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

FINAL REPORT
October 2001 — February 2005

Prepared for

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

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

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February 25, 2005

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ABSTRACT

The objective of this Direct Surge Control project was to develop a new internal method to avoid surge of pipeline compressors. This method will safely expand the range and flexibility of compressor operations, while minimizing wasteful recycle flow at the lower end of the operating envelope. The approach is to sense the onset of surge with a probe that directly measures re-circulation at the impeller inlet. The signals from the probe are used by a controller to allow operation at low flow conditions without resorting to a predictive method requiring excessive margin to activate a recycle valve. The sensor developed and demonstrated during this project was a simple, rugged, and sensitive drag probe. Experiments conducted in a laboratory compressor clearly showed the effectiveness of the technique. Subsequent field demonstrations indicated that the increase in range without the need to recycle flow was on the order of 19% to 25%. The cost benefit of applying the direct surge control technology appears to be as much as $120 per hour per compressor for operation without the current level of recycle flow. This could amount to approximately $85 million per year for the U.S. Natural Gas Transmission industry, if direct surge control systems are applied to most pipeline centrifugal compressors.

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1. EXECUTIVE SUMMARY

The Direct Surge Control research conducted by Southwest Research Institute® (SwRI®) and summarized in this report is part of the Natural Gas Infrastructure program under the U.S. Department of Energy (DOE), Office of Fossil Energy, National Energy Technology Laboratory (NETL), Strategic Center for Natural Gas and Oil (SCNGO) and the Delivery Reliability Program. This program fosters developments to enhance the U.S. Natural Gas Transmission system. The co-funders for this research included the Gas Machinery Research Council GMRC and Siemens Energy. The other partners were GMRC member companies, including Duke Energy Gas Transmission, El Paso Corporation, and Solar Turbines, Inc. The objectives of this project were to refine surge control, reduce surge margins, increase efficiency and flexibility in centrifugal compressors, and safely save energy and reduce operating costs.

Surge is a potentially damaging instability or collapse of flow through a centrifugal compressor that is usually avoided by recycling flow to maintain an arbitrarily selected minimum flow. However, recycling flow is inefficient and wasteful particularly when the flow maintained is higher than necessary to avoid surge. The approach to this research was based on previous research sponsored by the GMRC in which it was found that a re-circulation occurs in the outer diameter of the impeller inlet as surge is approached. If this impeller inlet re-circulation can be detected with a rugged and reliable internal probe, then this change in the local flow pattern can be used as a direct surge control signal.

To ensure that the results of this work would meet the industry's requirements, a set of specifications was developed by the industry oversight committee established for this project. A surge detection probe design process was developed and several probes for laboratory and field compressors were designed and fabricated according to the procedure. The design process requires that the probes are sufficiently rugged for the maximum flow rate and yet sensitive enough for the lowest or near surge flow rates. One of the key design requirements was that wiring from the internal probes be installed with a secure and reliable connection from the probe to the external amplifiers.

One of the major findings from the research was that the current approach for installing the pre-surge detection probe at the inlet face of the impeller was effective only for modern three-dimensional (3D) impellers. The re-circulating flow pattern was not present or sufficient for two-dimensional (2D) impellers in which the leading edges of the blades were recessed from the inlet of the impeller. It is estimated that between 66 and 80 percent of the centrifugal compressors in natural gas transmission service are of the modern 3D design. Test results using a 3D laboratory compressor showed that the re-circulating flow can be detected as a combination of a local increase followed by a rapid drop in the axial strain and a significant increase in the otherwise small tangential strain as the total compressor flow is reduced towards surge. This pattern of changing strain signals was repeatable at different compressor speeds and different rates of approach to surge and can be used as a reliable surge control signal. The nature of the changing strain signals can be explained by the expected impeller inlet flow patterns as surge was approached. Field test results in a modern 3D pipeline style compressor showed similar patterns of decreasing axial strain and increasing tangential strain. The field compressor strain signals contained small local peaks followed by a small additional decrease in the axial strain and a marked increases in the tangential strain as overall flow decreased and surge approached. There were some differences between the laboratory and field compressor strain data due to geometric differences between the compressors. However, if the correct algorithm, such as considering the difference between the axial and tangential strain was used, the crossing or near zero value of the difference indicated that the compressor was approaching surge. The increase in the operating range for the field compressor obtained by using the direct surge control methods ranges from 25.4% for the low speed operation to 19.2% for the high speed operation of the field compressor. Projecting the low pressure operations of the test field compressor to normal pipeline operating conditions, a savings in terms of fuel consumed of approximately 10 to 24 thousand standard cubic feet, MSCF, per hour, is expected for each compressor to which direct surge control is applied. Thus, with an average cost of gas of $5.00 per MSCF, the cost saving is between approximately $50 and $120 per hour per compressor for operation without the current level of recycle flow. Depending on the amount of low flow operations at each compressor, the saving in fuel gas could be as much as 85 to 200 MMSCF/yr at each compressor with direct surge control. Thus, using the direct surge control technology could result in a savings of approximately $50 to $85 million per year if applied to most of the compressors in the natural gas industry.

