International Journal of Engineering & Technology, 7 (4.35) (2018) 107-111

International Journal of Engineering & Technology

Website: www.sciencepubco.com/index.php/IJET

Research paper



Loss Factor of Elastomeric Dampers for Rotating Machinery Application

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Abstract

Elastomeric dampers have potential application in rotating machinery vibration control. They are however not widely used due to lack of reliable data on their loss factor. Most available data on these dampers are obtained from testing undertaken during stationary condition of the shaft. When the shaft rotates, the dampers are subjected to rotating load that may affect their loss factor. The effect of shaft rotation on the loss factor is experimentally examined in this work. Impact test was used to determine the frequency response function (FRF) of the dampers. For the dampers subjected to rotating load, the loss factor values derived from the FRF was found to be in good agreement with those determined from the half-power bandwidth method. The results further showed that the loss factor at resonant frequency determined from testing of the dampers under stationary shaft condition underestimates the values of the loss factor when the shaft is rotating. The effect of shaft rotation on the values of the damper's loss factor was more noticeable for the response in the *X*-direction as opposed to the *Y*-direction, indicating that pre-strain plays a more dominant role in influencing the loss factor of the dampers compared to the dynamic amplitude.

Keywords: *Elastomeric Damper; Impact Test; Loss Factor; Rotating Machinery*

1. Introduction

Elastomeric material have in the past being used as vibration isolators, absorbers and dampers in the control of vibration in machines and structures. This material has excellent dynamic characteristics for vibration control purposes over a wide range of frequency and temperature [1]. For rotating machinery application, however, squeeze-film dampers are favored over elastomeric dampers for the reduction of vibration response, particularly when the rotor is traversing one of its critical speeds. The squeeze-film dampers are widely utilized in rotating machinery despite having complex design and requiring auxiliary equipment such as oil-feed pump for their operation. Elastomeric dampers, although much simpler in design and maintenance, have not found much application in rotating machinery due to the lack of reliable data on their dynamic characteristics. The primary motivation to cut maintenance cost influences majority of the industries to utilize components that are less complex and those that have lower maintainability. Elastomeric dampers, if designed and installed properly, will require only minimal maintenance while still providing the much needed damping to reduce vibration response of rotating machinery.

Elastomeric dampers have dynamic stiffness and loss factor that are dependent on pre-strain, frequency, temperature and dynamic amplitude [2]. Friswell et al. [3] developed a model for elastomeric supports using internal variables that reproduced the frequency dependence of their modulus. Through numerical analysis, the authors demonstrated the correlation between the effects of frequency and temperature dependent modulus of the elastomeric supports on the dynamics of rotating machines. Zorzi et al. [4] evaluated the effectiveness of elastomeric and squeeze-film dampers on a supercritical power transmission shaft. Both these dampers were designed and incorporated into a customized test rig. The authors demonstrated that elastomeric dampers were a reliable source of external damping as they allowed the supercritical shaft of the test rig to operate at greater speed.

Since there are no available analytical expressions that allows the calculation of the loss factor of the elastomeric dampers, it can only be determined from measurements. Works in the past have utilized measurement techniques to determine the dynamics stiffness and loss factor of elastomers. Of particular interests are the work of Lin et al. [5] who used the driving point mobility measurements to determine the dynamics characteristics of elastomeric mount. The authors used complex frequency response function (FRF) from impact test to determine the mount's stiffness and damping coefficient. Evaluation of the dynamic characteristics at frequency bands below, at and above resonance was carried out.

On a much recent work done by Ooi and Ripin [6], the dynamic stiffness and loss factor of engine rubber mounts were determined using impact test. Both dynamic transfer stiffness and dynamic driving point stiffness were evaluated. By using impact test, a comparison of results between the dynamic transfer stiffness and the dynamic driving point stiffness was performed. The test results showed that the loss factor obtained from the transfer stiffness measurement exhibited a non-linear dependency on frequency while the loss factor obtained from the dynamic driving point stiffness measurement showed a linear dependency on frequency. The authors concluded that the dynamic driving point stiffness can only correctly represent the dynamic transfer stiffness for low range of frequencies.

