INCREASED FLEXIBILITY OF TURBO-COMPRESSORS IN NATURAL GAS TRANSMISSION THROUGH DIRECT SURGE CONTROL

Annual Technical Progress Report
October 2001 — April 2003

Prepared by

Robert J. McKee

May 2003

SwRI® Project No. 18.04990
DOE Award No. DE-FC26-01NT41163

Prepared for

U. S. Department of Energy
National Energy Technology Laboratory
3610 Collins Ferry Road
P.O. Box 880
Morgantown, WV 26507-0880

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SOUTHWEST RESEARCH INSTITUTE®
6220 Culebra Road
San Antonio, Texas 78238

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Approved:

Danny M. Deffenbaugh, Director
Mechanical and Fluids Engineering Department

DISCLAIMER

ABSTRACT

This preliminary Phase 1 report summarizes the background and the work on the "Increased Flexibility of Turbo-Compressors in Natural Gas Transmission through Direct Surge Control" project to date. The importance of centrifugal compressors for natural gas transmission is discussed, and the causes of surge and the consequences of current surge control approaches are explained. Previous technology development, including findings from early GMRC research, previous surge detection work, and selected publications, are presented. The project is divided into three Phases to accomplish the project objectives of verifying near surge sensing, developing a prototype surge control system (sensor and controller), and testing/demonstrating the benefits of direct surge control.

Specification for the direct surge control sensor and controller developed with guidance from the industry Oversight Committee is presented in detail. Results of CFD modeling conducted to aid in interpreting the laboratory test results are shown and explained. An analysis of the system dynamics identified the data sampling and handling requirements for direct surge control. A detailed design process for surge detection probes has been developed and explained in this report and has been used to prepare drag probes for the laboratory compressor test and the first field test.

The surge detection probes prepared for testing have been bench tested and flow tested to determine and calibrate their sensitivity to flow forces as shown in data presented in this report. The surge detection drag probes have been shown to perform as expected and as required to detect approaching surge. Laboratory test results of surge detection in the SwRI centrifugal compressor demonstrated functionality of the surge detection probes and a change in the impeller inlet flow pattern prior to surge. Although the recirculation cannot be detected because of the specific geometry of this compressor, there are changes that indicate the approach of surge that can be detected. Preparations for a field test had been completed at one point in the project. However, a failure of the surge probe wiring just inside the compressor case has caused a delay in the field testing. Repairs for the wiring in the compressor have been scheduled and the field test will take place shortly after the repairs.

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1. INTRODUCTION

The capacity and flexibility of the Natural Gas Transmission system within North America will increase over the next decade in order to meet the needs of our nation's population. To enable this expansion of pipeline capacity and to increase the system's responsiveness to the ever changing demands of energy buyers and sellers using technically advanced and cost-effective solutions, the Department of Energy (DOE) has established a Natural Gas Infrastructure program to foster research and development that will enhance the Natural Gas Transmission system within the United States. The development of a Direct Surge Control sensor and controller to increase flexibility of turbo-compressors is a part of the Natural Gas Infrastructure program. The objectives of the Direct Surge Control project are to refine surge control, reduce surge margins, increase efficiency and flexibility, and safely save energy and operating dollars.

The Direct Surge Control research program is being conducted by Southwest Research Institute® (SwRI®) with cooperation from the Natural Gas Industry, including two participating pipeline companies. Both Duke and El Paso Energy are providing input and ideas for the project and have offered to provide field sites and cooperation for testing and demonstrations. The co-funding partners are the Gas Machinery Research Council (GMRC), which is a member association of operating companies and equipment suppliers that support and decimate research that benefits the Natural Gas Industry and Siemens Energy and Automation, a GMRC member company interested in commercialization of a Direct Surge Control system. An Oversight Committee with representatives from the pipeline companies (Siemens) another GMRC member company (Solar Turbines), SwRI, and DOE has been established to provide industry's guidance and direction for this project. The entire "Increased Flexibility of Turbo-Compressors in Natural Gas Transmission Through Direct Surge Control" project is planned as a three phase and three-year program that includes verification of near surge sensing, development of a prototype surge controller, and testing and demonstration of the benefits of a prototype surge controller. The first phase will require more than one year (at least 18 months) and includes concept verification testing. This report is a summary of the efforts and results through most of the first phase.

1.1 BACKGROUND

Centrifugal compressors constitute a significant portion of the compression equipment used along North America's pipelines and represent approximately half of the installed Natural Gas transmission horsepower. In addition, turbo-compressors are used in gathering, processing, and other industrial plants that compress hydrocarbon gases and represent a major portion of all large gas compression equipment. Because of their economic advantages, turbo-compressors are becoming increasingly common and will continue to be installed in significant numbers. Centrifugal compressors operate by adding angular momentum to the gas within a rapidly spinning impeller and recovering that energy as pressure in a diffuser passage. Through this mechanism, turbo-compressors can effectively operate over a range of speeds, flows, and pressure rise. The pressure rise created by a centrifugal compressor is often expressed as head rise, in feet of suction gas, or as pressure ratio (discharge pressure divided by suction pressure) across the compressor. Figure 1-1 is a typical performance map for a pipeline centrifugal

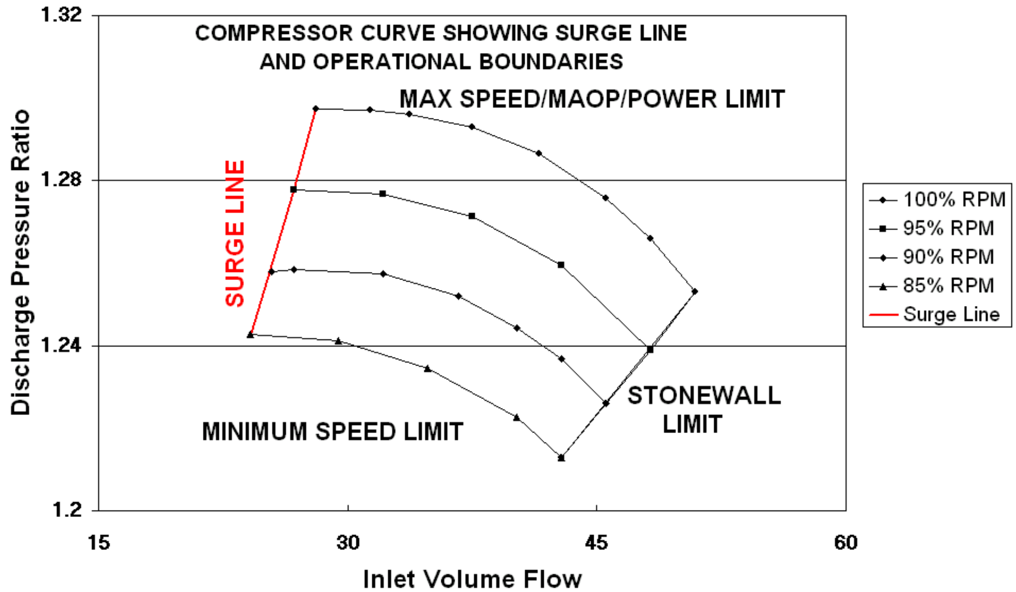


Figure 1-1. Typical Compressor Performance Map Showing Pressure Ratio as a Function of Inlet Volume Flow for Various Speeds with Operation Boundaries of Pressure, Speed, Power, Stonewall Flow and Surge or Minimum Stable Flow.

compressor, which shows the pressure ratio as a function of inlet volumetric flow for a range of compressor speeds and indicates that there are limits on the operating range of such a compressor. One or more of the maximum safe speed, maximum allowable discharge pressure, or maximum available power sets the upper limit of the operating map. At the high flow end of the performance map, the limit is set by stonewall or the maximum flow velocity that can pass through the compressor passages. If the stonewall condition limits flow, more flow can only be achieved with a physically larger compressor. In the low-pressure ratio portion of the performance map, the limit on operations is set by either the minimum speed at which the driver can provide power for the compressor or the minimum speed at which the compressor is effective. On the low flow side of the compressor curves, the actual limit is set by the flow instability known as surge. However, a surge control line is usually established with a margin above the actual surge condition, and this line actually sets the operational limit. Typical surge margins between the surge control line and the flow at which surge may occur is usually 10% of the design flow, and it is this margin and control of the surge limited operation of the compressor that is addressed by the current development.

Surge in centrifugal compressors is an instability in which a decreasing head for a decrease in flow, in contrast to the normal increase in head for a decreasing flow, causes an unstable or run away drop in flow. Surge is in effect a change in the slope of the compressor curve from the normal negative slope to a near zero or positive slope at the surge condition. The result of the decreasing head for a lower flow is that the entire compressor's flow decreases further, which further reduces the head, such that the flow pattern through the compressor collapses and flow within the compressor reverses, flowing from discharge to suction through the forward spinning impeller. Surge is a global instability in a centrifugal compressor, in contrast to local stalls, and results in a complete collapse and reversal of flow through the compressor.

Immediately following a surge event, the suction pressure is higher and the discharge pressure is lower, than prior to surge, so that the compressor is then able to compress (move gas) against the lower head and it begins to operate normally again. However, the original system condition that caused surge at a high head and low flow exists, such that as the compressor operates normally the head increases and flow returns to a low flow condition where surge is likely to occur again if the compressor and piping system do not change. The consequence of this surge, recovery, and repeated build up to surge is a surge cycle as shown in terms of flow and head in Figure 1-2 [PCRC Report No. TR 84-4[1]]. Such a surge cycle is an energetic event with flow reversing and then returning to a large magnitude forward flow before decreasing and again reversing during the next surge. The surge sequence with reverse flow through a forward spinning compressor causes extreme loads on the thrust bearings and large loads on other components of the compressor with the result that thrust bearing failures and other damage to the compressor can occur during surge. Surge is a source of large dynamic forces on the compressor elements and, hence, a flow phenomena that must be avoided.

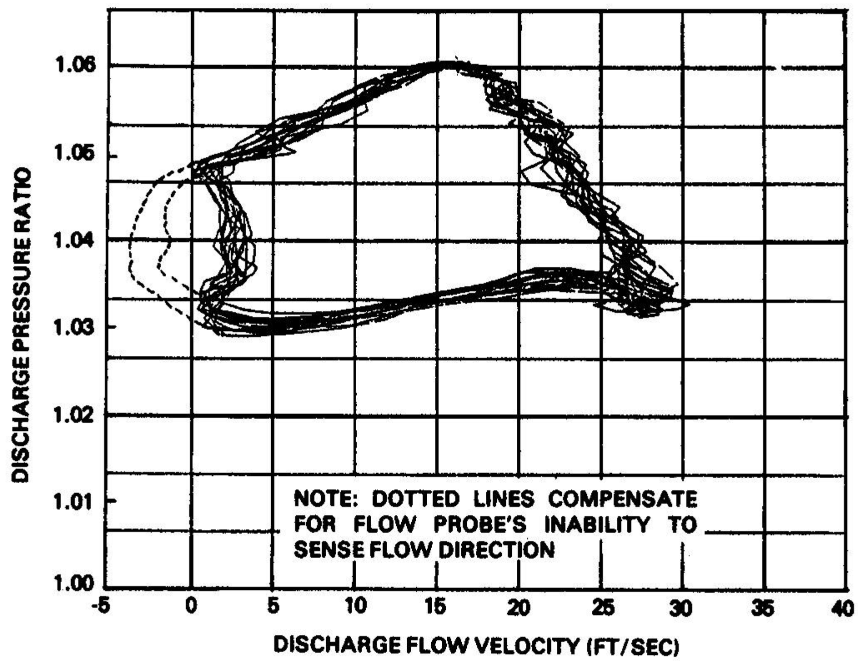


Figure 1-2. A Surge Cycle as Measured with Fast Response Pressure Transducers and a Hot Film Anemometer Shows Large Energetic Changes in Flow and Pressure.

Surge avoidance is essential for all centrifugal compressors and is normally achieved by selecting a minimum safe flow and recycling gas around the compressor to maintain no less than this minimum volumetric flow rate. The minimum allowable flow is known to be a function of speed or head, such that in addition to flow rate, head across the compressor is normally measured. In some systems, the speed is measured so the surge control line (minimum flow) can be adjusted based on what the head should be for the measured speed. Measurements used to place a compressor on its performance map in terms of flow and head relative to the surge control line are external measurements. A surge control line with some margin of flow above the flow at which surge is suspected is shown in Figure 1-3, which shows a typical surge margin.

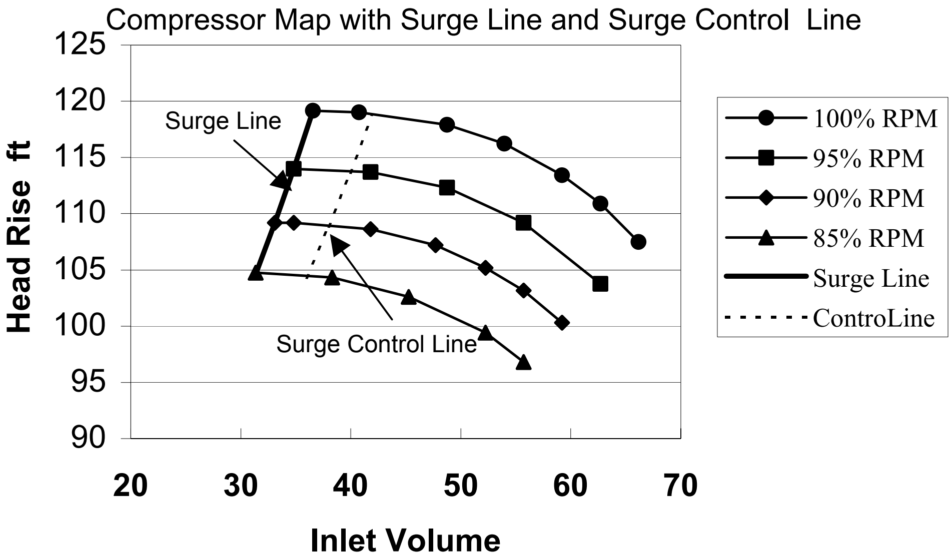


Figure 1-3. Typical Compressor Operating Map Showing the Approximate Surge Line and the Surge Control Line with a Surge Margin Used to Avoid Surge.

The reason that a margin above the suspected surge limit is required is that the exact condition at which surge occurs cannot be determined from external measurements. Research has shown that the precise head and flow at which surge occurs depends on speed, gas properties, piping characteristics, and dynamic disturbance in the system. Thus, the surge limit is different on the test stand than it is for a field installation, and the surge limit at a field site can change if gas properties or disturbances within the system change. In some compressors, a pressure pulsation or a vibration develops prior to surge and can be used as an indication that the compressor is approaching surge. The margin above surge, at which pulsation or vibration develops, is not known, depends on many factors, and may be large or small for different compressors. Pulsation or vibration seen in some compressors is a result of stall rather than surge and is not a good indication that the compressor is approaching surge. In other compressors or under different operating conditions, there is no pulsation or vibration prior to surge with the result that the compressor can reach surge unexpectedly. Because of these uncertainties as to the conditions at which surge occurs and the potential damage from surge, a significant margin of approximately 10% in flow is normally established between the suspected surge line and the surge control line to ensure that surge is avoided.

