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A REVIEW ON VIBRATION ANALYSIS OF GEARBOX RESONANCE

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ABSTRACT

Resonance frequency problems are often encountered in mechanical systems. When this occurs, the level of vibration is generally quite high, and this in turn often causes premature failure of machine components. Often, attempts are made to address the rotating elements and aligning coupled components. It is very important to maintain good balance and alignment. However in case of resonant conditions, the primary problem is generally not the magnitude of forcing function. The problem is that the forcing function, which may be of modest magnitude, matches a system resonance of natural frequency. General analysis of vibrating system is based on undamped forced vibrations. This analysis comprises determination of natural frequency of the system. Care should be taken in avoiding matching of this frequency with the frequency of external excitation and thus machines or machine parts are avoided from subjecting them to resonance conditions. Determination of effective mass and spring stiffness is a crucial part in the analysis. In complicated machinery where system may have many components interconnected to each other, it is difficult to determine effective mass directly. This can be carried out by preparing mathematical model to find effective mass of the system. The stiffness of the system may be found out by subjecting it to a known static force and measuring its deflection. When it not feasible to impart static force on the system, data from Impact hammer test may be employed.

We have done a case study of worm and wheel type gear box . The objective is to control vibrations in a worm and worm wheel type gear box by addition of the effective mass. The effective mass is relatively small part of the entire system. So it can be made possible to reduce the level of vibration with relatively small mass and simple in shape at required position on a gear box. Thus the resonance conditions in gear box can be controlled in a better way. Impact hammer test for the gear box, is performed and results obtained will be used to determine equivalent stiffness and the magnitude of required effective mass to shift the resonance from the present gear mesh frequency. It was observed that a change in a resonance condition will be confirmed by performing the Impact hammer test after adding the required effective mass.

Keywords: Frequency, Resonant Frequency, Static Deflection, Dynamic Deflection, Transfer Function, Quality Factor.

I. INTRODUCTION

Vibration is a universal phenomenon. The vibration are executed by mechanical systems which are made up of bodies interconnected by elastic elements and constrained to move relative to one another in a predetermined manner from the first input stage to the last out put stage. For purpose of analysis the body is treated as a finite collection of material particle. A particle in turn is idealized as a mass point whose dimension ignored in considering its motion. The motion of lumped entity is such as a particle in space is specified by its coordinates velocity and acceleration. The subject vibration deals with the behavior of bodies under the influence of oscillatory forces, which are frequently produced by unbalance in rotating machines or by the motion of the body itself. All bodies possessing mass and elasticity are capable of vibration.

With the ushering in of the industrial revolution the engineer emerged on the scene with vibration problems, which began to bedevil the design and construction of machines and structures. Vibration problems facing the industry were grappled by engineers such as Timoshenko and Den Hartog, who drew copiously from the stockpile of knowledge built up by astronomers, physicists and mathematics over the centuries.

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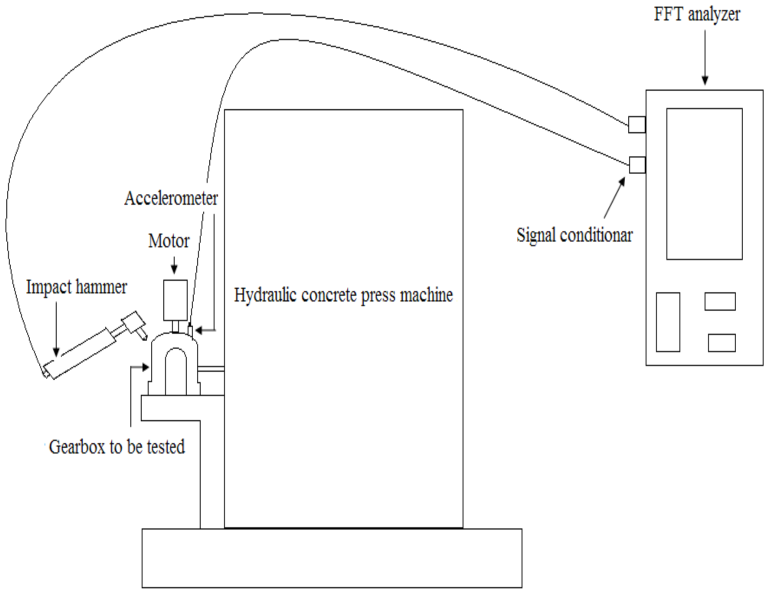
Vibration problems occur whenever there are rotating or moving parts in a machinery. Apart from the machinery itself, the surrounding structure also faces the vibration hazard because of this vibrating machinery. The common examples are locomotives, diesel engines mounted on unsound foundations, whirling of shafts, etc

EXPERIMENTATION:



A gearbox developed a resonance problem at the gear meshing frequency. The input shaft speed is 1440 RPM and there are 38 numbers of teeth on the input pinion.

