Add Seismic Mass . . . It's Easier

Richard L. Smith, R. L. Smith Enginering, Newmarket, New Hampshire

The addition of mass to control a machinery vibration resonance problem is compared to the addition of stiffness. Adding mass is less complicated than adding stiffness and will yield comparable results.

Quite often, particularly with newer, higher speed, and more lightly built machinery, vibration problems are encountered. Such is often the end result of economics. Like it or not, we live in an increasingly competitive world, a world where suppliers will meet the requirements of a specification in the most inexpensive method possible. The reason for this is simple. If he does not, the next person will.

Such a mindset will inevitably lead to the occasional machine with serious vibration problems. One cause for this is the trend toward higher speed machinery. For most fans and pumps all other factors being the same, the amount of product output is a function of the square of running speed. As such, doubling running speed will increase product output by a factor of approximately four, tripling a factor of approximately nine.

At the same time, the cost of increasing the speed of most machines is relatively low. So for a given product flow rate, the highest-speed machine will generally be the least expensive and most attractive to an equipment supplier. There are limits. For AC-induction motors, the upper limit will be line frequency, most often 60 Hz or 3600 CPM in North America. However, there is a factor worth considering that is inherently contrary to higher speed. That is centrifugal force. The general equation for force is:

$$F = MA \tag{1}$$

where F is force (pounds); M is mass, weight divided by gravitational acceleration in consistent units (Pounds/386 inches per sec^2); and A is acceleration (inches per sec^2). For a rotating system, centrifugal acceleration is determined by the following equation:

$$A = \omega^2 R \tag{2}$$

where ω is rotational speed in radians per second, and R is the radius of gyration (inches). So all other things being the same, the stress imposed on a rotating assembly is proportional to the square of the rotating speed. Such faults as the level of acceptable residual unbalance and the allowable misalignment have to be reduced proportionally to the square of the running speed to achieve the desired level of vibration. The other major factor is system restraints. As projects are becoming more and more influenced by budgetary costs, efforts are constantly being made to reduce these costs. This generally results in lighter, less rigid construction.

If no problems are encountered, these efforts will have been successful in reducing overall installation costs. However, making a structure lighter will generally reduce resonant frequencies. While at the same time, this increase of operating speeds can move such speeds into the range that will excite a system resonance. This can result in a very high, potentially damaging or unsafe level of vibration. If one is unfortunate enough to inherit such a problem, what is the best way to address the situation? The simple answer is that there is no single solution. Before proceeding, a basic discussion of resonances is in order. The simple formula for resonance is:

$$\omega_R = \sqrt{K / M} \tag{3}$$

where ω_R is the system resonant frequency in radians per second; K is system stiffness (force necessary to impart one unit of displacement, generally lb/inch); and M is the system mass (defined in Eq. 1). So changing either mass or stiffness will move the resonant frequency. However, because the result on the resonant frequency is a function of the square root, it may be necessary to change either the mass or stiffness significantly before the system resonance is moved enough to be out of the range of influence of operating speed.

On the other hand, the frequency range that will be affected by the system resonance is generally quite narrow. As such, one does not have to miss the resonance by much for it not to significantly affect system vibration.

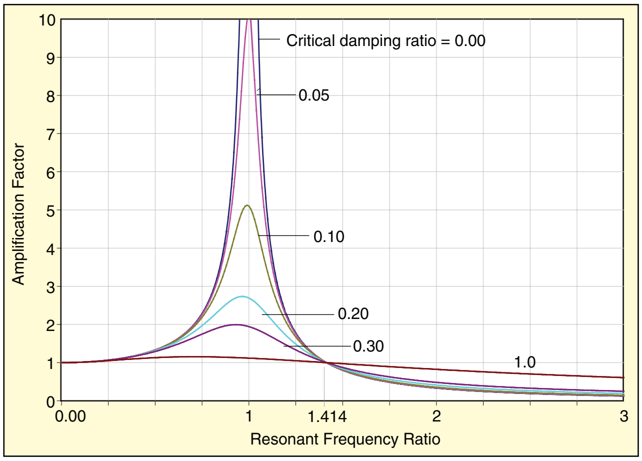

Figure 1. Amplification factor versus resonant frequency ratio; note that frequency range affected by system resonance is quite narrow.

Figure 1 is a plot of *amplification factor* versus *resonant frequency ratio*. The amplification factor is the ratio of *static displacement* versus *dynamic displacement*. Static displacement is the amount of displacement that will be imparted on a system by static force of a given magnitude. Dynamic displacement is the maximum displacement that will be imparted on the system by a force of the same maximum magnitude, but the magnitude of this force varies sinusoidally. The resonant frequency ratio is the ratio of a particular frequency to the system resonant frequency. At a resonant frequency ratio of 1, the system is running at the resonant frequency. The different plots in Figure 1 are for different amounts of system damping. Most systems fabricated primarily from structural steel have very little damping.

