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Diagnosing Machinery-Induced Vibrations of Structures and Foundations

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Abstract

When a structural natural frequency coincides with an operating speed, high vibrations of machinery, structures and foundations can occur and cause machinery or structural damage. This paper presents field measurement techniques to evaluate the effect of structural responses on machinery that can be employed prior to machine startup. Vibration measurement techniques are presented to aid in diagnosing structural problems. The use of a mechanical shaker and modal analysis in evaluating the structural response of several field structures is presented along with the solutions to correct the vibration problems. Vibration measurements (before and after modifications) are presented to show the changes in natural frequency and reduction of vibrational response. In one case, a mechanical shaker was used to define structural response and sensitivity of a structure to unbalanced machinery. Another case describes the successful modification of the foundation of a 10,000 HP axial fan.

Introduction

Vibrations of structures can be destructive, resulting in loss of production and expensive repair. Although the source of the vibrational energy is usually quite obvious (operating machinery), defining modifications to minimize the vibrations can be difficult for complex structures. Field measurements of the vibrations, and finite element modeling (FEM) techniques may be required to develop effective bracing or structural additions to detune a resonant frequency from the equipment speed.

Definition of the resonant member(s) is the key to correcting vibration problems. A vibration problem in an existing structure is usually identified by a failed component, or by noticeable vibration amplitudes. High vibration amplitudes simplify the diagnosis by bringing the resonant element to attention. It is a more difficult task to investigate a structure prior to its initial startup to define potential problems, although many problems can be evaluated with FEM programs. However, the use of a mechanical exciter or shaker to simulate the unbalanced forces of the operating machinery on a structure before startup can be an effective method to uncover structural resonances which could be potential problems.

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Vibration Severity

The most severe vibrations on structures which have machinery installed occur when the machine produces energy at a frequency which matches a natural frequency of the structure. This results in a resonant condition, which amplifies the vibrations. The severity of the vibrations depends upon the following factors:

1. Magnitude of exciting force
2. Frequency of exciting force
3. Natural frequencies and mode shapes of the structure
4. Input location of exciting force into the structure
5. Transmissibility of the exciting force to other parts of the structure

From the standpoint of resonance, if these factors combine in the worst relationship, large vibrations will occur. Identification of these factors and changing one or more is the key to solving the problem.

For ease of discussion, vibrations of structures can be separated into two categories: a) low frequency vibrations characteristic of overall structural movement of groups of beams and b) high frequency vibrations of individual beams or trusses. Low frequency vibrations usually involve a large portion of the structure and are more readily identified. Diagnosis of vibrations of structures involving high speed equipment is made difficult because of the many high frequency modes which could be excited.

Rotordynamics is the specialized study of rotating elements and the specific diagnosis of critical speeds has been discussed by other authors [1, (Atkins, et.al 85)] and will not be discussed in this paper.

Diagnostic Methods

Regardless of the complexity of the structure, the first step in the diagnosis procedure is to form a mental image of the potential vibration patterns or mode shapes. For large multi–level structures, walking around or standing at points of high vibrations can help to form a good mental image of the vibration pattern. With smaller structures, such as equipment skids, resonant members can be detected by feeling of the structure.

The potential sources of vibrational energy, their location and frequency must be identified. Identify the components which act as masses by moving as a rigid body and the components which act as springs or the primary flexible elements. Conceptualize the reaction of the structure to the imposed forces. Even though this conceptualized reaction of the structure may not be completely accurate, it should suffice to give guidance for vibration measurements.

Once a general feeling of the severity, frequency range, and patterns of vibrations are in mind, an objective or quantitative analysis would be the next step to obtain engineering data necessary to develop corrective action. Vibration measurements should be made at locations sufficient to describe its three-dimensional vibration mode shape. The mode of interest may be limited to a single deck, a few beams or columns, or the whole structure may be involved.

