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EXPERIMENTAL RESULTS OF AN ACTIVE TIP CLEARANCE CONTROL SYSTEM FOR A CENTRIFUGAL COMPRESSOR

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ABSTRACT

Increases in compressor efficiency and pressure ratio can be achieved by controlling the blade tip-to-shroud clearance. The design and test results of a fail-safe active tip clearance control system for a centrifugal compressor are presented. An electromagnetic actuator is used to pull the compressor impeller axially to reduce the tip clearance. The actuator acts against a preloaded Belleville spring. The controlled variable is an averaged, tip-to-shroud clearance which is measured with a capacitance displacement sensor and averaging electronics. A scheduled digital PI control law was used to account for the nonlinear effects of the electromagnet with electrical current and gap size. Closed-loop tip clearance control is demonstrated over the clearance range from 0.005 inches (0.127 mm) to 0.012 inches (0.305 mm). A simplified model of the system is presented along with open and closed-loop dynamic test results. Changes in the compressor performance due to tip clearance control are also presented.

NOMENCLATURE

ATCCS	Active Tip Clearance Control System
A	electromagnetic face area
B	mechanical damping, lbf/(mil/s)
CLC	Closed Loop Control
dx/dy	partial derivative of "x" with respect to "y"
E-M	Electromagnet
F	electromagnetic force, (lbf)
g	magnetic gap, (mil)
i	electric current, (amp)
K	mechanical spring constant, lbf/mil
mil	one thousandths of an inch, 0.001 inch, (0.0254 mm).
n	number of wire turns in electromagnet
OL	Open Loop
PWM	Pulse Width Modulated
ω_n	natural frequency

INTRODUCTION

As future gas turbine engines evolve, the need to increase specific fuel consumption, to extend the mean time between overhaul, and to improve the overall engine performance and operability are becoming the driving motivations for advanced designs. Tip clearance controls for the turbine and the compressor offer a solution to some of these stringent design requirements. For the compression system the advantages of tip clearance control are increased efficiency and pressure ratio, and better management of the compressor stall line leading to a reduction in the required level of stall margin. To implement clearance control the system must account for the transient behavior of the rotor blades relative to the shroud. Passive techniques can be used for slow transients caused by thermal variations but active control techniques are required to accommodate the transient behavior caused by aerodynamic loads, centrifugal unwrap, shaft whirl, machining variations and changes caused by wear. Active control can address these variations because the blade tip-to-shroud clearance is directly measured. This paper presents the design and test results of a fail-safe Active Tip Clearance Control System (ATCCS) for a centrifugal compressor using a capacitance blade tip-to-shroud clearance measurement.

In the following sections the literature on compressor tip clearance control is briefly reviewed. Then the test rig is presented along with the design of the electromagnetic actuator. Next the open-loop dynamic model of the system is presented along with the design of the digital control system. The closed-loop experimental test results are then presented along with the performance changes in the compressor due to tip clearance control and some additional observations under closed loop control. Finally, the results are summarized and recommendations for future research are stated.

LITERATURE REVIEW

Deterioration in the performance of turbine engines due to an increase in the blade to tip clearance have been reported in Mechalic and Ziemanski (1980) and Przedpelski (1982). The first reference deals with long term wear whereas the second reference deals with wear due to sand erosion. Senoo (1986), Venter (1992), and Frith (1994) have also examined the compressor performance variation with changes to the tip clearance. It was shown that active tip clearance control with an electromagnetic actuator could be accomplished in Weimer (1992). In this test, tip clearance was inferred from a position measurement made from the back of the impeller and the blade tip to shroud gap measurement was not used. Also, this design was not fail safe in the sense that failure of the actuator could lead to rubbing because the actuator pulled the impeller away from the shroud, increasing the clearance gap. The analysis of alternative fail safe configurations were presented by Jaw, Clifford, Fagan and Klusman (1994). The work presented here improves upon the design presented by Wiemer and continues the work published by Jaw, et al.

ATCCS HARDWARE DESCRIPTION

A diagram of the Active Tip Clearance Control System (ATCCS) for the 250-C30 compressor is shown in Fig. 1.

