Full Length Research Paper

Experimental determination of damping ratios at higher modes for use in modal superposition

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While carrying out the transient analysis, the major challenge is to provide accurate damping ratio data for higher modes as this is not reported. Hence an attempt is made to find out the damping ratios experimentally in this paper. Two approaches of experimental evaluation of damping ratios are discussed, for free and forced vibration, with four different materials namely: mild steel, aluminum, cast iron and plastics. For the first set, eleven cantilevers of uniform cross section with varying lengths are taken for evaluation. For the second set, fourteen components made of these materials used in air brake system of automobiles are also examined. For the free vibration, impact test is used to find the fundamental natural frequency and damping ratio. In forced vibration study, electro dynamic shaker is used to find the first three natural frequencies and damping ratios. Besides the above methods, natural frequencies are calculated analytically using finite element analysis (FEA). Based on the exhaustive experimental work carried out design guidelines on damping ratios for industrial problems are presented.

Key words: Finite element method, experimental determination of damping ratios at higher modes for different materials.

INTRODUCTION

One of the major problems encountered in transient analysis is the data needed on damping ratios corresponding to higher modes, when finite element method is used. It is well known that this depends on the material subjected to transient response and the natural frequency of the corresponding mode of vibration. This is the only method available to compute the damping matrix [C], even when one analyses transient vibration problems using time integration scheme (Ramamurti, 2012). Basic requirement for the transient analysis is to get the damping ratios corresponding to the first few modes if the concept of Rayleigh damping is used in modal superposition technique.

Since damping matrix is expressed as a combination of stiffness and mass matrices, two damping ratios are needed, one corresponding to the fundamental mode and the other one corresponding to a higher mode. These two are obtained experimentally in this article to determine the multipliers α and β.

These transient vibrations are experienced by marine structures subjected to severe impact due to floating ice bergs, rocket launchers during firing, machine tools like forging hammers and guillotine shears, inlet and delivery valves of compressors, brake pads of heavy duty vehicles during braking and space vehicles during launching.

Key researchers in the early era around 1950 like Ungar, Kerwin and Lazan addressed the damping problem. Companies like Bruel and Kjaer and LMS International collected considerable data, but they are not

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Abbreviations: **([C], [K], [M]),** Damping, stiffness, mass matrices of size (n*n); **(B, L, T),** width, length, thickness of the plate; **E,** Young's modulus; **[X],** eigen vector; **(f_1, f_2, f_3),** natural frequencies Hz; **x,** amplitude; **t,** time; **(α, β),** mass and stiffness multipliers; **ρ,** Density; **(η^1, η^2),** frequency ratios on two sides of resonance; **(ω_1, ω_2, ω_3),** natural frequencies rad/s; **(ζ_1, ζ_2, ζ_3),** damping ratios; **(S_1, S_2, S_3),** steel samples; **(A_1, A_2, A_3),** aluminum samples; **(C_1, C_2, C_3),** cast Iron samples; **(P_1, P_2, P_3),** plastic samples; **REV,** relay valve; **QRV,** quick release valve; **MTS,** modular door system; **BC,** brake chamber.
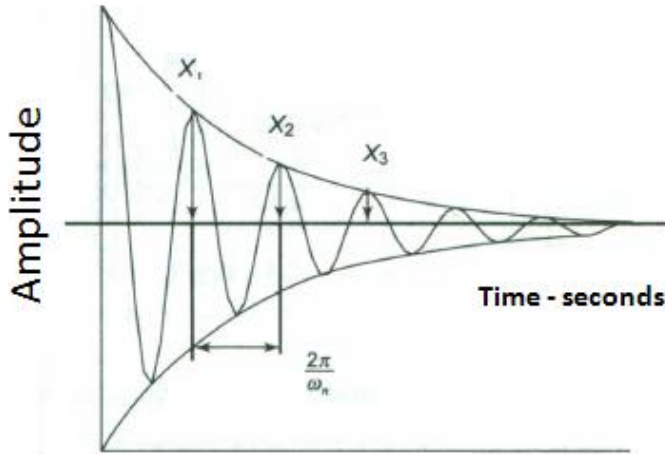
Figure 1. Free vibration response.

available in public domain. Besides most of them are not readily adaptable to modal super position technique. Standard packages like NASTRAN, ANSYS, ABAQUS and ADINA demonstrated Rayleigh damping concept through examples but did not spell out the procedure for computing the stiffness (β) and mass matrix multipliers (α). It is understandable since user manuals of these packages are not expected to provide design guidelines. They did not explain the basis of arriving at these coefficients. Obviously, these are not constants like acceleration due to gravity, but variables which depend on the material with which the vibrating member is fabricated and its first few natural frequencies.

There are not many papers on damping ratios of the first few modes of vibration reported in the last few years. Most experimental investigations are on vibration response of linear and non-linear systems (Bellette et al., 2008; Chow et al., 2005; Huang, 2005; Kuljamic et al., 2008; Mann et al., 2008; Issa et al., 2008; Li hen et al., 2008; Xiao et al., 2005; Zhang et al., 2004). There are a few experimental results on bodies surrounded by water (Grace et al., 2008; Li et al., 2010; Ramajayathilagam et al., 2004)

COMPUTING DAMPING RATIOS

There are two ways of experimentally figuring out the damping present in a system, one by the free vibration record and another by the forced vibration method (√2 method). By the first, one can compute the damping ratio for the fundamental mode only while by the second method it is possible to obtain the damping ratios for the first few modes.

Free vibration response when disturbed

It is well known, that any complicated system, when

disturbed, vibrates, response gradually decreases and, towards the last phase of decay, exhibits the response corresponding to the fundamental mode.

The response of a single degree freedom can be written as

$$x = \exp\left(-\zeta_1 \omega_1 t\right)(c_1 \cos \omega_1 t + c_2 \sin \omega_1 t) \qquad (1)$$

This can be represented by the curve shown in Figure 1. In this figure three successive amplitudes x_1, x_2, x_3 are separated by time difference

$$t = 2\pi/\omega_1 \qquad (2)$$

Ratio of successive amplitudes is given by

$$\frac{x_1}{x_2} = e^{2\pi\zeta_1} \qquad (3)$$

hence,

$$\log\left(\frac{x_1}{x_2}\right) = 2\pi\zeta \qquad (4)$$

Logarithmic ratio of successive displacements is equal to $2\pi\zeta$. By making use of the time period and successive amplitudes towards the dying phase of the multi degree freedom system one can compute ω_1 and ζ_1 of practical vibrating systems. Since these systems have very low damping ratios Equations (3) and (4) used in finite element methods are valid.

Determination of ζ by √2 Method

In the forced vibration method, the system is excited harmonically, at a chosen frequency and the system has the maximum response corresponding to a given natural frequency falling steeply on either side of this natural frequency. Shown in Figure 2, is a resonance zone of the frequency response curve of forced vibration. For a given natural frequency ω_i, let the maximum displacement be x_{max} corresponding to a given natural frequency. Let the frequency ratios for $x_{max}/\sqrt{2}$ values on either side of this resonance be ω_* and ω_{**}. If the natural frequency under consideration is ω_i, the frequency ratios η_1 and η_2 be ω_* / ω_i and ω_{**} / ω_i with the ratio $\eta_m = 1$ at the peak frequency. Let the damping ratio present in the system be ζ.

