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EFFECTIVE ISOLATION TREATMENT OF VERTICAL PUMPS INCLUDING SEISMIC RESTRAINTS

by

Lyn Greenhill

President

and

Valerie Lease

Engineering Analyst

DynaTech Engineering, Inc.

Roseville, California

Lyn Greenhill is the President of DynaTech Engineering, Inc., a consulting firm specializing in rotating machinery vibration analysis, testing, mechanical design, and engineering software, located in Roseville, California. Prior to founding DynaTech in 1995, Mr. Greenhill was employed as the Chief Engineer for a consulting company, a Senior Engineering Specialist for a rocket engine manufacturer, and as an engineer and supervisor for a gas turbine company. He has authored 15 technical papers for organizations such as ASME, AIAA, NASA, and the Vibration Institute in the areas of rotordynamics, machinery vibration, rolling element bearing mechanics, fluid film bearings, and turbine blade damping.

Mr. Greenhill holds a B.S. degree (Mechanical Engineering, 1975) and an M.S. degree (Mechanical Engineering, 1975) from Stanford University. In 1995, he was accredited as a registered Professional Mechanical Engineer in the State of California.

Valerie Lease has been an Engineering Analyst at DynaTech Engineering, Inc., in Roseville, California, since 1999. She is responsible for technical support to the company core engineering activities, such as creating and executing analysis models, building 3D solid drawings, reducing vibration test data, coordinating fabrication of hardware, and writing engineering reports.

Ms. Lease earned a B.S. degree (Mechanical Engineering, 1999) from the University of California at Davis and an M.S. degree (Mechanical Engineering, 2004) from California State University at Sacramento. She has also coauthored a technical paper on hydrostatic gas bearings.

ABSTRACT

Three vertical pumps at a wastewater transfer station a few miles away from the San Andreas Fault and next to an apartment building were the source of vibration and noise complaints. To attenuate the transmitted vibration, a typical isolation system was installed without a significant amount of analysis, and consisted of unrestrained pump base mount pads and flexible couplings on the suction and discharge pipes. Initial testing showed that while the base mount pads worked well, the lack of a positive seismic restraint made those devices unsuitable for use in an earthquake zone. The flexible couplings, however, actually increased vibration transmitted into the piping because of excessive stiffness.

Based on the initial test results and site observations, a new isolation system was required. Although isolation theory is relatively simple, this paper describes how a detailed three-dimensional analytical treatment was required to properly determine the properties of the isolators, including the influence of adequate seismic restraints. Test results are included showing the redesigned system effectively isolates the pumps.

INTRODUCTION

The Main Street Pump Station of the Sausalito-Marin City Sanitation District in Sausalito, California has three vertical pumps that transfer waste water from a wet well to a force main connected to the water treatment plant. All three of the pumps were the source of an ongoing transmitted vibration problem with an adjacent apartment building.

An initial set of vibration tests conducted on all three pumps showed that, in terms of isolation, the pump base mount pads worked well, but the flexible couplings at the inlet and discharge were too stiff and allowed a large amount of transmitted vibration. Another problem observed with the isolation system was that the base mount pads were not positively restrained and are therefore unsuitable for use in an earthquake zone. Sausalito is a few miles away from the San Andreas Fault.

This paper describes the process used to design an effective isolation system for the pumps. A primary design constraint was to replace the existing isolation devices with more effective hardware. Included is a description of the detailed analytical treatment of the pumps mounted on isolators and a discussion of the results. Two of the pumps are identical, and the third is smaller, requiring two completely different designs. For reference, the small pump is referred as Pump 1, and the two larger are labeled Pumps 2 and 3. After the new isolation hardware was installed and adjusted, vibration tests showed the redesigned system was effectively isolating the pumps and significantly attenuating any transmitted vibration.

DESCRIPTION

All three pumps are shaft coupled, center inlet, side discharge, single stage designs. A typical pump set is shown in Figure 1. An electric motor drives each pump through a shaft and set of couplings, with the shafts roughly 6 feet long. The couplings are U-joints, which accommodate some amount of misalignment and also isolate the pumps from the motors. The pumps are variable speed, with a range from 1470 to 1800 rpm for the small pump and 890 to 1200 rpm for two larger machines. The overall length of a typical pump assembly, from the top of the motor to the bottom of the pump housing, is approximately 10 feet.

