International Journal of Recent Development in Engineering and Technology
Website: www.ijrdet.com (ISSN 2347-6435(Online) Volume 5, Issue 6, June 2016)

Phase Tracking of Resonance Frequency to Avoid Fatiuge Failure

Mylaudy Dr. S. Rajadurai[1], B.Harinivas[2], D. Karthick[3]

[1]*Head, R&D Sharda Motor Industries Ltd, MWC, Chennai-603 002*
[2]*Senior Engineer, R&D, Sharda Motor Industries Ltd, MWC, Chennai-603 002*
[3]*Product Development Engineer, R&D, Sharda Motor Industries Ltd, MWC, Chennai-603 002*

***Abstract*— The vibration testing industry has made good use of sine vibrations to find the frequencies at which a particular device under test (DUT) resonates. These frequencies are important to the vibration testing because they are the frequencies at which the DUT vibrates with the peak amplitude and, therefore, may be the most harmful to the DUT. it can be assumed that the ideal phase value for a resonance is 90° under forced vibration. In reality, however, this theoretical phase value of 90° could be different because the phase value may be affected by the location of the accelerometer or due to a lag in the measurement instrumentation. This paper reveals with the automatic phase tracking of DUT to produce the maximum transmissibility value at a particular resonance under forced vibration using vibration controller. This technique makes fatigue tests more consistent with real-life excitations.**

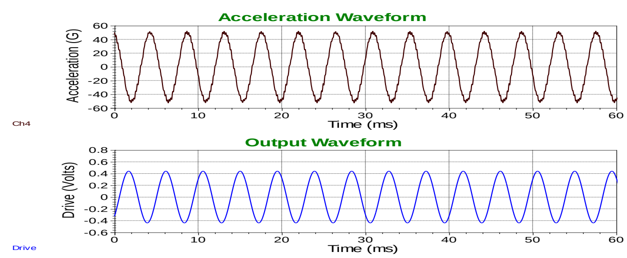
***Keywords*-- DUT, Amplitude, Resonance, Frequency, Phase, Forced Vibration, Real Life Excitations.**

I. INTRODUCTION

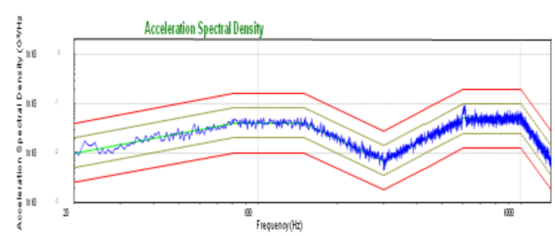
Vibration testing has introduced a forcing function into a structure, usually with some type of shaker. Alternately, DUT is attached to the table of a shaker. Vibration testing is performed to determine the response of a device under test to the defined vibration environment. The measured response may be fatigue life, resonant frequency or squeak. For relatively low forcing frequency, servo hydraulic shakers are used. For high frequencies, electrodynamics shaker are used.

The most common types of vibration testing services conducted by vibration test labs are sinusoidal and random vibration. Sinusoidal vibration is a special class of vibration. The structure is excited by a forcing function that is a pure tone with a single frequency. Sinusoidal vibration is not very common in nature, but it provides an excellent engineering tool that enables us to understand complex vibrations by breaking them down into simple, one-tone vibrations .Random vibration is also more realistic than sinusoidal vibration testing, because random simultaneously includes all the forcing frequencies and simultaneously excites all our product's resonances.

Under a sinusoidal test, a particular resonance frequency might be found for one part of the device under test and at a different frequency another part of the DUT may resonate. Arriving at separate resonance frequencies at different times may not cause any kind of failure, but when both resonance frequencies are excited at the same time, a failure may occur. Random testing will cause both resonances to be excited at the same time, because all frequency components in the testing range will be present at the same time.



1. (a)



1.(b)

Figure1 (a) Sinusoidal vibration, (b) random vibration

The test will be dwell at resonance frequency of DUT because it may cause violent swaying motion and even catastrophic failure in improperly constructed structure. In order to validate a mechanical stability of automobile component the structure should withstand at resonance condition.