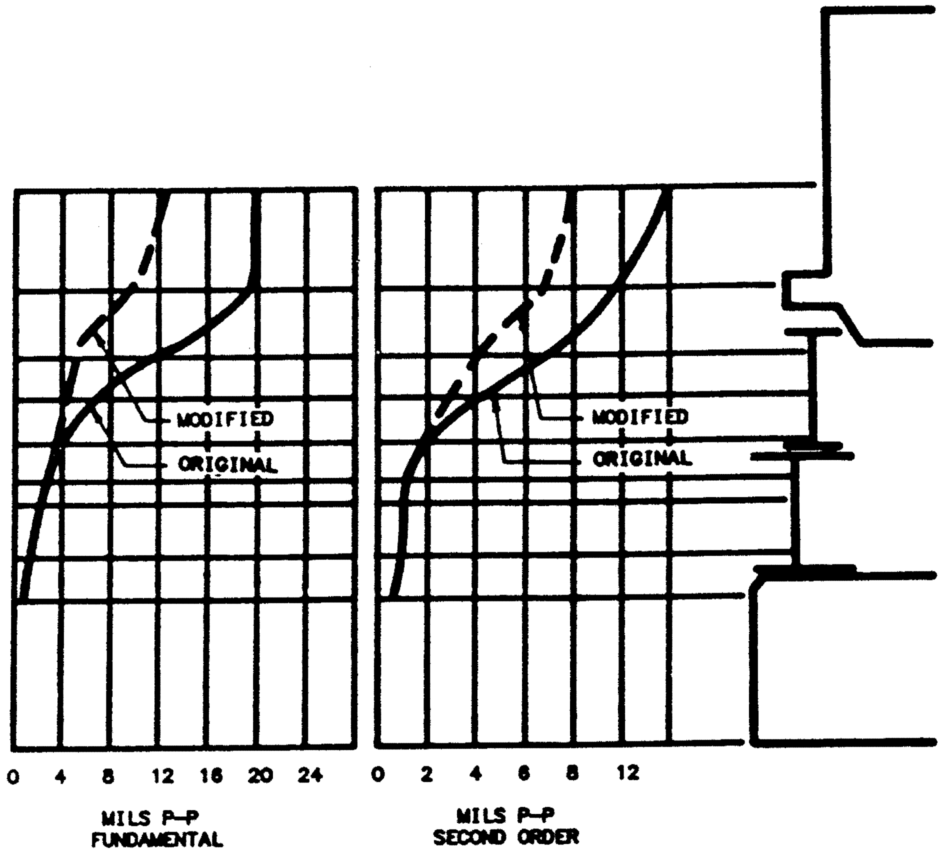

Figure 1: Vibration Profile Showing Skid Flexibility

Measurement Techniques

On a more quantitative basis, accelerometers or velocity transducers should be used to determine vibrational frequencies and amplitudes during normal operation of machinery. Two accelerometers and a dual channel oscilloscope displaying the complex vibration waveforms can be used to define the phase relationship between the signals. By keeping one accelerometer stationary as a reference, subsequent moves of the other accelerometer to measure amplitudes at various points on the structure can define the mode shape at selected frequencies. This method requires that the machinery speed remain constant while the measurements are being made. The speed should be set at the resonant frequency or mode to be identified.

As an example, the vibrational mode shape of the support skid of a reciprocating engine [2, (Wachel, et.al 85)] is shown in Figure 1. The mode shape at operating speed was obtained by positioning an accelerometer at various elevations and plotting the absolute motion, being sure that the phase relationship remained constant. Data was taken at the fundamental and second order of engine speed.

The vibration vectors are connected in a solid line representing the extreme motion along the side of the engine and skid. If the entire engine and foundation were rocking as a rigid body, the line would be straight. In this case the skid beams supporting the engine were inadequate in strength. The bend in the line indicated an area of greater relative flexibility. The skid was strengthened by adding braces and gussets. The dashed line shows the improvement in strength with the modifications.

As a general technique, this type of measurement should be taken at the resonant frequencies near the operating speed to define the elements which control the resonances. Measurements can also be made by using a reference accelerometer on the structure to trigger the data loading sequence of a real time analyzer to enable more accurate amplitude/phase data to be taken. The reference signal may be from a key phase signal which relates the vibrational maximum to the actual shaft position.

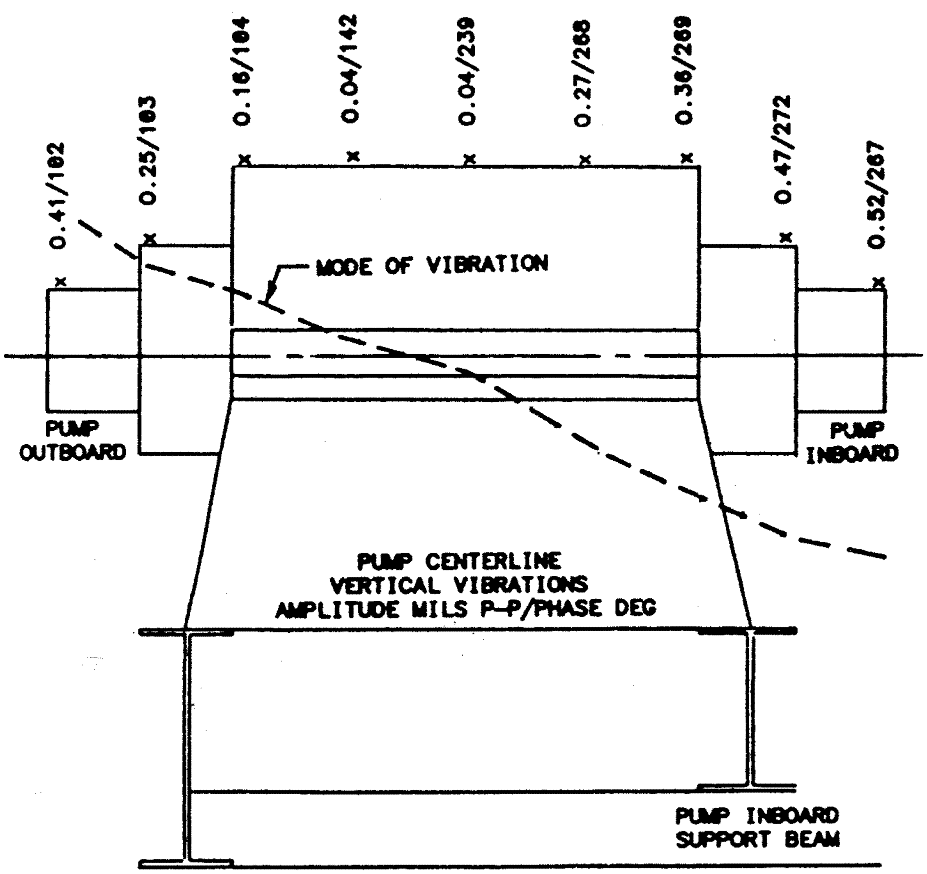

Figure 2: Vibrational Mode Shape of Feedwater Pump

Amplitude/phase data for a feedwater pump (Figure 2) are tabulated directly on the figure to aid in interpreting the mode shape. The vertical vibrations at the pump centerline are plotted as a dashed line. Although the phase difference from the outboard and inboard ends was 165 degrees (not 180 degrees), the mode was characterized by a rigid-body rocking motion of the pump case. The vertical vibrations on the pump inboard are plotted as a dashed line (Figure 3) and indicate that the horizontal support beam is one of the main flexible elements.

