SURGE DETECTION IN AN INDUSTRIAL AXIAL FLOW COMPRESSOR

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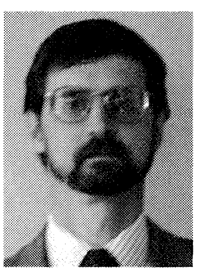

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ABSTRACT

Compressor surge can be a disruptive and potentially destructive phenomena in turbocompressors. A surge detection scheme utilized in shop and field testing of a 29,600 hp (22 MW) industrial axial compressor is described. Dynamic pressure transducers were installed in the compressor casing to monitor rotor blade-to-blade dynamic pressure changes indicative of rotating stall cells. Rotor shaft radial vibration was also monitored to correlate with the occurrence of surge.

The onset of surge could clearly be identified in both the shop performance testing and actual field operation. Low frequency, subsynchronous stall cells were detected prior to an actual flow breakdown and flow reversal. The detection of incipient surge allowed corrective measures to be taken to prevent the compressor from surging. The instrumentation used to detect incipient surge and the results of the compressor testing are described.

BACKGROUND

The subject compressor is installed to provide combustion air to a fluid catalytic cracking process in an oil refinery. The compressor is one component in a five body train comprised of a hot gas expander, axial compressor, steam turbine, speed reducing gear, and motor/generator. The compressor is driven by the hot gas expander during normal operation. The steam turbine and motor/generator in the train are used for startup, power supplementing, and electrical power generation.

The compressor was tested to assure maximum operational reliability by determining that no adverse operating conditions would be encountered during the unit startup/shutdown transients along with during normal operation.

The testing also provided a definition of the compressor surge limits that could be programmed into the antisurge control system. Axial compressors such as the one described often operate throughout a wide range of operating parameters that can drive the compressor into choke, surge, or stone walling. A minimum continuous operating time between inspections and refurbishment of the equipment is 35,000 hr.

The testing described herein was performed in conjunction with ASME performance testing of the compressor in a shop test facility. In addition, tests were conducted at the field installation under the actual operating conditions.

EQUIPMENT DESCRIPTION

The subject compressor is a fifteen stage industrial axial flow compressor. A photograph of the subject compressor on the test stand can be seen on Figure 1. The compressor is designed with double circular arc (DCA) rotor blades in the first three stages and NACA 65 blades in the remaining rows. The hub diameter of the compressor is 30.0 in (76.2 cm).

Variable geometry inlet guide vanes and six rows of stator blades are used in the inlet end of the compressor to increase the compressor's overall operating range. The remaining stator blade rows and exhaust guide vanes are fixed geometry. The blading is designed to be of the 50 percent reaction style.

The compressor operates at a fixed rotational speed as a result of the use of the motor/generator in the train of equipment. The compressor design conditions are highlighted in Table 1.

INSTRUMENTATION

The testing program implemented various kinds of instrumentation devices as depicted in Figures 2, 3, and 4. The compressor was

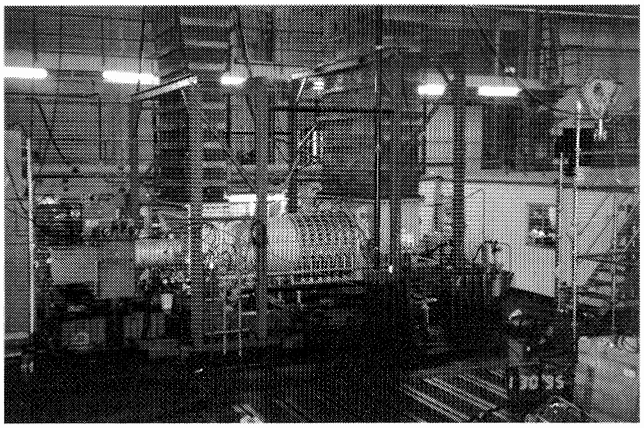


Figure 1. Tested Axial Flow Compressor.

Table 1. Compressor Design Conditions.

