Design and Performance Evaluation of Tuned Vibration Absorber for the Vibration Control of a Centrifugal Pump

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Abstract: With the prolonged use of domestic centrifugal pump, the severe vibration of the pump is resulted due to erosion, corrosion and an unbalance. An erratic sound coming out of pump makes the user uncomfortable. Here, Tuned Vibration Absorber (TVA) is used as a vibration controlling device. It is designed to reduce the RMS velocity below 20% of the circumferential speed at which the impeller is balanced. Natural frequency was found both by analytical and the numerical methods. A fan cover is considered as a TVA, tuned to the operating frequency, and dominating higher frequencies have been tackled by designing new TVA. Here with TVA, the vibration has been reduced to the permissible limit and the amplification factor is within the allowable limit.

Keywords: Allowable Amplification Factor, Centrifugal Pump, Tuned Vibration Absorber, Residual Unbalance

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Nomenclature: d_o: Fan cover outer diameter, d_i: Fan cover inner diameter, K: Stiffness of the TVA, m= Mass of TVA, ω_{cb}= Natural frequency of the TVA, ω_a= Natural frequency of the absorber, ω_b= Natural frequency of the cantilever beam, E: young's modulus of steel, I : second moment of inertia of the beam, $m_{b:}$ mass of cantilever beam, l_b: length of the cantilever beam, m_a= mass of the absorber, la: length from fixed end to absorber mass, P_1: point load due to absorber mass, P_2: equivalent point load of beam mass, y_1: deflection at point 1, y_2 : deflection at point 2, m_r: mass of rotor, ω_r: operating frequency, F:residual unbalance force of rotor, x_1:deflection of primary mass, X_{st}: static deflection of primary mass, p: Natural frequency of the primary mass, q: Natural frequency of the TVA mass.

1 INTRODUCTION

Domestic centrifugal pumps are finding wide application to lift water. Due to erosion of impeller and its vanes, bearings and also due to presence of residual unbalance force, the pump in its long run found producing erratic sound and large vibrations. The Tuned Vibration Absorber (TVA) consists of a mass, damping and stiffening elements. Fraham proposed the TVA system in 1909 for reducing the mechanical vibrations induced by monotonic harmonic forces [1].TVA solutions have been widely used for vibration control in mechanical engineering systems [2], [3]. The natural frequency of the TVA is tuned to the resonating fundamental frequency of the vibrating structure, so that a large amount of the structural vibrating energy is transferred to TVA and get absorbed. With attachment of TVA, the single degree of freedom system gets doubled. In its classical form, the natural frequency of TVA is tuned to match the natural frequency of the machine on which it is installed. The undamped TVAs are constant frequency applications, but range of the variation can be extended by using higher mass ratio as in Refs. [4] and [5]. Den Hartog [2] has given an optimal design theory for the undamped single degree of freedom in TVA, followed by many other methods developed to analyse the TVA under various types of excitation forces [14-17]. A dual mass cantilever beam is used as undamped TVA, tuned to operating frequency of the electric grass trimmer to control hand-arm vibrations as in Ref. [7]. TVA is also being used in machineries running at fixed rotational speed to curb large vibrations as in Ref. [9]. Presently, there are several types of TVAs in use in Japan, typically employing oil dampers, although viscoelastic dampers are also used. The TVA system has been successfully installed in slender skyscrapers and towers to suppress the wind induced structural excitations [11], [12]; such structures are used in CN Tower with height of 535m, six-story John Hancock Center, Center-point Tower of 305m height, and the tallest building in the world Taipei 101 Tower in Taiwan [13]. For the Centrifugal pump fitted onboard ship, TVA had been developed and fitted to the base frame. By attaching TVA at the base frame 35% reduction was attained in reference to the previous acceleration. Here vibration reductions of the centrifugal pump are attained by modifying the fan cover and using it as a TVA. It is necessary to note that an appreciable reduction in vibration has been attained.

Fan cover as TVA

The centrifugal pump operates with speed of 2800 rpm i.e. 46.67 Hz. The fan cover enjoys mass of0.2 Kg and d_o= 0.15m and d_i= 0.149 m; working as a TVA, its natural frequency is found to be equal to the operating frequency of the pump. Natural frequency of the fan cover is determined through the following equation:

$$\omega = \sqrt{\frac{3EI}{ml^3}} \tag{1}$$

m=0.2 kg, l=0.05 m. The fan cover is screwed to the motor body, acting as a cantilever beam. Thus the TVA is found to be tuned to the operating frequency of the pump.

