DYNAMIC ABSORBERS FOR SOLVING RESONANCE PROBLEMS

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ABSTRACT

Many experts in vibration analysis will agree that resonance is one of the five most common causes of excessive machine vibration along with unbalance, misalignment, looseness/weakness and distortion. Without question, the best solution to solving a resonance problem is to separate the component natural frequency from the responsible exciting force frequency. This can be accomplished by changing the component natural frequency by increasing or decreasing mass or stiffness, or by increasing or decreasing the exciting force frequency. Unfortunately, in many cases, such changes are not possible or cost effective. In addition, some machines may operate over a wide speed range such as VFD (Variable Frequency Drives) where modifications to separate the potential exciting force vibration frequencies may be virtually impossible.

Another possible solution to solve a resonance problem is to install a dynamic absorber, a device designed to have the same resonate frequency which, through its own vibratory force, will counteract the initial exciting force. This paper outlines the theory of dynamic absorbers, presents several case histories of resonate problems solved by the application of dynamic absorbers and details the design, fabrication and installation of dynamic absorbers to control resonance.

It should be noted that the use of dynamic absorbers is not a new technology. Dynamic absorbers have been used for many decades to control resonance problems.

THEORY OF THE DYNAMIC ABSORBER

Where a system is diagnosed as resonate, its vibration amplitude can be significantly reduced by purposefully creating an “anti-node” by attaching a second resonate spring-mass system in series. This device is called a “dynamic absorber”. For example, assume that the bearing pedestal in Figure 1 has been diagnosed as resonance. Attaching the dynamic absorber in series with the resonate pedestal as illustrated creates an anti-node for the bearing pedestal, significantly reducing the resonate vibration amplitude.

To help understand how a dynamic absorber works to control resonance, Figure 2 is a plot of the vibration amplitude and phase obtained on a machine during startup or coast down. This data, referred to as a Bode’ plot, clearly displays the response of the machine as it passes through a resonant or “critical” frequency. First, the vibration amplitude is significantly amplified at the resonate frequency. This is the frequency (RPM) at which the mass (M) or inertia force has become equal to the stiffness force (K). And, since mass (M) and stiffness (K) are 180 degrees out-of-phase, at this frequency, they are essentially equal in magnitude and cancel one another. In other words, at this frequency, the machine element essentially has no mass or stiffness to help control the exciting vibratory force. The result is a significant amplification in the vibration amplitude.

At the resonate or “critical” vibration frequency, only system damping (C) remains to control the amplification of vibration. Theoretically, if a machine or machine component was vibrating at a resonate frequency with no damping, the vibration amplitude would approach infinity. Of course, this is not possible. Some degree of damping is always available either through molecular friction, air resistance, etc.
Figure 1. Attaching a “tuned” resonate spring-mass system to a resonate structure, creates an “anti-node” to create an out-of-phase exciting force to effectively counteract the initial exciting force.

The important points to understand from the Bode’ plot in Figure 2 are as follows:

- When the machine is operating at a frequency below the resonate frequency, the machine is essentially controlled by stiffness (K). Thus, the motion of the machine is in-phase with the exciting force and 180 degrees out-of-phase with stiffness.

- When the machine is operating at a frequency above the resonate frequency, the machine is essentially controlled by mass (M). Because of this transition from stiffness controlled to mass controlled, above the resonate frequency, the motion of the machine is 180 degrees out-of-phase with the exciting force.

- When the machine is operating at a frequency at or close to the resonate frequency, the machine is essentially controlled by damping, and the motion of the machine lags the exciting force by 90 degrees.

Figure 2. A Bode’ plot showing vibration amplitude and phase response below, through and above resonance.
To understand how a dynamic absorber can effectively reduce the amplitude of resonate vibration, assume that the spring-mass system “A” shown in Figure 3 is resonate in the vertical direction and is being excited by the unbalance force “U”. If the system was operating below the resonate frequency, the motion of mass “A” would be essentially stiffness controlled and in-phase with the unbalance force. Or, if the system was operating above the resonate frequency, mass “A” would be mass controlled and vibrate 180 degrees out-of-phase with the unbalance force “U”. However, if the system is operating on or very near the resonate frequency, the resonate system “A” will vibrate 90 degrees out-of-phase with the exciting force of unbalance. In other words, the motion of machine “A” will lag the unbalance force by 90 degrees at resonate.

