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DEVELOPMENT OF SINE-ON-RANDOM ACCELERATED TESTS APPLIED TO TURBOCHARGER ELECTRONICS ACTUATORS

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Abstract. *The current market needs and the high products competitive pressure the companies to invest in product innovation with higher quality, lower cost and ensuring its reliability and durability. Thereat, it is necessary to invest in tests for indicate before that materials undergo damage or do not completely fulfill their functions during day-to-day use. Different vibration tests are involved in the conception phase: development, pre-qualification, qualification, endurance, certification and homologation. Accelerated vibration tests can provide results in a short time which are representative of real world, avoid over-design and under-design and obtaining identical failure in laboratory as in field. It is very important to know the vibration components which affect the product and obtain the excitation amplitudes values in each frequency to apply in shaker tests, reproducing the same failures modes due random and sinusoidal vibrations. The objective of this work is developing a procedure to generate a sine-on-random profile to test electronic actuators of Variable Geometry Turbocharger in accelerated vibration shaker tests. It was used the Taylor Methods and Maximum Response Spectrum/Fatigue Damage Spectrum as parameters to compare the accelerated Power Spectrum Density based on field data with standard procedure defined by ISO to develop shaker tests to electronics components used in automotive applications. The results show the benefits of this method to compare different types of tests and it was possible to determine the best profile to test components in a shorter time, submitted to both sine and random vibration.*

Keywords: *Reliability, Durability, Vibration, Accelerated Tests, Fatigue*

1. INTRODUTION

Reliability and durability are much important for products, especially components used in automotive applications. For their validation, field tests are performed to ensure that the product meets the structural durability as well as customer needs, such as comfort and safety.

To determine the component life tests that represent the actual conditions of the application are performed. However, during the short period of field testing is not always possible to check a fatigue failure or crack initiation (Xu Ke, 2011).

If no failure occurs when the component is tested in the field additional tests must be carried out in order to understand the components failure modes. Such tests may require a long time and high cost.

Therefore, to solve this problem accelerated tests, mainly accelerated vibration tests, must be performed (Cull, 2010).

A durability test has to meet the following criteria:

-The test should reproduce the same failure modes, which are observed in real conditions and which represents the application.

- The test cannot be accelerated to cause unrealistic failures due the high amplitudes, which may change the mechanical condition of the component.

Accelerating the tests the same failure modes of the field can be observed in a shorter period of time. Another important point is the possibility to determine the warranty period of the product and minimize the amount of "recalls" and spending aftermarket (Xu Ke, 2011).

In order to define an accelerated fatigue test that can be applied to electronic actuator turbocharger, this paper presents a comparison of spectra and evaluates the damage severities, by Fatigue Damage Spectra (FDS) and Maximum Response (MRS), in accordance with the following tests:

- Test with vibration signal based on the application that is accelerated for 20 hours – Taylor Method.

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- Test with sinusoidal vibration signal and other test with random vibration defined by ISO 16750-3 (2003).
- Testing with sine and random in a single test called "Sine-on-random".

It was possible to define a generic test, considering a single profile with a short duration to perform durability test independent of the application. Therefore in this work a methodology is defined for vibration testing on shaker to validate the component with a shorter time and to avoid that tests be conducted for each specific application.

The usage of FDS and MRS parameters is a technique already validated for accelerated testing, however, this work describes the use of this methodology for Sine-on-random testing, which is generic and allows the reduction of testing time and the amount of tests to validate new components.

2. BACKGROUND

2.1 Turbochargers Fundamentals

Turbochargers are thermal machines responsible for increasing the power of engines. The compressor is responsible for increasing the density of the air supplied to the combustion chamber. To increase engine power is necessary that a larger amount of fuel to burn and therefore a greater quantity of air is demanded. Turbine receives the gases resulting from the combustion process by exhaust manifold causing the turbine wheel rotation and thus transferring the movement to the rotor shaft of the compressor (Impeller).

In addition to their use in sports cars resulting in higher power, turbochargers are also used for applications with needs specific torque curves, also enabling the use of smaller engines, lighter, with less power loss and higher efficiency.

There are different ways to control the shaft speed and consequently the amount of air supplied to the combustion chamber. Highlights are pneumatic valves, also known as "wastegate" and electronic actuators. One of the most recent turbochargers architecture found in the market, which uses electronic actuator, is known as VGT, or even Varied Geometry Turbochargers.