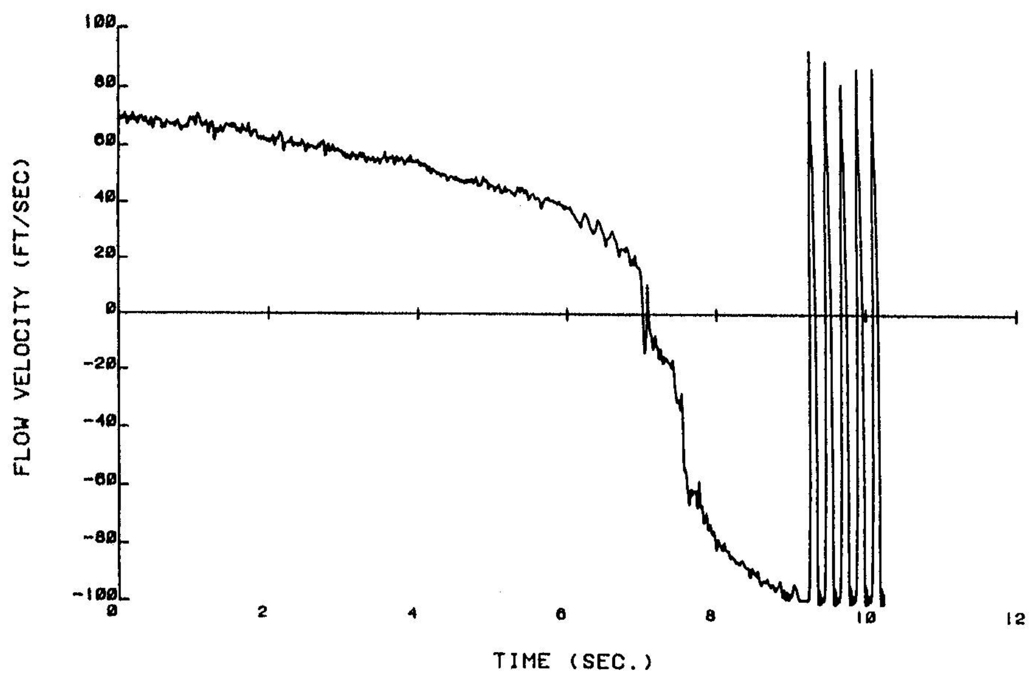
As the operating flow is decreased towards the surge control line, a recycle valve, which connects the discharge line to the suction line, is opened and flow that has been compressed is returned through a pressure loss to the suction side of the compressor. Regardless of the amount by which the throughput gas flow is reduced, the flow rate seen by the compressor can be maintained by opening the recycle valve further and recompressing flow that has already passed through the compressor. Opening the recycle valve tends to reduce the head developed by the compressor, such that there is a limit to the compressor's range if the pressure difference across the compressor increases beyond it capability. In some recycle valve installations that are intended to rapidly add flow to the suction line, the recycle line is closely coupled to the compressor discharge and suction, and there is no gas cooling in the recycle loop. In other recycle installations, the piping loop between the compressor discharge and the suction side is longer and includes cooling of the compressed gas before it is returned to the suction side of the compressor. Although recycle lines that include gas coolers are slower to respond as surge approaches, they can be used to control the flow rate for extended periods of times. Recycle valves are control valves that can be opened more or less, depending on the conditions required to avoiding surge. The overall effect of using a recycle valve is to increase flow seen by the compressor, reduce the head developed, and expend energy to compress gas that has already been compressed. Although the use of a recycle valve does prevent surge by maintaining a flow at or above the surge control line volume, it is inefficient and results in the use of large quantities of fuel to deliver reduced amounts of gas. Recycle operations significantly increase operating costs.

Although many centrifugal compressors could operate between 60% and 100% of their design flow if they operated down to the surge limit, they normally operated between 70% and 100% of their design capacity due to the surge margin established by the surge control line. All centrifugal compressors use their recycle valves at times, such as at start up or shutdown. The current surge avoidance and control methods result in recycle valves being used extensively and being opened well before the compressor is actually in danger of reaching surge. A survey of natural gas operating companies indicates that approximately 5% (in time) of all compressor operations make use of recycle flow. The loss in efficiency during recycle operations is at least 20% or more, which means that at least 1% of all fuel that is currently used to drive natural gas industry centrifugal compressor is used to recompress gas that has already been compressed and is, therefore, wasted. The purpose of the direct surge control project using measurements that are internal to the compressor is to reduce surge margins, use less recycle flow during operations, reduce wasted fuel, and reduce operating costs.

1.2 PREVIOUS TECHNOLOGY

The approach to implementing direct surge control in centrifugal compressors that is being developed in this study is based on previous research results and findings. At the start of the 1980's the understanding of surge was not complete and the simple external approach to keeping compressors away form the surge area of the performance map was the only available method. In the early 1980's, there were no known or reliable precursors to surge, although a number of signals and theories were tried. Starting about this time, the Gas Machinery Research Council (GMRC), formerly known as the Pulsation and Compressor Research Council (PCRC), funded research conducted at Southwest Research Institute to study the causes and processes involved in surge.

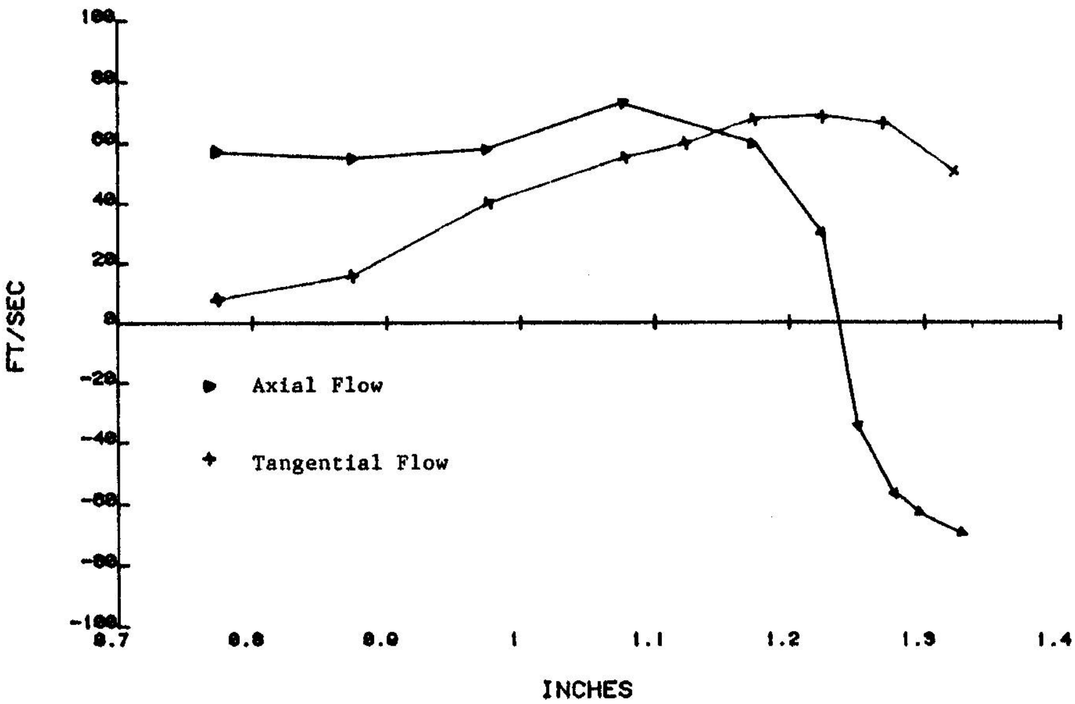
The early GMRC research eliminated many changes in compressor flows and conditions as indications of surge. Impeller exit and diffuser flows did not change in a unique manner as surge approached, although they were related to diffuser stall. The vibration and pulsation indications of rough operation did not always occur prior to surge and were caused by other mechanisms rather than by near surge conditions. One change that was found in a small compressor used for the first GMRC surge research, and later verified in a field compressor, was a change in the outer portion of the impeller inlet flow. The observed inlet flow pattern consists of a recirculation of flow on the outer edge of the impeller suction face, which develops as overall flow is reduced to the surge limit. The inlet area of the small laboratory compressor was instrumented for flow direction and magnitude and for temperature while system flow rate was slowly reduced until surge occurred. The most significant change observed as this class of compressor approached surge was a reduction in magnitude of velocity in the outer diameter area of the impeller inlet. This reduction in local flow was significantly greater than the reduction in average flow velocity. The outer ring of the inlet flow field also develops a tangential component in the direction of the impeller rotation as flow rate is reduced. As flow rate is reduced further and the surge limit is approached, flow in the outer part of the impeller inlet reverses and flows out form the impeller while the bulk flow is still positive towards the impeller. Figure 1-4 [PCRC Report No. TR 00-3[2]] is a plot of the magnitude of the inlet flow in the direction towards the impeller measured along the outer wall of the inlet passage. Figure 1-4 shows that the flow along the outer wall reduces rapidly as surge is approached, reaches zero, and begins to flow backwards while the bulk flow remains positive.



Near the Wall at Impeller Inlet

Figure 1-4. Change in Outer Wall Inlet Flow Signal as a Previous Small Laboratory Compressor Approaches and Reaches Surge.

The signal in Figure 1-4 would be useful as a surge avoidance and control signal if this change can be measured reliably in field compressors. Not only does the outer inlet flow area velocity reduce rapidly compared to the overall flow, but it reverses (crosses zero) and becomes significantly negative before surge occurs. The occurrence of surge is clearly shown in Figure 1-4 by the large swings in inlet flow from negative to a high positive and back again. Evidence that Figure 1-4 represents only the outer diameter portion of the inlet flow is shown in Figure 1-5 where the velocity profile across the radius of the impeller inlet is shown. Figure 1-5[2] shows the

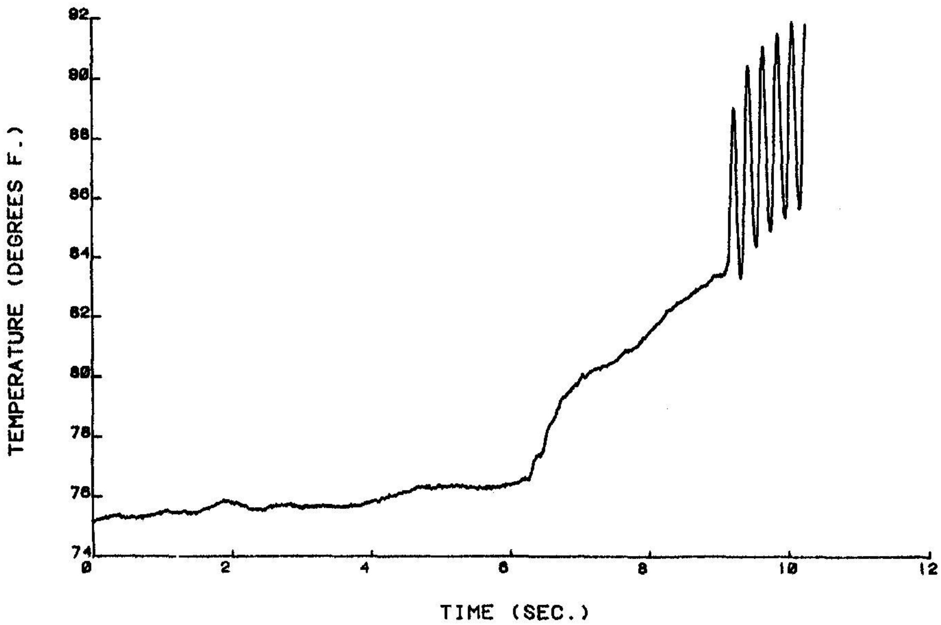


Axial & Tangential Velocity Along the Inlet Wall

Figure 1-5. Velocity Profile of Impeller Inlet Flow in the Axial and Tangential Direct Measured on a Small Laboratory Compressor Near Surge.

axial or towards the impeller flow at a steady near surge condition with positive velocity near the center three quarters of the radius, and a decreasing and then negative velocity in the outer one-quarter of the inlet radius. The inner or hub radius is 1.98 cm or 0.78 inches, and the outer radius of the inlet area is 3.4 cm or 1.34 inches as shown on the scale of Figure 1-5. The measured tangential component of the velocity is also shown in Figure 1-5 where the tangential component is seen to be small at the inner radius and to increase to a large value at the outer radius. One other measurement that helps to confirm the recirculating nature of the outer edge of the inlet flow is the temperature of the suction gas near the outer wall of the inlet passage as shown in Figure 1-6[2]. Figure 1-6 is a plot of the suction temperature near the outer edge of the inlet passage during the same period of time in which the flow rate was decreased towards surge as shown in Figure 1-4. The gas supply temperature was constant during this time, but the temperature of the gas near the outer wall increased a small amount at first and then markedly as the flow reversal condition developed. This increase in temperature is caused by the particle compression of the gas in the entrance area of the impeller, which then flows out from the impeller along the wall of the inlet. The development of this flow pattern at the face of an open inlet centrifugal compressor impeller, as shown by the early GMRC research, is an indication of approaching surge.

Since the GMRC research, similar findings of recirculating flow have been found and published by other researchers. The results of Kammer and Rautenberg's[3] work form 1985 is shown in Figure 1-7, where a strong inlet recirculation is shown and the increase in the tangential velocity component is confirmed. The work of Mizuki and Oosawa[4], published in 1985 and reproduced in Figure 1-8, shows that the recirculation zone increases in size as surge is



Near the Wall at Impeller Inlet

Figure 1-6. Gas Temperature Change Along the Outer Wall of a Small Laboratory Compressor as Flow Rate is Reduced Towards Surge.

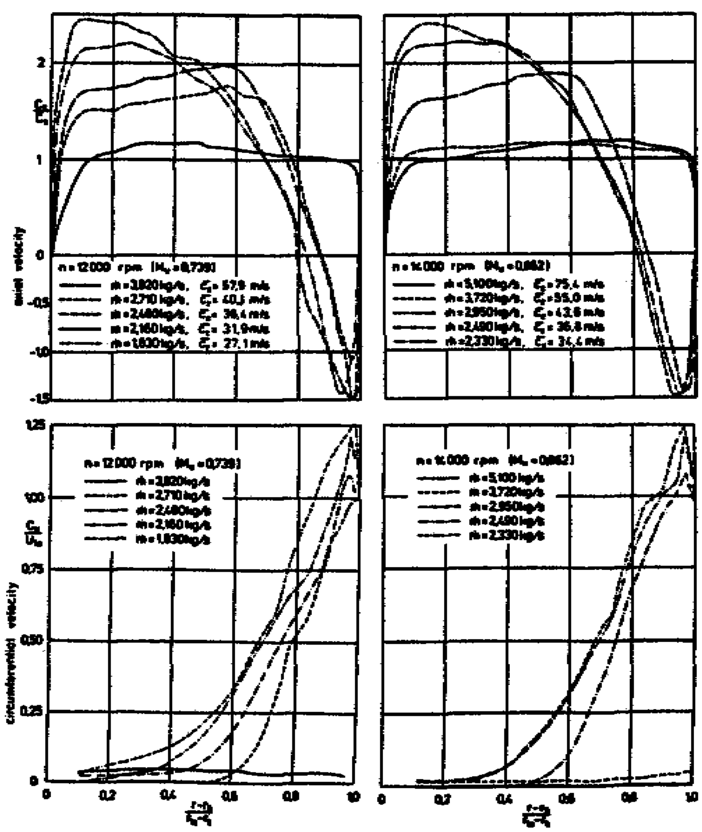


Figure 1-7. Inlet Axial and Tangential Velocity Profiles in a Centrifugal Compressor Approaching Surge as Published by Kammer and Rautenberg[3] in 1985.

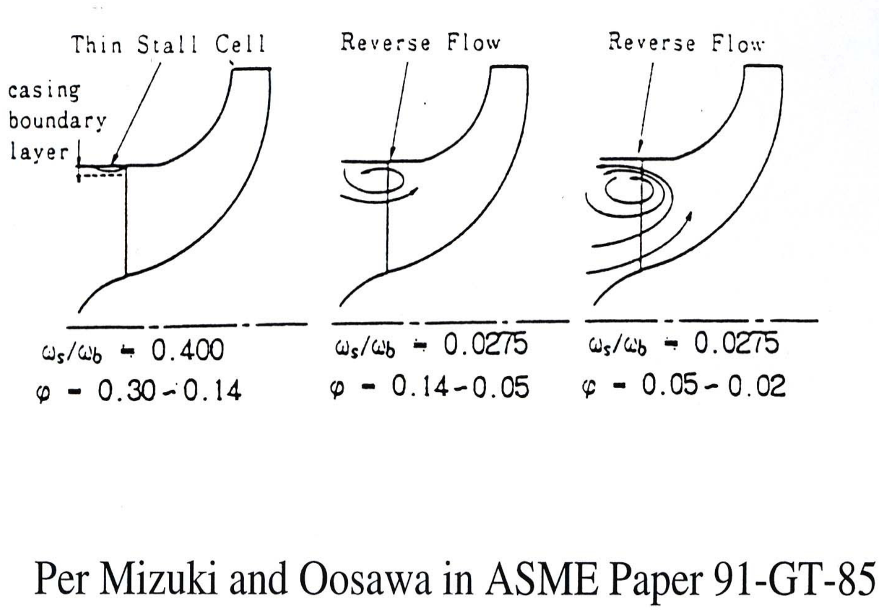


Figure 1-8. Centrifugal Compressor Impeller Inlet Flow Patterns as Flow is Reduced as Determined by Mizuki and Oosawa[4] in 1985.

approached. Other recent studies have referenced or alluded to similar flow patterns in both compressor and pump inlets, such that this behavior is now an accepted phenomenon.