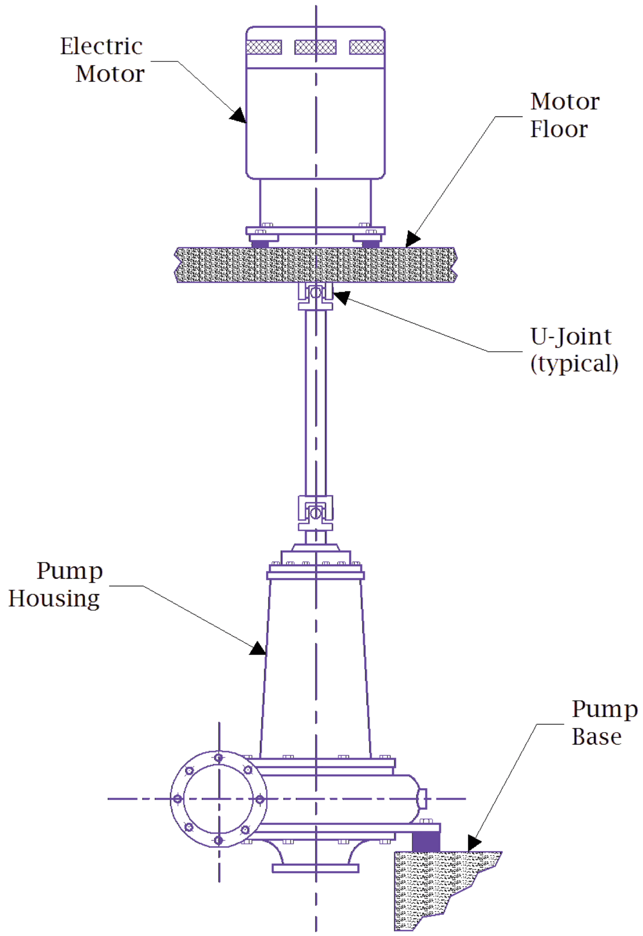

Figure 1. Typical Layout of Vertical Pumps at Pump Station.

Both the motor and pump are mounted on vibration isolation bushings, as shown in Figure 1. In addition, both the inlet and discharge connections between the pumps and associated piping are made with flexible couplings. This treatment was incorporated in an effort to reduce the transmitted vibration from the pumps to the surrounding structure of the pump station.

ORIGINAL CONFIGURATION VIBRATION DATA

Testing conducted on the original configuration showed that the flexible couplings were not as effective as the pump pads for attenuating vibration. The "transfer function" displayed in Figure 2, which is a plot of vibration measured on the inboard (pump side) and outboard (pipe side) of the flexible coupling, shows a modest 38 percent decrease in overall vibration, from 0.262 ips-rms in the upper plot (inboard side) to 0.162 ips-rms in the lower (outboard side). Although the amplitude is attenuated by 38 percent, the predominate 1/rev signal is transmitted across the coupling, which drives the discharge piping to create vibration in the force main.

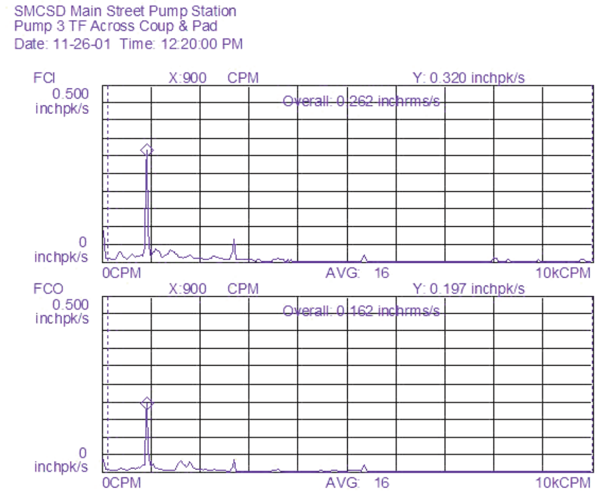

Figure 2. Discharge Flexible Coupling Transfer Function on Pump 3.

With the small pump, the transmissibility situation was much worse. As illustrated in Figure 3, the overall and peak vibration across the coupling is actually increasing, with the overall 6 percent higher and the peak 36 percent greater. In fact, the large peak in the outboard spectrum, due to 2/rev excitation from the pump, may even be a resonance of the piping system.

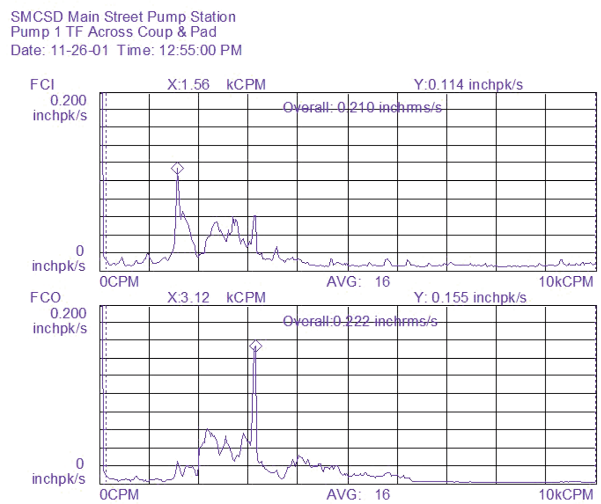

Figure 3. Discharge Flexible Coupling Transfer Function on Pump 1.

Based on this initial vibration survey, and a review of the design information relative to the selection of the couplings, it was concluded that the original flexible couplings were not soft enough to create an effective isolator for pump vibration. No rigorous analysis of the natural frequency situation of the pumps supported on the mount pads and connected by the flexible couplings was performed. While in many cases simply "soft" mounting such machinery produced acceptable results, obviously with these pumps a more concerted effort was needed to effectively isolate the station from the machinery vibration.

