Field Balancing Experiences

By Brian Howes and Bryan Long

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Introduction

This paper presents the results of a multi-plane, multi-speed balancing job performed by the authors of the Edmonton Power Rossdale Generating Station. Using the influence coefficient approach, a 60 megawatt steam turbine was balanced in-place, at two shaft critical speeds, as well as operating speed, using seven balance planes, and 10 noncontacting eddy current type vibration transducers.

Previous attempts by others at balancing using case vibrations had proved disappointing. This lack of success appears to have been because balancing was attempted at run speed only. The lack of sensitivity of the case readings may have been a contributing factor also.

Balancing of the unit was first attempted by the authors in December of 1979. Strict time limitations due to peak winter power demand, and the inability to access three balance planes resulted in improved but only marginally acceptable vibration levels.

The second balance attempt in June of 1980 followed a major overhaul which included cleaning out balance weight holes and re-alignment of the unit.

An attempt to reuse the December balance data was unsuccessful, apparently because of the effects of changes in alignment.

Description of the Machine

The machine is a two-case steam turbine (high pressure (HP) and low pressure – intermediate pressure (LP – IP) rotors) driving a generator.

The unit was installed in 1968 and had about 140,000 hours of operation without major overhaul in December of 1979. The requirement for the machine in recent years has been to provide peak load capacity during the morning and evening electricity demand peaks (two-shifting).

However, the unit had always been rough during coastdown, especially at the first critical speed (1400 cpm). In recent years it had gotten worse. The shutdown procedure was designed to decelerate the unit as rapidly as possible through the critical speeds by breaking vacuum on the LP section at about 3300 RPM and tripping the machine.

When the machine was shut down without breaking vacuum in December, 1979, vibrations of over 40 mils were observed don various probes. The machine “locked-in” on 1400 RPM at which point the deceleration stopped until the vacuum on the LP was broken.

After the final balance run of June, 1980, the same shut down procedure yielded levels of only 6 to 10 mils pp at various probes at 1400 RPM and no “lock-in” was observed.

Balancing Objectives

Consultations with plant personnel produced the following list of objectives presented in order of priority.

1. Reduce the vibration levels at the first critical (1400 cpm) so that the machine could be “two-shifted” without concern for its integrity.

2. Reduce vibration levels at run speed (3600 cpm) as much as possible.

3. Maintain or reduce vibration levels at the second critical (2400 cpm).

4. Emphasize reductions in the area of the thrust bearings and the governor end.

Other practical restraints were present which is typical for this type of balancing work.

1. Time to complete balancing was limited by demand for electrical generating capacity.

2. Time required to gather a complete set of balancing data without unexpected problems was about 3 to 4 days.

3. A finite number of holes were available for balancing at each plane. Weights had been added in some holes during previous balancing operations by others.

4. Thermal effects on the machine reduce the accuracy of vibration readings or increase the uncertainty of the readings.

Test Equipment

Although case vibration data were recorded for the “as found” and “as left” tests, using a seismic pickup, shaft motion relative to the housing was used to balance the machine.

Pairs of eddy-current proximity probes were mounted at 90B spacing along the machine as shown in the schematic diagram in Figure 1. In all, ten probes were used plus an eleventh probe for phase reference purposes mounted at the exciter end of the machine.

A two-channel tracking filter was used to measure amplitude and phase at each of the 10 probes. This instrument was accurate to $\pm$ 1 RPM in speed, $\pm$ 1 degree in phase and $\pm$ .01 mils pp in amplitude.

Data were recorded manually.

Machine Layout

A sketch of the machine is given in Figure 2 which shows the bearing locations and the balance planes.

Each balance plane had 12 holes in the HP rotor, and 24 holes in the LP-IP rotor.

No balance planes were available in the generator because the generator was pressurized with hydrogen. A decision was made to not depressure the generator unless the use of the other seven balance planes failed to produce acceptable results.

Balancing Program

The field balancing procedure used was that of influence coefficients with least squares minimization *. The required calculations were carried out using an in-house computer program “RMS4”. This program requires as input the “as-found” vibrations, the trial weight vibration data and information about the trial weights. Slow roll or runout data can also be entered at the analyst’s option. The analyst can also weight the data as desired. The program calculates the set of balance weights required to minimize the root mean square of all the vibration readings. It also predicts the residuals for that set of correction weights. Features allow the analyst to use influence coefficients derived independently and to ask for the residuals for any set of correction weights.

The “data weighting” feature allows the analyst to tailor the balance weights so that one mode of vibration is reduced more than another. This requires “operator insight”. As such, our use of the influence coefficient approach to balancing has some of the qualities usually ascribed to modal balancing.

* “A Least Squares Method for Computing Balance Corrections”, Thomas Goodman, Journal of Engineering for Industry, August, 1964.

Test Procedure

A. General Procedure

Initially the machine was started and run under 20 megawatt load.

Base line data were gathered from each probe for the “as found” system over a period of time (typically 6 hours or more). When changes in amplitudes and phases tended to stop, the load was removed and the data were taken near the second and first critical speeds.

The machine could not be held at a constant speed at the critical speeds of 2400 and 1400 RPM because vibrations would have been too great. As a result, speeds of 2300 and 1200 cpm were selected to gather the critical speed data. The selection of data gathering speeds was done to satisfy the following conditions:

i) Vibration levels must remain at a level such that they would not cause damage to the machine;
ii) Phase angles should not be changing rapidly with speed changes;
iii) The speed must be close enough to the critical speed that the mode is fully developed to get good balance sensitivity.

The recording of the data at 3600 cpm was straightforward. At 2300 and 1200 cpm, however, the speed of the unit had to be maintained manually. This turned out to be very steady during some tests, but in others the data were recorded as the unit passed through the desired speed. In such cases an interpolated value of vibration amplitude and phase was recorded for 2300 cpm or 1200 cpm. An attempt was made to keep the range of speeds to less than $\pm$ 50 RPM.

Trial weights were selected for each balance plane. The maximum weight that would fit in a balance hole was typically used. The location of the trial weights was selected with a view to reducing vibration levels but this did not prove to be successful in most cases because of the conflicting requirements at the three speeds.

Trial weight data were gathered in a similar fashion to the “as found” data. The run time to thermal stability was not so great when the machine was only down for a short time to install trial weights.

B. December, 1979

Only balance planes 3, 4, 5, and 7 were used.

C. June, 1980

In the beginning, new data were gathered at balance planes 1, 2 and 6. These data were combined with data from December for planes 3, 4, 5 and 7 to calculate a set of balance weights.

When results were evaluated as very poor, the weights were removed and a trial weight was put in to plane 3 and new data were taken. These showed that there were significant differences as discussed below.