As part of previous research sponsored by GMRC, an investigation of the types of sensors that would be best for measuring the flow reversal along the outer wall of an impeller inlet was conducted. In the laboratory, sensitive devices, such as hot film anemometers, have been used; however, these instruments are not sufficiently rugged for long-term use in a field compressor. A surge sensor to be permanently located immediately in front of a centrifugal impeller needs to be rugged and yet sufficiently sensitive to detect small changes in the local flows. The probe must also be compatible with the compressor environment, manufacturability, and reasonably easy to install. Without incorporating new technologies that are unproven, the type of sensor chosen for the surge flow pattern detection is a drag probe. A drag probe consists of a bluff body located in the flow field with strain gauges mounted on a beam or flexible element, which supports the drag body and senses the flow induced forces on the probe.

1.3 PROJECT ORGANIZATION

The overall approach for direct surge control is to develop a means to reliably measure the changes in internal flows that indicate approaching surge and then develop a controller to use the nearness to surge signal to avoid surge with less margin, less wasted energy, and more operational flexibility. To accomplish this in an orderly fashion, the project has been divided into three phases to verify the near surge sensing, to develop a prototype surge controller, and to demonstrate the benefits of direct surge control. The active involvement of the natural gas industry in project oversight, development of specifications, test site support, and definition of the required product or result is central to this project.

Phase 1 of the project consists of Task 1, to specify, model, and design the surge probes, and Task 2, to characterize the sensors and test their response to near surge compressor operation. The specification, modeling, and design activities in Task 1 include developing specification with the natural gas industry input, CFD modeling of impeller inlet flows at near surge conditions, and consideration of dynamic responses of the compressor systems to define sensor requirements. As the final part of Task 1, the probes are designed to satisfy the specifications, the expected extremes of flow conditions, and the dynamic responses of flow and the compressor system. In Task 2, which is part of Phase 1, the probes are characterized by bench testing and calibration, by testing in a laboratory flow loop and by testing in a laboratory compressor and a field compressor. The difficulties experienced, the results to date, and the lessons learned during Phase 1 are described in this preliminary report.

Phase 2, which includes Task 3 and 4, is planned as a one-year effort and is intended to develop the control algorithm and prototype controller and to field test direct surge control. Task 3 includes design of the control algorithm, involvement of the industry partners to identify target compressors, and specified requirements, implementation, and refinement of the control algorithm in the prototype controller through laboratory testing. Task 4 includes installing the prototype surge control system, including probes and controller in a field compressor and testing the system over a range of conditions, and a period of time to evaluate the surge control system and to quantify the benefits and reduced operating costs achieved. Following field testing of the surge controller, the algorithm and controller will be refined as necessary based on test results.

Phase 3, which consists of Task 5, is a demonstration of the direct surge control system. Task 5 includes preparing the control system for unattended operations with the industry's input, installing and checking out the system for extended operations with data logging, conducting a six-month demonstration test, and evaluating the results as to the performance of the direct surge controller and the benefits obtained. The overall project also includes a reporting task.

2. EXECUTIVE SUMMARY

As part of the DOE's Natural Gas Infrastructure program, this "Increased Flexibility of Turbo-Compressors in Natural Gas Transmission through Direct Surge Control" project is being conducted to verify near surge sensing, develop a prototype surge control system, and test and demonstrate the benefits of direct surge control. This project is co-funded by the Gas Machinery Research Council and Siemens Energy and Automation and has the support and direct participation of Duke and El Paso Energy companies and Solar Turbine as well as the interest of other GMRC member companies. Centrifugal compressors are a major and growing part of the natural gas transmission and related industries and their safe and flexible operating has a significant financial impact on industry's costs. Surge is an instability that limits the operation of compressors and causes the flow through a compressor to collapse. Surge is a damaging event and must be avoided. However, the current surge avoidance methods are based on external measurements, require large surge margins, result in recycling flow from discharge to suction, and thereby reduce efficiency and significantly increase costs. The goal of this project is to refine a surge avoidance method based on fundamental internal measurements that allows a compressor to operate over a wide range, closer to surge, with less recycled flow, and with less wasted fuel. The intent is to develop a direct surge control approach as a commercially practical product that the industry can use.

Previous research sponsored by GMRC identified an impeller inlet flow recirculation that develops as a compressor approaches surge and patented an approach to detect this flow change as a way to avoid surge. In the previous testing, an inlet flow reversal was measured on the outer wall of the inlet of a simple laboratory centrifugal compressor and has been found in numerous other compressors under detailed testing as demonstrated by various publications and recent findings. Specification for surge detection probes that will meet the industries' needs for a surge controller have been developed with industry guidance. CFD modeling conducted to aid in interpreting the laboratory test results show a small off-center recirculation zone in the SwRI laboratory compressor. An analysis of the system dynamics has identified data sampling and handling requirements for a direct surge control system. A detailed design process for surge detection probes that considers the compressor size and flow rates and conditions, ensures sensitivity, avoids overloads or dynamic failures, considers the signal wires, and provides a practical surge detection probe has been developed and used in this effort.

Laboratory test results in the SwRI centrifugal compressor do show a change in the impeller inlet flow prior to surge, but it is not the axial flow reversal that was expected. Because of the inlet geometry of the SwRI laboratory compressor, the recirculation is small, off-center, and too close to the impeller to be detected. A mostly tangential rather than axial change in the local flow pattern occurs at the surge probes in the SwRI compressors. Additional laboratory tests in another compressor will be conducted as part of Phase II of this project.

Preparations for a field test had been completed at one point in this project. However, a failure of the surge probe wiring just inside the compressor case caused a delay in the field testing. Repairs of the wiring in the compressor have been scheduled and the field test will take place shortly after the repairs.

3. EXPERIMENTAL

3.1 SPECIFICATIONS

In order to focus the direct surge development effort, the first step is to identify specification for a direct surge control system that will benefit the natural gas industry. The overall requirements for the sensor specified in this document are that it is simple, rugged, sensitive, cost-effective, easy to manufacture, installable in natural gas centrifugal compressors, and able to detect changes in flow conditions that indicate the closeness of surge in a useful manner. The general requirements for the specified surge controller are to receive the closeness to surge signal and to incorporate algorithms and flexible processing to produce outputs that prevent a compressor from reaching surge while maintaining an economical surge margin. Draft specifications were prepared and presented to the Oversight Committee for review. A meeting was held to discuss the specifications, and written comments were received form the Oversight Committee and industry representatives. The full specifications, as reviewed and approved by the industry Oversight Committee, are shown in the following paragraphs.

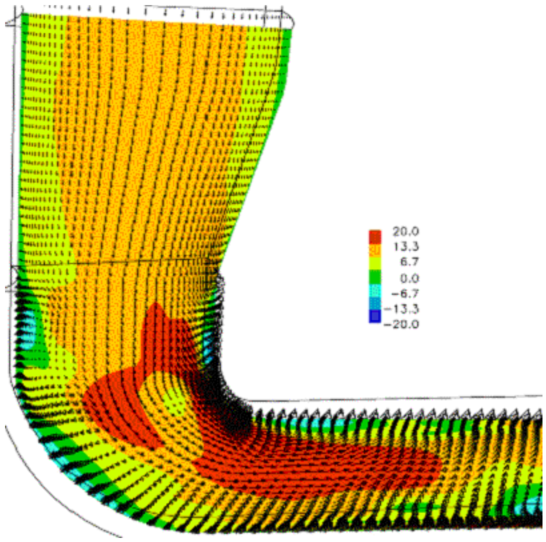
1. The sensor shall have sufficient sensitivity and rangeability to detect changes in flow direction, magnitude, and condition that indicate the nearness of compressor surge. The sensor shall respond to flows throughout the compressor's full range of operations and shall accurately and effectively measure low, near zero, and reverse local flows. The sensor shall measure local flow velocities from less than 3 feet per second to an upper near surge related limit. The sensor shall perform accurately and without drift at low, near zero, and reverse flow conditions over a long life (at least 10 years). The sensor shall be insensitive to thermal drift.
2. The sensor shall be rugged enough to withstand the maximum startup, stonewall, and overspeed flows in the compressor without damage or adverse effect on its sensitivity or stability. The sensor shall be compatible with the compressor environment and be robust. An element of redundancy shall be incorporated into the sensor system to improve its reliability. The sensor shall be such that its failure, mechanically or electrically, will not endanger the compressor in any way. The sensor shall be reliable.
3. The sensor shall not suffer from flow-induced vibrations under the full potential range of conditions. That is, the sensor's mechanical natural frequencies shall be well removed from any vortex shedding frequencies, and, in addition, no severe flow induced phenomena shall occur within the normal range of compressor operations.
4. The sensor shall be easy to manufacture, reasonable to install, simple in function, and cost-effective; that is, its cost will be small compared to the value that it provides.
5. The control signal from the sensor shall be reliable, directly related to the nearness of surge, and usable in a control algorithm to avoid surge and control the compressor with a small surge margin. The connection and electrical signal from the sensor, within the compressor, to the external control system will be reliable, durable, and safe for the natural gas compressor environment.

6. The exterior control system shall receive the surge control signal and shall incorporate algorithms and routines to control the compressor to avoid surge, minimize wasted fuel, and increase overall compressor flexibility.
7. The data acquisition and signal processing shall preserve the accuracy and address the nature of the signal (A-to-D precision, processing rate and speed of response, signal averaging, and a way to handle the non-linearity between flow and drag force) in such a way as to provide reliable and stable control.
8. The external control system shall consist of a Class I, Division 2, Group D system with a wall mounted NEMA 12 control panel. The PLC controller will be complete with a power supply, 16 digital, 8 analog, and 8 RTD inputs, 8 relay and 4 analog outputs, and MODBUS communication.
9. The surge control routines shall adapt existing logic and control procedures and shall use newly developed algorithms to control the compressor's proximity to surge.
10. The surge control system shall provide for surge control and shall interface with the compressor unit controller to allow for compressor starts, stops, and other operating modes and will provide, where necessary, speed change limits and alarm signals. The surge control system will maintain a safe (and adjustable) surge margin and will interface with the unit controller to accommodate unsupervised (no operator intervention) safe and reliable operation.
11. The surge control system shall provide a color graphical user interface, compressor map display, performance monitoring software, date storage capability, and high speed and remote communication capabilities.
12. The surge control algorithm and system shall be tested, evaluated, debugged extensively, and documented with drawings and instruction manuals.

3.2 MODELING

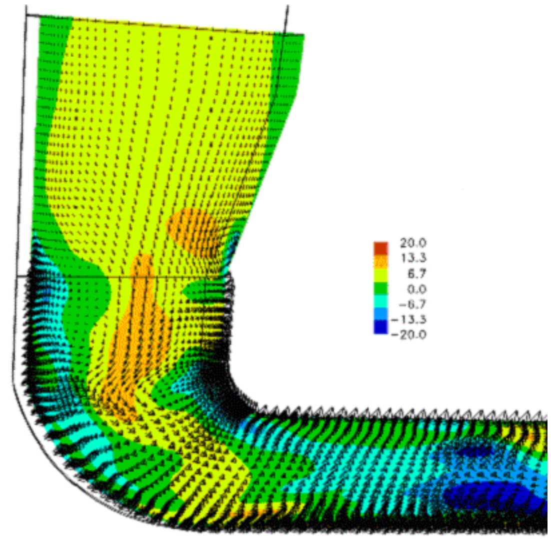
The second part of the work to develop a surge detection sensor is to model the impeller inlet flow and to consider the behavior of a sensing probe in that flow. A CFD model of the inlet flow channels, the approach to the compressor wheel, and the inlet side of the rotating impeller of the SwRI laboratory compressor was developed. The model included the effects of the rotating impeller surfaces and used periodic boundary conditions to model the effects of the multiple channels in which the SwRI laboratory compressor's flow is confined. The model's boundary conditions include a fixed flow rate at the upstream entrance and a fixed pressure at the simulated impeller exit. By reducing the model's flow rate and increasing the outlet pressure, the change from normal to near surge conditions for the compressor were simulated. One other generalized compressor configuration has been modeled, but no other specific compressor configuration has been modeled to date because the value of the results appear to be general rather than specific.

The key results of this modeling are displayed in Figures 3-1 and 3-2. Figure 3-1 compares the velocity vector field and contours of axial velocity (contravariant U velocity in m/s) along the channel centerline (k=20) for flow rates of 679 m^3/hr, or 400 ACFM in



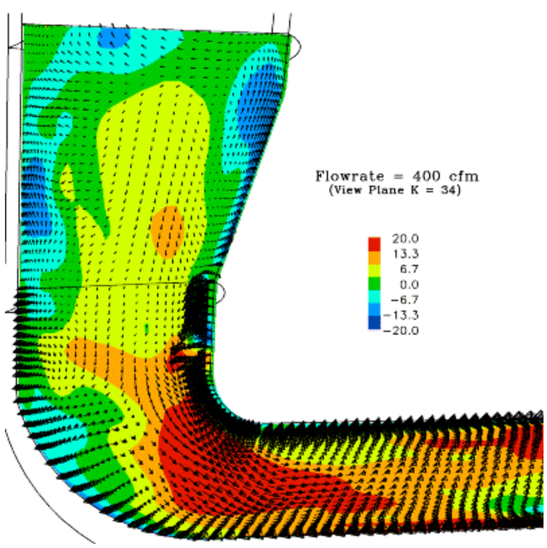
Q = 400 ACFM

Figure 3-1a. CFD Model of Inlet Centerline Velocity at Normal Flow Rate.



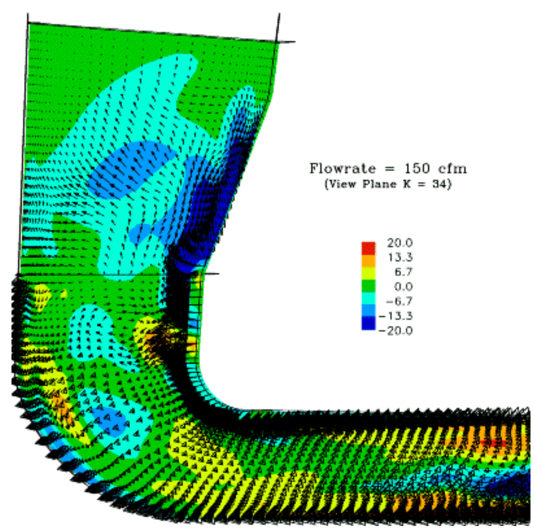
Q =150 ACFM

Figure 3-1b. CFD Model of Inlet Centerline Velocity at Near Surge Flow Conditions.



Q = 400 ACFM

Figure 3-2a. CFD Model of Inlet Sidewall Velocity at Normal Flow Rate.



Q =150 ACFM

Figure 3-2b. CFD Model of Inlet Sidewall Velocity at Near Surge Flow Conditions.

Figure 31a, and for the near surge flow rate of 255 m^3/hr, or 150 ACFM in Figure 3-1b. The rotational speed for this simulation is 8,000 rpm. At the higher flow rates, the velocity contours are uniform and considered normal with a lower velocity area near the inner wall and an accelerated area in the entrance to the impeller. At the near surge lower flow rate in Figure 3-1b, there is a recirculation region; however, it is small (of the boundary layer size) and is close to the impeller. Because of the inlet channel and accelerated flow at the impeller entrance, the recirculation in this compressor is small and occurs closer to the impeller than the surge detection probes are able to be located.

Impeller rotation is clockwise, with the direction of rotation in the positive direction when moving from grid plane k=6 to k=34. Figure 3-2 compares velocity vector fields and contours of axial velocity on the rotational direction side (k=34) of the inlet channel for the normal flow rate of 679 m^3/hr, or 400 ACFM in Figure 3-2a, and for the near surge flow rate of 255 m^3/hr, or 150 ACFM in Figure 3-2b. There is considerable more recirculation at the lower flow rate along the side of the inlet channel that is the direction of rotation than in the center of the channel as shown in Figure 3-2b. There is also some recirculation and other secondary flow along the sidewall at the higher flow rate. These figures and other model results confirm that a strong tangential velocity is present at near surge conditions as seen in previous research. However, the recirculating flow pattern in this compressor is difficult to detect due the geometry.

In general, the flow field structure in this turbo-machine results from a competition between the axial flow through the duct and the rotating shear flow in the shrouded inlet to the impeller. The axial flow moves essentially straight down the duct, but at the inlet to the shrouded impeller this axial flow must penetrate into a circumferential flow stream created by rotation of the impeller and the other duct flow streams. As a result, on the downstream side of the duct (in the corner regions and the downstream wall of the duct) some of this axial flow is entrained by the circumferential flow and then impinges on the downstream (direction of rotation) wall of the duct. This flow process is displayed in Figures 3-1 and 3-2. As the flow rate decreases, the axial flow has less kinetic energy for penetrating into the circumferential flow, and a larger volume of fluid is entrained and impinges on the downstream wall of the duct. This results in the development of recirculation cells near the duct exit.