Compressor Design Conditions		
	English Units	Metric Units
Inlet pressure	14.6 psi	1.03 kg/cm2a
Inlet temperature	68° F	20° C
Discharge press.	70.2 psia	4.94 kg/cm2a
Discharge temperature	437° F	225° C
Pressure Ratio	4.8	4.8
Flow	186,000 SCFM	318,000 NM3/hr
Power	29,600 Hp	22,080 Kw

fitted with typical industrial supervisory instrumentation that included:

- Rotor noncontact proximity probes for rotor radial vibration and axial position
- Embedment type thermocouples for bearing temperatures
- Electronic transmitters for inlet and discharge air pressures
- Probe thermocouples for inlet and discharge air temperatures
- Compressor flow nozzle instrumentation
- Inclinometer for stator vane angular position

In addition, the following instrumentation was utilized during the shop performance testing:

- Additional performance instrumentation per ASME power test code PTC-10
- Bearing housing triaxial accelerometers for bearing housing vibration
- Static pressure and temperature transducers as required by ASME power test code PTC-10
- Additional dynamic pressure transducers for interstage blade passing pressure profiles (Figure 4)

The dynamic pressure transducers were installed at various stages in the compressor during the two tests. Data were collected from stages 4, 8, 12, and 15 at several time periods to

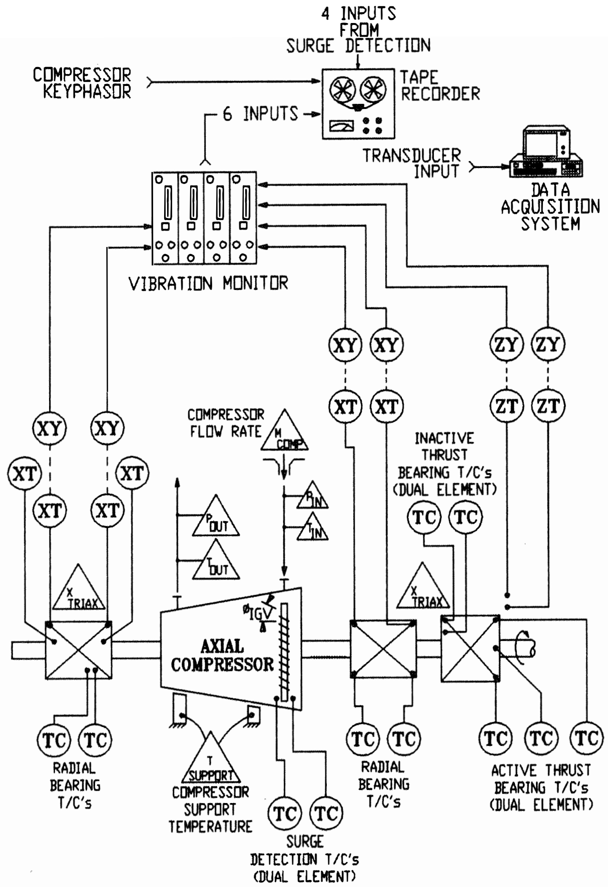
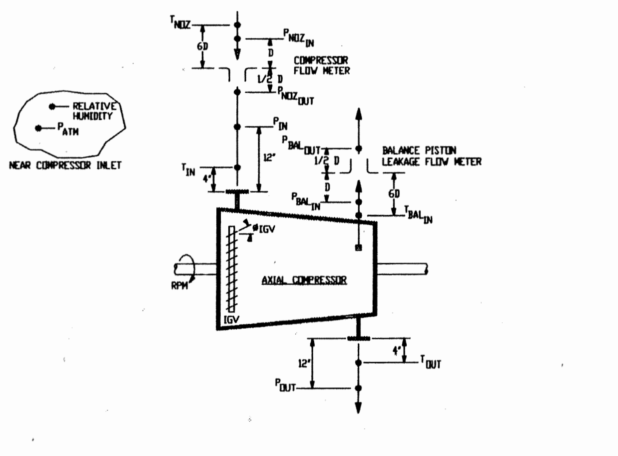


Figure 2. Mechanical Instrumentation System.



