A SOLUTION FOR VARIABLE SPEED VERTICAL PUMP VIBRATION PROBLEMS

by

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INTRODUCTION

A system for circulating (for cooling) and delivering highly toxic liquids was devised for a new chemical plant. Four 90,000 gallon tanks, each with two identical sump pumps, were mounted on load cells for accurate delivery and inventory control. The tanks were specially heat treated and stress relieved and welding upon them was *not* allowed. The cross-section view of a tank mounted pump is shown in Figure 1. The 25 hp motor was mounted vertically atop two cast iron distance pieces or "tripods" which housed the flexible shaft coupling and seal area. This was bolted to a very thick flange mounted on an extended tank nozzle. The nozzle was extended such that there would be easy maintenance access to the flange studs without removing any of the grating or piping when servicing a unit. A shaft tube through which a 1⅝ in shaft passed to drive a two-stage sump pump extended approximately 12 feet into the tank. Process fluid lubricated carbon bushings, spaced along the shaft tube, maintained the shaft position. The pump suction tube extended into a recessed well or sump and a discharge line, bracketed to the shaft tube, rose vertically to exit back at the top of the tank.

INITIAL FINDINGS

During initial packing run-in, high vibration was observed on all units. Since these were variable speed pumps, it was first observed that high vibration seemed to occur only at certain speeds. Field testing was begun by uncoupling the drive motor from the pump shaft, adding some wax to unbalance the motor, and tracking the vibration with a synchronous tracking filter as the motor coasted down from 1800 rpm to rest. Thus, the unbalance force acted as a variable frequency exciter, "ringing" all system resonances below 1800 cpm. A once-per-turn speed reference mark was supplied by reflective tape and an optical pickup. The apparatus used for data acquisition and reduction is shown diagramatically in Figure 2.
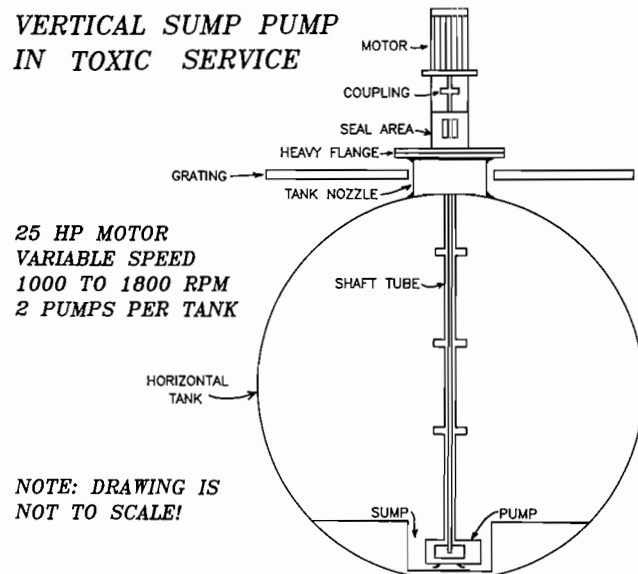

VERTICAL SUMP PUMP IN TOXIC SERVICE

MOTOR

COUPLING

SEAL AREA

HEAVY FLANGE

GRATING

TANK NOZZLE

25 HP MOTOR VARIABLE SPEED 1000 TO 1800 RPM 2 PUMPS PER TANK

SHAFT TUBE

HORIZONTAL TANK

NOTE: DRAWING IS NOT TO SCALE!

SUMP

PUMP

Figure 1. Diagram of Original Motor-Pump-Tank Assembly.

Two orthogonal structural resonances were observed, one longitudinal to the tank, referred to now as the east-west direction, and the other perpendicular to the tank, referred to now as the north-south direction. A rundown recorded from a velocity pickup mounted in the east-west plane on the drive motor is shown in Figure 3. A very pronounced resonance at 1240 rpm can be seen with an amplification factor of 25 (as calculated per American Petroleum Institute (API) Standard 617 [1]). This large amplification factor confirmed evidence that there was very low system damping and that it was probably structural in nature. Two smaller peaks can also be seen, one around 850 rpm and one around 1500 rpm; these will be discussed later. The Nyquist or polar form of the synchronous data shows these resonances clearly in Figure 4. The polar plot has been used with increasing frequency, due to its ability to show resonance phenomena without regard to runout, as is typical with non-contacting probes. The resonant frequencies, amplitudes, and especially their *phase relation* to each other are readily apparent in this format. The 90° phase shift from the initial vibration vector to the maximum vibration amplitude vector is also easily seen in this format. Rotor balancing can also be done simultaneously in several planes, with the polar form of the rotor response.
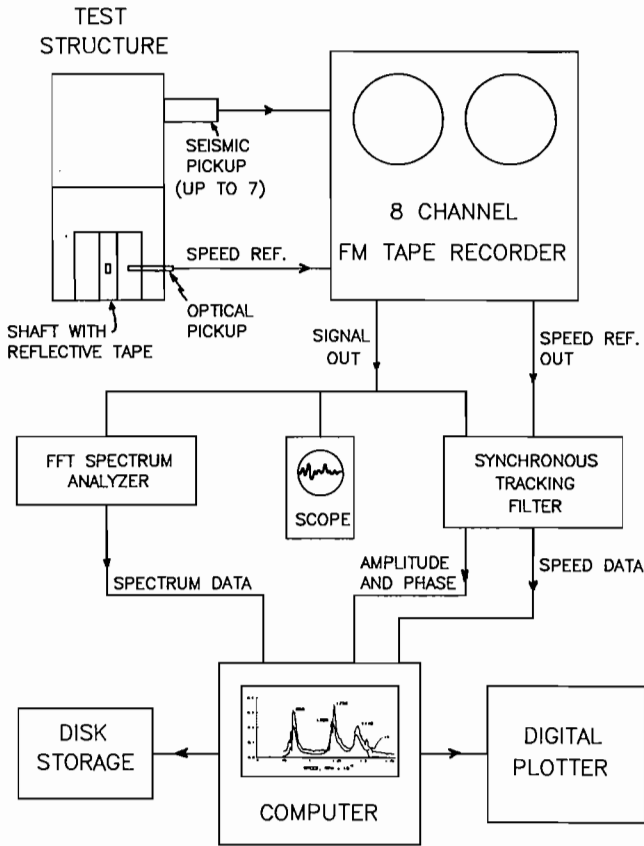
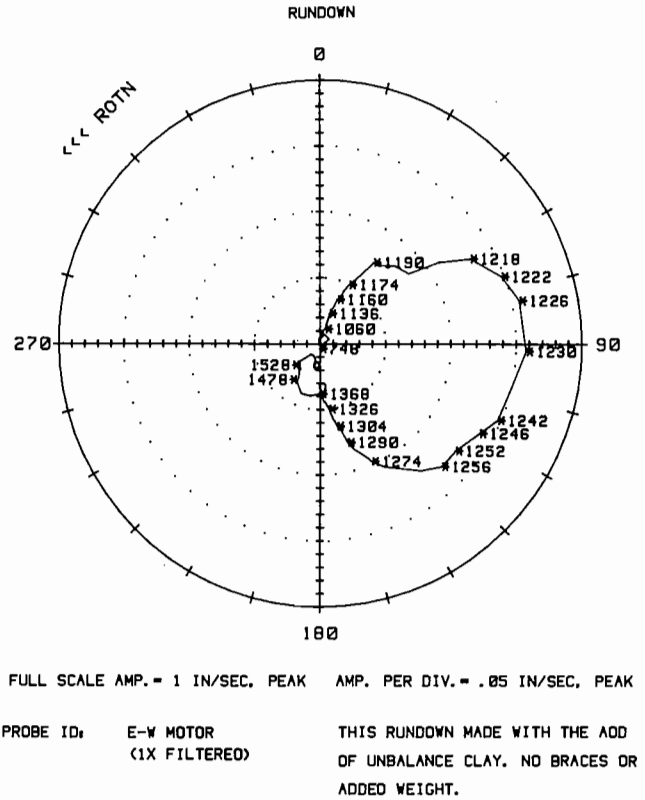Figure 2. Schematic Diagram of Data Collection and Reduction Equipment.

