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High Speed Turbocharger Instability

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ABSTRACT

Automotive turbochargers are known to have operation into the self-excited unstable region. In the past these instabilities have been accepted as unavoidable, but recent developments in analysis and instrumentation may make it possible to reduce or eliminate them. A test stand being developed at Virginia Tech has been used to measure the vibrations of a 3.9 liter diesel engine stock turbocharger with floating bushing journal bearings. Vibration spectrum content clearly identifies the shaft instabilities and provides the basis for additional evaluation of future bearing design modifications. Future research will be devoted to the elimination of the large sub-synchronous excitation.

Keywords: turbocharger, experimental, diesel, stability, rotor, dynamics.

INTRODUCTION

Turbochargers are turbomachines intended to increase the power of internal combustion engines. The first turbocharger was invented in the early twentieth century by the Swiss engineer Alfred Buchi who introduced a prototype to increase the power of a diesel engine. Turbocharging was not widely accepted at that time, but in the last few decades, turbocharging has become standard for most diesel engines [1] and is used in many gasoline engines as well. Engineers and other researchers are still searching for ways to improve turbocharger designs for better performance and lower manufacturing cost.

Since the earliest turbocharger prototypes, researchers have attempted to improve turbocharger reliability and increase turbocharger life. Since vibration-induced stresses and bearing performance are major failure factors, rotordynamic analysis should have been an important part of the turbocharger design process. A thorough rotordynamic investigation was, however, very difficult and relatively few studies were published.

Advances in rotordynamic analysis using up-to-date computation techniques have now made the dynamics of turbocharger rotor-bearing system a tractable problem. Manufacturers have begun using these tools to develop more dynamically stable turbochargers. Design improvements, however, cannot depend on computational analysis alone: on-engine test data are needed.

The work described here aims to evaluate the dynamic stability of a diesel engine turbocharger rotor-bearing system using both analysis and on-engine testing. A previous investigation [2] used a commercial finite element analysis (FEA) computer program [3] to model the dynamics of the turbocharger. That investigation demonstrated how linear analysis can be beneficial for understanding the dynamic performance of the turbocharger system; this research combines that work with on-engine testing.

1. BACKGROUND

A turbocharger consists basically of a compressor and a turbine coupled on a common shaft. The turbocharger increases the power output of an engine by compressing excess air into the engine cylinder, which increases the amount of oxygen available for combustion. Since output of reciprocating internal combustion engines is limited by oxygen intake, this increases engine power [4]. Since the turbine is driven using energy from the exhaust, turbocharging has little effect on engine efficiency. By contrast, a supercharger using power from the engine shaft to drive a compressor also increases power, but with an efficiency penalty. Most modern diesel engines are turbocharged, and many gasoline engines are turbocharged as well. The phrase “automotive turbocharger” will here be used to encompass both turbochargers used on passenger automobiles and those used in trucks and tractors.

An important factor in the design of an automotive turbocharger is the initial cost. The same power increase provided by the turbocharger can be provided by simply building a larger engine. Since engine weight is not a major part of overall weight for a diesel truck, the turbocharger is only competitive if it is less expensive than increasing engine size. For passenger cars the turbocharged diesel must compete with lighter and less expensive gasoline engines. To keep costs down while maintaining reliability, the designs of automotive turbochargers are usually as simple as possible. A common design assembly consists of a radial outflow compressor and a radial inflow turbine on a single shaft. Bearings are mounted inboard, with the compressor and turbine overhung as shown in Figure 1.

The compressor impeller in most turbochargers is made of aluminum (LM-16-WP or C-355T61). Aluminum (LM-27-M) is also used for the compressor casing, unless the compressor impeller is made from material other than aluminum. The turbine rotor sees operating temperatures of 750°C (1380°F) or more and must be of high strength so that thin airfoil sections can be used. A typical rotor material is 713C Inconel, a high nickel alloy. The turbine rotor casing also must withstand high temperatures, but strength requirements are lower. S.G. (spheroidal graphite) iron is used for operating temperatures up to 700°C (1290°F), high-silicon S.G. iron is used for temperatures up to 730°C (1350°F), and high nickel cast iron is used for higher temperatures. The shaft is usually made of high-carbon steel (C1144, EN 19C) to allow induction hardening of the journals. The turbine rotor in most common automotive turbochargers is connected to its shaft by a friction or electron beam welding method. The compressor wheel, or impeller, is usually a clearance or very light interference fit on the other end of the shaft. A locknut is used to hold the impeller against a shoulder on the shaft. Friction from the interference fit and/or nut clamping pressure is generally sufficient to transmit the torque, so no splines or keys are needed. Figure 2 shows the elements of a typical automotive turbocharger.