The control of the inlet gases port into turbine is done by electronic actuator coupled to the VGT turbocharger bearing housing. This control signal takes into account the shaft speed measured by a magnetic sensor. This signal is sent to the controller and thus obtains a larger or smaller gas flow going from compressor housing to the combustion chamber, according to Fig 1.

Figure 1. VGT Turbocharger (Cummins Turbo Technologies).

It is observed both low cycle and high cycle fatigue in turbochargers. Low cycle fatigue occurs mainly in compressor wheel since the rotor cyclically expands and contracts due to varying shaft speed over time. To low cycle fatigue calculation, rotor shaft speed signal is acquired in a representative duty cycle of the application and using the histograms of damage and SN curve of the material, it is possible to predict the failure rate over the years and even predict the warranty of components.

To obtain experimental SN curves of materials, tests are conducted in which turbochargers are placed in gas stand exposed to speed cycling test and verification the number of cycles necessary to bring the component to fail.

In the other hand, high cycle fatigue occurs mainly in the turbine wheel. This phenomenon occurs due to the vibration of the blades in resonance when exposed to vibration forced by the shaft rotation and its harmonics.

In order to avoid a high cycle fatigue on the turbine wheel, virtual analyzes are carried out through finite element method during the product development, so the natural frequencies do not coincide with the blades band excitation due to shaft rotation and its harmonics.

Other components coupled to the motor can also be exposed to forced vibrations at low frequencies due to rotation and its harmonics also called orders. This occurs, for example, with electronic actuator used for controlling rotation of turbochargers.

To be evaluated the high cycle fatigue in this type of component are used some experimental tests which simulates the application vibration and verified that it will resist due the stress.

The vibration tests are performed by applying an excitation to component and monitoring the structural integrity and its operation according to which it was designed. Such tests can be performed at different stages of development,

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production or product release. In the development stage the vibration testing can help identify possible improvements of the product, for example in your project. In the production step, the manufacturing quality can be evaluated by destructive and nondestructive tests and finally, in the release stage, vibration tests may be used to verify whether the product is suitable for the specified application (Silva, 2000).

Vibration tests are generally carried out using an electrodynamics driver (shaker), as shown in Fig. 2.

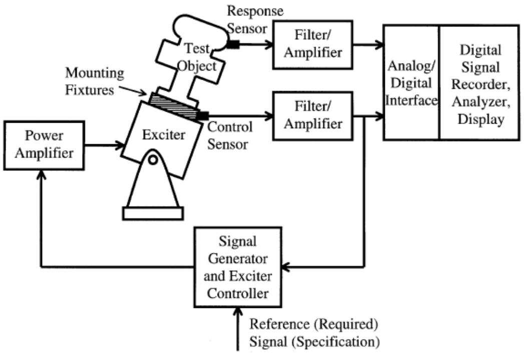

Figure 2. Electrodynamics driver used for vibration test (Silva, 2000).

In addition to the input signal, the response signals of component are also monitored, usually acquired by accelerometers or strain gages, providing signals that can be of acceleration, velocity, displacement, strain or stress. In some cases also are monitored values of temperature, pressure and current, which aim to ensure that the test is performed under the same conditions to which the product will be submitted in the application.

In the input signal are monitored amplitude, phase, the natural frequencies and damping factors, comparing the control signal with the expected in application. The representation of the output and input signals in the frequency domain are crucial in the evaluation of its spectral components, which is obtained by Fourier transform of the acquired signals in the time domain.

Frequency domain analysis are used to obtain the Power Spectral Density of signals, evaluating the best points for instrumentation, the vibration modes of the component and also enabling the calculation of the accumulated fatigue damage (Silva, 2000).

2.2 Taylor Method

The Taylor's method has being used to compose a single vibration spectrum of product that is representative of several application conditions.

In general, the test specifications can be divided into four stages, according to the experimental tests guide, Mechanical Environment (ASTE, 2010):

- Step 1: Life cycle definition that represents the application profile according to the type (ex: city transport, road transport, load vehicle, etc.); usage (Ex: 150 000 km per year), profiles (ex: 80% road - 20% city) and geographic location.
- Step 2: Determination of a representative application signal acquired or estimated in the application.
- Step 3: Test conditions determination
- Step 4: Test execution.

Using the signals obtained in Step 2 and the information from Step 1 the test conditions are defined, taking into account the percentage of time exposed to each application. The envelope method of PSD is then used in order to determine the vibration test signal.

