ACTIVE FEEDBACK CONTROL FOR STABILIZATION OF VIBRATION
IN ROTATING MACHINERY (24.034)

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It is a pleasure to write this quarterly report because of the recent progress. We demonstrated the special rotor having four closed loop servo-systems at the 1980 Symposium on Instrumentation and Control for Fossil Energy Processes. This rotor employs acoustic speaker coils to support and move the bearings at each end in the vertical and horizontal directions. The electronics for this system is through analog circuitry and is the means for controlling independently, in the four channels, spring stiffness and damping.

This rotor model demonstrates the feasibility of using active feedback control for minimizing vibrations. By controlling the spring stiffness in either the horizontal or vertical direction of either bearing one can demonstrate the change in orbit response of the system and in an extreme case can cause one end of the rotor to vibrate in a vertical plane only or if so desired can cause it to vibrate in the horizontal plane only. The fact that the individual support stiffness and damping, either vertical or horizontal at either end of the rotor, can be controlled individually promises much future experimental work for this rotor unit. For example, one can change the damping characteristics at the end of the rotor and through the instrumentation on this unit, which consists of six non-contacting sensing probes and proximitors, show the response of the center of the rotor.

The unit is presently supported by soft springs in both the horizontal and vertical directions at both ends of the shaft, in

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addition to the acoustic speaker coils. With this configuration the first rigid body mode of vibration is at approximately 600 rpm. At this running frequency the rotor motion becomes very large and would be unacceptable if it were a production machine. However, by changing the damping and/or the spring stiffness support through the four closed loop servo-systems, this unit can be run through the first mode and up to a speed exceeding 6000 rpm.

This rotor system has not been in use long enough for us to extract the essential data from it. However we have pursued several other facets associated with this feasibility study. Some of the theoretical work is included in the paper entitled "Active Feedback Control of Rotor Unbalance or Rotor Dynamic Instability" which was included in the 1980 Symposium previously cited and is included herein.

The usefulness of this feedback control loop system concept will rest on having an electro-hydraulic actuator system that can respond quickly enough to the desired motion of the machine. On the basis of a machine running at 10,000 rpm and assuming a required maximum force of 5000 pounds and a maximum displacement of the bearing of 0.005 inches there appear to be some commercial elements that can comprise part of a complete electro-hydraulic system. The physical hydraulic actuators that are commercially available will not meet the demands of the system. From a design point of view, we would consider using a steel diaphragm as the actuating mechanism being some 6 inches in diameter and with a thickness of 0.50 inches. On the basis of this diaphragm dimension and the design criteria of speed and force

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and motion one can calculate that a hydraulic power requirement of 4 horsepower and a flow rate of approximately 6 gallons per minute will satisfy the needs. Several MOOG servo valves for an example are available that can handle this demand. One such unit is the 33-180 Hybrid MOOG Servo Valve which would have a phase lag of some 22 degrees at the 167 Hertz frequency (10,000 rpm). Of course at lower speeds the phase lag would be reduced and therefore the control would be much better. The system likely proposed would probably include a two stage servo-valve so that the diaphragm actuator would be operated essentially under pressure in one direction only. Further theoretical work needs to be done to see whether or not one can control, to the necessary degree, the motion of the center of a rotor through actuation or motion of one end of the rotor only. It is clear that motion in both the vertical and horizontal direction at one end significantly reduce the amplitude of motion of vibration at the center of the shaft. Certain classes of machines however have enough cross coupling that it seems possible to control in say the horizontal direction only and attain the necessary reduction of vibration in the center of the shaft. This theoretical work will be part of the continuation of this research project and will require some rather sophisticated computer work. This means that at this stage the safest possible design approach will be to design with actuators in both the vertical and horizontal directions at the bearings using the electro-hydraulic system. Of course, it is anticipated that the necessary calculations would be made for a particular class of machine to determine

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whether or not there is sufficient cross-coupling in the response of the rotor so as to require only an actuator in a single direction.

It is felt that the work to date has been extremely positive. A digital control system has been built and at present the software is being implemented and checked out. We feel that the project is very much on schedule and the results of the testing to date have been much more positive than any of us anticipated.