The remaining balance planes were also redone.

Results

As a first step in attempting to balance the turbine in June, 1980, “as found” vibration readings were taken with the machine thermally stabilized. Trial weight runs were then made for each of the new balance planes (BP 1, 2, 6). In all cases readings were taken at 1200 RPM, just below the first critical, at 2300 RPM, just below the second critical and at 3600 RPM, normal run speed. Each run required several hours to remove the old trial weight, to insert the new and to achieve thermal stability. Data from the new planes were combined with the influence coefficients derived during December, 1979, balancing for the four original balance planes.

The program RMS4 was then used to calculate the set of balance weights required to minimize the residual RMS vibration. In this and other calculations the data were selectively weighted. The 2300 RPM data was weighted by 0.5 since passage through the second critical was relatively smooth. Readings from the governor end (test points 9 & 10) at 3600 RPM were weighted by 2.0 since this area was subjectively “rough”. Weights very close to those calculated were installed. Residual vibrations measured later showed generally increased vibration levels. A comparison of measured and calculated residual vibrations and influence vectors indicated very poor correlation; see Figure 3. It was concluded that the old influence coefficients were probably invalid.

The next step was to remove all balance weights and to go back to balance plane three (BP3) to generate new influence coefficients for that plane. These were subsequently compared to the old BP3 influence coefficients, as shown in Figure 4 and were indeed quire different. Actually, it was probably unreasonably optimistic to hope that the old coefficients would be valid even though the realignment had been described as “minor”. Figure 5 shows the change in vibration levels from December, 1979, to June, 1980, before and after the realignment.

Trial weight runs were then made for the remaining old planes and a set of correction weights was installed based on the new influence coefficients. The resulting improvement was not large (see Figure 6, balance run 2 results) partly due to the fact that one of the correction weights was later discovered to have been incorrectly placed. Based on the correction weights actually installed, some of the measured and calculated influence vectors are compared in Figure 7. The correlation was not outstanding but the influence vector magnitudes were reasonable and the errors in phase were quite consistent.

Based on these residuals and the same influence coefficients, a second correction weight iteration was calculated. Because of the consistent angular discrepancy noted above, these weights were rotated from the position calculated by the program. This time substantial improvements were achieved as shown in Figure 6. The reduction in vibration levels at the critical speeds (1400 and 2400 RPM) was even greater than the decreases at 1200 RPM and 2300 RPM. The residuals at 3600 RPM were judged to be reasonable. Correlation between measured and calculated influence vectors, Figure 8, was as good as that achieved during the preceding balance run, Figure 7. It would probably have been possible, therefore, to achieve a further reduction. This was not deemed essential, however, since the shaft orbits were now small enough compared to the bearing clearances of about 0.018 in.

Improvements achieved in case readings from December, 1979, to June, 1980, are shown in Figure 9.

Conclusions

1. Case and shaft-relative-to-case vibrations were reduced at all speeds.

2. Figures show the reductions at 3600, 2300 and 1200 RPM. Results at the critical speeds, particularly 1400 RPM are not shown because the machine had to be run rapidly through those speeds in the "as found" condition.

3. The shut down after the final balance run was not done in the normal fashion (ie: by breaking vacuum at 3300 RPM) because of an instrumentation problem that tripped the machine. The shut down, even at a slow deceleration rate, was remarkably smooth compared to a similar procedure done on the "as found" unit.

4. The changes in alignment produced both large changes in the "as found" vibration as well as in the influence coefficients.

5. The "thermal sensitivity" of the machine was reduced by the alignment changes. That is, the amount vibrations changed with time after a cold start-up was reduced. Also the time over which noticeable changes occurred, was reduced by the alignment changes.

6. Some of the residual vibrations still comes from misalignment.

7. Calculated influence vectors and residual vibrations differed significantly from measured values. Contributing factors were:

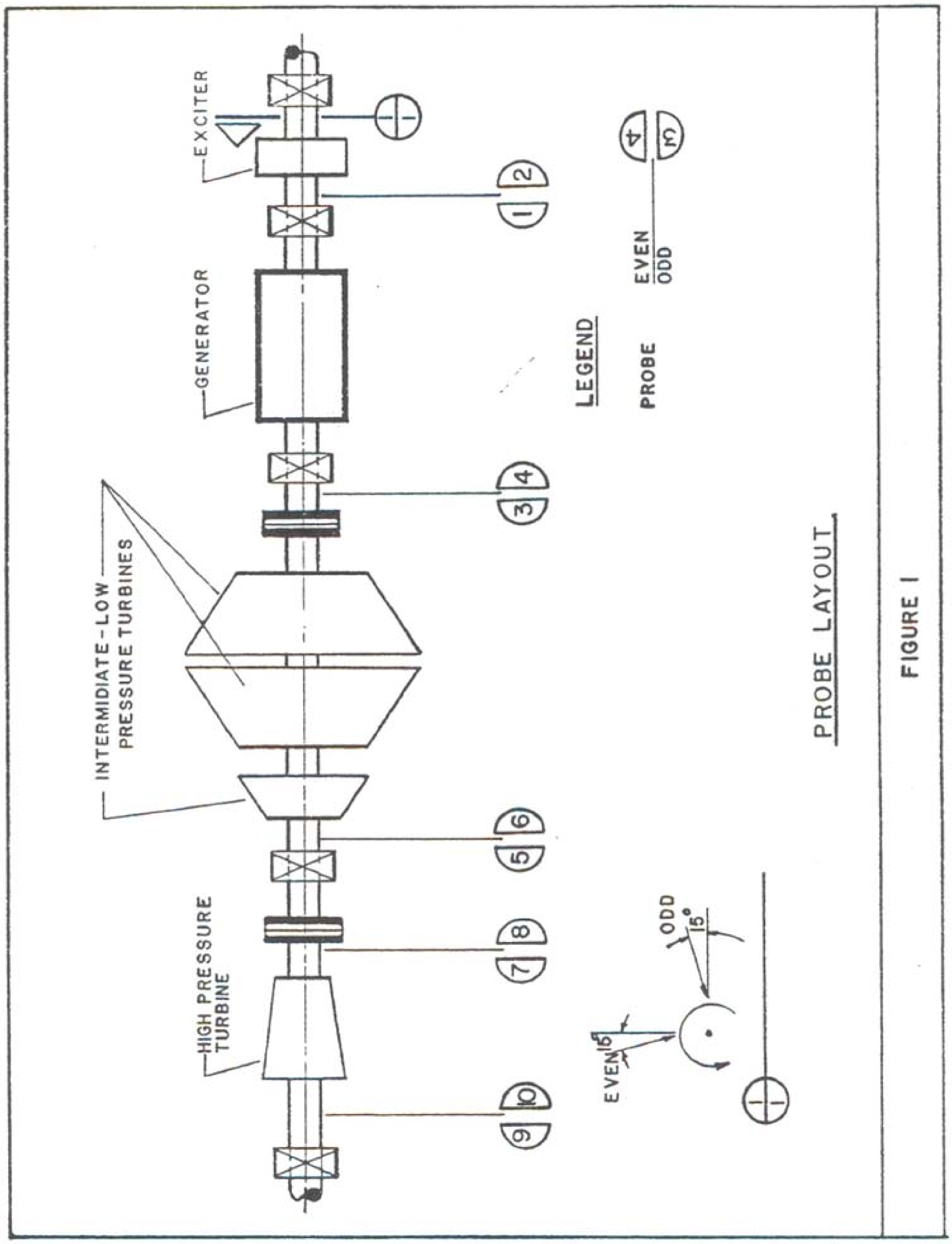
   a) Non-constant speed during data collection at 2300 and 1200 cpm.
   b) Variations in alignment with time due to changes in temperature of the machine.
   c) Variations of the influence coefficients with vibration amplitude due to nonlinear characteristics of journal bearings.

