# CASE HISTORIES OF SPECIALIZED TURBOMACHINERY PROBLEMS

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## INTRODUCTION

Vibration problems in turbomachinery occur due to many factors. Some of the problems could have been prevented if more detailed design analyses were performed. Other times the problems occur due to design extrapolations which are pushing the state of the art. The case histories of excessive vibrations and failures that will be discussed in this presentation are examples of those where additional design analyses would not necessarily have predicted or anticipated the problem that occurred because the analytical models are not sufficient to take into account all the variables.

The first case discussed concerns a steam turbine that had repeated blade failures. Blade failure problems and their solutions are usually disappointing to the design engineer since many times the procedure for solving them is of the "beef it up" philosophy. When blade failures occur, they normally occur near the root and the failures are normally associated with fatigue, most probably at the blade natural frequency. After thorough analysis, the manufacturer usually increases the cross-section near the root and tries to reduce the stress concentration factors. Most of the time this is the appropriate action and the success of this approach can be thoroughly documented. In those cases where this method does not work, it becomes increasingly difficult to obtain meaningful data because other than strain gaging the actual blade in its hostile environment, all other measurements only infer what the blade is doing. To properly measure the blade response therefore requires a tremendous effort, both in time and money, especially with the cost of downtime in some process applications. The next best approach would be to make external measurements which could be related to blade response. The case presented will discuss how the response of the blades was monitored by measuring the bearing housing vibrations with accelerometers and utilizing a real time spectrum analyzer in conjunction with other special automatic analyzing equipment so that a continuous spectral display versus speed could be presented. The end result of the data analysis is a version of the Campbell diagram with the amplitudes of vibration superimposed. These data presentation techniques can be used to gain more spectral information than could normally be obtained by single spectral analysis plots or by spectral time histories.

The second case discusses the field balancing of a unit which had a bearing housing support resonance. The unit had several available balance planes and influence coefficients were developed for four of the planes. The primary emphasis of the discussion is centered around the inconsistencies that can be obtained using vibration amplitude and phase data where nonlinearities and resonances appear in the support structure. A digital computer program was used to help select the best balance plane and to obtain the optimum balance for all combinations of balance planes. The digital computer can accept all the input data from proximity probes, shaft orbits, bearing housing vibrations, and shaft absolute vectors and optimize on all possible combinations of probes, speeds, and balance planes. The role of the engineer in the evaluation of balance data is also discussed.

The third case describes the excitation of a compressor shaft at its fourth critical which was 15 percent above the running speed. The shaft was excited by an acoustical resonance in the suction piping caused by variation of the inlet guide vane angles.

These case histories are typical of the types of problems which can occur. Many times it requires rather sophisticated instrumentation, such as real time analyzers and automatic data analyzer systems and specialized measurement techniques to determine the cause of the problems. The techniques illustrated in these case histories have been used successfully to investigate and solve a wide variety of problems.

## CASE I: BLADE FAILURES IN STEAM TURBINE

A three stage steam turbine driving a centrifugal air compressor in a catalytic cracking unit had a history of

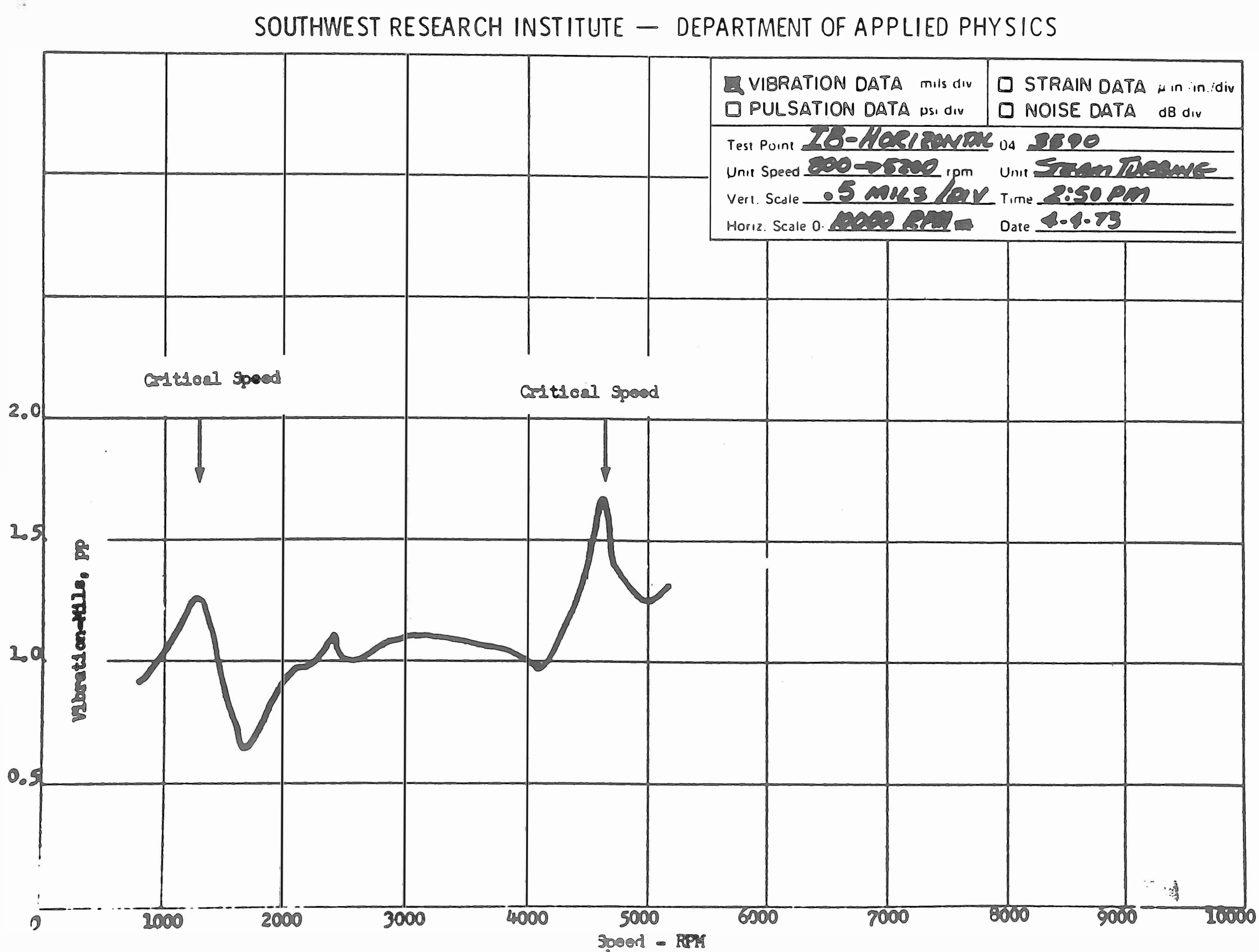



*Figure 1. Measured vibration response showing critical speed.*

vibration problems. These included excessive shaft vibrations, governor gear vibration problems, and failures of the first and third stage blades and shroud bands. This unit was rated at 4310 hp at 4575 rpm with a normal speed range of 3850 to 4800 rpm. Each of the three stages had 50 nozzle passages.

An investigation was made of blade failures to determine the causes and what steps could be taken to insure that the failures did not occur. Discussed in this presentation will be the steps involved in the investigation, the data obtained, and some of the data analysis techniques which were used to identify blade vibrations. These signature analysis techniques are applicable to other types of vibration problems.

To make a thorough investigation into the possible causes of blade vibrations, several calculations were made, including the torsional natural frequencies of the system, the lateral critical speeds of the turbine, and the natural frequencies and mode shapes of the first and third stage blades. In addition to the calculations, vibrations were monitored during the startup. Proximity probes were installed near the vertical and horizontal directions at the inboard and outboard ends of the shaft. An axial probe was used for a key phaser signal, so that a timing mark could be recorded. Accelerometers were used to monitor vibration signatures of the bearing housings and case on both the turbine and compressor.

*Summary of Calculations*

The torsional calculations showed that the running speed was between the first and second torsional natural frequencies (1146 and 11226 rpm). The critical speed was calculated to be in the running range (4800 rpm) for excessive bearing clearances. Figure 1 shows that the measured horizontal response at the inboard end of the turbine indicated a critical at 4600 rpm. The blade natural frequencies calculated are summarized in the Campbell diagram shown in Figure 2. The blades were calculated using partial fixity to compensate for the lack of total rigidity at the blade root. The nozzle passage frequency (50 times the running speed, 50X) can excite the second, third, and fourth tangential modes of the first and third stage blades on the normal running speed range (3850-4800 rpm).

*Summary of Field Investigation*

Vibration signature data was obtained on the turbine shaft, bearing housings, and case to determine if indications of vibrations at the blade natural frequencies could be depicted as a function of rpm. In this type of analysis, a dc voltage of the rpm signal is fed into the X-Y recorder and the rpm determines the baseline height on the spectrograph. The height above the baseline gives the amplitude of vibrations. This conveniently allows you to depict the Campbell diagram but with a vibration amplitude modulated on top of it. For ex-