From the expression for forced vibration response [18], one can write,

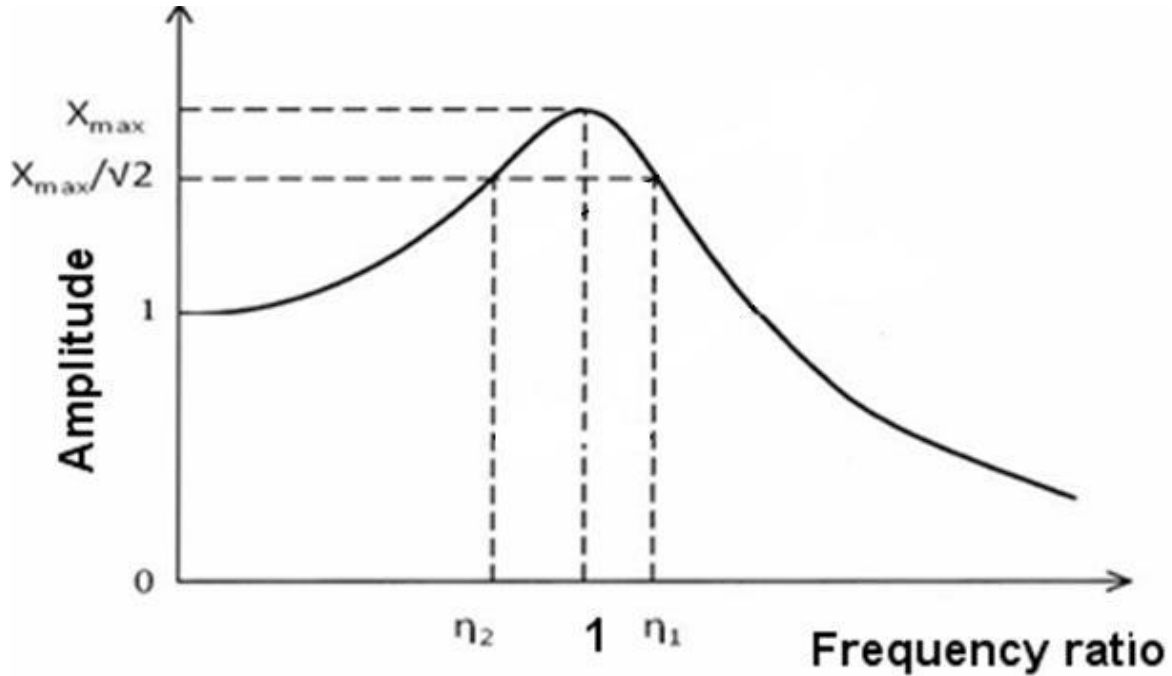$$\frac{1}{2}\left(\frac{1}{2\zeta}\right)^2 = \frac{1}{\left(1-\eta^2\right)^2 + \left(2\xi\eta\right)^2} \qquad (5)$$

Figure 2. Forced vibration response.

Since ζ, for practical systems is <<1.It can be proved that

$$\zeta = \frac{(\eta_1 - \eta_2)}{2}$$

(6)

where η_1 and η_2 being the two frequency ratios as shown in Figure 2.

This method is known as the √2 method. This forced vibration test is a very useful tool for computing the damping ratios of the first few modes even for multi degree freedom systems.

RAYLEIGH DAMPING

For the multi degree freedom system, the determination of damping matrix [C] is not easy. Since only [M] and [K] are available Rayleigh damping assumes that [C] can be expressed as a linear combination of [M] and [K]. It is expressed as

$$[C] = \alpha[M] + \beta[K]$$

(7)

$\alpha \, and \, \beta$ are matrix multipliers to be determined. Since K is expressed in N/mm and M in Ns^2/mm it implies from Equation (7) α must be in s^{-1} and β in s. If Equation (8) is pre multiplied by eigen vectors [X], and post multiplied by

$[X]^T$, one gets

$$[X]^T[C][X] = \alpha[X]^T[M][X] + \beta[X]^T[K][X]$$

(8)

From the basic properties of eigen pairs (Ramamurti, 2012)

$$[X]^T[M][X] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}, \quad [X]^T[K][X] = [\Omega^2] = \begin{bmatrix} \omega_1^2 & 0 & 0 \\ 0 & \omega_2^2 & 0 \\ 0 & 0 & \omega_3^2 \end{bmatrix}$$

(9)

If $[X]^T[C][X]$ = is expressed as [2 $\zeta_i\omega_i$] to make it diagonal and expanding Equation (9) one gets

$$2\xi_1\omega_1 = \alpha + \omega_1^2\beta$$
$$2\xi_2\omega_2 = \alpha + \omega_2^2\beta$$
$$2\xi_m\omega_m = \alpha + \omega_m^2\beta$$

(10)

Equation (10) indicates α and β are not constants, but matrix multipliers which depend on the first few damping ratios and the natural frequencies of the member analyzed. In this paper, forced vibration response is used for experimental determination of the first three natural frequencies and damping ratios. This approach became popular after the finite element method was successfully

used to solve problems in the area of dynamics with computational efficiency in mind.

RELEVANCE OF INVESTIGATION

It may appear, as if, "Computing Damping Ratios" and "Rayleigh Damping" are well known and are reported in most books. If one looks at the industrial scenario, the following may be observed. A marine steel structure experiences disturbance due to floating iceberg, probably once an hour, a multi barrel rocket launcher impact once in four seconds, a guillotine shear made of a steel housing every two seconds (in continuous mode of shearing of thick plates) and exhaust and inlet valves of a reciprocating compressor every 0.02 s (when the compressor runs at 3000 RPM). A typical case of two gear pairs in mesh provides two extremities of time duration, the first, a high speed pinion running at 3000 RPM every 0.02 s and the second, a girth gear running at 2 to 3 RPM every 30 s. All these disturbances are transients which completely die down before successive hits. With duration of disturbance varying very widely, the two parameters ζ and ω play a crucial role in the behavior of the vibrating member. They depend on the material, geometry and boundary conditions of the vibrating member.

If the damping is underestimated, the member receives hits before the influence of the previous hit disappears. If overestimated, the predictions negate what is being experienced. It is easy to compute the first few natural frequencies accurately analytically or experimentally, but damping ratios can be determined only experimentally. It is for this purpose this investigation is taken up for different materials commonly used in mechanical engineering.

PROCEDURE FOR INVESTIGATION

Impulse hammer

The hammer used excites the test structure with constant force over the frequency range of interest and produces an analog voltage pulse which is an exact representation of the input impulse (F.t) both in amplitude and phase. It uses an accelerometer of Low Impedance Voltage Mode (LIVM) which is designed to measure shock and vibration phenomena over a wide frequency range. It contains integral IC electronics that converts the high impedance signal generated by the piezo crystals to a low impedance voltage that can drive long cables with excellent noise immunity. These accelerometers utilize quartz and piezo ceramic crystals in compression and shear mode. The accelerometer is mounted on the test object at the position defined by the customer and the

impact is given using Impact Hammer at the specified point and in specified direction and the data is collected using Focus Dynamic Signal Analyzer. Figure 4 shows the arrangement for this test and Figure 3 the response.

Forced vibration test

One of the major problems encountered in forced vibration test is the range of excitation available in the exciter. Mechanical double mass eccentric exciter produces pure sinusoidal excitations, but the range is limited up to 100 Hz. While heavy engineering units are having lower natural frequencies (in the range 5 to 100 Hz), components in automobiles like inlet and exhaust valves have natural frequencies as high as 18000 Hz. It is impossible to cover this wide range experimentally. In this investigation, a heavy duty electrodynamic exciter with a range up to 3500 Hz is used to collect meaningful data (Figures 5 and 6).

The vibration testing unit consists of a signal generator, a power amplifier and a vibration shaker. In addition to this, instrumentations like accelerometers, signal conditioners, charge amplifiers, Fast Fourier Transform (FFT) are also used.

