

VIBRATION LIMITING OF ROTORS BY FEEDBACK CONTROL

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SUMMARY

This work reports some experimental findings of a three-mass rotor with four channels of feedback control. The channels are independently controllable with force being proportional to the velocity and/or instantaneous displacement from equilibrium of the shaft at the non-contacting probe locations (arranged in the vertical and horizontal attitudes near the support bearings).

The findings suggest that automatic feedback control of rotors is feasible for limiting certain vibration levels. Control of one end of a rotor does afford some predictable vibration limiting of the rotor at the other end. The next step in the work is to move from acoustic drivers to electro-hydraulics so that larger force levels may be experimented with.

INTRODUCTION

High speed turbomachines are subject to lateral vibrations, which may limit their performance level [1,2]. Machines are often required to operate at least 20% away from a rotor critical speed. They must also be designed so that, for machines operating above one critical speed, all clearances in the machine are sufficient to accommodate high amplitudes of vibrations near a critical speed. Lastly, machines are sometimes subject to subsynchronous vibrations, which can lead to machine failures [1]. One type of means for reducing these vibrations is that of active vibration control.

A number of analytical studies of active feedback controls for rotating machinery have been presented in the literature [3-6]. These have been primarily developed by researchers in the automatic controls area and the results are normally presented in terms associated with automatic control theory. A very limited number of experimental works have reported the use of active control for laboratory test rigs [7-9]. With the possible exception of the electromagnetic bearing, there has not been an application of active feedback control to an industrial rotor at this time.

This work considers unbalances in rotating machinery which might be compensated by use of active feedback control circuits. With proper algorithms, such closed loop automatic control devices could permit machinery to be run longer even though the unbalanced forces grew to significant values. Such an approach would have little impact for small and noncritical rotors. However, for very important systems for which an unscheduled shut-down would be costly, an analog or digital closed-loop feedback controlled force actuator(s) could be justified.

Also, for some very expensive turbines, pumps, and compressors, active feedback controlled force actuators would allow machinery to pass through the criticals with smaller clearances (because of controlled vibration levels) resulting in more efficient running. With such an application, the proposed devices would be functioning only during runup or rundown of the machinery permitting certain advantages for the designers of the machinery.

Values in the text are given in both SI and U.S. Customary Units. The measurements and calculations were made in U.S. Customary Units.

EXPERIMENTAL UNIT

A small test rotor has been constructed and is generally described by the schematic drawing of Fig. 1. It consists of a 12.7 mm (0.5 inch) diameter shaft supported on ball bearings which are 406 mm (16 inches) apart. For these tests, three masses, uniformly distributed between the bearings were attached to the shaft by simple set-screws. These disks weighed 8.9 N (2.0 lbs), 13.3 N (3.0 lbs), and 8.9 N (2.0 lbs) with the larger one at the center. The bearing carriers with the associated hardware each weighed 4.4 N (1.0 pound). Shaft weight is 4.0 N (0.90 pound).

Each bearing has mechanical springs at 11.7 N/mm (67 pounds per inch) in the vertical and horizontal directions plus loud speaker motors (acoustic drivers) attached to the bearings directed in the vertical and horizontal attitudes. This test rotor is unique in that forces may be applied to the bearings, through acoustic drivers that are independent from one another in the vertical and horizontal directions. These four acoustic drivers provide four independent routes for applying forces to the bearings.

The vertical and horizontal displacements near the bearings were measured relative to the static equilibrium position. A force was generated through an analog feedback loop in the same attitude but of the direction opposing the motion. At the same time, an additional force was generated that was proportional to the instantaneous velocity (and also oppositely directed). These forces were then applied to the shaft, through the ball bearings, by means of the acoustic drivers. These four acoustic drivers are independently operable. The controls for setting the constants for generating the forces, as a function of displacement or velocity or some combination of displacement and velocity, are also independent one from another.

EXPERIMENTAL RESULTS

A number of experimental runs were made. The ones reported here are for runs with the rotor fairly well balanced and with no control, a low level of control, and a high level of control. Control was limited in these cases to the velocity feedback loops so that, effectively, only damping was being controlled. In addition, the system was modelled and theoretical response was obtained on a digital computer.

The results of the first run are shown in Fig. 2. The lines with data points shown represents the experimental results and the continuous line is for theoretical results. There is an excellent match of critical speeds at 700 and 1200 rpm and a reasonable agreement of results for a critical near 6000 rpm. In the midrange ampli-

tudes are not in close agreement. It is noted that the displacement monitoring system had a limit at about 0.56 mm (22 mils).

For the second run the control loop gains were set to give an effective damping constant at each bearing of 0.047 N sec/mm (0.27 lb sec/in). This was for all four control loops. However, just as the run was about to start it was found that the current amplifier in the free-end vertical control loop was not functioning. The run was continued with data being taken at the motor end. The results here were very similar to those of the uncontrolled system except that the critical at 700 rpm is eliminated. The theoretical response did not agree well except for the location of the criticals.

A large damping factor was applied in the last experimental and theoretical runs. The results are shown in Fig. 4. The damping constant was estimated to be about 0.37 N sec/mm (2.13 lb sec/in), about eight times greater than the previous run. Results are dramatic over the range of all three criticals.