Lin et al. [5] and Ooi and Ripin [6] have done works focusing on the measurements of loss factor of elastomers subjected to unilateral load. This paper serves to expand the scope by evaluating the loss factor of the elastomeric dampers subjected to rotating load. This paper also includes the usage of rotor-mounted elastomers instead of the foot-mounted elastomers, which were more com-

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monly studied in previous works. An experimental test rig incorporating elastomeric dampers was designed, fabricated and commissioned. The FRF measurements were performed using impact test. Loss factor was determined from the experimental FRF data and the influence of rotating speed on the loss factor was examined.

2. Theoretical Background

The mathematical model of a rigid rotor supported by elastomeric dampers is represented by a single-degree-of-freedom (SDOF) system in this study. The system's equation of motion in the *X*- and *Y*-directions are

$$m\ddot{x} + c(\omega)\dot{x} + k(\omega)x = f_x(t) \quad (1)$$

$$m\ddot{y} + c(\omega)\dot{y} + k(\omega)y = f_y(t) \quad (2)$$

where *m* is the combined mass of half the rotor assembly mass and the bearing mounting mass, $c(\omega)$ and $k(\omega)$ are respectively the frequency dependent damping coefficient and dynamic stiffness of the elastomeric damper, and $f(t)$ is the external force acting on the rotor. Transformation of Eq. 1 and Eq. 2 in the frequency domain gives

$$F_x(\omega) = H(\omega)\, X(\omega) \quad (3)$$

$$F_y(\omega) = H(\omega)\, Y(\omega) \quad (4)$$

where F, X and Y are the Fourier transforms of *f*, *x* and *y* respectively. $F_x(\omega)$ is the input force in the *X*-direction and $F_y(\omega)$ is the input force in the *Y*-direction. $X(\omega)$ and $Y(\omega)$ are respectively the displacement response functions in the *X*- and *Y*-directions. $H(\omega)$ represents the complex receptance function. The real part, (Re (|H|)), and imaginary part, (Im (|H|)), can be obtained directly from the measured frequency response functions. They are shown in Eq. (5) and Eq. (6) for the *X*-direction.

$$\mathrm{Re}(|H|) = |X / F| \cos \text{Ø} \quad (5)$$

$$\mathrm{Im}(|H|) = |X / F| \sin \text{Ø} \quad (6)$$

Once the Re(|H|) and Im(|H|) parts are determined, the dynamic stiffness and damping coefficient can be estimated. The relationship between Re(|H|) and dynamic stiffness coefficient is shown in Eq. (7), whilst Eq. (8) shows the relationship between Im(|H|) and the damping coefficient.

$$\mathrm{Re}(|H|) = k(\omega) - m\omega^2 \quad (7)$$

$$\mathrm{Im}(|H|) = c(\omega)\omega \quad (8)$$

The loss factor, $\eta(\omega)$, is determined from Eq. (9).

$$\eta(\omega) = c(\omega)\, \omega / k(\omega) \quad (9)$$

3. Experimental setup

The test rig used in the experimental work is shown in Fig. 1. Four elastomeric dampers were installed on each bearing mounting and pedestal, as shown in Fig. 2. The elastomeric dampers were made of natural rubber having 50 durometer hardness.

Impact tests were undertaken using the PCB Piezotronics 086C02 impact hammer and the Dytran 3056B2. The impact hammer and the accelerometer were connected to the Ono Sokki portable FFT analyser model CF-7200. Two accelerometers were mounted on the bearing mounting, one each in the vertical (*Y*) and horizontal (*X*) directions. For the testing in the vertical direction (*Y*), the accelerometer is positioned at the bottom of the bearing mounting. Force, F_y was applied via impact hammer on the top of the bearing mounting. This impact was repeated ten times, and the responses were averaged to reduce noise in the data. Similar tests were undertaken in the *X*-direction.

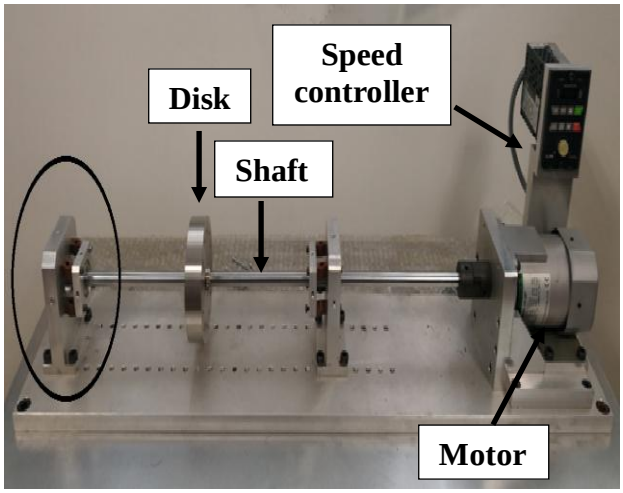


Fig 1: Rotor test rig.