Dynamic tests are performed with sinusoidal signal sweeping between the defined frequency range. The output of the signal generator is amplified in the power amplifier to a level sufficient to drive the vibration shaker. This vibration shaker converts this amplified signal to mechanical motion at the work table which transfers the same to the test samples. The mechanical motion at the table surface is sensed through the accelerometer. The electrical signal can be used for monitoring and feedback purposes to make the operation as a close loop. The acceleration level imposed on the specimen is measured and continuously compared with the program by the Digital signal controller. The difference between the command and the input signal is used to provide a continuous correction signal to the shaker, which causes this difference to minimum. Specifications of the experimental set up are shown in Table 1.

Digital amplifier

Model DSA 5K15, class D, Technology PWM, Output power 15 KVA (Suitable to drive shaker), Input power 415 V, 50 Hz, 3 phase, Frequency range DC to 3500 Hz.

System control unit

Sinusoidal signal generator routes the path of signal towards the power amplifier. It can use its own signal or signal from the external source and thereafter to the shaker after amplifications.

DUT: Steel Plate S1.   Condition: Impact in Z direction Response in Z direction
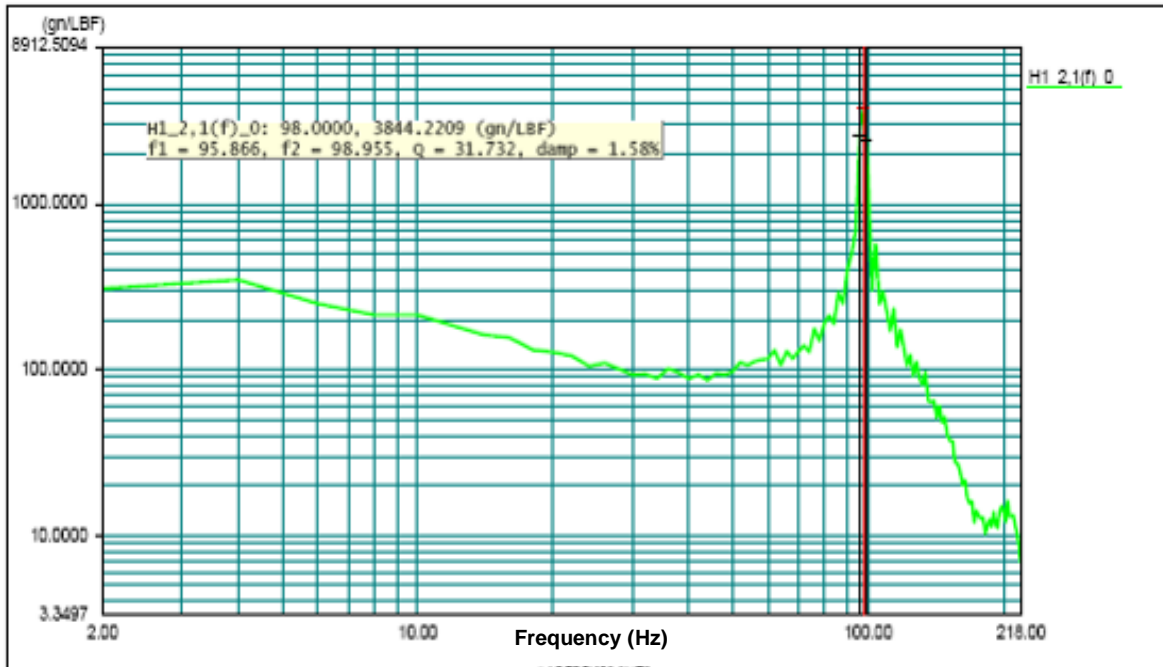

Figure 3. Frequency Vs Amplitude response –Impulse hammer (S_1).

DUT: Steel Plate S1.   Condition: Impact in Z direction Response in Z direction

Figure 4. Impact Hammer-Decoupled excitation.

Figure 5. Vibration shaker.

Figure 6. Experimental setup-forced vibration for the first set.

Accelerometers

Accelerometer of Model 2224C is a high precision Piezo electrical pickup over a wide frequency range. Nominal charge sensitivity 10 pC/g, Resonant frequency > 32000 Hz, mechanical shock limits 2000 'g', Sensing element shear ceramic **,** Transverse sensitivity < 3%, Mounting thread 10 to 32 coax type, Weight 15 g, Cable Microdot to Microdot 5 m length.

TESTS CONDUCTED

Twenty five specimens have been chosen for the analysis. The first set of 11 specimens are cantilevers of

three varying lengths (Table 2) and four different materials (Mild Steel, Cast Iron, Aluminium and Plastics) of uniform cross section (Figure 4). It is well known that their first three natural frequencies in bending mode are decoupled. This program is expected to give controlled values of damping ratios. The other set of fourteen components constituting four materials (Mild Steel, Cast Iron, Aluminum and Plastics) (Ramamurti et al., 2010) are used in air brake system of automobiles and heavy vehicles. These details are shown in Table 3. Test specimens are shown in Figure 7. The first few mode of vibrations are coupled. These are also studied to find out how the complex systems behave.

Assumed ratio of values of E/ρ (taken from Hand books) are the same for steel and aluminium and hence computed natural frequencies of these two will be the same for geometrically identical plates.

All the 25 specimens listed in Tables 2 and 3 are modeled using finite element method (ANSYS) and the first three natural frequencies are computed. Figure 8 shows the first three modes for S_1 using finite element method. The specimens are subjected to free vibration test using impulse hammer. Figures 9, 10, 11 and12 show the free vibration records for steel, aluminum, cast iron and plastics. Forced vibration on electro dynamic exciter is also performed. The results on damping ratios are shown in Figures 13 to 16. Firm lines indicate the response for plates of uniform rectangular cross section (uncoupled modes) whereas dots represents products of manufacture of complicated shapes (coupled modes).

RESULTS AND DISCUSSION

From the Tables 2 and 3, and Figures 13 to 16, the following observations have been summarized and are shown in Table 4

The plots of ω Vs ζ for the first and third modes are made available for the four materials investigated (Figures 13 to 16). ζ obtained by two approaches were compared. Damping ratios ζ_1 for all the specimens computed by using logarithmic decrement approach and Tables 2 and 3 by using the $\sqrt{2}$ method with electro dynamic exciter agree well.

(i) ζ values of these materials are in the following order aluminum < mild steel < cast iron < plastics.
(ii) Plastics exhibit a maximum value of ζ_1 4.1% whereas for mild steel it does not exceed 2.9%.
(iii) Values of ζ_3 are less than the corresponding values of ζ_1, for most of the specimens tested (Tables 1 and 2).