2.3 PSD Envelope Method

Random vibrations are generally represented by power spectral density (PSD). The envelope method determine a single PSD of a particular event, obtained by the composition (envelope) of the frequency components of several PSD calculated using the signals measured in different parts of the application.

Due to limitations of control systems of the shakers used to perform accelerated testing, the resulting PSD should contain a reduced amount of breaking points, usually 30 points are used to represent the spectrum.

The PSD used in the test is the combination of several PSDs of each application condition, but simplified in a single spectrum defined by segments, not necessarily of the same length, as illustrated in Figure 3, according to experimental tests guide Mechanical Environment (ASTE, 2010):

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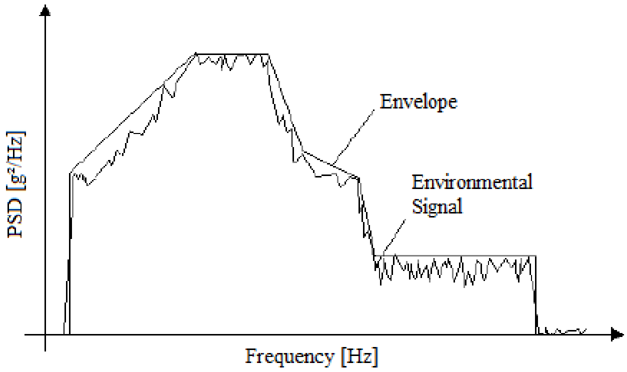

Figura 3. PSD Envelope.

Some factors should be highlighted for the PSD determination:

- As the PSD has only 30 points, the smoothness of the original curve is reduced, but it must keep the main harmonic components of spectrum, avoiding that it becomes purely random.
- The energy of the PSD after the application of envelope must be same as the energy of the PSD with just 30 points.

This energy is calculated by the overall level of the signal, which will be used in the accelerated test, Eq. (1):

Grms_acceleratedPSD=Max[GrmsSpeedPeakHold]*Amplification factor (1)

The GrmsSpeedPeakHold is the maximum overall level observed considering the overall levels calculated for each speed during the test of engine speed sweep (from lower to higher engine speed, assembled on the dynamometer, at idle condition). The amplification factor is obtained by Eq. (2), according to the experimental tests guide, Mechanical Environment (ASTE, 2010):

$$Amplification\ Factor = \left(\frac{T_{real}}{T_{specification}}\right)^{\frac{1}{b}} \quad (2)$$

The b value is the exponent of Basquin relation, which represents the material behavior. The T_{real} is the expected time of component failure in the field, or the time that it wants simulate if the failure will be occur; $T_{specification}$ is the expected time for the accelerated test. The test time used in this work was 21 hours, and b = 0.09 (Copper material). Although the turbocharger is composed for several materials, the lowest b value was used to avoid that test is severe for the material of shorter life.

2.4. Equivalent Damage Method

The calculation of the damage accumulation in the frequency domain has been used to compare the tests severity of different applications and to validate the usage of accelerated tests.

The parameters Fatigue Damage Spectrum (FDS) and Maximum Response Spectrum (MRS) are used to define the accelerated tests represent the same failure modes observed in the field. However sometimes it is not possible to get access to the internal parts of the components and it is necessary to measure only the vibration signal of the structure. This method has strong relation with the need to define tests obtained by measured signals.

Vibration levels, which excite the electronic actuators of turbochargers, can be obtained from the vibration signal measured on bearing housing, since the turbocharger is the main vibration source of the actuator.

The system response for fatigue analysis is evaluated in terms of displacement due it is directly proportional to the energy responsible for the failure (Miles, 1954).

Once the vibration input signal of the system is known, Biot (1932, 1933) developed a methodology which allows the determination of the Maximum Response Spectrum (MRS) and Fatigue Damage Spectrum (FDS).

Based on this works, Bendat (1964) and Lalanne (2002) obtained the expression for the calculation of Extreme Response Spectrum (ERS), defined according to Eq. (3).

$$ERS_{accel}(f_n) = \sqrt{\pi . f_n . Q . G_{\ddot{Z}}(f_n) . \ln(f_n . T)}$$
$$ERS_{disp}(f_n) = \frac{ERS_{accel}(f_n)}{(2.\pi . f_n)^2} \quad (3)$$

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Where:
- "disp" index is the displacement response;
- "accel" index is the acceleration response;
- T : Duration of the excitaion signal;
- $G_{\ddot{z}}(f_n)$: PSD amplitude of the input acceleration signal in the frequency f_n