In the area of the input bearing, there is very high level of vibration at the gear meshing frequencies. The overall vibration level is determined with the help of FFT analyzer as shown in Figure 1 & 2. or axial and radial positions respectively.

Existence of a resonance at an approximately the gear meshing frequency can be confirmed by impact hammer test.

In this case, only a part of machine is involved in the resonance, and the system is much complicated to try to derive any sort of an estimate the effective mass. In such machine, it is possible to derive a value for spring constant by measuring deflection imparted by a known force. From the spring constant, the effective mass can be determined.

In this case, however, it is feasible to impart a static force on the system. In order to overcome this shortcoming, data from impact hammer test were employed. This was done in two steps. First system compliance, m / N, would have to be determined from existing data.

It is possible to perform a impact hammer test with units of compliance directly. By performing the test, data were recorded in units of mobility; cm / sec./ kg. Units of velocity in mm / sec. Can be converted to units of displacement by following equation:

From fig.3,

$$D = v / \pi f$$

Where,

v = velocity, m/ sec/kg.

f = frequency, Hz.

D = displacement, m, peak to peak.

$D = 0.3183\, v / f$

$= (0.3183) \times (57 \times 10^{-3}) / 120.2$

$= 0.9 \times 10^{-3}$ m

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This equation is generally used to convert vibration from units of velocity to displacement. However, there is no reason that it cannot be used to convert a transfer function from units of velocity per units of applied force. Thus the dynamic compliance will be 0.9 x 10^{-3} m /kg. By inverting the value of dynamic stiffness K_d can be determined.

K_d = 1 / D

= 1 / 0.9 x 10^{-3} m /kg

= 1.05 x 10^{-3} kg/m

The next step is to determine the amplification factor using the following equation

$$\frac{XK}{F_0} = \frac{1}{\sqrt{\left[1-\left(\omega / \omega_n\right)^2\right]^2 + \left[2\xi\, \omega / \omega_n\right]^2}}$$

Where,

X =Dynamic deflection.

K = Spring constant.

F_o = Pick dynamic factor.

ω = Forcing frequency

ω_n =Natural frequency.

ξ = Critical damping ratio.

XK/ Fo = Amplification factor.

Every thing on the right side of the equation is known except the critical damping ratio. The critical damping ratio is the ratio of actual system damping to that of critical damping. Critical damping is minimal amount of damping required to prevent vibration when a system is displaced and released.

The critical damping ratio can be determined from the transfer function. Two methods will be employed to determine the critical damping ratio. The first is the half power method. In determining a damping of the system, the parameter 'Q' is often employed. This parameter is measure of the sharpness of the resonance and is defined as

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

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

In using the half power method, the total transfer function is employed. The top trace in figure is the total transfer function. The total transfer function will generate a roughly symmetric peak the center of which will be the resonant frequency. The steepness of the peak is a measure of amount of damping in the system. In the half power method, the frequency of resonant value is compared with the frequency at which the half power values on either side of resonance occur. The half power value is 0.707 multiplied by peak value.

The Q value is,

$$Q = \omega_n / (\omega_1 - \omega_2)$$

Where,

ω_n = Resonant frequency.

ω_1 = Frequency at which the upper half power amplitude occurs.

Approximately 157.07 rad/sec.

ω_2 = Frequency at which the lower half power amplitude occurs,

Approximately 62 rad/sec.