The CFD results show that for the flow rate of 255 m^3/hr, or 150 ACFM, the flow in the downstream wall region of the duct has a flow structure characterized by significant regions of flow separation, which extend up through the channel transition section. As flow continues to decrease toward surge, the size and intensity of these recirculation regions will continue to grow.

These simulations assumed an initial turbulent intensity of 10% of the inlet flow kinetic energy and an initial turbulent viscosity of 100 times greater than the dynamic viscosity of the working fluid (air, μ = 2.0e-05 N s/m^2). These values may have an impact on simulation results (although simulations with variation of these values were performed and resulted in only minor differences). In addition, all other boundary condition specifications, such as duct inlet velocity and static pressure on the outlet side of the impeller are based on compressor curves.

The other part of the modeling effort was the analysis and consideration of dynamic responses and behavior of the compressor system and the detection probes as surge is approached. There are a number of high frequency disturbances that will be present during operation of a centrifugal compressor and may appear in the surge detection signal. Among these higher frequencies are the blade passing and acoustic noise frequencies from the compressor and the natural mechanical frequency of the surge detection probes that will intentionally be placed above most excitation frequencies. As a centrifugal compressor approaches surge, other frequencies that may appear are the ones noted by operators as rough operating conditions that sometimes develop before surge. These lower frequencies include indications of stall that may (or may not) develop as surge is approached, control system and valve variations during restricted operations, or fluid fluctuations or instabilities at low flow conditions. The variations and changes that occur just prior to surge are also low frequencies set by piping system dynamics and compressor curve interactions. Therefore, the surge detection system should be able to detect and follow low frequency changes and ignore or not be affected by high frequency signals or noise that has nothing to do with surge. The frequencies of interest for a surge detection system are subsynchronous to the compressor operating speed. The result of these considerations is that the surge probe and detection system should sample and process information sufficiently rapidly to follow changes slower than the compressor operating speed but should filter out and ignore frequencies higher than the compressor fundamental frequency. A data sampling rate of 200 to 300 samples per second, depending on the compressor speed, is suggested with the probe signal low pass filter to eliminate frequencies above this range. Given these general considerations, when the control system does detect oscillations in the signal, the sensitive and reaction times of the controller to approaching surge should be at its maximum.

3.3 DESIGN PROCESS

The objective of designing and installing a surge detection probe for any centrifugal compressor is to be able to measure the change (reversal) of the inlet flow pattern near the outer wall of the impeller inlet flow channel that indicates the approach of surge. The intent is to use this changing signal to control the compressor to be able to operate close to surge while safely avoiding surge. The type of probe used in this design approach is a drag probe, which includes a drag body placed in the flow stream and a supporting beam with strain gauges to measure the forces imposed. The strain signal must be led out of compressor to the control system. The steps and some guidelines for the design are as follows:

1. Calculate the gas density and flow velocity range in the inlet near the impeller inlet and determine the potential drag forces on a probe from maximum excess flow to surge or extreme low flow conditions.
2. Size the drag body to be not more than 25% of the width of the channel and at least 10% of the channel width outside of the boundary layer (20% is ideal). The drag body should not block more than 2% of the inlet flow area.
3. Size the bending beam cross-section and length to be able to safely and reliably support the drag probe against the maximum force, including a potential particle impact, and yet provide adequate signal for low flow or flow reversal detection near surge. The probe must be sensitive enough for low surge level flows and yet rugged enough for the maximum flows.

4. Check that the vortex shedding frequency (based on Strouhal number) will not correspond with the mechanical natural frequency or the running and blade passing frequencies.
5. Design the probe and a probe holder to penetrate the outer wall of the impeller inlet channel, such that the drag body is a short distance (from 0.5 to 1.5 channel heights) in front of the impeller. The probe holder should provide a fulcrum for the bending beam and securely retain the probe in the correct position in the flow field at the impeller inlet
6. Plan for the strain gauge installation to be such that the bending signal in doubled, tension on one side added to compression on the other, and that the temperature effects on the strain in the element is balanced. Arrange strain gauges to monitor the axial and tangential flow forces and mark and control the probe orientation to know which signal indicates which direction.
7. Arrange for the wires and the signal from the probe to exit the compressor pressure case through a safe, leak proof, and durable fitting or connection that does not violate or invalidate the hydro test and pressure rating of the compressor case.
8. Provide the basic input for the strain gauge amplifier circuits to process the signal, which includes a full bridge strain gauge amplifier with sufficient sensitivity to present the surge flow signal as a reliable output.
9. The sampling frequency for probe data should be approximately two to three times the running speed of the compressor as this will track the subsynchronous frequency that may develop from stall that occurs before surge. The processing will need to identify and control the compressor to limit the amplitude of this subsynchronous flow variation. The sampling rate will be less than and the output will be unaffected by blade passing frequencies and mechanical vibrations of the surge detection probe.

When the drag probes have been designed, they are fabricated by machining the drag element and its supporting bending beam out of a single piece of stainless steel in this case; although, other compatible materials could be used in the future. The bending beam is generally square in shape to accommodate the strain gauges and to purely separate the axial from the tangential directions for sensing the flow induced forces. Photographs of two of the first surge detection probes fabricated for laboratory testing are shown in Figure 3-3. The probes in Figure 3-3 included one longer and one slightly shorter drag probe for use in the SwRI laboratory compressor. Note the small holes that are drilled above the bending beam for the strain gauge wires to be routed out of the supporting tube. After the strain gauges are mounted on all four sides of the bending beam and the wires are installed thought the exit holes, the strain gauges and wires are coated with a protective coating. A tube fitting for retaining the surge probe and threading into the outer wall of the inlet channel is attached to the drag probe and a photograph of the completed sensor with its wires inside a flexible tube for protection is shown in Figure 3-4.

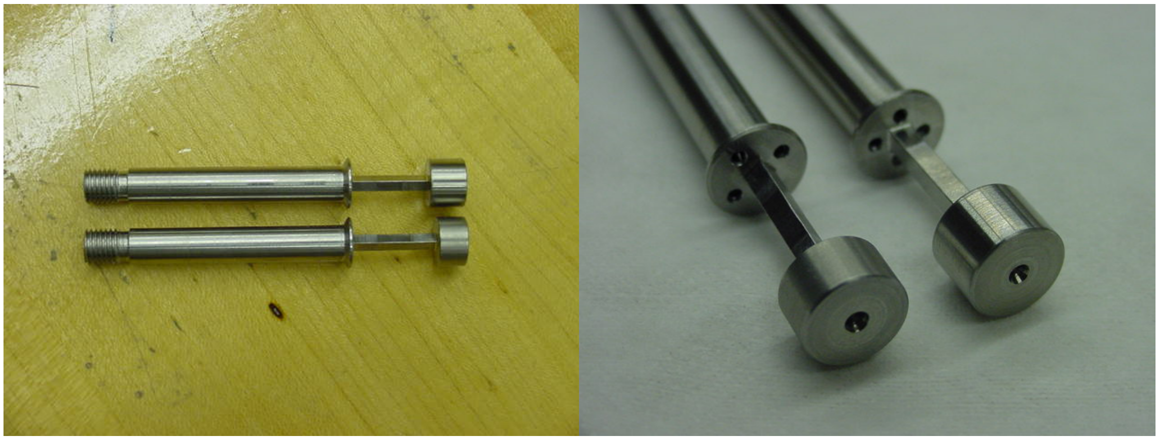


Figure 3-3. Photograph of the First Surge Detection Drag Probe Fabricated for a Laboratory Centrifugal Compressor.

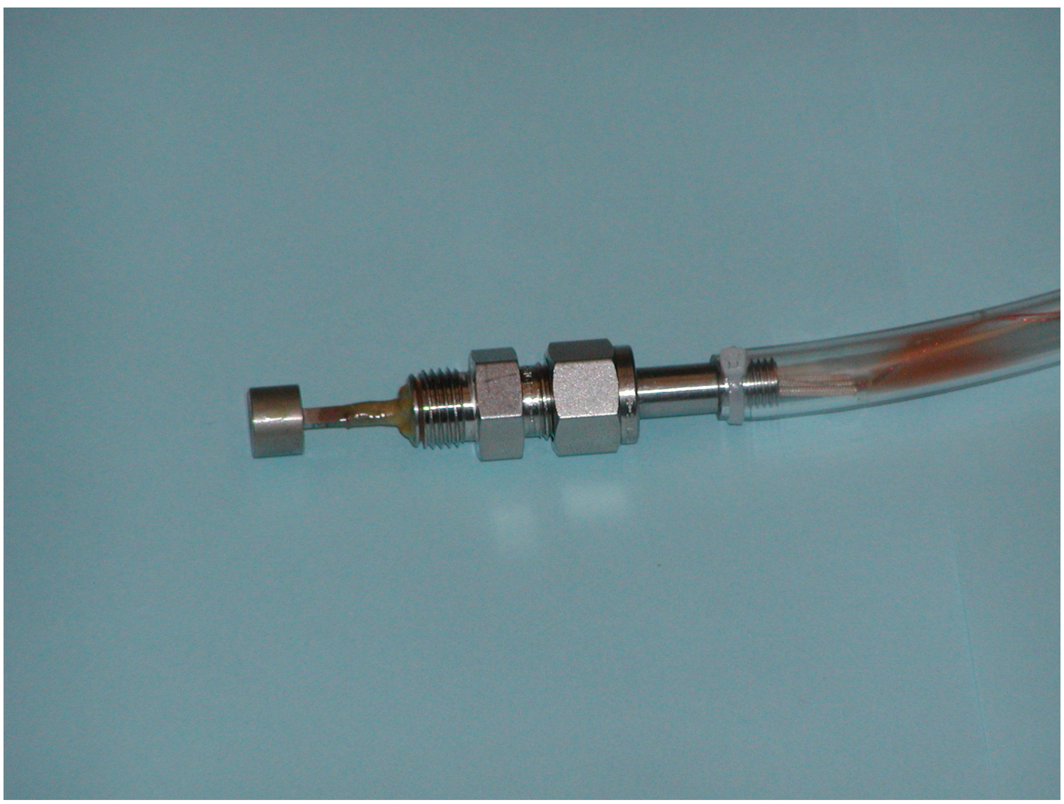


Figure 3-4. Photograph of a Completed Surge Detection Probe Ready for Installation in the Laboratory Compressor.

4. RESULTS AND DISCUSSION

4.1 BENCH AND FLOW TEST RESULTS

The first test to characterize the response of surge detection probes is a static bench test in which selected masses (weights) are hung on the probe and the magnitude of the strain is recorded to confirm the probe's sensitivity to force. There are four strain gauges on each probe, and they are wired in pairs so that there are two directions sensitive to forces, the axial and the tangential directions. The force and response in each direction can be positive or negative. Therefore, the probes being bench tested were turned with each of the four sides up and weights were hung so that gravity provided a known force in each direction. In the case of the small probes to be used for the laboratory compressor, the masses used were from 100 to 500 grams (approx 1.09 pounds force maximum). The strain level magnitude (absolute value) was recorded for each direction as a function of the mass applied. The results for the slightly longer probe are shown in Figure 4-1 where it can be seen that the four directions are nearly identical in sensitivity and that the results agree with ideal calculations of expected strain. The results for the shorter drag probe are shown in Figure 4-2 where the agreement with ideal calculations is good and one of the strain gauges has lower sensitivity than the others by approximately 3.6%. The sensitivity of one gauge being low is most like a result of an installation effect. Since the detection of surge is based on a change from the normal flow condition and hopefully a reversal of flow, a small difference in the magnitude of the signal will not prevent detection of surge.

A bump test was conducted as an additional part of the bench testing to determine the mechanical natural frequency of the surge detection probes. The result of the bump test for the shorter laboratory surge detection probe is shown in Figure 4-3 where the mechanical natural frequency of 932 Hz is clearly indicated. The mechanical natural frequency of the longer laboratory drag probes was 903 Hz, which is well above the operating speed of the compressor.

To confirm the response of the probes to flow velocity each of the laboratory surge detection probes was installed in a straight test section pipe and exposed to known flow loop velocities. The flow facility pressure was held constant at various pressures from 15 to 100 psia such that the flow-induced forces were similar to compressor inlet flow forces. The strain output in one direction, axial or tangential, was recorded as a function of flow rate and plotted as shown in Figure 4-4. Figure 4-4 shows the strain at different flow rates for three repeated tests and indicates a consistent output as a function of flow. Strain is linearly proportional to force on the drag probe, which is dependent on density times velocity squared and increases parabolically as shown in Figure 4-4. Except for the uncertain increase in strain at very low flows, the response of the long laboratory probe in the 2-4 strain gauge direction is as expected.

The long drag probe was turned 90 degrees in the flow section piping so that the other direction of sensitivity could be tested and Figure 4-5 shows the results. During the test of the long probe in the 1-3 strain gauge direction, two of the tests were conducted at a gas temperature of approximately 84°F, and a third test was conducted after the flow loop heated up to approximately 130°F. In Figure 4-5, it appears that the strain for the third test is lower than it should be; however, the density of the gas is lower at the higher temperature, and when the drag

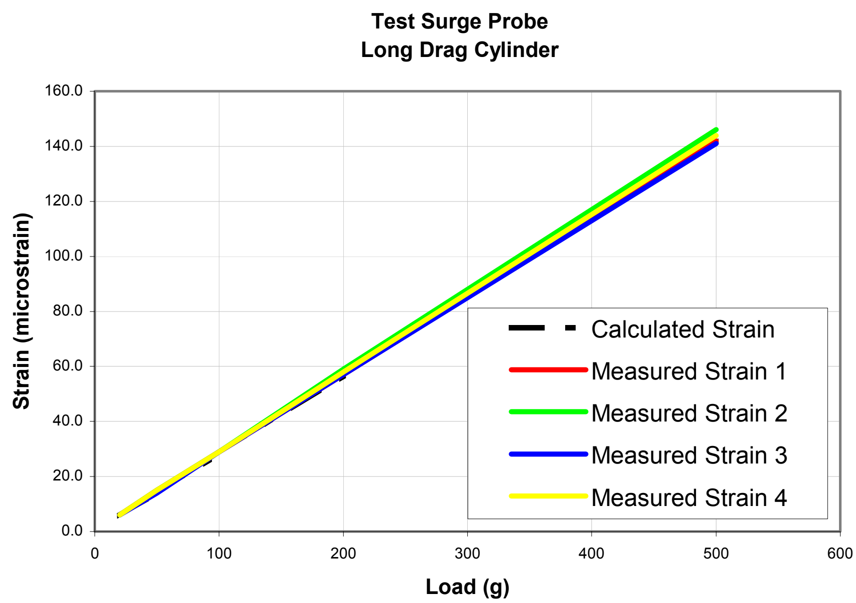


Figure 4-1. Bench Test Results Showing Strain and a Function of Load for All 4 Drag Probe Sensitivity Directions for the Long Probe.

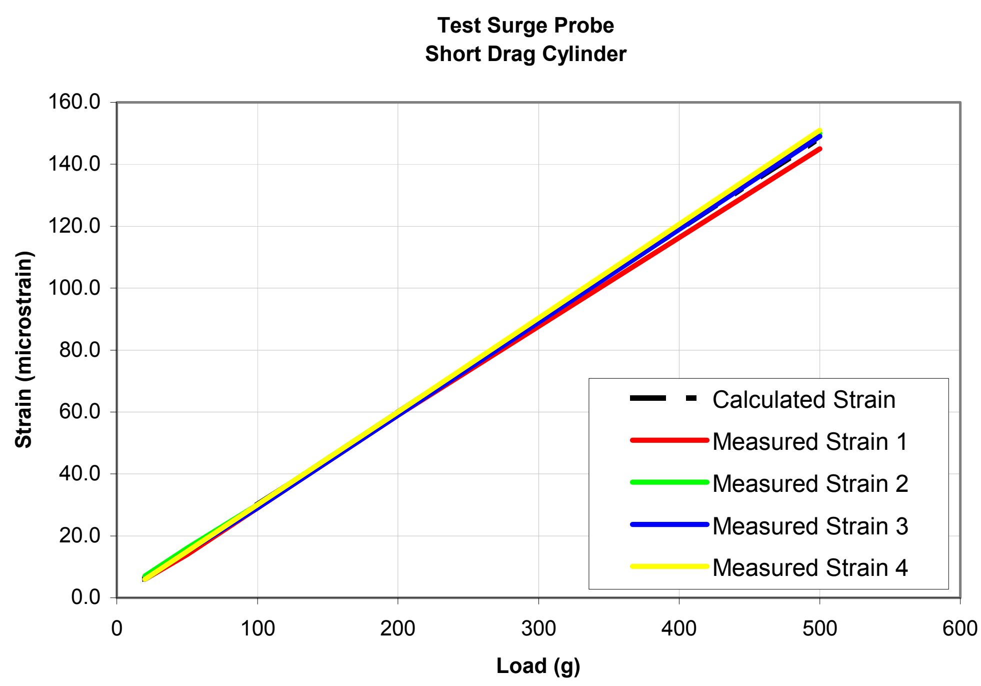


Figure 4-2. Bench Test Results Showing Strain and a Function of Load for All 4 Drag Probe Sensitivity Directions for the Short Probe.