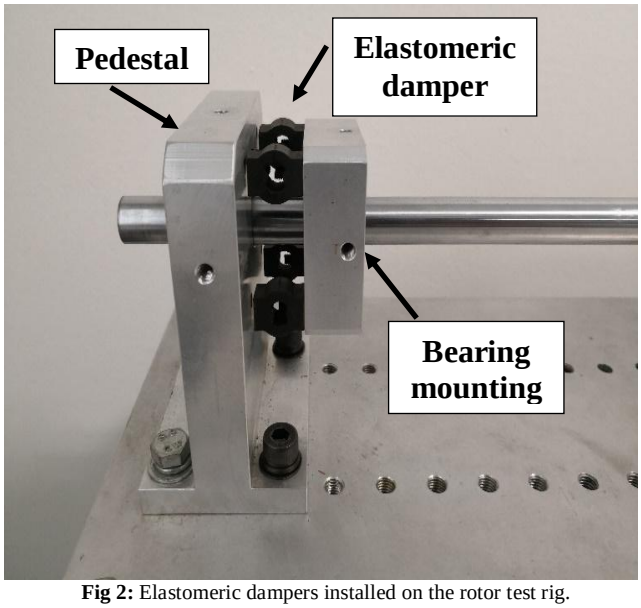


Fig 2: Elastomeric dampers installed on the rotor test rig.

4. Result and Discussion

Impact test was undertaken for both the cases of non-rotating and rotating shaft. The FRF magnitude and phase responses were measured, and the frequency-dependent loss factor of the elastomeric damper was derived using Eq. (9). The frequency range of interest in this work was from 30 Hz to 200 Hz. The value of the mass, *m*, which is the combined mass of half the rotor assembly and the bearing mounting, was measured to be 0.746 kg. The FRF magnitude and phase responses for the test undertaken in the vertical (*Y*) direction is respectively shown in Fig. 3 and Fig. 4. The fundamental natural frequency of the system was determined to be 80 Hz, Fig. 3.

The loss factor values of the elastomeric damper in both the *X*- and *Y*-directions, for the case of non-rotating shaft, are respectively shown in Fig. 5 and Fig. 6. The loss factor values of the damper showed large variation as a function of frequency in both the *X*- and *Y*-directions. Nadeau and Chapoux [7] in their work have also explained that there were some uncertainties in the loss factor data which makes the loss factor curve less smooth. Sensitivity of the loss factor to the boundary conditions and to the rigid body modes

are the factors contributing to these uncertainties. Similar observation was reported by Lin et al. [5].

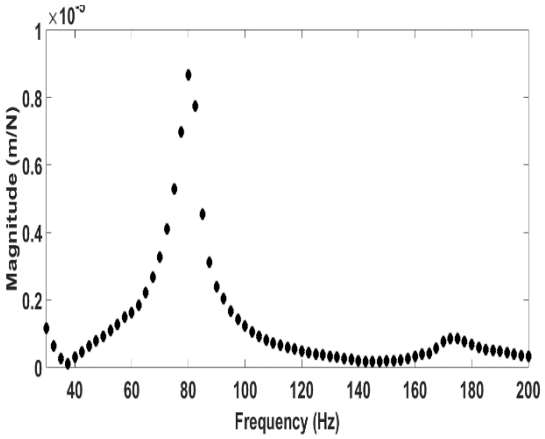


Fig. 3: Magnitude response vs frequency for vertical (*Y*) direction.