International Journal of Recent Development in Engineering and Technology
Website: www.ijrdet.com (ISSN 2347-6435(Online) Volume 5, Issue 6, June 2016)

During test condition, the components which are fixed along with fixture assembly will undergo critical vibration with peak amplitude. So the component fixture assembly should be designed in such a way. This paper mainly deals with tracking the resonance frequency of fixture and more concerned in phase value of resonance experimentally to provide real life excitation.

II. Phase Representation

We may assume that the mass is going to oscillate up and down with a sinusoidal oscillation of amplitude (A). Let's assume that time starts when the oscillation passes through the rest position.

Displacement = x = Asin (ωt)
Velocity =dx/dt = Aωcos (ωt)
Acceleration = dv/dt = -Aω²sin (ωt)

The displacement(x), velocity (v), acceleration (a) are plotted against phase angle shown below

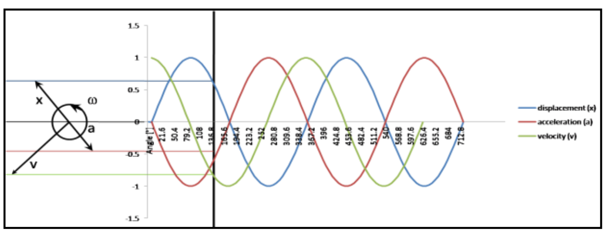
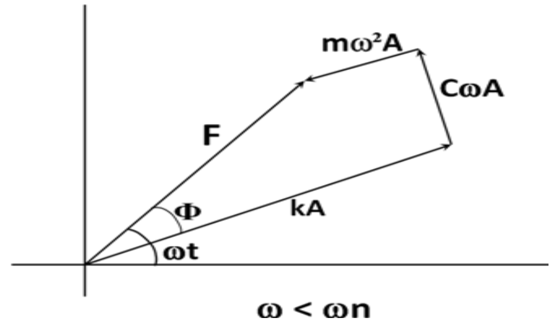


Figure 2 Phase representation of displacement, velocity and acceleration.

From the vector graph we can see that the velocity vector is 90° in front of displacement and the acceleration is 90° in front of velocity.

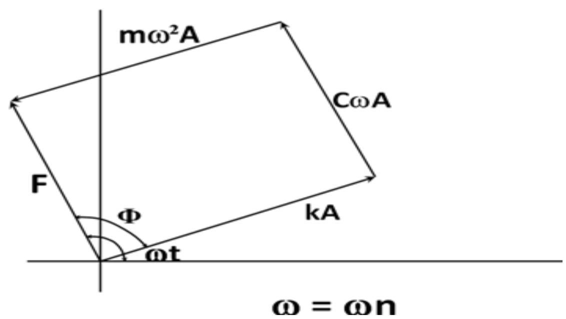
A. $\omega < \omega n$

When the value of ω is very small then the inertia and damping force are reduced considerably resulting in small value of phase (Φ).hence the phase angle is reduced and F is balanced by spring force (KA) which is almost equal and opposite in magnitude.



B. $\omega = \omega n$

When the value of ω is equal to ωn, resonance occur with the phase angle 90°.hence the spring force is equal and opposite to inertia force mω²A and excitation are balanced by damping force.



C. $\omega > \omega n$

At high frequency, inertia force increases very rapidly and its magnitude is very large. Damping and spring force are small in comparison. The phase angle approaches 180°.

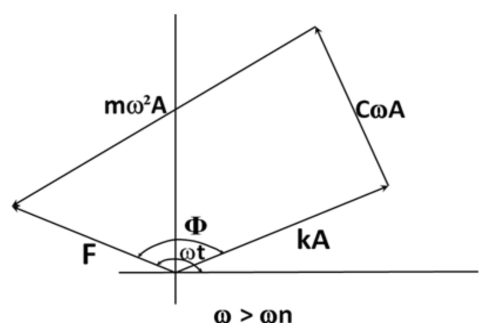


Figure 3 Vector diagram for forced vibration with varying frequency ratio (a). ω < ωn, (b). ω = ωn, (c). ω > ωn

Where, $m\omega^2A$ = inertia force, $c\omega A$ = damping force, Ka = spring force.

