Experimental results of a novel tilting pad bearing operating in a small high speed turbocharger

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ABSTRACT

High speed turbochargers are known to operate with limit cycle vibration as a result of fluid-film instability. The goal of this research is to have a stable synchronous response with a minimum of non-synchronous contribution excited only by the engine dynamics and exhaust pressure pulsations. Previous papers have documented experimental results of the fixed geometry bearing designs. This paper documents a new tilting pad bearing concept that has replaced the fixed geometry bushings, with a minimum of modification to the stock bearing housing. The summary of the on-engine testing over the last two years is documented in this paper.

1 INTRODUCTION

Turbochargers are 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. Turbo-charging has now become standard for most diesel engines [1] and is also used in many gasoline engines. Engineers and other researchers are still searching for ways to improve turbocharger designs for better performance and lower manufacturing cost. Since vibration-induced stresses and bearing performance are major failure factors, rotordynamic analysis should be an important part of the turbocharger design process.

Advances in rotordynamic analysis using modern computation techniques have made the dynamics of the turbocharger rotor-bearing system much easier and manufacturers can now use these tools to develop more dynamically stable turbochargers. Design improvements cannot depend on computational studies alone and on-engine test data are needed to validate the analytical predictions.

For the past seven years, the research at Virginia Tech has studied the dynamic stability of a diesel engine turbocharger rotor-bearing system using both analysis and on-engine testing. The initial analytic investigation [2] used a commercial finite element analysis (FEA) computer program [3] to model the dynamics of the turbocharger. That investigation demonstrated how linear and non-linear analysis can be beneficial for understanding the dynamic performance of the turbocharger system.
The goal of the current research project is to have a stable synchronous response with a minimum of non-synchronous contribution excited only by the engine dynamics and exhaust pressure pulsations. This paper documents a new tilting pad bearing concept that can replace the small fixed geometry bushings, with a minimum of modification to the stock bearing housing [4]. The summary of the on-engine testing over the past two years is documented in this paper.

2 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 the output of reciprocating internal combustion engines is limited by the oxygen intake, this increases engine power [5]. 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.

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 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, therefore splines or keys are not required.

Many 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. The primary consideration in the rotodynamic 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. Almost all rotors of automotive turbochargers exhibit both forced vibrations and self-excited vibrations [5, 6].

Forced vibrations from imbalance are harmonic and occur at the turbo 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 usually be minimized by designing the rotating element so that no natural frequencies are close to the desired operating speed range. Thermal bowing is the only exception to this previous statement.

Self-excited vibrations usually occur at frequencies that are a fraction, rather than a multiple, of shaft speed. The sub-synchronous 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, the aerodynamic forces on the rotor and the hydrodynamic forces in the bearings.

Rotordynamic design of turbochargers has been based on both linear and nonlinear vibration analysis [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. However, with floating bushing bearings many turbochargers show high levels of sub-synchronous vibration [3, 7, 10].

In recent years developments in computational methods have made the analysis of self-excited vibration easier, faster, and more reliable [11-15]. 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, 16-20]. The goal of the work presented here is to provide additional experimental data from on-engine testing of a novel cage design tilting pad bearing concept [4].

3 DESCRIPTION OF THE ENGINE TEST STAND

The turbocharger used for the experimental work is 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 [17]. 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 increase turbocharger speed by providing engine load, not to measure engine performance. Fuel, coolant, exhaust, and control connections were made as needed. Figure 2 shows the engine on the Virginia Tech IC Engine Laboratory test stand. 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. In addition, a velocity sensor was mounted on the side of the engine to document the basic engine parameters.
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.

4 RESULTS AND DISCUSSION

The test results have been documented using an existing commercial PC based data acquisition and reduction system. The limitation on the reference shaft frequency was much lower than required for this high speed turbocharger. This made it necessary to use a Keyphasor® conditioner to allow the actual shaft speed to be reduced by a factor of three (3), thereby permitting a shaft speed of up to 180 kRPM to be documented for spectrum content. All full speed and loaded test runs use a Keyphasor® factor of 1/3. The vibration probes are mounted to monitor the special target nut at the compressor inlet end of the turbocharger shaft.