| PROBE ID: | E-W MOTOR (1X FILTERED) | THIS RUNDOWN MADE WITH THE ADD |
| PROBE ID: | E-W MOTOR (OVERALL) | OF UNBALANCE CLAY. NO BRACES OR ADDED WEIGHT. |

Figure 3. Bodé Plot of Original East-West Motor Vibration.

FULL SCALE AMP. = 1 IN/SEC. PEAK   AMP. PER DIV. = .05 IN/SEC. PEAK

| PROBE ID: | E-W MOTOR (1X FILTERED) | THIS RUNDOWN MADE WITH THE ADD OF UNBALANCE CLAY. NO BRACES OR ADDED WEIGHT. |

Figure 4. Polar Plot of Original East-West Motor Vibration.

The same rundown recorded from a velocity pickup mounted in the north-south plane is shown in Figures 5 and 6. The major response, with an amplification factor of 26, now appeared at 1450 rpm. There was also a lesser response around 1250 rpm. An example of a series of "rap" tests made with the unit not running is presented in Figure 7. Besides confirming the structural resonances already mentioned, note a response around 800 to 900 cpm and the absence of any response peaks above 1800 cpm.

FIELD TESTS BEFORE ANALYSIS

At first it was thought that the asymmetry in the responses was due to the cutouts in the distance pieces that allow access to the coupling and seal areas. However, reorienting these pieces resulted in no significant change in the vibration amplitude, frequency or direction. Later analysis proved that the majority of the system flexibility was attributable to the extended tank nozzle design.

The next step was to attempt to alter the mounting arrangement to move the resonances out of the operating range. Several techniques have been successfully applied by others in solving similar problems. In one case, stiffening and bracing were effective in moving the resonance above the operating range [2]. In another case, rubber mounts were used to lower the resonance below the operating range [3]. Due to the critical nature of this service, the elastic mounting was not considered, for safety reasons. Stiffening the assembly was considered; however, this proved to be very difficult, because the motor could not be braced to an adjacent wall, since the tank was on load cells and the automatic control systems would be adversely affected. Welding or bolting to the tank were also eliminated as solution options, due to metallurgical and code restrictions. Several attempts were made at bracing the motor with jackbolts and other brackets with little success. The maximum frequency
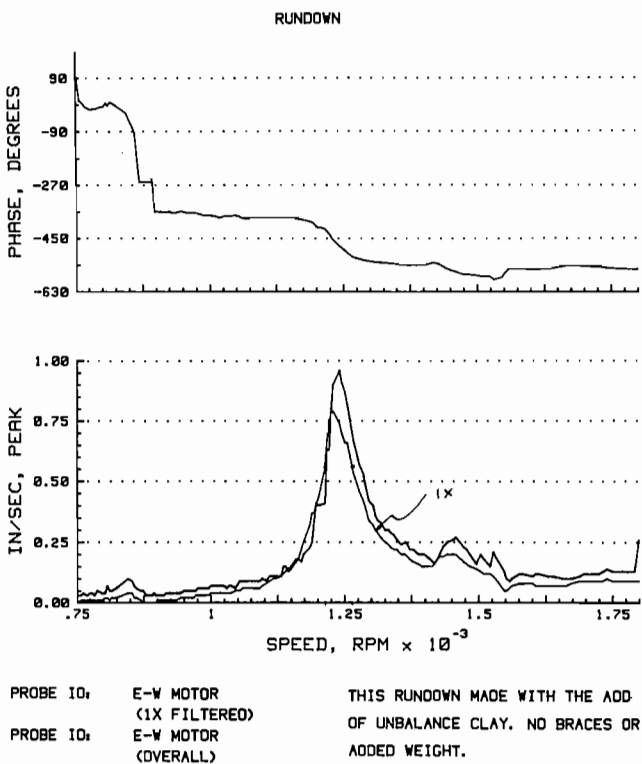
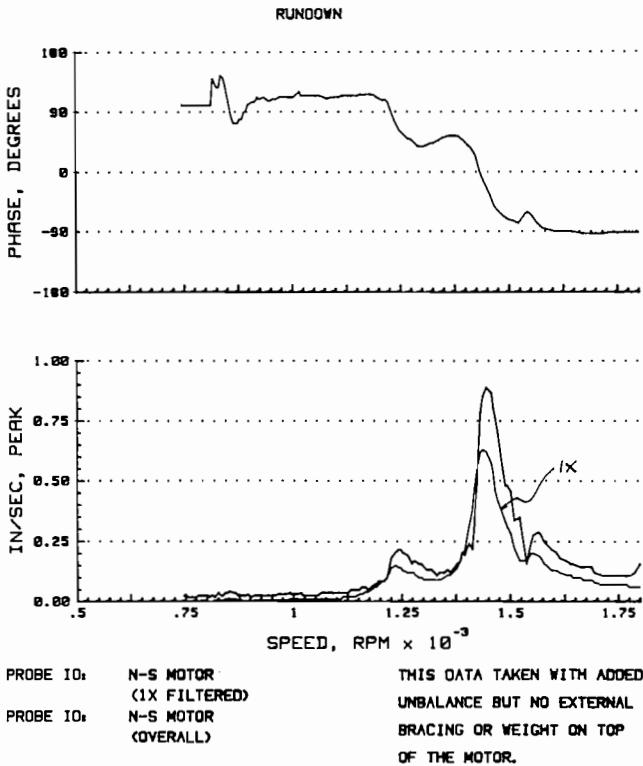*Figure 5. Bodé Plot of Original North-South Motor Vibration.*

Figure 6. Polar Plot of Original North-South Motor Vibration.

shift obtained by stiffening was a 100 to 125 cpm increase in the natural frequencies.