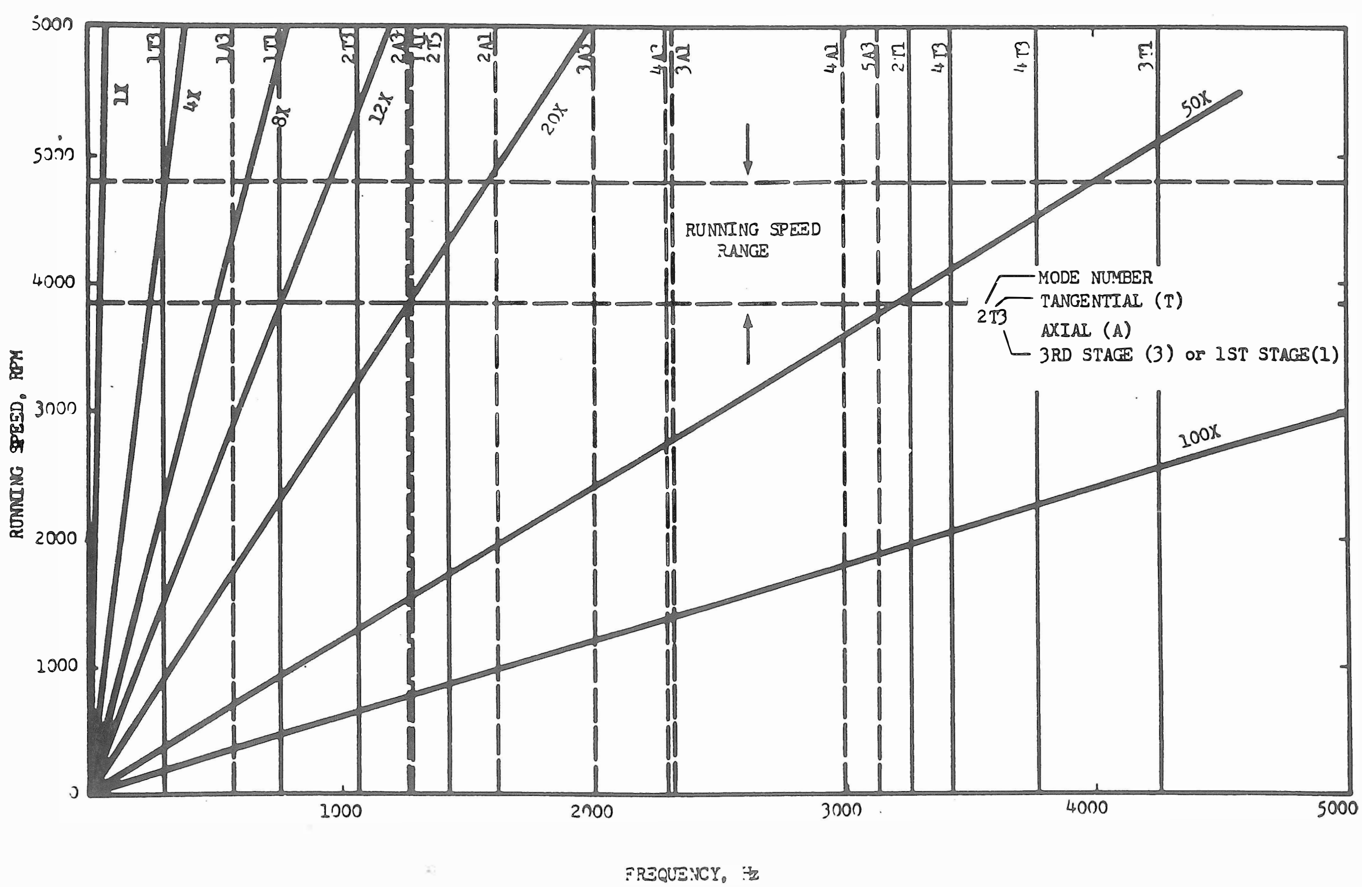



*Figure 2. Campbell diagram.*

ample. Figure 3 gives a spectral analysis versus rpm (0-1000 Hz) showing the lower excitation harmonics (1X through 14X running speed). Figures 4 and 5 show that the blade passage frequency excitation (50X) and its multiples excite the higher tangential modes of the first and third stage blades. For example, when the component at 50 times rpm passes through the 3100-4300 Hz range, large components are measured on the bearing housing. When the 100X component passes through this frequency range, it also excites these modes. The same is true of the 150X component at the lower speed. The significance of this is that by this data presentation method, seemingly insignificant responses or spikes in the vibration signatures can be shown to reveal information hidden within the data.

Figure 6 is a spectral analysis (0-1000 Hz) showing the excitation of possible blade frequencies when passing through the critical speed. As reported in a previous paper (*1*), when a unit runs on a critical speed, it can excite blades at their own natural frequencies. This phenomenon occurred during this field study near 4600 rpm which, from Figure 1, was the measured critical speed. Frequencies near the blades' first tangential modes and the first axial mode of the third stage blades were excited. The exact mechanism by which this energy couples is difficult to define; however, it has been observed repeatedly in the field. Table 1 gives a comparison of the calculated blade natural frequencies to frequency components measured on the bearing housings. Although it can be argued that the fact that these frequencies are there may not indicate that the blades are vibrating, the data presentation technique reinforces the assumption that it is blade natural frequencies since the same component is excited by other multiples of the blade passage frequency.

Figure 6 shows another interesting phenomenon. It is well known that centrifugal force causes the blade root to become more fixed as the speed is increased, thereby increasing the blade natural frequency. Figure 6 shows that near some of the calculated blade natural frequencies, an increase in the natural frequency is observed as a function of speed, until the speed reaches a certain level and then the frequency remains the same. This could be indicative of the effort of centrifugal force upon the blade natural frequency. It is also interesting to note the presence of components at the blade natural frequencies at all speeds. This means that blades do not have to be excited by integer harmonics of running speed.

[1]Numbers in parentheses refer to similarly numbered references in bibliography at end of paper.

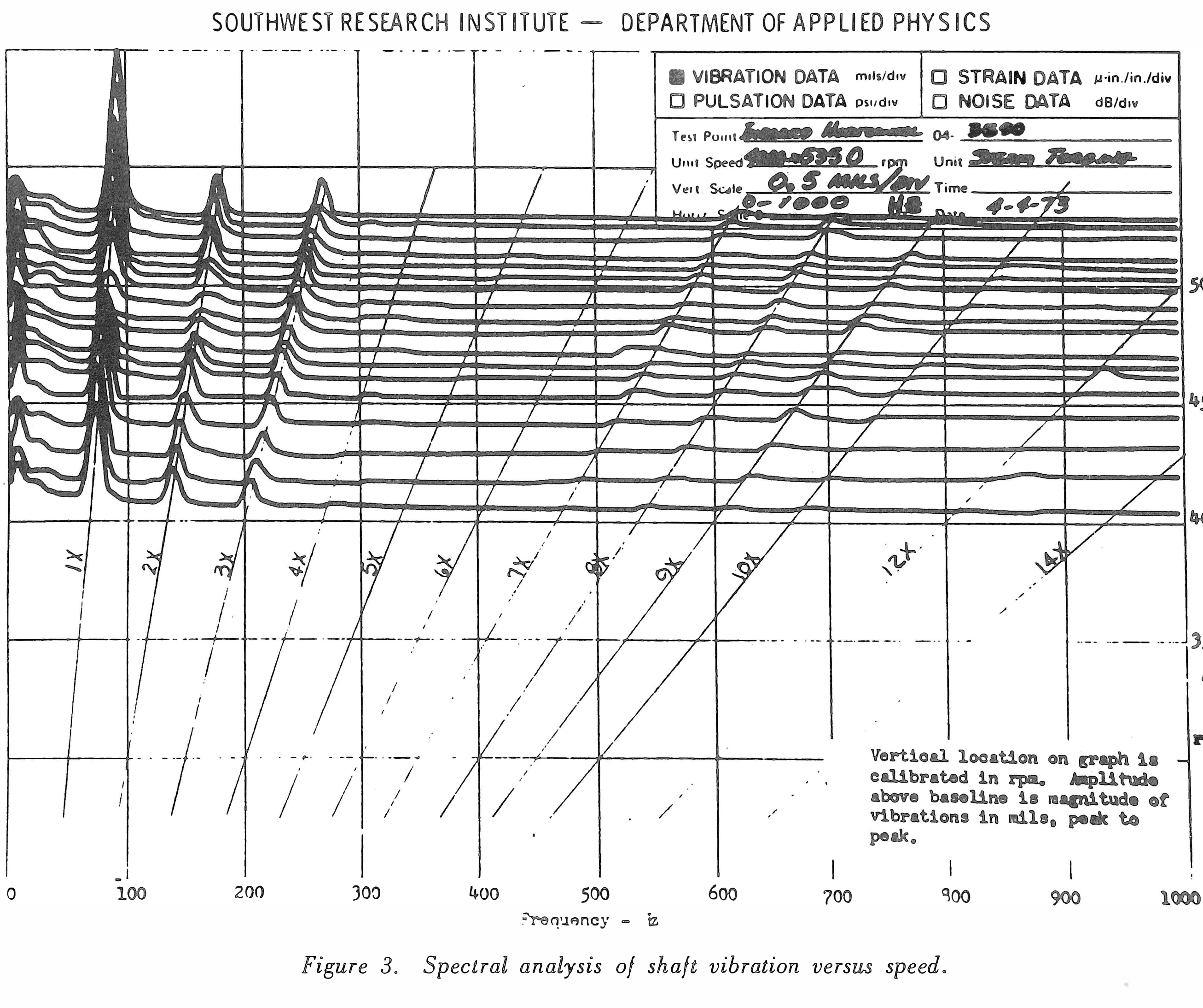



*Figure 3. Spectral analysis of shaft vibration versus speed.*

TABLE 1. SUMMARY OF CALCULATED AND MEASURED BLADE FREQUENCIES

| | | First Stage | |
|---|---|---|---|
| | | Calculated | Measured (a) |
| First Tangential | (1T1) | 759 | 780 |
| First Axial | (1A1) | 1275 | 1300-1500 |
| Second Axial | (2A1) | 1619 | |
| Third Axial | (3A1) | 2313 | 2500,2650 |
| Fourth Axial | (4A1) | 2991 | |
| Second Tangential | (2T1) | 3260 | 3200,3300 |
| Third Tangential | (3T1) | 4260 | 4200 |
| | | **Third Stage** | |
| | | Calculated | Measured (b) |
| First Tangential | (1T3) | 300 | 300, 310<br>325, 360 |
| First Axial | (1A3) | 580 | 440, 450<br>500, 570 |
| Second Tangential | (2T3) | 1080 | 890,1000 |
| Second Axial | (2A3) | 1270 | |
| Third Tangential | (3T3) | 1430 | |
| Third Axial | (3A3) | 2000 | |
| Fourth Axial | (4A3) | 2300 | |
| Fifth Axial | (5A3) | 3130 | 3150 |
| Four Tangential | (4T3) | 3436<br>3773 | 3750,4100 |

(a) Measured on outboard bearing housing
(b) Measured on inboard bearing housing

During the fields test, the unit was run with unequal pressure distribution around the nozzle periphery to study partial admission effects. When compared to the full admission spectra, the difference was considerable. The net result is that amplitudes of the lower harmonics were larger, as can be seen by Figure 7.

The significance of the case is that by utilizing accelerometers on the bearing housings, positive identification of the excitation of blades at their natural frequencies can be depicted by utilizng data presentation techniques which utilize real time spectral analysis versus rpm, load, etc. This powerful technique can allow one to obtain more information from data than can normally be obtained utilizing single spectral plots. The composite spectral plot is a Campbell diagram with the vibration amplitudes superimposed on excitation harmonic lines. Another significant aspect is that the excitation of blades at their own natural frequencies was shown to occur when the unit was running on a critical speed.

## CASE II: MULTIPLANE BALANCING OF STEAM TURBINE

Upon startup of a steam turbine in a chemical plant, shaft vibrations and bearing housing vibrations were above the allowable limits. To reduce the vibrations, balancing was attempted. Difficulties were encountered in the balancing procedures due to a resonance associated