It is important to highlight that both terms ERS (Extreme response Spectrum) and MRS (Maximum Response Spectrum) can be found in the literature. This MRS parameter is used to evaluate the maximum vibration levels, indicating the maximum amplitudes that the component may be submitted in the application. According to Rice (1954) and Lalanne (2002), the fatigue damage spectrum can be obtained directly from PSD, as seen in the Eq. (4).

$$FDS(f_n) = \frac{f_n T K^b}{C}\left[\frac{QG_{\ddot{z}}(f_n)}{2(2\pi f_n)^3}\right]^{\frac{b}{2}} \Gamma(1+\frac{b}{2}) \tag{4}$$

The Fatigue Damage Spectrum indicates how much damage is accumulated per frequency during the test. The following equations describe better the parameters used in Eq. (4) (Halfpenny, 2001):

- Wöhler curve:

$$N = CS^{-b} \tag{5}$$

S-N: N is the number of cycles for the fatigue with stress amplitude S.

- Gamma Function

$$\Gamma(g) = \int_0^\infty x^{(g-1)} e^{-x} dx \tag{6}$$

- Dynamic Amplification factor:

$$Q = \frac{1}{2\xi} \tag{7}$$

- K: Proportionality constant between stress and strain of the material - Hooke's Law.

It is important to highlight that the dependence of K, C, b and K' does not bound this methodology due to the purpose of this work is the comparative analysis of tests severity. Those parameters only must be the same for different test conditions.

The MRS of accelerated test must be higher than MRS of application signal, however must be lower than SRS, since this is related to an impulsive input which have high energy.

Other concepts of fatigue analysis can be found in Halfpenny (1999, 2001), Bishop (1989) e Downing (1982).

The MRS and FDS were calculated for the accelerated envelope PSD of application, standard ISO 16750-3 (2003) PSD and sine-on-random PSD and compared in this work.

2.5 Standard ISO 16750-3 (2003)

The vibration test methods consider several severity levels of vibration that are applied to electrical and electronic equipments. It is recommended that the vehicle manufacturers and suppliers choose the test method, the environmental temperature and the vibration parameters according to the mounting components location.

The established values are applied to the components mounting on the vehicle. Using a different condition of assembly on the shaker the loads can vary. If the electronic control unit (ECU) is assembled on a vehicle support, all tests of shock and mechanical vibration shall be made with this support.

Once the setting conditions of the component on the shaker are established, a sine sweep should be performed with 1 octave / min. The test must be applied to each of the three perpendicular axes. The scope of test profiles and test duration are defined to check the fatigue failures.

Input accelerations that are outside the frequency bands of the test should be apart considered. The parameters of accelerated vibration test are described in the section 4.1.3.2.2 of the standard. They should be applied to electronic components coupled to the engine of load vehicles.

The engine vibrations can be divided into two types: sinusoidal vibration derived from engine orders (operational components), and random for all other sources of engine vibration, for example, valves closing and combustion. Therefore, it is necessary to test components using both random and sinusoidal tests.

The procedure of the test presented in ISO 16750-3 (2003) can be determined by IEC 60068-2-80 (2005), which states that the test should be carried out as a combination of sine and random signals. In another way, these tests can be performed sequentially.

If the component presents natural frequencies below 30 Hz, an additional test has to be performed for 32 hours in each direction.

According to IEC 60068-2-6 (2007) standard the sine test must be carried out for 94 hours in each direction and the amplitude values for the sine test at each frequency are presented in Tab. 1 (ISO 16750-3 (2003)).

Table 1. Acceleration values established by ISO 16750-3 (2003) for sine tests.

Frequency [Hz]	**Acceleration Amplitude [m/s²]**
20	11.4
65	120
260	120
260	90
350	90
350	60
520	60

According to IEC 60068-2-6 (2008) standard the random test must be carried out for 94 hours in each direction and the amplitude values for the random PSD are presented in Tab. 2 (ISO 16750-3 (2003)).

Table 2: Acceleration values established by ISO 16750-3 (2003) for random tests.

Frequency [Hz]	**PSD [(m/s²)²/Hz]**
10	14
20	28
30	28
180	0.75
300	0.75
600	20
2000	20

According also to IEC 60068-2-6 (2008) 32 additional hours of random test in each direction should be performed when the frequencies are below 30 Hz, as seen in Tab. 3.

Tabela 3: Acceleration values established by ISO 16750-3 (2003) for random tests below 30 Hz.