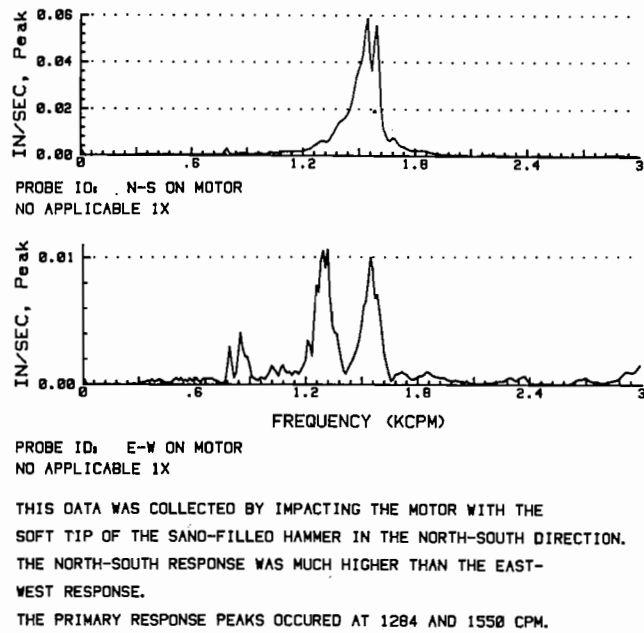As is typical, the pump manufacturer was not supplied with the complete mounting details of these units beforehand and,

Figure 7. Impact Response of Original Motor Assembly, Both Directions.

while a vendor supplied solution is desirable in a case such as this, the pump manufacturer was neither at fault nor of much help in diagnosing the problem.

The resonance observed around 850 cpm had not been explained yet, so velocity pickups were placed along the shaft tube in the tank and the uncoupled motor was again allowed to coastdown and act as a variable frequency system exciter. While the shaft tube responded to the motor assembly resonances at 1240 rpm and 1440 rpm, there was also a sharp resonance at 850 rpm, as is shown in Figure 8 and Figure 9. Rap testing confirmed this to be a structural resonance, as can be seen in Figure 10. The various signals collected along the length of the shaft were all in-phase, suggesting the first mode of an over-hung system (cantilever swinging of the entire assembly).

ANALYSIS

The shaft tube and pump arrangement were analyzed using a method by Strazzabosco and Primak [4] for finding the first natural frequency of a vertical shaft. Their analysis includes the stiffness of the shaft and its supports, the mass of the pump and the method of attachment to the top of the tank. Using the calculations provided, a natural frequency of 828 cpm was predicted, which was in good agreement with the natural frequency measured. The equations and a short computer program for this calculation are shown in the APPENDIX. While 850 rpm was in the range of desired speeds, internal bracing of any kind was virtually impossible, and it was decided to tackle to motor assembly resonances first.

The analysis of the entire system was performed by Harry Karabinis, a structural engineer with Monsanto Corporate Engineering in St. Louis, Missouri. He modelled the system on a finite element analysis computer program called STRUDL, which was originally developed by MIT's Civil Engineering Systems Laboratory and improved by McDonnell Douglas Company. This analysis confirmed that the tank nozzle was the "soft" member. As a result of that analysis, a spring-mass detuner was designed to be placed on top of the motor to alter the system resonances. This approach is similar to the spring-masses which are called resonant or dynamic absorbers and are often applied to fixed-speed, single-frequency systems. *Howev-*
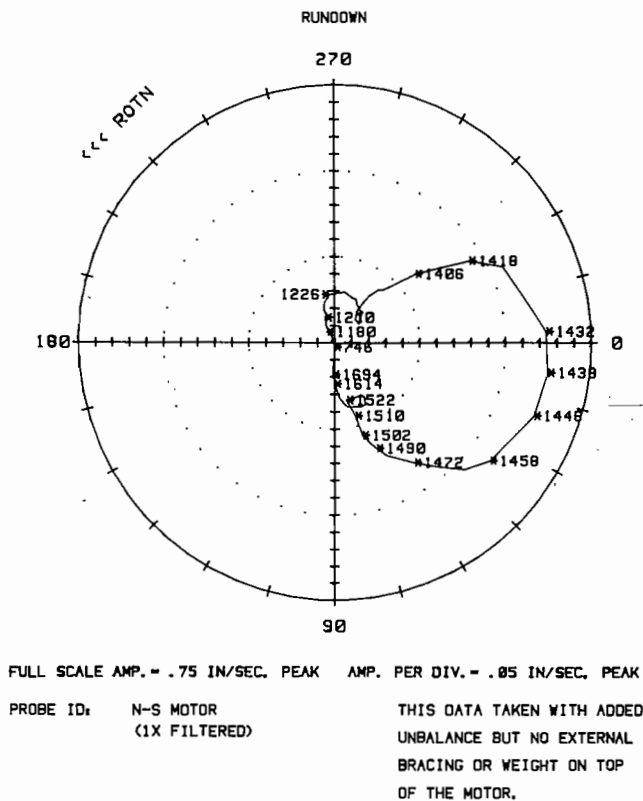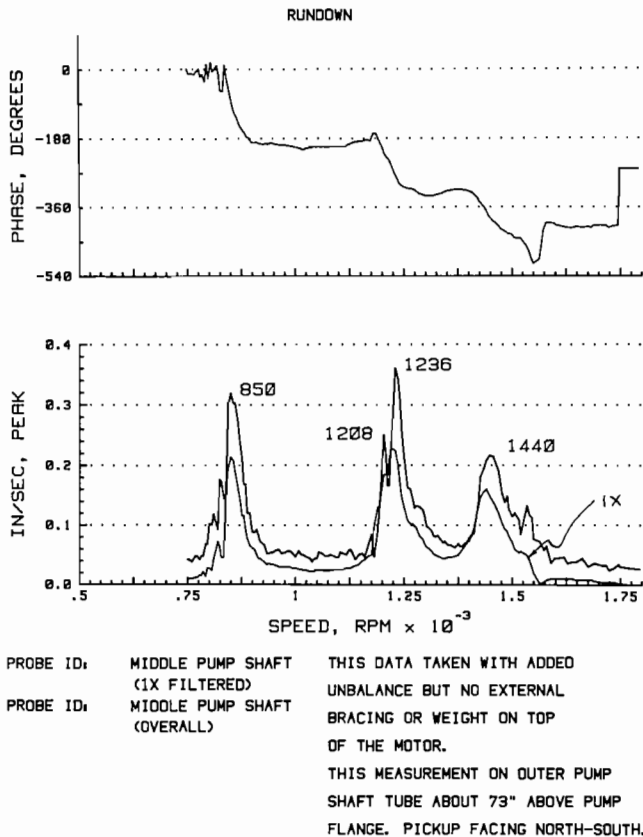
PROBE ID: MIDDLE PUMP SHAFT   THIS DATA TAKEN WITH ADDED
            (1X FILTERED)   UNBALANCE BUT NO EXTERNAL
PROBE ID: MIDDLE PUMP SHAFT   BRACING OR WEIGHT ON TOP
            (OVERALL)   OF THE MOTOR.
                                   THIS MEASUREMENT ON OUTER PUMP
                                   SHAFT TUBE ABOUT 73" ABOVE PUMP
                                   FLANGE. PICKUP FACING NORTH–SOUTH.