PTC-10 PERFORMANCE TEST DATA

SYMBOL	DESCRIPTION	TRANSDUCER(S)
RPM	ROTOR SPEED	SHOP DRIVER
P_{IN}	COMPRESSOR INLET STATIC PRESSURE	8 STATIC PRESSURE
T_{IN}	COMPRESSOR INLET STATIC TEMPERATURE	4 TYPE J T/C's
ϕ_{IGV}	IGV ANGLE	INCLINOMETER/MANUAL
P_{OUT}	COMPRESSOR DISCHARGE STATIC PRESSURE	8 STATIC PRESSURE
T_{OUT}	COMPRESSOR DISCHARGE STATIC TEMPERATURE	4 TYPE J T/C's
T_{NOZ}	FLOW METER UPSTREAM TEMPERATURE	4 TYPE J T/C's
$P_{NOZ_{IN}}$	FLOW METER UPSTREAM PRESSURE	4 STATIC PRESSURE
$P_{NOZ_{OUT}}$	FLOW METER DOWNSTREAM PRESSURE	4 STATIC PRESSURE
P_{ATM}	ATMOSPHERIC PRESSURE	1 STATIC AMBIENT PRESSURE
RH	ATMOSPHERIC RELATIVE HUMIDITY	
$P_{BAL_{IN}}$	BALANCE PISTON LEAKAGE FLOW METER UPSTREAM PRESSURE	2 STATIC PRESSURE
$P_{BAL_{OUT}}$	BALANCE PISTON LEAKAGE FLOW METER DOWNSTREAM PRESSURE	2 STATIC PRESSURE
$T_{BAL_{IN}}$	BALANCE PISTON LEAKAGE FLOW METER UPSTREAM TEMPERATURE	2 TYPE J T/C's

Figure 3. Performance Instrumentation.

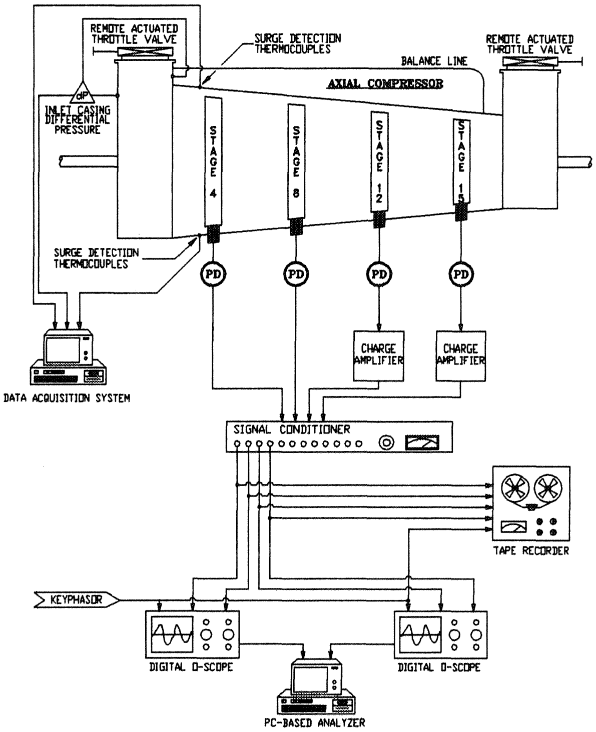


Figure 4. Dynamic Pressure Transducer Data Systems.

determine when flow instabilities were created in the compressor. These transducers were installed radially through the outer casing in the mid chord region of the rotor blades.

In this configuration the probes also gave an insight into the quality of the flow through the respective stages. Under ideal flow conditions, such as operation around the compressor design point, the pressure profile will exhibit a smooth sinusoidal characteristic in the rise and fall of the waveform from the suction surface of one blade to the pressure surface of the next blade. Each passing blade passage displays the same repeatable waveform characteristic.

At high blade incidence angles and stable flow, the boundary layer on one blade surface will thicken while consecutive blade passages will continue to display the original pressure rise characteristics. The flow in the cascade becomes irregular from one blade passage to the next as the blade incidence exceeds the stalling angle. As the compressor approaches and enters surge, the pressure rise characteristic becomes unstable and grows in amplitude to the point of causing saturation of the pressure transducer signals.

The installation of multiple dynamic pressure transducers in a single stage allows the determination of the propagation of stall cells, their number and angular velocity during rotating stall. The phase angle between the multiple pressure transducers determines the number of stall cells, while the rate of propagation can be inferred from the frequency of the stall cells.