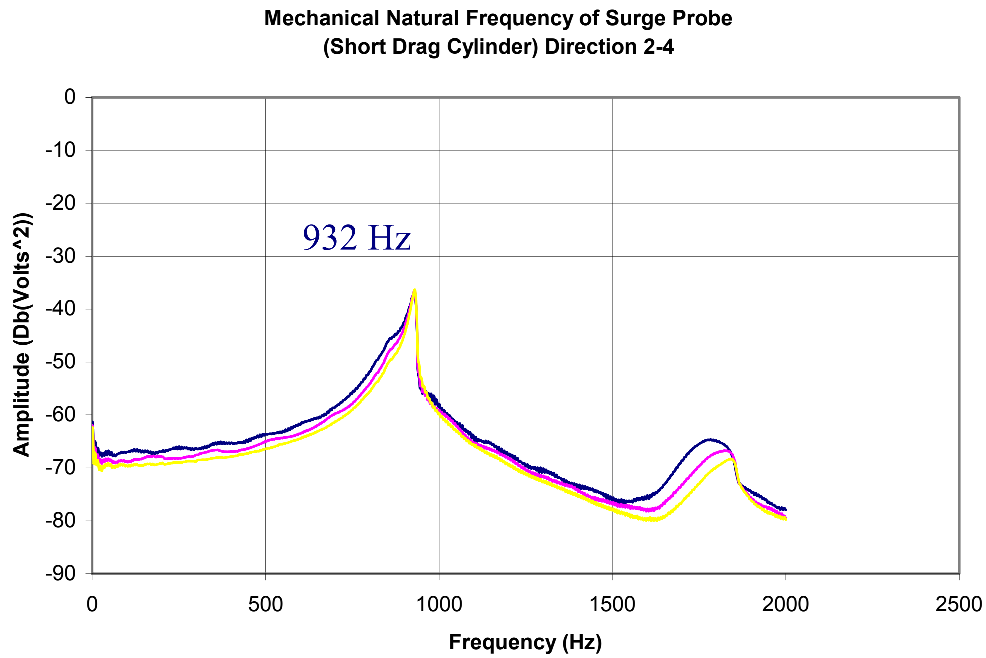


Figure 4-3. Results of Bump Test of Short Laboratory Drag Probe Showing the Mechanical Natural Frequency of the Probe.

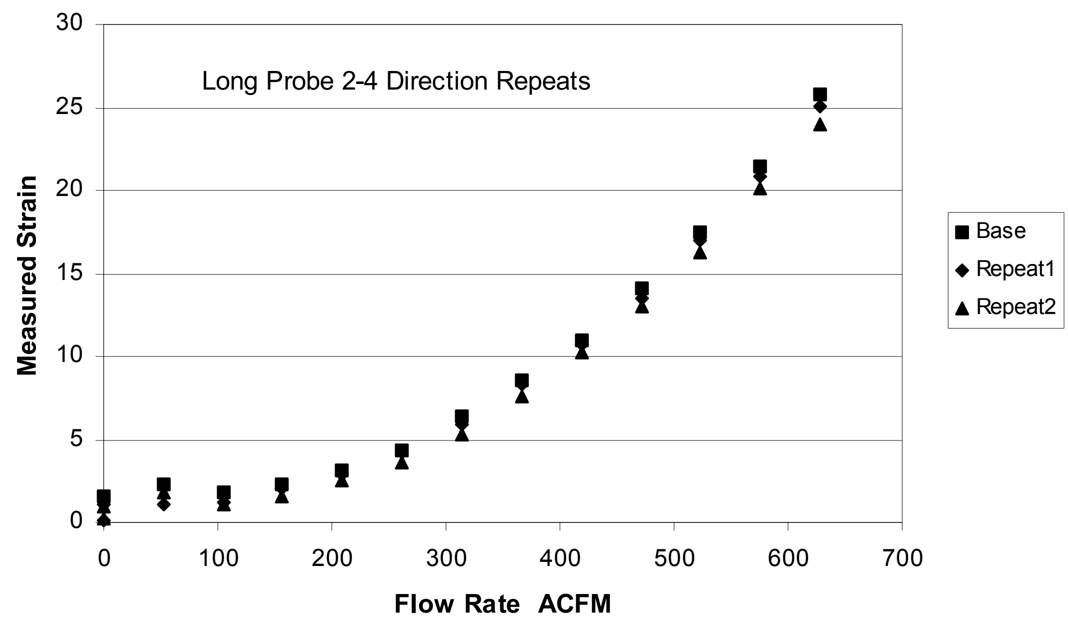


Figure 4-4. Strain As a Function of Flow Rate During Repeated Flow Facility Tests of the Long Laboratory Drag Probe in the 2-4 Direction.

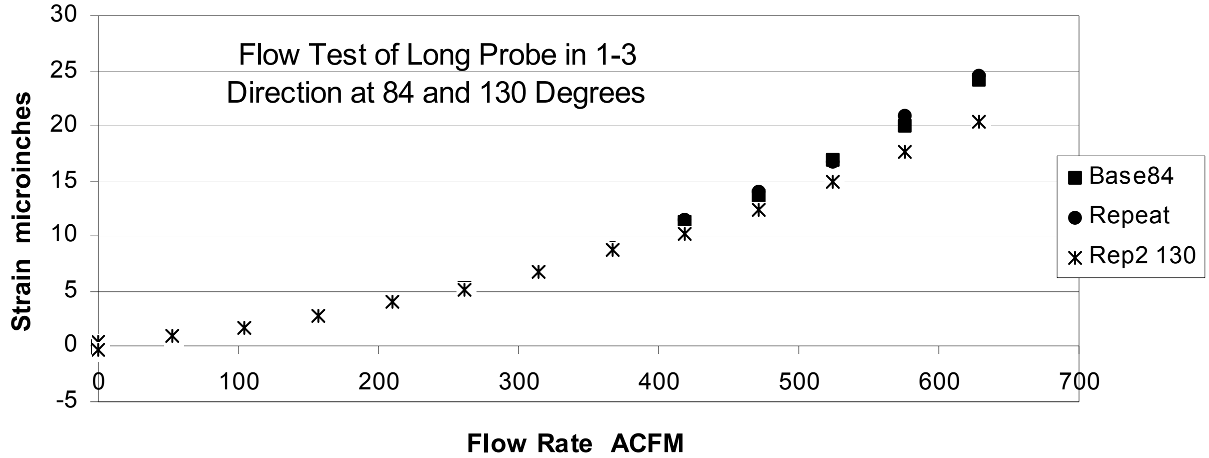


Figure 4-5. Strain As a Function of Flow Rate During Repeated Flow Facility Tests of the Long Laboratory Drag Probe in the 1-3 Direction.

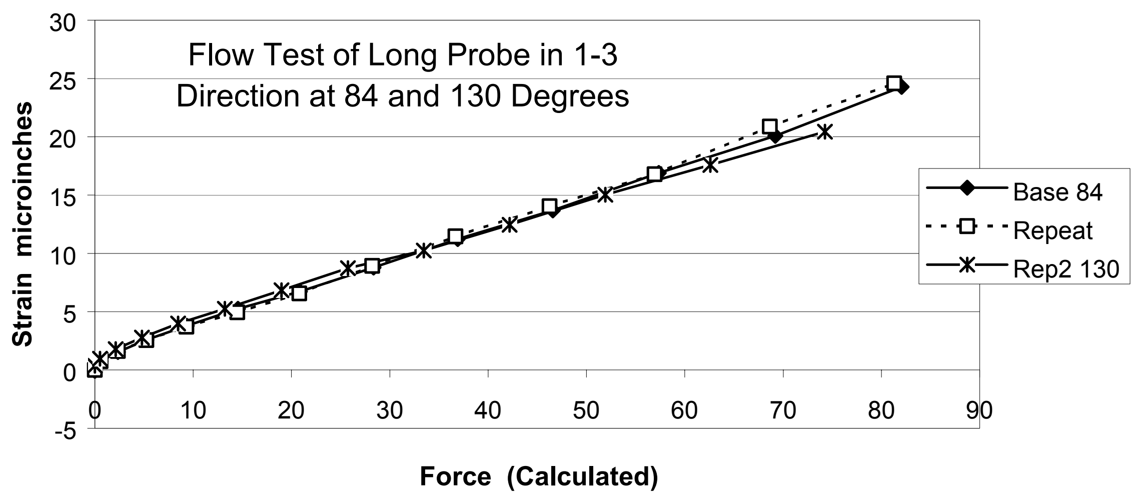


Figure 4-6. Strain As a Function of Calculated Drag Force During Repeated Flow Tests of the Long Laboratory Drag Probe in the 1-3 Direction.

force is calculated and the strain is plotted as a function of force as shown in Figure 4-6, the agreement between the test runs is within 10% or better for all of the test points.

A test of the short laboratory probe in the 2-4 strain gauge direction shows an important feature of the drag probes, and that is that they are equally sensitive in the reverse flow direction as in the forward flow direction. Figure 4-7 is a plot of the strain from the short drag probe with flow in the direction of the 2-4 strain gauge axis as a function of flow rate. The strain is negative indicating that the flow is in the reverse direction as far as the probe orientation is concerned. However, the magnitude of the strain as a function of flow is essentially the same as for the other probe and directions. When corrected for the applied flow force, the strains from the two tests shown in Figure 4-7 are essentially identical as a function of force as shown in Figure 4-8. The resulting strain sensitivity per unit flow force constitutes a calibration of the probes.

4.2 LABORATORY COMPRESSOR TEST RESULTS

Once the experimental surge detection probes are bench and flow tested, they are ready for installation in a laboratory compressor. The compressor was disassembled and the inlet flow channel was removed from the compressor case for drilling and taping. The inlet channel in the top half of the compressor case is shown in Figure 4-9 where the shaft way is clear in the center of the picture and the space of the impeller is at the top of the picture. The annular inlet flow area to the impeller is divided into 14 channels by walls, which are essentially lengthy guide vanes in the inlet annuals. The probes were mounted to be in the center of selected passages as close to the impeller as practical. The lower half of the inlet passage section showing some of the dividers and channels with two surge detection probes mounted in place are shown in Figure 4-10. A close-up of one of the probes in an inlet channel is shown in Figure 4-11. It is the metal of the impeller inlet seal and shroud division wall that prevents the surge detection probe form being installed closer to the impeller. Two pictures of the lower half of the compressor case with rotor, including the shaft and impeller and one of the two surge detection drag probes, are shown in Figure 4-12. The location of the probe, the restriction of the stage division wall, and the routing of the wires to the compressor case exit are shown in Figure 4-13.

The laboratory compressor was then assembled and other instrumentation, including suction and discharge pressure and temperature, and several local pressure measurements were added to the compressor suction and discharge passages. A measured compressor curve for the SwRI centrifugal compressor operating at a low suction pressure of 28 psia is shown in Figure 4-14. Most of the early testing shown in the following results was performed at the lowest continuous operating speed of 8,000 rpm. The results of several exploratory tests are discussed in the following paragraphs.

The general trend in surge detection probe response is that as flow rate increases the axial strain from both the short and long probes increases. The starting strain is normally set to zero and some offset, positive or negative, takes place as the compressor comes up to operating conditions. The change in strain is referenced to this initial offset. An example of the change in axial probe strain as flow rate decreases and then increases is shown in Figure 4-15. Although the compressor was close to surge at the lowest flow rate in Figure 4-15, surge did not occur

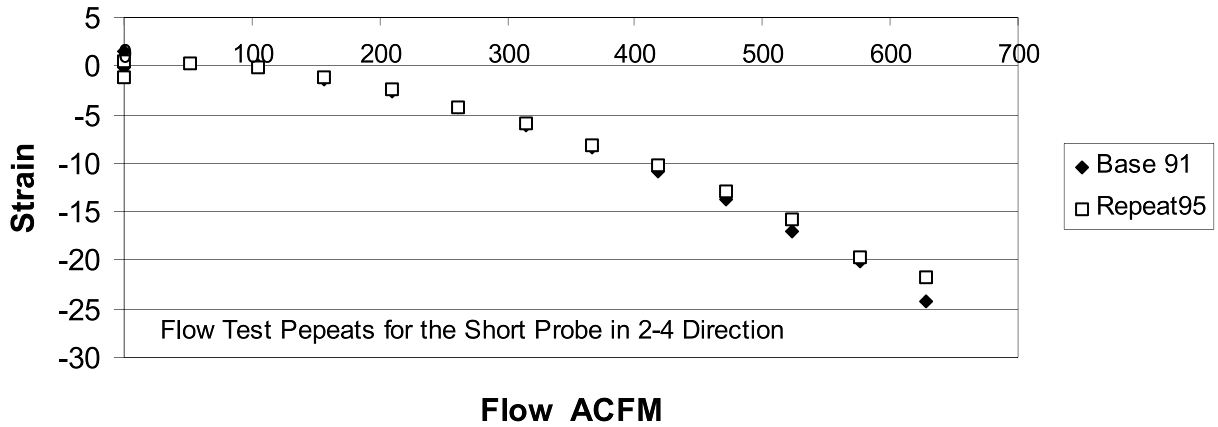


Figure 4-7. Strain As a Function of Flow Rate During Repeated Flow Tests of the Short Laboratory Drag Probe in the 2-4 (Negative) Direction.

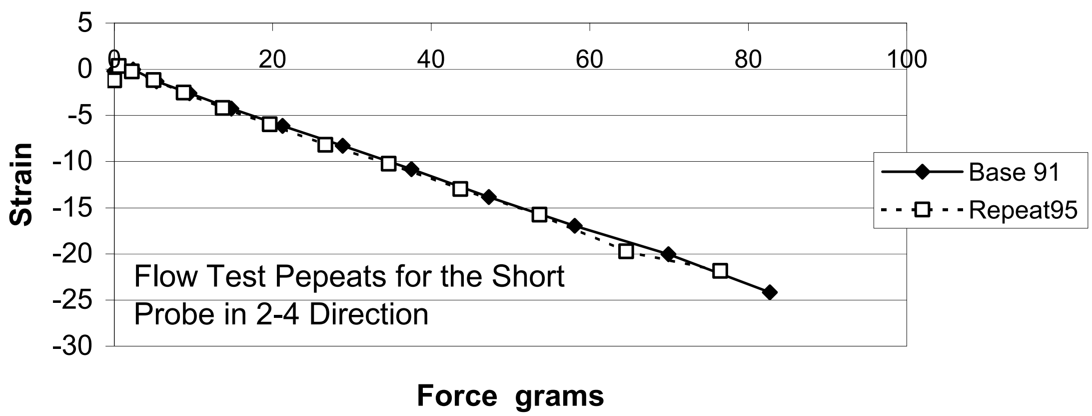


Figure 4-8. Strain As a Function of Calculated Drag Force During Repeated Flow Tests of the Short Laboratory Drag Probe in the 2-4 (Negative) Direction.

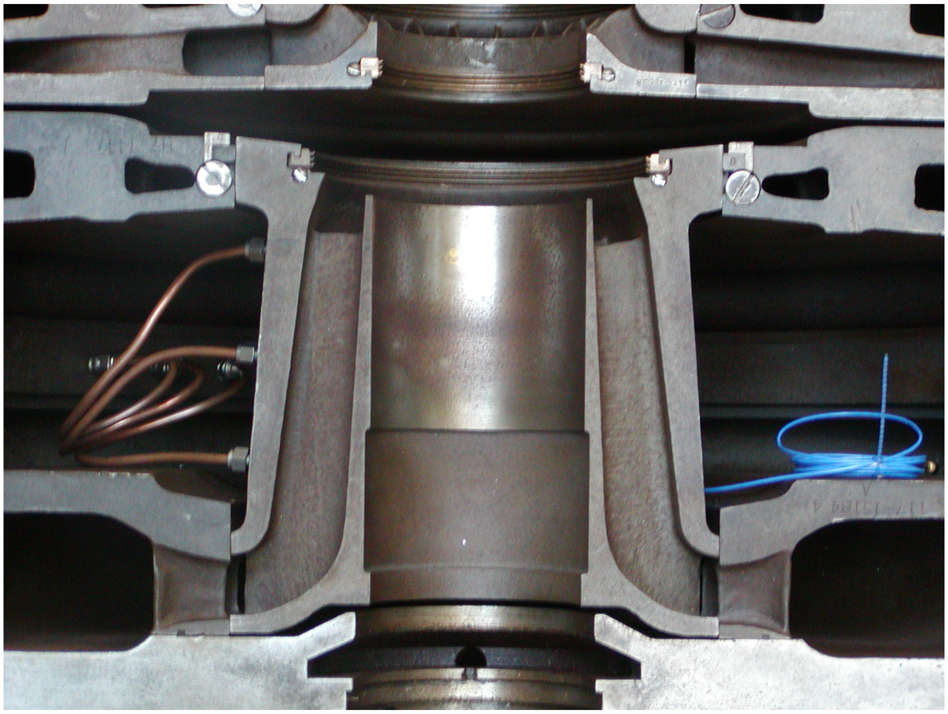


Figure 4-9. Photograph of Inlet Channel in Open Compressor Case Showing the Shaft Way, the Impeller Space, and the Channel Walls in the Inlet Annular.