Figure 8. Bodé Plot of Pump Shaft Tube Vibration, North-South Direction.

er, most references preclude use of this solution on variable frequency machines. This is because two "sidebands" will appear on either side of the frequency being suppressed and, thus, make application to variable speed machinery difficult. However, this phenomenon can be applied to a structural system if the produced sidebands are properly placed in the design.

To show how this design can be applied, consider the simple two-spring-mass system shown in Figure 11. When an additional mass m is added via spring k to an existing mass-spring M and K, the new system will have *two* modes of motion, since it now has *two* degrees of freedom. The first mode will have masses m and M "in-phase" with flexing at spring K. The second mode will have masses m and M moving relative to each other 180° "out-of-phase," with flexing occurring at both k and K. Assuming harmonic motion, the equation of motion for this system can be shown to be:

$$w^4 - ((k+K)/M + k/m)w^2 + (kK + K + k)/(mM) = 0$$

which has the solutions:

$$w^2 = (k+K)/2M + k/2M \pm \sqrt{\left(\frac{k+K}{2M} + \frac{k}{2m}\right)^2 - \frac{kK + K + k}{mM}}$$

The complete derivation of these equations can be found in references [5] and [6]. If a wide range of stiffnesses and masses is chosen, the solution can then be plotted nondimensionally, as a family of curves, as shown in Figure 12. The curves represent the ratio of w_m/w_M as a function of mass ratio m/M. The two new system resonances w_1 and w_2 are then read on the ordinate as a ratio with the original natural frequency of the added detuner alone (w_m).

FULL SCALE AMP. = .3 IN/SEC. PEAK   AMP. PER DIV. = .02 IN/SEC. PEAK

PROBE ID: MIDDLE PUMP SHAFT   THIS DATA TAKEN WITH ADDED
             (1X FILTERED)   UNBALANCE BUT NO EXTERNAL
                                    BRACING OR WEIGHT ON TOP
                                    OF THE MOTOR.
                                    THIS MEASUREMENT ON OUTER PUMP
                                    SHAFT TUBE ABOUT 73" ABOVE PUMP
                                    FLANGE. PICKUP FACING NORTH–SOUTH.
                                    THREE DISTINCT RESONANCES ARE
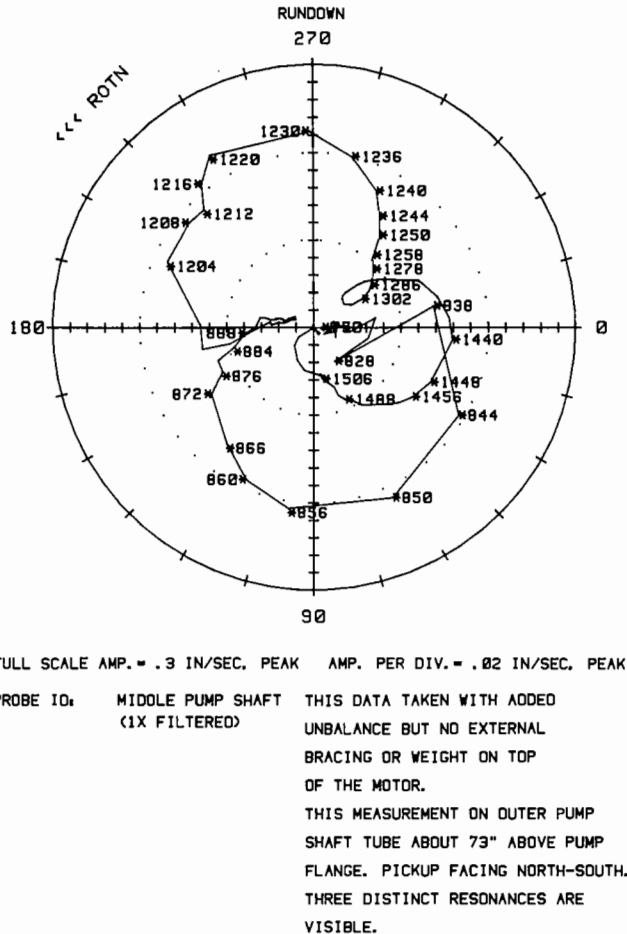   VISIBLE.

Figure 9. Polar Plot of Pump Shaft Tube Vibration, North-South Direction.

Note that as the mass of m is increased, the w_2 (lower, in-phase) frequency will decrease, assuming constant stiffnesses. Also, as the stiffness of the spring k is increased, the w_1 (higher, out-of-phase) frequency will increase. It is necessary to adjust both parameters in order to choose the proper detuner.

For example if k = K and m = M ($w_m = w_M$), then one can follow the $w_m/w_M = 1.0$ line over to the m/M = 1.0 line and read the new system frequencies from Figure 12 as $0.618w_m$ and $1.618w_m$. Thus, if $w_m = 1000$ cpm, then the new system would have an in-phase natural frequency at 618 cpm and an out-of-phase natural frequency at 1618 cpm. A plot similar to Figure 12 was shown at the First International Pump Symposium in May 1984 in the tutorial on the critical speeds of pumps by Chamieh [7].

The amplitude as a function of frequency of a two-mass system can be derived (as shown in [6]) and nondimensionalized, thus

$$X(K/F_o) = \frac{(1 - (w/w_m))}{[1 + k/K - (w/w_M)]^2 [1 - (w/w_m)] - k/K}$$

where F_o is the applied force. If this equation is plotted for the case previously discussed where k = K and m = M, the two system resonate peaks at 0.618 and 1.618 times w_m are clearly indicated, as shown in Figure 13.

It is useful to consider the best way to get the maximum "spread" between the two modes to create the largest safe operating range. To make the in-phase mode as low as possible, one would like to have m be large, compared to M. To make the
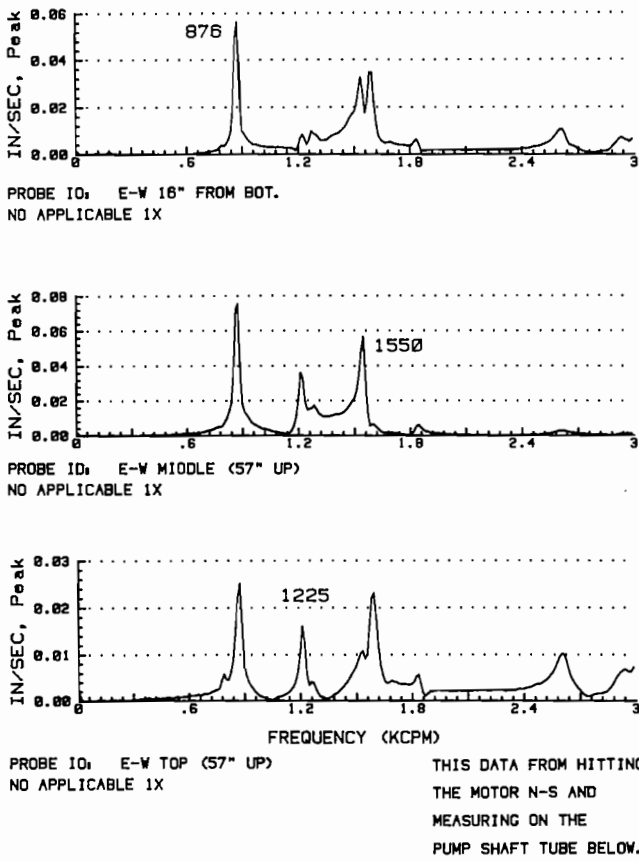
PROBE IO: E-W 16" FROM BOT.
NO APPLICABLE 1X