Most automotive-size turbochargers incorporate floating bushing journal bearings. These bearings are designed for fully hydrodynamic lubrication at normal operating speeds. For low cost and simple maintenance, turbochargers use the engine oil system for lubrication instead of having a separate system. Ball bearings are not used for most commercial engine applications because of higher first cost, shorter life, and difficult access for replacement. Special high performance engines in automotive racing applications can afford the added expense of ball bearings, and current designs make use of ceramic ball elements.

The primary consideration in the rotordynamic design of high-speed machinery is to control and minimize vibration. Large-amplitude vibration is undesirable in that it generates noise and can have large amplitudes that cause rotor-stator rub. In most rotating machinery, the dominant vibration is a forced response to rotor imbalance. There exists, however, another class of vibration termed rotordynamic instability or self-excited vibration. Vibration of this type requires a different design approach to ensure that rotating machinery is quiet and durable. Almost all rotors of automotive turbochargers exhibit both unbalance vibrations and self-excited vibrations [4, 5, 6].

Figure 1. A typical assembled turbocharger rotor

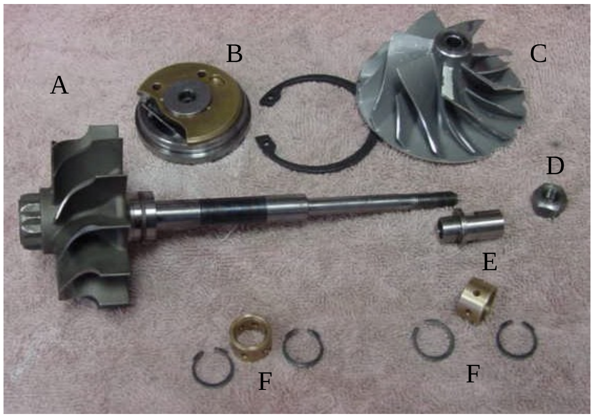


Figure 2. Common turbcharger shaft components

A - turbine and shaft B – thrust bearing assembly
C- compressor impeller D – stock shaft nut
E – test nut and shaft probe target
F – floating ring bearing with retaining rings

Unbalance vibrations are harmonic vibrations occurring at shaft speed. They are generally driven by either mass eccentricity in the rotor or shaft bow. Mass eccentricity is a result of manufacturing tolerances, while shaft bow can be due to manufacturing tolerances or thermal effects. Unbalance vibrations can be minimized by designing the rotating element so that its natural frequencies are well away from operating speeds.

Self-excited vibrations usually occur at frequencies that are a fraction, rather than a multiple, of shaft speed. Such subsynchronous (below shaft speed) vibrations do not require a driving imbalance in the rotating element, but are due to the interaction between the inertia and elasticity of the rotating elements and the aerodynamic forces on the rotor and hydrodynamic forces in the bearings. These instabilities usually increase in amplitude as the running speed increases until nonlinearities in the system forbid higher amplitudes or failure occurs.

There are various mechanisms for self-excited vibration [6], but the subsynchronous instabilities in turbochargers are commonly referred to as whirling or whipping. Whirling or whipping arises as the viscous fluid circulates in the journal bearing clearance with an average velocity of about one-half the shaft's surface speed. The work described here addresses the effects that fluid film bearing design can have on the whipping/whirling instability in turbochargers.

Until recently, rotordynamic design of turbochargers has been mainly based on analysis of forced harmonic vibrations [7, 8, 9]. It was found that floating bushing bearings were more resistant to self-excited vibration than plain journal bearings, and these became widely used. Even with floating bushing bearings many turbochargers show high levels of subsynchronous self-excited vibration [2, 7, 12].

In recent years developments in computational methods have made the analysis of self-excited vibration easier, faster, and more reliable [10-14]. Such analysis is becoming a fairly common part of the turbocharger rotordynamic design process. There has been only limited experimental data for verification of modeling results [7, 14, 15]. The goal of the work presented here is to provide experimental data from on-engine testing for comparison with modeling results.