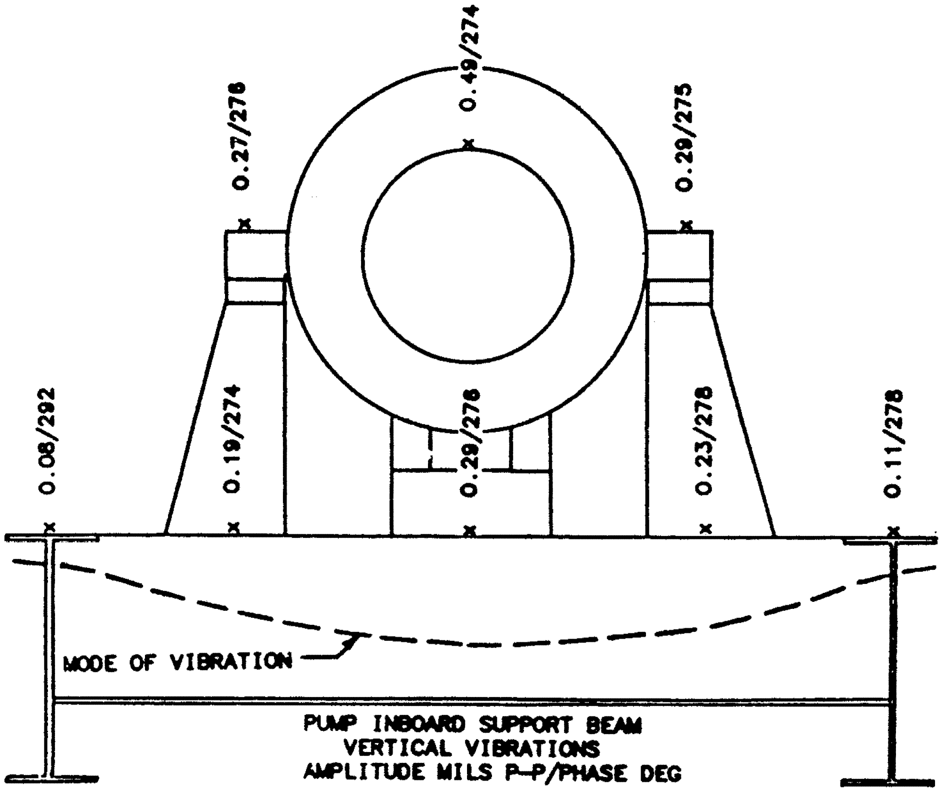

Figure 3: Flexibility of Pump Support Beam

Variable Speed Shaker

The resonant frequencies of a structure can be defined by changing the speed of the equipment causing the shaking until the vibrations are maximum. For structures without variable speed equipment, mechanical vibrators as shown in Figure 4 can provide a variable frequency force.

Mechanical shakers are convenient to use for structural investigations because they can be operated with a minimum of auxiliary equipment or processes. Small compact shakers can be mounted almost anywhere and require little energy since the primary purpose is to produce the shaking force. Various types of shakers are available. Shakers A and B are air driven vibrators which can be operated with typical plant air supplies (100 psi, 699 kPa; 100 cfm, 2.8 m^3/min). Air flow causes a spherical ball to whirl around a circular race to create the unbalanced force which is a rotating force vector in the plane of the race. Air driven vibrators are made by several manufacturers and are sized for different speed and force ranges. Shaker C has an eccentric weight with adjustable unbalance and can be driven by an air motor or electric motor. It also produces a rotating force vector. These shakers are used to simulate the shaking forces of rotating equipment.