PROBE IO: E-W MIDDLE (57" UP)
NO APPLICABLE 1X

PROBE IO: E-W TOP (57" UP)
NO APPLICABLE 1X

THIS DATA FROM HITTING
THE MOTOR N-S AND
MEASURING ON THE
PUMP SHAFT TUBE BELOW.

Figure 10. Impact Response of Pump Shaft Tube, Various Locations, North-South Direction.
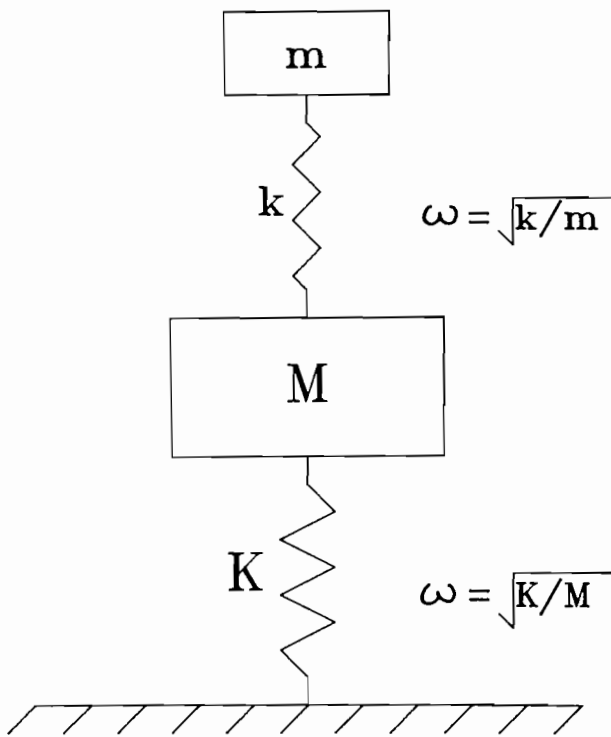

$$\omega = \sqrt{k/m}$$

$$\omega = \sqrt{K/M}$$

Figure 11. Schematic Diagram of Simple Two-Spring-Mass System.
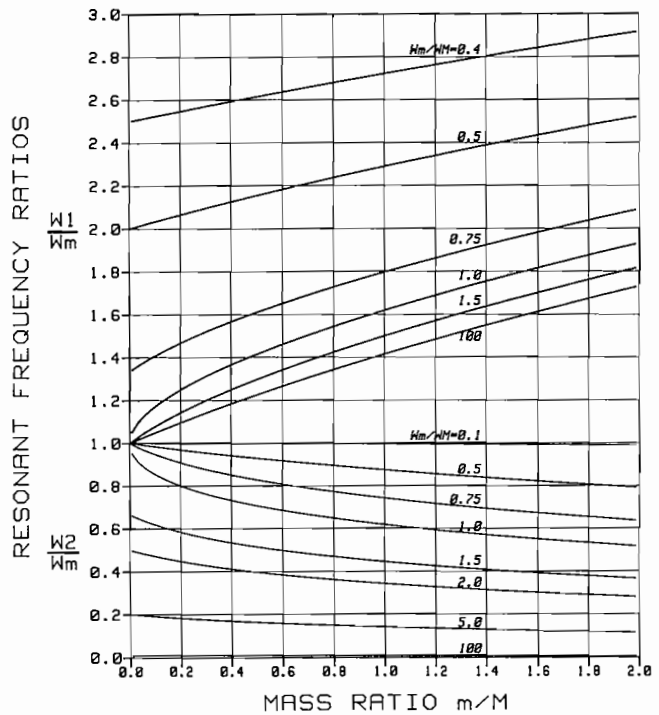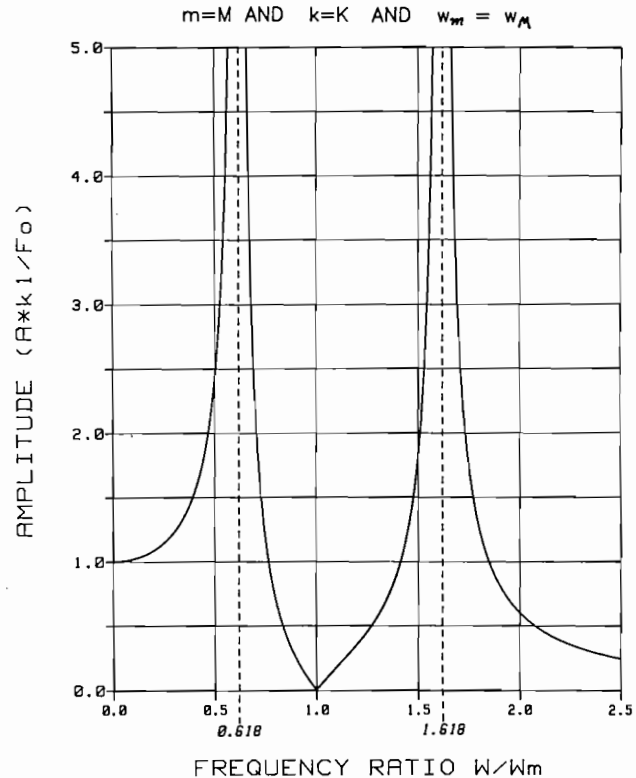

Figure 12. Nondimensional Resonant Frequencies Versus Mass Ratio.