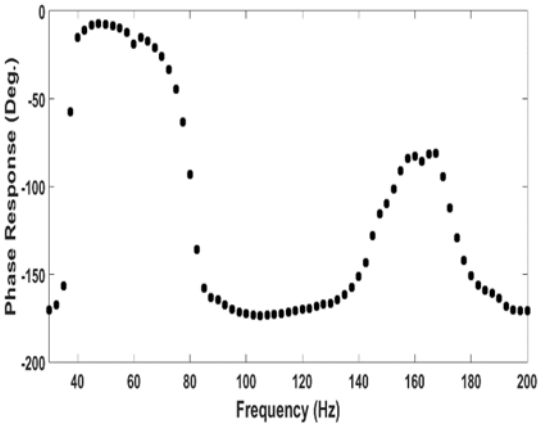


Fig. 4: Phase response vs frequency for vertical (*Y*) direction.

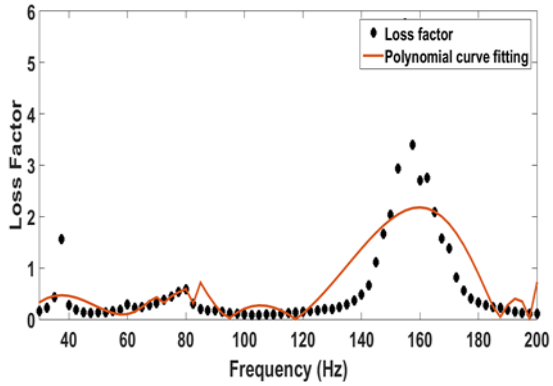


Fig. 5: Frequency dependent loss factor of the elastomeric damper in the vertical (*Y*) direction.

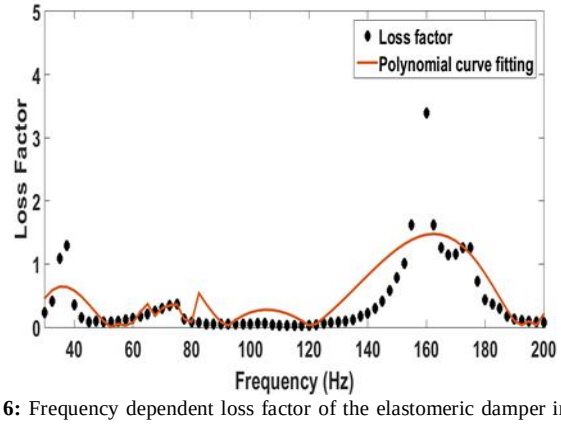


Fig. 6: Frequency dependent loss factor of the elastomeric damper in the horizontal (*X*) direction.

Piecewise polynomial curve fitting of the frequency dependent loss factor data was undertaken for both the *Y*- and *X*- directions, and these curves are shown in Fig. 5 and Fig. 6, respectively. The loss factor data were subdivided into three regions: (a) below the resonant frequency band, (b) within the resonant frequency band and (c) above the resonant frequency band. This was necessary due to the considerable variation of the loss factor values over the frequency range between 30 Hz and 200 Hz. Within the resonant frequency band in the *Y*-direction, the range was from 75 Hz to 82.5 Hz. For the loss factor values across this frequency range, a cubic curve function is fitted to the data using least-squares method, as given in Eq. (10):

$$\eta_1(f) = a_1 f^3 + b_1 f^2 + c_1 f + d_1 \quad (10)$$

The values of constants are $a_1 = 3.5 \times 10^{-10}$, $b_1 = 0.189569$, $c_1 = -23.409$ and $d_1 = 7.53 \times 10^6$. The coefficient of determination, R^2, is 0.98, indicating that the loss factor values obtained from the polynomial function represented well the experimental determined loss factor. The loss factor at resonant frequency is further evaluated by using the half-power bandwidth method, Eq. (11).

$$\eta = \frac{\Delta\omega}{\omega_n} = 0.50 \quad (11)$$

The loss factor obtained from Eq. (11) is 0.50 and this agrees fairly well with the loss factor of 0.59 derived from the polynomial function in Eq. (10) at 80 Hz; the percentage error being 15.25%. The difference in these values exist due to the fact that the half-power bandwidth method is highly dependent on the values of $\Delta\omega$ and ω_n used, and these values vary with different frequency resolution [5]. For the case of the loss factor in the horizontal (*X*) direction, the polynomial curve which was fitted to the measured data is shown in Fig. 6. Within the resonant frequency band, the range is from 70 Hz to 80 Hz. A fourth order polynomial was fitted to the data using least-squares method, as given in Eq. (12).