STRUCTURAL ANALYSIS OF ORIGINAL CONFIGURATION

The fundamental objective in isolation system design is to lower predominate system natural frequencies below primary excitation frequencies. The basic concepts behind this vibration treatment are discussed in most textbooks, such as Thomson (1972). In order to determine the natural frequencies of the pumps mounted on isolators, a finite element structural analysis is required to properly characterize the modes from all six degrees-of-freedom. In a multipoint constraint system such as these pumps, the combined interaction of all the connections is difficult to adequately address using hand calculations. Harris and Piersol (2002) discuss multiple degree-of-freedom isolation design, and to arrive at reasonable solutions, a great deal of simplification is required, such as equal isolator spring rates and symmetric structures. With these pumps, none of the simplifications are possible.

The analysis was initially conducted on the two larger pumps, since these run at lower speeds and were thought to be the primary source of vibration into the pump station. In the model, all significant mechanical elements of the pumps were represented, including the housings, couplings, inlet elbow, and baseplate. Due to the lack of symmetry in the overall structure, a full three-dimensional representation was required. The finite element mesh constructed to model the pump is shown in Figure 4.

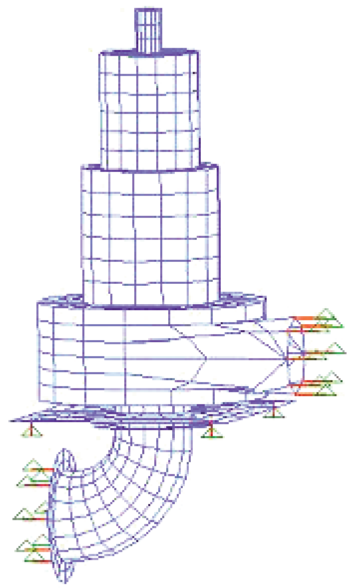

Figure 4. Structural Analysis Finite Element Model for the Large Pumps.

The model primarily consists of solid elements, with the elbow created from plates and the flexible couplings and pads represented by springs connected to ground. Density of the elements was adjusted to obtain a proper inertia match with the actual hardware. The stiffness of the springs was set based on manufacturer data for the particular isolation device, which for the couplings was 7584 to 11,020 lbf/in (depending on direction), and for the pads 500 lbf/in. No attempt was made to verify these values, or the equivalent with the replacement hardware. It is recommended that static stiffness tests be performed if increased accuracy in frequency predictions is required.

Using the finite element model as shown in Figure 4, the natural frequencies of the two large pumps are listed in Table 1 for the first six modes of interest.

Table 1. Natural Frequencies, Original Configuration, Large Pumps.

Mode Shape	Frequency (rpm)	Frequency (Hz)
Translation Z	200	3.3
Translation X	371	6.2
Translation Y	532	8.9
Rotational X	843	14.1
Rotational Z	939	15.6
Rotational Y	1077	18.0

Results in the table show two rigid body modes within the speed range of the system (890 to 1200 rpm). These two modes, which are rotational displacements in the X and Y direction, are at 939 rpm and 1077 rpm, respectively. For reference, the first six mode shapes listed in Table 1 are displayed in Figure 5 through Figure 7, and show significant flexure of the expansion joints as well as the pump pads. There is essentially no localized deformation of the actual pump housing, because the mounts are much softer than the pump structure, and thus the reason why these frequencies are referred to as rigid body modes.

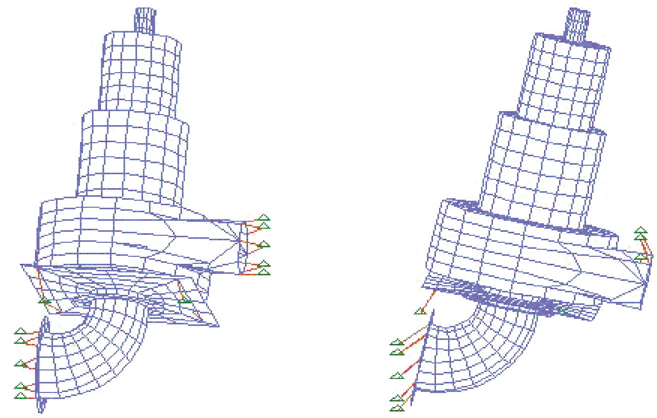

Figure 5. Rigid Body Modes 1 and 2 from Finite Element Analysis of Large Pumps.

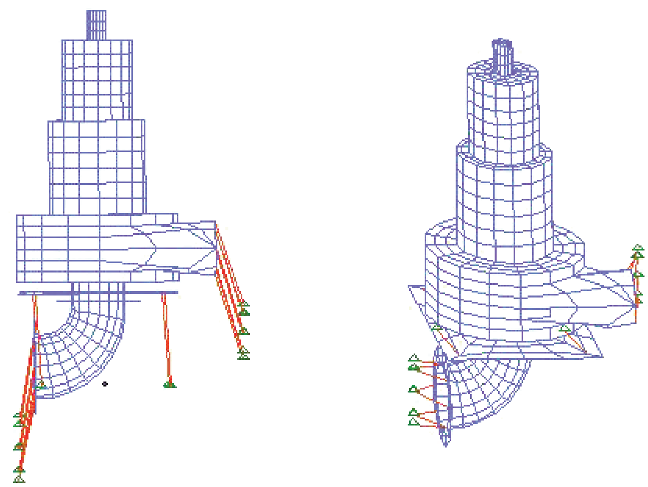

Figure 6. Rigid Body Modes 3 and 4 from Finite Element Analysis of Large Pumps.