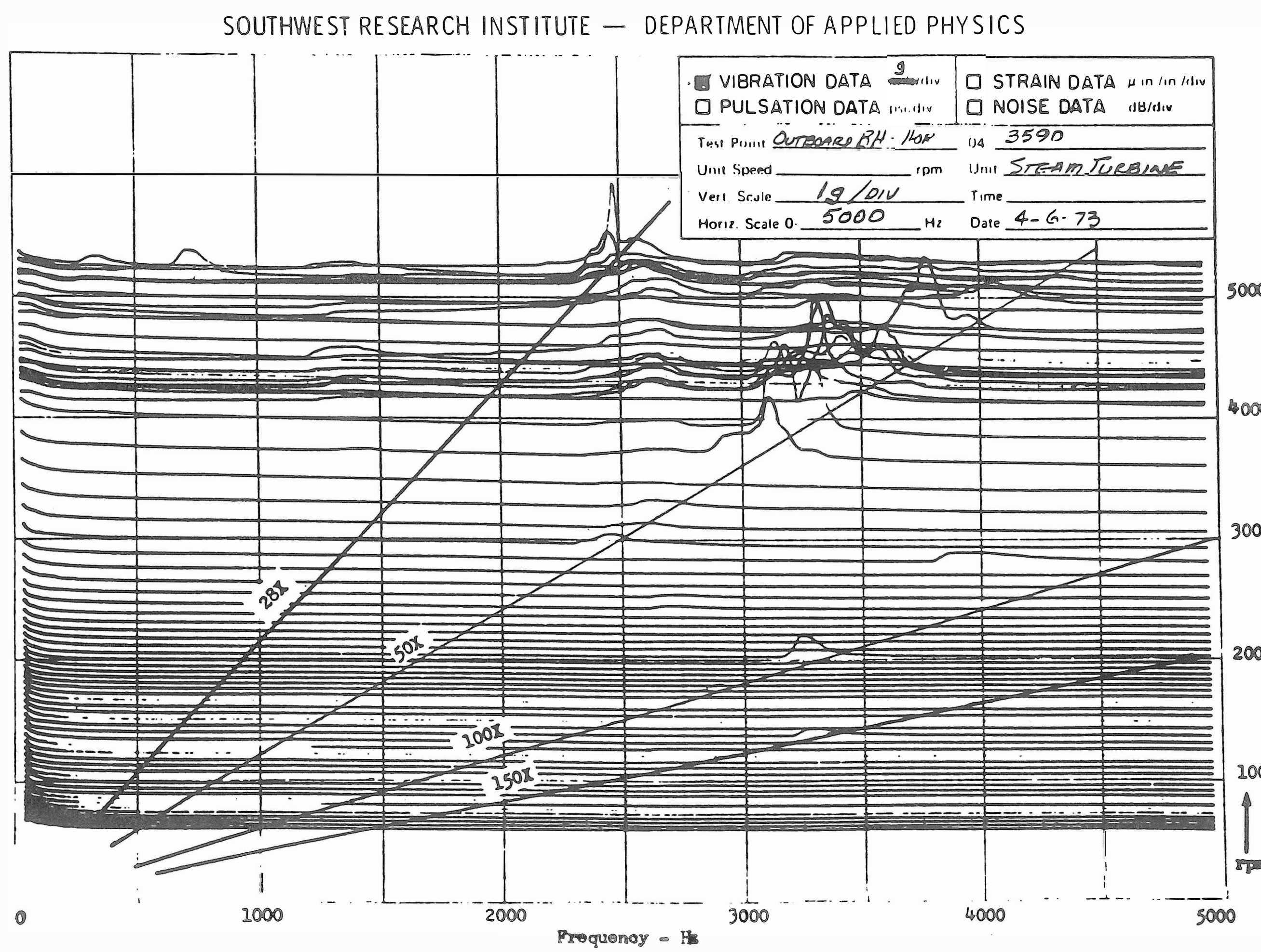



*Figure 4. Outboard bearing housing vibrations versus speed showing nozzle excitation frequencies.*

with the inboard bearing housing support structure. The authors were asked to assist in the balancing and to determine the exact location of the pedestal resonance by the use of an air driven shaker. This case is presented to illustrate some of the difficulties encountered in balancing and to show how digital computers can be used to optimize the balance of flexible rotors with multiple balance planes.

Figure 8 shows the turbine and compressor arrangement and the relative location of the balance planes. A list of the vibration data points is also given. Several possible combinations of these data points could be used to balance the rotor. Simple theory indicates that any two probes in a plane can be used. This means that the following combinations could be used: proximity probes—vertical and horizontal, bearing housing—vertical and horizontal, shaft absolute—vertical and horizontal. If the vibration signals 90 degrees apart are properly added, the combined orbits can be obtained and these can also be used for balancing. This preponderance of data quickly leads the engineer to ask the question as to which data is best to use. If the system is linear and the rotor and its supporting structure have no resonances in or near the selected balance speeds, theory indicates that any combination of probes should give the same balance solution. In the real world, the assumption of a linear system and no effects from rotor or structural resonances is seldom met. This means that inconsistencies in data will occur and that arbitrary selection of balance planes may not lead to a satisfactory balance condition unless extensive trial and error techniques are pursued.

Since in this case a resonance was suspected in the bearing housing support structure and verified by a shaker test (Figure 9), all of the listed data points were carefully measured so that a complete record of all data points was available. The data was checked to make sure that all the readings repeated. Bracing was added to the bearing housing support structure to try to move the natural frequency above the running speed range. After this, trial balance weights were added to each of the four balance planes. After an analysis of the vibration data, balance weights were added at balance plane 4. The vibrations were acceptable after two tries; therefore, balancing was terminated (Figure 10).

The decision to add the balance weights only to balance plane 4 was reached after detailed analysis of all data, including numerous vector plots and the use of a balancing digital computer program which the authors have developed. It would be desirable to have a computer program which would calculate exact balance

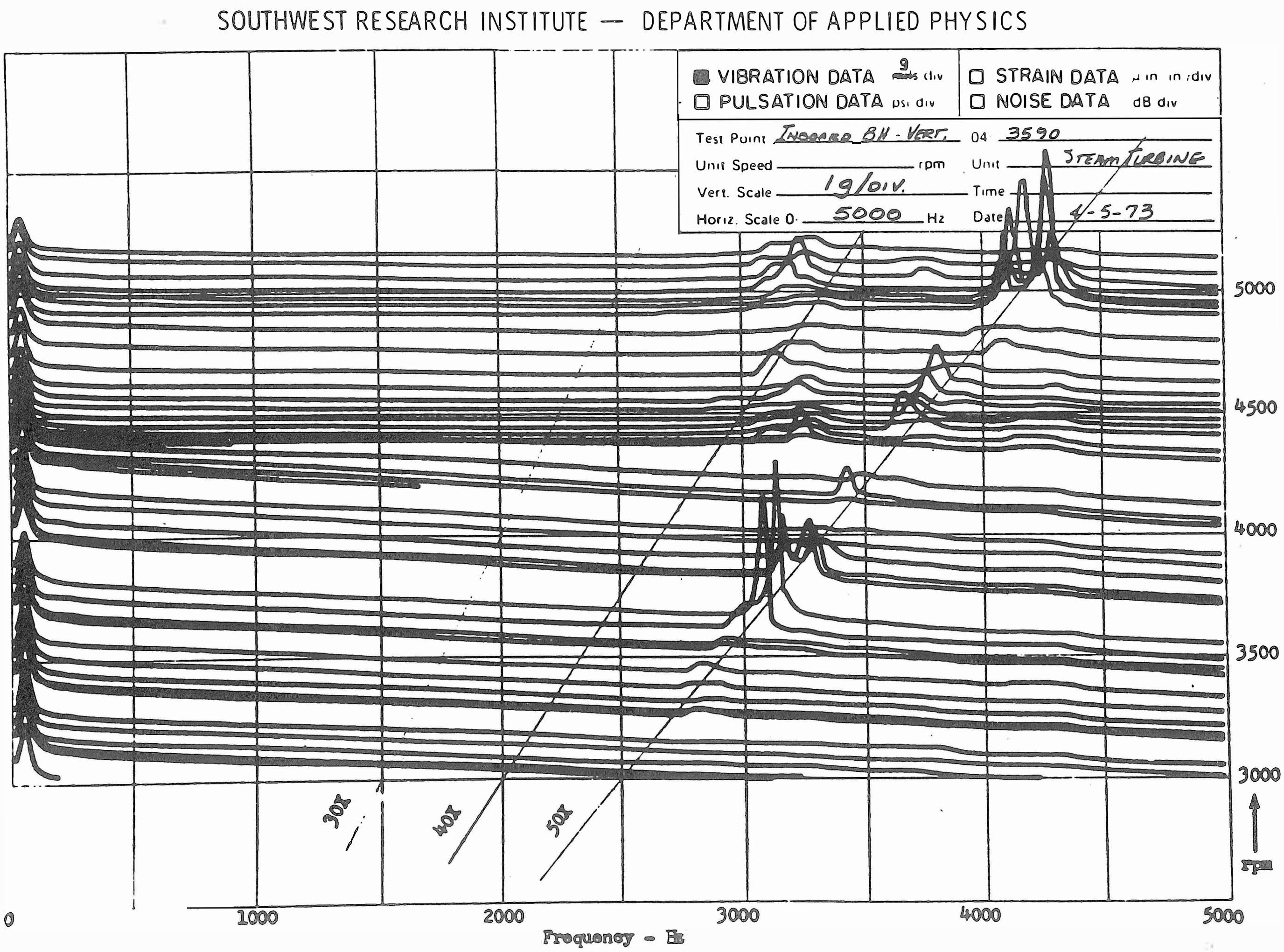



*Figure 5. Inboard bearing housing vibrations versus speed showing nozzle excitation frequencies.*

weights and angles required at each balance plane without inconsistencies. So far, this is not the case, and some judgment on the part of the engineer is still needed. This is primarily due to the nonlinearities and resonant effects in the system. The advantage of the computer is that it can be programmed to handle large amounts of data and perform optimizing analyses. For example, when four balance planes are available, there are 15 different balance plane combinations that could be used for the final balance. To calculate by hand even the two plane balance combinations would be prohibitive from a time standpoint.

The computer program employs the least squares mathematical technique similar to those of Goodman (2) and Lund (3) to solve balance problems by using field data to develop an empirical influence coefficient matrix which will predict the response of the turbine rotor o specific unbalances. The influence coefficient data is obtained in a step-by-step manner by locating an unbalance on each plane individually and obtaining the vibration data (amplitude and phase angle) near each bearing.

A flow chart of the computer program is shown in Figure 11. The program calculates all the possible balancing combinations based on selected test points and speeds. Optimizing criteria are then applied to the calculated balance weights to determine minimum expected vibration, minimum added weight, maximum reduction per added weight, and minimum sum of squares vibration at all test points. Since this leads to numerous solutions, the engineer must still use his judgment to determine which procedure is best. Theoretically, the best balance will be obtained by adding weights at all available balance planes: however, in field balancing, it is desirable to keep the balance planes to a minimum since considerable effort is required to reach some of the balance planes. Most of the time a "perfect" balance is not obtained due to practical limitations such as time and the fact that all that is normally needed is a satisfactory balance within acceptable vibration criteria. What this means, in reality, is that considerable time can be obtained utilizing only one or two balance shots and a minimum of balance planes.

Since the computer can calculate all the balance combinations, it is a simple matter to compare the balance weight solutions and determine which plane or combination of planes will give the minimum vibrations. For this example case, Table 2 summarizes the one plane, two plane, and three plane balance solutions. To reduce the table, some solutions utilizing balance plane 1 were