4.1 Description of the new cage tilting pad bearing

The goal of the 2010-2011 project team was to design a new support system or a new bearing for the turbocharger while still using a stock bearing housing as shown in figure 1. It was soon evident that the available space for any added support or a new bearing was very limited. The decision was made to attempt to design a tilting pad bearing that could be installed in the current bearing housing. This dictated that the volume available was in fact the same as the stock floating ring bearing. Standard tilting pad designs have an outer housing or shell that allows the pads to be retained and offers a surface for the pad pivots to contact for support. The use of a bearing housing was not possible, and small pins to hold the pads was undesirable. The decision was made to make the pads from a bearing bronze ring that would have the proper pivot height and arc lengths for a four pad bearing. A cage design holder would position the pads and retain the pads during assembly. The design of the pads and the precision cage to retain the pads was possible by the project team but the machining was left to the ME department machine shop instrument makers, who were willing and able to hold the required tolerances. The pad ring with proper bore was machined into four pads with the required pivot height and arc length, using a custom fixture shown in Figure 3. The design was a 4 pad, 60% offset, average 50% preload and with a bearing clearance of 75 micron. The L/D of the bearing was 0.67 and was determined from the stock bearing length minus the length required for the cage retainer ring pocket design.

The bearing is made with optional o-rings to hold the pads prior to insertion into the bearing body. The bearing housing bore is 15.875 mm and the shaft is 10.98 mm. The design requires holding 5-10 micron accuracy, which is typical for turbocharger tolerances.

Figure 4 shows the cage and pads for two bearings, with the o-rings that are used to hold the pads in the cage prior to inserting the cage into the bearing housing. Figure 5 shows a close-up of the cage and pads in a check tool with the pad tilting
much more than when the shaft is inserted. The initial bearing clearance of 75 micron is about three times the stock bearing inner fluid film clearance. It was thought that the tilting pad would have adequate stiffness with this clearance. In fact, the actual operating clearance is not known at either the hot nor the loaded condition.

4.2 Test results for original cage design

The baseline vibration that is used for this project is the spectrum for the original stock floating bushing bearings. The result for a March 2011 loaded run is shown in Figure 6. Here both the first and second mode instabilities are evident. This is a typical result similar to repeated runs since the initial full load run in April 2006 [18].

The cage design tilting pad bearing was installed in both ends and the resulting no load run is given in Figure 7, where initially, after the initial start-up transient due to lack of oil, the turbo was running pure synchronous for the first time in all the testing since 2006. It seemed, during the warm-up, that the problem was solved. Even for the initial increase in speed, the vibration was dominant synchronous. Evaluation of the centerline plot for the compressor end probes, revealed that the cage was whirling (moving) slowly and the sub-synchronous vibration was not acceptable.

Figure 6 Spectrum content for stock floating ring design, 3/24/2011
The results were not totally a success, but the fact remained that the vibration was synchronous up to 40,000 rpm. The hope was that a modified cage design would allow the desired results to be obtained.

4.3 Test results for spin restrained cage, no o-rings, loaded

The evaluation of the test results revealed that the cage was whirling slowly in the clearance space of the bearing housing. The excessive vibration was also noted to be worse for the hot running condition following the initial start run-up to design engine speed of 2500 RPM. The turbo can operate at speeds of 80-98 kRPM with a no load condition and engine speeds of 2500 – 2800 RPM. The cage was redesigned to prevent the cage from whirling and the bearing bore had three 1 mm divots milled into both bearing bores to match the three similar tangs on the outer cage ring. These changes are evident in the new cages shown in Figure 8. Figure 9 shows the bearing cage and pads in place for the modified compressor end housing.

The bearings were installed without the o-rings and the resulting loaded run is shown in Figure 10 where the instability is essentially missing until about 65 kRPM.
The instability comes in strong when the engine is loaded at a turbo speed of just over 90 kRPM and the turbo goes to a top speed of just over 140 kRPM. The instability seems to jump from 15 kcpm to 20 kcpm.