Figure 13. Theoretical Undamped Amplitude Response when $m = M$ and $k = K$.

out-of-phase mode as high as possible, it is necessary to have k as large as possible, compared to K. Consider a system with $k = 10K$ and $m = 10M$ (note that w_m still equals w_M). The response of such a system is shown in Figure 14. In this case, the in-phase resonance will occur at 29 percent of w_m and the out-of-phase resonance will occur at 345 percent of w_m. How-
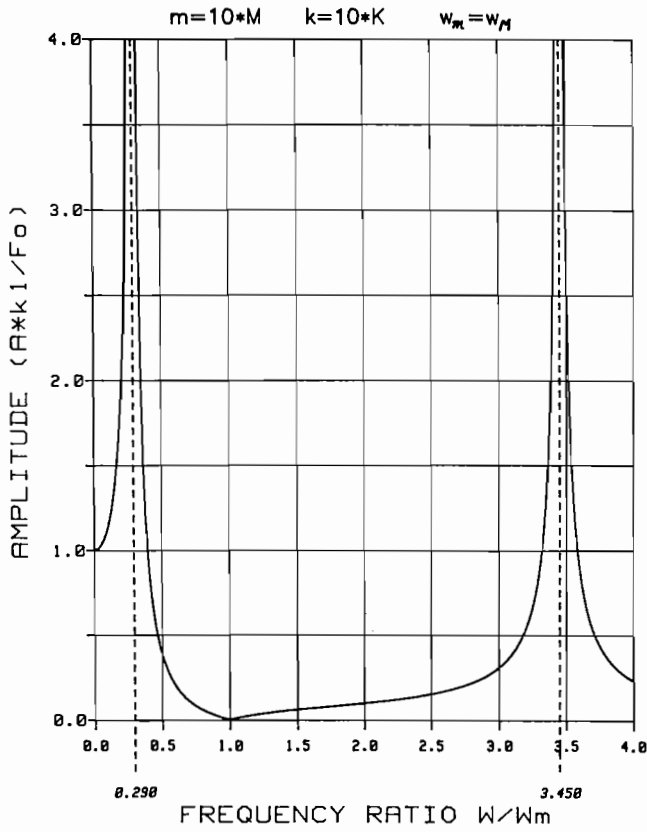
Figure 14. *Theoretical Undamped Amplitude Response when $m = 10M$ and $k = 10K$.*

ever, this would be a most unusual and unwieldy system, with the mass and stiffness of the absorber ten times the sizes of the original system!

From a practical standpoint, the size of a system detuner will be limited by the physical nature of the original system. The detuner chosen for this case is shown in Figure 15. This design was considered the largest practical size easily handled in the field. The initial prototype was an adjustable tube-in-tube arrangement for fine tuning the stiffness in the field. It was found that good results were obtained when $m = 120$ lb and tube length was 24⅝ in, which yielded $k = 12,385$ lb/in, so that the tuner resonance alone was $w_m = 1906$ cpm. With the natural frequencies of the motor assembly known from the test results shown above, as well as its mass of 360 lb, the w_m/w_M ratio was $1906/1250 = 1.52$, and the mass ratio m/M was equal to 0.333. Using Figure 12, new system frequencies are predicted at 0.525 $(1906) = 1000$ cpm and 1.21 $(1906) = 2306$ cpm. Solving the amplitude equations for these conditions confirms the two peak amplitudes at those frequency ratios, as indicated in Figure 16.

RESULTS

In the field, the adjustable tube-in-tube test detuner was installed and a series of tests were run at various heights. Once the optimum height and weight were determined, final response data were taken. A rundown with the detuner attached and a velocity pickup mounted in the east-west plane is shown in Figures 17 and 18. The speed range of 1000 to 1800 rpm is now completely free of resonances, with two resonant peaks observed below 1000 rpm. The new lower motor assembly resonance is at 780 cpm, and the shaft tube resonance is still at 850 cpm. Almost identical results for the north-south plane (except that the new lower motor assembly resonance is at 940 cpm) are shown in Figures 19 and 20.
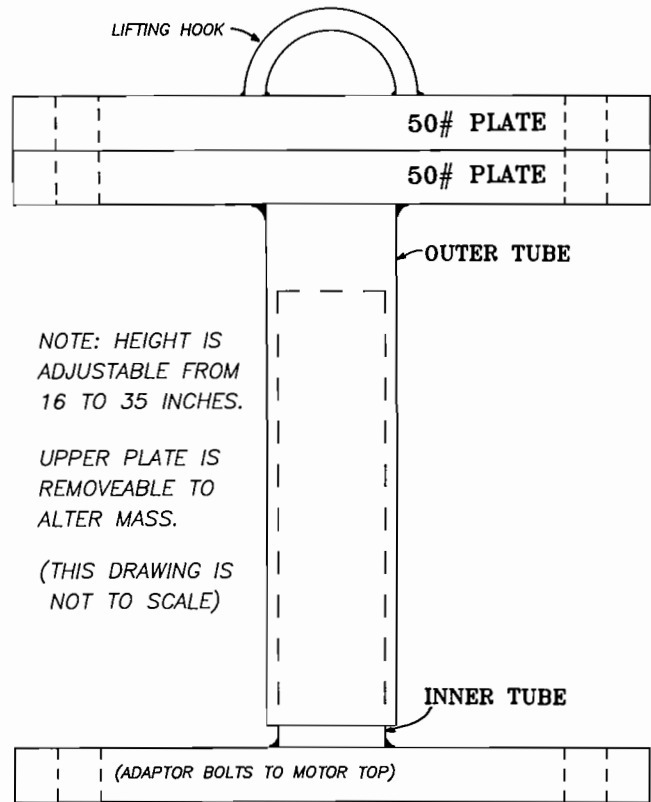

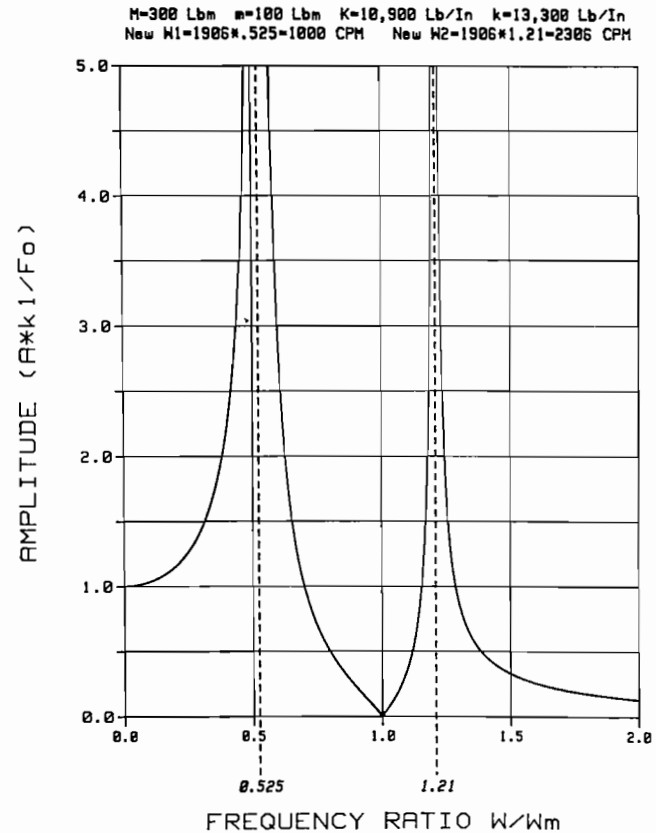Figure 15. *Adjustable Tube-in-Tube System Detuner.*