If spring-mass system “B” (a dynamic absorber) is attached to “A” and “B” is “tuned” to the same resonate frequency, the motion of “B” must lag its exciting force by 90 degrees also. And, since “B” is attached to “A”, the exciting force for system “B” is the vibratory force from system “A”. As a result, the motion of system “B” lags the motion of system “A” by 90 degrees, and the vibration of system “A” lags the unbalance exciting force by 90 degrees. Thus, the vibratory motion of system “B” lags the unbalance exciting force by a total of 180 degrees (90 + 90 = 180). The force of vibration generated by the dynamic absorber (system “B”) is 180 degrees out-of-phase with the exciting force of unbalance (U), and since both of the opposing forces are acting on system “A”, the vibration amplitude of system “A” is effectively reduced. The purpose of the dynamic absorber (B) is to generate, through its own resonate vibration, a vibratory force equal in intensity and 180 degrees out-of-phase with the exciting force.

![Figure 3](image3.png)

Figure 3. The force generated by the mass “B” is 180 degrees out-of-phase with the exciting force. With opposing forces acting on mass “A”, the vibration of system “A” is effectively reduced.

**APPLICATIONS FOR DYNAMIC ABSORBERS**

The dynamic absorber is by no means a new tool for controlling resonate vibration. They have been used for many years as a possible solution to many resonance problems. In his book “Mechanical Vibrations”, first published in 1934, Mr. Den Hartog covers the theory and practical applications of the dynamic absorber in great detail.

Many leading automobile manufacturers have utilized the dynamic absorber for many years to control annoying and potentially damaging resonance problems. Because an automobile must operate over a wide range of operating speeds, with numerous variable exciting force frequencies from the engine, transmission, drive shaft, axles and other rotating components, it would be nearly impossible to totally avoid resonance problems. For example, one leading automaker incorporates a dynamic absorber bolted to the transmission extension housing to control resonance problem in the drive train. In another instance, a dynamic absorber fastened to the exhaust catalytic converter controls an annoying resonance of the exhaust system. In both cases, whenever the vehicle reaches an operating speed to excite a resonate condition, the dynamic absorber effectively “opposes” the exciting force to minimize the
vibration amplitude. At any other operating speed, the dynamic absorber is ineffective and has no effect on vehicle vibration.

From the above examples, it should be apparent that the use of the dynamic absorber can be quite effective in controlling resonate conditions on machines which must operate over a wide range of operating speeds. Increasing or decreasing the system mass or stiffness to change a resonate frequency in these cases would be of little value since changes would simply move the resonate frequency to another operating speed. In addition, the dynamic absorber can be used to minimize damage to machines such as pumps, machine tools, refrigeration equipment and other machines that must be operated over a wide range of operating speeds or continuously started and stopped and which must pass through resonate or critical frequencies each time. Although the vibration amplitude may be quite acceptable at certain operating speeds, a machine subjected to high levels of vibration each time it passes through a resonate frequency may experience damage and premature failure.

**DYNAMIC ABSORBERS TO DIAGNOSE RESONANCE**

Another useful application for the dynamic absorber is to verify a resonate problem where other analysis or diagnostic techniques are not possible. For example, if it is not possible to shutdown a machine to perform a coast-down test such as a cascade plot, Bode’ plot, polar (Nyquist) plot or to perform a “bump” (impact) test with the machine shut down, it may be possible to temporarily attach a dynamic absorber to the machine while it is operating. In many cases, a dynamic absorber can be applied to the machine or machine component suspected of being resonate while the machine is operating. If a dynamic absorber, temporarily bolted or clamped to the machine or machine element suspected as being in resonance, and properly tuned, effectively reduces the vibration amplitude, the problem is truly a resonance problem.