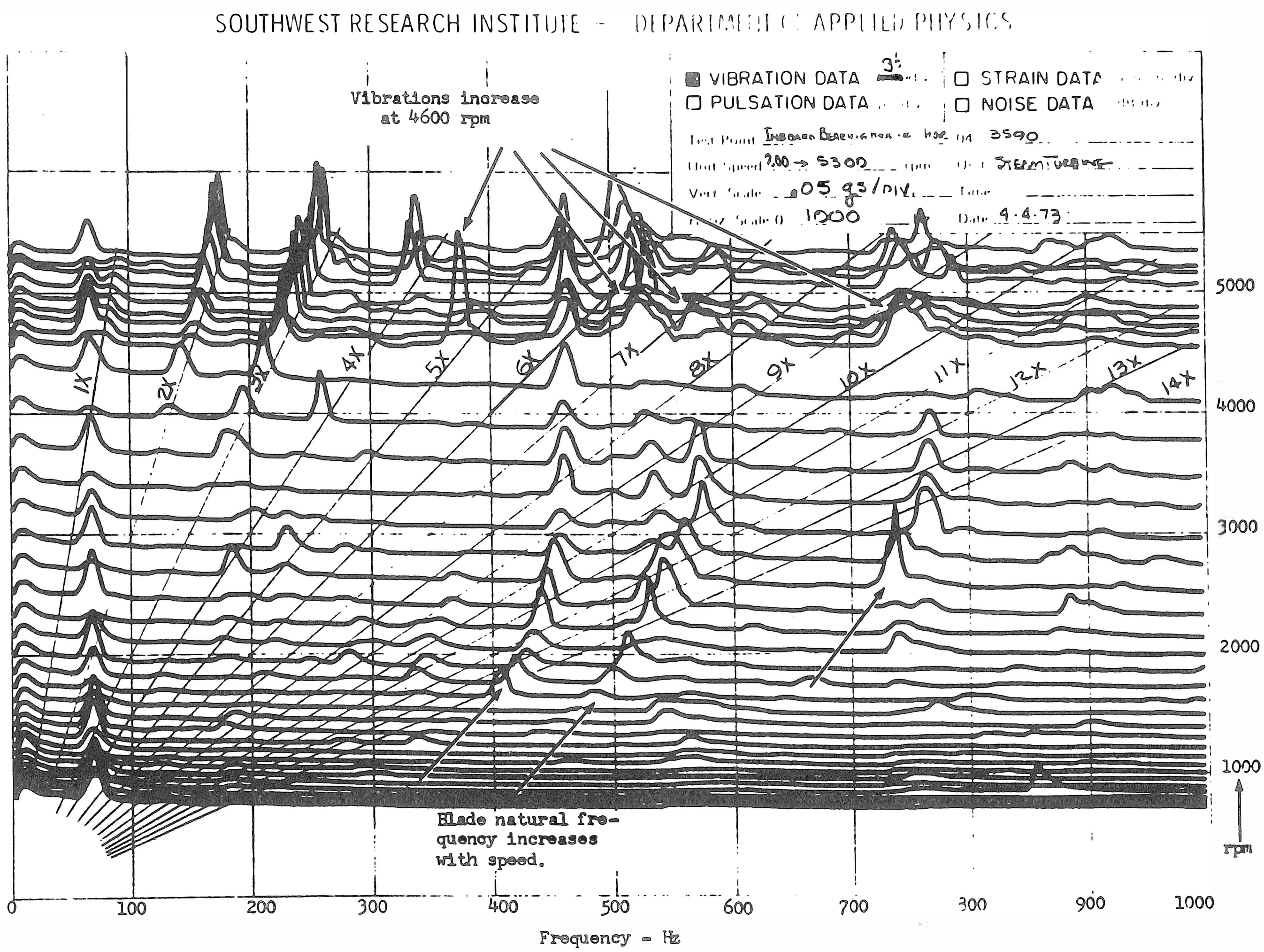



*Figure 6. Excitation of lower blade natural frequencies as a function of speed.*

omitted, even though they were calculated. Balance plane 1, which was the coupling, was left out on the two and three balance plane solutions because it has certain disadvanages when used as a balance plane. The disadvantages are that the weights have to be carefully marked and reinstalled if the units are uncoupled for any reason. The effect of the weight on the system response is also a function of the system misalignment.

Table 2 consists of 40 single plane balance solutions, 18 two-plane solutions and 6 three-plane solutions. Obviously, it would not be practical to do these calculations by hand while the balancing is taking place; however, the computer solution is rather straightforward once it has been programmed. This computer program has been adapted to a time share terminal, and thus could be used right at the machine with a portable teletype machine, if necessary. Normally, the data is transmitted back to the central computer facility by telephone or Telex and run in a batch mode so that all possible combinations can be calculated at one time.

A study of the data in Table 2 shows that:

1. The balance solutions for the different probes are not consistent. For example, for the single plane balance solution, the ratio of maximum to minimum weight required was 148/17, 48/9, and 112/55 for balance planes 1-4 respectively. The deviation in angle was 57, 213, 51, and 23 degrees.

2. If the comparison is made for only the cases where four speeds and four probes were used (runs 3 and 6), the difference between solutions based on the proximity probes and the absolute shaft vectors become closer. Statistically, this means that more consistent data will be obtained if more probes and speeds are used.

| | *BP1* | *BP2* | *BP3* | *BP4* |
|---|---|---|---|---|
| Maximum/Minimum | 33/31 | 23/13 | 85/37 | 109/55 |
| Angle Difference | 44° | 151° | 21° | 15° |

3. For the single plane balance, plane 4 had the lowest expected residual for all cases except runs 3 and 9. Also, the balance angle deviation was smallest for balance plane 4. Based on these criteria, balance plane 4 appears to be the best plane for a single plane balance.

4. If the magnitude of weight that has to be added is considered, one might select balance plane 2 based on

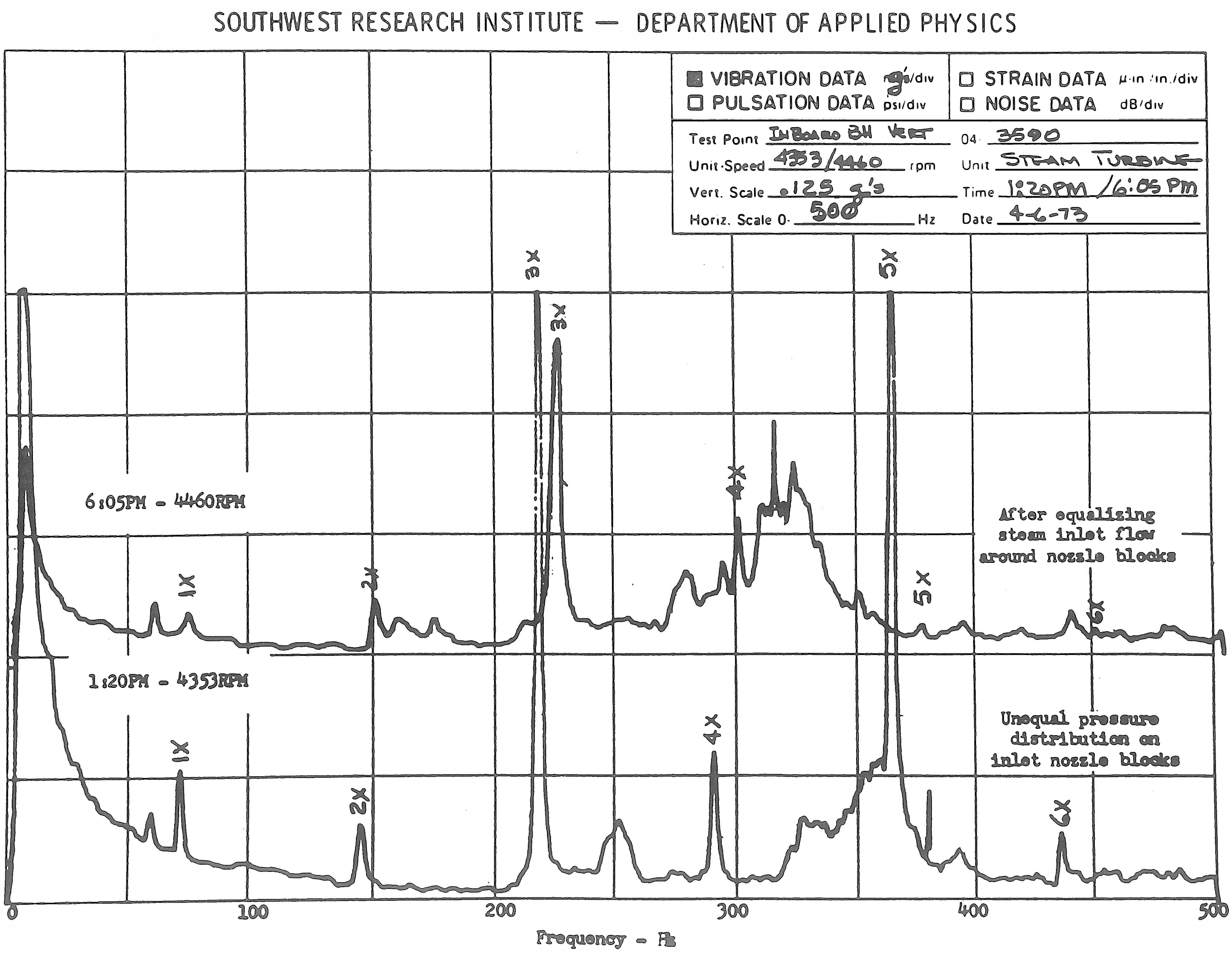



*Figure 7. Comparison of vibrations for partial and full admission steam inlet.*

the data in run 3 which indicated that 13 gm's at 208 degrees could gain the same results as 55 gm's in BP4. However, due to the inconsistent phase angle and the 5:1 ratio between maximum to minimum weight predictions, one mistrusts the balance plane 2 data.

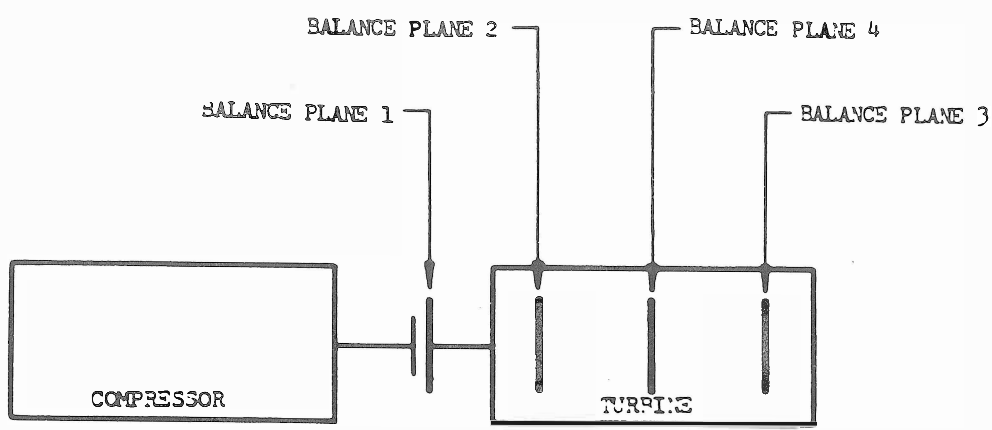
5. When the run 13 and 16 solutions for the two plane balancing are compared, it can be seen that the data are consistent for balance planes 4 and 2, whereas for balance planes 4 and 3, the predicted balance weight on balance plane 3 differs by a factor of 8:1 and by an angle difference of 111 degrees. For balance planes 3 and 2, the data is consistent for all cases.

6. Solutions such as given in runs 11, 12, and 13 for balance planes 4 and 3 where exceedingly large weights are predicted is one problem that is common when using least squares computer analyses. The computer calculates the weight needed and will not recognize an unrealistic weight unless it is programmed to do so. Usually, large weight solutions can be traced back to nonlinear or resonant effects in the data.

7. For the three plane balance cases, considerable differences are noted in the predicted solutions. If only the proximity probe data had been taken and the influence coefficients determined from the data for balance planes 2, 3, and 4, then the three plane solution would be as given in run 19. Imagine the anguish and indecision that an engineer would have if the computer told him to put approximately ½ lb on balance plane 4. When large weight solutions are predicted, the next balance plane will normally have a large weight opposite it. For example, run 19 has 264 gm's at 12 degrees for balance plane 4 and 187 gm's at 198 degrees for balance plane 3, or 187 degrees away.