For some reason, whenever a resonance-related problem is encountered, the first approach most often considered is to add stiffening. It is surmised that the reason for such thinking is a desire to move the resonance up and completely out of the frequency range of operating speed even during start-up and coast-down. Such thinking is fine for the next generation of the machine that will be fabricated from scratch and under controlled conditions. In an existing system, however, implementing such changes can be challenging. First, a stiffening device generally requires two points of attachment, one on the vibrating system and one on the substrate. On each, there must be an attachment point in relatively close proximity to each other. Also, the substrate must have adequate inertia so that it is not just pulled along with the system. And the stiffener must have adequate rigidity that it is not easily deformed by the system vibration.

On the other hand, adding mass is generally much simpler. First of all, a seismic mass requires only to be attached to the primary system. Second, in general, it is relatively easy to test such a device. The mass can be temporally attached even when the machine is operating. Also, if the size of the mass must be adjusted, it is generally quite easy to add or remove mass.

Let us see the results of an example. The subject machines were four more or less identical pumps. Each was a vertically mounted lubricating oil pump for a large induced-draft fan. There were two pumps for each fan. These pumps are relatively small, but having at least one operational is required for fan operation, even when the fan is off line and on turning gear. As such, the consequences of not having at least one pump available for service at all time would be very grave. All of these pumps had a high level of vibration (see Figure 2).

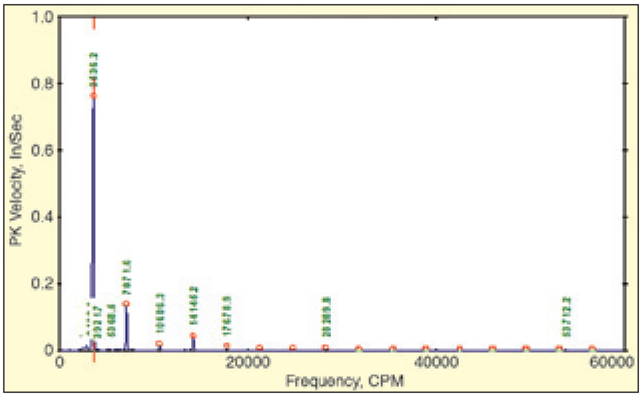

Figure 2. Frequency spectra obtained from subject pump immediately before seismic mass was attached; note high level of vibration (~0.8 inches per second, IPS, at fundamental running speed, ~3540 CPM).

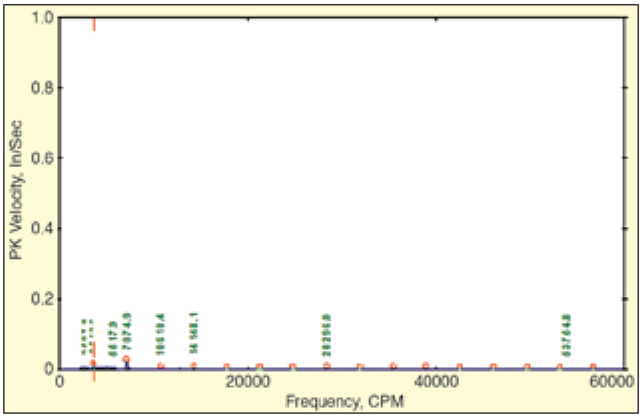

Figure 3. Frequency spectra obtained from subject pump immediately after seismic mass was attached; note dramatic reduction in level of vibration.

As noted previously, the motors are mounted vertically, but there were also side mounts should the application have been horizontal. The side mounts made an ideal site on which to attach a seismic mass. A steel flange weighing approximately 20 pounds was attached to the side mounts with C-clamps.

Figure 3 is the frequency spectra obtained immediately after the seismic mass was attached. For clarity, the amplitude scale is the same in both Figure 2 and 3. After the seismic mass was attached, running speed vibration amplitude went from approximately 0.8 IPS to 0.01 IPS, a reduction of almost two orders of magnitude.

In summary, whenever a resonance-related vibration problem is encountered, it is possible to detune the system so that the resident forcing function, quite often fundamental running speed, will no longer excite the resonance. This generally will dramatically reduce the level of vibration. The two basic ways that this can be done are by adding either stiffness or mass to the primary system. In an existing system, adding mass is generally far less complicated to implement and will yield comparable results. S/V

The author can be contacted at: rlsmith_eng_pe@myfairpoint.net.