2. DESCRIPTION OF THE ENGINE TEST STAND

The test turbocharger was installed on a 3.9 liter 130 HP 4 cylinder diesel engine. Design and setup of the test stand was performed by Mechanical Engineering senior students as an undergraduate design project [16]. The engine was installed, using its stock mounts, on a heavy cast-iron test base. It was coupled to a chassis dynamometer with a flywheel adapter plate and a shaft floating between two universal joints. The purpose of the dynamometer was to control turbocharger speed by controlling engine speed and load, not to measure engine performance. Fuel, coolant, exhaust, and control connections were made as needed. Figure 3 shows the engine on the Virginia Tech IC Engine Laboratory test stand.

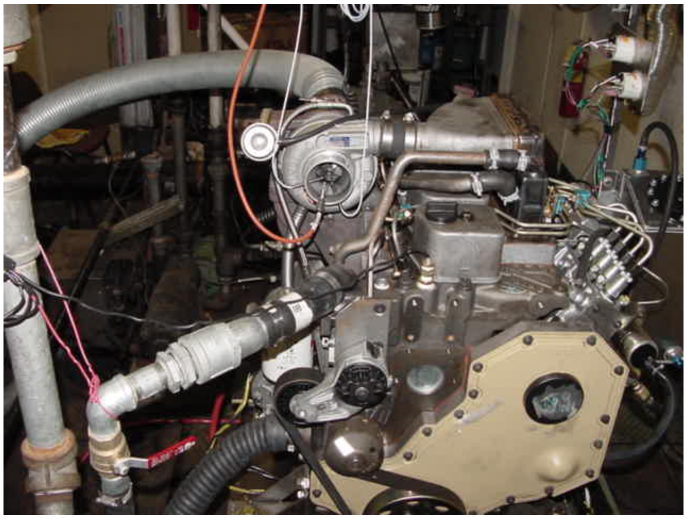


Figure 3. Diesel engine on test stand at Virginia Tech

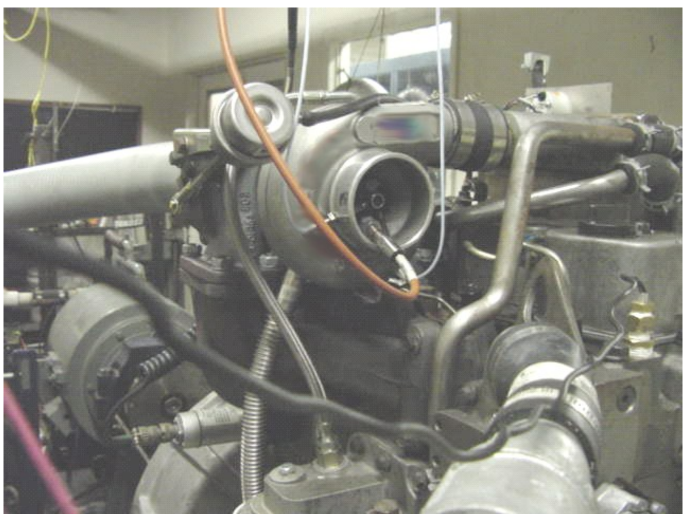


Figure 4. Compressor inlet with optical speed sensor

The basic engine instrumentation included engine rpm from an optical speed sensor monitoring the main shaft, coolant temperature gauge and a turbo boost gauge. Additional engine parameters can be monitored in future testing if desired. A second optical speed sensor and non-contact displacement probes were added to the turbocharger (see Figure 4). In addition, a velocity sensor was mounted on the side of the engine to document the basic engine vibration. The use of available instrumentation was helpful but special order equipment was also required to monitor this high frequency and small diameter shafting. In addition, the lighting for the optical speed sensor target is critical for proper speed detection.

Turbocharger shaft deflection was measured by two special order eddy current proximity sensors measuring the horizontal and vertical displacement of the custom small diameter impeller shaft target nut. An attempt was made to mount two additional proximity sensors to measure displacement of the shaft between the turbine and compressor rotors. Clearance between the probe tip and the housing was inadequate for the sensor to detect the shaft.

Initially, data collection was limited to turbocharger speeds below 60,000 rpm, since that was the maximum data collection speed of the pc based data acquisition system available for the project. Initially the team was investigating ways to reduce the once per turn signal, but a commercial Keyphasor® conditioner was found that could give the desired speed signal reduction. Changing the Keyphasor® to 1/*n* caused the test speed monitor to respond to every *n*th timing mark, so that the actual maximum test speed was increased to 60,000 x *n* rpm.

The instrumentation used for the project is further described in Table I.