8. Based upon an assessment of the data given in Table 2, it was decided that the best single plane balance could be obtained at balance plane 4. In trying to determine which weight to install, all ten solutions were plotted in Figure 12. Since only a 23 degree difference was calculated, the location was fairly straightforward, and a weight was selected which was in the middle or average of the predicted amplitudes. Figure 10 gives the results of this balance shot. This first attempt showed considerable improvement; however, a second trim balance was added to move the residual unbalance closer to the center of the polar diagram. The solid figures on the diagram represent the vibrational amplitudes on either end of the turbine, and the open circles in the center show the reduction of the common or static unbalance. The trim balance reduced the vibrational amplitudes through the speed range from 5700 to 7200 rpm to less than 1 mil p-p. This example demonstrates that a satisfactory balance of the turbine was attained with one balance plane.

9. A further examination of Table 2 shows that a better balance could be attained by using balance plane 2 with balance plane 4. Balance plane 2 is preferable to plane 3 since the weight on plane 4 was not significantly




| VIBRATION DATA POINTS | METHOD OF MEASUREMENT |
|---|---|
| INBOARD VERTICAL (IB V) | PROXIMITY PROBE |
| INBOARD HORIZONTAL (IB H) | PROXIMITY PROBE |
| OUTBOARD VERTICAL (OB V) | PROXIMITY PROBE |
| OUTBOARD HORIZONTAL (OB H) | PROXIMITY PROBE |
| INBOARD BEARING HOUSING VERTICAL (IB BH V) | SERVO ACCELEROMETER |
| INBOARD BEARING HOUSING HORIZONTAL (IB BH H) | SERVO ACCELEROMETER |
| OUTBOARD BEARING HOUSING VERTICAL (OB BH V) | SERVO ACCELEROMETER |
| OUTBOARD BEARING HOUSING HORIZONTAL (OB BH H) | SERVO ACCELEROMETER |
| SHAFT ABSOLUTE INBOARD VERTICAL (SA IB V) | PROXIMITY PROBE-SERVO ACCELEROMETER |
| SHAFT ABSOLUTE INBOARD HORIZONTAL (SA IB H) | PROXIMITY PROBE-SERVO ACCELEROMETER |
| SHAFT ABSOLUTE OUTBOARD VERTICAL (SA OB V) | PROXIMITY PROBE-SERVO ACCELEROMETER |
| SHAFT ABSOLUTE OUTBOARD HORIZONTAL (SA OB H) | PROXIMITY PROBE-SERVO ACCELEROMETER |

*Figure 8. Turbine balance planes and data points.*

TABLE 2. BALANCE CORRECTIONS FOR COMBINATIONS OF PLANES

A. Single Plane Balance Solutions

| Run No. | Type of Data | No. of Probes | No. of Speeds | Balance Plane 4 Wt. | Balance Plane 4 Angle | Balance Plane 4 Max Resid. | Balance Plane 3 Wt. | Balance Plane 3 Angle | Balance Plane 3 Max Resid. | Balance Plane 2 Wt. | Balance Plane 2 Angle | Balance Plane 2 Max Resid. | Balance Plane 1 Wt. | Balance Plane 1 Angle | Balance Plane 1 Max Resid. | Lowest Predicted Residual |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | Proximity Vertical | 2 | 4 | 56 | 5 | 1.08 | 24 | 18 | 1.17 | 25 | 231 | 1.16 | 25 | 104 | 1.09 | BP4 |
| 2 | Proximity Horizontal | 2 | 4 | 83 | 358 | 0.62 | 54 | 353 | 0.93 | 9 | 120 | 1.15 | 34 | 81 | 0.90 | BP4 |
| 3 | Proxim. Vert. & Hor. | 4 | 4 | 55 | 8 | 1.10 | 37 | 3 | 1.18 | 13 | 208 | 1.22 | 31 | 90 | 1.06 | BP1 |
| 4 | Shaft Absolute, Vert. | 2 | 4 | 92 | 4 | 1.37 | 56 | 344 | 1.81 | 16 | 130 | 2.16 | 20 | 51 | 2.03 | BP4 |
| 5 | Shaft Absolute, Hor. | 2 | 4 | 110 | 345 | 1.22 | 89 | 340 | 1.69 | 48 | 55 | 2.42 | 148 | 80 | 1.66 | BP4 |
| 6 | Shaft Abs. Vert. & Hor. | 4 | 4 | 109 | 353 | 1.59 | 85 | 341 | 1.84 | 23 | 57 | 2.77 | 33 | 46 | 2.95 | BP4 |
| 7 | Proxim. Vert. & Hor. | 4 | 2 lowest | 67 | 359 | 0.64 | 41 | 9 | 0.87 | 27 | 268 | 0.86 | 17 | 85 | 0.91 | BP4 |
| 8 | Shaft Abs. Vert. & Hor. | 4 | 2 lowest | 71 | 346 | 0.85 | 62 | 327 | 1.36 | 28 | 65 | 1.78 | 42 | 58 | 1.26 | BP4 |
| 9 | Proxim. Vert. & Hor. | 4 | 2 highest | 67 | 4 | 1.11 | 37 | 4 | 1.19 | 12 | 174 | 1.23 | 34 | 91 | 1.07 | BP1 |
| 10 | Shaft Abs. Vert. & Hor. | 4 | 2 highest | 112 | 354 | 1.55 | 90 | 343 | 1.80 | 32 | 67 | 2.81 | 34 | 47 | 2.93 | BP4 |

B. Two Plane Balance Solutions

| Run No. | Type of Data | No. of Probes | No. of Speeds | Balance Planes 4 & 3: Bal. Plane 4 Wt. | Bal. Plane 4 Angle | Bal. Plane 3 Wt. | Bal. Plane 3 Angle | Max Resid. | Balance Planes 4 & 2: Bal. Plane 4 Wt. | Bal. Plane 4 Angle | Bal. Plane 2 Wt. | Bal. Plane 2 Angle | Max Resid. | Balance Planes 3 & 2: Bal. Plane 3 Wt. | Bal. Plane 3 Angle | Bal. Plane 2 Wt. | Bal. Plane 2 Angle | Max Resid. | Lowest Pred. Resid. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 11 | Proximity Vertical | 2 | 4 | 272 | 347 | 197 | 160 | 0.47 | 77 | 346 | 48 | 248 | 0.74 | 68 | 330 | 62 | 241 | 0.85 | BP4&3 |
| 12 | Proximity Horizontal | 2 | 4 | 199 | 358 | 111 | 173 | 0.37 | 98 | 357 | 18 | 267 | 0.54 | 88 | 350 | 31 | 261 | 0.81 | BP4&3 |
| 13 | Proxim. Vert. & Hor. | 4 | 4 | 255 | 352 | 173 | 167 | 0.59 | 90 | 358 | 34 | 255 | 0.87 | 79 | 348 | 47 | 252 | 1.01 | BP4&3 |
| 14 | Shaft Absolute, Vert. | 2 | 4 | 86 | 340 | 29 | 63 | 1.15 | 119 | 356 | 41 | 259 | 0.81 | 101 | 341 | 51 | 266 | 1.36 | BP4&2 |
| 15 | Shaft Absolute, Hor. | 2 | 4 | 161 | 358 | 71 | 202 | 0.98 | 112 | 357 | 42 | 318 | 0.84 | 136 | 336 | 61 | 269 | 1.45 | BP4&2 |
| 16 | Shaft. Abs. Vert. & Hor. | 4 | 4 | 102 | 336 | 22 | 56 | 1.37 | 135 | 352 | 39 | 272 | 1.26 | 116 | 346 | 48 | 288 | 1.59 | BP4&2 |

C. Three Plane Balance Solutions

| Run No. | Type of Data | No. of Probes | No. of Speeds | Bal. Plane 4 Wt. | Bal. Plane 4 Angle | Bal. Plane 3 Wt. | Bal. Plane 3 Angle | Bal. Plane 2 Wt. | Bal. Plane 2 Angle | Max Resid. |
|---|---|---|---|---|---|---|---|---|---|---|
| 17 | Proximity Vertical | 2 | 4 | 380 | 17 | 300 | 201 | 42 | 153 | 0.38 |
| 18 | Proximity Horizontal | 2 | 4 | 224 | 359 | 150 | 178 | 14 | 107 | 0.34 |
| 19 | Proxim. Vert. & Hor. | 4 | 4 | 264 | 12 | 187 | 198 | 29 | 179 | 0.46 |
| 20 | Shaft Absolute, Vert. | 2 | 4 | 96 | 355 | 17 | 345 | 38 | 269 | 0.67 |
| 21 | Shaft Absolute, Hor. | 2 | 4 | 152 | 339 | 84 | 113 | 58 | 345 | 0.70 |
| 22 | Shaft Abs. Vert. & Hor. | 4 | 4 | 98 | 337 | 46 | 27 | 38 | 296 | 1.07 |

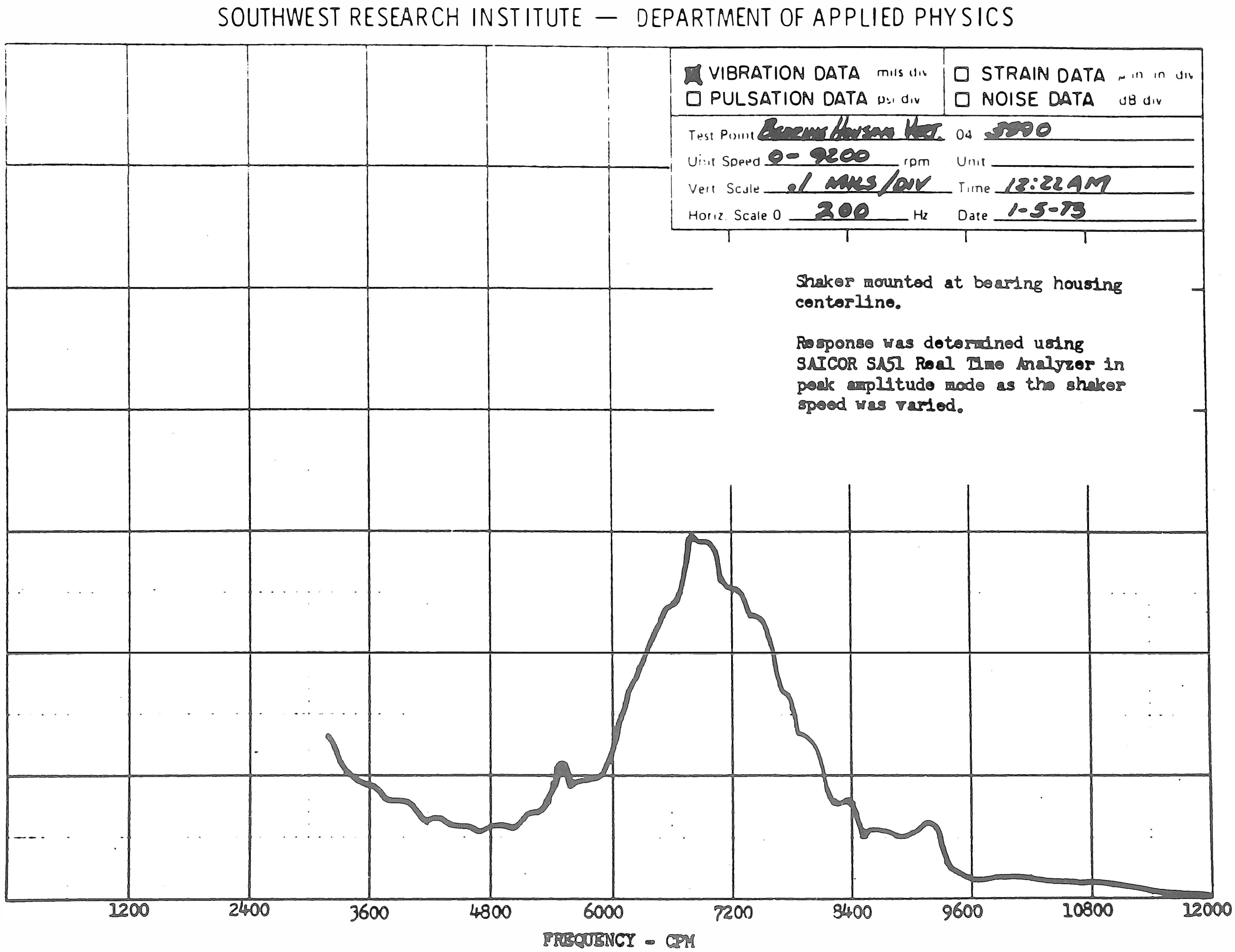



*Figure 9. Resonance frequency of bearing housing determined by shaker test.*

changed, plus the fact that the weights are unrealistic for plane 4 when plane 3 is used. This means that additional improvement in balance could be attained without adjusting the weight on the fourth balance plane but merely by adding weight to the second balance plane. Figure 12 indicates the balance weight region for the second balance plane which is centered at approximately 270 degrees, and a weight magnitude of approximately 18-47 grams required. This was not attempted in the field due to the time limitation, as startup was imminent. It would have been the logical choice for the next balance shot; however, the vibrations were already within specified limits.