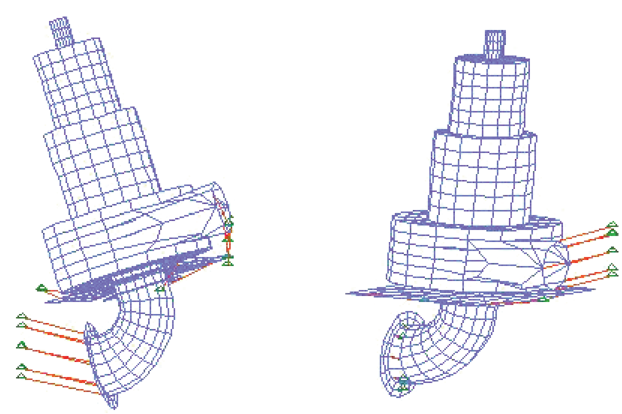

Figure 7. Rigid Body Modes 5 and 6 from Finite Element Analysis of Large Pumps.

ISOLATION SYSTEM REDESIGN

In order to create an effective isolation system, the rigid body modes listed in Table 1 must ideally be located well below the minimum operating speed of the pumps. The exact separation must be determined through analysis, by changing the stiffness of the isolation system components and examining the resulting response. The frequency of the higher modes is being controlled by stiffness of the flexible couplings, which created a design problem to find suitable hardware that would be soft enough to lower the modes to replace the existing couplings. Another design constraint was to replace the baseplate mount pads with something as soft as the current pneumatic isolators, without requiring significant changes to the mounting structure. The results of this search are discussed in the next two subsections.

Pump Base Pads

The original mount pads, one of which is shown in Figure 8, provide an acceptably soft support. However, the configuration is not adequate in a seismically active location, since no positive retention is incorporated. In addition, these pads require continual maintenance to ensure a constant air pressure.

Figure 8. Original Pneumatic Baseplate Isolator.

The replacement of these pads with a seismically restrained vibration isolator is a much more acceptable choice. Various sizes of these isolators are available to match the spring rate of the existing pneumatic pads or achieve any other required stiffness. The selected replacement is shown in Figure 9. Height of this support can be adjusted by the locking nuts, which also control how much static load each isolator is carrying. Seismic restraint is provided by the rectangular housing surrounding the spring.

Figure 9. Replacement Spring Isolator and Wide Arch Flexible Coupling.

Inlet and Discharge Flexible Couplings

The original flexible couplings are shown in Figure 10. The couplings have a single short arch in the center, which was filled to prevent flow disruption. This configuration makes the couplings very stiff, approximately 8472 lbf/in axially. Although the couplings have some damping, such large stiffness values cause the natural frequencies of the system to fall into the running speed range, providing an ideal means to transmit vibration from the pump to the piping and walls.

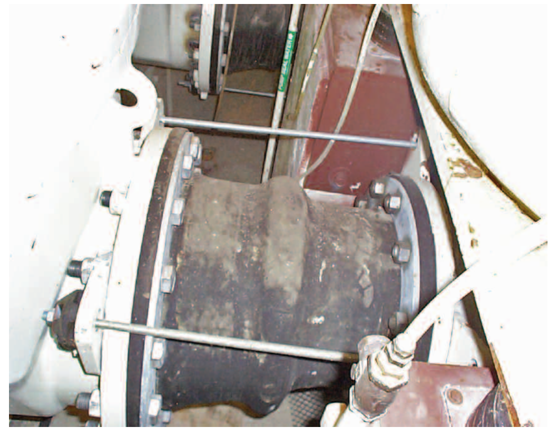

Figure 10. Picture of Original Flexible Coupling.

Various types of couplings were considered in order to find a suitable alternative. In general, most of the rubber designs were no softer than what was originally used. Metal bellow configurations were considered, and are soft enough, although the cost is significant.

As a result of discussing the stiffness requirements with coupling manufacturers, an alternative was found by using a wide arch design between the two flanges, which allows the coupling to have much more flexibility and a correspondingly lower spring rate. This coupling is shown in Figure 10. The arch, because of its wider width, does not require filling to prevent entrapment problems, which further decreases the stiffness.

The wide arch coupling, in an equivalent size to what was originally used, has an axial stiffness of approximately 1500 lbf/in, an 82 percent reduction. Since the exact stiffness was not known, a finite element model was used to determine the spring rate in all directions. An example of the mesh used for these calculations is shown in Figure 11. The coupling is composed of various layers of rubber and reinforcing cord, which were represented in the model by laminate layers. It should be noted that the stiffness of the coupling is not equal in all directions, further complicating the isolation model.