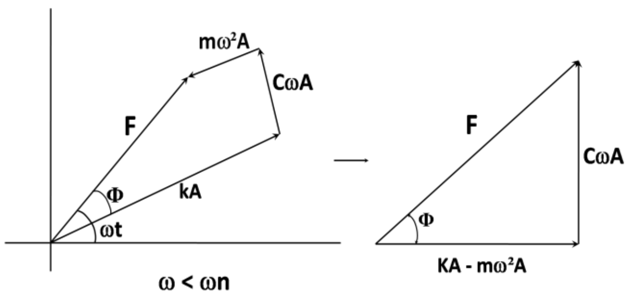


Figure 4 The diagram shows that the applied force (F) is at angle Φ relative to displacement (x).

Applying Pythagoras theorem we have

$F^2 = (KA - m\omega^2 A)^2 + (C\omega A)^2$

Where A is the amplitude of vibration,

$$A = \frac{\frac{F}{K}}{\sqrt{\left(1 - \frac{m\omega^2}{k}\right)^2 + \left(\frac{C\omega}{K}\right)^2}}$$

$$\frac{C\omega}{K} = \frac{C}{CC} * \frac{CC}{2m} * \frac{2m}{k} * \omega = \frac{2\varsigma\omega}{\omega n}$$

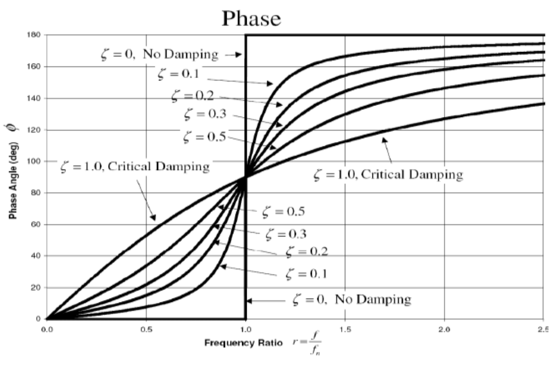


Figure 5 Phase angle variation with respect to frequency ratio for various damping ratios

Damping ratio (ς) is defined as the ratio of damping constant (C) and critically damping constant (Cc).

$$A = \frac{\frac{F}{K}}{\sqrt{\left(1 - \frac{m\omega^2}{k}\right)^2 + \left(\frac{2\varsigma\omega}{\omega n}\right)^2}}$$

$$\tan\Phi = \frac{2\varepsilon\left(\frac{\omega}{\omega n}\right)}{1 - \left(\frac{\omega}{\omega n}\right)^2}$$

$$\tan\Phi = \frac{C\omega A}{KA - m\omega^2 A}$$

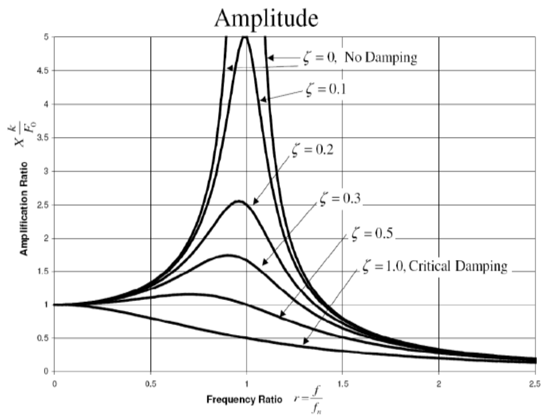


Figure 6 Amplitude variations with respect to frequency ratio for various damping ratio

III. FRQUENCY RESPONSE ANALYSIS (NUMERICAL METHOD)

CAE plays an inevitable role in design and validation of vibration assembly to reduce the number of proto types by understanding the structural behavior in the initial phase of the design. Simulation of those iterated theoretical method is used as an effective tool for the complete modal optimization. Optimization necessitates design modification and crucial changes are orienting the model with respect to the application. Finally the optimized modal is experimented and correlated with the simulation results, to eradicate the assumption on the failure assembly.

Harmonic response analysis is done by the RADIOS software with excitation force on the base plate of the fixture under specific condition shown below.

IV. MODAL ANALYSIS

The base plate is fixed to the shaker so the boundary condition for the model is given as fixed support on the bottom face of the fixture.