Frequency [Hz]	**PSD [(m/s²)²/Hz]**
10	50
30	30
45	0.1

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2.6 Sine-on-random Tests

Sine on random tests, known as Mixed Mode Tests, are divided into two steps: one sinusoidal and other random. As said previously, those tests are important for the products that have vibration spectrum composed of random and sine components.

There are military standards that define the sine on random tests (SOR) as a sine sweep superposed by a random vibration. The MRS and FDS parameters are used to validate the method that sums the sine components (narrow band PSD) with the random PSD (Cho, 2010), as seen in Eq. (8).

$$G_{SOR}(f_s) = G_R(f_s) + \frac{U(f_s)^2}{\Delta f} \tag{8}$$

where f_s and $U(f_s)$ are the frequency and the sinusoidal amplitudes, G_{sor} e G_r are the sine on random PSD and the random PSD, and Δf is the PSD frequency resolution.

Once the PSD equivalent to the SOR test is know, the Eq. (3) and (4) are used to calculate the MRS and FDS parameters.

2.7 Analysis of Engine Vibration Orders

Order analysis is widely used for vibration signals related to components coupled to engines. The Waterfall plots are based on a time-frequency transform, but the Y-axis or ordinates has the engine speed variation (rotation -Tacho) and the X-axis or absciss has the frequency. As the engine speed varies with the time, it is possible to obtain the vibration spectrum for each rotation speed.

For most of the rotating machines the vibration components are related to the rotation speed and its harmonics, named orders.

$$fi = \frac{\omega}{60} * i \tag{9}$$

where f_i is the vibration frequency related to the i^{th} engine order. The main orders are multiple of 0.5 and the highest amplitudes are related to the 2^{nd} and 4^{th} orders (engine firing orders) for 4 cylinders engine and 3^{rd} and 6^{th} orders for 6 cylinders engine.

The vibration Peak hold spectrum and order peak hold are also analyzed beyond the waterfall plots. The highest amplitudes for each frequency of the spectrum, which are related to each engine rotation speed, are presented in the peak hold spectrum. The overall levels, which are obtained from each engine order for all rotations, are presented in the order peak hold.

An example of the Waterfall plot for the vibration signal of a 6 cylinders engine is presented in Fig. 4. As observed in the plot, the sloping lines, highlighted by numbers 3 and 6, indicate the 3^{rd} and 6^{th} engine orders. As this engine has 6 cylinders those orders have the highest amplitudes and are related to the vibration harmonics due to the piston movement.

Each order can also be observed in the spectrum, as seen in the Fig. 5, where the spectrum peak hold for the same vibration signal observed in the Waterfall plot, is presented. The same frequency band observed for 3^{rd} and 6^{th} orders can also be seen in the spectrum peak hold of the Fig. 5.

Once the tests are dependent of the engine speed sweep test, non stationary signals at the time, the vibration PSD used in this work are obtained from the Spectrum peak hold of acquired data.

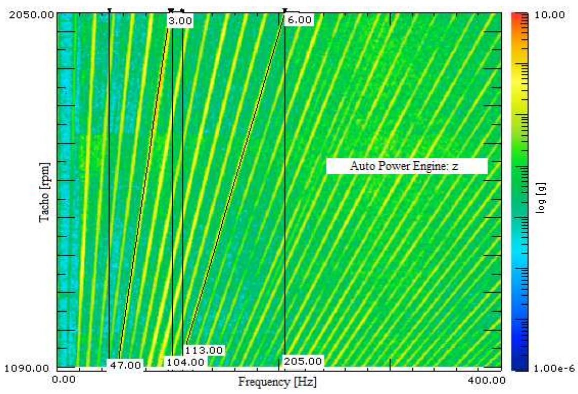

Figure 4. Vibration Waterfall plot (6 cylinders engine)

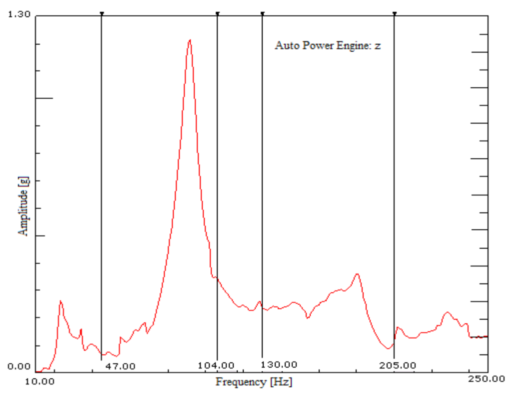

Figure 5. Vibration Spectrum Peak Hold (6 cylinders engine)

3. METHODOLOGY

A speed sweep up test from lower to higher engine speed at idle condition, with the engine full loaded assembled on the dynamometer, was carried out. Vibration data (at three directions) from the engine block, from turbocharger housing and from turbocharger electronic actuator were acquired. The engine rotation was also acquired.