An additional advantage to the computer technique is that it has the ability to predict the expected vibrations for the actual added weights. This is important since many times the exact weight cannot be put into the desired spot. The expected residual vibration for the proximity probe, including runout, was calculated for the actual added weight in the fourth balance plane. This data is compared with the actual measured vibration data in Table 3.

## CASE III: ACOUSTICAL EXCITATION OF SHAFT VIBRATIONS

A gas turbine driven compressor unit had several reoccurring problems, including: (1) Slipping of compressor coupling on shaft; (2) Bearing failures in turbine; (3) Rotation of diaphragm; (4) Failure of inlet guide vanes; (5) Excessive noise levels. These units ran from 13000 to 14200 rpm. The compressors did not operate on their original design point due to changes in plant operating characteristics after the units were purchased. Due to this, high inlet guide vane settings were necessary to control pressure rise.

The critical speed map for the compressor is given in Figure 13. The mode shapes for the first four criticals are given in Figure 14. It can be seen from the mode shape that the anti-nodes for the fourth critical speed are near the two impellers. The unit runs between the third and fourth criticals.

A field study was conducted to measure the vibrations, pulsations, and noise of the units. Figures 15-18 summarize the significant results. At certain inlet guide vane settings, the shaft vibrations would drastically increase, and the noise in the compressor and piping would also increase. Shaft vibrations and pulsations were recorded simultaneously. Figure 15 is the spectral analysis of the shaft vibrations versus the guide vane settings. The unit speed was held constant and the guide vane settings were gradually changed. Each vertical step represents about 5 seconds. At the lower guide vane settings, the predominant frequency was at the running speed. When the guide vane setting was increased above 45 degrees, a 1 mil vibration component at 252 Hz (15120 cpm) suddenly appeared. This frequency matched the calculated fourth critical. Notice that the amplitude is as large as the running speed component. Suction pulsation recorded during this same time period revealed the cause of these nonsynchronous vibrations. Figure 16 shows that very little pulsation was present at the compressor running speed (220 Hz) or 252 Hz; however, when the guide vane setting was increased above 45 degrees, a large acoustical resonance was excited at 252 Hz. The amplitude eventually reached 110 psi p-p. This acoustical resonance caused the shaft to vibrate at its fourth critical which was above running speed. The fact that the impeller was at an anti-node helped the energy to couple into the lateral shaft vibrations.

Figure 17 is a photograph of the complex wave of the suction pressure and the shaft vibration just as the acoustical resonance occurs. Notice the beating of the shaft vibration wave at a 30 Hz beat frequency. This is expected since the two frequency components are close (220 and 250 Hz). Even though the suction piping had an acoustical resonance, the pulsations did not pass through to the discharge. Figure 18 gives the discharge pressure during the same time interval.

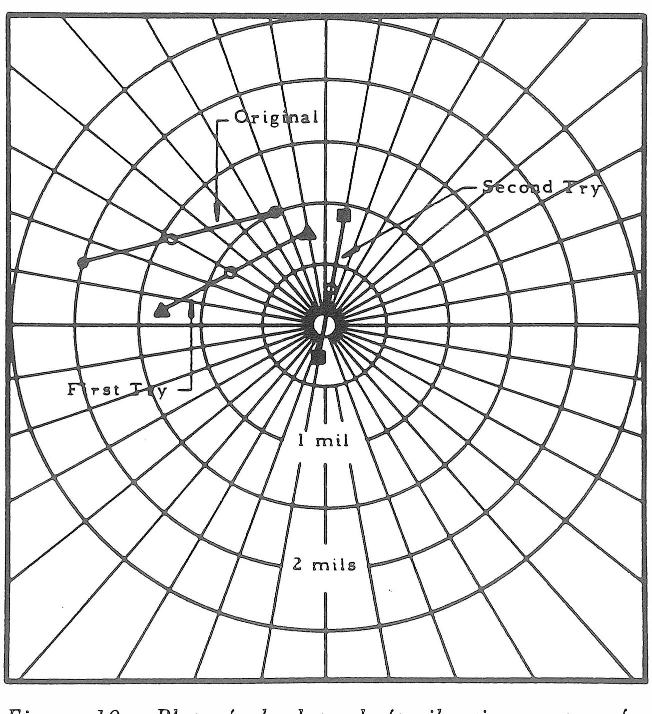



*Figure 10. Plot of absolute shaft vibration vectors for steam turbine.*

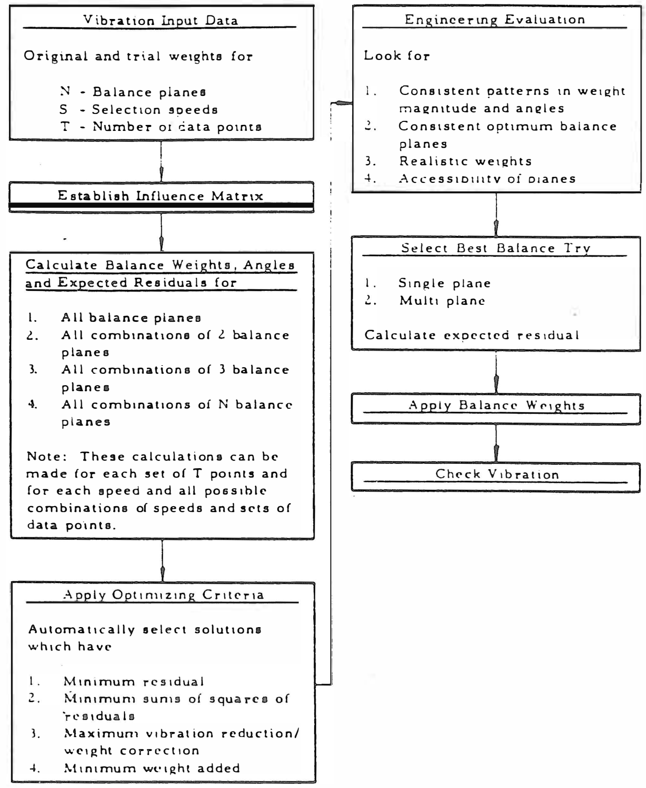



*Figure 11. Logic flow diagram of optimizing balance program.*

Since the excessive vibrations were found to be related to the high negative guide vane settings, avoidance of these operating conditions eliminated the problems. Another solution would have been to change the acoustical characteristics of the suction piping. An orifice plate inserted into the suction line drastically reduced the suction pressure pulsations and vibrations. The wheels in these units were changed to more properly match plant pressure and flow requirements.

## CONCLUSIONS

This paper has discussed three case histories of vibration problems in turbomachinery. These three cases illustrate the types of problems which can occur when the analytical solutions that are available are not sufficient to accurately take into account the system variables. Based upon the data obtained and the analysis of these problems, certain conclusions can be made:

1. By measuring vibrations on the bearing housing of a turbine, it is possible to determine if the blades are vibrating at their natural frequencies. This type of data analysis requires the use of real time spectrum analyzers and automatic data monitoring instrumentation such that the vibration signatures can be displayed as a function of speed as in a Campbell diagram. These techniques can be applied to other applications to measure low energy signals.

2. The balancing of a rotor is more difficult when a bearing housing support or rotor resonance is near the running speed range. This is because the vibration amplitude and phase data will have inconsistencies which

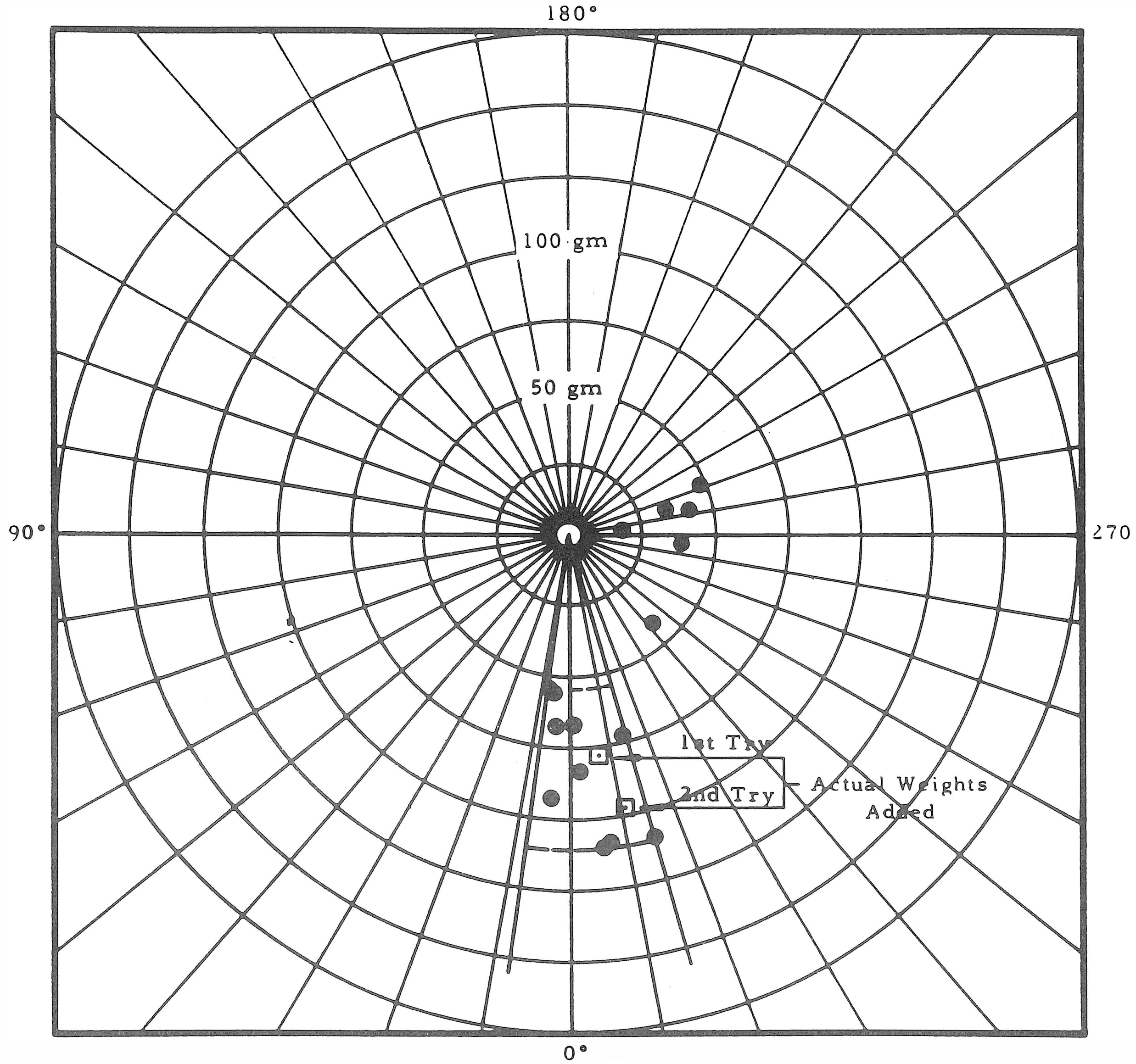



*Figure 12. Balance weight corrections using different probes.*

hinder the solution for the balance weights. Also, non-linear effects in a system complicate the solution for the balance weights.