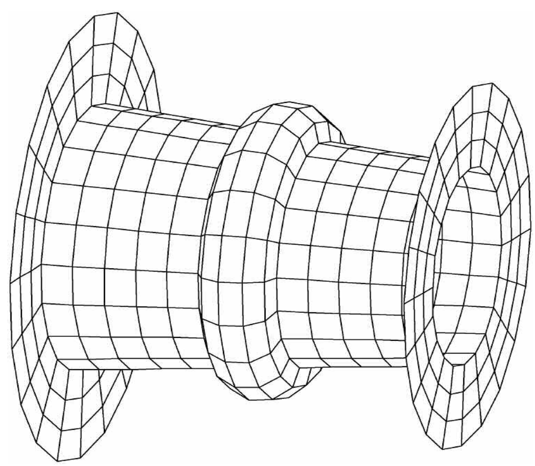

Figure 11. Finite Element Model of Wide Arch Flexible Coupling.

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In order for the flexible couplings to be seismically restrained, control rods are used, which also prevent overextension during operation. These control rods, shown in Figure 10, apply a compressive force across the coupling to counteract the extension due to operating pressure. Typically, the amount of axial expansion is less than a tenth of an inch (0.10 inches). To properly isolate, the preload in the control rod springs must be adjusted so that at operation, there is very little, if any, axial clamp force exerted by the control rods on the coupling. For variable speed pumps, this implies some compromise is needed, and in this application, it was decided to bias the preload such that at minimum speed, the clamp force would be zero. When the pump is not running, the clamp force compresses the coupling and the amount of deformation is limited by a sleeve on the outside of the rod, between the two flange faces. It should be noted that the axial stiffness of the control rod spring must also be included in the finite element model. With the original couplings, shown in Figure 10, the control rods were not spring preloaded so were adjusted to be loose when the pump was not operating, and rubber washers were placed under the nuts in an attempt to isolate transmission of vibration.

Using the stiffness of the wide arch coupling, and including the influence of the control rods, the model was rerun for natural frequencies. Results showed a 47 percent reduction in the sixth mode, placing it 35 percent below minimum running speed (Figure 12).

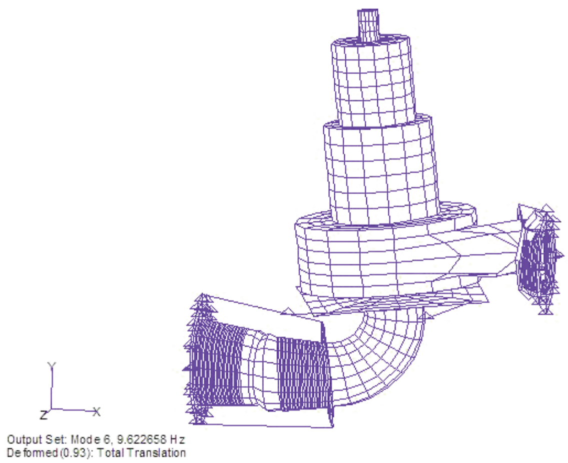

Figure 12. Sixth Mode Shape of Redesigned Isolation System for Large Pumps.

A summary of results for this revised model is given in Table 2. Comparing these results with the original values in Table 1, the revised configuration has all the rigid body frequencies well away from the running speed range. It would be desirable to have greater separation, however, this was not possible without significant hardware changes to the pump inlet and discharge pipes, which does not satisfy the overall design objective. The controlling connection is the discharge flexible coupling, which is relatively short and could not be designed with lower stiffness without significant length increase.

Table 2. Natural Frequencies, Revised Configuration, Large Pumps.

Mode Shape	Frequency (rpm)	Frequency (Hz)
Translation Z	125	2.08
Translation X	193	3.21
Translation Y	291	4.85
Rotational X	447	7.45
Rotational Z	478	7.97
Rotational Y	577	9.62

Transmissibility Analysis

To assess the effectiveness of the proposed isolation system, a transmissibility analysis was performed by applying a speed dependent forcing function to the finite element model, to simulate the lowest vibration harmonic being transmitted from the pumps (unbalance). The forcing function magnitude was set to a value of one, so it was in essence a normalized amplitude, and the point of application was at the center of the model, in the axial plane of the impeller. This position will excite all six natural frequencies, since it is not at the center of mass. Pertinent results from the analysis consist of forces at the springs to ground, where the isolation devices are located. The ratio of forces produced at the spring divided by the applied force provides the measure of isolation effectiveness, or transmissibility. For a properly designed isolation, the transmissibility should be less than one, and preferably less than 20 percent.

Results from the analysis of the original and redesigned configurations are displayed in Figure 13 through Figure 15 for pump baseplate mount pads, the inlet coupling, and the discharge coupling, respectively. There is a dramatic difference in the response between the two designs, as the figures show. For the operating speed range of 890 to 1200 rpm, the transmissibility of the existing configuration is very high, much greater than one except near the upper end of the frequency range. Of course, this is due to the presence of the natural frequencies in the operating range, and it also explains why the pumps are not isolating the vibration from the surrounding structure.