2. INTRODUCTION

2.1 CENTRIFUGAL COMPRESSORS AND THE DIRECT SURGE PROJECT

The capacity and flexibility of the Natural Gas Transmission system within North America will increase over the next decade and a half 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 DOE Office of Fossil Energy, National Energy Technology Laboratory, Strategic Center for Natural Gas and Oil has established a Natural Gas Infrastructure program under the Delivery Reliability 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 were to refine surge control, reduce surge margins, increase efficiency and flexibility, and safely save energy and operating dollars.

This Direct Surge Control research program was conducted by Southwest Research Institute® (SwRI®) with the cooperation from the Natural Gas Industry. Duke Energy Gas Transmission, El Paso Corporation, and Solar Turbines, Inc. provided guidance and ideas for the project and offered support and test sites. The co-funding partners are the Gas Machinery Research Council (GMRC) and Siemens Energy and Automation, a member company. GMRC is a member association of operating companies and equipment suppliers that supports and disseminates research that benefits the Natural Gas Industry. Siemens is interested in commercialization of a Direct Surge Control system. An Oversight Committee with representatives from the natural gas pipeline companies, Siemens, Solar Turbines (another GMRC member company), SwRI, and DOE was established to provide the industry's guidance and direction for this project. The entire "Increased Flexibility of Turbo-Compressors in Natural Gas Transmission Through Direct Surge Control" project was conducted as a three phase program that included verification of near surge sensing, development of a prototype surge controller, and demonstration testing of the benefits of a prototype surge controller. The first phase included concept verification and testing. The second phase included laboratory and field testing and development of the prototype controller. The third phase included the final field and demonstration testing.

2.2 BACKGROUND

Centrifugal compressors constitute a significant portion of compression equipment throughout the energy industry and are a key element for operation of natural gas pipelines. Simple pipeline style centrifugal compressors are reliable, efficient, and function well when operating near their design conditions. However, flow rates and pressure rise requirements during pipeline operations vary widely and result in compressors reaching various operating limits. A typical performance map for a pipeline compressor, which shows pressure ratio as a function of inlet volumetric flow, is shown in Figure 2-1, which indicates some of the limits on operation of typical centrifugal compressors. Maximum speed, discharge pressure, or power limits occur at the top of the map where the pressure rise reaches a limit for the compressor. The