INFLUENCE ON DYNAMIC STRESS

As per international standards, for continuous operation, the threshold of vibration disturbance is of the order of 2 mm/s. It is presumed that very severe vibration due to

Table 1. Specifications of shaker and subsystem

Subsystem	Model	Shaker	SEV 150
Electro dynamic shaker	SEV 150	Rated force	1500 Kgf (Peak SINE)
Digital power amplifier	DSA 5K15	Operating frequency range	1 Hz to 3500 Hz
Digital system interface	SCU 011C	Stroke	40 mm
Piezo electric accelerometers	2224 C (2 nos)	Acceleration bare table	80 'g'
'SINE" S/W package	SVC 2205-FW	Maximum velocity	1200 m/s
		Moving Platform diameter	240 mm

Table 2. Details of standard plates investigated (set-1)

Plates	L mm	B mm	T mm	Finite element method		Free vibration method (Experimental)		Forced vibration method (Experimental)			
				ω_1	ω_3	ω_1	ζ_1	ω_1	ω_3	ζ_1	ζ_3
S_1	420	50	4	132	2262	138	0.0145	128	2284	0.0150	0.0030
S_2	220	50	4	522	9117	540	0.0115	548	9192	0.0130	0.0064
S_3	220	50	8	1043	18234	993	0.0104	1087	19578	0.0110	0.0100
A_1	420	50	4	132	2262	138	0.0085	130	2314	0.0096	0.0031
A_2	220	50	4	522	9117	528	0.0200	547	9079	0.0097	0.0090
A_3	220	50	8	1043	18234	1005	0.0180	1030	18414	0.0088	0.0090
C_1	420	50	8	201	3569	214	0.0300	196	3625	0.0208	0.0031
C_2	220	50	8	817	11520	754	0.0360	804	11451	0.0144	0.0080
P_1	375	50	4	50	880	63	0.0320	63	921	0.0300	0.0260
P_2	185	50	4	239	4191	264	0.0360	283	4659	0.0330	0.0450
P_3	220	50	8	371	5623	415	0.0230	425	5329	0.0177	0.0180

Table 3. Details of components tested (Set-2)

Components	Material	Finite Element method		Free vibration method (Experimental)		Forced vibration method (Experimental)			
		ω_1	ω_3	ω_1	ζ_1	ω_1	ω_3	ζ_1	ζ_3
RE 6	Steel	477	1903	483	0.0270	444	1717	0.0290	0.0110
QRV	Steel	1149	4090	852	0.0200	784	3596	0.0220	0.0160
BC type 12	Steel	1746	6729	2418	0.0290	2500	4481	0.0290	0.0080
MTS	Steel	3623	8664	3443	0.0270	3237	7546	0.0270	0.0130
Fixture	Steel	3631	9710	3321	0.0140	3257	9555	0.0120	0.0110
RE6	Al	477	4803	483	0.0140	440	4506	0.0140	0.0100
QRV	Al	5400	11828	6063	0.0100	6195	10687	0.0100	0.0040
MTS	Al	3623	8664	3443	0.0190	3670	7516	0.0150	0.0120
Flange (compressor)	Al	5510	10499	4423	0.0180	4668	10675	0.0130	0.0080
Compressor (over hang)	CI	2022	3700	2150	0.0240	2023	4668	0.0200	0.0087
Compressor	CI	5015	7320	4724	0.0370	4324	6014	0.0330	0.0046
Compressor	CI	5510	10499	4461	0.0290	4706	10710	0.0280	0.0100
QRV	Plastics	823	4012	911	0.0440	924	3522	0.0410	0.0035
MTS	Plastics	3623	8664	3443	0.0380	3675	7053	0.0400	0.0030

transients acting for a very short period of time will not exceed 300 mm/s. The ratio of fall in amplitude for transient vibration is of the order 150 (300/2). Fall in amplitude for every oscillation for viscous damping is $e^{2\pi\zeta}$.

Steel				
	RE6 Valve	BC Type 12	Fixture	MTS
Aluminum				
	RE6 Valve	QRV	Flange-Compressor	MTS
Cast Iron				
	Comp-Over hanged	Flange compressor	Compressor	
Plastics				
	MTS	QRV		

Figure 7. Test specimens.

1st natural frequency is 132 rad/s	2nd natural frequency is 522 rad/s	3rd natural frequency is 1043 rad/s
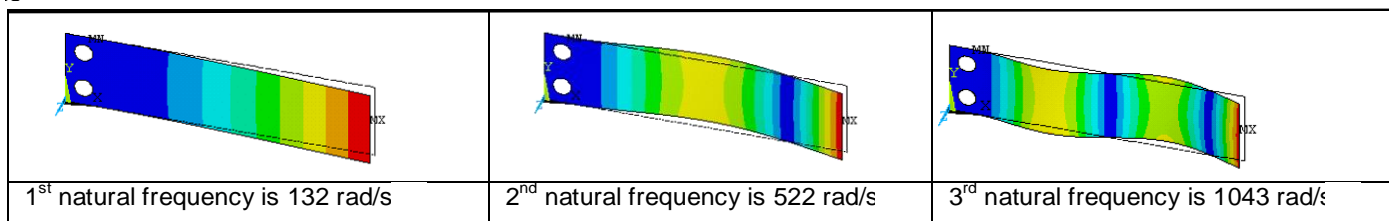
Figure 8. Finite element analysis of S_1 along with its mode shape.

Response of Steel Plate-S_1

Response of steel plate S_2
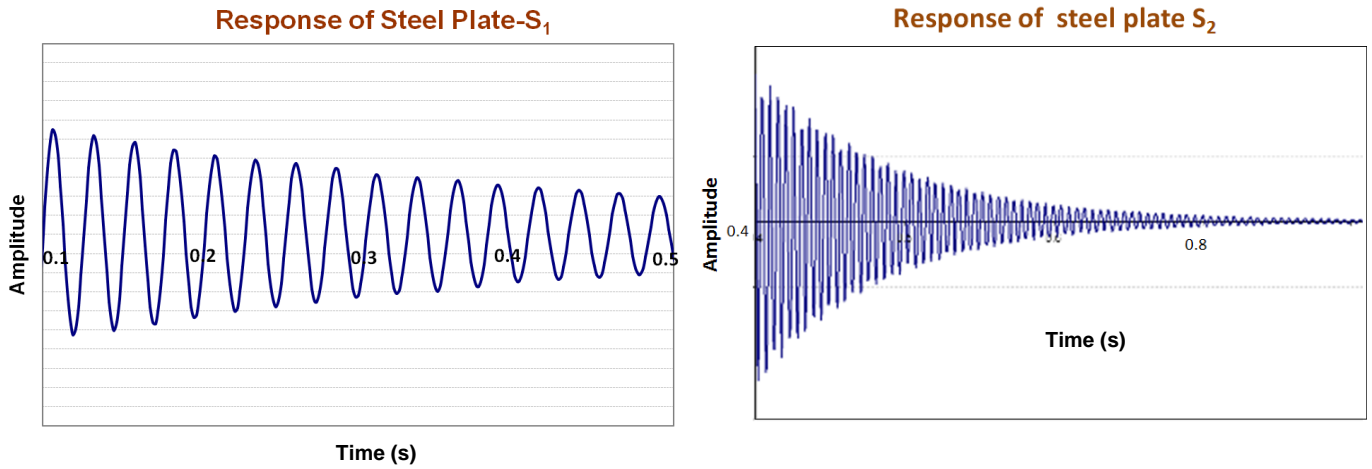
Figure 9. Free vibration response of steel S_1, S_2.