Thus,

$$Q = \frac{120.2}{157.07 - 62} = 1.27$$

And

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

$$\xi = \frac{1}{2 \times 1.27}$$

$$= 0.39$$

Now that value has been obtained for one critical damping ratio ξ,

The dynamic amplitude factor can be determined,

$$\frac{XK}{F_0} = \frac{1}{\sqrt{[1-1^2]^2 + [2 \times 0.39 \times 1]^2}} = 1.27$$

From the dynamic amplification factor and dynamic spring constant, the static spring constant can be determined as,

$$\frac{XK}{F_0} = \frac{K}{F_0/X} = \frac{K}{K_d}$$

There fore,

K = 1.27 x Kd

= 1.27 x 1.05 x 10^3

= 1338.8 kg / m

By rearranging terms in the resonant frequency equation , the effective mass can be determined ,

m = k / $(\omega_n)^2$

Where,

C = Resonant frequency

= 120.2 rad/sec

Therefore,

m = (1338.8 kg/m) / (120.2 Rad./sec.)^2

= 0.092 kg – s^2 /m

This mass can be converted to a weight by multiplying by gravitational acceleration, 9.81m/s^2

Weight = 0.092 kg – s^2 /m x 9.81m/s^2

= 9.02 N

This is relatively small part of the total system ca be involved in the resonance. In this case, gearbox weight is approximately 20 kg, but the effective mass of resonance system is 0.092 kg.

Now all of the parameters of the resonant system are known. If we refer back to fig.1, the frequency range from approximately 219.9 rad/sec to 245 rad/sec has relatively small amplification factor. If the system can be returned so that this curve is shifted to the left such that the gear meshing frequency falls in to the trough between 219.9 rad/sec and 245 rad/sec, the level of vibration should go down significantly. Care must be taken not to lower the resonance to much.

We will try to move the anti resonance, currently at 245 rad/sec, down to the gear meshing frequency, 219.9 rad/sec. In order to do this, the resonance will be lowered approximately 245 – 219.9 = 25.14 rad/sec. This can be done by increasing the effective mass of the system.

The new resonant frequency for which the system will be tuned will be,

120.2 – 25.14 = 95rad/sec.

Going back to resonant frequency equation,

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$$m1 = k / (\omega_n)^2$$
$$= (1338.8 \text{ kg /m}) / (95 \text{ rad/Sec.})^2$$
$$= 0.148 \text{ kg} - S^2 /m$$

Or, Weight = 1.45 N

This is the weight of the new effective mass of the system. In order to implement a change in the effective mass, a concentrated inertial mass, of the weight of which is equal to the new, target effective mass minus the weight of the current effective mass should be bolted to the input shaft flange

Concentrated weight = 0.188 – 0.092

= 0.056 kg

= 0.549 N

The transfer function indicated that the resonance had indeed been shifted to approximately the target frequency and out of the frequency range where gears would excite resonant frequency as shown in fig.4.

A vibration survey taken immediately after the inertial mass had been installed revealed that the level of vibration had been reduced. which is shown in figure 5 and 6.

In this case, it is possible to realize a very significant reduction in the level of vibration with a relative simple fix. The cost of fabricating and installing the inertial mass was minuscule compared to the cost of the gearbox.

Fig.1
Vibration spectrum of gearbox before mass modification
(Radial position)
Overall vibration level = 2.2

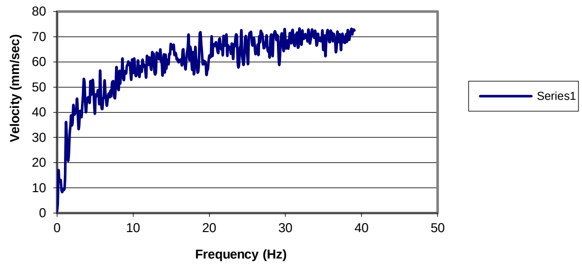


Fig.2
Vibration spectrum of gear box before mass modification
(Axial position)
Overall vibration level =2.9