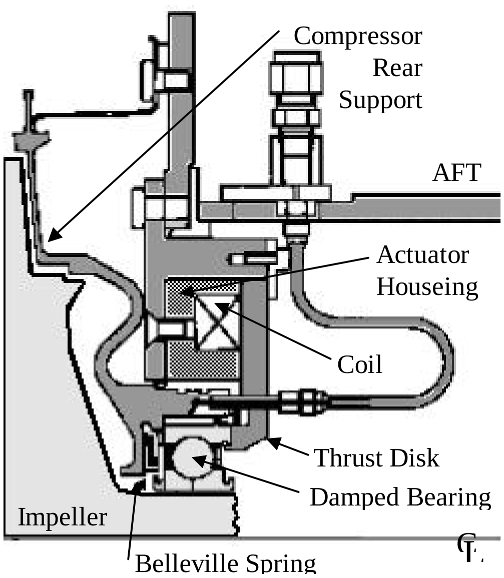

Figure 1 Centrifugal Compressor with ATCCS

In Figure 1, the aft part of the compressor is on the right and air enters the compressor from the left. The impeller is symmetric about the **c**enter**l**ine, so only the top half is shown. The stationary electromagnetic **coil** in this application pulls the **thrust disk** forward (left), loading the outer race of the **bearing** and compressing the **Belleville spring**. The **rear support** also deflects. The effective spring rate is the sum of the spring rates for the rear support and Belleville spring. The spring is the fail safe device. The spring was sized such that clearance is maintained whenever the actuator is off. The spring rate is approximately 84.8 lb./mil (14.85e03 N/mm). When the electromagnet is off or fails only the aerodynamic forces pull the impeller forward against the spring. The aerodynamic force is approximately 700 lb. (3110 N) at 30,000 rpm. This implies that the open-loop impeller will deflect 8.2 mils (0.21 mm) at 30,000 rpm. If the zero speed tip clearance is 20 mils, then this implies a blade tip clearance of 11.8 mils (0.3 mm) at 30,000 rpm. The configuration for the electromagnet, spring, and tip clearance sensors are different than those used by Wiemer (1992). A squeeze film damper was added to the thrust bearing to accommodate the radial and axial clearances necessary to allow for the rotor movement.

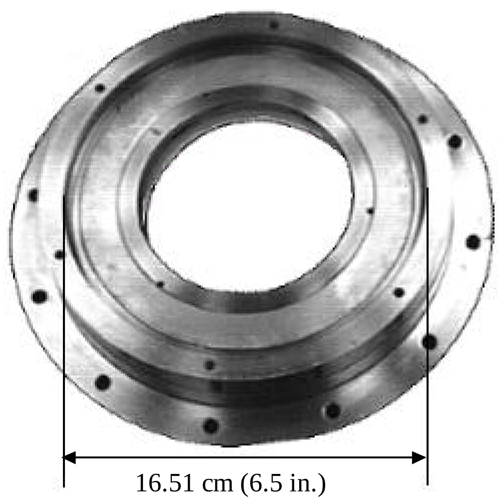

Figure 2 Electromagnetic Coil

An electromagnetic actuator was designed to fit around the rear support The actuator is a two-pole single-coil device shown in Figure 2. The characteristics of the device follow: outer diameter: 6.5 inches (16.5 cm) ; inner diameter: 3.95 inches (10 cm); width: 0.9375 inches (23.8 cm); 1300 lb. (5782 N) force at a gap of 5 mils (0.13 mm) and a current of 8.0 amps. The test coil consisted of 156 turns of 18 AWG polyimide coated magnet wire encapsulated with 500° F (260° C) epoxy. The electrical current is provided by a PWM servo amplifier (Advanced Motions Controls). The servo amplifier was used in a voltage controlled current source configuration.

The blade to tip clearance sensor consists of a capacitance displacement sensor with once-per-revolution averaging electronics. This means that the sensed tip clearance measurement is updated at a variable rate depending on the rotor speed. At 48,000 rpm the update rate was 800 Hertz and at 30,000 rpm the update was 500 Hz. This update was at a sufficiently high rate such that the sensor dynamics need not be and were not modeled.

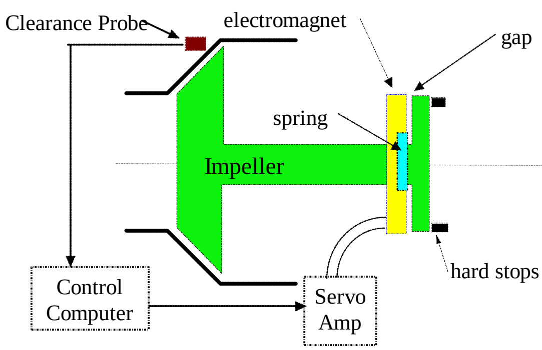

Figure 3 Control System Configuration

OPEN-LOOP DYNAMIC MODEL

The open-loop system is made up of several subsystems as shown in Figure 3 The "plant" is a second order mechanical system. The electromagnetic actuator couples the mechanical and electrical systems. The sensor is the clearance probe. The individual subsystems are examined below.