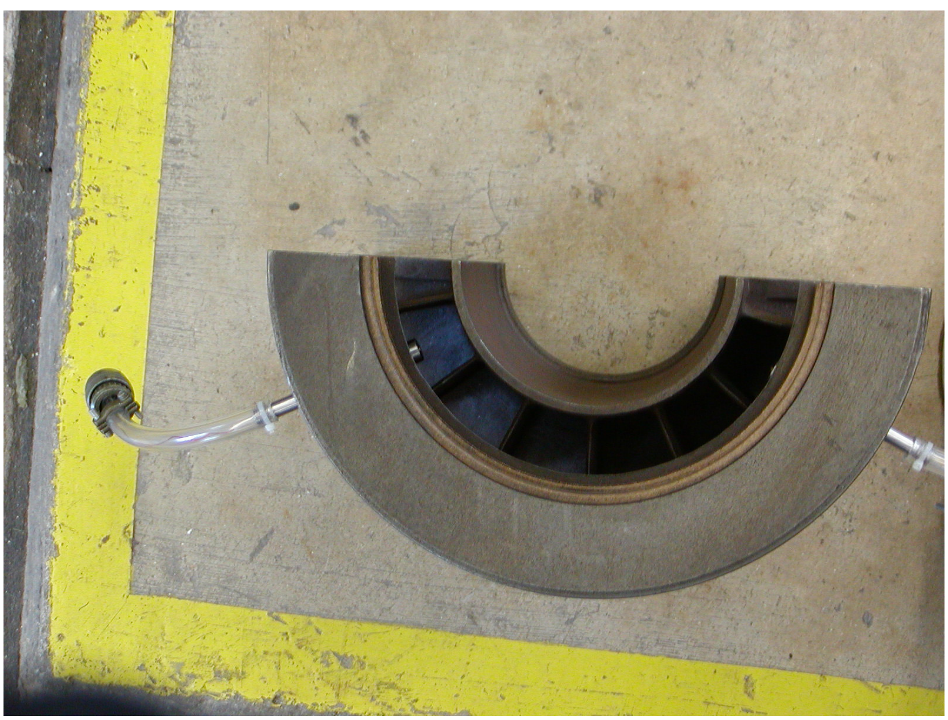


Figure 4-10. Photograph of Half of the Inlet Channel Removed from the Compressor Case with Two Surge Detection Probes Installed.

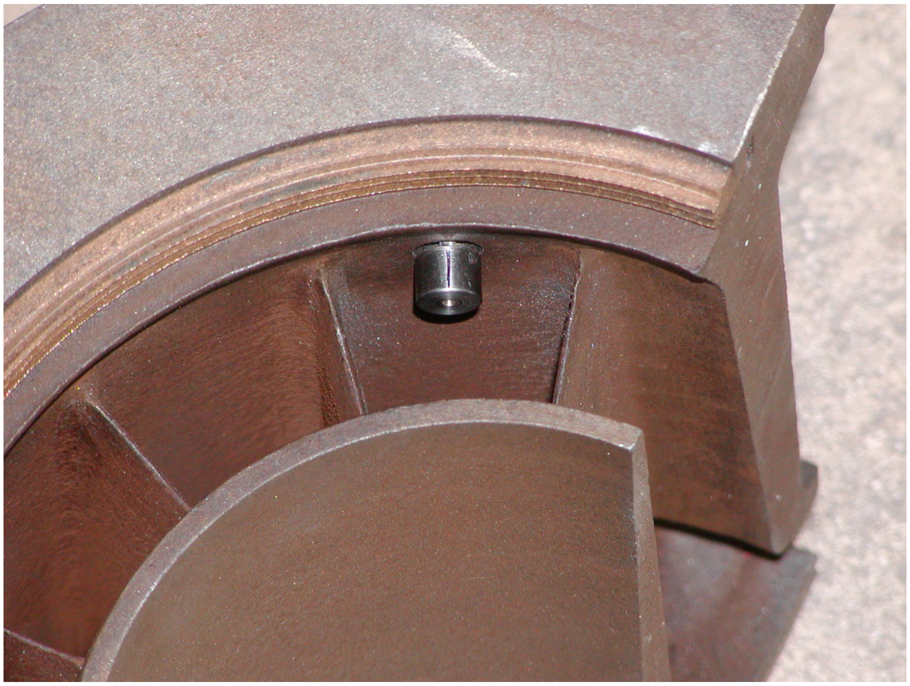


Figure 4-11. Photograph of Surge Detection Probe in an Inlet Channel of the Laboratory Centrifugal Compressor.

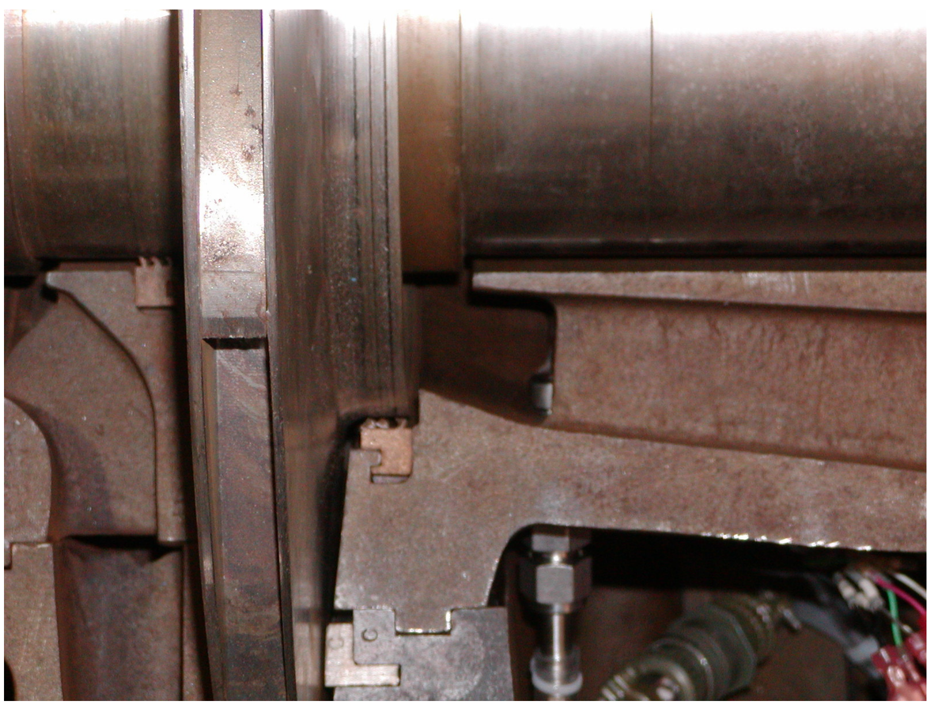


Figure 4-12. Photograph of Surge Detection Probe Installed in the Open Laboratory Centrifugal Compressor with Shaft, Impeller and Inlet Channel Shown.

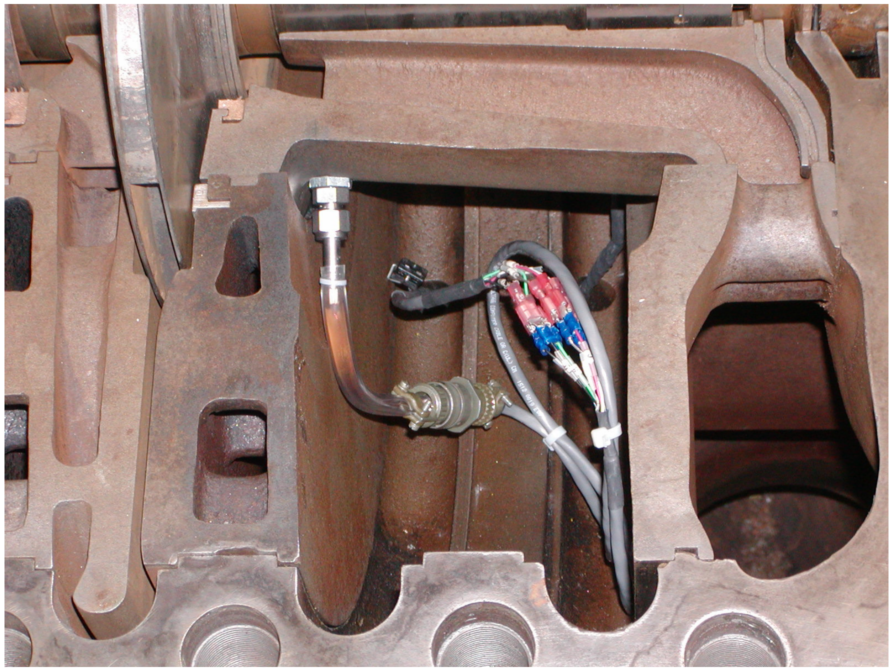


Figure 4-13. Photograph of Surge Detection Probe Installed in the Open Laboratory Centrifugal Compressor with Inlet Channel, Division Wall, and Wiring Shown.

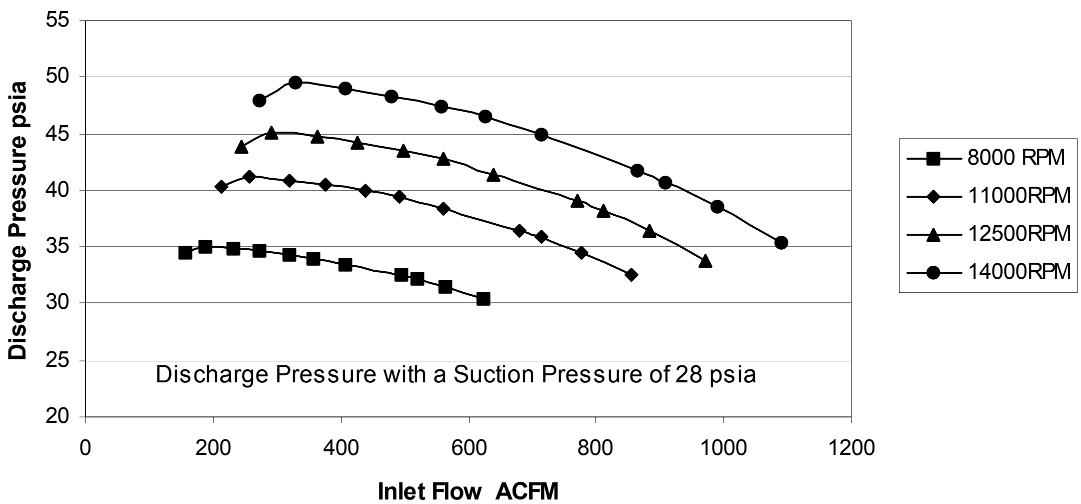


Figure 4-14. Measured Performance Map for the SwRI Laboratory Compressor with a Constant Suction Pressure of 28 psia for Various Operating Speeds.

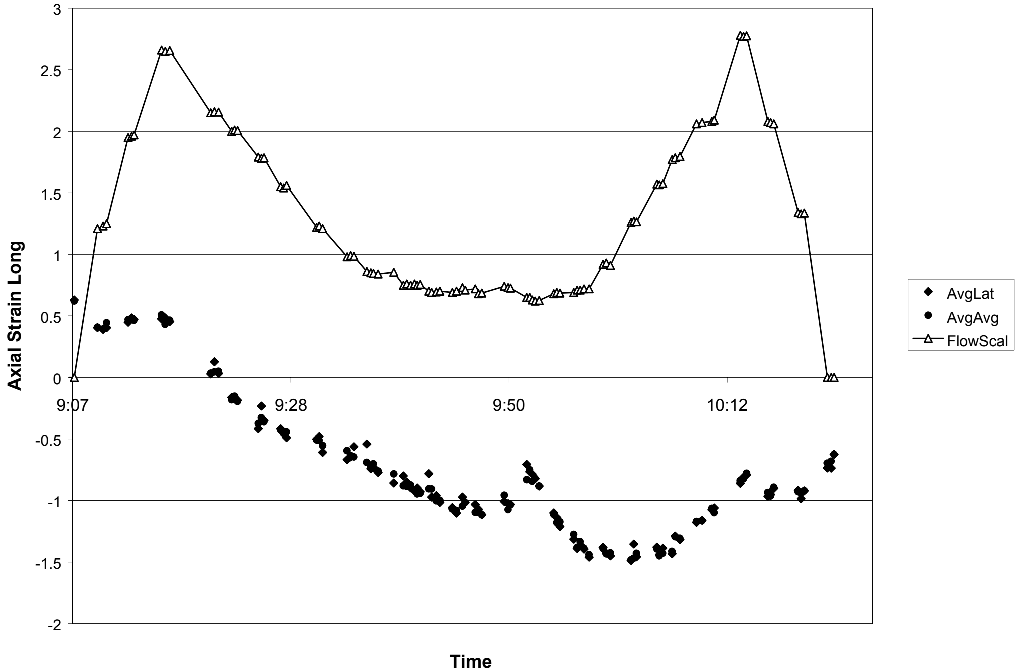


Figure 4-15. Axial Drag Probe Strain Changes as Flow in the SwRI Laboratory Compressor Decreases Towards Surge and Then Increases. Flow is a Comparative Value from which the Increase in Strain at Constant Low Flow Near Surge is Evident.

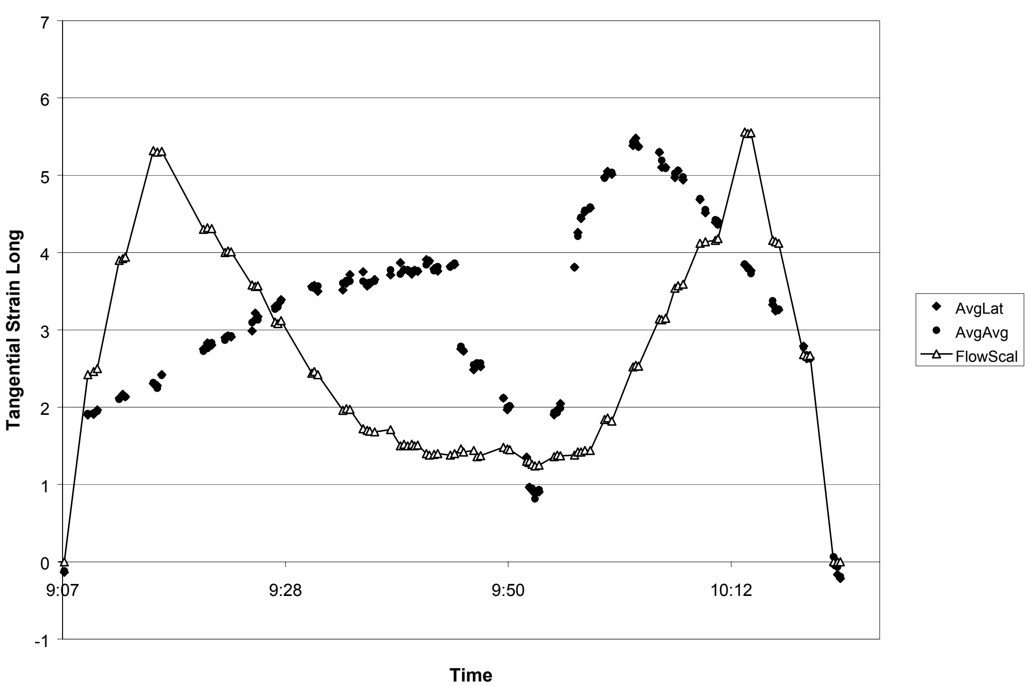


Figure 4-16. Tangential Drag Probe Strain Changes as Flow in the SwRI Laboratory Compressor Decreases Towards Surge and then Increases. Flow is a Comparative Value from which the Decrease in Strain at Constant Low Flow Near Surge is Evident.

during this particular test. However, a change (small increase) in the axial strain did occur at the lowest flow condition, which is close to surge. The change in the tangential flow and strain is more complex than the change in the axial strain.