Testing C.F pump in running condition without TVA:

The following velocity spectrum is obtained Through FFT analyser. The R.M.S velocity is 5.195 mm/s. The pump impeller is balanced to the grade G6.3 [10]. The maximum allowable vibration limit amounts to 6.3 mm/s RMS. The vibration readings show a RMS velocity of 5.195 mm/s which is 20% beyond the permissible vibration limit.

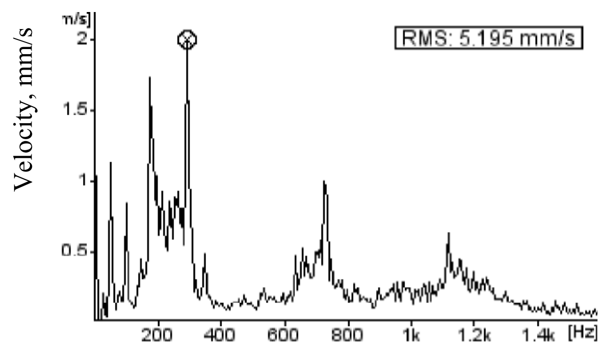


Fig. 1 Velocity response of pump before attaching TVA

Looking at the velocity response, the 293 Hz frequency shows highest velocity pick around 1.9 mm/s. The operating frequency has been taken care by the tuned fan cover that is attached to the pump. To reduce the RMS velocity, thus, to less than 20% of the maximum allowable vibration limit, we have tuned our absorber at frequency of 293 Hz.

Physical design and tuning of TVA

TVA is designed and manufactured as shown in Fig. 2. As discussed in previous section, at 293 Hz the TVA has to be tuned. For that purpose, an absorber with the mass of 0.75 Kg was considered where the stiffness of the absorber was determined to be 2539200 N/m using the relation. As TVA is to be welded to the relatively light weighted cover, three TVAs are considered to be acting in parallel. The stiffness of each TVA would be 846400 N/m.

Fig. 2 TVA welded to the fan cover

For the analytical determination of natural frequency of TVA, the horizontal beam is assumed to be one system and the absorber mass at the end of beam is counted as another. Neglecting TVA system damping, the natural frequency of the TVA ω_{cb} can be determined analytically by:

(1) Dunkerleys approximation.

(2) Rayleigh method.

(1) Dunkerleys approximation

Dunkerleys approximation as in Ref. [1] and Ref. [2] may be presented as:

$$\frac{1}{\omega_{cb}^2} = \frac{1}{\omega_b^2} + \frac{1}{\omega_a^2} \qquad (2)$$

Wherein E= 200 GPa, m_b= 0.05495 Kg, l_b= 0.07m, m_a= 0.2198 Kg, la= 0.055m.The natural frequency based on the calculation is ω_{cb}= 288 Hz.

(2) Rayleigh method as in [2]

An energy method used to find the natural frequency of the beam whenever it is subjected to transverse loading. A good estimation of fundamental frequency can be made by assuming the suitable deflection curve for the fundamental mode.

$$\frac{\omega^2}{2g}\Sigma \mathrm{Py}^2 = \frac{1}{2}\Sigma \mathrm{Py} \qquad (3)$$

P_1= 2.156 N, P_2= 0.539N.

Equating the maximum kinetic energy to maximum potential energy, the static deflection at two points were obtained by using the following relations:

$y_1=M_1ga_{11}+ M_2ga_{12}$,(4)

$$y_2=M_1ga_{21}+ M_2ga_{22}. \qquad (5)$$

The calculated values of y_1 and y_2 are 9.8646e-7 and 3.09199e-6 respectively.

$$\omega = \sqrt{\frac{g\,\Sigma \mathrm{Py}}{\Sigma \mathrm{Py}^2}} \qquad (6)$$

The natural frequency of TVA is calculated to be 292 Hz.

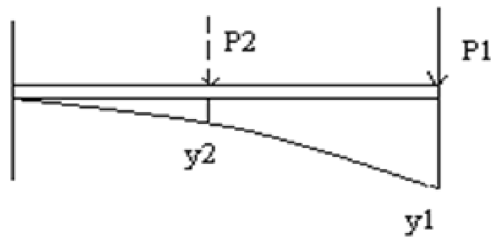


Fig. 3 Cantilever beam with assumed deflection

2 MODAL ANALYSIS

The FE modal analysis is carried out to determine the mode shape and the natural frequency of the TVA. The meshed model of TVA is shown in Fig. 4, where SOLID 45 elements were used to mesh the model. The fundamental frequency of 296 Hz is obtained from the modal analysis.

This shows somewhat nearer to the natural frequency that is found with the analytical methods. The reason for the difference is beam length, considered 70 mm in FE modal analysis, as the 20 mm to be welded to the cover. The length of 20 mm is given a fixed boundary condition i.e. all DOFs which are constrained.