Mechanical System

The inertia of the impeller, the Belleville spring, and system damping make up a second order system. The natural frequency of the system can be estimated from $\sqrt{k/m} = 225$ Hz; however, the system was found to be grossly overdamped. Figure 4 shows the response of the system to a step change in the servoamp input voltage. The clearance changes from about 9.6 to 11.3 mils with a 90% rise time of about 50 ms. The fine oscillation are 60 Hz noise. and are not part of the system dynamic response. The oscillation with a period of about 6 Hz was prevalent in all the data. The mechanical system was

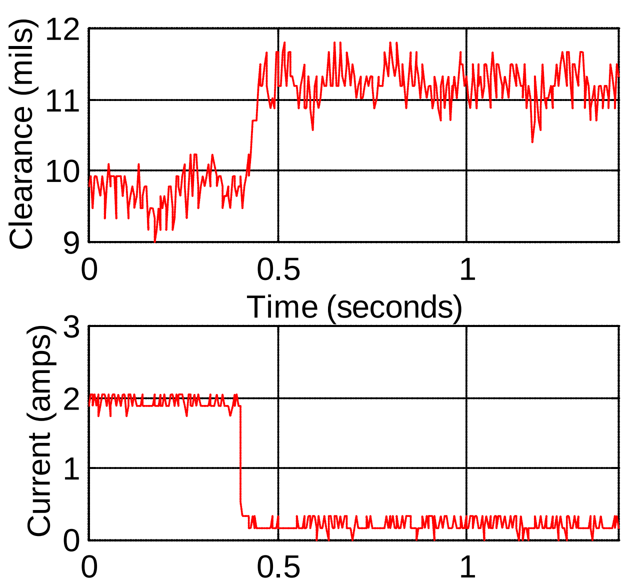

Figure 4 Mechanical System Open-Loop Step Response

modeled as a first order system, as shown in Figure 5, with just a spring rate, (84.8 lbf/mil) and an identified value for the damping, (6 lbf/(mil/s)). The nominal aerodynamic force shown in Figure 5 is used to establish equilibrium by balancing the aerodynamic, spring and electromagnet forces.

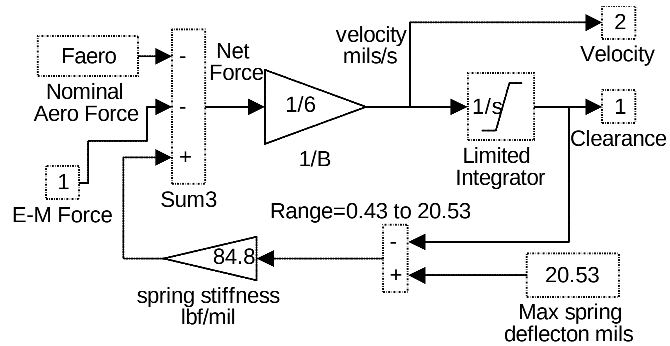

Figure 5 Mechanical Subsystem Block Diagram

Electro-mechanical System

The mechanical system is coupled to the electrical system via the electromagnet. Electromagnetic force is modeled as a two dimensional table. The output of the table is force and the inputs are the ampere-turns and the magnetic gap. The force of the electromagnet is proportional to the square of the current, i, and the face area of the electromagnet, A, and inversely proportional to the square of the gap, g, as in equation (1).

$$F \sim \frac{n^2 i^2 A}{g^2} \qquad (1)$$

The estimated force relationship is plotted in Figure 6. This data resulted from the electromagnet design and was not verified experimentally.