After the compressor reaches a normal operating condition and the flow begins to decrease, the tangential strain, from both the long and short probes, increases as shown in Figure 4-16. However, at a condition near the lowest flow tangential, strain decreases while there is little change in overall flow rate as shown in near the center of Figure 4-16. This result implies a change in the flow pattern involving the tangential flow that will be shown to be related to the closeness of surge.

The results of a later test, including a surge event, indicate the same type of changes discussed above, and along with other results confirm that the impeller inlet flow pattern does change prior to surge. The general rules that apply for probe output during normal operational changes, before surge is approached, are that as compressor flow rate decreases, the axial direction strain decreases and the tangential strain increases. When the compressor approaches the actual surge condition, data shows that the axial strain does not continue to decrease but rather increases slightly, and the tangential strain does not continue to increase but decreases sharply prior to surge. These changes are related to the inlet flow pattern change but are not the expected recirculation in the axial direction. As shown in the CFD modeling, the recirculation in the SwRI compressor occurs closer to the impeller than the location of the surge detection probes, the largest area of recirculation is the downstream side of the inlet channels, and the flow field in the inlet area near surge is fully 3-dimensional.

The suction pressure from one of the later laboratory compressor tests of the surge detection probes is shown in Figure 4-17. During these operations, the suction pressure is not entirely steady due to the gas supply and the compressor seal leaks that must be balanced. The curve plotted in Figure 4-17 that is marked as flow is a scaled flow signal, such that it fits on the same plot as the other data, and this proportional representation of flow will be used on all of the figures presented in this discussion. The flow curve ends at the moment when surge occurred, which was at 11:29 a.m. on the time scale. The suction pressure increased suddenly at 11:29 a.m., as would be expected when a compressor surges, and the compressor was then shutdown at 11:35 a.m.

Figure 4-18 shows the discharge pressure during the same tests, including the responses of the discharge pressure to suction pressure variations and the sudden drop in discharge pressure at 11:29 a.m. when surge occurs followed by the drop in discharge pressure when the compressor is shut down. The two axial drag probe signals during this later surge detection test are shown in Figures 4-19 and 4-20 where the decrease in strain as compressor flow decreases and a small increase in axial strain starting at 11:27 a.m. are shown. The increase in axial strain at a near surge flow, but prior to surge, is an indication of the change in inlet flow pattern that indicates approaching surge. The axial strain also shows an increase in strain when surge occurred at 11:29 a.m., which implies that the drag probes at this location are outside of the main reverse flow stream.

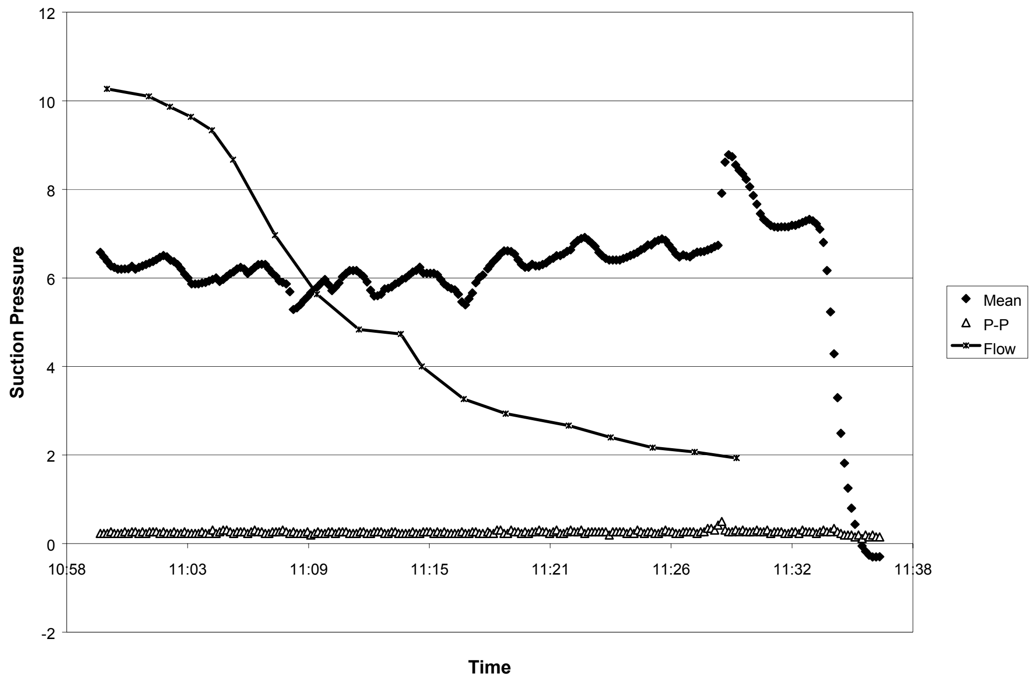


Figure 4-17. Suction Pressure and Relative Flow During a Surge Test of the SwRI Laboratory Compressor with Surge Detection Probes Showing Surge at 11:29.

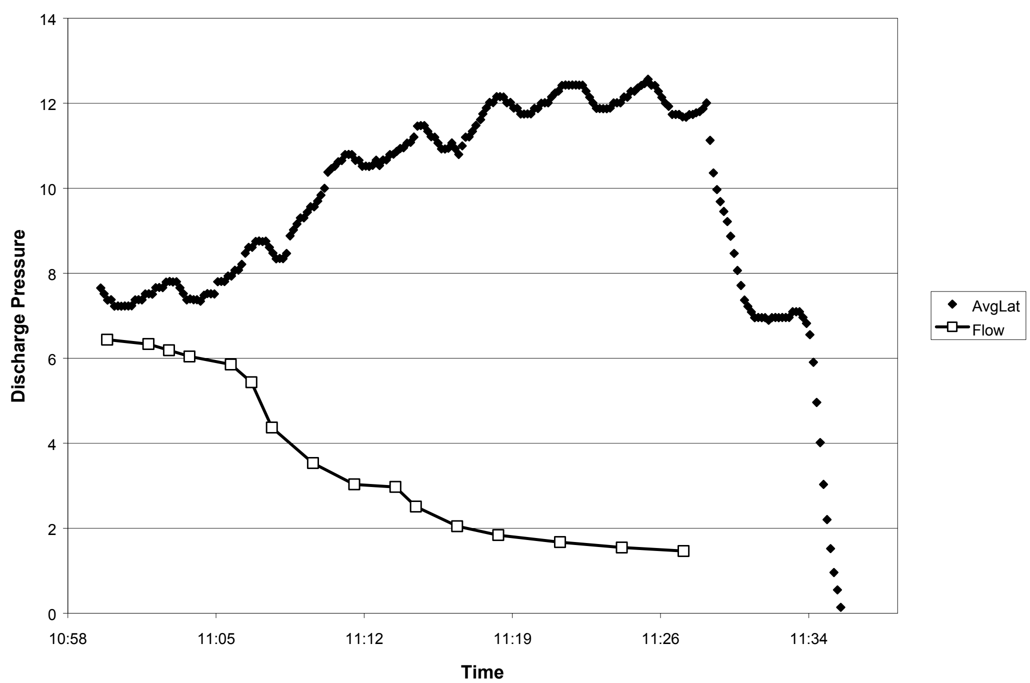


Figure 4-18. Discharge Pressure and Relative Flow During a Surge Test of the SwRI Laboratory Compressor with Surge Detection Probes Showing Surge at 11:29.

The tangential signal for the short and the long drag probes with the same scaled flow signal curve, as in the previous figures, are shown in Figures 4-21 and 4-22. As the operational flow rate through the compressor decreased, the tangential component of the flow at the drag probe locations increased, as indicated in Figures 4-21 and 4-22, until conditions prior to surge when the tangential drag suddenly decreased prior to surge. Note that when surged occurred, the tangential strain increased to a level above its previous magnitude and then decreased again prior to the compressor being shut down, showing that the tangential drag was responding to the surge cycle. As flow rate in the compressor decreases and increases in the tangential component of the inlet flow and, hence, an increase in the tangential strain is expected. The sudden drop in the tangential strain implies that a recirculation that develops downstream of the probes and to the side of the channel results in a reduction of the tangential velocity at the probe location. Flow returning or flowing out from the impeller face along the side of the channel need to flow out from the rotation direction wall towards the center of the channel in order to balance the mass flow and this reduces the tangential velocity near the outer edge of the recirculation zone, which affect the drag probes.

In order to better understand the velocity pattern in the impeller inlet area, velocity profile measurements were made with a hot film anemometer at different flow rates. The axial velocity profiles in an inlet channel at the same relative position at which the surge detection probes are located for flow rates from 595 m^3/hr, or ACFM, to 231 m^3/hr, or ACFM (see Figure 4-12), are shown in Figure 4-23. The corresponding tangential velocity profiles, which were taken by turning the hot film anemometer 90 degrees, are shown in Figure 4-24. These velocity profiles show the strong effect of the pinch or rapid reduction in channel height just prior to the probe location. This reduction in channel height (diameter) accelerates the flow along the top or outer wall of the channel and produces the stepped or two level velocity profile seen in Figure 4-23. Only at the lowest flow rates is the difference between the inner and outer velocities reduced. This acceleration is also believed to force the recirculation, caused by approaching surge, further downstream in the inlet channel. The tangential velocity profile, shown in Figure 4-24, experiences low velocities on the inside or smaller diameter wall as expected and increases in magnitude at higher diameters; however, the relative shape of the curves in not directly proportional to overall flow rate.

The result of this series of surge detection test with the drag probes in the SwRI laboratory compressor is that the inlet flow pattern does change as surge is approached, and these changes could be used to indicate the nearness of surge. However, because of the narrow inlet channel geometry in this compressor and the restricted location of the drag probes, the changes are not a clear recirculation, including a reversal of the axial strain that was expected. Although, the SwRI compressor does show changing inlet flow signals as surge is approached, it is not a good candidate compressor for development of a general surge avoidance controller based on inlet flow changes with detection by drag probes.

4.3 PREPARATIONS FOR FIELD TESTING

A pipeline centrifugal compressor operated by El Paso Energy and located in Columbus, Mississippi was selected as the first field test compressor for the direct surge control probes.

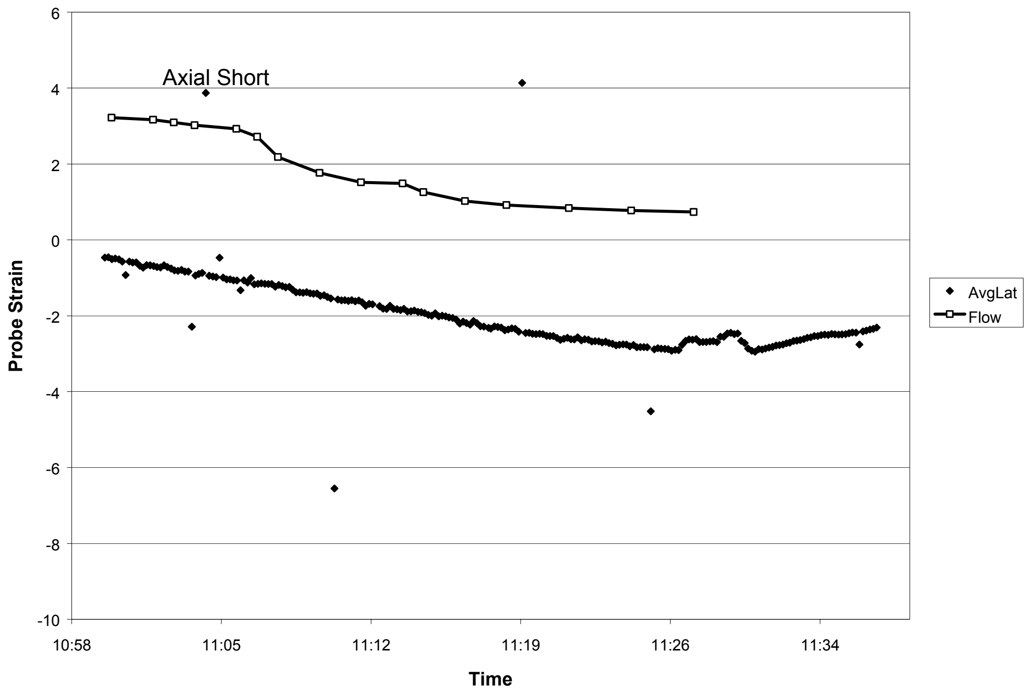


Figure 4-19. Axial Strain from the Short Surge Detection Probe and Comparative Flow During a Surge Event Test in the SwRI Laboratory Compressor Showing a Small Increase in Signal Starting at 11:27 and Prior to Surge at 11:29.

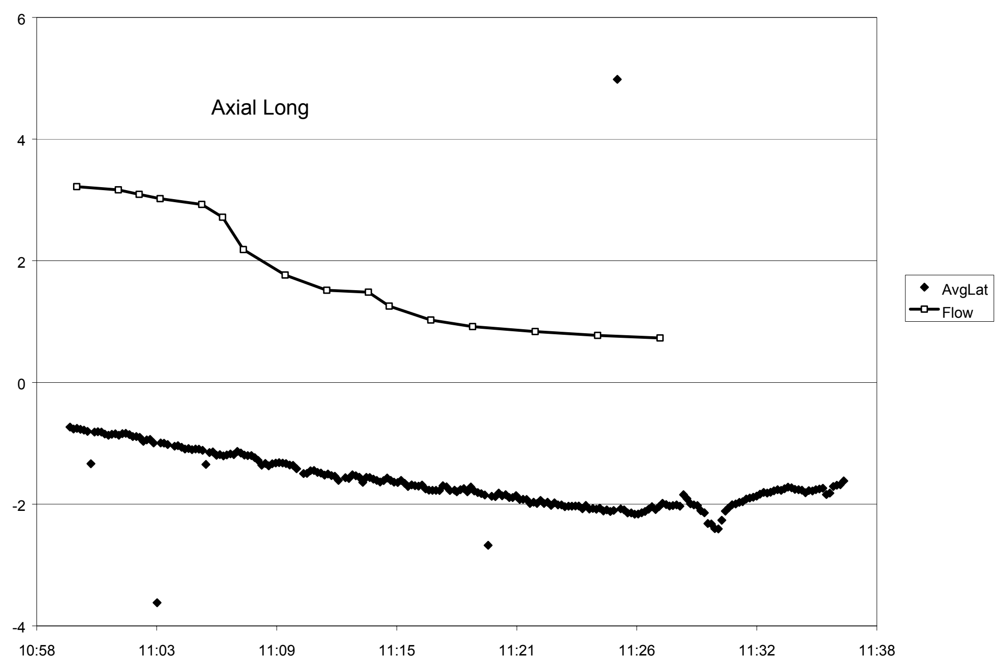


Figure 4-20. Axial Strain from the Long Surge Detection Probe and Comparative Flow During a Surge Event Test in the SwRI Laboratory Compressor Showing a Small Increase in Signal Starting at 11:27 and Prior to Surge at 11:29.

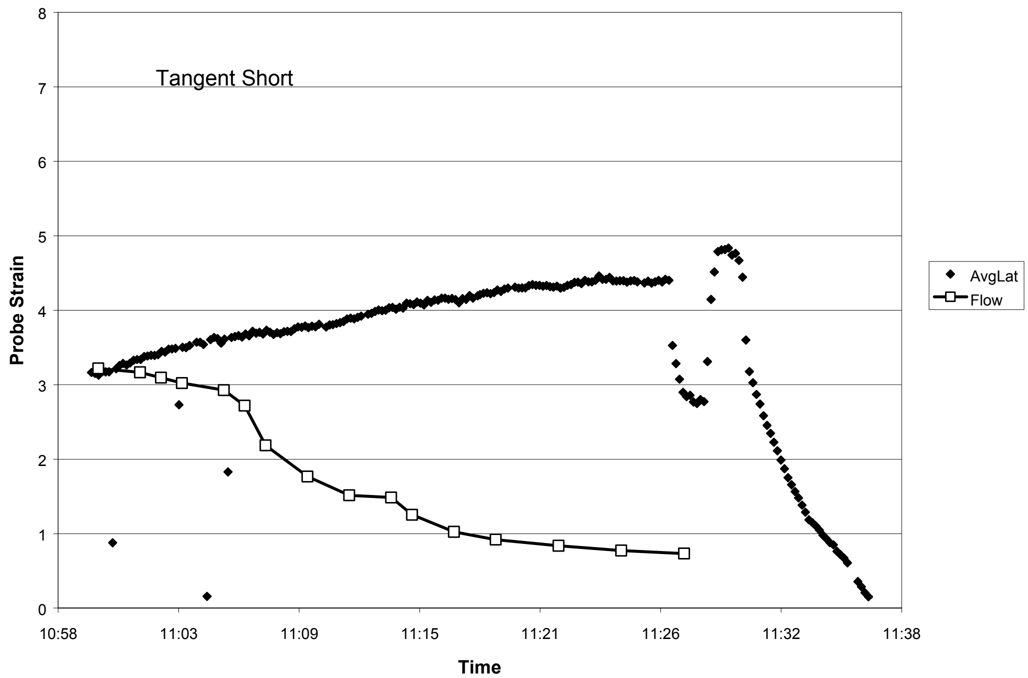


Figure 4-21. Tangential Strain from the Short Surge Detection Probe and Comparative Flow During a Surge Event Test in the SwRI Laboratory Compressor Showing a Large Decrease in Strain Starting at 11:27 Prior to Surge at 11:29.