If the vibration problem is not resonate, attaching dynamic absorber will actually create a resonate problem and effectively amplify or increase the vibration amplitude. In other words, adding a dynamic absorber can only do one of two things. If the vibration problem is truly a resonate problem, applying a dynamic absorber will reduce the vibration amplitude. However, if the excessive vibration is not due to resonance, applying a dynamic absorber will actually create a resonate condition and increase or amplify the vibration amplitude. To illustrate, a 4-stage centrifugal pump in a petroleum pipeline was suspected to operating at resonance. The inboard (drive-end) bearing had a vibration amplitude in excess of 1.0 in/sec (25.4mm/sec) but only in the vertical direction. The horizontal and axial vibration amplitudes were well within acceptable levels. Because of the extremely high levels of vertical vibration on the pump bearing, a resonance of the bearing in the vertical direction was suspected. The frequency of the high-level vibration was the vane-passing frequency or approximately 25,000 CPM (416.67 Hz.). The vane-passing frequency was generated by the pump’s 7-vane impellers operating at rotating speed of approximately 3571 RPM (3571 RPM x 7 vanes = 24,997 CPM). Since the pump could not be shut down to perform other analysis techniques such as coast-down tests or “bump” tests to verify a resonate condition, a dynamic absorber was designed, fabricated and attached to the pump bearing by bolting it to one of the six bolts securing the upper portion of the bearing. When the dynamic absorber was “fine tuned” to the exciting force frequency, the vibration amplitude of the pump bearing virtually doubled from 1.0 in/sec (25.4 mm/sec) to 2.0 in/sec (50.8 mm/sec). Since the dynamic absorber failed to reduce the pump bearing vibration, bearing resonance was excluded as a possible cause of excessive vibration. Later tests determined that the high level of vibration was the result of improper impeller design for the specific product being pumped at the time of the vibration analysis.

Temporary dynamic absorbers can usually be attached by bolting or with “C” clamps. If the temporary dynamic absorber does effectively reduce the vibration, it can also be left in place to minimize the vibration until a more permanent solution is implemented.
A CASE HISTORY

Figure 4 illustrates a dynamic absorber attached to a 3600 RPM (60 RPS) horizontal centrifugal pump in a major petroleum refinery. The pump shown is one of two identical direct-coupled, motor-driven pumps that were experiencing very high levels of vibration in the horizontal direction, but only on the pump bearing. The horizontal vibration amplitude of the pump bearing was typically 1.3 to 1.5 inches/second (33 to 38 mm/sec), whereas the vertical and axial vibration amplitudes, by comparison, were less than 0.15 inches/second (3.8 mm/sec). Vibration amplitudes on the motor were typically less than 0.12 inches/second (3.0 mm/sec) in all directions.

According to plant maintenance and engineering personnel, the pumps exhibited high levels of horizontal vibration on initial installation and required bearing and seal replacements approximately every four weeks of operation.

Figure 4. Dynamic absorbers effectively reduced the resonate vibration on these centrifugal pumps from 1.3 to 1.5 inches/second (33 to 38 mm/sec) to less than 0.09 inches/second (2.3 mm/sec).

Due to the fact that the horizontal vibration amplitude was nearly ten (10) times the vertical and axial amplitudes, a resonate condition was strongly suspected. An amplitude-versus-frequency (FFT) analysis disclosed that the frequency of vibration was 3600 CPM (60 RPS) or 1 x RPM.

To test for resonance, one of the pumps was shut down and a “bump” test was performed in the horizontal direction on the pump bearing using a rubber mallet. The bump or impact test verified that the horizontal natural frequency of the pump bearing was, in fact, 3600 CPM (60 Hz). A similar pump test was later performed on the second pump showing that it was also resonate in the horizontal direction at 1 x RPM.