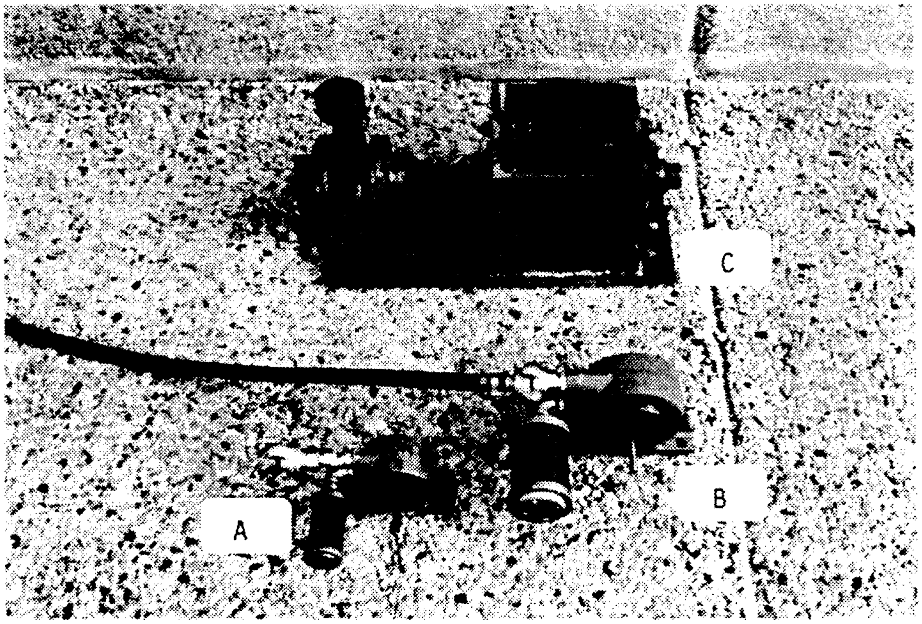

Figure 4: Mechanical Vibrators

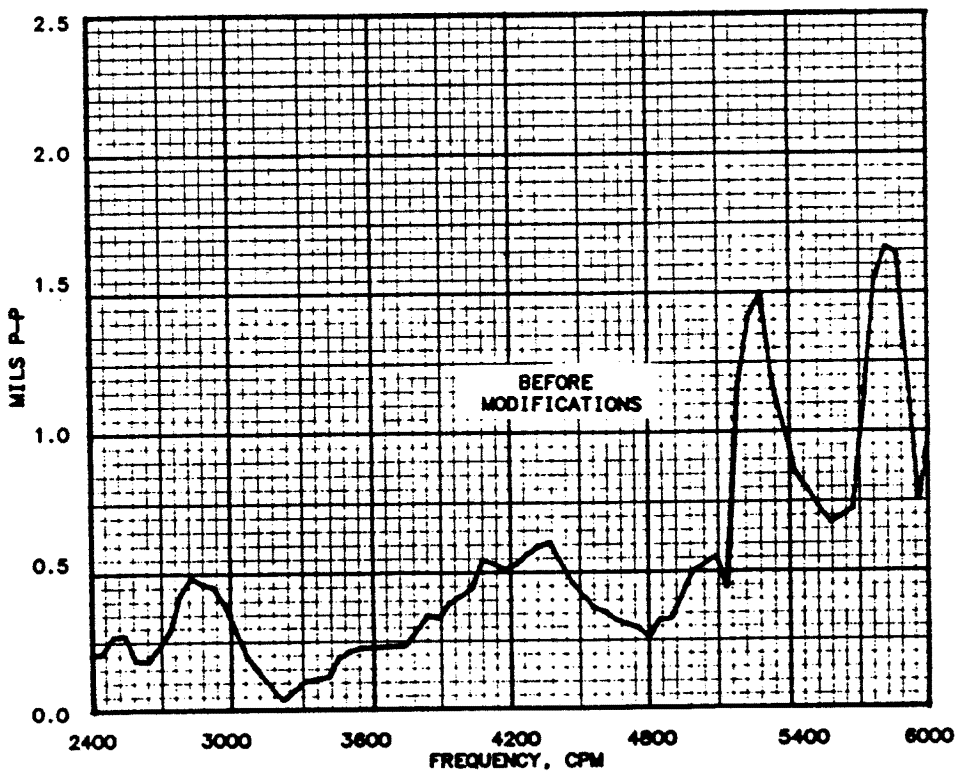

Figure 5: Bearing Housing Response of Steam Turbine

Shaker B was used to define the resonant frequencies and mode shapes of the bearing housing of a steam turbine. The response curve of Figure 5 was produced by changing the shaker speed from 2400 to 6000 cpm. The major resonant frequencies at 5250 and 5750 cpm were horizontal modes of the bearing housing which is supported by the exhaust duct. The normal operating range of the turbine is 3000 rpm to 5800 rpm. Bracing was added inside the exhaust duct to stiffen the support of the bearing and the shaker test was repeated. The response curve in Figure 6 shows that the resonant frequencies were moved above 6000 cpm.

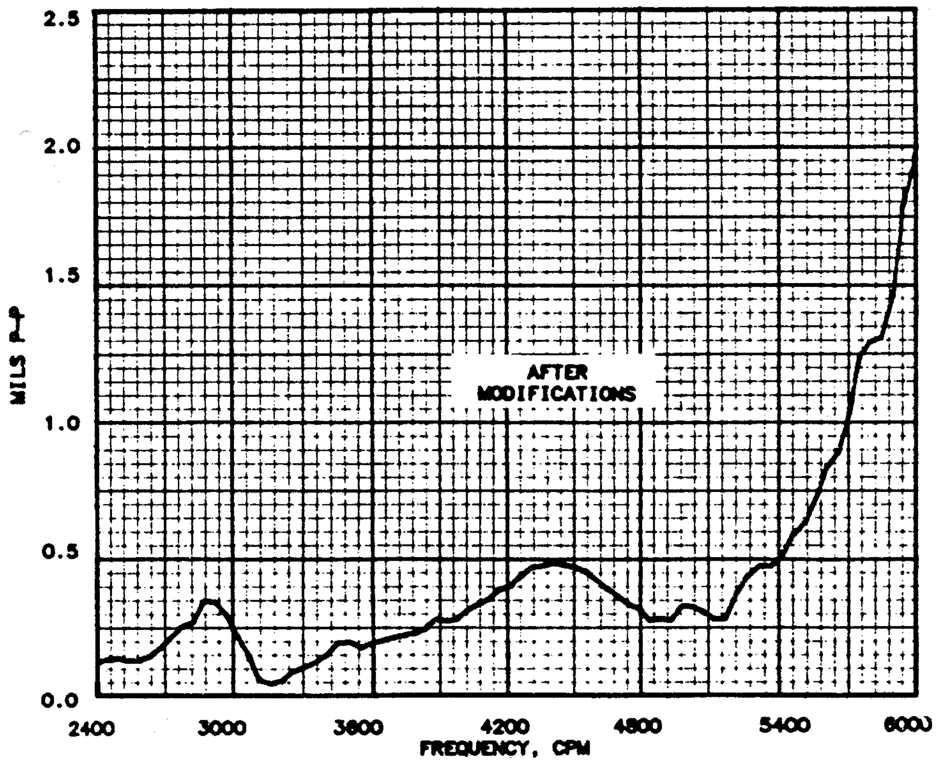

Figure 6: Bearing Housing Response After Modifications

Impact Tests

An impact test is a simple technique which can be used to excite resonances of flexible substructures. Generally, the lower modes of vibration will be excited. The mode of vibration is strongly influenced by the:

1. direction of impact
2. interface material
3. impact velocity
4. contact time
5. impact location

For best results, the impact velocity should not be greatly different from the vibrational velocity of the vibrating member. For example the lowest beam mode of a piping span can be excited with a rubber mallet by applying the impact with a forceful, medium-speed swing. However, a sharp rap with a steel-faced hammer could produce a higher mode of beam vibration, or possibly a pipe shell wall resonance. These higher energy modes will be quickly damped and difficult to identify. Using modal analysis techniques, the input spectrum can be evaluated and the impact can be adjusted to obtain the best results for the structure ~~being tested~~.