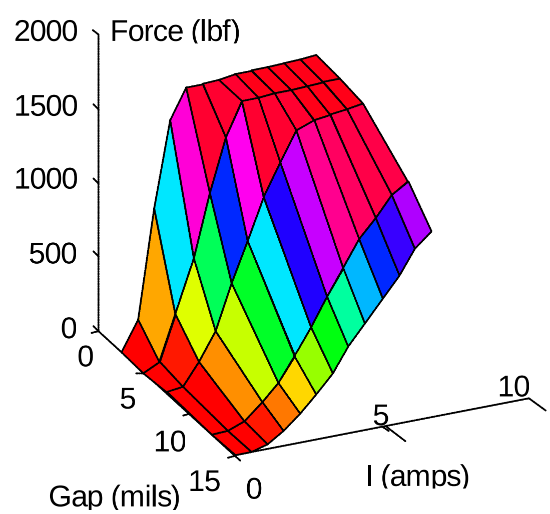

Figure 6 Electromagnetic Force vs Current and Gap

Coupling the electromagnet (E-M), to the mechanical system, (mbk2), results in the block diagram in Figure 7. Note that the path from clearance to the electromagnet is a positive feedback, which is destabilizing. Also, the feedforward path gain from the electrical current to electromagnetic force changes as a function of current and gap.

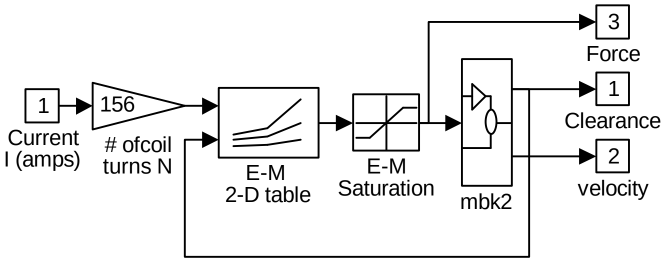

Figure 7 Electromagnetic Table and 2nd Order System

Besides the typical limits involved in a real control system, the electromagnet is the only nonlinearity in the system that has to be considered in the control design. By linearizing the two dimensional electromagnetic force table, an understanding of the local linearized dynamics can be obtained. Figure 8 shows the partial derivative of force with respect to current (dF/di) and Figure 9 shows the partial derivative of force with respect to the gap, (dF/dg). This data was obtained from the local slope of the surface plotted in Figure 6. Using these partial derivative functions to linearize the system shown in Figure 7 leads to the block diagram in Figure 10, (the "E-M Saturation" block has been removed). The gains DFDI=dF/di and DFDG=dF/dg are functions of the local operating point. The pole of the local linear first order model of the system was found to shift back and forth on the real axis and is unstable at some operating points determined by the current and clearance. Thus, it is difficult to have acceptable performance over the entire operating region with a single linear control system. A number of different linear control designs were attempted.

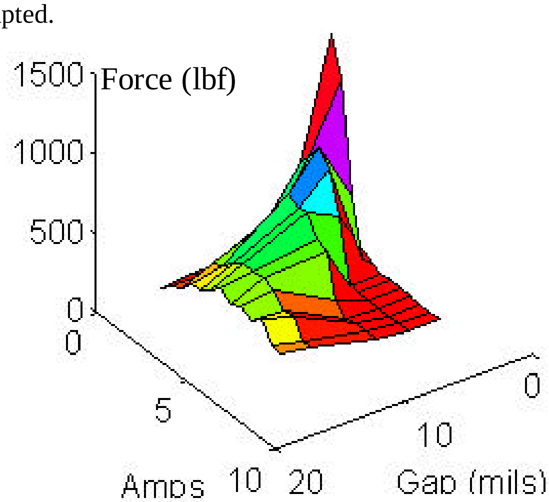

Figure 8 Partial of Force with respect to the Current, (dF/di)

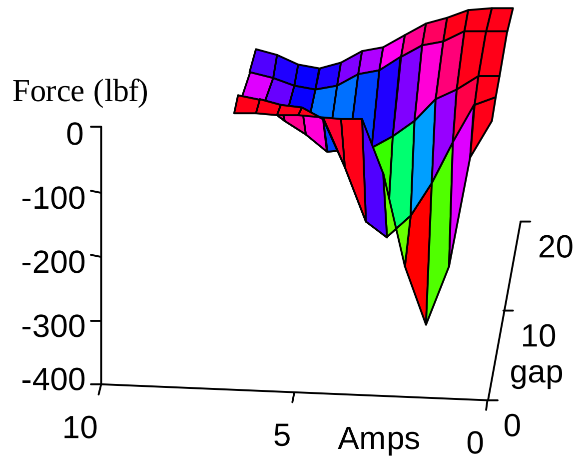

Figure 9 Partial of Force with respect to the Gap, (dF/dg)