Table I Instrumentation for experimental work

2 - 8 ch PC based data acquisition	2 – optical speed sensors
4 - non-contact shaft probes	2 – velocity seismic probes
2 – amplitude, phase and speed vector filters	1 – boost pressure gauge
2 – normal probe drivers	1 – Keyphasor® conditioner
2 - x-y Oscilloscopes	1 – temperature gauge
2 – special probe drivers	2 – power supplies
1 - 400 line FFT Analyzer	1 – engine dyno

3. RESULTS AND DISCUSSION

3.1 Test Results for No Load Low Speed Initial Test Run

The speed limitations of the data acquisition system required the first run to operate below 60,000 rpm. The cascade plot of that run, Figure 5, shows the low order engine harmonics and also the onset of a tracking bearing instability that was near 25% of synchronous speed (0.25x) at just over 50,000 rpm operation. The synchronous (1x) component is below 2 mil-pp and the vibrations are essentially all forward whirl components. This first run shows a small tracking component that starts at 1x and ends at about 0.56x. Future runs do not show this tracking frequency.

Figure 6 gives example shaft orbits at the top speed. These orbits show the large whirl, Fig 6a, and illustrates the dominant instability overpowering the small synchronous component of vibration, Fig 6b.

The waterfall plot, Figure 7, more clearly shows the tracking components. The turbo 1x is 1.58 mil-pp at 50,200 cpm, while the largest instability is 3.95 mil-pp at 14,100 cpm and the small tracking frequency is 0.68 mil-pp at 28,200 cpm. Further study of these components will provide valuable information to more fully understand the turbocharger instability.

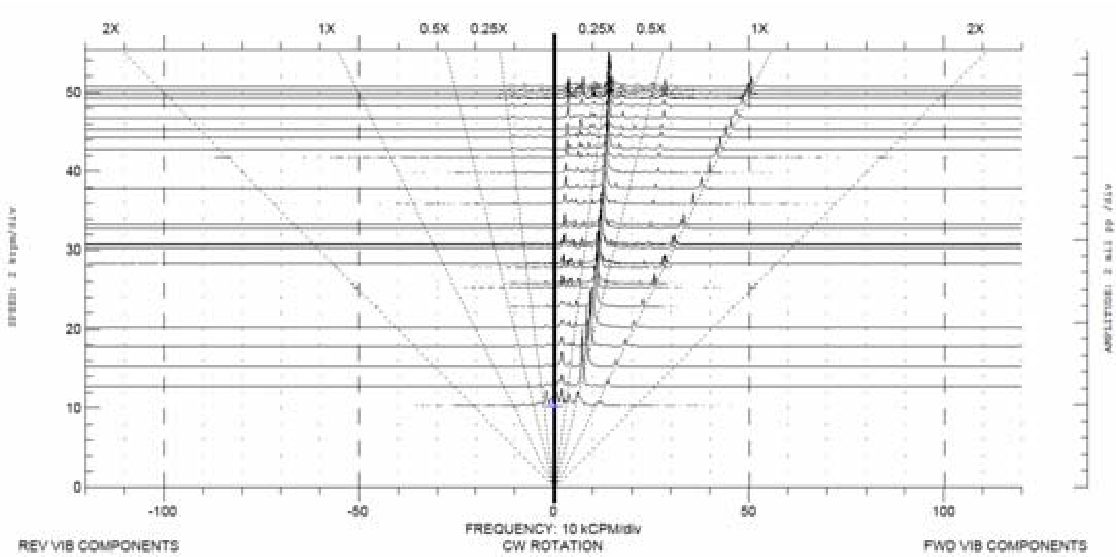
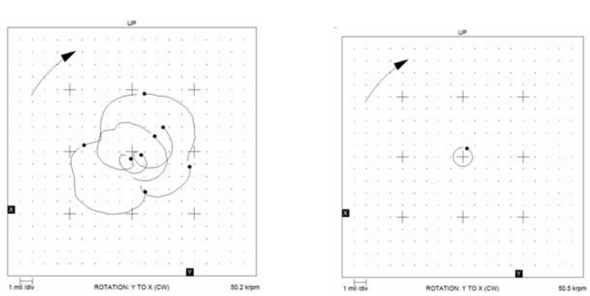


Figure 5. **Cascade plot for first run of engine April 11, 2006, with turbo speed limited to less than 60,000 rpm and Keyphasor® at 1/1. Results are for compressor shaft target nut horizontal and vertical probes.**



(a) **(b)**

Figure 6. **First run, speed just over 50,000 rpm.**
a) Orbit for direct amplitudes.
b) Orbit for synchronous amplitudes.