International Journal of Recent Development in Engineering and Technology
Website: www.ijrdet.com (ISSN 2347-6435(Online) Volume 5, Issue 6, June 2016)

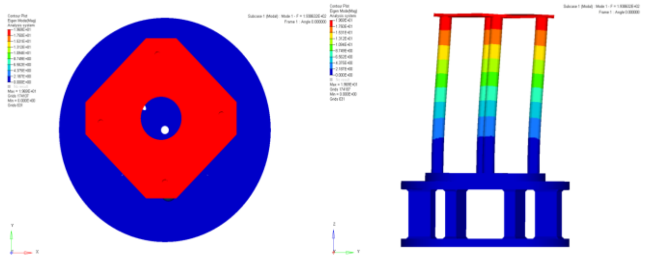


Figure 7 Modal analysis

S.No	Modal Analysis	Frequency Hz	Mode Shape Description
1	First Mode	193	Combined longitudinal and lateral mode
2	Second Mode	194	Combined longitudinal and lateral mode
3	Third Mode	310	Torsion about Z axis
4	Fourth Mode	707	Lateral mode

From the theoretical method we see that the phase angle is 90 degree when the frequency ratio is equal to 1. But in reality, however, this theoretical phase value of 90° could be different because the phase value may be affected by the location of the accelerometer or due to a lag in the measurement instrumentation.

V. Frequency Response Analysis (Experimental Method)

An electrodynamics shaker is used to provide an excitation force to the structure based on the principle of Fleming's left hand rule. An electrodynamics shaker has two separate coils, one is stationary and other is dynamic. A stationary magnetic field is produced in a field coil by constant DC supply and the dynamic magnetic field is produced in an armature coil by varying AC supply. Hence, electrodynamics shakers are controlled by vibration controller which is used to interface between the user and shaker system. This is used to produce high speed digital signal processing and ensure the efficiency of input excitation to counter the sudden change in response of the shaker system.

Dynamic transducers are used to measure the input excitation force and the resulting vibration responses.

Piezoelectric accelerometers are used as a function of transducer.

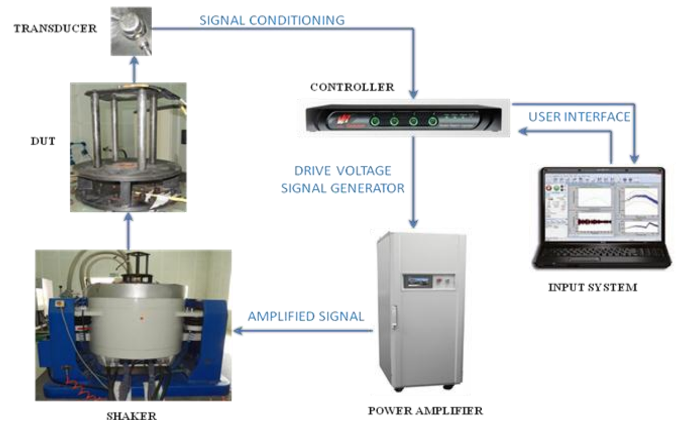


Figure 8 Experimental setup

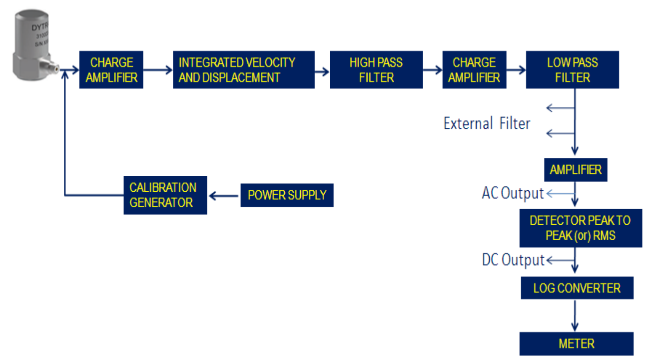


Figure 9 Block diagram of measurement unit

Certain natural and manufactured materials like quartz, tourmaline, lithium sulfate, and Rochelle salt generates electrical charge when subjected to a deformation or mechanical stress. A typical piezoelectric transducer (accelerometer) is shown in Fig. (7). in this figure, a small mass is spring loaded against a piezoelectric crystal. When the base vibrates, The load exerted by the mass on the crystal changes with acceleration, hence the output voltage generated by the crystal will be proportional to the acceleration. The main advantages of the piezoelectric accelerometer include compactness, ruggedness, high sensitivity, and high frequency range. The output voltage of the crystal is given by

$$E = vtpx$$

Where v is called the voltage sensitivity and t is the thickness of the crystal. Px is the pressure due to force applied by the mass with respect to area as shown in figure below