Figure 16. *Theoretical Undamped Amplitude Response for Actual Detuner Applied to Motor Assembly.*
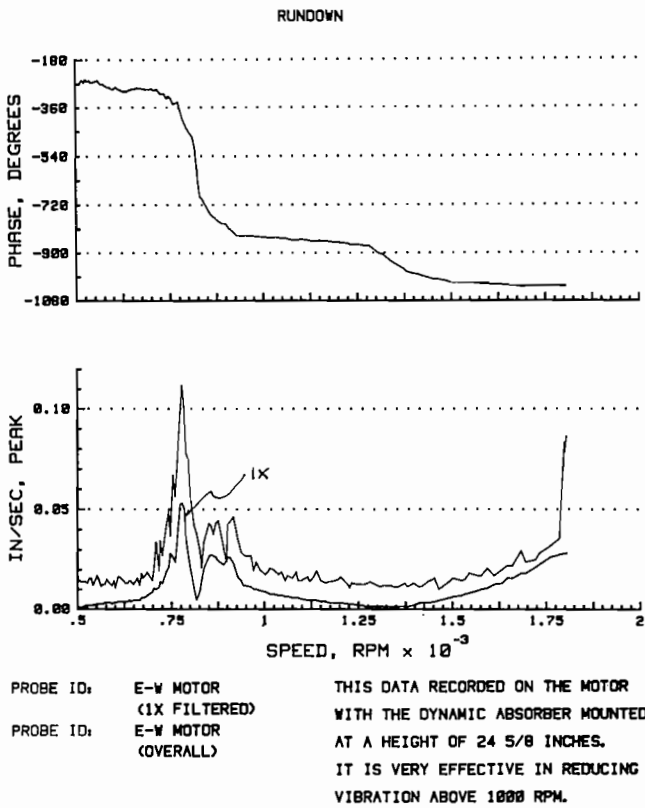
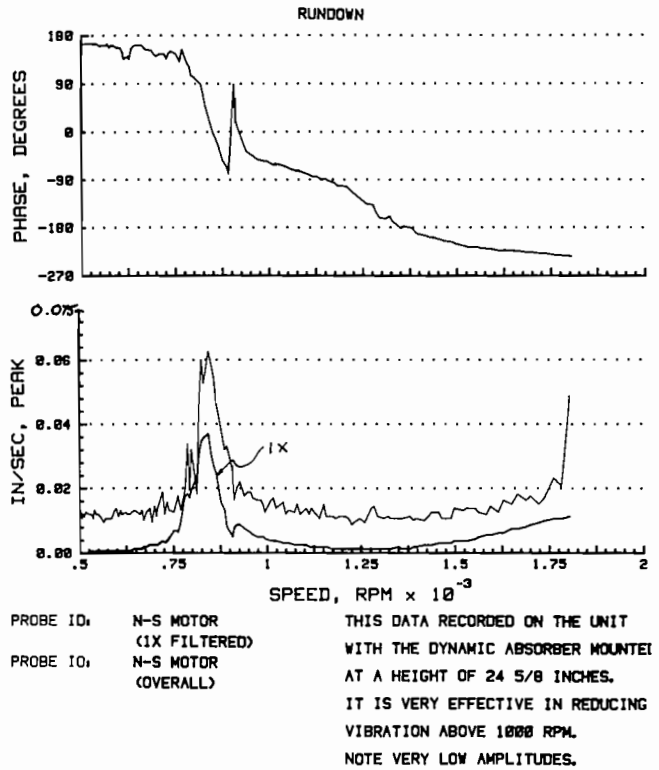Figure 17. Bodé Plot of Final Modified East-West Motor Vibration.

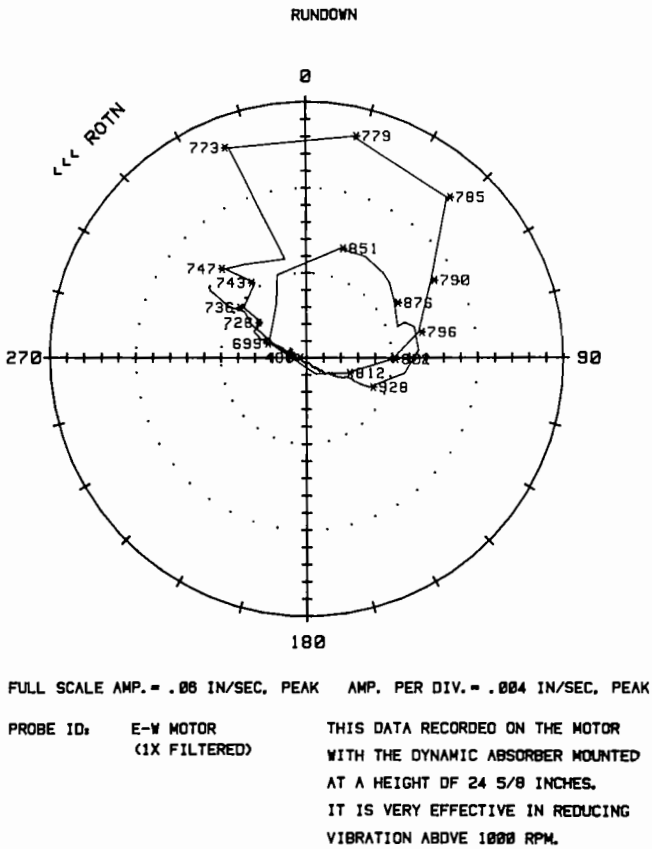Figure 19. Bodé Plot of Final Modified North-South Motor Vibration.

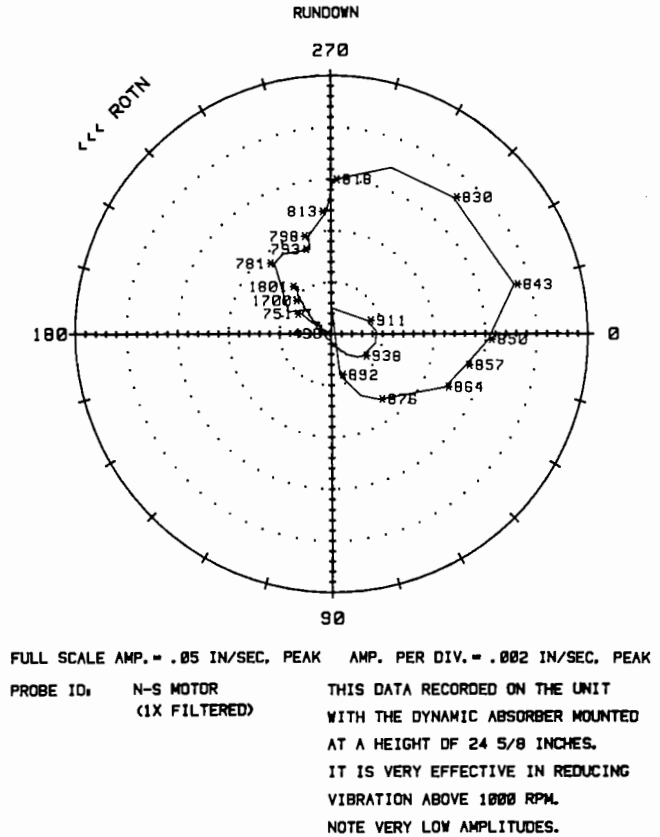Figure 18. Polar Plot of Final Modified East-West Motor Vibration.