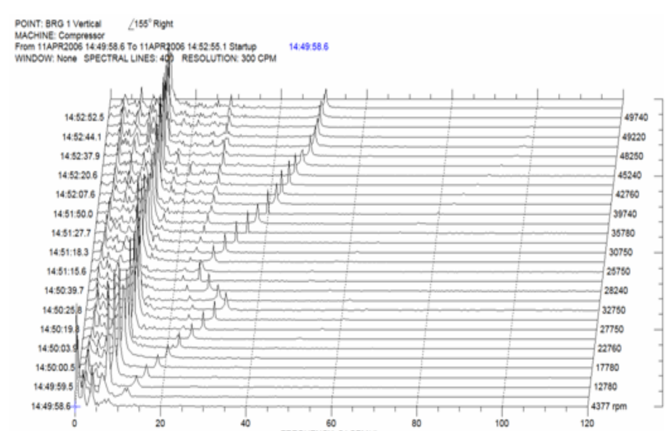


Figure 7. **Waterfall plot of vertical probe for operation up to 50,200 rpm.**

3.2 Test Results for No Load Engine Full Speed Operation

The following week a Keyphasor® conditioner was available to allow the altered speed signal and the engine was tested at full speed, but still without the dyno connected, to see the max turbo speed for this condition. It was very interesting to see the turbo was able to go to near 100,000 rpm, without any load on the engine. This fact could be very useful for future proof tests of bearing stability, since the dyno is not required to achieve these high speeds. However, the engine operation at such high speed and no load is not generally a good idea for long duration operation. Figure 8 shows the waterfall plot for the first run to top engine speed and no load, which occurred on April 18, 2006. The turbo is noted to be operating at 97,200 rpm for the max point recorded and a new, higher frequency instability is noted to occur at 80,000 rpm with a frequency of 33,900 cpm. The lower instability at 18,000 cpm seems to drop out at the same time the new one occurs. The reverse actions occur on decel. The vibration remained essentially forward whirl for all frequency components. The largest instability was noted to have an amplitude of 4.35 mil-pp at a frequency of 35,400 cpm when the turbo was at max speed of 97,200 rpm.

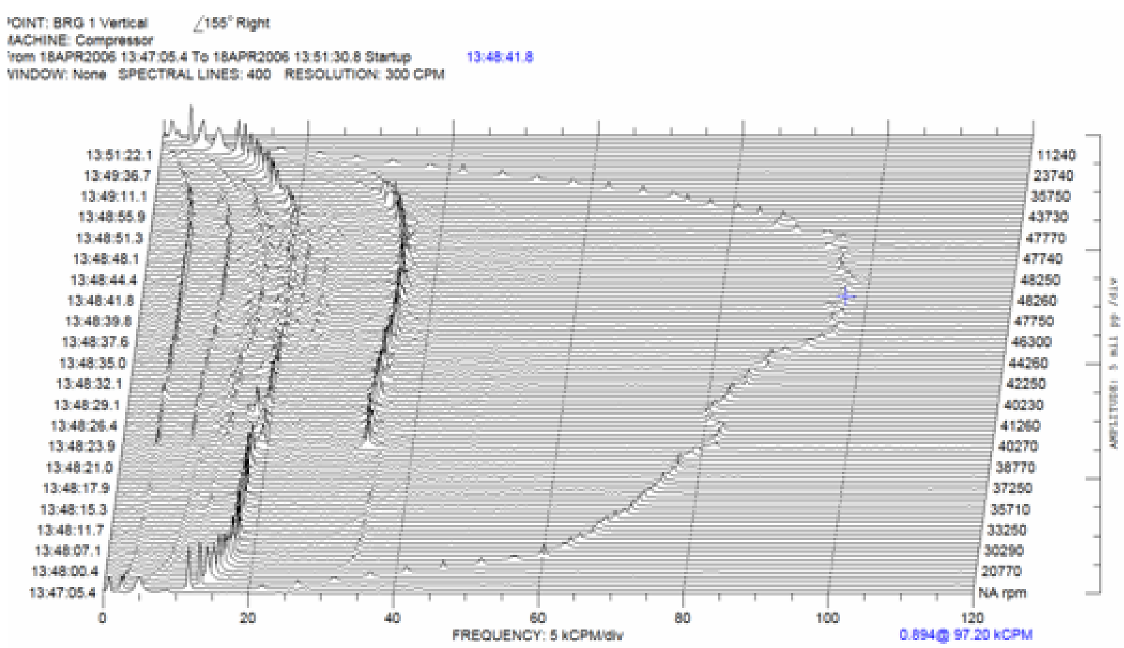


Figure 8. **Waterfall of vertical probe for first full speed run of engine April 18, 2006. Turbo top speed of 97,200 rpm and dominant instability at 35,000 cpm. Keyphasor® at ½ so turbo speeds are 50% of actual frequency.**