International Journal of Recent Development in Engineering and Technology
Website: www.ijrdet.com (ISSN 2347-6435(Online) Volume 5, Issue 6, June 2016)

Table 1
Accelerometer specification

ACCELEROMETER SPECIFICATION	
Mounted resonant frequency (kHz)	70
Mounting torque (Nm)	2±0.2
Maximum acceleration Range (g)	500
Temperature range (°c)	-54 to 165
Sensitivity (mV/g)	19.97
Output impedance (Ω)	≤ 100
Bias voltage (V)	10.3

VI. Cause Of Errors In Accelerometer

Following parameter causes accelerometer error during vibration test.

A. Powering

Many FFT analyzers and vibration monitors are available with internal accelerometer power supplies. If the user plans to drive long cables the low frequency response may be affected by the input impedance of the measuring instrument.

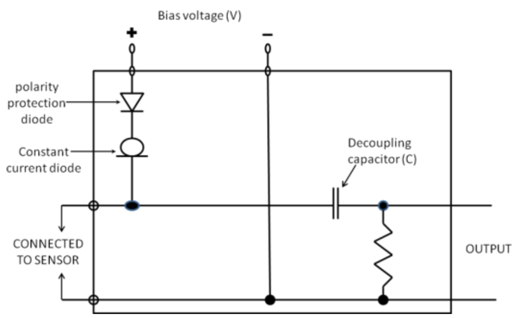


Figure 10 Power supply of accelerometer

Above figure shows a schematic of a simple, single channel power supply coupler for K-shear accelerometers. DC power is supplied from an 11.3-volt source such as regulated power supply or batteries. One advantage of the piezoelectric system is the fact that simple two-wire coaxial cables are used for both power and signals. Because the signal and power both share the same line, it is necessary to include the capacitor C to decouple the DC from the measurement instrument input. Operating within the voltage range of 11-15 Vdc assures full undistorted ±5 volt output amplitude. If the source voltage is reduced, distortion and clipping of signals will occur if one attempts to use the full amplitude range of the accelerometer.

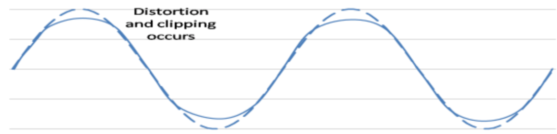


Figure 11 Example of output distortions and clipping when supply voltage is low

B. Accelerometer Mounting

Accelerometer mounting and his position also causes error in acceleration measurement. Different types of mounting position given bellow.

Table 2
accelerometer mounting types

Mounting methods	*Advantage*	*Disadvantage*	*Remarks*
Stud	Best coupling of sensor to test specimen for highest frequency response.	Require threaded hole in specimen.	Control mounting torque, use silicon grease
Adhesive mounting pad	Allows stud mounting, provides electrical isolation	Lowers resonant frequency.	Pads are usually epoxied to test specimen.
Adhesive wax	Ideal for light weight unit	Limited temperature range, not suitable for sensors	
Adhesive cement	Good, strong coupling of sensor to specimen, higher temperature capability then wax	Difficult to remove sensors	Suitable for more permanent application and high frequency measurement
Adhesive tape	Easy application	Only for low g, low weight accelerometer	Only for limited application
Magnetic base	Easy and quick installation	Adds considerably to mass loading, lowers resonant frequency	Limited to ferromagnetic materials

International Journal of Recent Development in Engineering and Technology
Website: www.ijrdet.com (ISSN 2347-6435(Online) Volume 5, Issue 6, June 2016)

VII. Resonance Tracking Methods

Whenever the natural frequency of vibration of a machine or structure coincides with the frequency of the external excitation, there occurs a phenomenon known as resonance, which leads to excessive deflections and failure. The controller tracks the resonance frequency at particular place of channel 3 and 4 mounted over DUT.

To understand this method swept sine test was conducted under the following condition.

In this case the resonance table was indicating that the fundamental mode of Resonance at 745.69 Hz, by considering the peak amplitude of acceleration, where the measured transmissibility is 32.71 (G/G) and the measured phase angle value is -99.89°.