Once the resonate conditions of both pumps were confirmed, the decision was made to “solve” the problem utilizing dynamic absorbers. Attempting to change the horizontal natural frequency of the pump bearing by increasing mass was totally out-of-the question. In this case, the resonate probably could have solved by increasing the horizontal stiffness of the pump bearing by redesigning the bearing vertical support brace to increase horizontal stiffness and raise the bearing horizontal natural frequency. However, plant engineering elected to utilize the dynamic absorber approach to solve the resonate vibration problem and fabricated and installed the absorber shown in Figure 4.
The dynamic absorbers were flame-cut from a ¾ inch thick steel plate with the base portion dimensioned and drilled to match the bearing cap bolt pattern. Once installed and properly “tuned”, the dynamic absorbers effectively reduced the vibration amplitudes on both pumps from the initial levels of 1.3 to 1.5 inches/second (33 to 38 mm/sec) to levels less than 0.09 inches/second (2.3 mm/sec), a most dramatic improvement.

The dynamic absorbers were installed on these two pumps in March, 1980. The refinery was closed in January, 1990. According to plant maintenance personnel, over this time span of nearly ten years, neither pump experienced a bearing failure.

**DESIGNING A DYNAMIC ABSORBER**

Designing a dynamic absorber to solve a resonance problem is a fairly straightforward procedure; however, since the intensity of the dynamic force exciting the resonance is not usually known, some trial-and-error experimentation may be necessary.

Figure 5 shows a sketch of a typical dynamic absorber, along with the dimensions and calculations needed to determine the amount of weight (W2) needed at a specified distance (a) on a piece of flat or rectangular bar stock of length (L), having cross-sectional dimensions of (b) and (h) to achieve the desired natural frequency (fn). For those with little or no experience with designing dynamic absorbers, several trial calculations may be necessary to establish reasonable absorber dimensions and weights. However, after a few attempts, establishing the initial dimensions based on the required natural frequency will become easier.
Figure 5. Dimensions, values and equations needed to design a dynamic absorber.

Establishing the “L” and “a” dimensions

The first dimension to be estimated is the freestanding length (L) of the dynamic absorber. This dimension will depend on the natural (resonate) frequency desired as well as the unobstructed space available for the absorber. Typically, a dynamic absorber designed for a resonate frequency of 3600 CPM (60 Hz) may have a length of 10 to 14 inches (25.4 to 35.5 cm), depending on the cross-sectional dimensions of the absorber bar stock being used.

\[ W_2 = \frac{(2.114 \times 10^5) E l}{N_f^2 (3a^2 L - a^3)} - \frac{0.75 w L}{3a^2 L - a^3} \]

WHERE:  
\[ N_f = \text{DESIRED NATURAL FREQUENCY (CPM)} \]
\[ w = \text{WEIGHT PER INCH OF SPRING (lbs)} = b \times h \times d \]
\[ E = \text{MODULUS OF ELASTICITY} \]
STEEL = 29,000,000 psi
ALUMINUM = 10,000,000 psi
COPPER = 16,000,000 psi
IRON = 18,000,000 psi
\[ l = \frac{b (h^3)}{12} \]

MOMENT OF INERTIA FOR  
RECTANGULAR CROSS SECTION  
“b” and “h” dimensions in inches

\[ W_2 = \text{REQUIRED WEIGHT IN POUNDS (lbs)} \]

DENSITIES (d):  
STEEL = 0.262 lbs/in³
ALUMINUM = 0.099 lbs/in³
COPPER = 0.321 lbs/in³
IRON = 0.260 lbs/in³
If the resonate frequency is less than 3600 CPM (60 Hz), the length (L) will typically be greater. For example a dynamic absorber for a resonate frequency less than 1000 CPM (16.7 Hz) may be more than 24 inches (61 cm). On the other hand, a dynamic absorber designed to control a vane-passing resonate frequency of, say, 25,000 CPM (417 Hz), may have a free-standing length (L) of only 5 or 6 inches (12.7 to 15.24 cm). Again, a little “trial-and-error” effort may be required to establish some reasonable dimensions for the dynamic absorber.