In nonsymmetric members or structures, a resonant mode has a preferred direction of motion. Impact testing should be approached on an experimental basis by trying variations of impact velocity, direction, etc. Numerous frequencies may occur which make the mode shapes difficult to identify. The objective of the testing should be to identify structural resonant frequencies which occur within the machinery operating frequency range. In modal analysis testing, the accelerometer remains at one location and an impact hammer is used to excite the structure at selected locations.

The response frequency spectrum in Figure 7 was obtained by shocking a speed governor assembly on a steam turbine. The mode at 8000 cpm was a cantilever mode which resulted in high vibrations as the turbine speed approached 8000 rpm.

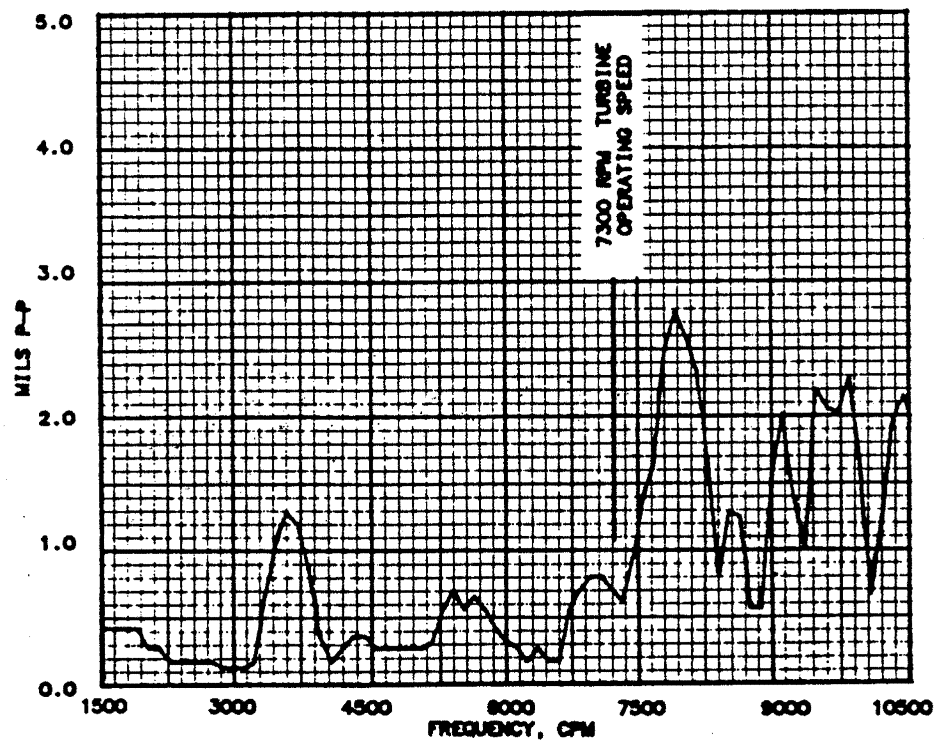

Figure 7: Response Spectrum from Impact of a Speed Governor

Development of Modifications

The first step to developing corrective modifications is to identify the resonant components of the structure. The most effective solutions are generally to strengthen the flexible member (or primary spring) which controls the natural frequency.

Complex steel structures may literally have hundreds of natural frequencies and associated modes, which can complicate the process of locating the flexible elements which control the resonant frequency. Only the resonances near or in the operating speed range need to be addressed. However, structural modifications may move new modes of vibration into the speed range in addition to moving the targeted mode away.

Modifying critical support structures of rotating equipment to move resonances out of the operating speed range requires detailed measurements of the stiffness characteristics. The effective stiffness of the composite structure should be known as a guide to determine how much additional stiffness would be required to move resonant frequencies; however, measuring the static stiffness of large structures can be an difficult task. Many times a more practical approach is to model the structure with a FEM program, which can also be used to evaluate possible modifications.

The accuracy of the FEM simulation can be improved by comparing the dynamic response (frequency, amplitudes and mode shapes) with the field results and making adjustments to the analytical model to compensate for simplifying assumptions or inadequate modeling. When the analysis is capable of modeling the actual response, it can be used to determine the effectiveness of additional braces, increased beam sections, or other structural modifications to strengthen the structure.

Two examples of structural resonances are discussed to demonstrate the procedures used to develop viable modifications to control the vibrational energy.

Case History — Coal Crusher House

High vibration levels resulting in support rod failures of a coal separator screen repeatedly occurred on a coal crusher structure at a power plant. Measurements were made with the crushers and separator screens operating to determine the severity of the situation. With the equipment off, a shaker was used to excite the resonances near the operating speed.

The major equipment in the coal crusher house consists of the coal conveyors, the separator screens and the secondary crusher. The separator screen is isolated from the structure by coil springs (4) at each corner with a spring rate of 1460 lbs/in (255 N/mm). The nature of the coal separator is to generate a shaking force with a motor-driven eccentric weight operating at 780 rpm, to vibrate the screen so the coal will fall through the screen and onto the conveyor.

The initial survey identified the highest vibrations at the beams supporting the separator screen. Vibrations were predominantly 780 cpm (the screen operating speed); however, some small vibration components occurred near 900 cpm which is the crusher drive motor speed. The beam supporting the west end of the screens had vibration levels of 13 to 17 mils (0.33 to 0.43 mm) peak–peak; however, the vibrations on the east end supports were 4 to 7 mils (0.10 to 0.18 mm) peak–peak. For steel structures of this type, vibrations greater than 10 mils (0.25 mm) peak–peak are considered unacceptable.