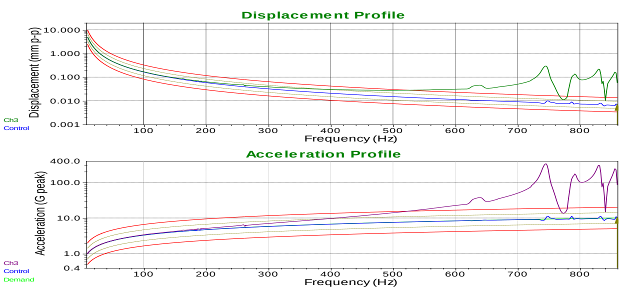


Figure 12 Displacement (mm peak to peak) and acceleration (G peak) variation with respect to frequency (Hz), the red mark indicates an error produced in the reference channel

A. Default phase tracking

Since the theoretical phase of a linear spring-mass system is 90°, many vibration controllers will set the phase automatically to 90° at resonance tracking. The measured transmissibility ratio at 90° phase is 33.8 (G/G).

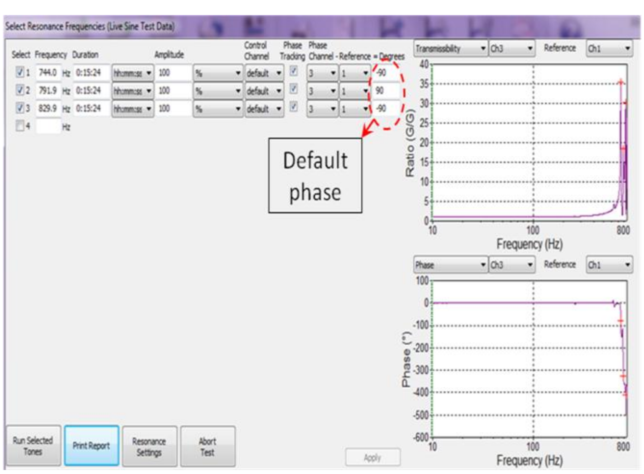


Figure 13 Default phase tracking method

B. Manual phase tracking

The actual phase of the resonance that produced the peak transmissibility was not the default value (–90°) or the predicted value by the software (–99.89°). As can be seen in the peak transmissibility was a completely different value (–77.60°).