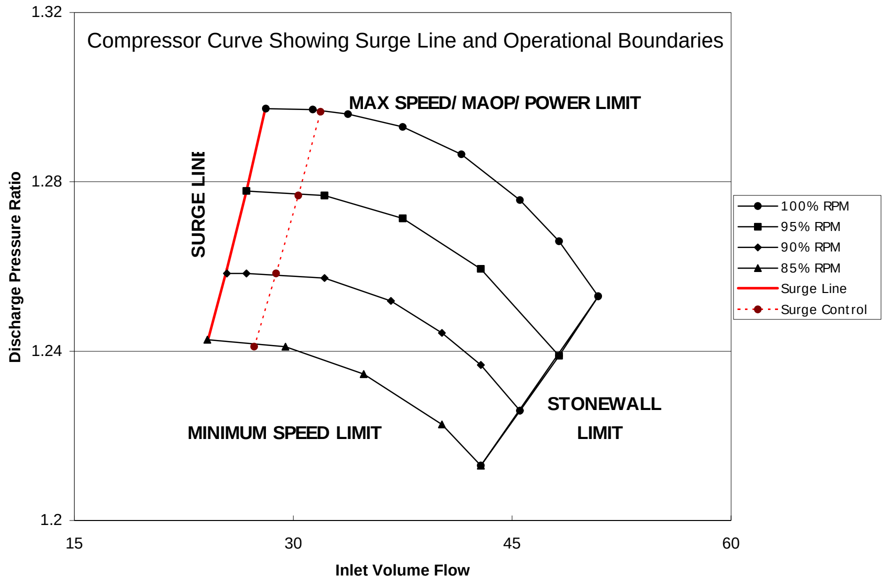


Figure 2-1. Typical Compressor Performance Map Showing Pressure Ratio as a Function of Inlet Volume Flow for Various Speeds with Operation Boundaries Showing a Surge Line and a Current Technology Surge Control Line

high flow or stonewall limit is reached on the right side of the performance map, and the minimum stable or minimum driver speed limit is reached at the bottom or low pressure rise (low head) area of the compressor map. The limit for low flow operations is set by a flow instability known as surge, which is expected to occur along the low flow limit shown as the solid red line on the left side of the figure. The exact location on a compressor map at which surge occurs is not normally known and is affected by many factors, including gas composition, piping configuration, parallel operating units, and the calculation of system parameters. As a result of this uncertainty in the external conditions at which surge will occur and the uncertainty in the calculations, a surge control line is usually estimated and placed on the map. The surge control line usually has a 10% of design flow margin above the expected surge line as shown by the dotted line. With no other indication of when surge will occur, old surge control systems usually maintained a compressor's flow at or above the surge control line by recycling flow in order to protect the compressor from surge. The new direct surge control system's intent is to protect the compressor and optimize the operations.

Surge is a complete breakdown of the flow pattern within a centrifugal compressor's impellers and passages, which results in a reversal of flow through the compressor. This global collapse of flow through a centrifugal compressor results when too little flow is being forced against more pressure rise than the compressor can generate at that operating condition. Surge usually occurs suddenly and with little warning, although in some, but not all cases, unsteady pressures and vibrations that are due to stall or off-design operating conditions precede surge. The pressure variations and vibrations that are seen before surge in some cases are not fundamental indications of approaching surge and can occur well before surge is imminent or not at all prior to surge. When surge does occur, both flow rate and head decrease rapidly and gas flows backwards through a forward spinning impeller, which causes large dynamic loads on thrust bearings, blades, and other components of the compressor. Surge at full speed and head is a dramatic and violent event that disrupts throughput, can cause damage to a compressor, and can lead to significant downtime. Surge during normal operation should always be avoided.

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, if the original system condition that caused surge at a high head and low flow continues to exist, the compressor will cause the head to increase and flow to decrease until surge occurs again. The consequence of this surge, recovery, and repeated build up to surge is a surge cycle as shown in terms of flow and pressure ratio in Figure 2-2 [Ref. 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.

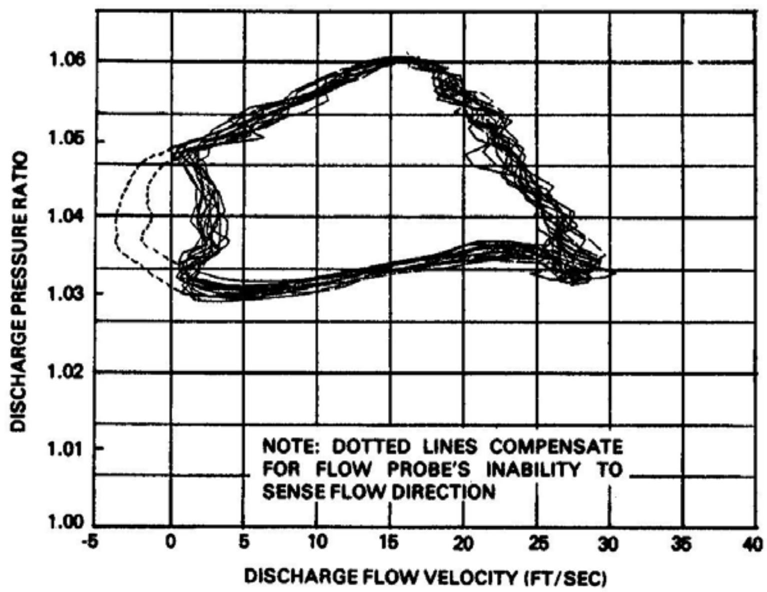


Figure 2-2. A Surge Cycle as Measured with Fast Response Pressure Transducers and a Hot Film Anemometer Shows Large Energetic Changes in Flow and Pressure