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ACTIVE FEEDBACK CONTROL
OF ROTOR UNBALANCE OR ROTORDYNAMIC INSTABILITY

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SUMMARY

This paper discusses some of the considerations involved in the use of feedback control as a means of reducing the unbalanced response of a rotor or of eliminating or alleviating rotor dynamic instability. A simple model of a mass on a flexible shaft is used to illustrate the application of feedback control concepts. A description is given of a system assembled at the University of Virginia which uses feedback control individually adjustable in the vertical and horizontal directions to support the shaft bearings.

INTRODUCTION

The feasibility of using active feedback control of rotor dynamics by the active control of some or all of the forces in the bearing support system is the subject of research just started at the University of Virginia. If such an approach is feasible, or even partially so, many benefits could accrue to those now having problems of rotor dynamic instability.

The potential value of this work, if successful, will be measured by some very large annual savings. With this approach, large turbines, pumps, compressors, motors, and alternators can be designed with much smaller clearances and hence, higher efficiencies. Presently, clearances must be such as to accommodate the larger motion excursions that occur as the rotating machine passes through certain "critical speeds." This active feedback control approach can move these criticals in such a way as to avoid running through them.

With certain types of compressors, pumps, and turbines, the unbalance of the rotating element changes as fouling occurs. This active feedback control approach coupled with adaptive elements will, perhaps, permit running a machine longer between necessary shutdowns for overhaul and cleaning.

The technology associated with this active feedback control approach may be considered as a type of insurance. It is easy to imagine a designer adding another stage to a turbine and additional length to an alternator and producing a unit that cannot be brought up to the operating speed because of a large vibration amplitude at a critical speed. A "fix" employing passive squeeze film dampers or bearing modifications may not solve the problem. If this new

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application of feedback control has been developed sufficiently, this technology may be the most (or only) expeditious solution means. It is certainly too early to attach any numbers to this technology, but saving even one billion dollar electric power generation plant may be cost effective. The singular use as a back-up technology to the centrifuge uranium enrichment program may also justify the development of this technology.

To study this approach a rotor system using control concepts is now being assembled. It consists of a flexible shaft mounted horizontally in ball bearings at the ends and having provisions for mounting one to three masses. The bearings are each supported by two high fidelity speaker motors which have a linear range in excess of any expected transients. These motors are mounted at ninety degrees to each other. Induction type proximity sensors are used to sense shaft position at the bearings and at the central mass. The position signals are then input to the control system which supplies gain and compensation for the servo amplifiers driving the motors.

The system is designed to use either analog control or computer control of the individual control loops. Initial efforts will use relatively simple control algorithms but ultimately it is expected that the system will have aspects of pattern recognition and will be adaptive to running speed and to other system parameters.

To illustrate some of the potential benefits of using feedback control a simple model is analysed here from the control standpoint. Parameters for the model are selected to be close to those for a system which was the subject of a doctoral dissertation by Marvin Taylor at the University of Virginia in 1979. This work reported on the active feedback control of a rotating cup containing a steel ball which was free to rotate around the inner periphery of the cup. The cup was mounted on a slender shaft, positioned vertically and supported at the upper end only. This study was limited in that motion was both sensed and controlled in the plane of unbalance. However, Taylor was able to control not only the vibration amplitude at the critical speed, but also the whirl of the ball in the cup.

NOMENCLATURE

a_i	polynomial coefficients	k	shaft stiffness, N/cm (lb/in)
$A(s)$	loop gain	k_1	support stiffness, "
c	shaft damping, N s/cm (lb sec/in)	K	control gain
c_i	support damping, "	L	load disturbance
$G(s)$	control transfer function	m	rotor mass, Kg (lb sec ² /in)
$G_c(s)$	overall feedback transfer function	p_i	control and system poles
j	$(-1)^{\frac{1}{2}}$		

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r	unbalance position, cm (in)	α	ratio of support stiffness to shaft stiffness
s	Laplace variable	γ	damping ratio
u	control signal	ϕ_i	phase angles
z_i	control and system zeros	ω	frequency, rad/sec
Z	displacement of mass center, cm (in)	ω_n	natural frequency
Z_1	displacement of bearing center, cm (in)		

