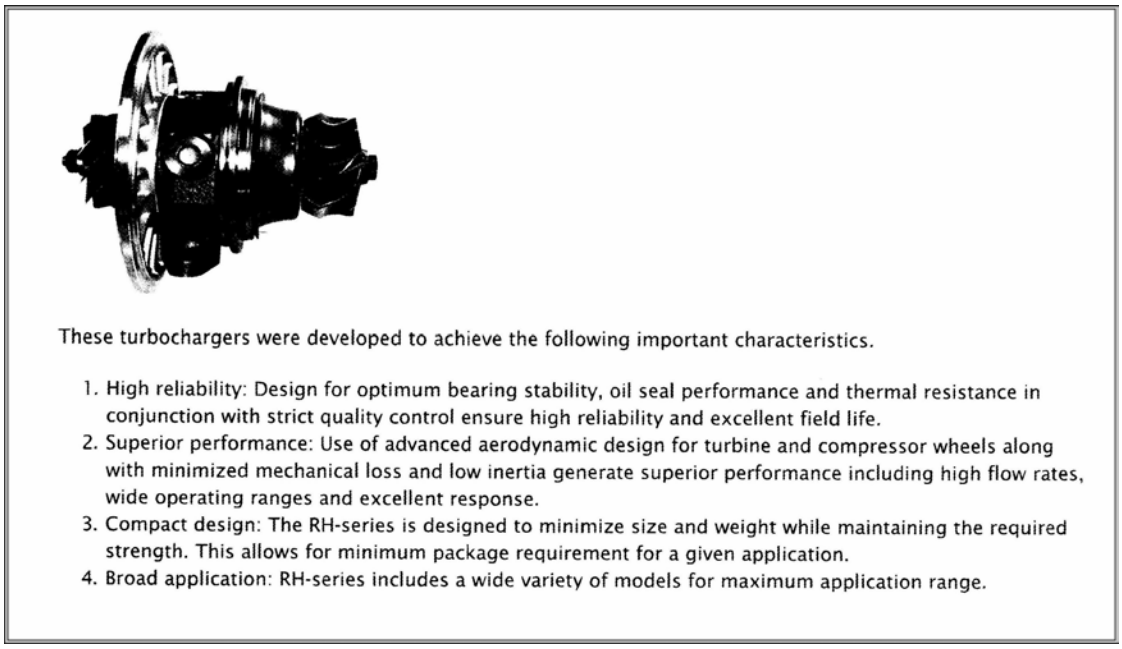


Fig. 5.1 Major Turbocharger Manufacturer Claims On Stability and Reliability

Figure 5.1 represents the the claims of a major turbomachinery manufacture concerning the stability and reliability of their turbochargers. Turbochargers which use floating bush bearings cannot be optimized for stability. These bearings are inherently unstable at all speeds. The reliability of these bearings decreases dramatically when they are employed in small high-speed turbochargers. This is due to the dynamics involved and the difficulty of properly lubricating these bearings

5.2 Conclusions and Recommendations

Figure 5.2 shows the wear marks on the compressor and turbine end bearings. The compressor end bearing shows full wear marks around the circumference. The turbine end bearing shows the narrow wear band due to the conical whirl motion of the turbocharger. Figure 5.3 shows that the compressor internal clearance has been opened up due to wear from under one mil to 10 mils. This is a major cause of failure for the turbochargers. The enlargement of the internal clearance of the compressor bearing allows the compressor wheel to rub against the intake volute causing turbocharger failure.

The turbocharger compressor end bearing is too narrow to carry the dynamical loading and it is also insufficiently lubricated to prevent internal wearing.