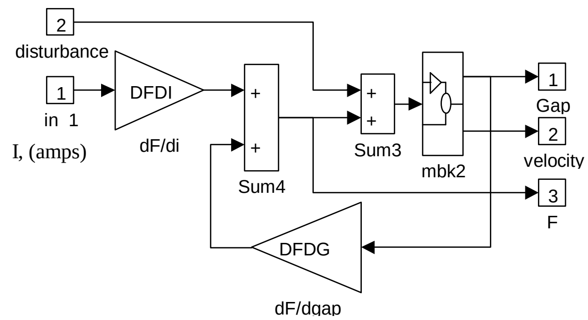

Figure 10 Linearized Version of Figure 7

The actuator dynamics, consisting of the PWM amplifier and the electric coil, are dominated by the coil inductance, (0.050 Henries) and resistance, (2.5 Ohms at room temperature). The PWM frequency is 27kHz and this frequency is attenuated by the inductance. So the servoamp was modeled as a linear voltage controlled current source. The fictitious op-amp and "idot" gains of 800 and 0.009 were modified to fit the open-loop step response test data of the electrical system. The complete linearized, open-loop model is comprised of the block diagrams in Figure 5, 10 and 11.

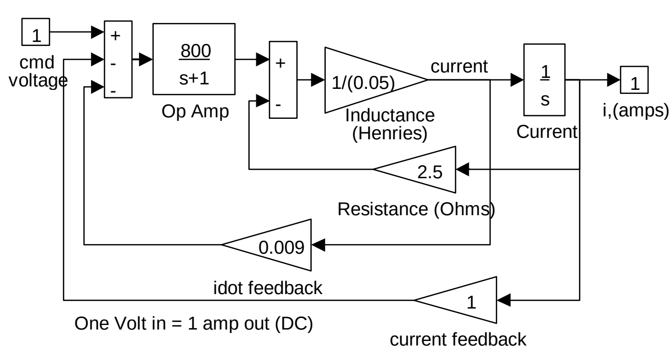

Figure 11 Servo-Amp and Electromagnet Electrical Model

THE CONTROL DESIGN

The control design would have been more involved had the mechanical system not had been so overdamped. With the damping, the mechanical system response was 10 times slower than the L/R coil dynamics. Thus the velocity or acceleration sensors that are typically used in electromagnetic control systems were not required. Instead, a gain scheduled PI control system was used. The control law was designed at three nominal operating points and the gains were scheduled between those points. A H-infinity control design scheme was used to build-in robustness due to uncertainty in the electromagnetic force. The subsequent model reduction resulted in a first order PI control system. Standard phase and gain margins were used to check the closed loop robustness of the reduced order controller for the linearized system at three operating points. The PI gains were scheduled with the clearance and the electric current. A filtered version of the controller output was used for the current since electric current follows the controller output fairly well. The controller block diagram is shown in Figure 12. This continuous design was discretized and implemented at a 2 kHz sampling frequency using an ISI AC100 and AutoCode™. The controller bandwidth is less than 10 Hz, but the computer was also used for high speed data acquisition.

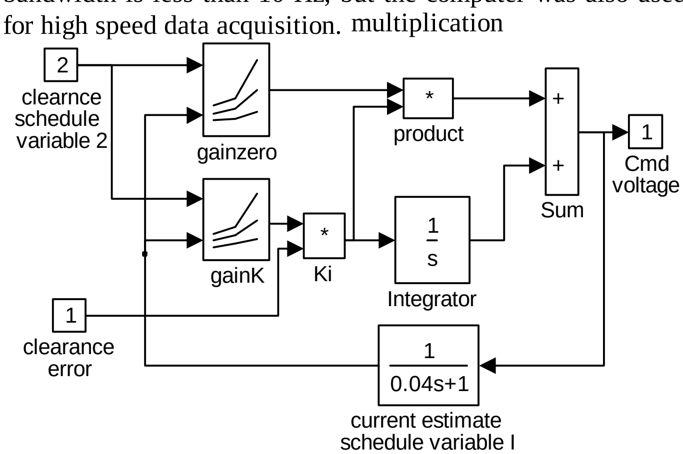

Figure 12 Scheduled PI Controller Block Diagram

CLOSED LOOP RESULTS

Figure 13 shows the regulation at 12.6 mils (0.32 mm) and then a step down to 6 mils (0.15mm). Note that the sensor used was not ideal and had a noise amplitude of about ±0.5 mil (0.013mm). Better sensor hardware would solve this problem. The step response has a large overshoot (2 mils (0.05mm) for the 6.6 mil (0.16mm) step). The 30% overshoot in Figure 13 is due to the difference between the model and the plant as the responses shown are with the controller designed on the model and no additional modifications were made to the controller during implementation. Refinement of the gain scheduling should reduce the oscillations at the lower clearances.