An Aside

An aspect of balancing large machines outside the area of vibrations and computer programs involves supervision of the personnel actually working on the machine changing weights (typically a hot, tiring job).

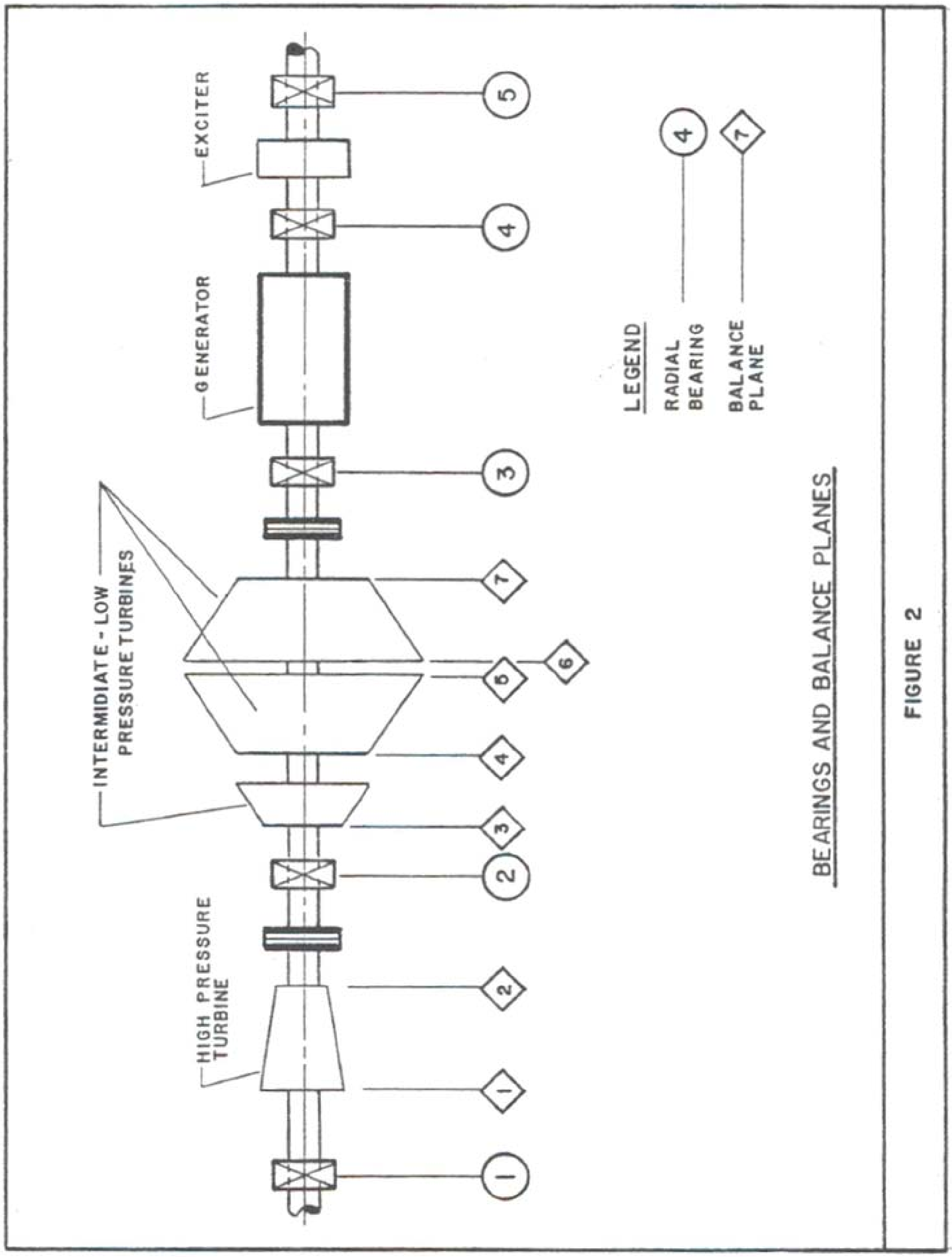
In this job, a weight was not put in the right hole and one weight fell out during a trial run. These events affected the balancing operations.

Our recommendation based on this experience is that very close supervision is necessary up to the point that the working personnel resent the presence of the supervisory staff. When many overtime hours are involved in a job good relations with all personnel is essential to get the job done in a timely manner. Thus, over-supervision causing resentment could be counter-productive.