3.3 Test Results for Full Load Engine Full Speed Operation

The engine was coupled to the dyno and on April 19, 2006 the first run to full speed and full load was documented using a Keyphasor® of 1/3, meaning every 3rd timing mark was counted. Hence, the actual speed is 3 times the documented turbo speed for each spectrum. The spectrum content is valid as reported. The cascade plot, Figure 9, has a smaller frequency range, the same as the previous plots so the highest turbo frequency is not observed. However, the lower frequencies are better resolution in fact. The lower instability at 15,000 cpm drops out between turbo speed of 51-54,000 rpm while the new, higher instability occurs now at turbo speed of 54,000 rpm with amplitude of 2.96 mil-pp and a frequency of 27,300 cpm. The next scan at 57,300 rpm turbo speed has the instability at 3.26 mil-pp and frequency of 27,900 cpm. At top turbo speed of near 137,000 rpm, the instability is 3.79 mil-pp at a frequency of 35,400 cpm.

The next day, the plot range was corrected and another full speed, full load run was documented. Figure 10 shows the waterfall plot presentation of the instabilities onset and drop out, similar to the previous run, but now the turbo speed is very clearly evident in the plot. Figure 11 documents selected shaft orbits during the decel. The instability is observed to be over 14 mil-pp at top speed and just under 10 mil-pp for the lower speeds shown in this figure.

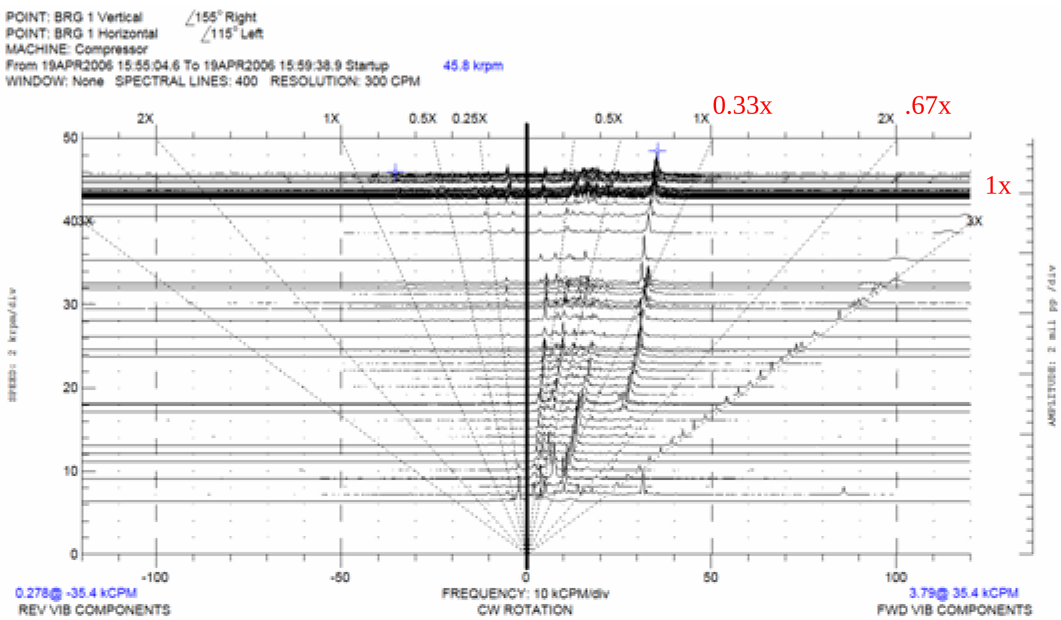


Figure 9. **Cascade plot for first full load full speed run of engine April 19, 2006, Keyphasor® now at 1/3 but frequency range not adequate to show actual 1x speed when operation at full speed. Compressor shaft target nut shows same higher dominant instability at 35,400 cpm at max turbo speed.**