The simplest solution would be to reduce the speed of the shaft with the eccentric weight which would require a change in drive sheave diameter; however, the coal capacity would be adversely affected. A detailed investigation using the shaker was made to aid in developing structural modifications to increase the resonant frequencies.

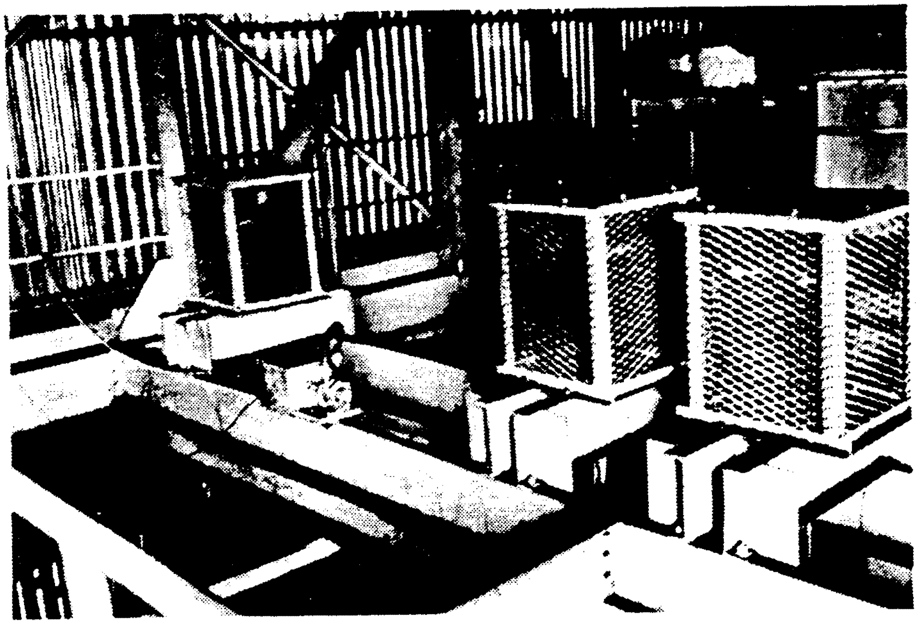

Figure 8: Mechanical Shaker to Excite Structural Response

The shaker was mounted on a major structural beam near the attachment of the screen hanger as shown in Figure 8. With all the auxiliary equipment off, the shaker was operated at speeds from 500 cpm to 1200 cpm to determine the natural frequencies and response of the structure. Major resonances were found at 800 cpm, 950 cpm, and 1050 cpm as shown in Figure 9. An overall vibrational mode of the building structure did not occur; however, some structural modifications could help to reduce the maximum vibrations of the support beam. The support beam was stiffened by adding a beam and braces under the original beam as shown in Figure 10 to increase the section modulus and shorten the span.

The isolation properties of the supporting spring hanger could be optimized to attain further reduction of the transmitted forces; however, special custom designed springs would be required which would be expensive.

This example demonstrates that the simplest and direct solutions may not be acceptable. A slower speed would avoid the resonance but would affect capacity. Softer springs would reduce the transmitted forces and vibrations but were economically unacceptable. Strengthening the support beam was the other choice and it could be accomplished quickly at a lower cost.

Case History — Axial Induced Draft Fan

Induced draft boiler fans (900 rpm; 10,000 HP, 7457 kw) at an electric generating plant experienced high vibrations on the fan housing and foundation. The concrete grout