The installation of the dynamic pressure transducers was critical to assure consistent and predictable data. The pressure probes are separated at a pre-determined circumferential spacing, away from the stator vane flow wakes. Also, the dynamic pressure transducers must be installed such that the transducer diaphragms are flush with the flow path outer diameter. This installation detail is critical, since improper installation can lead to dampened pressure responses and potential interference with 1/4 wave Helmholtz resonances in the casing pressure tap passages.

The dynamic pressure transducers were selected to be capable of the required static pressure and temperature levels. The high temperature 480°F (250°C) of the discharge end stages required the use of a high temperature type dynamic pressure transducers for these stages.

The transducers also had to have suitable frequency response and dynamic pressure ranges to monitor the pressure pulsations generated by the rotating blades.

The dynamic pressure transducer output signals were viewed in a real time mode on dedicated oscilloscopes, FFT analyzers and simultaneously recorded independently to a WB GR-1 data storage recorder for post test analysis.

Static pressure transducers were installed in the interstage area during some of the testing to get the individual stage pressure rise data.

SHOP TESTING

A 28 point performance test was conducted to define the compressor's operating envelope. Seven stator vane settings with three operating points per vane setting were executed to define the performance characteristics. Also, compressor surge points were defined at the seven vane angles settings to complete the definition of the performance map envelope. The shop test was conducted with the compressor in a suction throttled condition to reduce the shop driver requirements to 18,800 hp (14 MW).

The dynamic pressure transducers were capable of monitoring the rise and fall of the blade to blade passing pressure profile. The predominant frequency of the blade passing pressure pulses was 2.2 kHz, which corresponded to the compressor rotational speed multiplied by the number of blades.

A typical, normal, time base signature from two dynamic pressure transducers is shown in Figures 5 and 6. The same data is also depicted in the frequency domain in Figure 7.

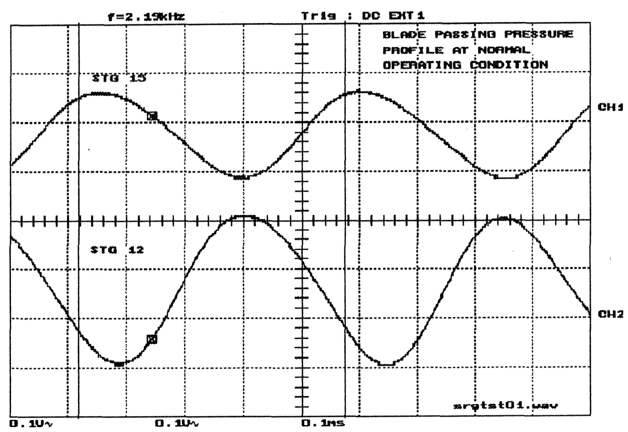


Figure 5. Dynamic Pressure Transducer Time Base Signature for Stages 12 and 15 (Δt = 0.1 ms).

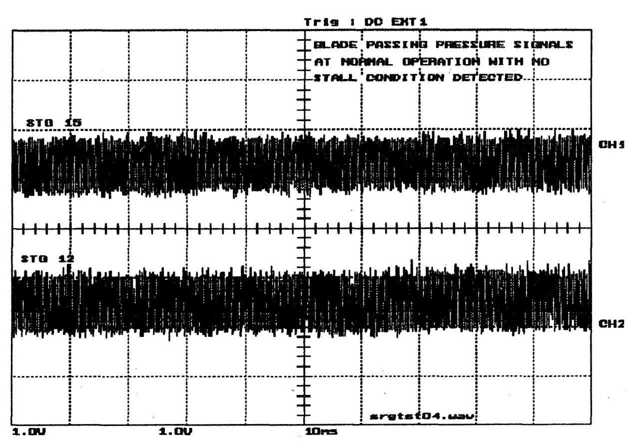


Figure 6. Dynamic Pressure Transducer Time Base Signature for Stages 12 and 15 (Δt = 10 ms).

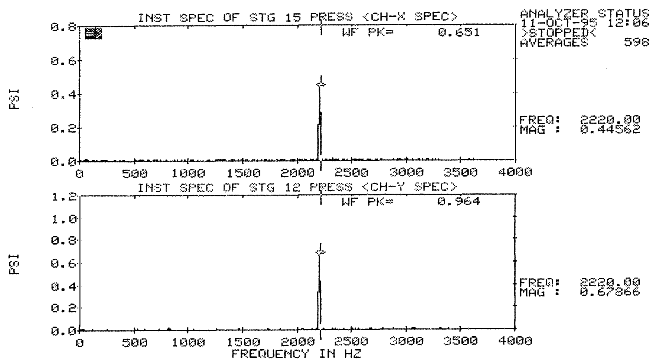


Figure 7. Dynamic Pressure Transducer Signals Displayed in Frequency Domain.