PROBE LAYOUT

FIGURE I



BEARINGS AND BALANCE PLANES

FIGURE 2

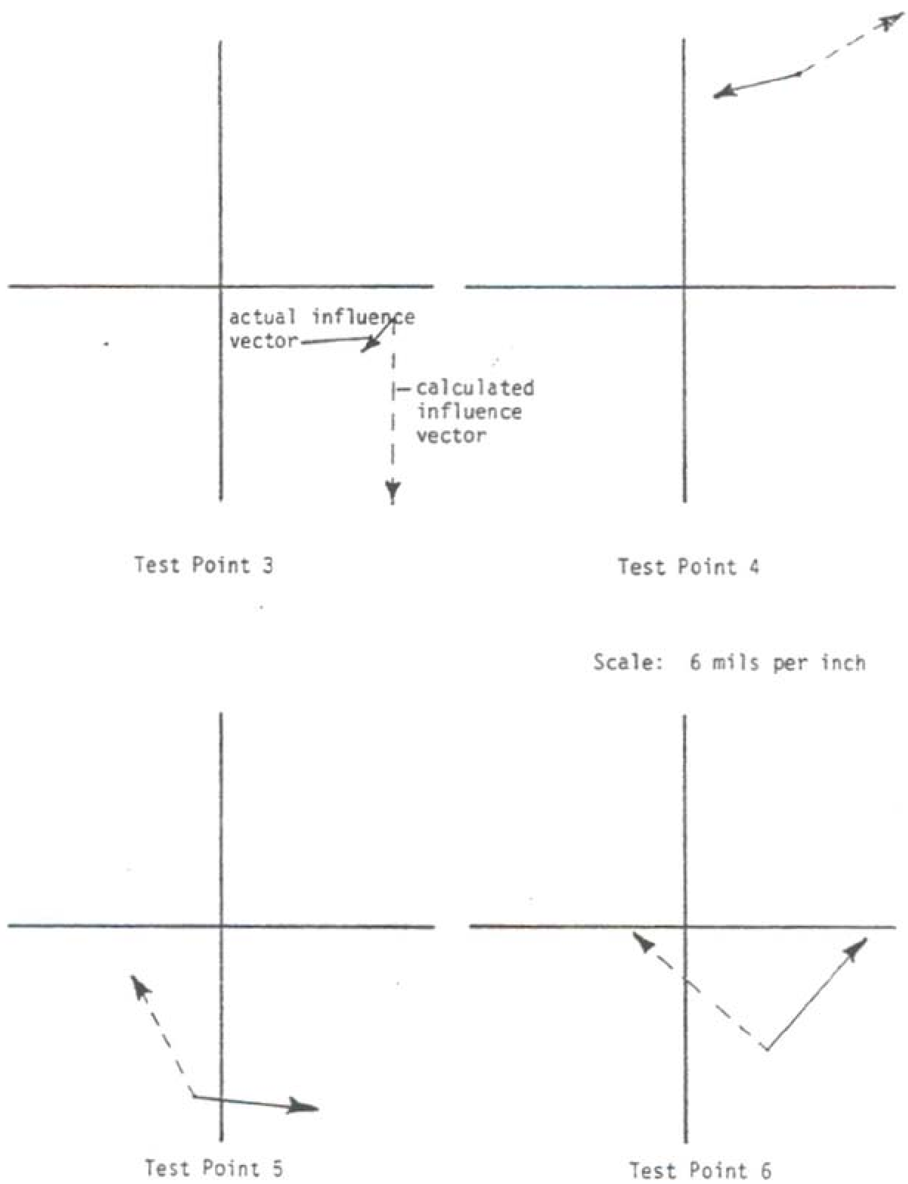


Figure 3 – Comparison of measured and calculated residual vibrations and influence vectors after first balance run.

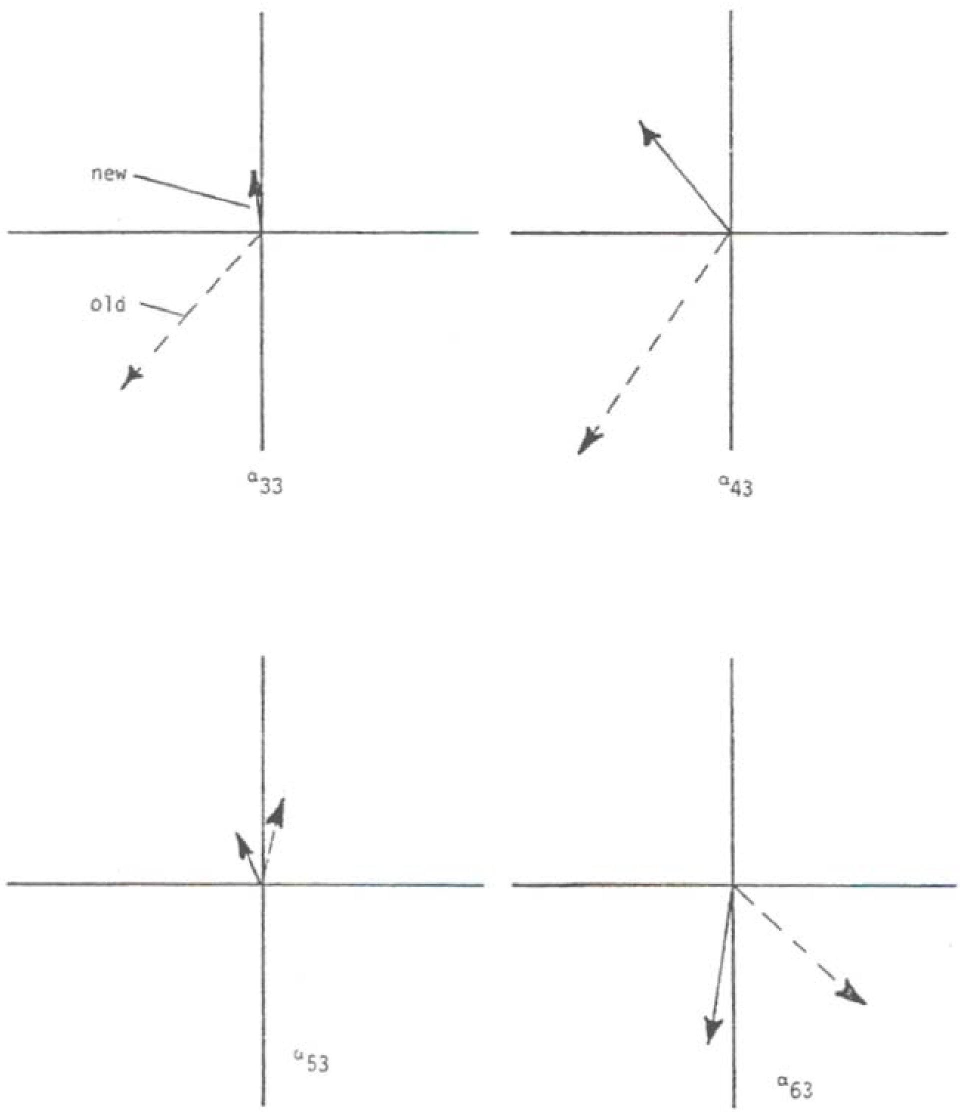


Figure 4 – Comparison of old and new influence coefficients.