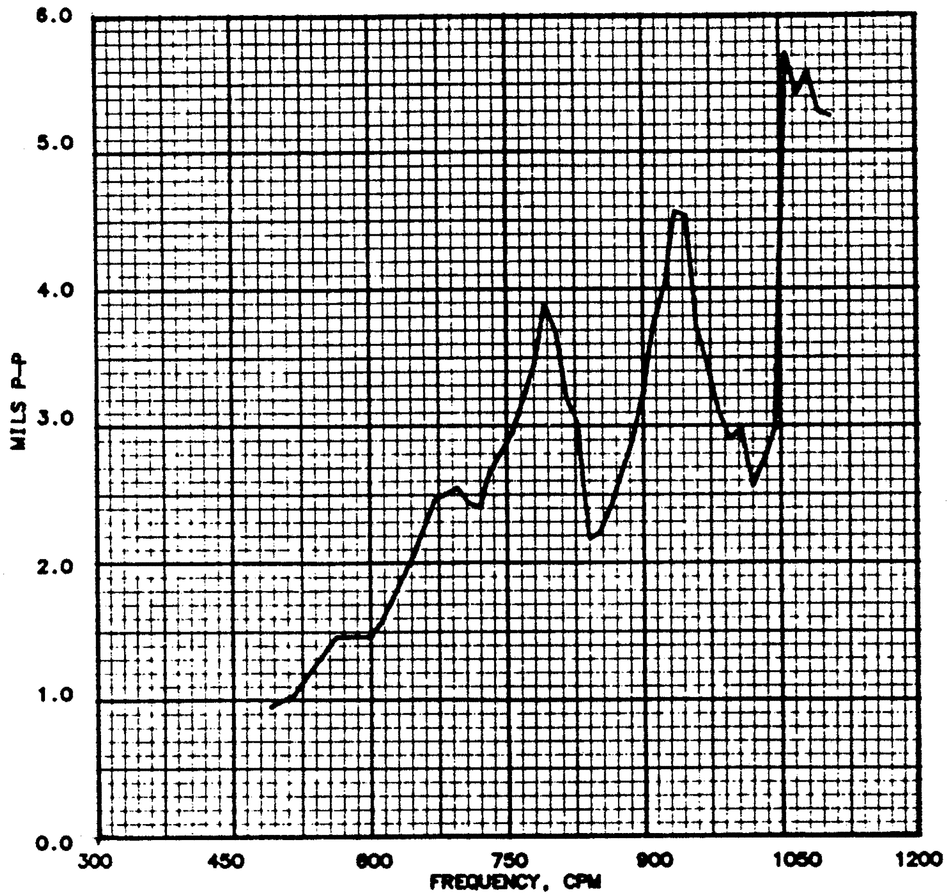

Figure 9: Structural Response Excited by Shaker

cracked on both fans and the units were regrouted during a plant outage. The ID fans were balanced after the maintenance outage and the vibrations were reduced to less than 1 mil (0.03 mm) peak-peak. After a period of operation, the vibrations increased. It was suspected that the increase in vibration level was related to differential motion between the individual concrete foundations. The fan foundation consisted of two independent blocks supported by piles. There was no common connection between these blocks and the motor foundation.

After two years of operation, the vibration levels on the B fan increased to approximately 11 mils at the bearing housing. Weld failures occurred in the welded case which allowed the case to flex and the rotor rubbed the second stage discharge seal.

The foundation was tested to determine its vibrational pattern and natural frequencies. The tests were performed by attaching a mechanical shaker on top of the fan case (Figure 11) to simulate the unbalanced shaking forces of the fan during operation. The shaker could be operated well above the fan speed to 2000 rpm to locate resonances both above and below the operating speed. Accelerometers were attached to the foundation and fan to develop the response curve as the shaker speed was varied and to measure the frequencies and amplification factors of the fan/foundation system resonances.

Shaker tests were made with the axial fan at rest on the foundations as originally built. The resonant frequency of the fan was 1050 cpm as shown in Figure 12. The amplification factor (Q) of the fan foundation natural frequency was approximately 5.

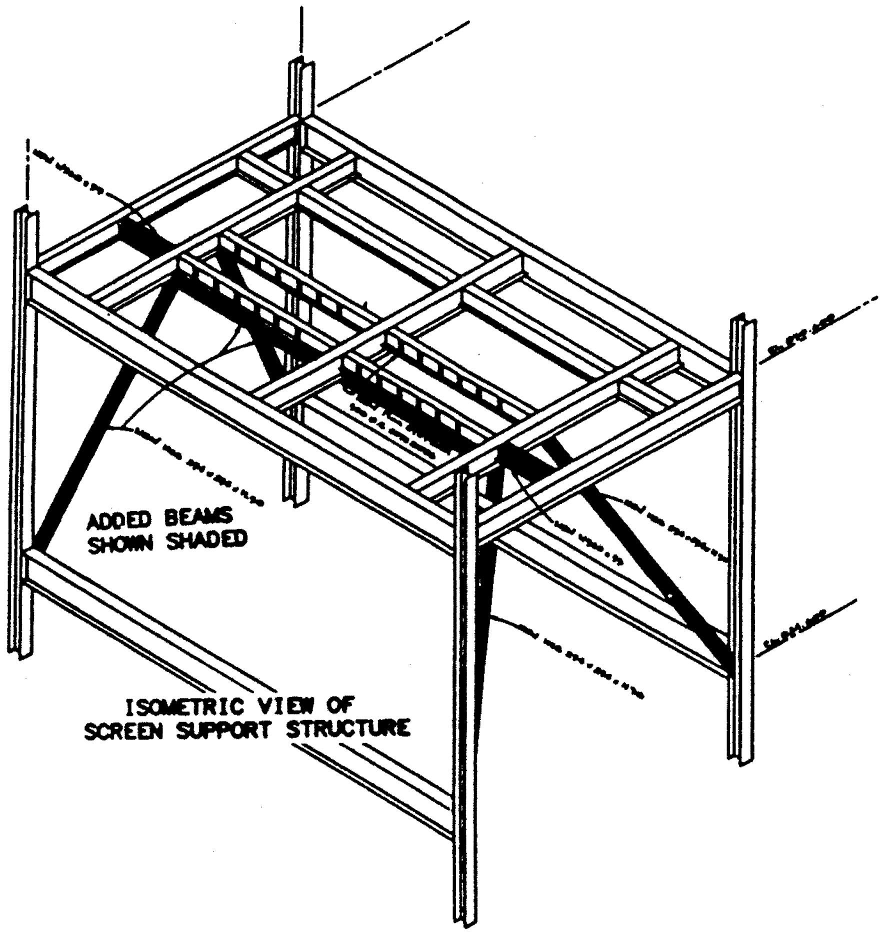

Figure 10: Modifications to Support Structure

Typical amplification factors are 2–3 for mat type foundations.

The vibration mode shape at resonance was obtained by operating the shaker at the resonant frequency and measuring the vibration amplitude at points in the horizontal and vertical directions. The vertical motion of the foundation supporting the fan indicated that the foundation blocks were moving independently.

Modifications to Solve the Problem

The field test results and foundation calculations were used to design modifications to increase the foundation frequency and significantly reduce the vibration of the fan. The objectives of the modifications were to tie the separate fan foundations together to move as a unit and to increase the foundation stiffness by causing the piles to act as guided cantilevers.

For the greatest improvement, the structural modifications had to be strong but comparatively light-weight. For this reason, steel frames were designed to provide strength without the mass of the concrete. Photos of the installed frames are shown in Figure 13. The attachment between the frames and the existing concrete blocks was a very important element. Any flexing in the bolted joint would severely reduce the potential stiffness added by the frames.