4.4 Test results for modified oil groove cage, with o-rings, loaded

When the turbo shaft was removed and examined, it was evident that a slight contact was evident on the shaft in the corresponding cage areas. The minor marks were buffed away and the cage bore was opened by another 254 microns to prevent any future contact. In addition the oil feed groove was ground back and the leading edge of the pad was further blended to improve oil delivery to the pads. The bearings were installed with the o-rings. The loaded run result is given in Figure 11 where the improved performance lasts until near 80kRPM and then a jump to 20 kCPM occurs when the engine is loaded. The top turbo speed is consistent with the previous run but with larger synchronous response and the non-synchronous content is larger as well.

Figure 10 Spin restrained cage design, 60% offset, no o-rings, loaded
Case no. Turf1104

Figure 11 New cage design, modified oil groove, over-bored cage ID, with o-rings and loaded with the dynamometer, Case no. Turf1130
4.5 Test results for mod oil groove, overbore, no o-rings, and no-load
The bearings were rebuilt without the o-rings and a no load run was made. This initial start and run-up, as shown in Figure 12, seems to have less sub-synchronous but the lower 16 kCPM frequency is larger than desired above 75 kRPM. This was the last run of the fall 2011 semester. It is clear that the undesired vibrations get larger when the engine is loaded, or even when the heat soaks into the bearing housing. The unloaded bearing housing temperature can be over 300 deg C and the loaded runs go over 500 deg C. That the thermal conditions are producing a larger than desired bearing clearance is one likely reason for the poor performance at high speeds. The lack of the proper volume of oil getting to the pads is also a concern and will be further evaluated for future tests.

![Figure 12](image-url)

**Figure 12** New cage design, modified oil groove, over-bored cage ID, with no o-rings and no-load condition, Case no. Turf1131

4.6 Test results for modified cage, reduced clearance pads, and with load
The bearings were rebuilt with new pads that have a smaller pivot radius and a smaller bearing clearance. The preload is near 50% and the offset is 60%. The new radial bearing clearance was 18micron (0.7 mil). The clearance is about half of that used for the results shown in the previous Figures 7, 10, 11, and 12. Figure 13 presents the best run to date at full load with a tilting pad bearing. The results are very promising except for a small amount of sub-synchronous content during the engine ramp to full speed. At 12% over design speed, the load was applied to pull the engine back to design speed of 2500 rpm. The turbo goes to a max speed of 138,500 rpm with 66 micron p-p (2.6 mil p-p) on the vertical probe. The highest sub-synchronous peak was 133 micron p-p (5.24 mil p-p) at a frequency of 14,250 cpm while the turbocharger speed was 67,000 rpm with 1x vibration of 34.5 micron (1.36 mil p-p). More runs are in progress and the latest results are encouraging. The goal to have the original second mode instability at 39,00 cpm reduced to a lower frequency and with smaller amplitudes has been achieved to some degree. The desire to produce a turbocharger with zero sub-synchronous excitation may not be possible, and it may not be desirable in fact. This is because a totally synchronous orbit could more easily excite a synchronous thermal instability.

Current work is in progress to design the optimum tilting pad bearing for this particular turbocharger. When the engine is at full temperature, it seems that the higher harmonics of engine crank shaft speed may become important, but this may save the system from the thermal instability.
5 CONCLUSIONS

The testing to date has provided increased confidence that the on-engine testing can be used to evaluate different bearing designs by rebuilding the turbocharger utilizing a consistent procedure for comparison to previous runs. The current results give great hope that the current cage design tilting pad bearing will be able to operate as desired once the bearing clearance and oil flow can be as desired at hot loaded running conditions. Current test results indicate that a bearing radial clearance below 38 micron (1.5 mil) and larger than 8 micron (0.3 mil) will be required for the optimum clearance. The range of preload and offset has not been considered in the current work.

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