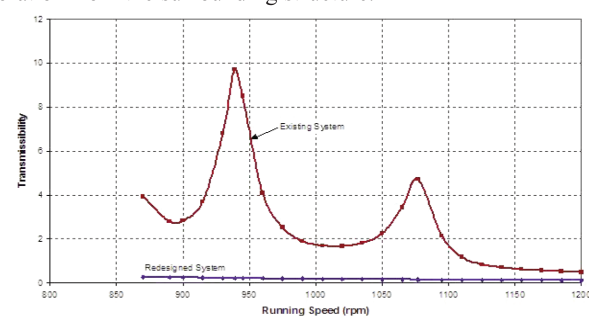

Figure 13. Transmissibility Comparison for Baseplate Mount Pads.

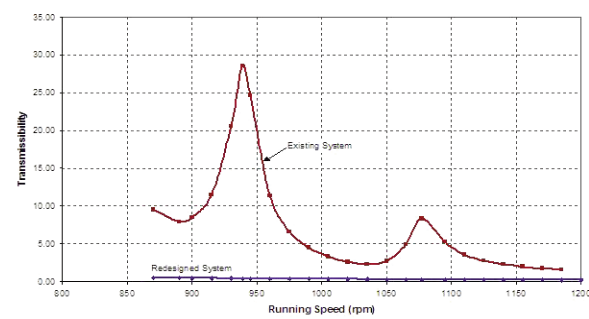

Figure 14. Transmissibility Comparison for Inlet Flexible Coupling.

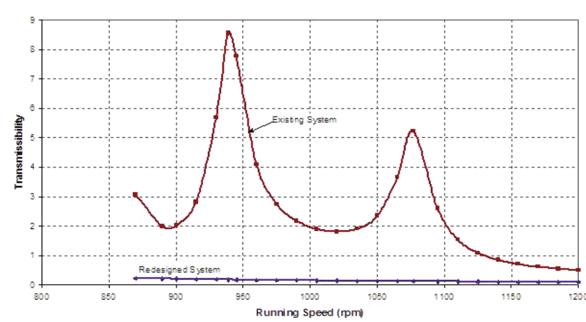

Figure 15. Transmissibility Comparison for Discharge Flexible Coupling.

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In contrast to the original configuration results, the proposed redesign response has very good transmissibility, with a value always less than unity, and closer to the desired 0.10 to 0.20 magnitude. This marked difference is entirely due to keeping the natural frequencies away from the operating range by properly specifying the stiffness of the isolation hardware.

A summary of the isolation effectiveness is provided in Table 3, which lists selected values of average transmissibility at various running speeds. By average, it is meant that the three values of transmissibility for the mount pads, inlet coupling, and discharge coupling are averaged to obtain a single comparison value. The improvement column is simply the ratio of the two average transmissibility values, and indicates the amount of decrease that can be realized with the redesigned system. As the table shows, the effectiveness of the new system is no less than 70 percent, and the improvement ranges from 5.7 to 58 times. At higher speeds, the redesigned system is particularly good, with better than 80 percent effectiveness.

Table 3. Isolation Effectiveness Ratios for Large Pumps.

Running Speed (rpm)	Average Transmissibility		Improvement
	Current System	Proposed System	
890	4.24	0.32	13.3
939	15.62	0.27	57.9
1020	2.03	0.21	9.7
1077	6.09	0.18	33.8
1170	0.98	0.15	6.5
1200	0.80	0.14	5.7

SMALL PUMP ISOLATOR SYSTEM

An analysis of the small pump was performed in order to determine if the same type of isolation system could be used to decrease transmitted vibration. Similar to the process used for the two large pumps, a 3D finite element model of the small pump was created and then analyzed for natural frequencies. A plot of the model used for this analysis is displayed in Figure 16. Note that because of the pump size, the mounting baseplate is at the bottom of the pump, rather than just under the volute as with the big pumps. In fact, this baseplate was actually an inertia mass, designed to increase the overall weight of the pump to make the selection of isolator spring rates easier.

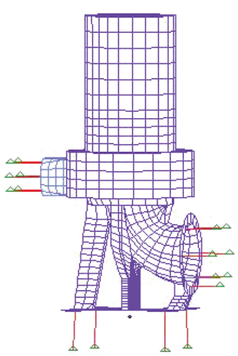

Figure 16. Structural Analysis Finite Element Model for Small Pump.

Results from the natural frequency analysis showed that with similar soft spring mounts and wide arch rubber joints, sized for use with small pump, the closest rigid body mode was predicted at 940 rpm. A plot of this mode is displayed in Figure 17, showing it is primarily a rotational displacement. The frequency of this mode is 36 percent lower than the minimum running speed of 1470 rpm, and is roughly equal to the separation obtained with the larger pumps. Again, the controlling element is the discharge coupling, which is relatively short and cannot be lengthened without major modifications to the piping. As such, the same type of isolation hardware can be used, sized appropriately for this small pump.

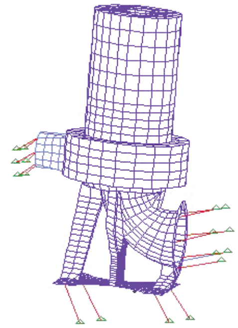

Figure 17. Rigid Body Mode 6 from Finite Element Analysis of Small Pump.