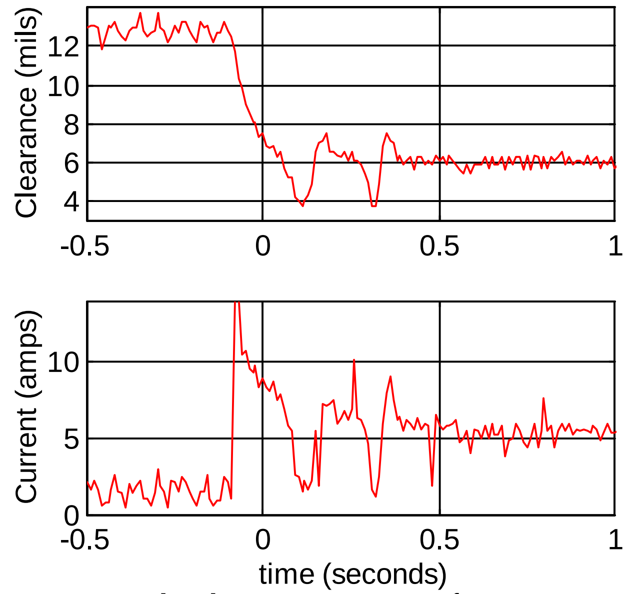

Figure 13 Closed Loop Step Response from 12.6 to 6 mils

Figure 14 shows the fail safe feature of this ATCCS. When the control system is turned off (or fails), the Belleville spring force increases the clearance. In Figure 14, the control system was regulating at 10.6 mils (0.27mm) and was suddenly turned off. The clearance increased to an equilibrium position at about 13.6 mils (0.345mm). This equilibrium is determined by the spring rate and the aerodynamic loading on the impeller which is a function of the compressor operating point. Also notice in Figure 14 the first order response of the uncontrolled system as it settles to this new operating position.

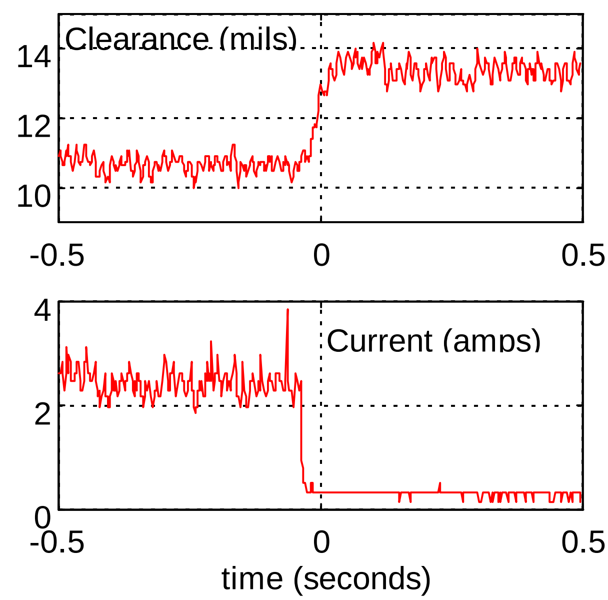

Figure 14 Fail Safe Response When Controller is Turned Off

During the rig test a clearance audit was taken. The compressor operating point was determined at constant corrected speed and a bleed port was adjusted to change the operating point on the speed line. This procedure was executed for both the open-loop and the closed loop control system. During closed loop control the tip clearance was regulated at 5 mils (0.127mm) less than the nominal open-loop clearance. Figure 15 shows the results for the change in the operation on the performance map for the 80.5% and 90% speed operating points. At 80.5% speed the data shows a 1.8% increase in airflow and a 2% increase in pressure ratio. At the 90% speed line the airflow and pressure ratio both increased by 1.9%.