As an example, assume that a dynamic absorber is needed to solve a 3600 CPM (60 Hz) resonate frequency, and it was decided to make the absorber bar 12 inches (30.5 cm) long. The next dimension to be established is the distance along the bar (dimension “a”) for placing the adjustable tuning weight (W2). Normally, the “a” dimension will be 10% to 20% less than the “L” dimension to allow ample movement for adjusting or “fine tuning” the position of the tuning weight (W2) to achieve the exact desired results.

For an absorber 12 inches (30.5 cm) long, an “a” dimension of 8 to 10 inches (203 to 254 cm) would be appropriate.

Establishing the “b” and “h” dimensions

Next, the cross-sectional (b and h) dimensions of the dynamic absorber bar must be selected. The cross-sectional dimensions of the absorber bar should be rectangular. Since nearly all structural resonance problems are typically directional, the dynamic absorber must be designed to have a natural frequency in a specified direction. If, for example a round rod was used, the radial stiffness of the rod will be equal or uniform in all radial directions (360 degrees). As a result, the round rod could vibrate or “whip” in a circular or conical fashion rather than in one specific direction. This might reduce the amplitude of vibration in the initial direction of resonance; however, it may create a resonance problem in a direction perpendicular to the original resonance. The same may be true if the absorber bar stock had a square cross-section. Therefore, round and square bar materials should be avoided. Virtually any rectangular bar stock can be used. Again, a little trial-and-error may be necessary in establishing suitable “b” and “h” dimensions.

Selecting the appropriate dynamic absorber material

Another decision that must be made is to select the material (steel, aluminum, brass, etc) to be used in fabricating the dynamic absorber. This choice may depend on factors such as possible problems with rust, corrosion or adverse reaction with other materials or chemicals.

For a given set of “L”, “a”, “b” and “h” dimensions, bars of different materials will have different levels of mass and stiffness and, thus, different natural frequencies. The effective stiffness of a specified material is quantified by its modulus of elasticity, sometimes referred to as Young’s Modulus. This is a number that represents the ratio of a stress applied to a material (either compression or tension) and the resulting strain or deflection which results. For example, in Figure 5, the modulus of elasticity (E) for steel is given at 29,000,000 PSI (pounds per square inch). This means that if a block of steel were compressed with a force of 29,000,000 pounds per square inch, it would compress a distance of 1 inch. Aluminum, on the other hand, is not as stiff as steel and requires only 10,000,000 PSI to achieve 1 inch of deflection. The modulus of elasticity for common dynamic absorber materials is given in Figure 5 for reference purposes.

If a different material will be used in the fabrication of the dynamic absorber, refer of an engineering handbook for the appropriate modulus of elasticity number as well as the material density “b” in pounds per cubic inch.

The values for modulus of elasticity and density given in Figure 5 are approximate values and may vary from the actual values by as much as 10%, depending on the exact metallurgy of the material being used. However, these minor errors are of little importance since the dynamic absorber can be finely tuned after installation to compensate for minor differences in materials, dimensions and the actual desired natural frequency.
Calculating the moment of inertia

Another value, which affects the natural frequency of the dynamic absorber, is its moment of inertia \( I \). Since the total mass and stiffness of the dynamic absorber bar is determined not only by the material selected but also by the relationship between the “b” and “h” dimensions (i.e., the thickness relative to the width), these dimensions must be taken into consideration as well. Since dynamic absorbers are typically used to control directional structural resonance problems, the equation for calculating the moment of inertia for rectangular bar stock is given in Figure 5. Normally, the dynamic absorber is designed around the lowest natural frequency (i.e., in the direction across the “h” dimension) and is presented as such in Figure 5. The dynamic absorber can be designed to be resonate in the stiffer direction (the “b” direction) by simply reversing the “b” and “h” dimensions in the equation, Figure 5.

A sample dynamic absorber calculation

With the dimensions and values of a sample problem listed below, the following outlines the calculations needed to determine the amount of weight \( W_2 \) for a dynamic absorber for solving a 3600 CPM (60 Hz) resonate problem.