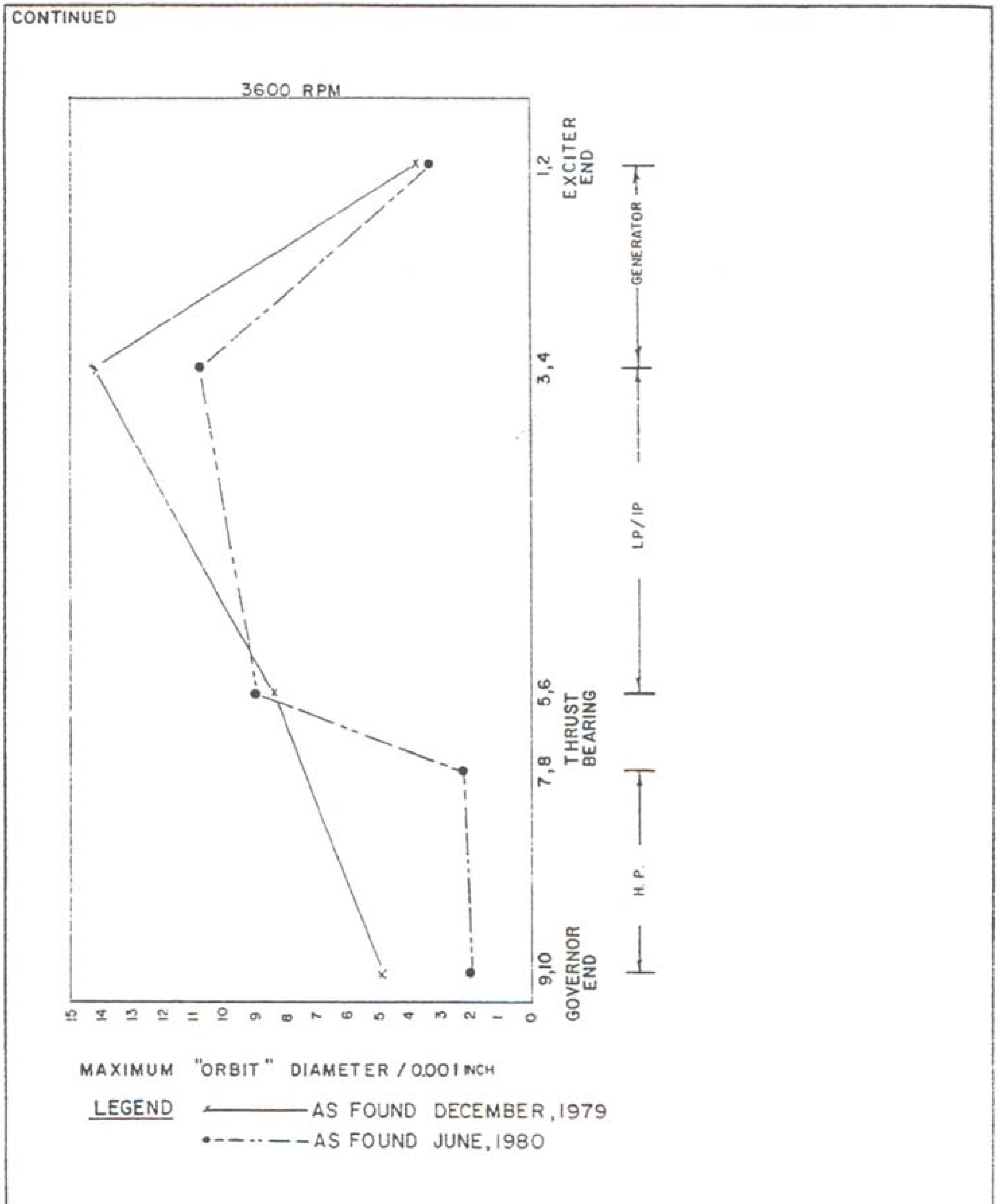


Figure 5 – Vibration Change Due to Realignment

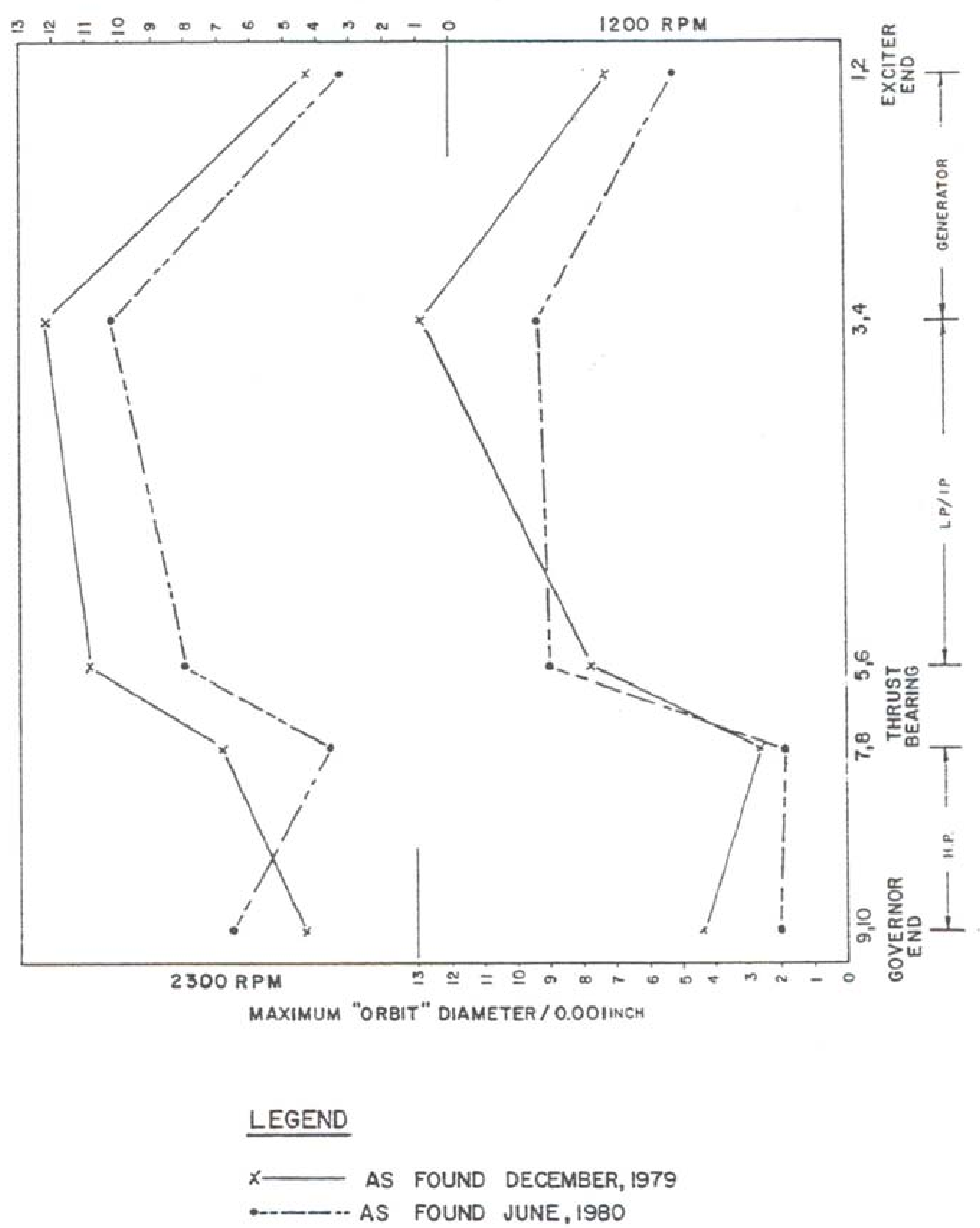


Figure 5 – Vibration Change Due to Realignment

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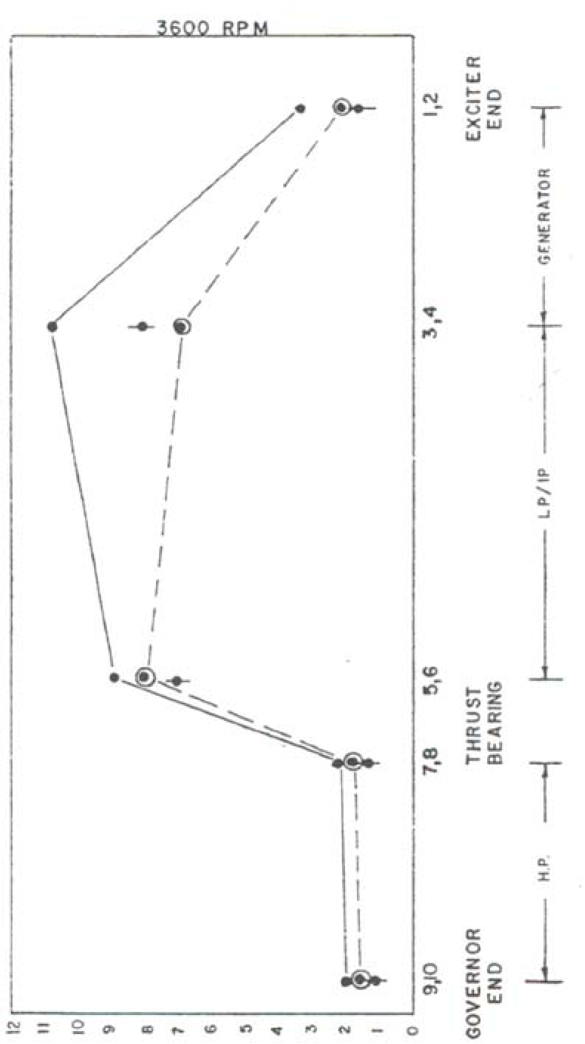

MAXIMUM "ORBIT" DIAMETER/0.001INCH

LEGEND
- AS FOUND JUNE,1980
- BALANCE RUN # 2 RESULTS
- BALANCE RUN # 3 RESULTS (FINAL)