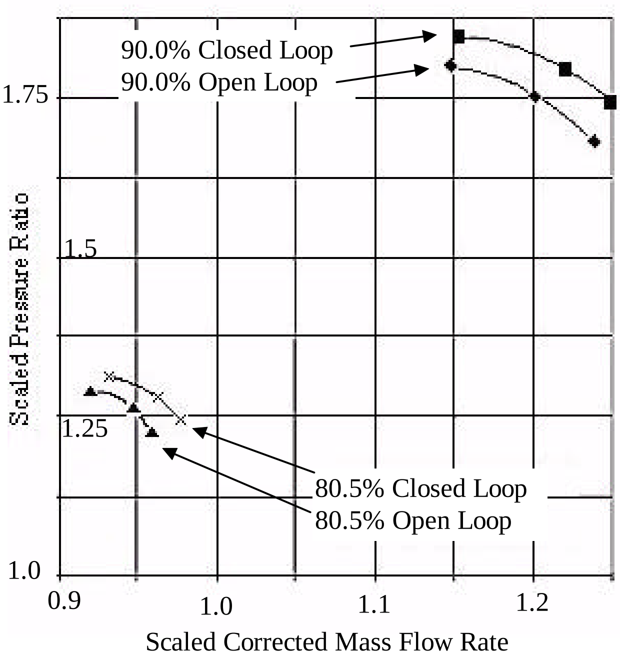

Figure 15 Changes in Operating Point Due to Controller

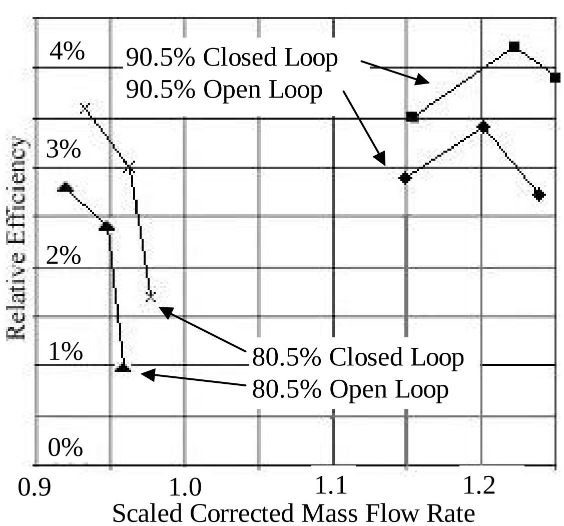

Figure 16 Changes in Efficiency Due to Controller

Figure 16 provides the corresponding changes in the compressor efficiency with tip clearance control active at the 80.5% and 90% speed operating points. Note that the indicated efficiency increased by at least 0.5 percent for a 5 mil (0.127mm) reduction in the tip clearance.

OBSERVATIONS

The PWM servo amplifier is appropriate for this application, but for this rig test a linear amplifier would have caused less problems. The PWM switching frequency appeared in the capacitance sensors and better isolation and grounding would help to alleviate this problem.

The compressor was surged during closed loop control. A gross deflection of about 4 mils (0.10 mm) occurred during surge with and without the controller active. When active, the controller did return the clearance back to the original setpoint after the surge. The best the control system can do is to turn the current off and allow the spring to pull the impeller back away from the shroud to increase the tip clearance. However, this is exactly what would happen under open-loop conditions with the controller off. While the electromagnetic can assist in the damping of the oscillations, it is up to the spring to resist the loss in clearance during a surge event.

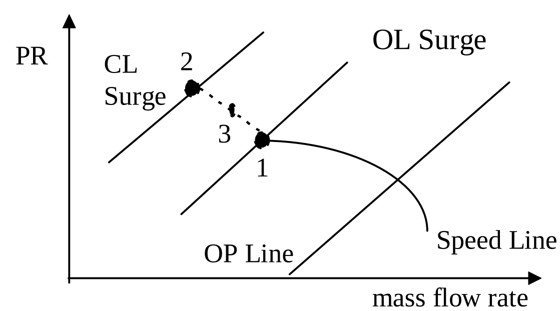

Figure 17 Operating Points on the Compressor Map

One test of interest was to determine the clearance response if the controller electric current was lost while near the surge line during closed loop control. Refer to the sketch of the compressor map in Figure 17 for the positions of the test points. Surge was found at point #1 during open loop (OL) operation. The controller was then turned on and the clearance was reduced by approximately 5.0 mils (0.127mm). This established a new surge point at point #2 on the closed loop (CL) line. It was hypothesized that by moving the operating point to point #3 and turning the controller off, that the compressor should drop into surge because the operation was beyond the OL surge line established at point #1.