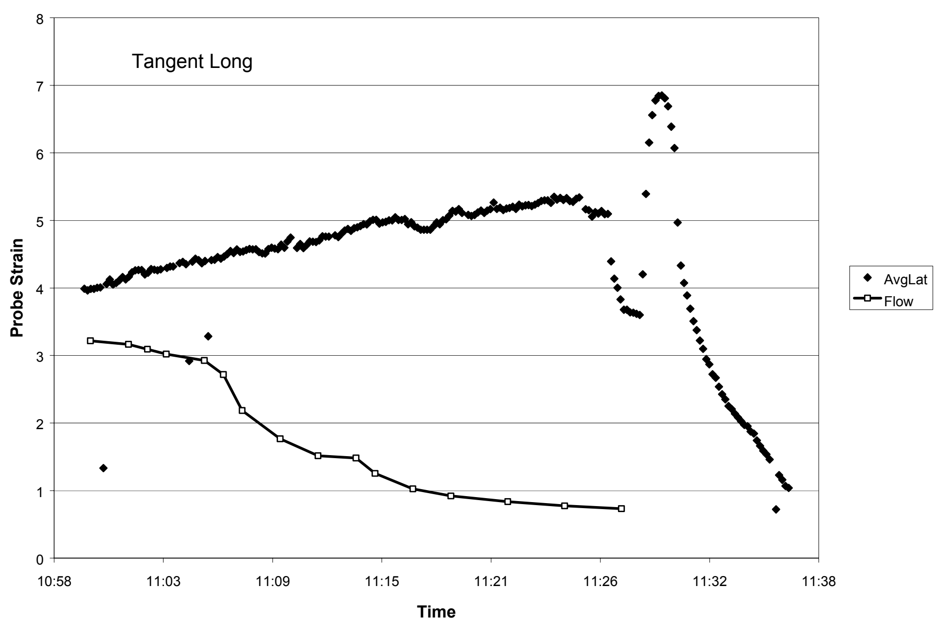


Figure 4-22. Tangential Strain from the Long Surge Detection Probe and Comparative Flow During a Surge Event Test in the SwRI Laboratory Compressor Showing a Decrease in Strain Signal Starting at 11:27 Prior to Surge at 11:29.

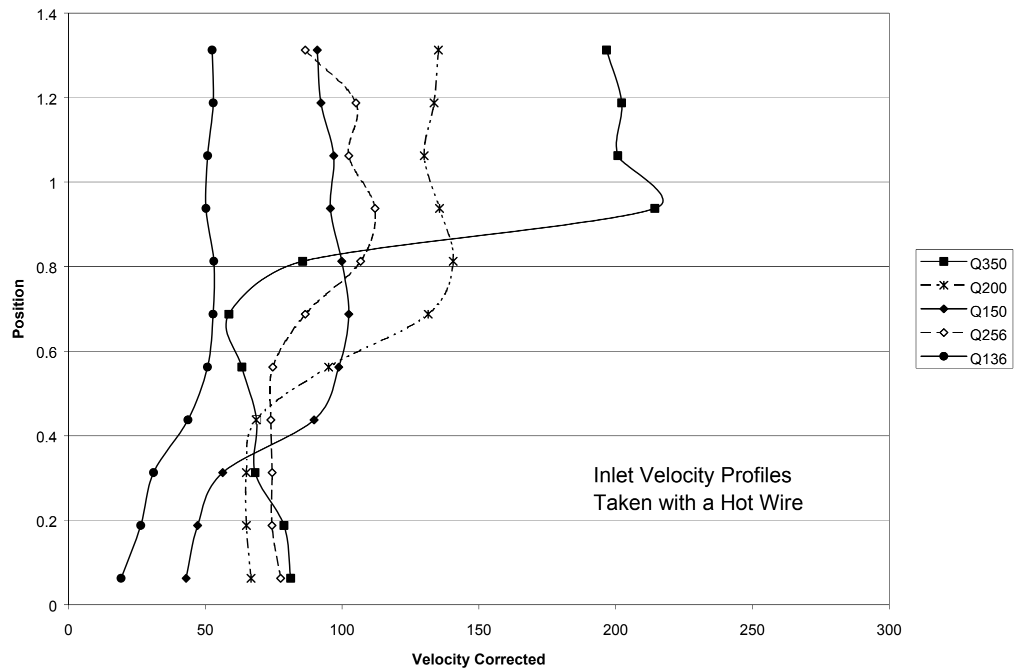


Figure 4-23. Axial Direction Inlet Velocity Profiles in the SwRI Laboratory Compressor at Different Flow Rates Measured with A Hot Film Anemometer.

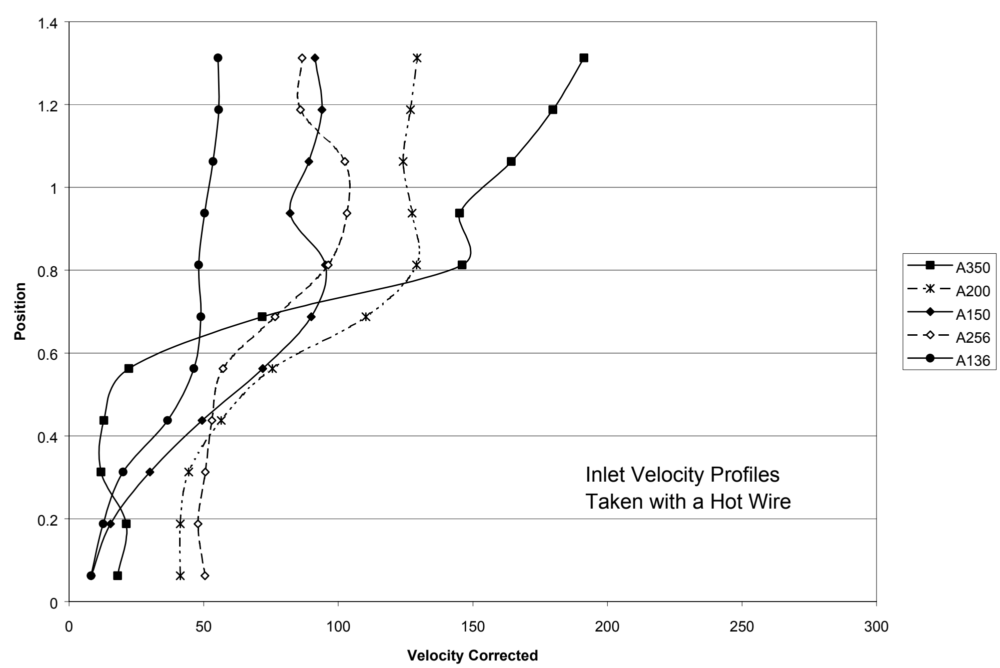


Figure 4-24. Tangential Direction Inlet Velocity Profiles in the SwRI Laboratory Compressor at Different Flow Rates Measured with A Hot Film Anemometer.

Operating conditions, a compressor map, and general drawing for this compressor were obtained by El Paso. Not only were surge detection drag probes designed for this compressor in accordance with the design process discussed in Section 3.3, but also a plan for installation and routing of the compressor wires out of the compressor case was developed. A rough cross-sectional drawing of the field site compressor with the drag probes and wire installation plan sketched is shown in Figure 4-25. The design calls for the drag probes (two) to be mounted through the inlet flow guide as close to the impeller as allowed by the diffuser wall and seal. The wires from the probes are then to be run in a flexible tubing conduit through the dead space between diaphragm walls, out through the suction diaphragm under the curved suction flow guide, and finally through a hole in the guide to a flange on the side of the compressor case where a pressure seal will be used for the wire penetration.

The flange where the pressure seal is located is present on this compressor because it previously had adjustable inlet guide vanes, which are now fixed vanes. A hole in the suction flow guide is also present because of the rod that was previously used for the inlet guide vanes. The hole in the suction diaphragm, the hole in the suction flow guide, and the unused flange on the side of the compressor case are key to getting the wires out of the compressor for this particular installation. A Connex multiply wire compression fitting is used for the pressure seal, and if heavy solid core wires are used, this has proven to be a completely effective seal.

After the field probes had been fabricated with the stain gauges mounted and the strain gauge wiring connected to wires within the tubing, the entire internal direct surge detection system was prepared for installation. With the field compressor disassembled, holes were drilled and tapped in the inlet flow guide and the probes were installed. The two surge detection probes mounted in the split apart compressor are shown in Figure 4-26 where the impeller, the outside of the inlet flow guide pulled back a short distance from the impeller, and the suction diaphragm with the flexible conduit passing through it can be seen. Figure 4-27 shows the suction side of the diaphragm with the compressor bundle assembled and the surge detection probes mounted in place at the inlet of the impeller. Figure 4-28 shows both the suction flow guide and the compressor ready for installation of the suction flow guide with the conduit with the drag probe wires ready for the final stage of installation.

The final field test has not been conducted because the wires to these probes failed approximately two days after the compressor was placed in service. The failure of the wires occurred in the short distance between the curved suction flow guide and the flange where the pressure seal in located. This part of the wire and conduit path is in the suction flow stream and it appears that insufficient preparations were made to secure the wires at this point. Prior to the wire failure, everything including the probes, the strain gauges, the continuity, and the pressure seal had been in place and functioning as expected. Plans have been made to repair the internal wiring and to conduct the field test shortly after completing the repair of the field installation.

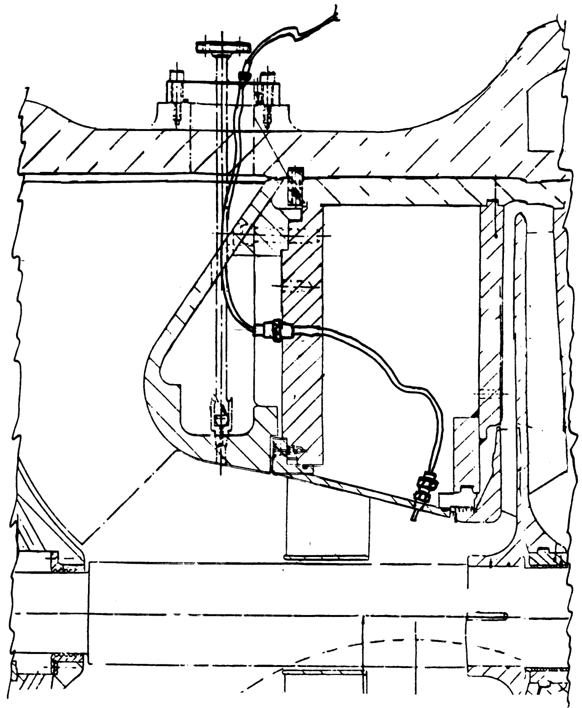


Figure 4-25. A Cross-Section Drawing of the Field Site Compressor with the Surge Detection Probe Location and Wiring Route Shown.

Figure 4-26. Photograph of the Surge Detection Probe Installation in the Split Field Site Compressor Showing the Probe Mounting on the Inlet Flow Guide, the Flexible Conduit for the Wiring, and the Wire Penetration of the Suction Division Wall.

Figure 4-27. Surge Detection Probes Installed in Front of Field Site Compressor Impeller.

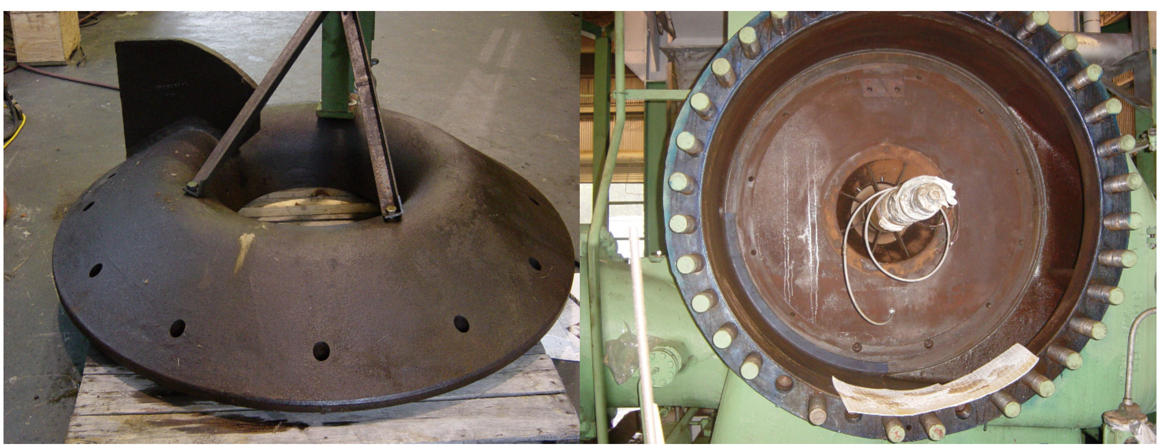


Figure 4-28. Inlet Flow Guide and Field Compressor Ready for Installation of the Suction Flow Guide on the Division Wall. The Flexible Conduit with Probe Wires is Shown in the Compressor Where it Penetrates the Suction Division Wall.

5. CONCLUSIONS

The following conclusions result from the industry's needs for this project and from the work that has been performed to date.

1. The motivation for a direct surge controller development is to refine surge control, reduce required surge margins, increase flexibility and efficiency, and safely save energy and reduce operating costs.
2. The background for this research is previous research sponsored by GMRC that identified an impeller inlet flow pattern (reversal) that indicates the approach of surge in centrifugal compressors and a method to detect that flow change.
3. Detailed specifications for the surge detection probes and the surge controller have been developed with gas industry input in order to address the industry's needs for a fundamental, accurate, rugged, flexible, and cost-effective surge control system.
4. Modeling of the flow field in the laboratory compressor has been performed to assist in interpretation of the laboratory test results. CFD models indicate that the recirculation zone in the laboratory compressor is small, off center, and closer to the impeller than the surge sensing probes due to the compressor's inlet geometry. Analysis of the system dynamics indicates that surge detection sampling should identify frequencies less than the compressor's running speed and should filter out higher frequencies.
5. A detailed design process for surge detection drag probes has been developed and explained in this report. The design process accounts for the size, flow rate, and gas properties in the individual compressor, the strength required to support the probe at the maximum flow and the strain sensitivity needed at minimum flow, the mechanical natural frequency, and the excitation frequencies so that the sensor will not fail dynamically, temperature compensation, and a means to pass the signal wires out of the compressor case.
6. The designed and fabricated surge detection sensors have been bench and flow tested to demonstrate and calibrate their sensitivity to flows and to confirm their satisfactory operation.
7. Testing of the surge detection probes in the SwRI centrifugal compressor demonstrated that the sensors do respond to changes that occur prior to surge and successfully indicate the compressors approach to surge. However, the probes are not located within the small recirculating flow in this compressor with the result that signals are not as expected and are primarily tangential changes rather than an axial flow reversal.
8. Preparation for a field test of the surge detection sensors, including selection of the compressor, design and fabrication of the probes, and installation of two surge detection drag probes and their wiring within the field compressor was completed. A failure of the wires (due to vibration) just inside the compressor case has caused a delay and requires an internal repair before the field test can be conducted.

6. REFERENCES

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8. LIST OF ACRONYMS AND ABBREVIATIONS

°F	Degrees Fahrenheit
ACFM	Actual Cubic Feet per Minute
CFD	Computational Fluid Dynamics
cm	Centimeters
DOE	Department of Energy
GMRC	Gas Machinery Research Council
Hz	Hertz
m/s	Meters per Second
m^3/hr	Cubic Meters per Hour
N s/m^2	Nutant in Seconds per Square Meters
NEMA	National Electrical Manufacturers Association
PCRC	Pulsation and Compressor Research Council
psia	Pounds per Square Inch Absolute
rpm	Revolutions per Minute
SwRI®	Southwest Research Institute®
μ	Viscosity