Figure 6 – Vibration Change Due to Balancing

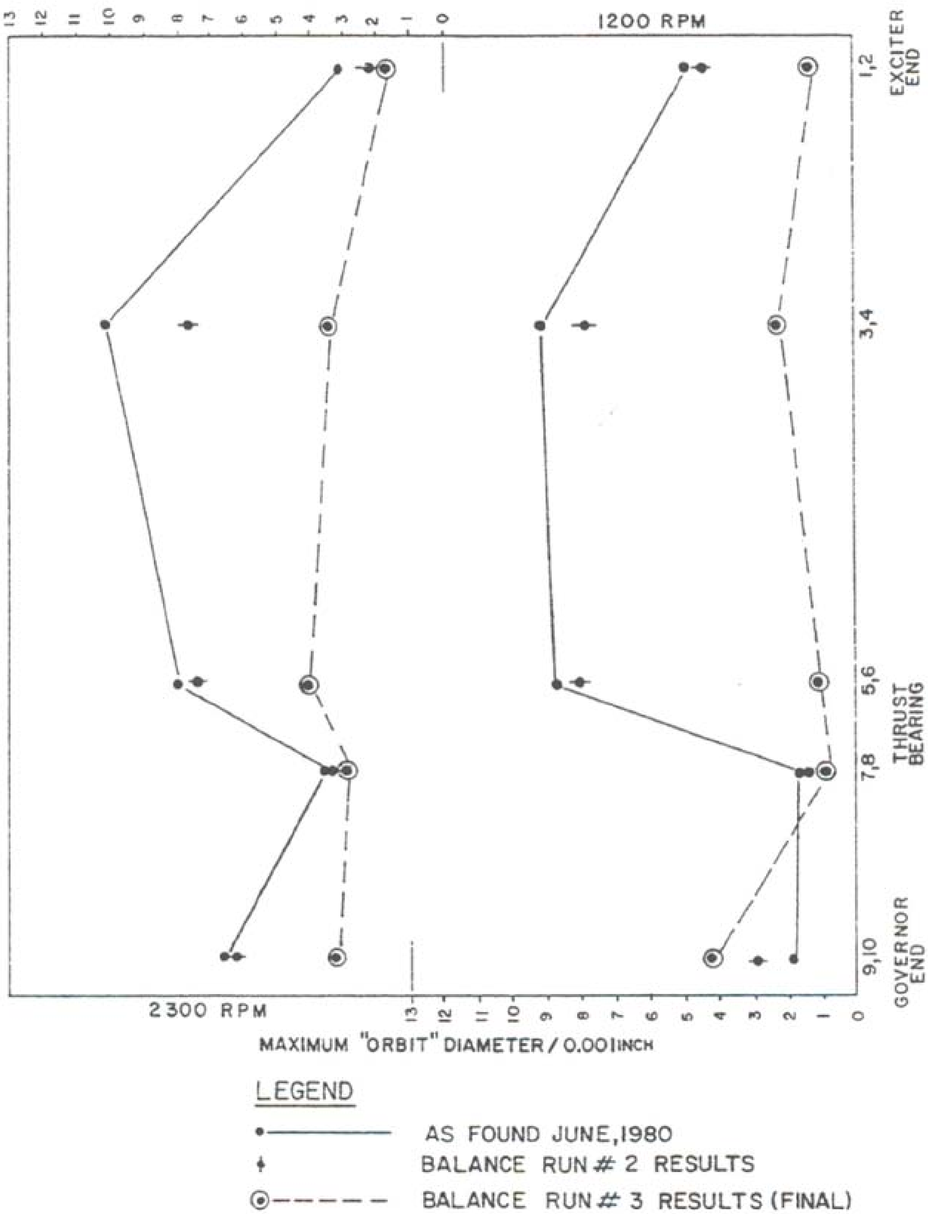

Figure 6 – Vibration Change Due to Balancing

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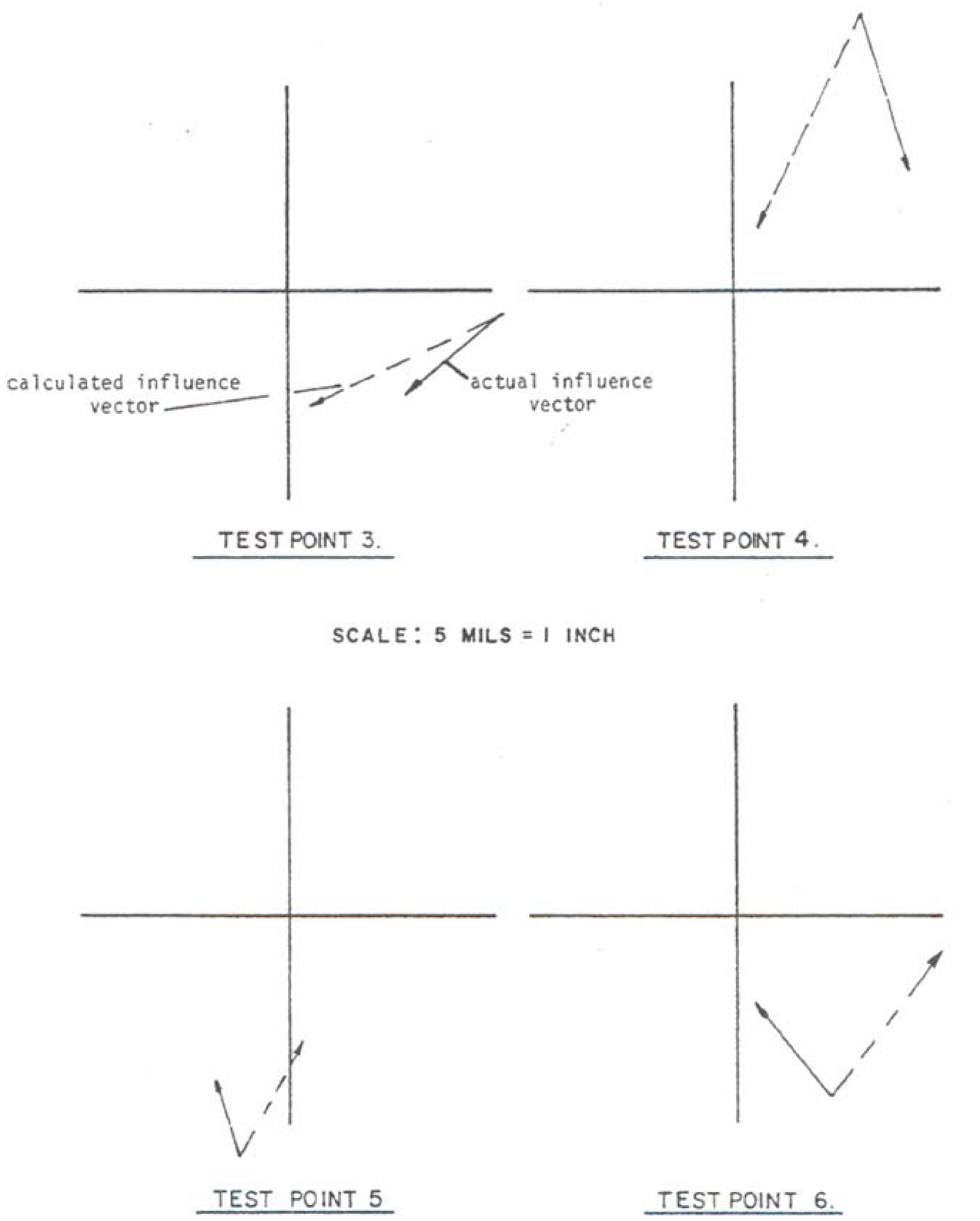