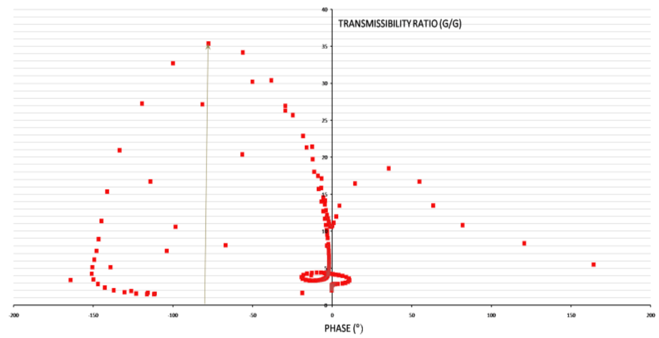


Figure14 Phase vs. transmissibility

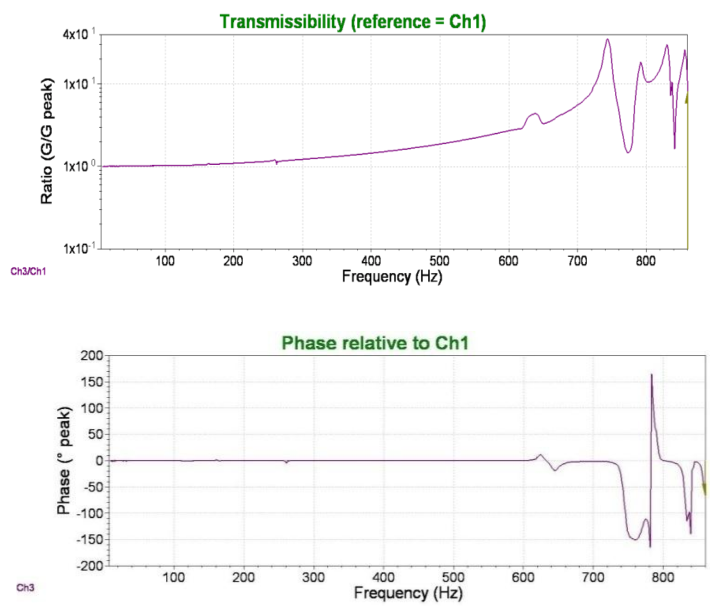


Figure15 Phase and transmissibility variation with respect to frequency

VIII. Exploratory Result

When the advanced user-defined SRTD phase-tracking method is used on the hot vibration fixture, the phase angle was manually adjusted to –77.60°. Then SRTD test dwelt at that phase value of -77.60 at resonant frequency of 744 Hz. At this phase setting and frequency setting, the product experienced peak transmissibility near 35.39 G/G.

International Journal of Recent Development in Engineering and Technology
Website: www.ijrdet.com (ISSN 2347-6435(Online) Volume 5, Issue 6, June 2016)

The normal resonant frequency-tracking method would have set the resonant frequency to 745.6 Hz and would have dwelt there with a phase value of –99.89°. This would have produced approximately a peak transmissibility of 32.71 G/G. The advanced user-defined SRTD phase-tracking method gives approximately a 7.5% increase in the peak transmissibility value compared to the resonant frequency-tracking method.

IX. Conclusion

The results of these tests indicate that the test engineer ought to manually control the SRTD phase-tracking to find the most accurate location for the peak transmissibility of a resonance .This technique makes fatigue tests more consistent with real-life excitations.

REFERENCES

[1] Dr. Sivanandi rajadurai, M.Guru Prasad, R.Kavin, M.Sundaravadivelu, M.Balaji, Computational simulation to ascertain hot vibration test assembly for converter validation
[2] Improving SRTD Testing with Resonance Phase Settings, Joel Minderhoud, Vibration Research Corporation, Jenison, Michigan
[3] "Fundamentals of Sound and Vibrations" by KTH Sweden.
[4] Measuring vibration, Brüel & Kjær
[5] Signal processing for effective vibration analysis Dennis H. Shreve, IRD Mechanalysis, Inc Columbus, Ohio November 1995
[6] Mechanical Vibrations, Singiresu S. Rao, University of Miami.
[7] Electrodynamic Shaker Fundamentals George Fox Lang, Data Physics Corporation, San Jose, California

Abbreviations

SRTD	: Sine Resonance Track Dwell
DUT	: Device Under Test
FFT	: Fast Fourier Transform
AC	: Alternate Current
DC	: Direct Current
CAE	: Computer Aided Engineering

AUTHOR'S PROFILE

Dr. S Rajadurai, born in Mylaudy, Kanyakumari District, Tamil Nadu, India, received his Ph.D. in Chemistry from IIT Chennai in 1979. He has devoted nearly 35 years to scientific innovation, pioneering theory and application through the 20th century, and expanding strides of advancement into the 21st century. By authoring hundreds of published papers and reports and creating several patents, his research on solid oxide solutions, free radicals, catalyst structure sensitivity, and catalytic converter and exhaust system design has revolutionized the field of chemistry and automobile industry. Dr. Rajadurai had various leadership position such has the Director of Research at Cummins Engine Company, Director of Advanced Development at Tenneco Automotive, Director of Emissions at ArvinMeritor, Vice-President of ACS Industries and since 2009 he is the Head of R&D Sharda Motor Industries Ltd. He was a panelist of the Scientists and Technologists of Indian Origin, New Delhi 2004. He is a Fellow of the Society of Automotive Engineers. He was the UNESCO representative of India on low-cost analytical studies (1983-85). He is a Life Member of the North American Catalysis Society, North American Photo Chemical Society, Catalysis Society of India, Instrumental Society of India, Bangladesh Chemical Society and Indian Chemical Society.

Harinivas is a senior engineer in the Hot Vibration Lab of Sharda Motor Industries Limited (Research and Development Center). He completed his Diploma in Electronics & communication. He has also completed B.E in Electronics & communication Engineering. He has done so many Hot vibration testing and actively involved in product development. He possesses strong knowledge in Vibration analysis. He has benchmarked many systems.

D. Karthick is a product development engineer in Sharda Motor Industries Limited (Research and Development Center). He completed his B.E in aeronautical engineering. He has actively involved in vibration analysis and Benchmarking.