THE ILLUSTRATIVE SYSTEM

The system to be modeled and controlled is shown in Fig. 1. For this study complete symmetry is assumed and the bearings are considered as massless. The springs provide the basic support and the control force is additive at the same point. This allows the system to be considered as being of third order. The equations of motion are:

$$c_1 \dot{Z}_1 + k_1 Z_1 + k(Z_1 - Z) = u \quad 1.$$

$$m\ddot{Z} + c\dot{Z} + k(Z - Z_1) = L \quad 2.$$

where L represents the loading at the mass due to unbalance and u is the control force to be applied. We take the Laplace transform and solve for transfer functions of Z and Z_1 with input u. The unbalance force we will consider as being a load disturbance. In addition, the shaft damping is considered as negligible. The transfer functions are:

$$Z = \frac{(c_1 s + k + k_1) L + ku}{mc_1 \left[s^3 + \frac{k + k_1}{mc_1} s^2 + \frac{k}{m} s + \frac{kk_1}{mc_1} \right]} \quad 3.$$

$$Z_1 = \frac{(ms^2 + k) u + kL}{mc_1 \left[s^3 + \frac{k + k_1}{c_1} s^2 + \frac{k}{m} s + \frac{kk_1}{mc_1} \right]} \quad 4.$$

The denominator of both equations will have one real root and a pair of complex roots. Using this fact and noting that we need to separate the load and control effects, we can write:

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$$Z = \frac{(s + z_1)}{m(s + p_1)(s^2 + a_1 s + a_0)} \left[L + \frac{ku}{c_1(s + z_1)} \right] \quad 5.$$

In this form a block diagram can be drawn. This is shown in Fig. 2 where control and sensor blocks have also been added. In this instance the sensor is assumed to give position and velocity information.

An actuator, the control motor, is assumed to have the transfer function

$$\frac{p_2}{s + p_2} \quad 6.$$

and the controller is assumed to have the form of a gain and a lead-lag function. Combining these factors the overall control transfer function is:

$$G_c(s) = \frac{-K(s + z_2)(s + z_3)}{(s + p_3)(s + p_1)} \quad 7.$$

Combining this with the system transfer function gives a loop gain $A(s)$ which can be used to plot a root locus which, in turn, can be used to select the gain and set the control poles and zeros. This function is:

$$A(s) = \frac{K_1(s + z_2)(s + z_3)}{(s + p_1)(s + p_2)(s + p_3)(s^2 + a_1 s + a_0)} \quad 8.$$

As mentioned earlier, the parameters for this system were chosen to be approximately equivalent to Taylor's system. This had $k = 20.8$ lb/in and $\omega_n = 85.5$ rad/sec. A number of systems were considered having second order parameters close to these. These are listed in Table 1. Here the entering point is the first column, which is the ratio of support stiffness to shaft stiffness. The second column is the support damping. For each pair of values of these parameters the real pole and zero, the complex poles, and the quadratic coefficients are listed. The parameters chosen for this study are in the row marked with an asterisk.

Root loci of this system for two different values of the control zero are given in Fig. 3. Also, taking advantage of the vertical symmetry of root locus plots, the lower half plane shows a locus of the system complex poles for the spring rate ratio equals one and for the damping factor being variable. The point on the locus for c equals .5 is marked with an x, the standard open loop pole symbol. Its complex conjugate is used in the upper half plane as the starting point for the root loci.

Considering these loci, the solid line is the complex locus with the system motor pole at -200 and a zero at -1. The control compensator pole is

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at -400 and its zero is at -100.

The dashed line represents the same system except the compensator zero is moved to -40. The effect is to move the complex locus farther to the left of the imaginary axis. In addition, a new complex locus appears near the real axis.

A root locus is a plot of all possible roots of the closed loop system with K as the variable parameter. Thus the effect of closing the control loop is very apparent in this case. The principle effect is that the original system poles are eliminated and a new set of poles are established. These poles can be located at any particular points on the branches of the loci by picking a particular value of the control gain K . The poles on the upper complex branch are the ones of particular interest since they will tend to dominate the dynamics of the system. They could be selected for maximum damping or to move them far away from the original system natural frequencies. They represent a quadratic pair in the closed loop system. Considering them only, their damping factor is represented by the $\arccos \gamma$, measured from the negative real axis. In Fig. 3 the maximum γ lines are shown for both upper branches and the minimum γ line is shown for the lower complex branch.