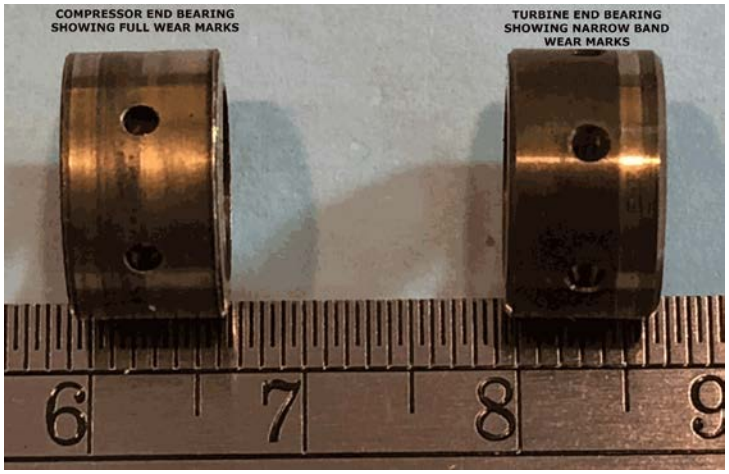


Fig. 5.2 Comparison of Compressor and Turbine Floating Bush bearings

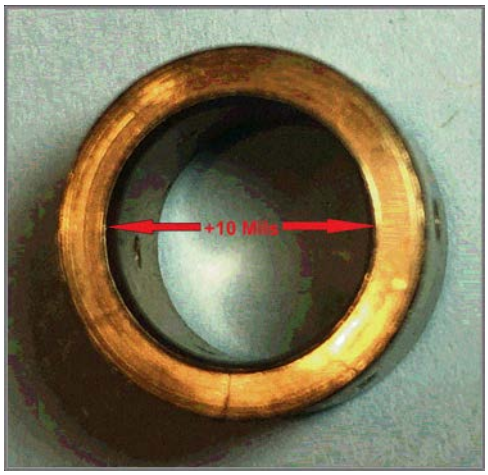


Fig. 5.3 Compressor Bearing Showing 10 Mil Clearance

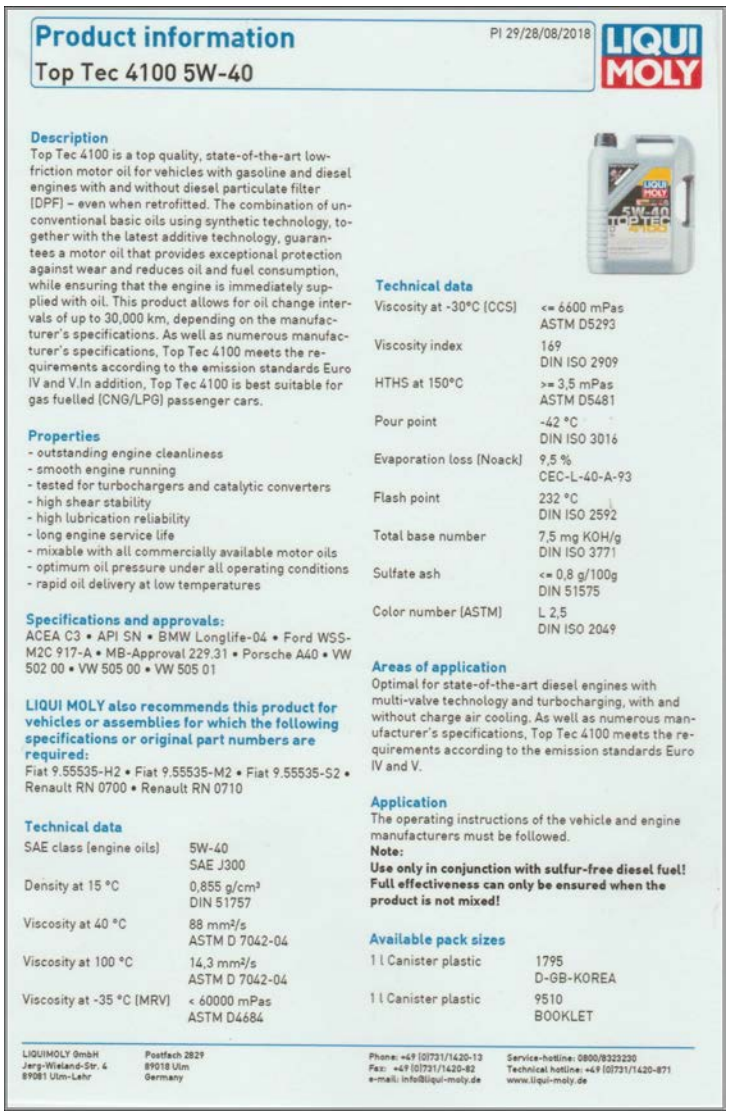
It is recommended that the compressor end bearing be doubled in length in order to carry the required dynamical loading. Such a design change may be resisted by the turbocharger manufacture as this would increase bearing friction losses and would increase turbocharger lag. The other recommendation is that an external central groove be added to the compressor end bearing in order to provide better lubrication for both the outer and inner bearing surfaces.