Determining the allowable amplification factor

(1) Residual unbalance force

For rotor balanced as per G6.3 grade, the residual unbalance force can be calculated by considering m_r= 2.5 kg, ω_r= 48 Hz [7]. The value of F was calculated as 4.61475 N.

(2) Static deflection of main mass under excitation force

Static deflection X_{st} is calculated by considering main mass of 7 kg. Stiffness of main mass is counted as K= 601216.4978 N/m.

$X_{st} = \frac{F}{K}$, the value calculated amounts to 7.6756 µm.

(3) Allowable amplification factor

$$X = \frac{X_1}{X_{st}} \tag{7}$$

Based on the calculation, it is concluded thatx_1 = 0.30408 µm, and the allowable amplification factor from Eq. (7) is found to be 0.0396.

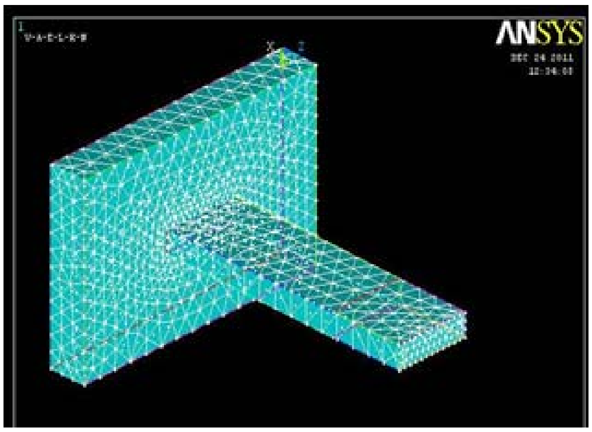


Fig. 4 FE Meshed model of TVA

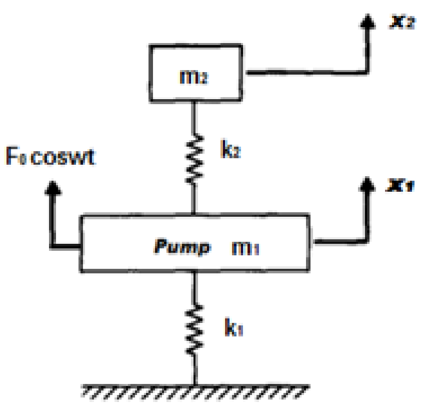


Fig. 5 Pump system and TVA model

Before attaching TVA, the displacement RMS value is 149.26 µm and corresponding amplification factor X= 19.446. The amplification factor is found to be 4.9 times more than the allowable value. After attaching TVA, the displacement RMS value has been reduced to 11.456 µm and corresponding X =1.4925 which is far below the allowable amplification factor value.

Determining the safe operating range around the resonant frequency

For the combined system shown in Fig. 5, the equations of motion are as follow:

$$m_1\ddot{x}_1 + k_1x_1 + k_2(x_1 - x_2) = F_0\, sin\omega t \tag{8}$$

$$m_2\ddot{x}_2 + k_2(x_2 - x_1) = 0 \tag{9}$$

Here, x_1= Displacement of the primary system, assume it harmonic, i.e., $x_1 = X_1 \cos\omega t$, x_2= Displacement of the TVA system, assume it harmonic, i.e.. $x_2 = X_2 \cos\omega t$. Considering mass ratio µ = K_1/K = m_2/m_1, the two new natural frequencies are as follow:

$$\omega_{n1,2}^2 = p^2\,[1 + \mu/2 \pm \sqrt{\mu + \frac{\mu^2}{4}}\,] \tag{10}$$

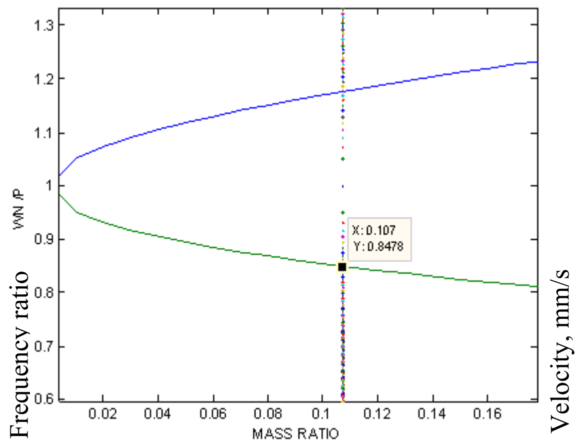


Fig. 6 Two new natural frequencies in relation to mass ratio of the TVA

The mass ratio for our case is 0.107 and for that, two new natural frequencies obtained from Fig. 6 are 0.8478 and 1.1768 times the resonant frequency. The 'C' curve as in Fig. 6 is used to determine the safe operating range. By simulating the amplitude ratio for equation (10) the same safe frequency range is faced.