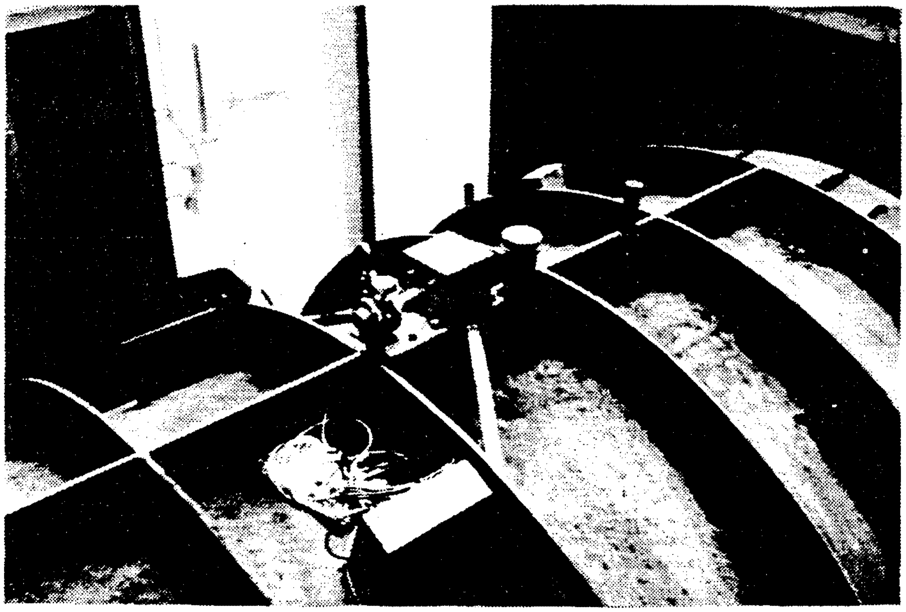

Figure 11: Mechanical Shaker Mounted on Fan Case

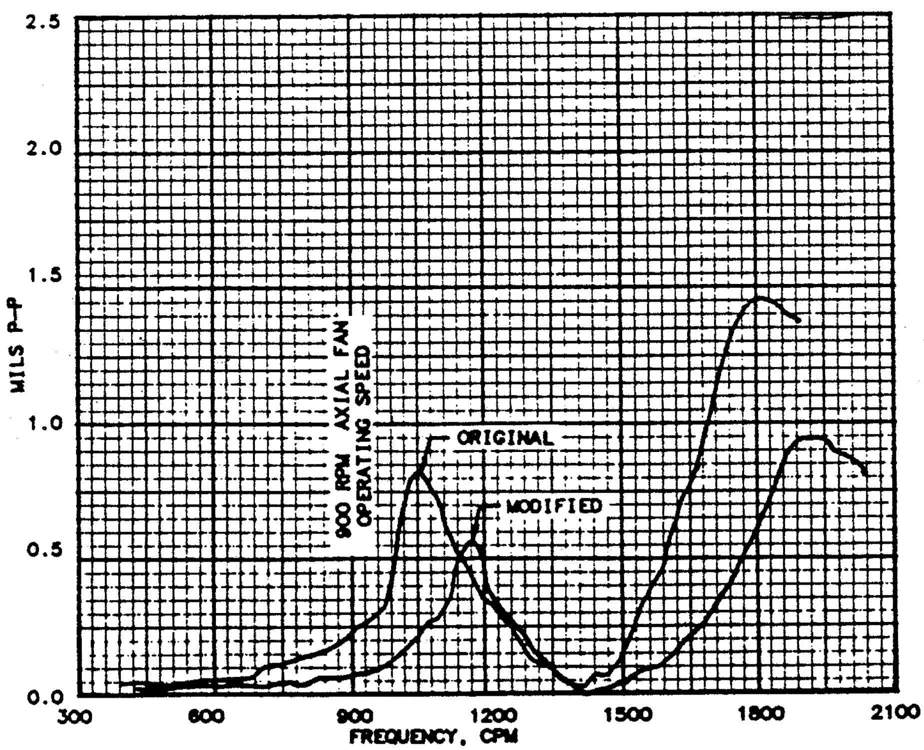

Figure 12: Vibrational Response of Axial Fan

The structural modifications were designed to withstand the maximum rotating unbalance of the fans specified by the manufacturer. The preload in the bolts attaching the frames to the concrete blocks was sufficient to maintain contact between the frames and blocks under operating conditions with a damaged blade. The frames were grouted to the concrete with a high compressive strength non-shrink epoxy grout. The purpose of the bolt tension and grout was to maintain solid contact between the frames and the concrete blocks for extreme conditions of unbalance or blade loss.

After the fan was re-installed on the modified foundation, the shaker test was repeated to evaluate the effect of the steel braces. Identical forces were applied with the shaker installed at the same position, so the vibrational response could be directly compared. The resonant frequency increased from 1050 cpm to 1170 cpm which resulted in significantly lower vibration levels at operating speed. The increased foundation stiffness was evidenced by smaller vibration amplitudes on the foundation and fan. Comparisons at the top of the foundation (Figure 12), indicated that the foundation stiffness was increased by a factor of 3.

The fans have been operating since these modifications were made (Spring, 1983) without vibration problems or abnormal maintenance.

Figure 13: Frames Installed between Foundation Blocks

References

[1] Atkins, K. E., Tison J. D., and Wachel, J. C. "Critical Speed Analysis of an Eight-Stage Centrifugal Pump", *2nd International Pump Symposium* , Texas A & M, April, 1985.

[2] Wachel, J. C., Baldwin, R. M., and Szenasi, F. R., "Dynamic Vibrations of Stationary Engines", ASME Paper, 78–DGP–1, 1978.

Appendix

Conversion of Dimensions

mils	inches	millimeters
1	.001	.025
2	.002	.051
3	.003	.076
4	.004	.102
5	.005	.127
10	.010	.254
15	.015	.381
20	.020	.508