Response of Steel Plate-S_1

Response of steel plate S_2
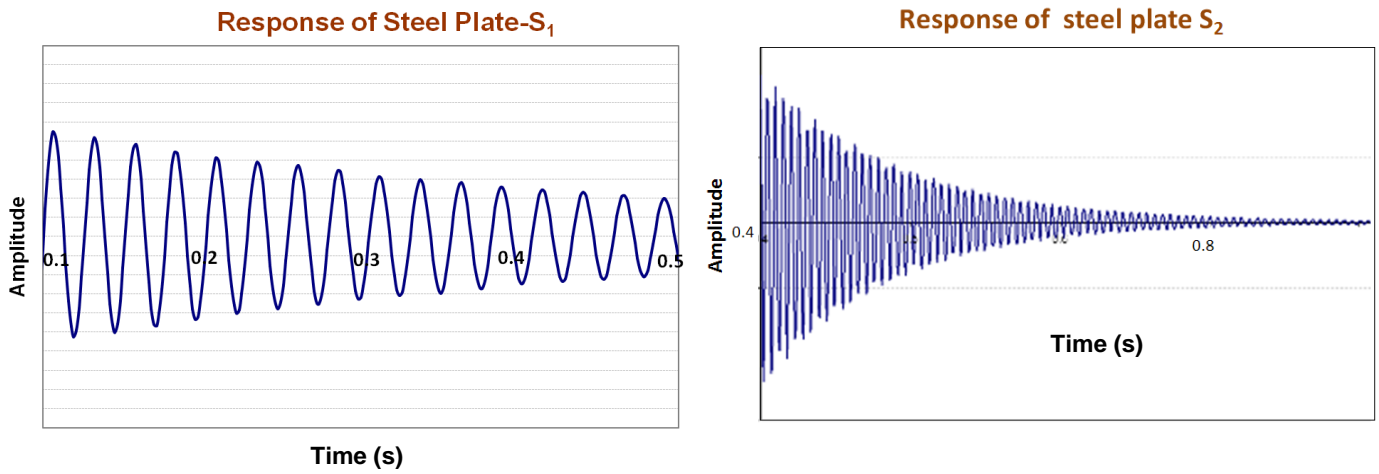
Figure 10. Free vibration response of steel A_1, A_2.

Response of Cast Iron Plate-C_1

Response of Cast Iron Plate-C_2
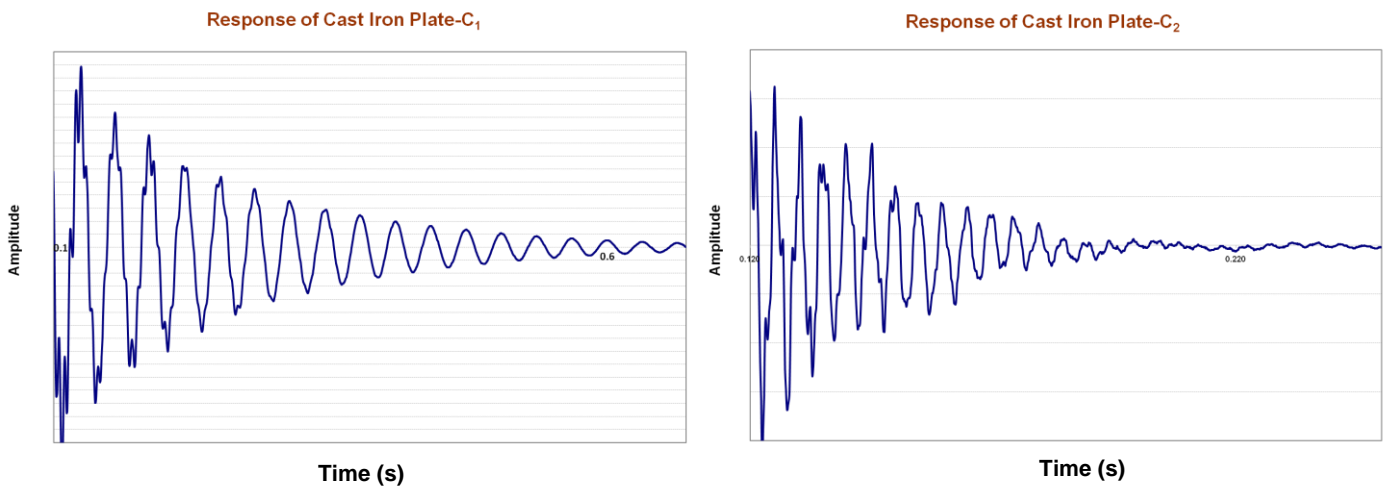
Figure 11. Free vibration response of steel C_1, C_2.

Response of Plastic Plate-P$_1$

Response of Plastic Plate-P$_2$
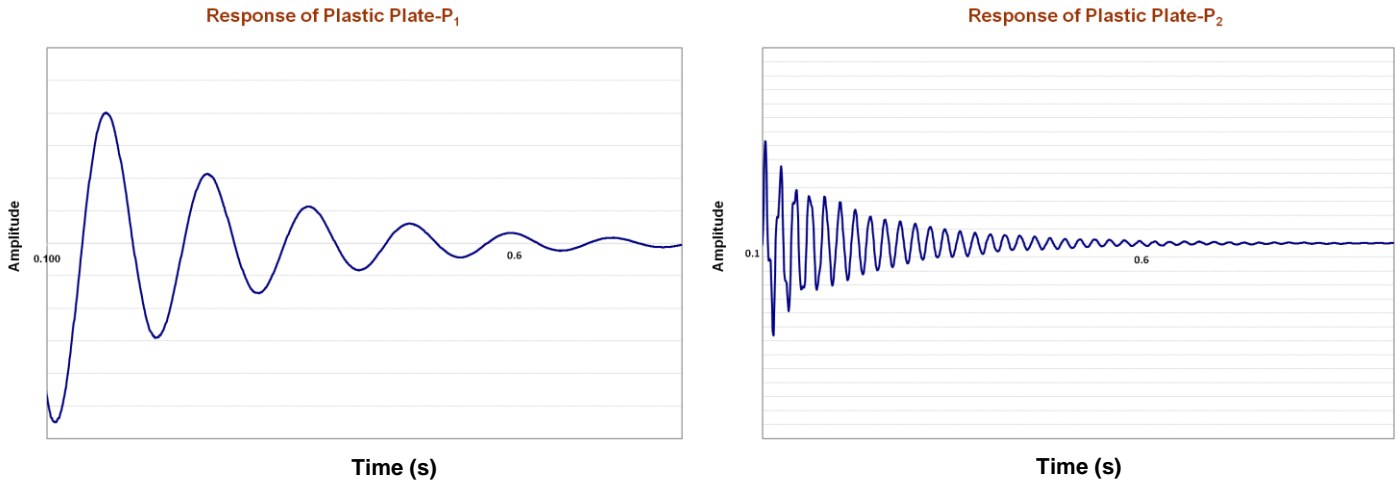
Figure 12. Free vibration response of plastics P$_1$, P$_2$.

Natural frequency Vs ζ of steel
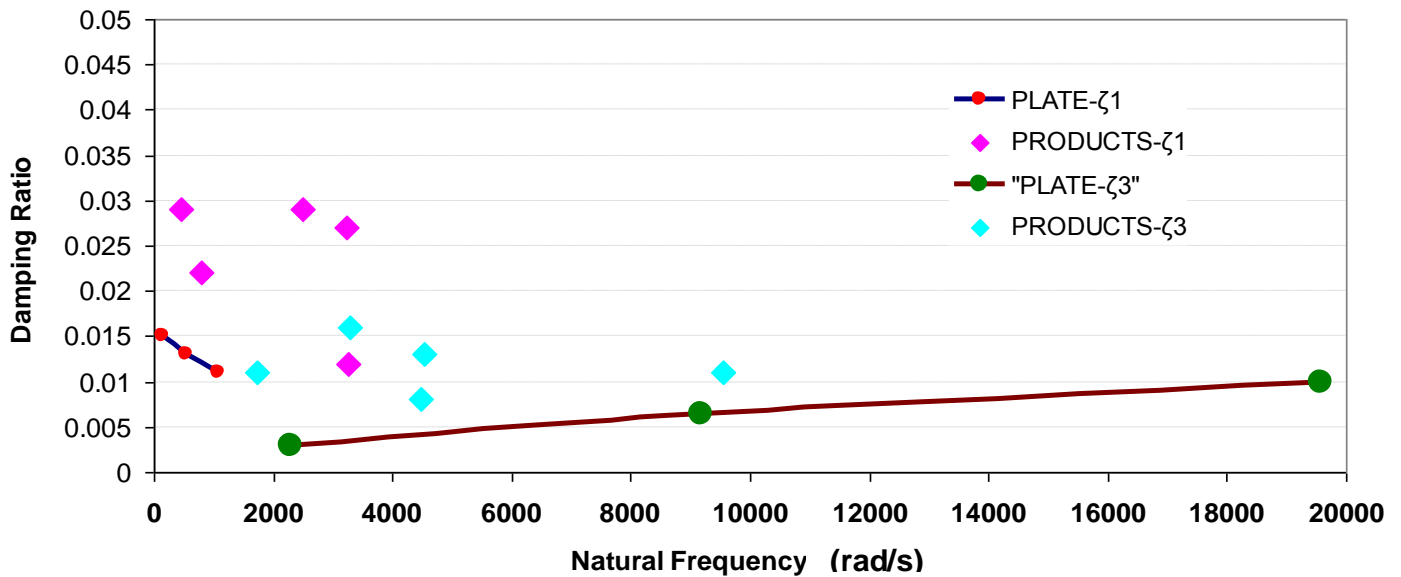
Figure 13. ω Vs ζ for steel.