It is also apparent that the current synthetic lubricant being used for these turbochargers does not have the sufficient boundary lubricant properties in order to prevent excessive wear. A superior synthetic lubricant is recommended such as the Royal Purple synthetic oil or the German manufactured Moli lubricant which is listed in the VW 502 bulletin as an approved lubricant.

Also in order to improve turbocharger life, a one minute engine warm up is recommended for winter driving conditions. On engine shutdown, after extended operation, the engine should be allowed to idle for one minute to allow the turbocharger to reduce in speed while all the bearings are fully lubricated.

Rapid acceleration or deceleration of these 4 cylinder engines with the existing turbocharger is also not recommended.

The major extreme pressure additive used in this lubricant is moly disulfide which is very effective in preventing wear in bronze bushing bearings. The specification sheet shows that this particular oil has a viscosity index of 169 which is exceptional. In the original specification of oil viscosity index properties by the SAE is that the best Pennsylvania oil had a viscosity index of 100 while the worst oil, a golf benzene ring oil had a viscosity index of zero. A viscosity index of zero would indicate that the oil behavior at room temperature would be like syrup but at operating temperature of 200° it would be more reminiscent of water! The modern turbochargers operating at high speeds at high temperatures requires a high viscosity index in order to properly lubricated and protect the journal bearings.



Product information
Top Tec 4100 5W-40

PI 29/28/08/2018

LIQUI MOLY

Description
Top Tec 4100 is a top quality, state-of-the-art low-friction motor oil for vehicles with gasoline and diesel engines with and without diesel particulate filter (DPF) – even when retrofitted. The combination of unconventional basic oils using synthetic technology, together with the latest additive technology, guarantees a motor oil that provides exceptional protection against wear and reduces oil and fuel consumption, while ensuring that the engine is immediately supplied with oil. This product allows for oil change intervals of up to 30,000 km, depending on the manufacturer's specifications. As well as numerous manufacturer's specifications, Top Tec 4100 meets the requirements according to the emission standards Euro IV and V.In addition, Top Tec 4100 is best suitable for gas fuelled (CNG/LPG) passenger cars.

Properties
- outstanding engine cleanliness
- smooth engine running
- tested for turbochargers and catalytic converters
- high shear stability
- high lubrication reliability
- long engine service life
- mixable with all commercially available motor oils
- optimum oil pressure under all operating conditions
- rapid oil delivery at low temperatures

Specifications and approvals:
ACEA C3 • API SN • BMW Longlife-04 • Ford WSS-M2C 917-A • MB-Approval 229.31 • Porsche A40 • VW 502 00 • VW 505 00 • VW 505 01

LIQUI MOLY also recommends this product for vehicles or assemblies for which the following specifications or original part numbers are required:
Fiat 9.55535-H2 • Fiat 9.55535-M2 • Fiat 9.55535-S2 • Renault RN 0700 • Renault RN 0710

Technical data

SAE class (engine oils)	5W-40 SAE J300
Density at 15 °C	0,855 g/cm³ DIN 51757
Viscosity at 40 °C	88 mm²/s ASTM D 7042-04
Viscosity at 100 °C	14,3 mm²/s ASTM D 7042-04
Viscosity at -35 °C (MRV)	< 60000 mPas ASTM D4684

Technical data

Viscosity at -30°C (CCS)	<= 6600 mPas ASTM D5293
Viscosity index	169 DIN ISO 2909
HTHS at 150°C	>= 3,5 mPas ASTM D5481
Pour point	-42 °C DIN ISO 3016
Evaporation loss (Noack)	9,5 % CEC-L-40-A-93
Flash point	232 °C DIN ISO 2592
Total base number	7,5 mg KOH/g DIN ISO 3771
Sulfate ash	<= 0,8 g/100g DIN 51575
Color number (ASTM)	L 2,5 DIN ISO 2049

Areas of application
Optimal for state-of-the-art diesel engines with multi-valve technology and turbocharging, with and without charge air cooling. As well as numerous manufacturer's specifications, Top Tec 4100 meets the requirements according to the emission standards Euro IV and V.

Application
The operating instructions of the vehicle and engine manufacturers must be followed.
Note:
Use only in conjunction with sulfur-free diesel fuel!
Full effectiveness can only be ensured when the product is not mixed!