In Fig. 7 the allowable amplification factor for the primary system is shown. For the frequency ratio greater than 1, the amplitude of primary and absorber systems are in phase opposition which contributes to prevent the vibration of the primary system.

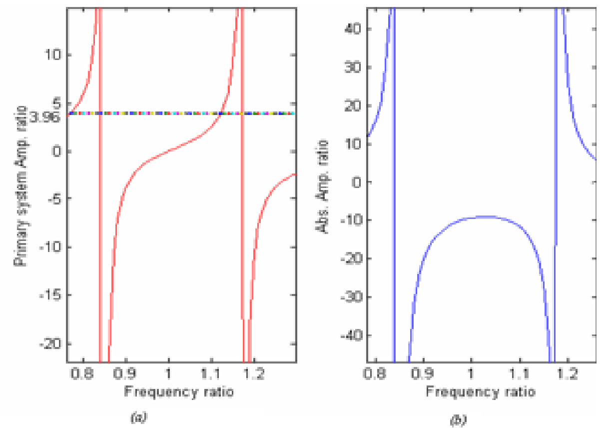
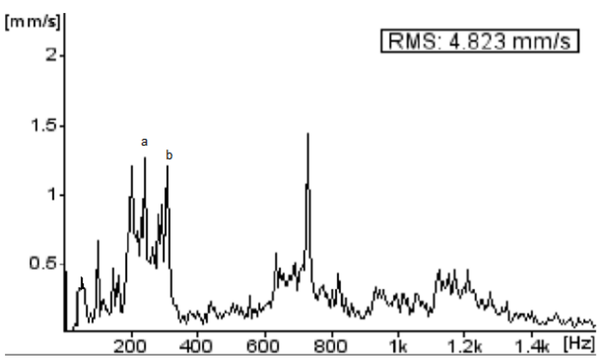


Fig. 7 Amplitude ratio to frequency ratio of primary system (a), Absorber system (b)

Testing the C.F. pump in running condition with TVA

After attaching the TVA, the vibration response - shown in Fig. 8 - indicates a RMS velocity of 4.813 mm/swhich is 20%beyond the permissible vibration limit. The velocity response of 293 Hz frequency comprises the velocity of 0.654 mm/s. Prior to attaching the TVA, the velocity at 293 Hz was 1.418 mm/s, almost 54% reduction in velocity is obtained.



Frequency, Hz

Fig. 8 Velocity response of Pump after attaching TVA, a): 243Hz, b): 312 Hz

Prior to attaching TVA as shown in Fig. 1, we have 1.418, 0.978, 0.814 mm/s velocity for the frequency values of 293, 291, 296 respectively. Now with the attachment of the TVA, the velocity has been reduced to0.892, 0.658, 0.654 mm/s for the frequency values of 243, 312 and 293 Hz respectively. Here it can be considered that for the frequency range of 243-312 Hz, the vibration is considerably reduced. For the new safe frequency range, the lower and higher frequencies obtained experimentally are somewhat lower than the theoretical calculations.

After attaching the TVA, the erratic sound experienced before the attachment is reduced. The sound intensity reduction has not been measured but determined based on the experience of the daily user of the same pump. Here in Fig. 8 it is noticed that the 48 Hz frequency shows lower velocity value of 0.4 mm/s. The TVA is welded to the cover and thus becomes bulky; The screws that keep the cover tight to the pump body are only 5mm deep, so the connection of the cover to the pump has been loosen during the running. The remedial action, accordingly, is to tapa hole of 10mmin depth and using corresponding long screws with washer.

3 CONCLUSION

Attaching the TVA to fan cover of a pump, results in reduction of sound and vibration considerably. The RMS velocity of the centrifugal pump (with TVA) is reduced to 4.813 mm/s which is 20% beyond the maximum allowable vibration limit of 6.3 mm/s RMS. Through using the TVA, the amplification factor is 1.4925 which is 62% less than the allowable amplification factor. The displacement RMS value has been also reduced to 11.456 µm which is 13 times less than that of the TVA-free state. The new lower and higher natural frequencies are 17% and 7% respectively within the resonant frequency of 293 Hz.

Future scope

Further researches may comprise accomplishing the case design modification, i.e. instead of attaching TVA separately to fan cover, it can be considered as a part of the casing, making the fan cover free to attain the operating frequency of the pump.

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