Hence the ratio of fall in amplitude in one second ($e^{2\pi\zeta}$)$^{\omega/2\pi}$ = $e^{\zeta\omega}$. If $e^{\zeta\omega}$ is to have a maximum value of 150, then $\zeta\omega$ should be of the order of 5 ($e^5 \sim 150$).

To study the effect of damping ratio, as an example delivery valve stopper used in reciprocating air compressor is taken for investigation. This is very light with very high fundamental natural frequency of 18000 Hz.

Figure 17 shows the delivery valve stopper sub assembly with a shoulder held by helical spring. Figure 18 shows the stopper alone for the delivery valve (Ramamurti et al., 2010). This experiences, dynamic stress due to movement of ring valve which discharges air from the compressor. Since this valve stopper is subjected to repetitive shock pulses when the force acts for a very short period of time Figure 19, transient dynamic analysis is carried out using mode superposition technique.

In order to know the influence of damping ratio on the dynamic response of the valve stopper, four cases are
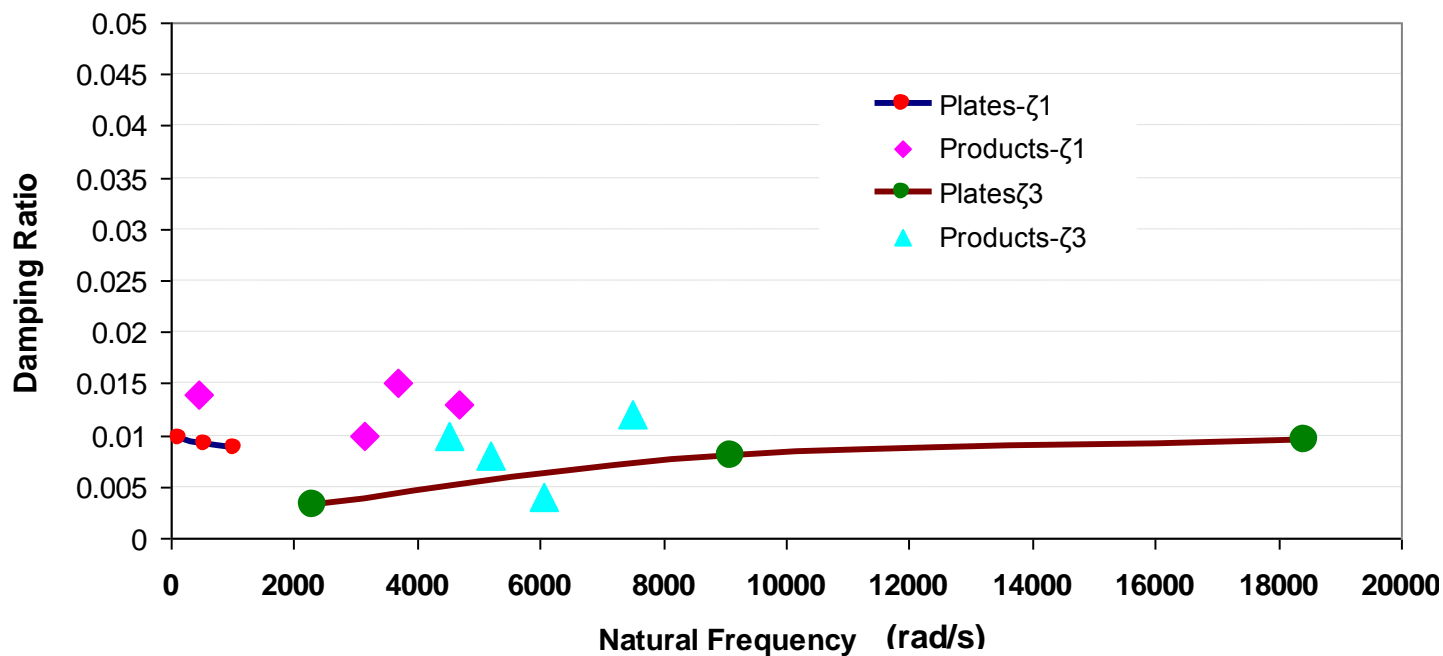
Natural frequency Vs ζ of Aluminium

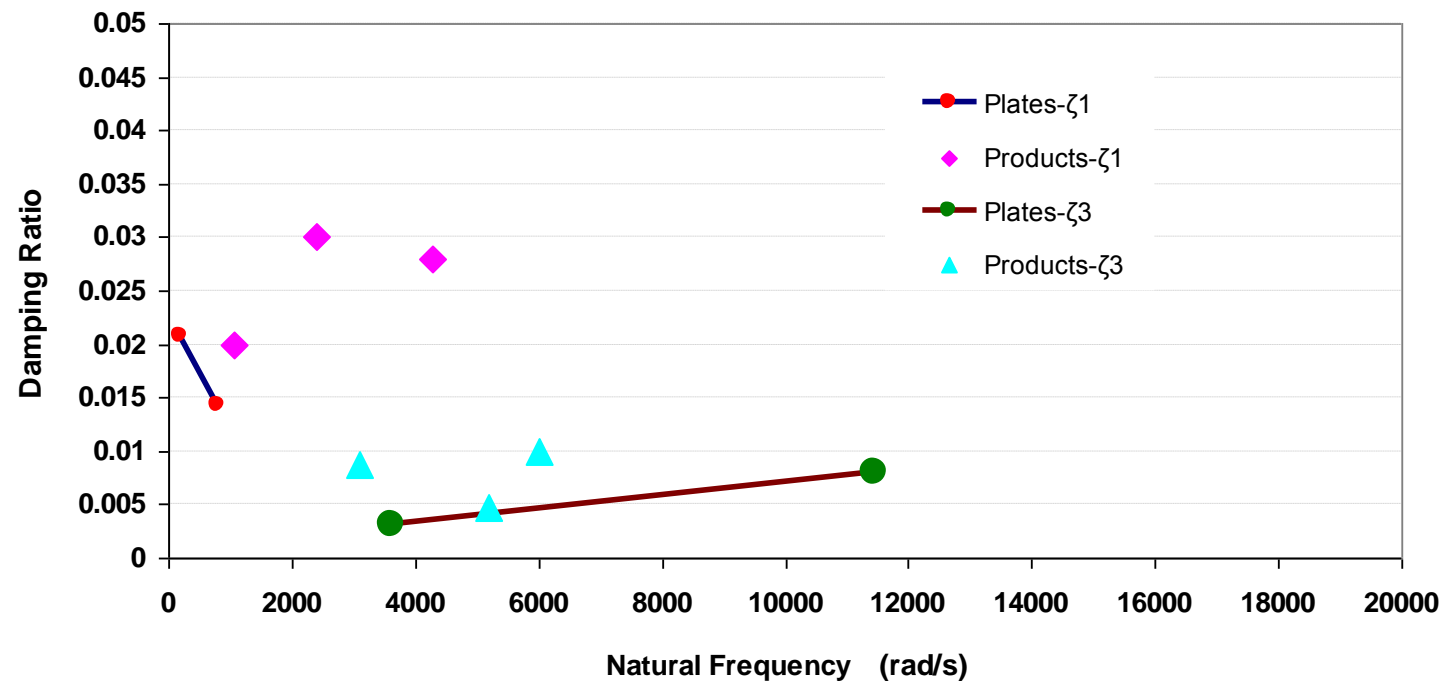**Figure 14.** ω Vs ζ for aluminum.