FIELD TESTING

Similar dynamic pressure data was collected on an identical compressor in the field installation.

The testing was used to confirm that the compressor was operating without any aerodynamic instabilities throughout the entire operating range. The field test was conducted without suction throttling and under the full and normal process conditions.

RESULTS

The testing was found to be extremely successful in the detection of incipient and full surge in the compressor. The dynamic signals from pressure transducers were found to be remarkably detailed to allow controlled and continuous operation just below the full surge point or at the incipient surge point of the compressor before surge could be detected by any other means.

Under closed stator conditions such as during a compressor start up, flow instabilities were detected in the middle stages of the compressor. The flow instabilities showed up as low frequency sinusoidal oscillations superimposed on the blade to blade pressure fluctuations. The blade to blade pressure distributions typically showed irregular and smaller peak amplitudes during this low flow condition.

The cause of these fluctuation signals at the low flows is not completely understood, although hub section stalling, acoustical resonance, or blade flutter is suspected. Subsequent compressor field operation excluded closing the stator vanes into the region where the instabilities were noted and a minimum compressor discharge pressure was thus defined.

The data suggested that certain stages in the compressor were not performing useful work.

During the compressor performance test, the compressor was intentionally brought into a surge condition. The interstage dynamic pressure transducers detected flow instabilities well before the compressor actually encountered full surge.

A full compressor surge occurrence was defined when the compressor ceased to produce a forward flow and the flow reversed momentarily. The flow reversal occurrence was determined from audible noise, rotor vibration, discharge pressure, and suction flow instabilities, as observed on the test supervisory instrumentation.

As the compressor operating point moved toward the surge condition, the dynamic pressure transducers began to detect flow instabilities as a boundary layer growth in a flow passage developed, indicating an incipient surge condition. As the flow instabilities developed and grew in magnitude, the blade passing waveforms were observed to modulate by a superimposed low frequency carrier type wave signature. Just prior to surge, the low frequency sinusoidal pressure waveform became well developed and grew in magnitude. This low frequency flow instability can be seen in Figures 8, 9, and 10. Figure 8 can be directly compared to that of Figure 6.

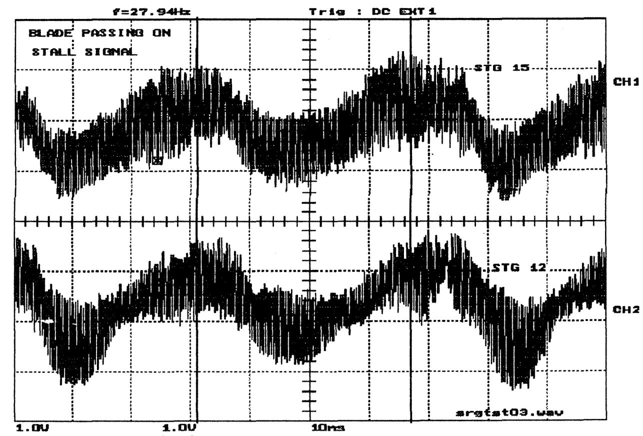


Figure 8. Time base Waveform of Compressor Operation During Incipient Surge.

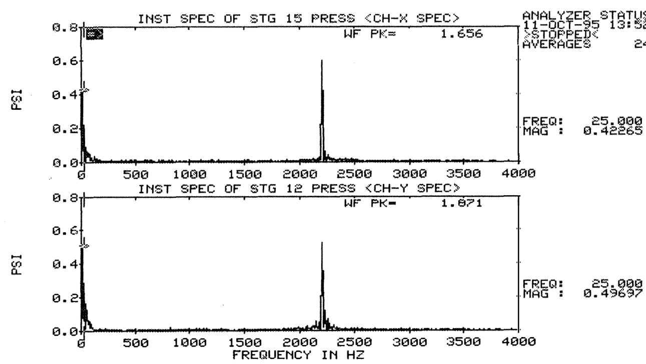


Figure 9. Frequency Spectrum During Compressor Incipient Surge.