This was demonstrated on the rig as shown in Figure 18. The compressor was initial at point #2 with a clearance of about 11 mils. Then the controller was turned off. As the clearance increased, the compressor moved towards point #1 and surged.

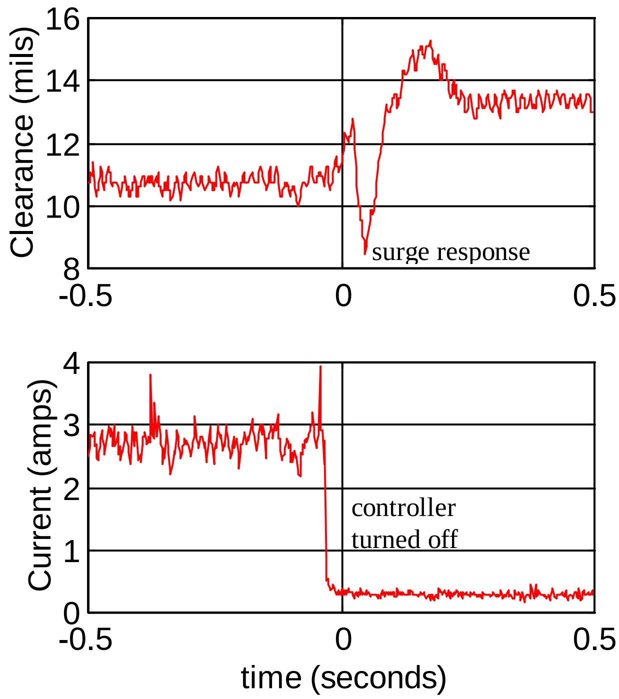

Figure 18 Induced Surge

Under closed loop control two mechanical differences were observed. First, the vibration level of the compressor was reduced by 60% as seen in Figure 19. While obtaining quantitative whirl data was not possible due to the lack of a sufficient number of clearance probes, and the associated spatial clearance information, the reduction in vibration amplitude infers that the whirling amplitude is reduced. The second effect occurred when the compressor was in closed-loop control. The test stand operator would notice a distinct improvement in controllability of the rig speed in the sense that speed variations were almost nonexistent and the time to get on point was minimal.

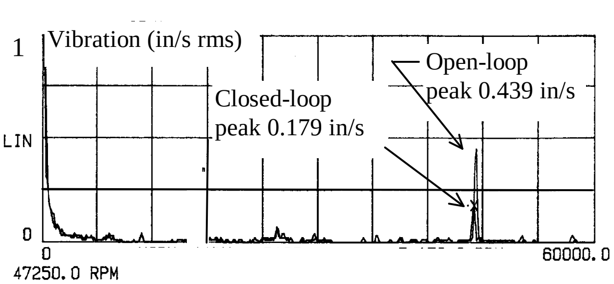

Figure 19 Spectrum of Rotational Vibration Levels

SUMMARY

The following conclusions can be drawn from this program.

1. The electromagnetic actuator design provided adequate force to run the test.
2. The mechanical system was well damped, apparently due to oil in the air gap.
3. The control system was able to control the clearance to approximately 4.0 mils. With further refinement of the gain scheduling, it is believed that control at small clearances is possible.
4. The performance effects of tip clearance changes were measurable and agreed with general experience on 250 engines. On average a 5 mil (0.127mm) reduction in the clearance resulted in a 1.9% change in pressure ratio and flow, and an efficiency increase of 0.5%.

The performance benefits of active tip clearance control described in #4 above are shown in this report. Also, a device such as the one tested could reduce the time to build a compressor. Much effort goes into “stacking” a compressor rotor. With an active tip clearance system a compressor could be built initially with larger clearances and then pulled in. The size of the electromagnet may be reduced if clearance control is only required near the cruise operating condition.

It would be useful to run durability tests on the electromagnetic actuator used in this program. Testing could be carried out on either a static or rotating rig. Durability of the coil is also a major issue in deploying this technology.

The 2 dimensional force table for the electromagnet needs to be validated. Control system design was based on an incomplete model. A validated model for the electromagnet would be the first step in refining the control system to prevent limit cycles at the smaller clearances.

The issues involved with the mechanical connection between the compressor and the turbine would have to be addressed before this technique could be implemented in an engine. This would involve a small axial motion while transmitting a torque through the shaft.

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