The order spectra of engine vibration signals were compared with the spectrum peak hold of actuator vibration signal.

It is important to highlight that the order spectrum can be represented in function of the frequency instead of function of the orders number, as commonly. The order spectrum can be determined multiplying each order by the fundamental frequency obtained from the spectrum peak hold. Therefore this procedure is performed identifying the frequency value of first order by the comparison between the waterfall plots and the spectra peak hold. The other orders are multiplied by this fundamental frequency and the amplitudes of each order can be shown in function of the frequency.

Using the waterfall and spectrum peak hold analysis it was possible to verify if the vibration components of the actuator are due to forced vibration (operational modes) or natural modes. Therefore, this identification is much important due to the purpose of this work is to determine a accelerated profile of vibration based on a random test, a sine test or the sum of them, which directly depend on the main actuator vibration components.

The vibration signals of turbocharger were used as input to the shaker test, once this signal can be considered as an excitation source of the actuator, which is assembled on the bearing housing of the turbocharger. Using these signals, the accelerated profiles were generated for 20 hours testing in accordance with Eq. (1) e (2) of section 2.3.

The PSD based on the application has a frequency range from 20 to 2000 Hz, also used in the ISO 16750-3 (2003). The standard uses this frequency band because the displacements are low in frequencies upper than 2000 Hz, causing little fatigue damage for the tested part.

The random PSD and the amplitude values for the sine test were obtained by ISO 16750-3 (2003) and the sine on random PSD was determined based on Eq. (8). As the standard specify the test duration of 94 hours for each step at each direction, in this work were considered both a 188 hours test (94 hours for sine test + 94 hours for random test) and a 94 hours test (both tests running in parallel) for the sine on random condition.

Both sine on random profiles obtained were accelerated for 20 hours and compared with standard ISO 16750-3 (2003) test and the test based on accelerated profile of the application, in this case the speed sweep up of the engine.

The MRS and FDS parameters were calculated for the generated PSDs by LMS-Mission Synthesis software. The calculations were performed first for the PSD of the application, second for the sine on random PSD (composed by the sine test + random test) and the third for the specified tests of the standard ISO 16750-3 (2003), such as sine, random and an additional one considering the frequencies below 30 Hz.

Finally the MRS and FDS curves for each condition were compared and they severities were evaluated.

4. RESULTS

The comparison between the spectrum peak hold of the actuator and order spectrum of the engine for the signals acquired during the speed sweep up, is presented in Fig. 6. The curves presented in Fig. 6 are the spectrum of the vibration at vertical direction. Spectra of other directions (axial and horizontal) were not presented due to they have the same behavior that the vertical direction.

As seen in the Fig. 6 the vibration signal of the actuator presents harmonic components multiple of 0.5 engine order. Therefore, the vibration actuator can be related to the forced vibration. As the actuator vibration presents harmonic components, it was necessary to determine an accelerated test considering both as random as sine profiles.

The turbocharger orders were not considered in this work because its speed can reach 200000 rpm and its orders are above the 6^{th} order (frequencies above 2 kHz).

Using the spectra established by the ISO 16750-3 (2003) the sine on random test was obtained.

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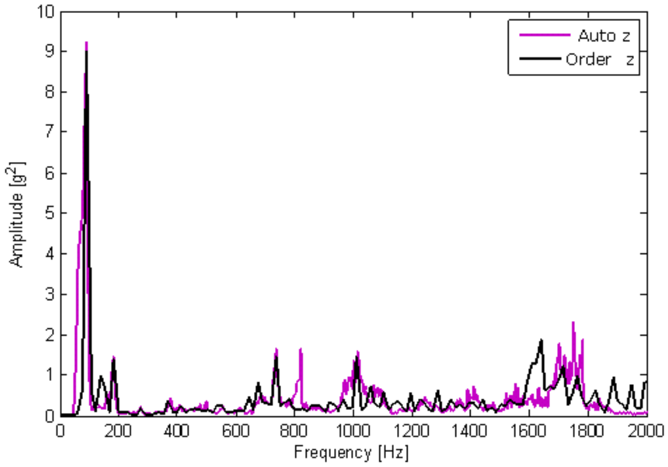

Figure 6. Actuator vibration spectrum and engine order at vertical direction.