Natural frequency Vs ζ of cast iron

Figure 15. ω Vs ζ for cast iron.
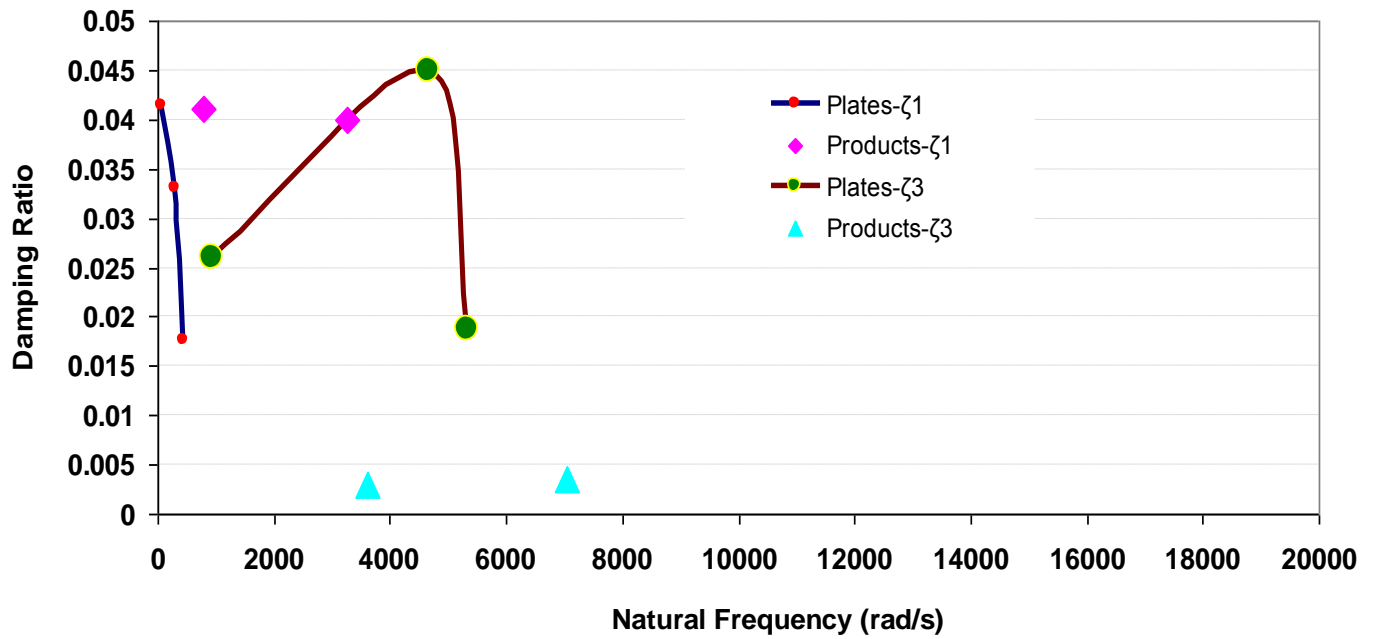
Natural frequency Vs ζ of plastics

Figure 16. ω Vs ζ for plastics.

Table 4. ζ and f of different materials.

Material	f $_1$ (Hz)	ζ$_1$	f $_3$ (Hz)	ζ$_3$
Steel	21 to 518	0.011 to 0.029	273 to 1521	0.003 to 0.016
Aluminum	21 to 985	0.01 to 0.015	368 to 2930	0.003 to 0.012
Cast Iron	31 to 748	0.014 to 0.033	576 to 1822	0.003 to 0.01
Plastics	10 to 584	0.017 to 0.041	146 to 1122	0.003 to 0.045
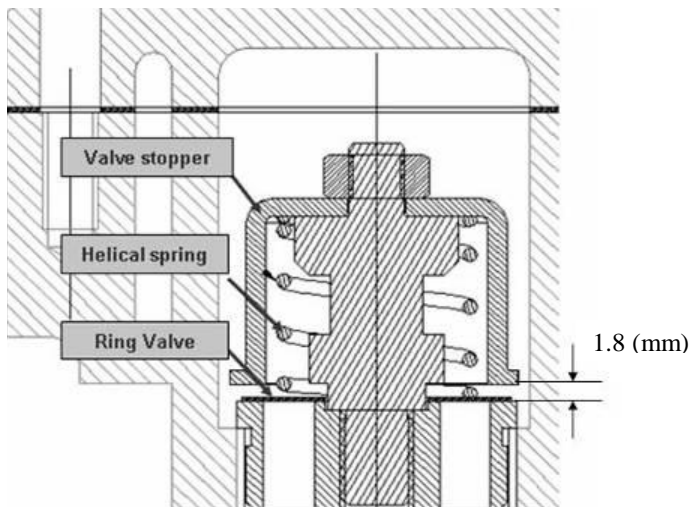
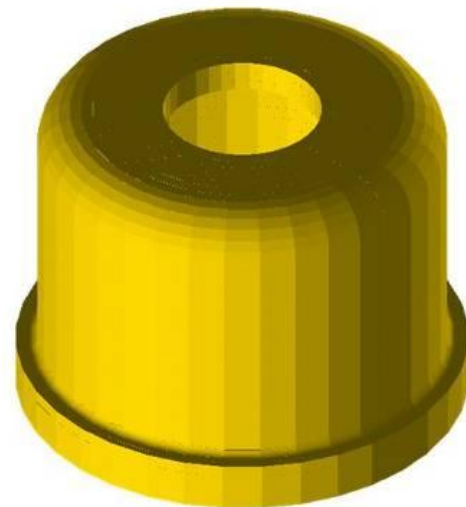**Figure 17.** Delivery valve assembly in the cylinder head.