Material = Carbon Steel

Modulus of Elasticity = 29,000,000 PSI

Density = 0.282 pounds per cubic inch

L = 12 inches

a = 10 inches

b = 0.75 inches

h = 0.50 inches

\[ F_n = 3600 \text{ CPM (60 Hz)} \]
An easy way to solve the problem for calculating the dynamic absorber tuning weight (W2), is to first calculate and list all the values needed to solve the equation shown in Figure 5.

\[
\frac{b}{12} = \frac{0.75(0.125)}{12} = \frac{0.09375}{12} = 0.007813
\]

\[
3600\text{ CPM}^2 = 12,960,000
\]

\[
a^2 = 10 \times 10 = 100
\]

\[
a^3 = 10 \times 10 \times 10 = 1000
\]

\[
w = b \times h \times 0.262\text{ lbs/ln}^3 = 0.75 \times 0.50 \times 0.262\text{ lbs/ln}^3
\]

\[
= 0.1058\text{ lbs}
\]

\[
L = 12 \times 12 \times 12 \times 12 = 20,736
\]

By simply inserting these values into the dynamic absorber main equation shown in Figure 5, the amount of the adjustable tuning weight (W2) is calculated as follows:

\[
W2 = \frac{211,400 \times 29,000,000 \times 0.007813}{12,960,000 \times (3 \times 100 \times 12 - 1000)} - \frac{0.75 \times 0.1058 \times 20,736}{(3 \times 100 \times 12) - 1000}
\]

\[
= 1.419 - 0.634
\]

\[
= 0.785\text{ lbs}
\]

From the calculations outlined above, a weight of 0.785 pounds located at a distance of 10 inches on a piece of carbon steel bar stock 12 inches long and having cross-sectional dimensions of 0.5 inches (h) and 0.75 inches (b) would produce a dynamic absorber with a natural frequency (Fn) across the “h” dimension of 3600 CPM (60 Hz).

After the calculations have been completed, the amount of the tuning weight (W2) will determine whether or not the design is reasonable. If, for example, the weight for W2 turned out to be 200-300 pounds on a piece of bar stock 12 inches in length, this would be totally unreasonable. In fact, the steel weight (W2) of 0.785 pounds, from the example outlined above, is rather small when it must include the weight of all bolts, washers and nuts needed to secure the tuning weight to the dynamic absorber bar. Therefore, it may be advisable in this case to reduce the “L” and/or “h” dimensions or increase the “b” or “h” dimensions to achieve a more realistic weight for W2, perhaps somewhere between 2 and 4 pounds.

If, after performing the calculations, it is found that W2 turns out to be a negative (-) value, this means that the bar selected for the dynamic absorber already has a natural frequency less than the desired natural frequency (Fn). If this is the case, the length of the bar must be reduced or the “b” and/or “h” dimension(s) must be increased to increase the resonate frequency of the absorber bar.
More trial-and-error may be needed

From the example outlined above, it should be apparent that some trial-and-error attempts may be necessary in order to derive reasonable or realistic dimensions and weights for a dynamic absorber. The goal, of course, is to create a dynamic absorber which is capable of generating a vibratory force that is equal in intensity but 180 degrees out-of-phase with the original force exciting the resonate vibration. Unfortunately, the intensity of the original exciting force frequency is rarely known. Even after reasonable weights and dimensions have been calculated, and the dynamic absorber installed and fine-tuned or adjusted, further modifications may be required. If the dynamic absorber is too small, it may not have the vibratory energy to counteract the original exciting force.

To illustrate, Figure 6 shows two vertical lube-oil pumps on a large cargo ship. The vertical pumps are located in the stern of the ship, behind a large diesel engine used for main propulsion. From initial installation, both pumps exhibited extremely high levels of vibration. Initial attempts to reduce pump vibration through extensive stiffening provided little or no improvement. The pumps are 120 horsepower, and operate at 1200 RPM (20 RPS). The pumps are positive displacement, with a discharge pulsation of 2 x RPM or 2400 CPM (40 HZ).