$$\eta_2(f) = a_2 f^4 + b_2 f^3 + c_2 f^2 + d_2 f + e_2 \quad (12)$$

The values of constants were determined to be $a_2 = 2.3 \times 10^{-45}$, $b_2 = -1.3 \times 10^{-9}$, $c_2 = 0.49607$, $d_2 = -56.1829$ and $e_2 = 1.65 \times 10^3$. The coefficient of determination, R^2, is 0.98. The half-power bandwidth method is used again here at the resonant frequency to evaluate the value of loss factor. The loss factor value for the horizontal (*X*) direction at resonant frequency determined through the half-power bandwidth method gives $\eta = 0.37$. On the other hand, the loss factor value obtained from Eq. (12) is $\eta = 0.36$, giving a percentage error of 2.78%. The FRF was also measured for the case of shaft rotation at speeds of 500, 1000, 1500, 2000, 2500 and 3000 rpm. Loss factor values were determined using Eq. (9). Polynomial curve was fitted to the loss factor data. The influence of rotor speeds on the loss factor of the damper in the *Y*-direction are shown in Figs. 7-9 for different frequency regions, i.e., below the resonant frequency band, within the resonant frequency band and above the resonant frequency band. Such a significant variation of loss factor with frequency was also shown in the recent work of Vangipuram et al. [8]. They used a simple time-domain based identification technique to determine the frequency-dependent dynamic stiffness and loss factor of elastomeric mounts. The half-power bandwidth method was again used within the resonant frequency band region to evaluate the loss factor values at resonant frequency. Table 1 compares the loss factor obtained from the curve fitted function within the resonant frequency band and also the loss factor value determined from the half-power bandwidth method for all rotational speed considered in this work. With the exception of the case for non-rotating shaft, the comparison in Table 1 showed that the loss factor at the resonant frequen-

cy determined from both methods gave considerably similar results, with percentage difference less than 10%

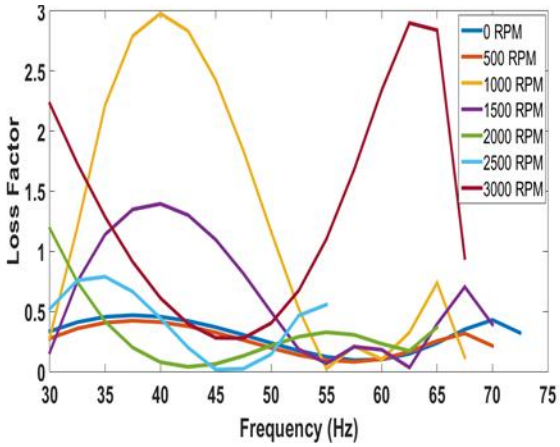


Fig. 7: Curve fitted loss factor of the damper below the resonant frequency band for various rotor speed: *Y*-direction.

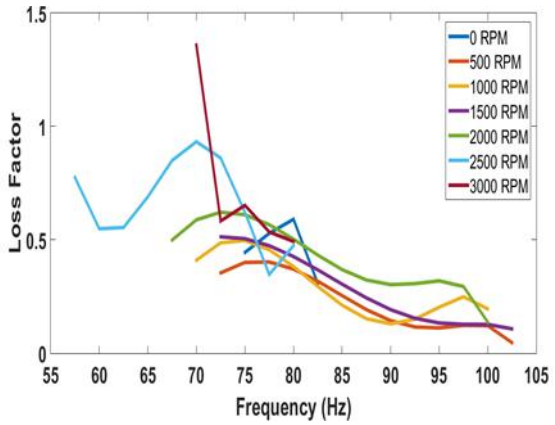


Fig. 8: Curve fitted loss factor of the damper within the resonant frequency band for various rotor speed: *Y*-direction.