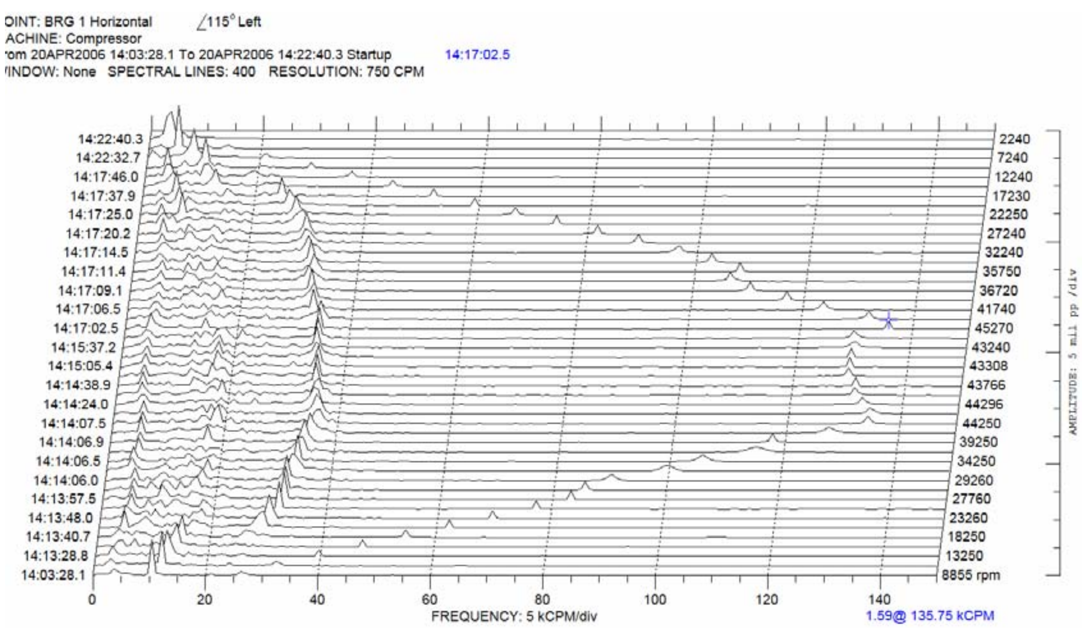


Figure 10. **Cascade plot for second full load full speed run of engine April 20, 2006, Keyphasor® now at 1/3 and frequency range adequate to show actual 1x speed when operation at full speed. Compressor shaft target nut shows same higher dominant instability at near 35,000 cpm.**