The pumps normally operate one-at-a-time, with one serving as a backup or standby. However, high levels of vibration were noted regardless of which pump was in operation.

The most significant vibration was 0.72 in/sec (18.3 mm/sec) at the top motor bearings in a direction perpendicular to pump discharge. Vibration amplitudes measured in other directions were well within acceptable levels. The vibration frequency was 2400 CPM (40 HZ), which was equal to the discharge pulsation frequency. Again, the vibration amplitude and frequency characteristics were virtually independent of which unit was operating.

Due to the fact that the vibration was extremely directional in nature, a condition of resonance was suspected. When vibration readings were taken on the discharge piping, the piping was found to be vibrating at an amplitude of 3.0 in/sec (76.2 mm/sec) in a direction perpendicular to pump discharge or parallel to the manifold piping connecting the discharge valves of the two pumps. Vibration amplitudes measured in other directions on the discharge piping were, again, well within acceptable levels.

Due to the extremely high amplitude of directional vibration on the discharge piping, piping resonance was strongly suspected. As a result, both pumps were shut down and a bump or “impact” test performed on the discharge piping to determine the natural frequency of the piping. This bump test revealed that the piping had a natural frequency in a direction perpendicular to pump discharge at 2400 CPM (40 Hz), which was equal to the 2 x PRM inherent discharge pulsation frequency, and verified the problem as resonance.

In this case, the decision was made to attempt to solve the resonate vibration by installing a dynamic absorber in lieu of stiffening the piping or changing piping configuration. Changing piping configuration, perhaps by installing a flexible bellows between the pump and discharge valve, would have been preferable. However, this solution was rejected as being too costly and time consuming.

Stiffening the piping by adding braces between the piping and ship’s hull or superstructure was also rejected as a possible solution. This may have created problems, particularly in rough seas where relative motion of the hull and piping could create undue stresses. For these reasons, the dynamic absorber seemed to be the quickest and most economical solution.

As a first attempt, a single dynamic absorber having cross-sectional dimensions of \( b = 0.75 \) inches and \( h = 0.50 \) inches was used. The absorber had an “L” dimension of approximately 14 inches. When the absorber was installed and fine-tuned to the resonate frequency, the original vibration amplitude of the piping was reduced from the original amplitude of 3.0 in/sec (76.2 mm/sec) to approximately 2.2 in/sec (55.9 mm/sec). This reduction verified that the problem was truly a resonance problem. However, once tuned to the resonate frequency, the dynamic absorber could be seen to “whip” or vibrate at the top a distance of approximately 5 inches (12.7 cm). In other words, the dynamic absorber, although reasonable in design dimensions, was simply too small and incapable of generating a vibratory force sufficient to counter the original exciting force.
A second dynamic absorber with identical dimensions was attached in tandem with the first absorber. Once properly tuned, it also exhibited approximately 5 inches (12.7 cm) of whip, but did further reduce the vibration amplitude to approximately 1.9 in/sec (48.3 mm/sec).

Because of the extremely high level of absorber vibration, both absorbers failed due to fatigue after approximately 1 to 2 hours after installation.

Based on these results, a second dynamic absorber was designed, larger dimensionally than the first, to produce or generate a larger counteracting vibratory force. The cross-sectional dimensions were increased from \( b = 0.75 \) inches and \( h = 0.50 \) inches to \( b = 1.5 \) inches and \( h = 0.75 \) inches to increase the amount of weight \( W_2 \) needed and, thus, the vibratory force generated by the dynamic absorber.

This larger dynamic absorber, once installed and fine-tuned, reduced the discharge piping vibration from the original 3.0 in/sec (76.2 mm/sec) to approximately 1.6 in/sec (40.6 mm/sec). Although this was a considerable reduction, the new absorber could still be observed to vibrate or “whip” approximately 0.25 inches. For an absorber with an “L” dimension of only 14 inches, a vibration or “whip” in excess of 0.25 inches would still be considered excessive and indicative of short fatigue life. In other words, even this larger dynamic absorber design was insufficient to produce a vibratory force necessary to counteract the original exciting force of hydraulic pressure pulsations.