Available pack sizes

1 l Canister plastic	1795 D-GB-KOREA
1 l Canister plastic	9510 BOOKLET

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Fig 5.4 Specifications For 5W-40 LIQUI MOLY Synthetic Oil

Also in order to improve turbocharger life, a one minute engine warm up is recommended for winter driving conditions. On engine shutdown, after extended operation, the engine should be allowed to idle for one minute to allow the turbocharger to reduce in speed while all the bearings are fully lubricated. Rapid acceleration or deceleration of these 4 cylinder engines with the existing turbocharger is also not recommended.

5.3 Summary

The inexpensive bronze floating bush bearing is the most common bearing type currently being used in most automotive and diesel engines. The plain journal bearing, for example, is the most unstable of the fluid film bearings. A rotor with a plain journal bearing could not operate much above 8000 RPM before encountering self-excited oil film whirl.

The use of the floating bush bearing was developed in the 1930s with the concept of reducing bearing friction losses. The floating bush is free to spin and develops external forces which, if properly lubricated, keeps the bearing ring from contacting with the rotor casing.

Based on only the bearing linear stiffness and damping coefficients, these turbochargers are predicted to be unstable at all speeds, exciting the first or second critical speeds depending upon their operating speed. What prevents failure of these bearings is the nonlinear fluid film forces developed in the outer film. However, if insufficient lubrication is provided to the outer film, then the rotating ring will contact the rotor casing.

With these ultra high speed turbochargers operating at speeds of over 200,000 RPM, it is exceedingly difficult to provide sufficient oil to lubricate both the outer and inner ring surfaces of the floating bush bearing. The operational characteristics of the automobile are considerably different from that of a commercial turbocharged diesel. The diesel engines run over long periods of time without shutting down the engine. The larger floating bush bearings of the diesel engine and the lower operating speeds ensures adequate lubrication for the floating bush bearings. The intermittent operation of the automobiles, particularly under cold weather conditions, can place a strain on the integrity of the floating bush bearings.

The floating bush bearing designs for the turbine and compressor locations are always identical. However, it is the compressor end bearing that has the severest loading applied to it. This is because at high speeds, the excitation of the second mode causes larger amplitudes and forces to be transmitted to the compressor end bearing. This is caused by the dynamics of the system in which the turbine is considerably heavier than the compressor. From a rotor dynamics standpoint, therefore the compressor bearing should be larger to carry the higher bearing loads.

Huge internal bearing wear of up to 10 mils was observed on the failed compressor end bearings. This high bearing wearing was not observed with the turbine bearings. Increasing the width of the compressor end bearing to provide better load carrying capacity would be severely resisted by the turbocharger manufacturers as this would further result in increased turbo lag, which is one of the major criticisms of turbocharged engines.

What is required at this stage is the design of a superior bearing system, such as a rolling element bearing design with ceramic bearings in an oil damper cartridge, or the use of air foil bearings, which are used in some aircraft air compressor designs. Failure to improve these high-speed turbochargers will eventually lead to more automotive manufacturers resorting to a hybrid engine design, rather than to continue to employ these unreliable turbochargers with their four cylinder engines.

6 Improving the Life Expectancy of the 2 L Four-Cylinder Engine Turbochargers

Although the floating bush bronze bearings are very common for a variety of turbocharged engines, including large diesels and six-cylinder gasoline engines, they are unsuited for use in these ultra speed high-speed applications which are exceeding speeds of 200,000 RPM.

These floating bush turbocharger bearings always operate in an unstable mode exciting subsynchronous whirling of either the first or second critical speeds or both may be excited. As previously mentioned, these instabilities cannot be removed by any combination of the floating bush bearing inner and outer ring clearances, contrary to the claims of the manufacturers.

It should be noted that the turbo compressor designers are exceedingly competent in the aerodynamic design of the turbine and compressors for the efficient design. However, when it comes to the design of the bearings and the understanding of the rotor dynamics, the turbo charger designers in general have insufficient knowledge in this area. Actually, on a grade basis, they receive an **A+** for the design of the impellers, but **F** when it comes to the design of the bearings and their understanding of rotor dynamics.