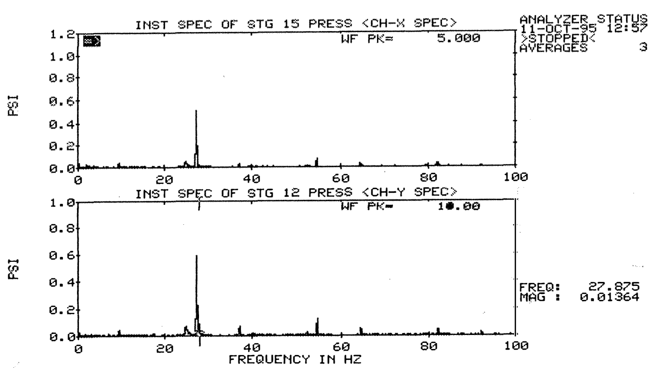


Figure 10. Frequency Spectrum During Compressor Surge (zoomed to stall frequency).

As the compressor approached the surge point, the low frequency stalling amplitudes increased dramatically. The amplitude of the low frequency oscillations exceeded the blade to blade pressure amplitude by more than a factor of three. The frequency range of the oscillations was found to be from 26 to 32 Hz, which corresponded to roughly one half the rotor speed frequency. During the field testing, subsynronous frequencies at 40 percent and 50 percent of the blade passing frequency were noted. The low frequency was noted throughout all compressor stages although the magnitude of the pressure oscillations varied significantly from stage to stage. The strongest flow instabilities were detected on the discharge end of the compressor.

The magnitude of the pressure fluctuations during the presurge subsynchronous aerodynamic instabilities was found to be comparable to the average pressure rise per blade row. During a surge event, the dynamic pressures increase dramatically. Surge dynamic pressures in excess of ± 30 psi (± 2.2 kg/cm2) were encountered. The maximum value of these pressure surges could not be determined since the transducers saturated at the 30 psi signals. These large pressure fluctuations are significant from a blade structural loading standpoint and are potential sources of significant blade excitation and alternating stress.

The compressor was found to operate in the incipient surge condition with good stability. A compressor performance test point could be held and taken within 0.1 psi (0.007 kg/cm2) of the surge point without fear of surging the unit.

The testing showed the actual compressor surge line to be higher than that predicted by analytical calculations (i.e., thus yielding a greater operating range). The flow instabilities and surge conditions were found to occur primarily toward the discharge end of the compressor and propagate towards the inlet.

Some difference was noted in the wave form shapes between the testing in the shop environment and those of the field. The dynamic signals were sinusoidal in shape during the shop test while the field signals were more stepped or saw tooth in shape. The characteristics may be different due to the differences in flow conditions and power levels that were experienced during the two tests (throttled inlet conditions during shop test).

The technology used to test the subject compressor and the knowledge gained was considered to be very successful and has future potential.

Dynamic pressure testing will allow compressor surge characteristics to be determined without intentionally surging the compressor. Testing can also be repeated at any time to determine the effects of compressor fouling on the surge characteristics. This type of test will potentially allow compressor anti-surge control systems to be set up with lower control margins since the actual surge points can be clearly identified.

The identification of a surge line without surging the compressor is desirable from a user standpoint since the process will not have to be subject to upset conditions. Not subjecting the compressor to multiple surge conditions is also desirable from a compressor life standpoint. In addition, future surge control backup systems may be developed to detect incipient surge prior to an actual flow reversal.

ROTOR SHAFT VIBRATION

Throughout the testing periods, the rotor and bearing housing vibrations were closely monitored to determine if adverse operation such as a surge condition could also be detected via rotor shaft vibration. Indeed, compressor surge was detected in the vibration instrumentation. The rotor vibrations were found to increase and become unstable at the point, or just prior to, the actual surging of the compressor. During a surge event, subsynchronous vibrations were found to occur on both ends of the rotor. A comparison of the vibrations just prior to surge and during the surge event can be seen in Figures 12 and 13. Operation during light incipient surge is shown in Figure 11. Note that no subsynchronous activity is present.