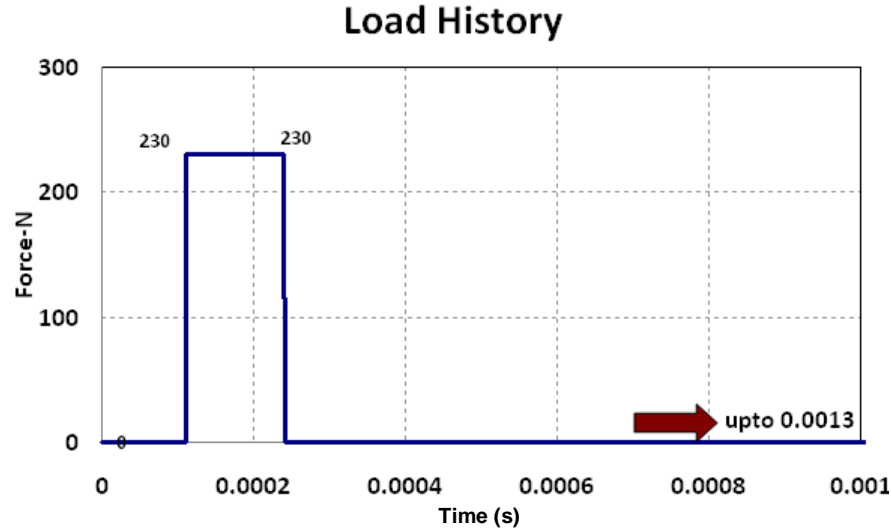

Figure 18. Valve stopper.

Load History

Figure 19. Shock pulse as input to delivery valve.

Table 5. Dynamic response of valve stopper.

Case	ζ_1	ζ_3	Maximum dynamic stress (MPa)	Time of decay(s)
1	0.015	0.01	40.70	1.5 e-3
2	0.015	0.02	40.50	1.4 e-3
3	0.010	0.01	40.83	2.04 e-3
4	0.010	0.02	40.60	2.01 e-3

taken for the analysis By varying ζ_1 and ζ_3 obtained from the experimental findings transient dynamic analysis is carried out. For the first two cases, ζ_1 is assumed to be 0.015, ζ_3 is varied with 0.01 and 0.02. For the second set ζ_1 is assumed to be 0.01 and ζ_3 is varied with value 0.01 and 0.02.The dynamic stress obtained in these cases is shown in Figure 20 (i) to (iv).

When ζ_1 is kept constant, the effect of ζ_3 on dynamic stress is negligibly small and the time of decay (fall in ratio of amplitude around 150) is also small. When ζ_3 is kept constant at 0.01 if ζ_1 is altered from 0.01 to 0.015, the maximum value of dynamic stress varies from 40.83 to 40.7 MPa. When ζ_3 is 0.02 the dynamic stress varies from 40.6 to 40.5. With ζ_1 0.01 the dynamic stresses dies down at $2.01e^{-3}$ s whereas it takes $1.4e^{-3}$ s for ζ_1 of 0.015 (Table 5).

Conclusion

Experimental evaluation has been carried out to find out the damping ratios for free and forced vibration using four different materials. Besides the above, the natural frequencies have been calculated analytically using finite element analysis (FEA). The natural frequency values obtained using the analytical method, free and forced vibration methods have a fairly good agreement with each other for both standard plates investigated and components tested. It was observed that for all the four materials covered in this paper Figures (13 to 16), ζ_1 has never exceeded 4.5% (mostly less than 2%) Table 4. The value of ζ_1 obtained by free vibration method and $\sqrt{2}$ method agree well. Since the natural frequencies cover a wide range of values in industrial parlance, the damping ratios ζ_1 and ζ_3 obtained give a very good database for a designer using steel, aluminium, cast iron and plastics. As an example, in "Influence on Dynamic Stress", axis symmetric element is used for modeling a valve stopper transient vibration study. Figures 13 to 16 provide good data base for computing ζ_1 and ζ_3. After computing the first few natural frequencies of a member which is to be analyzed (eigen pairs) these damping ratio values are to be used in modal super position technique. "Influence on Dynamic Stress" also indicates that the effect of damping ratio ζ_3 on dynamic stress is negligible whereas ζ_1 has some influence. However, it is premature to conclude that the change in damping ratio has marginal influence on the magnitude of dynamic stress. Since Equation 8 is the basic relationship connecting the damping matrix [C] with stiffness and mass matrices [K] and [M], in our opinion, the picture is likely to change if other types of finite elements are used for modeling.
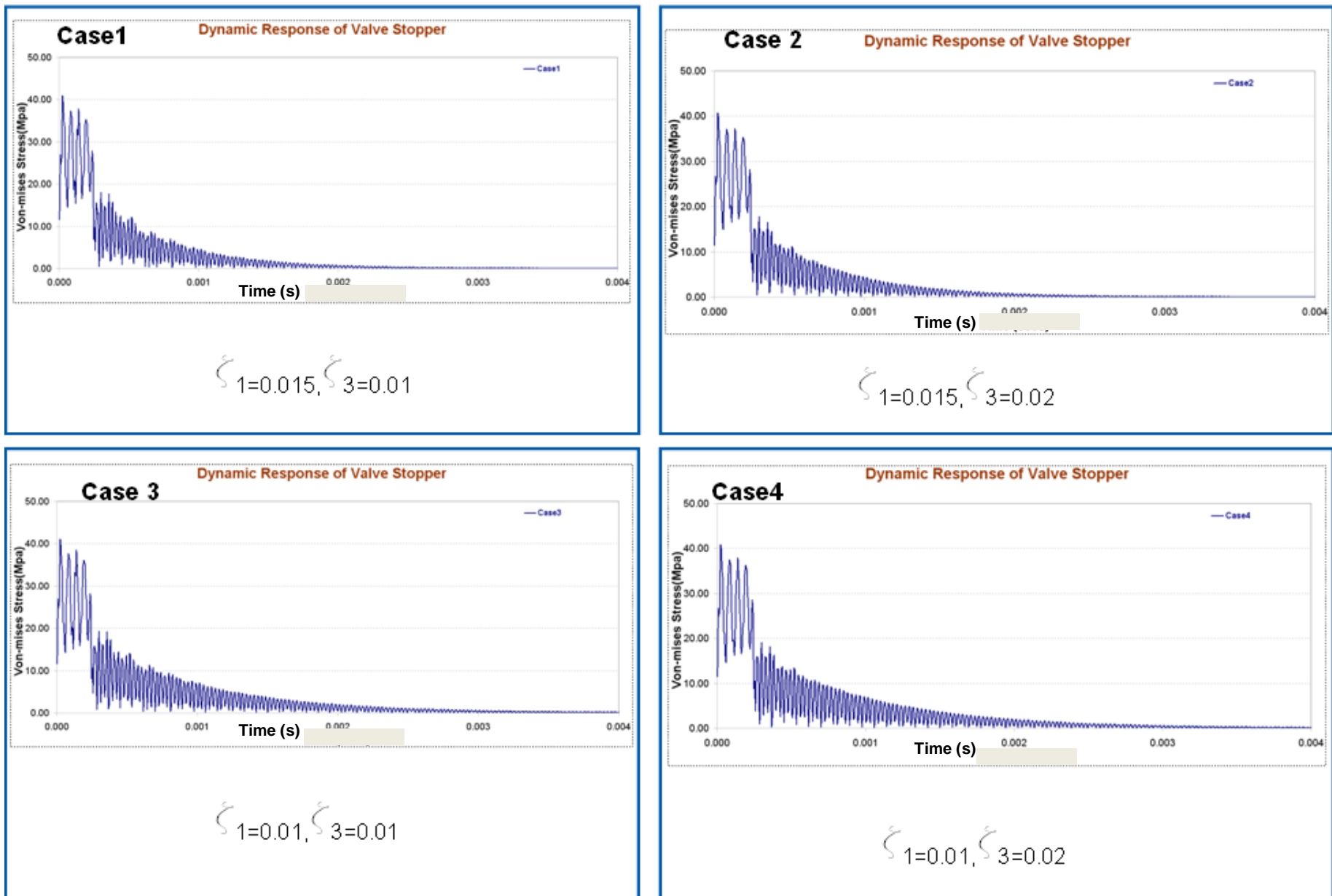
Figure 20. Dynamic stress plot.

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