3. When complete vibration data is taken, including relative shaft vibration, bearing housing vibrations, absolute shaft vibrations, plus relative and absolute shaft orbits, there are many combinations of probes which theoretically can be used for the balancing procedure. In practice, one normally selects one or more sets of data and uses these to perform the graphical and analytical solution for the balance weights for a single or two plane balance. This technique can lead to unrealistic solutions and can result in costly trial and error balance procedures.

TABLE 3. COMPARISON OF PREDICTED TO MEASURED VIBRATIONS FOR HORIZONTAL PROXIMITY PROBES

| Speed | Original | | | | Predicted | | | | Measured | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Position 1 | | Position 2 | | Position 1 | | Position 2 | | Position 1 | | Position 2 | |
| rpm | Amp. | Angle | Amp. | Angle | Amp. | Angle | Amp. | Angle | Amp. | Angle | Amp. | Angle |
| 7246 | 0.64 | 173 | 1.4 | 240 | 0.98 | 206 | 0.42 | 229 | 0.74 | 280 | 0.44 | 158 |
| 6901 | 0.53 | 262 | 1.2 | 214 | 0.52 | 286 | 0.43 | 168 | 0.86 | 276 | 0.43 | 139 |
| 6398 | 1.2 | 254 | 0.64 | 180 | 0.64 | 271 | 0.32 | 146 | 0.80 | 269 | 0.38 | 108 |
| 5735 | 1.1 | 237 | 0.38 | 130 | 0.57 | 268 | 0.29 | 140 | 0.59 | 270 | 0.38 | 83 |

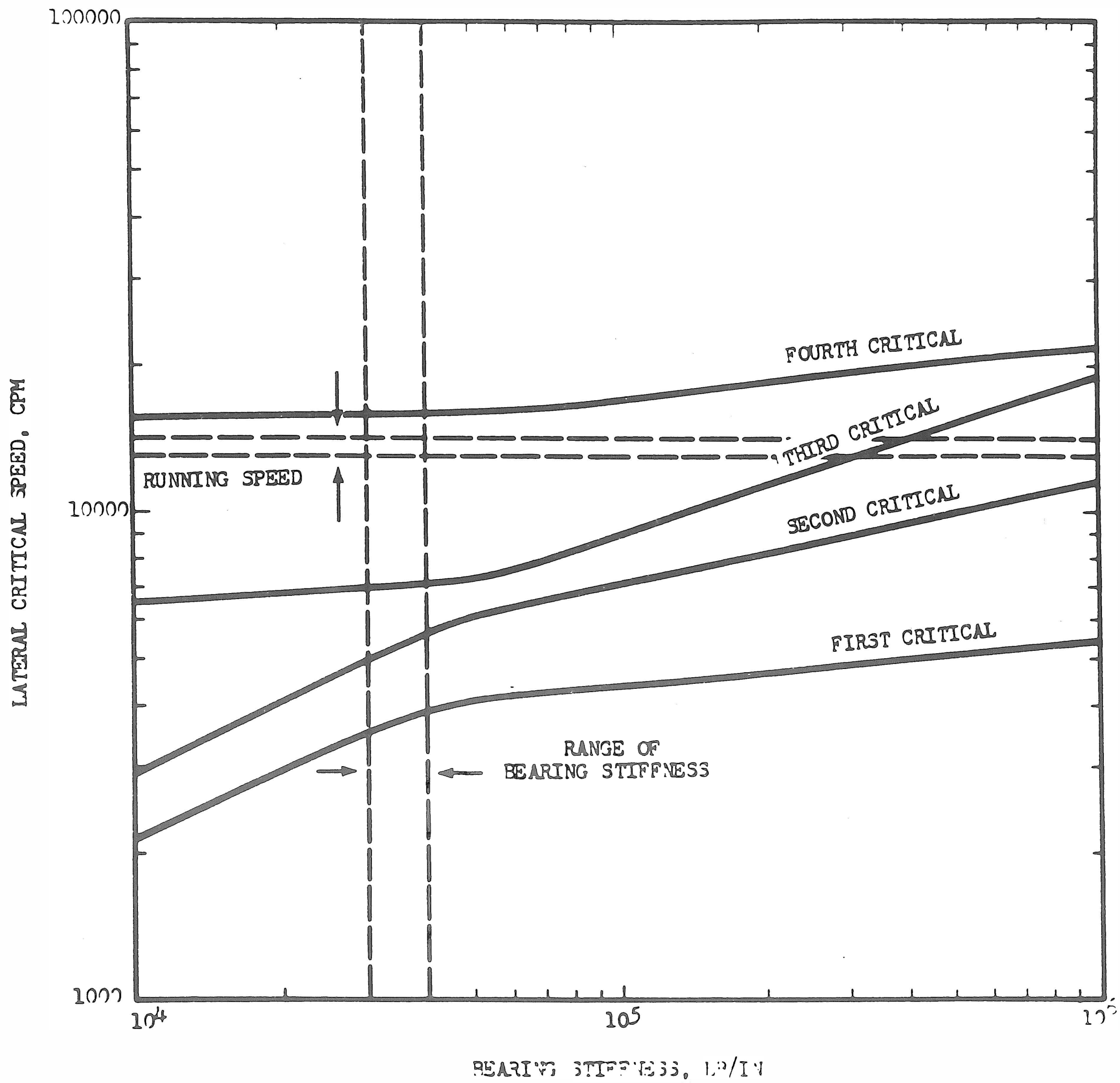



*Figure 13. Critical speed map.*

4. A digital computer program has been developed which can be used for multiplane balancing for any combination of balance planes, test speeds, and any number of vibration inputs. This program considers all possible balancing combinations and helps the engineer to determine the best balancing procedure. Several optimization schemes have been programmed to help eliminate the inconsistencies caused by nonlinearities and by resonances in the rotor and support structure.

5. The evaluation of all possible sets of vibration data leads to a zone of correction weights and angles for particular balance planes. The engineer must then decide the amount of the weight and the angle. Statistically, the more speeds and probes that were used in the analysis, the smaller the correction weight zone.

6. Nonsynchronous shaft vibrations can be excited by acoustical resonances in the piping system. This type of problem illustrates the desirability of performing detailed spectral analysis of vibration problems so that the exact cause can be established.

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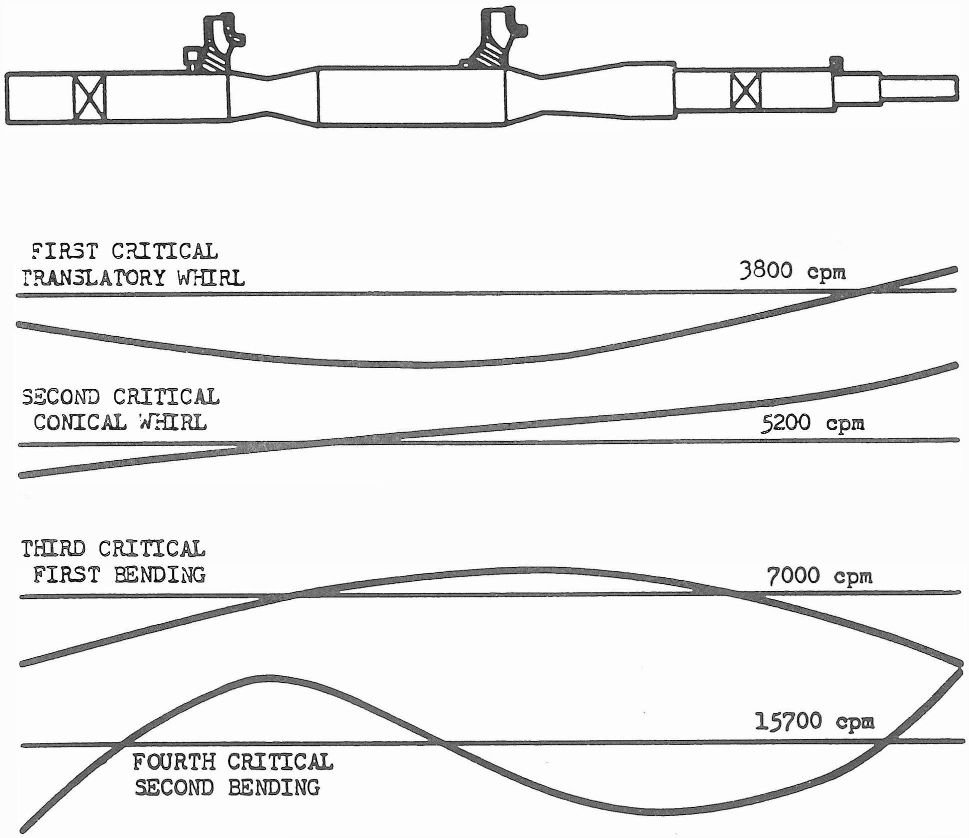