In Fig 7 is presented the comparison between the sine on random accelerated spectra based on the ISO 16750-3 (2003), considering the original test of 94 hours (ISO_24_94) and the sum of tests with 188 hours (ISO_24_188); and the accelerated spectra based in the application signal (engine sweep test), considering the spectra for each direction (centralx, centraly e centralz). It is important to highlight that all profiles were accelerated for 20 hours.

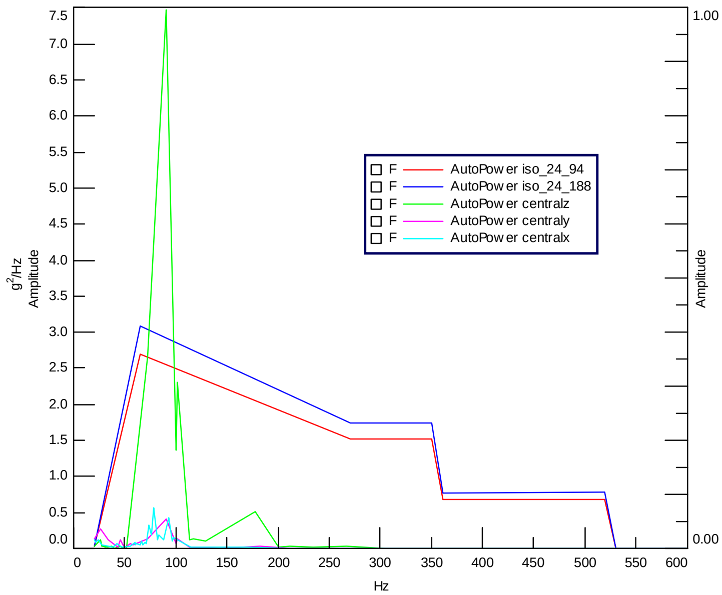

Figure 7. Comparison between accelerated PSD's of Sine-on-random tests specified by ISO and application accelerated PSD.

As observed in Fig. 7 there is a highest peak in the frequency band from 50 Hz to 100 Hz, for the application PSD at Z direction. However there is a wide bandwidth for the sine on random spectra. As there are big differences among the curves presented in Fig. 7, the MRS and FDS parameters are calculated in order to evaluate the severity of the tests and the fatigue damage of the tested part.

The MRS's and FDS's of sine on random tests, of the application profile and those defined by ISO 16750-3 (2003) (iso_sine, iso_random e iso_30Hz) are presented in Fig. 8 and Fig. 9, respectively.

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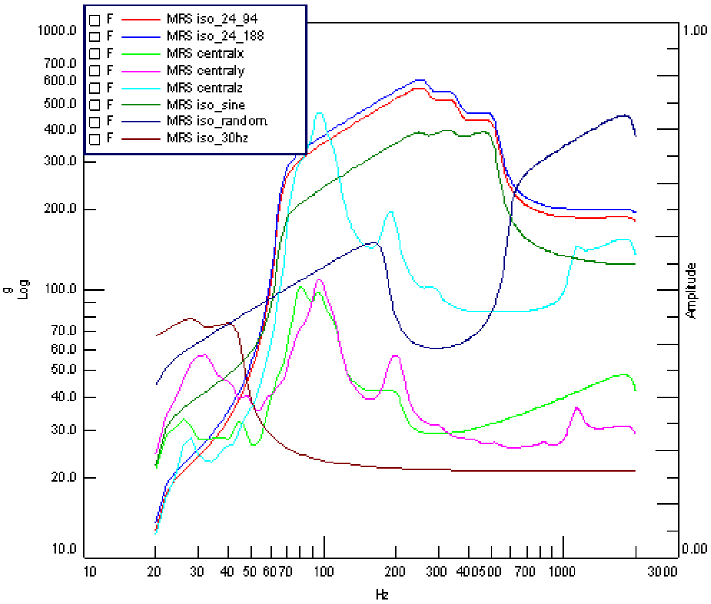

Figure 8. Maximum Response Spectrum Comparison.

According to the Fig. 8 it is important to discuss that:

- The sine on random curves has the same behavior as the original curves of the ISO 16750-3 (2003) but with higher amplitude. This amplitude is higher due to the SOR profile is obtained from the sum of sinusoidal components and the random spectrum.
- However at low frequencies the sine test presented higher amplitudes than the SOR. It has been already provided by the standard ISO 16750-3 (2003), which advise an additional random test, named ISO_30Hz, that also accumulate damage in low frequencies.