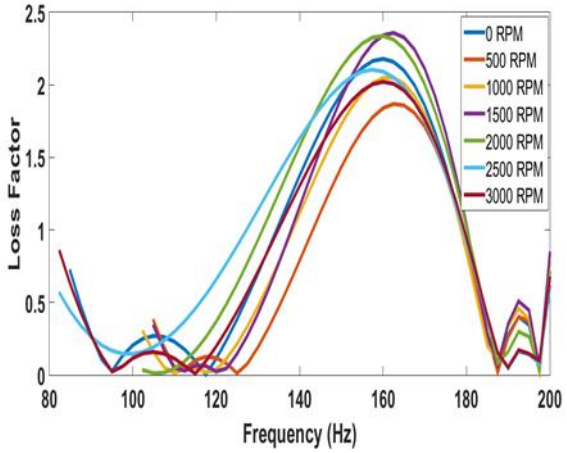


Fig. 9: Curve fitted loss factor of the damper above the resonant frequency band for various rotor speed: *Y*-direction.

Table 1: Loss factor at the resonant frequency determined from the polynomial curve fit function and the half-power bandwidth method: *Y*-direction.

Speed (RPM)	Resonant Frequency (Hz)	Curve Fitted Loss Factor	Half-Power Bandwidth	Percentage Difference (%)
0	80.0	0.59	0.50	15.25
500	80.0	0.37	0.38	2.70
1000	75.0	0.50	0.47	6.00
1500	72.5	0.51	0.48	5.88
2000	75.0	0.61	0.60	1.64
2500	77.5	0.35	0.32	8.57
3000	52.5	0.67	0.70	4.48

Figs. 10-12 show the influence of various rotor speeds on the loss factor of the damper in the *X*-direction comparison of the curve fitted loss factor for different frequency regions, i.e., below the resonant frequency band, within the resonant frequency band and above the resonant frequency band.

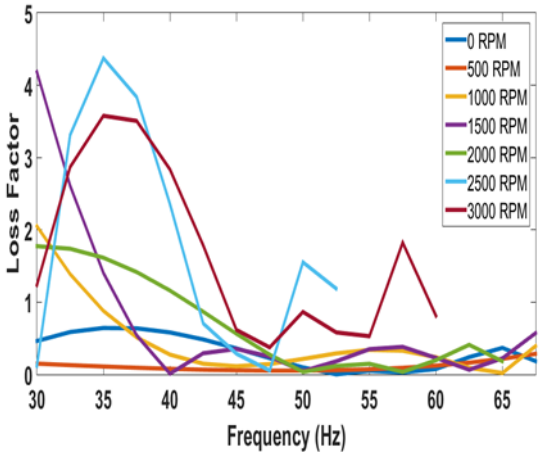


Fig. 10 Curve fitted loss factor of the damper below the resonant frequency band for various rotor speed: *X*-direction.

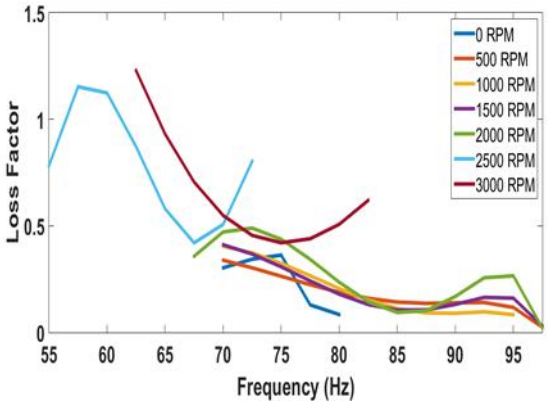


Fig. 11: Curve fitted loss factor of the damper within the resonant frequency band for various rotor speed: *X*-direction.

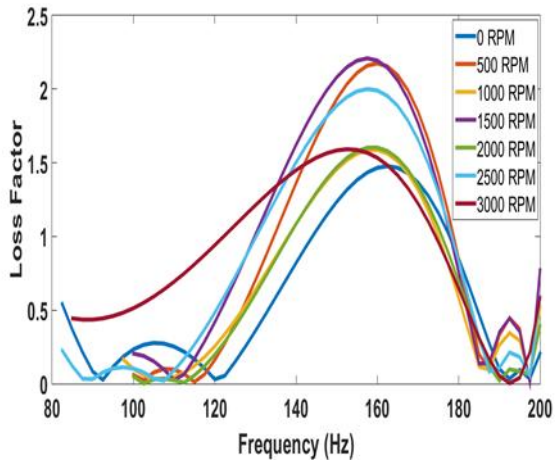


Fig. 12: Curve fitted loss factor of the damper above the resonant frequency band for various rotor speed: *X*-direction.