DISCUSSION AND CONCLUSIONS

This limited set of experiments has shown some of the effects that may be obtained with the use of feedback control. Here, the analog control signals were applied in both directions at the motor end and only to the vertical direction at the other end. For Fig. 4, the motor end resulted in a small circular orbit and simultaneously a very flat ellipse at the other end. The effects obtained here are through the use of velocity feedback only with the velocity gain setting at about one fourth of its full scale values.

These runs were made using analog control only. The experimental rig has a complete digital control system, also. The full potential for the analog system has yet to be explored. However, it is known that it would use some adaptive ability, either speed sensitive or logical, in order to fully develop its potential.

The digital control system is potentially far more flexible. In this experimental system it can be brought into play by simply turning it on and flipping a switch. The digital unit in the control loop is an Intel 8086 single board computer. This offers a sixteen bit manipulator. We can anticipate some very interesting control algorithms with this system which at the present time has been used for a simple look-up algorithm only. There will be a possibility of even employing an algorithm that "learns" what combination of displacement and velocity (when multiplied by appropriate constants) will produce the minimum response for the system at each speed for that particular rotor condition. Then as the rotor condition changes (through erosion, accumulation of dirt onto the disks, etc.) this "learning" algorithm could accommodate the changing disturbance.

The acoustic drivers are small, yielding forces that are small. Typically, the output for the drivers is approximately 15 watts maximum but with the proper phasing does produce significant results. It is felt that electro-hydraulics will be the preferred means of applying forces in an industrial atmosphere. Preliminary calculations indicate that electro-hydraulics will be satisfactory directly for speeds up through 10,000 rpm. For speeds above that, it will be necessary to employ one or more revolutions of the shaft to make the necessary calculations as part of the control algorithm. However, this time delay (or more specifically the rotation delay)

will not likely be significant as rotors must go through many cycles before amplitudes change an appreciable level.

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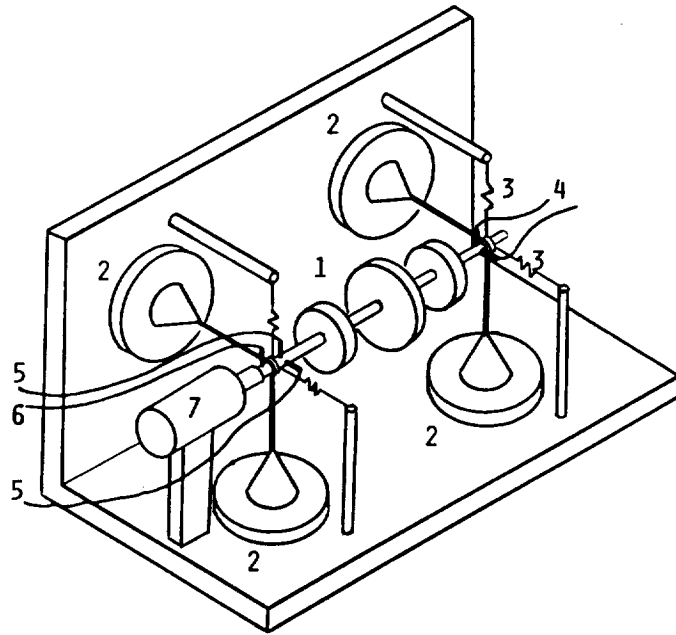


Figure 1. - Schematic diagram showing three-mass rotor 1, loud speaker motors 2, mechanical support springs 3, non-contacting probe signals for measuring the position of the shaft at the outboard bearing 4, similar signals for the motor-end bearing 5, key phasor signal for speed measurement 6, and a variable speed motor 7.