Figure 7 – Comparison of measured and calculated residual vibrations and influence vectors at 3600 RPM after second balance run.

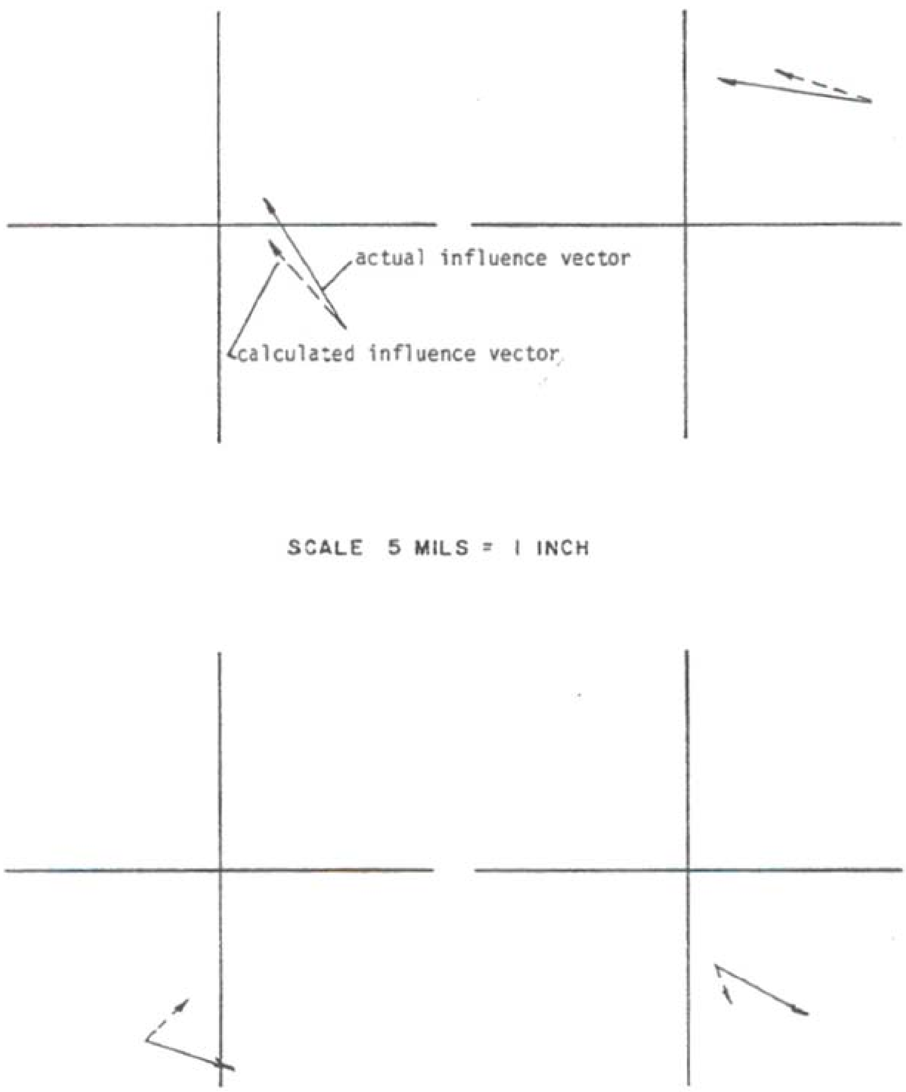

Figure 8 – Comparison of measured and calculated residual vibrations and influence vectors at 3600 RPM after third balance run.