REDESIGNED CONFIGURATION TEST RESULTS

After the completely reengineered vibration isolation system was installed on all three pumps and properly adjusted, a vibration survey was conducted to determine if the rebuilt machines had satisfactory vibration levels according to Hydraulic Institute (HI) standards and to evaluate the isolation system used for overall effectiveness. Testing of the isolation effectiveness was made by taking measurements on either side of the inlet and discharge couplings, and from the pump baseplate to the support pillars. A duplicate set of measurements was made across each coupling and baseplate to ground.

Small Pump Vibration

Due to performance issues with the original pump, the small machine was a complete replacement. The new pump was factory tested over the operating range and had very good vibration results, producing overall amplitudes around 0.10 ips-rms. The original isolation system was replaced with an inertia base, seismically safe support springs, more flexible inlet and discharge couplings, and control rods across the couplings to prevent excessive extension during startup. All of the isolation components were optimized using the 3D finite element natural frequency model of the pump, support system, and connections.

The results of testing the small pump mounted on the isolation system showed low levels of overall vibration throughout the speed range. The vibration at maximum speed (1800 rpm) was slightly lower than at minimum speed (1470 rpm). Normally, the magnitudes of vibration would be higher at maximum speed. However, the influence of the isolation system is affecting the vibratory characteristics by attenuating motion at higher speeds. Review of the transient capture data showed that no resonances or unusual peaks were present in the data over the tested speed range.

The highest overall amplitude of 0.175 ips-rms was produced when operating at minimum speed. When compared to the HI 9.6.4 vibration limit, the peak vibration is 56 percent of the 0.31 ips-rms limit. The vibration is slightly higher than what was measured at the factory test, which is to be expected with the much more flexible mounting system.

Steady-state vibration spectra for the small pump at minimum speed are shown in Figure 18 for the upper transverse measurements (PUX and PUY, at the top of the bearing housing). The only noticeable peak is at the 1/rev and is at an average level of 0.194 ips-pk. Below the 1/rev frequency, any structural resonances of the pump isolation system would be noticeable, and there is a small peak in the PUY location. At the minimum speed, the separation between the mounting system resonances and running speed is also at a minimum, although very little response is observed. With the isolation design for the small pump, it was possible to select support springs and couplings that provided sufficient separation between the highest resonance and minimum speed, and the spectral data confirm this response.

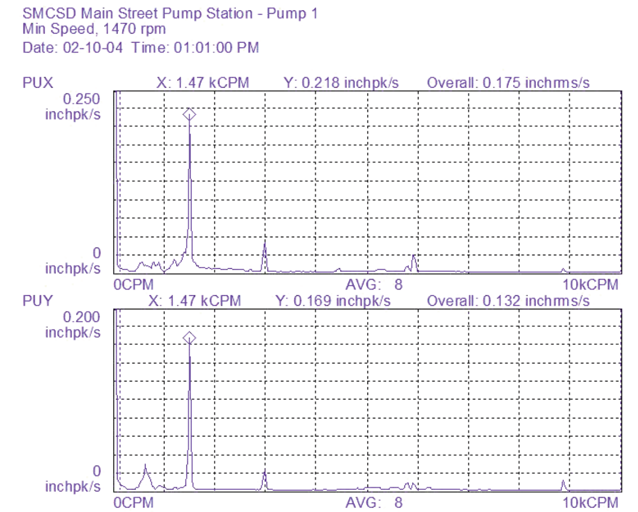

Figure 18. Vibration Spectra for Minimum Speed Operation of Small Pump.

Large Pump Vibration

Previous testing of the large pumps had indicated excessive unbalance, and during the subsequent teardown of the pump station to install the new isolation hardware, the impellers were balanced to API standards. With the new isolation components, the overall vibration values found during testing were all well under the limits of 0.33 ips-rms as required by the HI vibration standard. Maximum amplitude was 0.173 ips-rms at the maximum speed, which is 52 percent of the allowable.

A set of spectra at minimum speed from one of the large pumps, in this case Pump 2, is shown in Figure 19. The marker in both plots is set at operating speed, which is 930 rpm. Also present are harmonics at 2/rev, 3/rev (vane pass), and a few other higher harmonics. Below the operating speed frequency are several peaks corresponding to the natural frequency of the pump supported by the isolators and connected to the piping with flexible couplings.

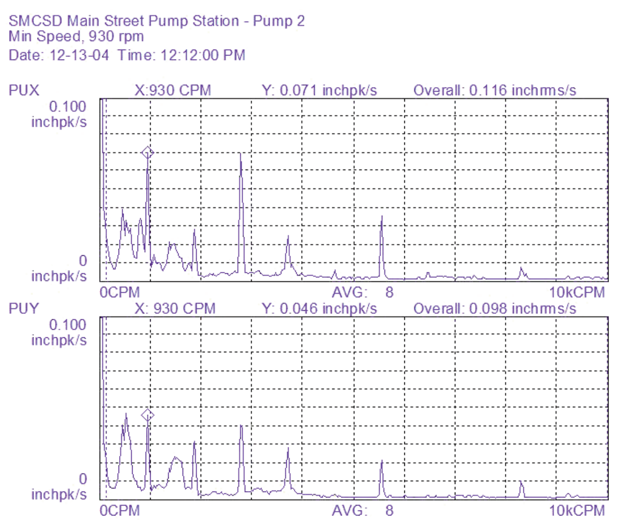

Figure 19. Typical Vibration Spectra for Minimum Speed Operation of Large Pump.