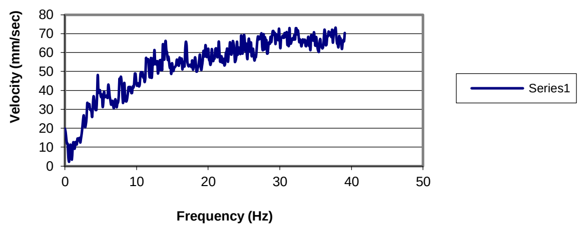


Fig.3
Transfer function before mass modification by Impact Hammer test

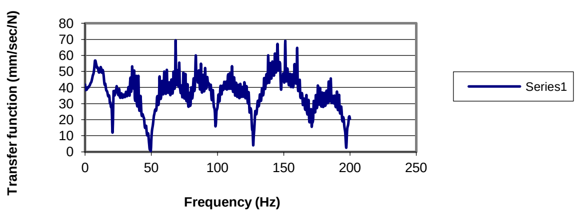


Fig.4
Transfer function after mass modification by Impact Hammer Test

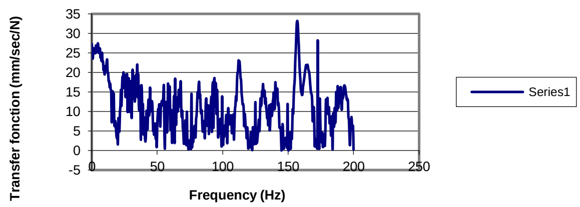


Fig.5
Vibration spectrum after mass modification by Impact hammer test (Radial position)
Overall vibration level = -6.4

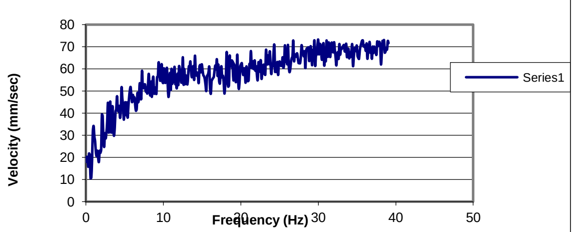
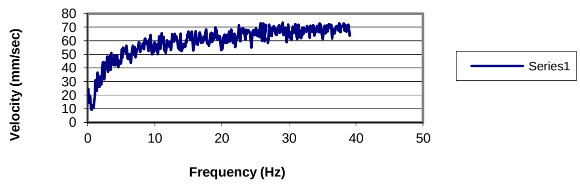


Fig.6
Vibration spectrum after mass modification by impact hammer test (Axial position)
Overall vibration level = -4.5



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II. RESULT

A gearbox developed a resonance problem at the gear meshing frequency. The input shaft speed is 1440 RPM and there are 38 numbers of teeth on the input pinion.

In the area of the input bearing, there is very high level of vibration at the gear meshing frequencies. The overall vibration level is determined with the help of FFT analyzer for axial and radial positions respectively. Existence of a resonance at an approximately the gear meshing frequency can be confirmed by impact hammer test.

Before mass modification, at resonance condition the overall vibration level for axial position is 2.9 and for axial position is 2.2.this vibration level is carried out by FFT analyzer and accelerometer, which is placed on casing of input shaft of gearbox for axial and radial positions. The natural frequency as 19.921 Hz and max. velocity function (Transfer function) was 57 x 10^{-3} m/sec/N. And then with the help of derived formulae the concentrated inertial mass is calculated this mass was 0.056 Kg. Then this calculated mass is attached on gearbox casing at input shaft. Then again impact hammer test is carried out, It has been observed that the resonant or natural frequency is shifted to 18.75 Hz. By installing gearbox on machine the overall vibration level is determined, it is reduced to –4.5 for axial position and – 6.4 to radial position. The overall vibration level is reduced by addition of small inertial mass equal to 0.056 Kg compared to total mass of the gearbox equal to 18.36 Kg.

III. CONCLUSION

Modal analysis is the process of determining modal parameters of linear time invariant system. Experimental modal analysis is verification or correction of results obtained through analytical analysis. and spectrum analysis is also important tool for vibration measurement by transfer function and it refers to time domain representation. It is evident that vibration measurement using modal technique is based on Fast Fourier Transform analyzer (FFT). The FFT signal using spectrum analyzer, which converts directly time domain to frequency domain. In practice piezoelectric transducer can measure very small amplitudes of vibration. The impact hammer test is carried out to determine natural or resonant frequency of gearbox.

Because the resonant frequency is the function of square root of both the effective mass and stiffness, it is often not practical because a significant change in resonance by changing either of these parameter when the entire structure is a part of vibrating system.

However, in case of gearbox, the effective mass of the vibrating system is relatively small part of the system, and adding the inertial mass is very effective in controlling the vibration level.

These techniques were implemented on worn and worn wheel type gearbox. The vibration spectrum and auto spectrum were plotted for both axial and radial positions. By calculating concentrated inertial mass, it is attached at input shaft casing of gearbox because it is source of vibration. Again by taking impact hammer test it is observed that the natural or resonant frequency of gearbox is shifted and vibration level at axial and radial positions are reduced.

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