Figure 20. Polar Plot of Final Modified North-South Motor Vibration.

Extensive rap testing was conducted to assure the absence of structural resonances in the desired frequency range. One example of those tests (Figure 21) shows the range of 1000 to 2000 cpm to be free of any response peaks. The shaft tube resonance and lower motor system resonance are visible, along with the upper motor system resonance, which appears at 2160 cpm versus 2300 cpm as predicted. Remember, there were no response peaks between 2000 and 3000 cpm before the detuner was mounted.
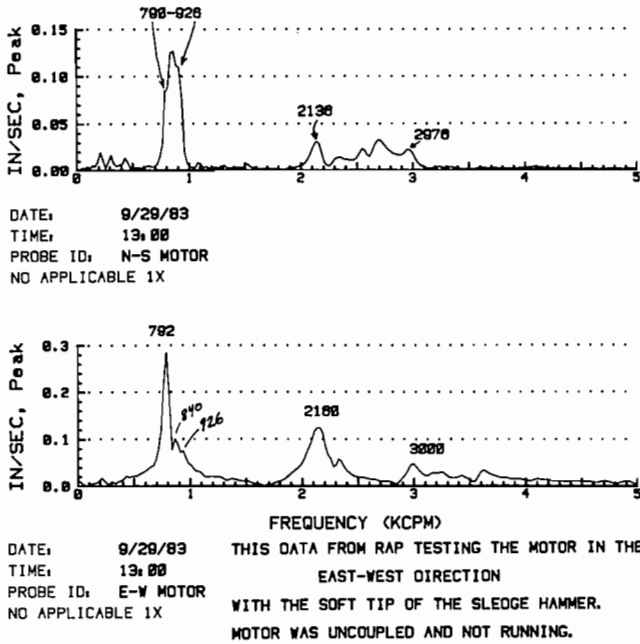

Figure 21. *Impact Response of Final Modified Motor Assembly, Both Directions.*

The final system is shown in Figure 22. The final design was slightly different than the test rig, but it achieved comparable results. The range of 1000 to 1800 rpm remained resonance free. Due to physical as well as time restraints, no attempt was made to detune the 850 cpm shaft tube resonance. The lower motor system resonance had been moved into this range and moving it further would have been difficult. To avoid these lower frequency resonances, the speed range of 700 to 1000 rpm was "jumpered out" in the motor speed control panels, so operation in that range is operator impossible. The resultant modified system allows complete automatic control of the delivery and inventory control system for minimum energy usage and safe vibration avoidance.

CONCLUSIONS

A set of variable speed vertical pumps in critical toxic service were modified by changing the system from one which had a single structural resonance within the operating range to a two-spring-mass system with two resonances, both outside the operating range. This was accomplished by the addition of a second mass, attached via a pipe to the top of the drive motor. It has been shown that a simple two-spring-mass system can be used successfully to model a real system and solve a potentially complex problem. The solution was simple, easy to implement, and easy for the plant to maintain. Differences in the actual versus predicted results are due to the complex interaction of the motor resonances with the shaft tube resonance. These units have now operated flawlessly for over 18 months.

As more and more variable speed motor applications are introduced to save energy and integrate automatic control into
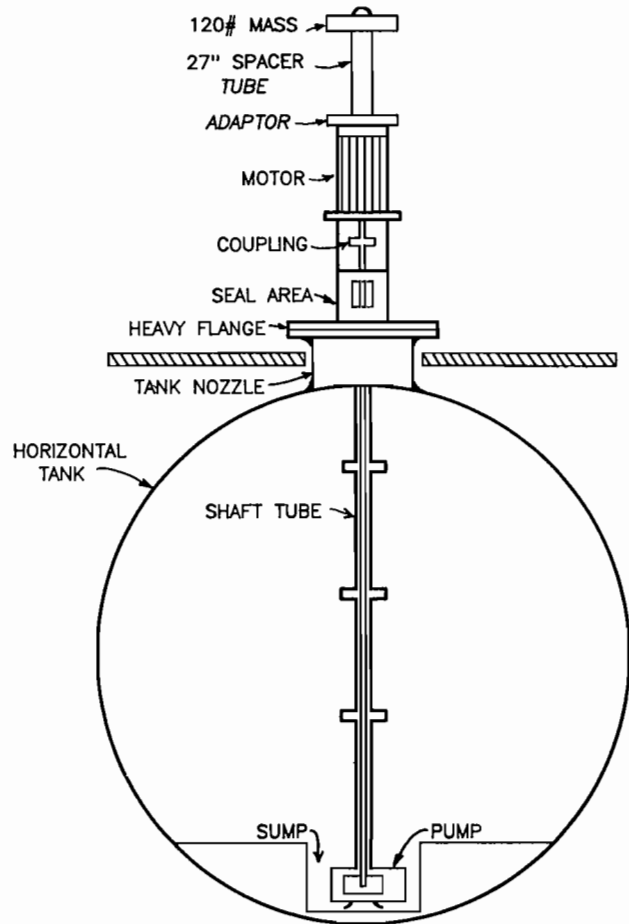

Figure 22. *Final System with Detuner Installed.*

process systems, structural resonance problems are likely to be seen with increasing regularity. Future installations of this type should be examined during the design phase, to avoid field problems. Putting the responsibility on the pump manufacturer would seem the logical thing to do. However, this may go beyond the expertise of most vendors, particularly since most vertical pump installations in the past have been those with fixed speed. Producing safe reliable pump systems designs incorporates much more than delivering the required head and flow. Thus, the mechanical engineer must develop the ability to peform design audits to predict these potential problems and provide engineered solutions.

APPENDIX

Marenco and Vergani [4] presented the derivation of the equations used to calculate the natural frequency of vertical shaft assemblies. These equations were used to write the following short BASIC computer program to perform the calculations.

Program line	Comment on line
10 E = 3.0E07	Modulus for steel
20 I = 159	Inertia (in^4) of tube + shaft
30 B = 12.75*12	12¾ feet long tube
40 A1 = 3/(2*B)	computed quantity
50 Ep = 15	location of center of gravity of pump (15 in from end of tube)
60 G = 386	gravity
70 A = PI*36	area (PI = 3.14159...)

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 80  K = (I/A)^{1/2}              radius of gyration
 90  W = 156                     weight of pump
100  M = W/G                     mass conversion
110  J = M*K^2                   pump moment of inertia
                                 (lb-in-sec^2)
120  Wn = (3*E*I/(B^3*           natural frequency (as given in [4])
     (M*(1 + Ep*A1)^2 +
     J*A1^2)))^{1/2}
130  Fn = Wn*30/PI    convert to cpm
140  PRINT "Natural frequency = ";Fn;" cpm."
150  STOP
RUN
```

Natural frequency = 828 cpm.

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ACKNOWLEDGMENTS