The principal source of failure and limited turbocharger life is connected specifically with the compressor end bearing. This bearing, in particular, is **overloaded** and **under lubricated** as well as being **unstable in all speeds**, as previously mentioned. The inner bearing clearances are normally set to be less than one mil. Over continued operating cycles, the compressor bearing clearance is worn open due to excessive loads and under lubrication. This eventually allows the compressor impeller to contact the volute resulting in failure. It is apparent that a superior bearing needs to be redesigned for the compressor bearing location in particular.

There are a number of actions, however, that may be currently taken that would assist in improving the life of these turbochargers. Some of these recommendations are as follows:

1. Change the synthetic lubricant currently in use.

It is apparent from reviewing the failed parts and damage on the thrust bearing area that the current synthetic lubricant, Castro Edge, is insufficient to provide the necessary boundary lubricant properties to prevent excessive bearing wearing. It is recommended that the Castro Edge be replaced by the approved VW 502 lubricant **Liqui Moly**. Another superior lubricant is Royal Purple, but it is not currently on the VW 502 approved list.

2. Avoid a cold start ups.

A cold startup of these turbochargers presents a problem since the bearings are not fully lubricated. It is recommended that at least one minute of warm up is to be performed before driving off. This is a common procedure with diesel engines in which the engine is warmed up for several minutes before driving.

3. Avoid hot shutdowns.

In this situation the driver abruptly shut off the engine after a long drive in which the engine has reached operating temperatures. An abrupt hot shutdown can cause a loss of lubricant flow to the turbocharger bearings.

It is important to continue to supply coolant to the turbocharger casing after shutdown to avoid carbonization of the oil in the turbocharger. This has been shown to be a cause of failure with the turbine end bearing.

4. Avoid abrupt speed changes.

If the turbocharger is operating at full power, and the vehicle is abruptly decelerated, then there is the possibility of surging the compressor. Surging of the compressor occurs when the compressor is operated at a high pressure ratio and there is a an abrupt reduction of airflow to the engine. Surging of a compressor can create violent axial and radial forces to act on the turbocharger. This can create high loadings on the journal bearings leading to excessive wear. Also considerable distress was observed on the thrust bearing runners due to excessive axial loading and poor lubrication.

5. Avoid maximum power demands at high speeds.

This condition occurs, for example, if one were driving up a steep mountain grade such as Afton at 70 miles an hour requiring the turbocharger to operate at full power levels. In addition to the instability caused by the floating bush bearing themselves, additional un-stabilizing forces are generated in the turbine and compressor areas. These are referred to as the Alford forces.

These forces can cause the turbocharger to whirl violently in the first mode. The turbocharger will admit a loud distinct tone which is related to the excitation of the rotor conical first critical speed. The excitation of this violent subsynchronous whirl mode is extremely damaging and the conical motion of the turbocharger may cause impact of the impeller with the volute leading to an ***instant fatal heart failure***. ***(In some cases the shaft will shear off at the turbine location.)***

6. Revise the initial compressor and volute clearance set up.

With an open faced centrifugal impeller, the tighter the clearance is with respect to the volute, the higher will be the efficiency. These clearances are set without regard to the amplitudes of motion of the rotor. To improve turbocharger life, it is highly recommended that the automobile manufacturers work with the turbomachinery designers to increase the clearance between the impeller and the volute. This will cause a drop of a few points in efficiency, but it will help to prolong the life of the turbocharger. Such a change, however, may be expected to be met with great resistance from the turbocharger manufacturers.

7. **Pretreat the shaft and volute intake with molybdenum disulfide**.

The pre-treatment of the shaft and volute intake with the molybdenum disulfide will help considerably reduce the wear experienced on the compressor end bearing. The treatment on the volute will also help prevent violent failure of the shaft with impact of the compressor wheel. As the Director of the Gas Bearing Laboratory at the Franklin Institute, the tilting pad bearings and shaft were pretreated with molybdenum disulfide. This was an important process as air has no boundary lubricant properties. Moly, as it is referred to, has excellent chemical properties of low friction. When treated on a clean steel shaft, it interacts with the steel to form a molecular level coating which greatly reduces the friction. The treatment of Moly on the volute intake is of particular importance. For example, if the aluminum impeller should rub on the steel volute, the coefficient of friction may exceed one. The compressor wheel may be visualized as a high speed gyroscope. The contact of the impeller with the volute may cause it to precess in a backwards fashion. When this occurs, the turbine torque will cause the shaft to snap at the turbine location causing instant failure.

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