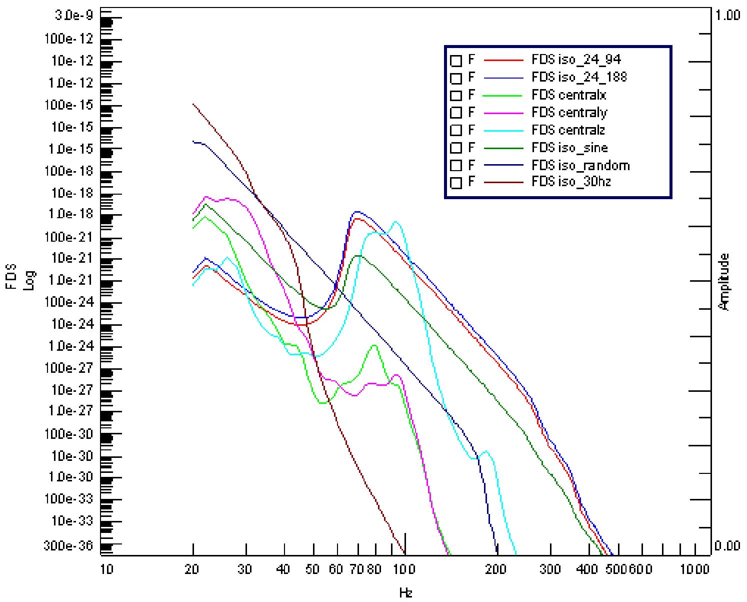

Figure 9. Fatigue Damage Spectrum Comparison.

According to the Fig. 9 it is important to discuss that:

- The sine on random curves present greater damage than the application profile, except for the PSD profile of the actuator at Z direction.
- In accordance to the results, the SOR test based on ISO 16750-3 (2003) can be used to validate new components in a different application, since it is a generic test. If there is no failure observed on the part submitted to SOR test, the future applications can be validated just comparing the FDS and MRS, obtained in the engine sweep test and accelerated for 20 hours, against these limits established by SOR test, without to be necessary shaker tests.

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5. CONCLUSION

With this work it was concluded that:

- The presented methodology for combining a sine wave superimposed on the random signal can be used to "sine-on-random" tests applied to electronics actuators and can be further extended to test any other type of product, since known the amplitudes of the sine and random signals separately.
- It is proposed to use the amplitudes defined by ISO 16750-3 (2003) and accelerated to 20 hours to generate the test SOR and validate the development of new actuators. Once tested in the "shaker" and confirmed their integrity can be concluded that the actuator withstand the severe conditions which may be subjected in automotive applications.
- The evaluation of the actual failure of the component under test can be done by its post-test analysis. The electronic actuator Cummins VGT turbocharger can be tested through software which simulate their conditions of work and receives electronic signals control, which can be used to verify its integrity after the test.
- The occurrence of structural cracks of the actuator can be checked through fluid penetrating analysis or repeating the experimental modal analysis on the shaker and comparing the new values of natural frequencies obtained with the values previously identified. If it is not found large differences in natural frequencies it is concluded that equipment has not suffered structural change.
- The use of FDS and MRS parameters is of great importance in the validation of new applications, since its damage are lower than SOR test performed and validated, it can be confirmed their durability in the field just through theoretical analysis, performing FDS and MRS comparison, without the needs more tests on the shaker.
- It is observed that damage accumulated in application evaluated is less than the maximum provided in ISO 16750-3 (2003), so this application can be validated if the actuator does not fail in the test performed with SOR signals based on ISO 16750-3 (2003).
- The PSD_ISO188 have amplitudes higher than the PSD_ISO94 because simulates a test originally longer in a new test "sine-on-random", but both with 20 hours of "shaker", it means that PSD_ISO188 test is more severe than the test PSD_ISO94 because it simulates in 20 hours a test of longer duration.
- If through comparison between FDS curves it is observed that a new application accumulates more damage than the SOR test, it is necessary to perform the test shaker considering the accelerated vibration profile of application for a test of 20 hours.
- During development of a new product, if the actuator fails in generic SOR test based on ISO, another test may be defined in this case by considering the application vibration spectrum. The product may be released then for that particular application, taking into account the needs to acquire a vibration signal representative of worst case real conditions. In this case, different tracks of the vibration signals can be acquired and then used the technique of envelope PSD.

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