The half-power bandwidth method was also applied within the resonant frequency band region to evaluate the loss factor values of the elastomeric damper in the horizontal (*X*) direction. The comparison between the loss factor obtained from the curve fitted function within the resonant frequency band and also the loss factor value determined from the half-power bandwidth method is

given in Table 2. The percentage difference for both cases of non-rotating shaft and rotating shaft was less than 10%, indicating that the polynomial function, determined from the measured loss factor data, could fairly accurately predict the loss factor of the damper at the resonant frequency. The influence of the dynamic characteristics of elastomeric dampers on the vibration response of a rigid rotor at resonance was shown in the recent work by Zakaria et al. [9]. Earlier work by Akturk and Gohar [10] have also shown the effectiveness of these dampers to reduce vibration amplitude at resonance. They further highlighted the importance of the dynamic characteristics of these dampers to ensure their effectiveness in reducing the vibration amplitude of the rotor.

Table 2: Loss factor at the resonant frequency determined from the polynomial curve fit function and the half-power bandwidth method: *X*-direction.

Speed (RPM)	Resonant Frequency (Hz)	Curve Fitted Loss Factor	Half-Power Bandwidth	Percentage Difference (%)
0	75.0	0.36	0.37	2.78
500	72.5	0.30	0.28	6.67
1000	70.0	0.40	0.43	7.50
1500	70.0	0.41	0.42	2.44
2000	70.0	0.47	0.43	8.51
2500	67.5	0.42	0.41	2.38
3000	50.0	0.86	0.80	6.98

The effect of rotor speed on the loss factor of the elastomeric damper is shown in Fig. 7 and Fig. 10, which respectively represent the values for the *Y*- and *X*-directions, for below the resonant frequency band region. The variation of the loss factor for various rotor speed is seen to be quite significant in this region. Such variation was seen to be less significant for the case of above the resonant frequency region, as shown in Fig. 9 for the *Y*-direction and in Fig. 12 for the *X*-direction. For the region within the resonant frequency band, the variation of loss factor with rotor speed is observed to be more significant in the *X*-direction, Fig. 11, as compared to the *Y*-direction, Fig. 8. In the *Y*-direction, the damper is under pre-strain due to the weight of the rotor assembly, whilst in the *X*-direction the damper is essentially statically unloaded. Therefore the damper's loss factor in the *Y*-direction is influenced both by pre-strain as well as dynamic amplitude due to shaft rotation, whereas the loss factor in the *X*-direction is only influenced by the dynamic amplitude. The results in Fig. 8 and Fig. 11 indicate that the pre-strain effect on the damper's loss factor is more dominant as compared to the effect of dynamic amplitude.

Results shown in Tables 1 and 2 allow the effect of rotor speed on the loss factor in the region of resonance to be quantified. In the *Y*-direction, the loss factor values obtained using the half-power bandwidth method at 0 rpm and 3000 rpm are respectively 0.5 and 0.7. In the *X*-direction, on the other hand, the difference is more significant with values of 0.37 and 0.8 corresponding to 0 rpm and 3000 rpm, respectively. These results generally indicate that the damping determined from testing on a stationary shaft underestimates the actual damping when the shaft is rotating. Although damping is beneficial in the resonance region, too much damping can however lock up the support, and consequently reducing the effectiveness of the dampers.

5. Conclusion

The impact test method has been used to experimentally evaluate the loss factor of elastomeric damper for application in rotating machinery. The outcome of the work showed that, for the elastomeric damper under rotating load, the curve fitted loss factor derived from the impact test FRF is fairly accurate for the prediction of the damper's loss factor within the region of resonant frequency. A comparison with the more established half-power bandwidth method showed that the difference was less than 10%, for the case of the dampers subjected to rotating load. It was also shown that the loss factor determined from a stationary shaft underestimates the actual damping that is provided by the elastomeric damper during rotation of the shaft. This was seen to be more pronounced in the *X*-direction response, further indicating that the preload was more dominant as compared to dynamic amplitude in influencing the loss factor of the elastomeric dampers.

Acknowledgement

The financial support from the Ministry of Higher Education, Malaysia, under the Fundamental Research Grant Scheme (FRGS), FRGS/1/2015/TK03/UNITEN/02/6, is gratefully acknowledged.

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