As a result, a second, identical dynamic absorber with the larger dimensions was attached to the opposite end of the discharge manifold, as illustrated in Figure 6, to work in parallel with the first absorber. The two absorbers, working in parallel, reduced the piping vibration from 3.0 in/sec (76.2 mm/sec) to approximately 0.6 in/sec (15.2 mm/sec). This resulted in a reduction of pump vibration from 0.72 in/sec (18.3 mm/sec) to less than 0.2 in/sec (5.1 mm/sec). Although this vibration level was considered acceptable by shipboard personnel, the vibration level could possibly have been reduced even more by attaching additional absorbers to work in parallel with existing ones, or by further increasing the dimensions of the absorbers to further increase the counteracting forces generated by the absorber resonate vibration.

![Figure 6. Two dynamic absorbers working in parallel were required To generate a force sufficient to counteract the original exciting force from pump hydraulic pressure pulsations.](image-url)
Another case history

Figure 7 illustrates another interesting example of dynamic absorber used to control a floor resonance problem in a newly constructed food processing plant. Among the machines installed on the equipment floor was a large vibrating conveyor, operating at approximately 900 oscillations-per-minute. With the vibrating conveyor in operation, the equipment floor exhibited extremely high amplitudes of vibration of nearly 20 mils peak-to-peak (508 micrometers peak-to-peak), which could ultimately cause structural failure. An operating deflection shape analysis of the floor was performed, verifying that the floor was, in fact resonate at the 900 CPM oscillating frequency.

At the time, it was estimated that stiffening the floor by adding additional “T” beams would cost approximately $50,000 US dollars, whereas the cost to design and install dynamic absorbers (including materials) would be approximately $4,800 US dollars. For this reason, and to avoid downtime costs, the dynamic absorbers were installed, resulting in a significant reduction in equipment floor vibration from approximately 20 mils peak-to-peak (508 micrometers peak-to-peak) to 1.3 mils peak-to-peak (33 micrometers peak-to-peak).

The dynamic absorbers illustrated in Figure 7 were fabricated from carbon steel, and have the following dimensions:

1. L = 40 inches
2. a = 30 inches
3. b = 6 inches
4. h = 2 inches
5. W2 = 293.5 pounds

Figure 7. Dynamic absorbers used to control a resonate floor vibration problem in a food processing plant. (Photo courtesy of Mr. James Dreymala of Dreyco Mechanical Services – Houston, TX)
Conclusions

From the discussion and examples presented, it should be apparent that the dynamic absorber can be an effective tool for not only verifying a resonance problem, but also as a possible solution where separating the natural and exciting force frequencies is not possible or cost-effective.

However; when considering the application of dynamic absorber, it is most important to keep the following points in mind:

1. The bar stock used for the dynamic absorber should be rectangular in cross-section in order to create an absorber that is resonate in a specific direction.

2. The dynamic absorber must be rigidly attached to the resonate machine or structural component and oriented to vibrate in the direction of resonance.

3. When adjusting the position (a) of the absorber tuning weight (W2) along the bar for fine tuning, move the weight in small increments, perhaps 1/16 inch (0.16 cm) at a time, while observing the vibration amplitude. Once the dynamic absorber has been finely tuned to its resonate frequency, the machine or structural vibration will either decrease (if the vibration problem is truly resonance), or it will increase (if the initial vibration problem was not resonance).

4. If the dynamic absorber, once finely tuned, effectively reduces the vibration, note the amount of vibration or “whip” of the absorber itself. If the absorber whips or vibrates excessively, it may be too small and will likely fail from fatigue. This can be resolved by simply making a dynamic absorber that is dimensionally larger, or by adding additional absorbers to work in parallel.

5. If the dynamic absorber will be used as a permanent solution to a resonance problem, be sure that present as well as future maintenance, engineering and operations personnel are advised of its purpose, so that it will be reinstalled if it is ever necessary to remove it for equipment maintenance.