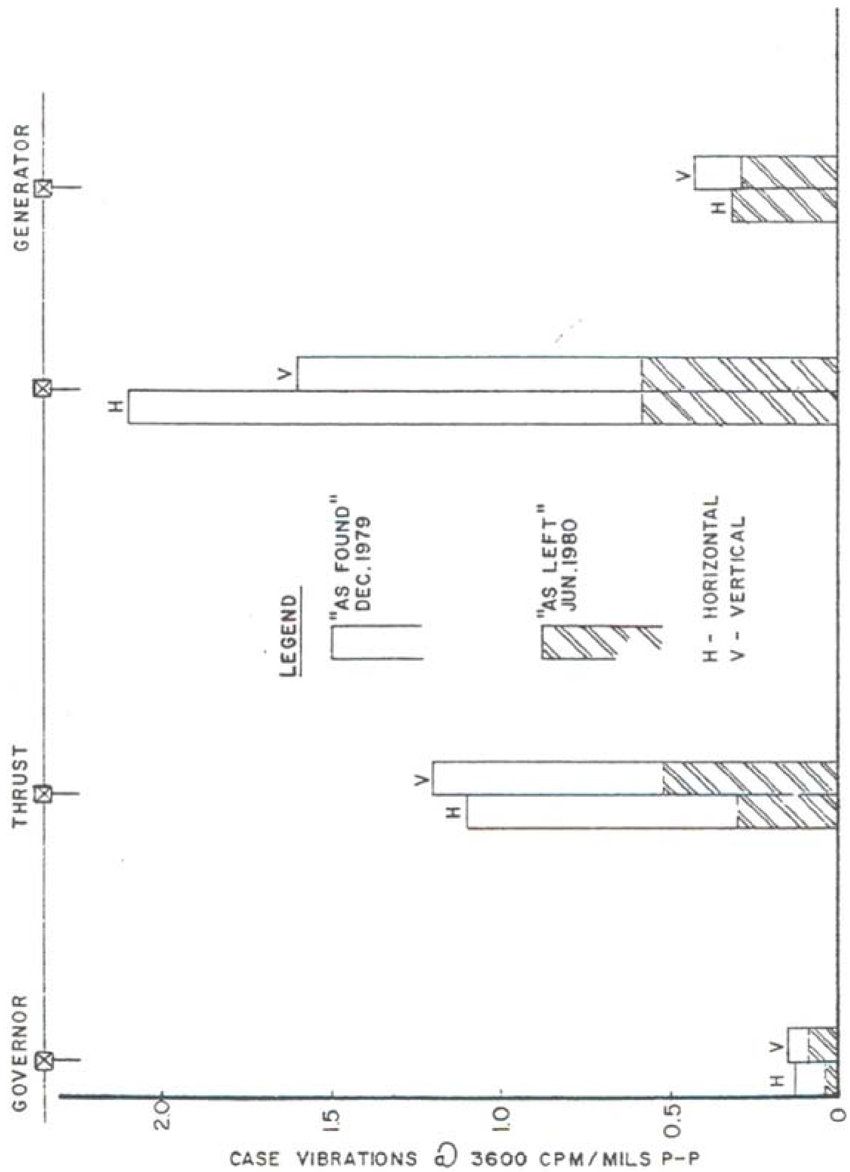

Figure 9 – Case Vibrations

DISCUSSION

"FIELD BALANCING EXPERIENCES"

B. Howes and B. Long

Prepared by G.J. Brown

The first commentator suggested that in the situation where there is little prior knowledge about a machine, then a computer based balancing system would be of great assistance. The use of this system would enable a complete set of polar and rotor plots to be obtained during the run up or run down of the machine. This would alleviate the problems of making accurate measurements close to the critical speed while trying to hold the speed manually, especially difficult for phase measurements. This more complete data would have made the choice of where to make the balance corrections much easier. The author concurred with this view, pointing out that equipment limitations in the field situation had to date made this approach difficult.

A second commentator emphasized the importance of obtaining the machine response in the form of a vibration amplitude versus machine speed plot, so that the critical speed can be accurately pin-pointed. This data can then be used in an incremental balance procedure to ensure that correct balance is obtained at the critical speed. In this method the machine is balanced using influence coefficients calculated at the closest speed to the critical speed that can be safely maintained. After balance the machine is run closer to the critical speed, and new influence coefficients are calculated and balance corrections made. This approach will allow balance at critical speed ensuring the lowest vibration amplitude in the machine.

A question regarding the inclusion of the generator in the balancing exercise was raised, it was pointed out that if the culprit out of balance was actually in the generator and an attempt was made to correct this by balancing the turbine, then the bearing influence coefficients could be significantly in error. In the study reported, the generator had not been included. It was hydrogen filled and had not been depressurized, even though there was some chance it had been previously damaged, and may not be in a good state of balance.

In response to a second question, it was noted that the highest vibrations measured were on the low pressure rotor and these had not been significantly affected by a change in vacuum. In the first machine balance, weights had been added to all available planes in the machine, in the second balance the major weight changes had been in the low pressure rotor. It was also noted that in spite of the high vibration level prior to balance, and the corkscrew pattern in the polar plots at running speed, there was no reported evidence of bearing contact on subsequent strip down of the machine. The sensor recording these high vibrations was somewhat displaced from the bearing itself, and the structure was a very rigid one. The maximum allowable shaft amplitude to avoid contact should be around 25% of the bearing clearance.

The importance of mechanical or electrical runout in the vibration measurements was questioned. During this study these had not been significant, zeroing of the data had kept runout below one mil.

The last question asked was whether on subsequent machine strip down, there had been physical evidence of damage, large enough to cause the high vibration levels seen before balancing of the machine. In fact, it appears that this machine had a history of being very rough from the day it was installed, and the conclusion drawn was that balancing technology in 1968 was just not good enough to provide adequate balance in the machine.