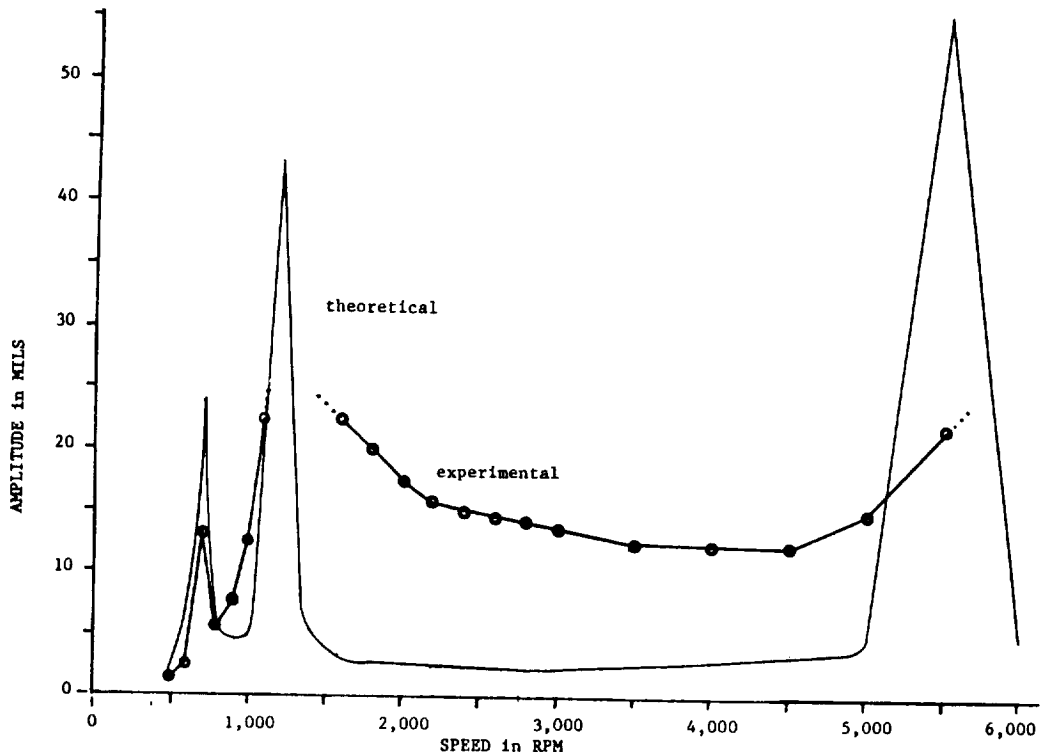


Figure 2. - No feedback control.

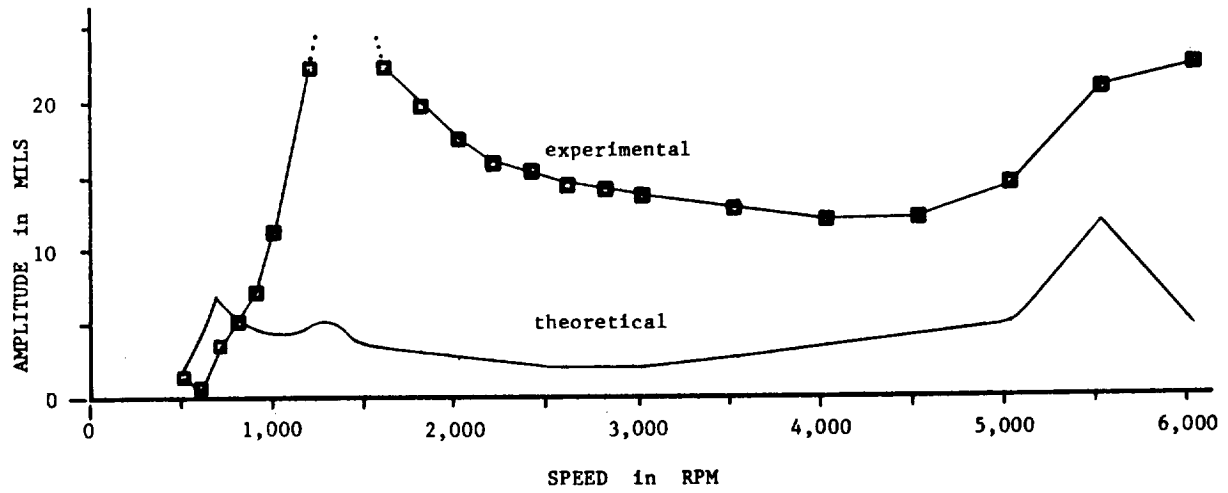


Figure 3. - Small velocity feedback control.

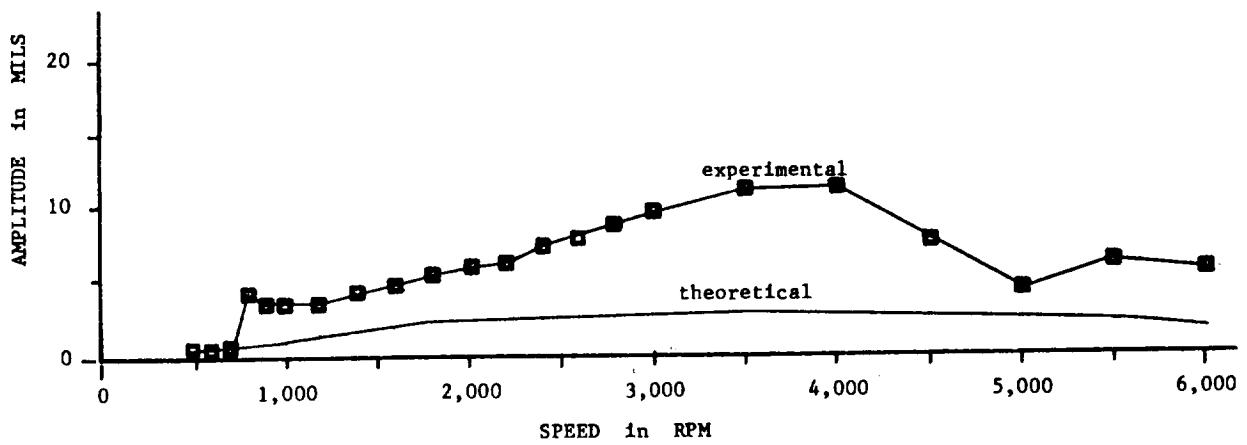


Figure 4. - Large velocity feedback control.