Four values of gain were chosen for calculation of the closed loop frequency responses of the system. The resulting closed loop complex poles are represented by the numbered squares on the upper branches. The results are shown in Fig. 4. Curve 1 is for poles located at point one on the solid locus. The resonant peak is about $1.6 \times$ approximately 200 rad/sec. Curve 2 is for the dashed locus with the complex poles having about the same imaginary value as the first case and with the control zero at -40. Curve 3 is from the same locus with the control gain lowered to place the upper complex poles at near maximum γ . For curve 4 we return to the solid locus and raise the gain to place the complex poles at near the point where the locus crosses the imaginary axis, the point of control loop instability. The response is unbounded for practical purposes at about 270 rad/sec.

THE EXPERIMENTAL MODEL

This has already been described and is very similar to the model of Fig. 1 without the symmetry limitation. Also, the control system almost completely replaces the spring-damper support system and the bearing masses become significant. The analog control system will be of the same form as in the example. All its parameters will be adjustable and there is provision for making it somewhat adaptive to rotor speed.

The computer control is effected by replacing the control block in Fig. 2 by a single block Intel 8086 microprocessor and peripherals. Input and output are through appropriate analog to digital and digital to analog converters. This microprocessor can handle 16 bit words which allows greater flexibility and precision but which operates a bit slower than some 8 bit microprocessors. However, preliminary estimates indicate that a computation cycle can be completed about every 10 degrees of rotation of a shaft turning at 6000 rpm. The initial programming of the microprocessor will be of a proportional plus rate type with adaptivity to rotor speed and potential for

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adapting to other parameters.

CONCLUSION.

The example presented illustrates some of the ability of feedback control to modify the basic characteristics of a rotor system. Potential advantages include the possibility of easily negotiating critical speed ranges; lossless electronic damping; and adaptation to changing load conditions, age, lubricant characteristics, and other factors. In addition, computer control offers the potential of parallel, or even on line, analysis of system characteristics as a means of recognizing potential trouble and of adapting the control system in an optimum manner.

Yet to be considered in detail are a myriad of the problems of real systems. These include all the present problems which are well known in rotor dynamics and adds the problems of control systems. These latter include bandwidth limitations of control components; design of control motors; power and force levels; nonlinearities, particularly signal limiting; observability of appropriate system states; selection of optimum control algorithms; computation speed; and many others. Some recommendations with respect to the feasibility of solutions for some of these problems will be made in the near future. An adequate number of problems will remain for several years of research effort.

ACKNOWLEDGEMENTS

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TABLE 1
SYSTEM CHARACTERISTICS

α	C_1	REAL	COMPLEX	QUADRATIC COEFFICIENTS			ZERO
0.5	0.1	-296	$-8 \pm 50j$	1	16	2539	-312
	1	-10.6	$-10 \pm 84j$	1	21	7095	-31
	5	-2	$-2 \pm 85j$	1	4.1	7298	-6
1	0.05	-829	$-2 \pm 60j$	1	4.4	3629	-833
	0.1	-408	$-4.4 \pm 61j$	1	8.9	3692	-417
	* 0.5	-53	$-15 \pm 74j$	1	31	5695	-84
	1	-22	$-10 \pm 82j$	1	20	6871	-42
	5	-4	$-2 \pm 85j$	1	4	7298	-6
	10	-2	$-1 \pm 86j$	1	2	7311	-4
	20	-1	$-.5 \pm 86j$	1	1	7310	-2
2	0.1	-621	$-2 \pm 70j$	1	4	4848	-625
	1	-46	$-8 \pm 80j$	1	17	6549	-63
	5	-8	$-2 \pm 85j$	1	4	7280	-12
5	0.1	-1249	$-.5 \pm 78j$	1	1	6025	-1250
	0.5	-245	$-2 \pm 78j$	1	5	6138	-250
	1	-117	$-4 \pm 80j$	1	8	6415	-125
	5	-21	$-2 \pm 85j$	1	4	7229	-25
	10	-10	$-1 \pm 85j$	1	2	7294	-12
10	0.1	-2084	$-.2 \pm 81j$	1	0.4	6503	-2084
	0.5	-414	$-1 \pm 81j$	1	2	6530	-416
	1	-205	$-2 \pm 81j$	1	3	6613	-208
	5	-38	$-2 \pm 85j$	1	4	7161	-42
	10	-19	$-1 \pm 85j$	1	2	7270	-21

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Fig. 1 Physical Model

Fig. 2 Control Model

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Fig. 3 Root Locus

Solid Line $z_2 = -100$

Dashed Line $z_2 = -40$

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Fig. 4 C. L. Frequency Response

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