The interesting conclusion is that the dynamic pressure transducers could detect an incipient surge condition well before the flow instabilities excited the machine mechanically. The radial vibration probes were found to give little warning that the compressor flow was about to break down. During the surge event, the rotor did, however, respond to the aerodynamic flow reversal.

RECOMMENDED FUTURE WORK

During the installation and compressor testing, it was found that the dynamic pressure transducer instrumentation was susceptible to handling and mechanical damage due to their delicate and lightweight design. Never the less, the instrumentation performed well throughout the test periods of several days. However, the long term reliability of instrumentation should be questioned since the removal of the instrumentation after 16,000 hr showed it all to be nonfunctional.

Development work is required to produce high temperature dynamic pressure transducers that are capable of providing long term reliable operation if they are to be used for continuous real time monitoring when the machine is in service.

It is not believed that the dynamic pressure transducers will give sufficient warning of a surge condition if the compressor is in a rapidly changing operating mode of operation. Only under controlled operating conditions, did the dynamic pressure probes give excellent warning and controllability. Further work is needed to determine if this type of instrumentation can be applied as a surge control input.

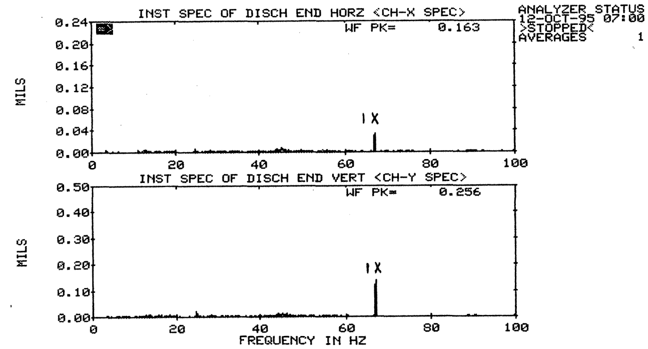


Figure 11. Discharge End Rotor Vibration (mild incipient surge conditions).

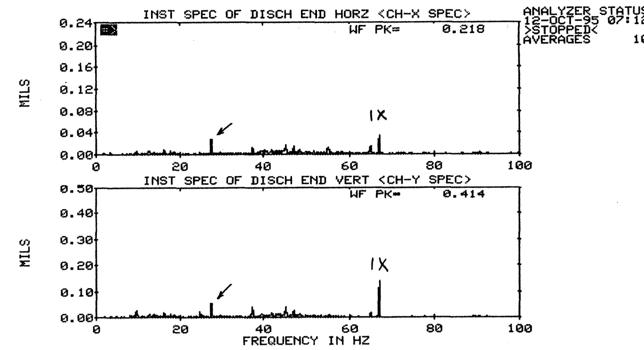


Figure 12. Discharge End Rotor Radial Vibration (well developed incipient conditions).

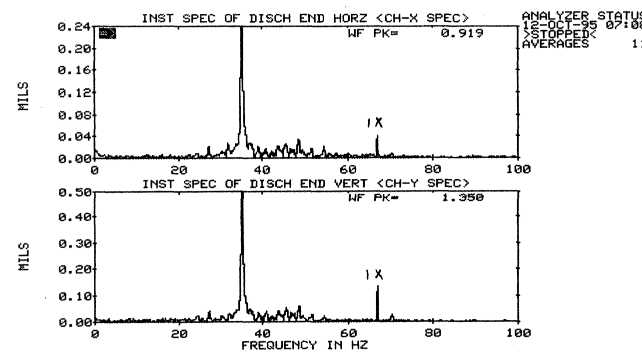


Figure 13. Discharge End Rotor Radial Vibration (during full surge conditions).

CONCLUSIONS

- The detection of compressor aero instabilities can be detected prior to a complete flow breakdown or reversal.
- Dynamic pressure transducers are an effective method of detecting flow instabilities and surge in axial flow compressors.
- Potentially greater compressor operating ranges can be identified with flow path dynamic pressure transducers.
- The dynamic pressure transducers permit very predictable and controllable operation of the compressor up to surge.
- Mechanical vibrations respond during compressor surge but do not provide adequate resolution or forewarning to prevent surge.
- The currently available instrumentation is not suitable for long term compressor operation or as a surge control input.

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