Isolator Effectiveness

A set of vibration tests was performed on the three pumps to determine the effectiveness of the new isolation equipment, to compare with analytical predictions, and to make sure the isolation devices are working properly. These measurements were conducted by placing accelerometers on either side of the isolator in question, in both vertical and horizontal orientations with the flexible couplings, and at two locations on the pump bases. The pumps were running at minimum speed, which is the condition at which the isolation will be least effective. The vibration measured across a device should show a marked decrease in amplitude if the isolator is working properly. Recall that with the original configuration, the vibration increased across some of the isolation equipment, especially at the couplings. An amount of vibration reduction or attenuation can be calculated from the decrease, with a desired reduction of 70 percent being the design target.

A table showing the resulting overall vibration found at each point as well as the attenuation (percent decrease in vibration) can be seen in Table 4 for the small pump. These measurements are the average from the two sets of readings across each component.

Table 4. Measured Vibration Across Isolators on Small Pump.

Position	Overall Vibration (ips-rms)	Attenuation
Inlet Inboard	0.060	82%
Inlet Outboard	0.011	
Outlet Inboard	0.078	45%
Outlet Outboard	0.043	
Base Inboard	0.056	80%
Base Outboard	0.011	

For the large pumps, the resulting overall vibration as well as the attenuation is listed in Table 5, again averaged for both pumps and the two measurement locations on each isolation component.

Table 5. Measured Vibration Across Isolators on Large Pumps.

Position	Overall Vibration (ips-rms)	Attenuation
Inlet Inboard	0.202	78%
Inlet Outboard	0.044	
Outlet Inboard	0.124	64%
Outlet Outboard	0.045	
Base Inboard	0.072	82%
Base Outboard	0.013	

When comparing the testing originally conducted, which had transmissibility values in some case greater than unity, with the values listed in the tables, a significant improvement in isolation effectiveness is observed. The minimum attenuation is 45 percent on the small pump with the outlet coupling, similar to the large pumps. This particular coupling is relatively short and could not be configured with low enough flexibility to achieve the desired 70 percent reduction. At the other locations, the attenuation meets or exceeds the design goal.

Subjectively, the vibration at the station on the walls and piping was significantly reduced using the new isolation system. Very little vibration can be sensed on the structure, and at the adjacent apartments, it is not possible to tell the pumps are operating while standing next to the station.

It should be noted that no modal testing of the pumps on the new isolation systems was conducted, to verify the frequency predictions of the models. Certainly, this type of testing is recommended, especially if the final performance is undesirable, however in this situation, the very acceptable end result precluded the need to continue making vibration measurements.

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CONCLUSIONS

This paper describes the analytical and empirical evaluations conducted to design an effective isolation system for a set of pumps at a wastewater pump station in Sausalito, California. The overall objective of the project was to specify replacement isolation hardware for the inlet coupling, discharge coupling, and pump base mounts that did not require extensive modification of the existing piping and supports yet would significantly reduce transmitted vibration to the surrounding structure. New hardware was needed because testing indicated that the original flexible couplings at the inlet and discharge did not isolate very well, and the pump base mounts were not seismically restrained. A summary of the results generated by the analysis and testing performed on this project may be listed as follows:

- In order to fully characterize the vibration of the pumps on isolators, it is essential to perform an analysis with a 3D finite element model. All six modes of rigid body motion must be considered, using accurate stiffness values of the isolation components. With a complicated isolation system such as these pumps, simple hand calculations are inadequate.
- The original isolation configuration results in rigid body modes within the operating speed range of the pumps. This is primarily due to the stiffness of the inlet and discharge flexible couplings being too high. With this adverse natural frequency situation, the original isolation system is not effective in attenuating vibration created by the pumps.
- Results from a forced response analysis showed the transmissibility of original configuration, on an average basis, varied from 0.8 to 15.6, clearly unacceptable.
- Replacement isolation hardware was specified that analytically showed a substantial improvement in isolation effectiveness. Using the 3D model, the stiffness of the isolation devices was adjusted to keep the modes sufficiently under the minimum operating speed of the pumps. With the couplings, precise values of stiffness were not obtained from the manufacturer, requiring detailed finite element models of the hardware to be made. Using this proposed system resulted in a significant drop in natural frequencies and corresponding improvement in transmissibility, to a value 0.3, or 70 percent isolation effectiveness.
- Based on test measurements, the new isolation components all showed substantial decreases in transmitted vibration, with a minimum reduction of 45 percent and a maximum of 82 percent, compared to the design goal of 70 percent. Because the discharge flexible couplings could not be obtained with sufficiently low stiffness, the desired reduction was not achieved. In addition to this improvement, all of the isolation components are now seismically safe.

Given the relatively tight quarters in the pump room, any additional reduction in vibration would be difficult to achieve. The vibratory performance of the equipment is considered to be very good, and the low transmissibility has alleviated any complaints from the adjacent apartments.

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