*Figure 14. Vibration mode shapes.*

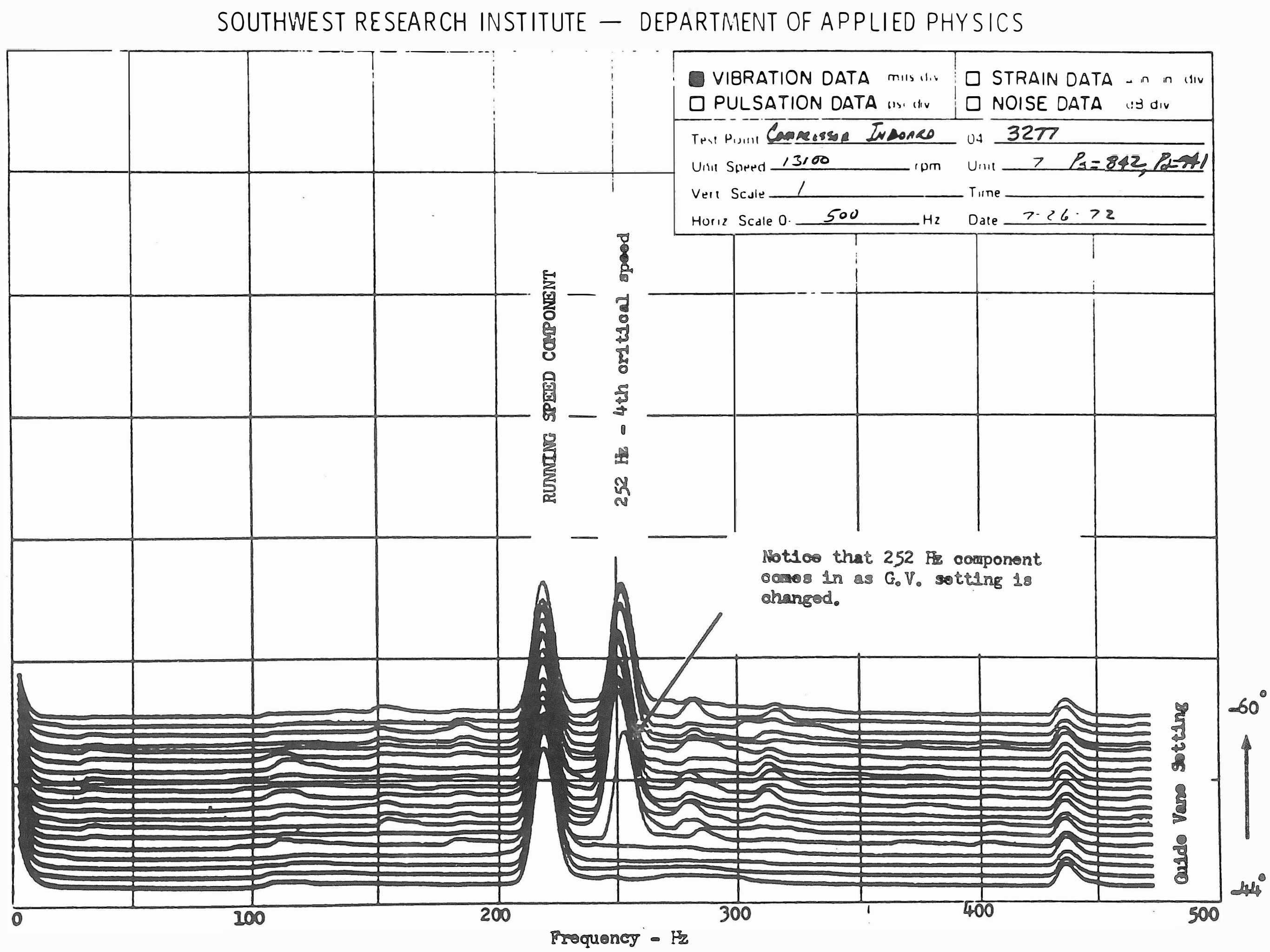



*Figure 15. Compressor shaft vibrations versus guide vane setting.*

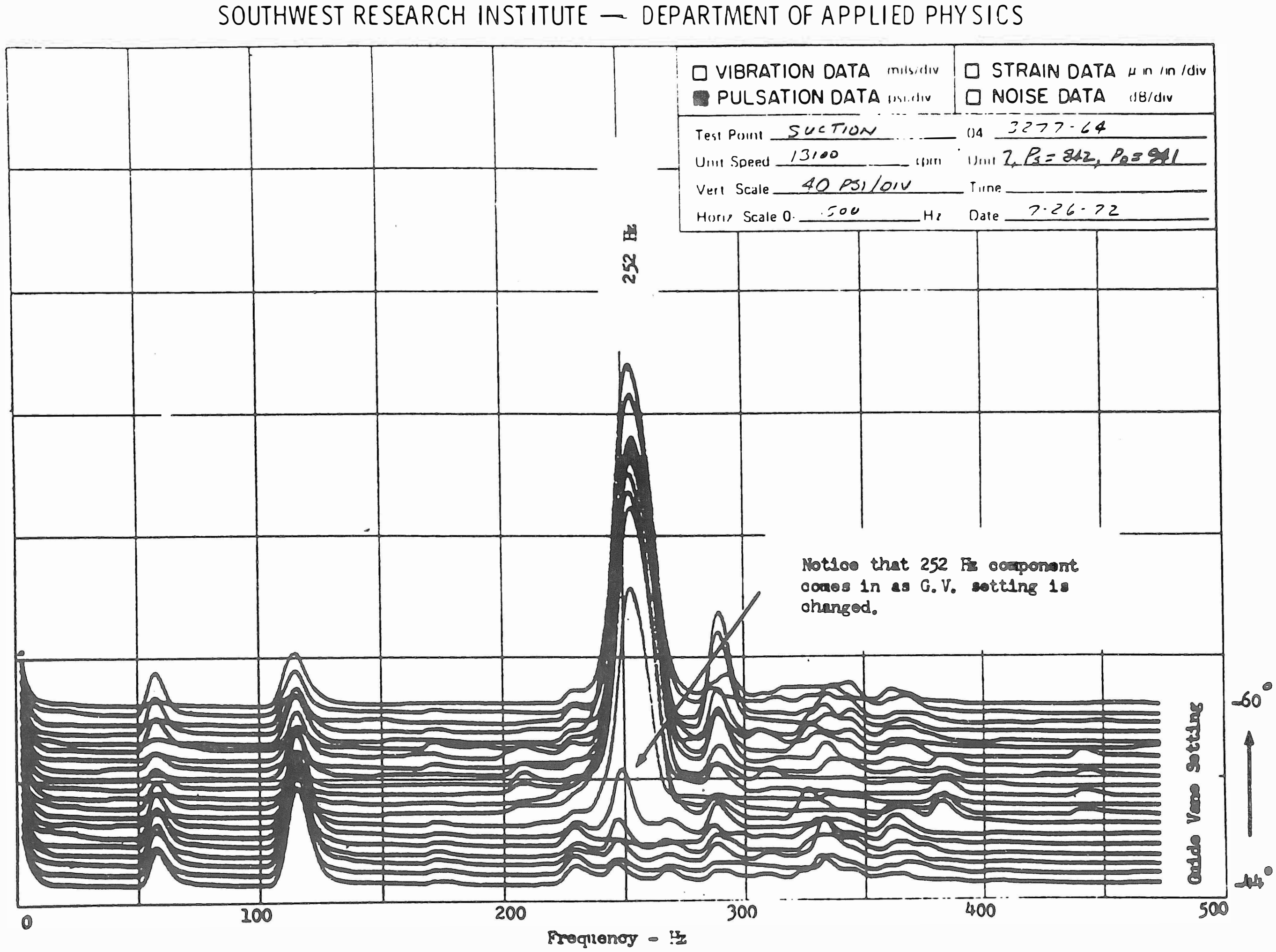
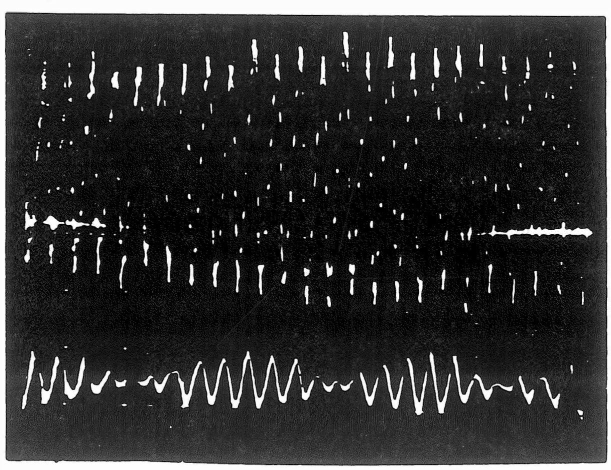



*Figure 16. Compressor suction pulsation versus guide vane setting.*



CHANNEL 1 10 PSI/DIVISION

CHANNEL 2 5 MILS/DIVISION

TIME BASE 10 MS/DIVISION

*Figure 17. Suction pressure and shaft vibration as acoustical resonance occurred.*

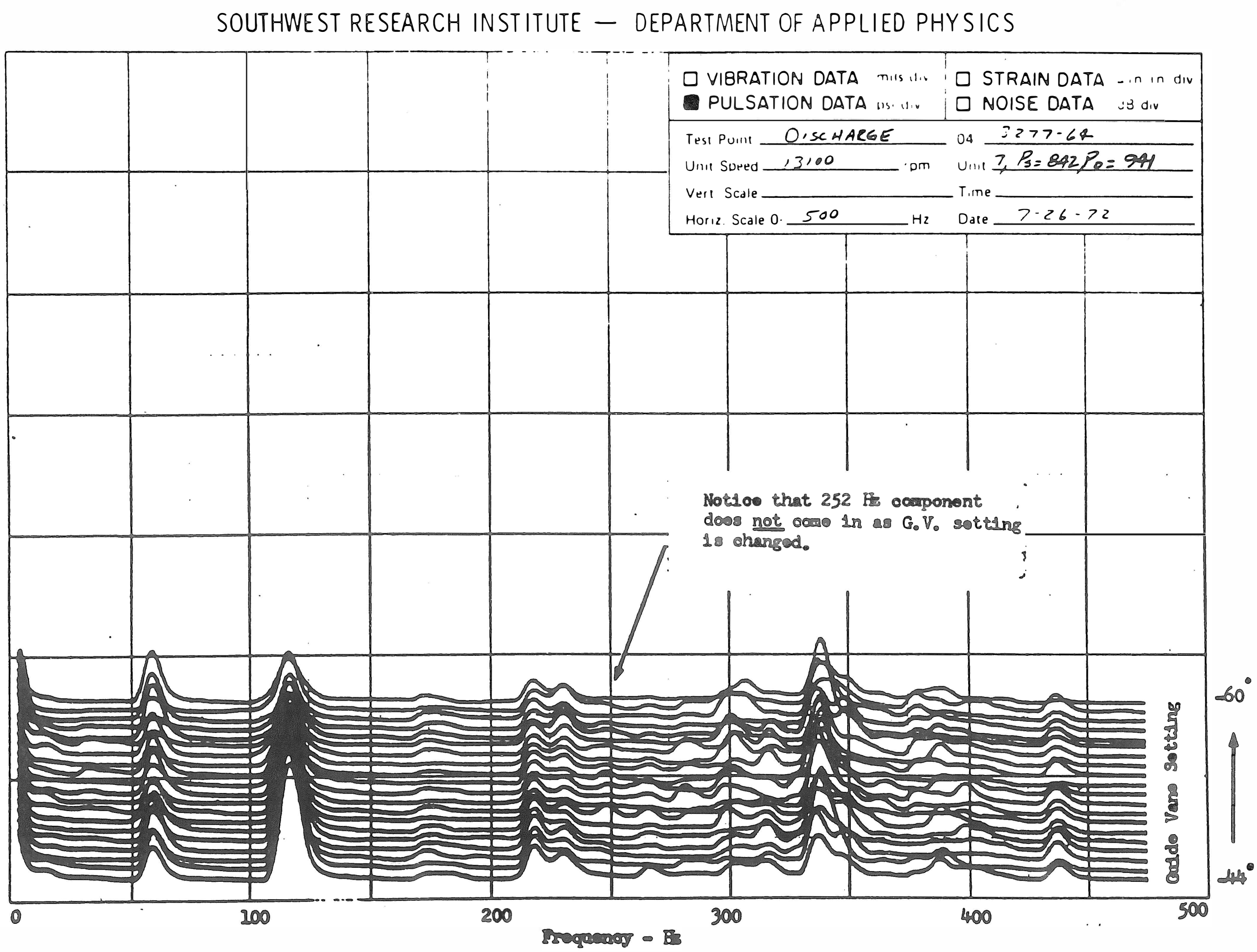



*Figure 18. Compressor discharge pulsation versus guide vane setting.*

## ACKNOWLEDGMENT

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