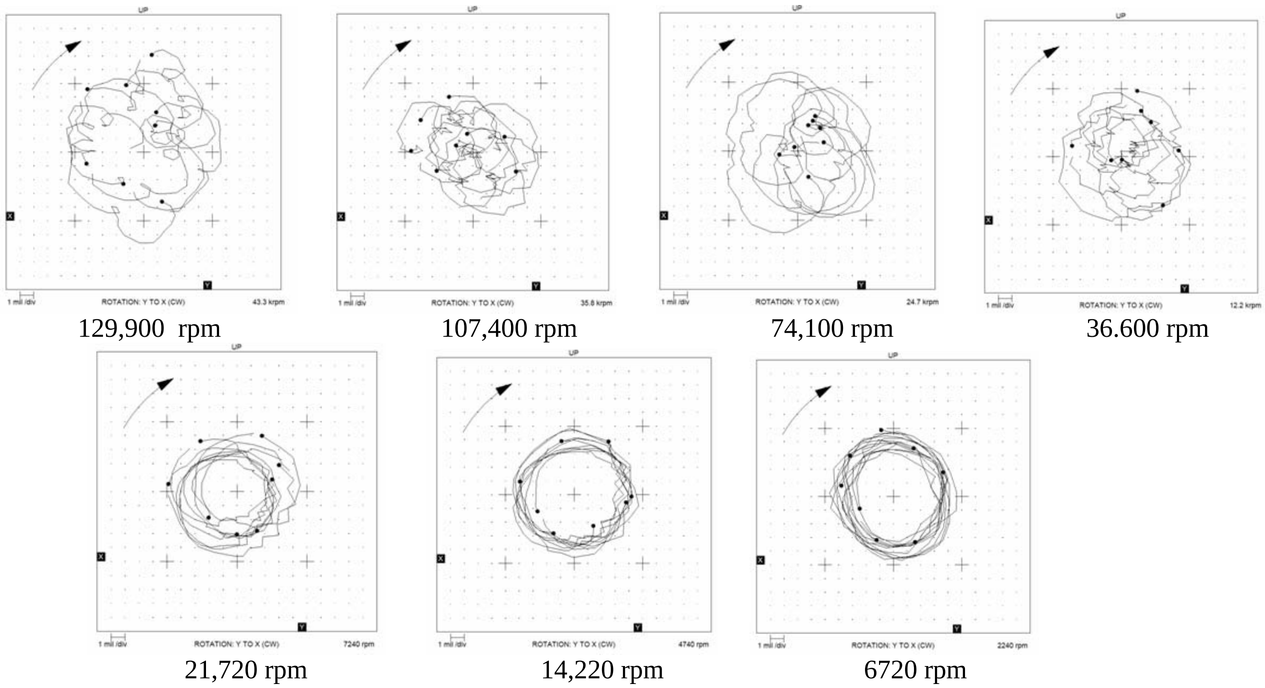
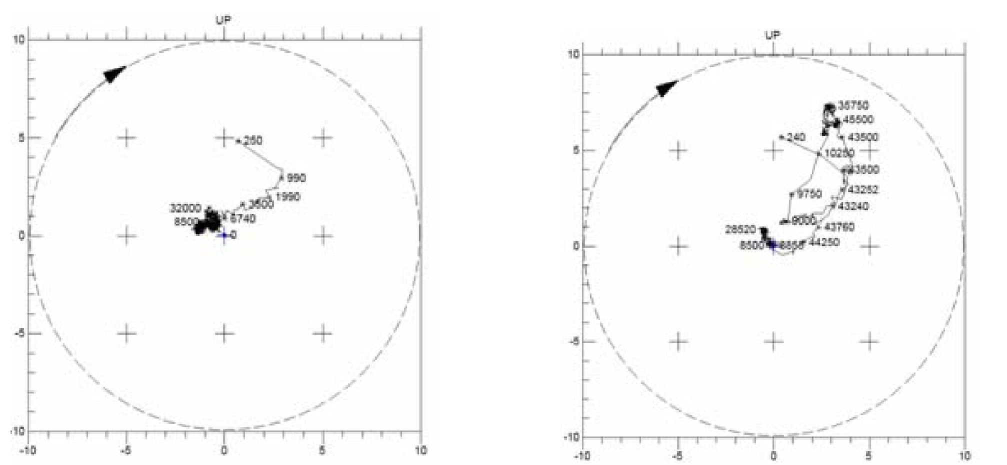


Figure 11. Orbits for selected points of final decel of second full load, full speed run of the engine April 20, 2006.

3.4 Shaft Centerline Result for No Load and Full Load Conditions

An interesting result is observed when the probe DC voltage is tracked during the test run. The results for the no load and a full load condition of Figure 8 and 10 respectively, are presented in Figure 12. For the no load case, the shaft is centered for almost all of the run and then goes up on an arc as the speed drops below about 15,000 rpm, actual speed. The loaded case has more offsets as the engine is loaded and then at 30,000 rpm it re-centers and then comes up the same arc to the upper portion of the bearing. The previous analysis [2] indicates this turbo has a small up load at the compressor end bearing, hence the upward movement as speeds drops.



(a) April 18, no load, ref Figure 8 (b) April 20, full load, ref Figure 10

Figure 12. Centerline plot of compressor end shaft at target nut for a) no load and b) full load. Keyphasor® at 1/3, speeds shown are 1/3 of actual shaft speed in rpm.

CONCLUSIONS

The initial tests for the 3.9 liter 130 HP diesel engine with stock turbocharger operating at no load and full load were successful in providing insight into this particular turbocharger's dynamic performance. It is believed to be typical of high speed turbochargers operating on floating ring bearings. All runs to date have been run with the same factory built test turbocharger with a special shaft target nut to provide a proper target for the non-contact probes used to detect the shaft vibration. The reduction and evaluation of this information, a small portion of which is included in this paper, permits the following conclusions:

1. It is possible to capture the shaft motion over the entire operating speed range of this turbocharger with the special data acquisition system employed in this project.
2. The dominant vibration is a result of subsynchronous self-excited instability.
3. A lower frequency and higher frequency instability have been detected in all full speed runs to date.
4. Further analysis is required to fully identify the source of the instability energy.
5. Tests to date indicate a side load force when the engine is loaded at full speed, that moves the shaft eccentric.
6. Additional testing with different bearings is necessary to further identify the bearing influence on these instabilities.

Additional testing is currently underway with a new project team and the current plans are to run numerous bearing designs with additional instrumentation as funding allows.

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