Surge avoidance is essential for pipeline compressors and is normally achieved by recycling gas around the compressor to maintain a flow no less than the surge control flow rate. Recycling flow is achieved through a pipe connection (recycle line) that carries some discharge gas back to the suction through a restrictive control valve known as a recycle valve. Recycling flow lowers the pressure rise (head) the compressor can generate, consumes extra fuel or power from the driver, and significantly lowers overall efficiency. If the actual approach of surge could be detected, then the centrifugal compressors could be operated with less recycled flow and a higher overall efficiency.

The reason that a margin above the suspected surge limit is required is that the exact condition at which surge occurs cannot be determined accurately 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, but these behaviors are a result of stall and not true indications of 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. 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, the potential damage from surge, and the safety of employees, 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.

In current compressor controls, 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 its 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 returns 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.

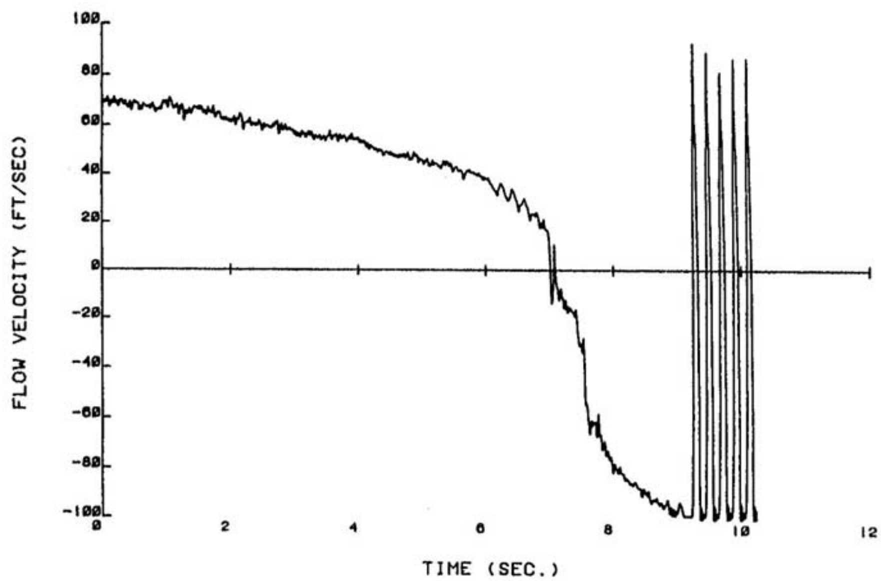
Although many centrifugal compressors could operate between 60% and 100% of their design flow if they operated down to the surge limit, they normally operated between 70% and 100% of their design capacity due to the surge margin established by the surge control line. All centrifugal compressors use their recycle valves at times, such as at startup 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 indicated 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 compressors 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 was to reduce surge margins by removing uncertainties, use less recycle flow during operations, reduce wasted fuel, and reduce operating costs.

2.3 PREVIOUS TECHNOLOGY

In the early 1980's, the GMRC funded research at SwRI to study the processes involved in surge. At that time, there were no known or reliable precursors to surge, although a number of signals and theories had been tried. The early GMRC research eliminated many changes in compressor flows and conditions as indicators of surge. Tests in the early work showed that the vibration and pulsation indicative of off-design operation do not always occur prior to surge and are caused by other mechanisms rather than by near surge conditions. One change was found that does indicate the approach of surge in small laboratory compressors and that change was a re-circulation in the outer portion of the impeller inlet. This observed impeller inlet flow re-circulation pattern has subsequently been verified in other laboratory and field compressors and has been reported by other authors [Refs. 2 & 3].

In the early GMRC funded research, the inlet area of the small laboratory compressor was instrumented for flow direction and magnitude. The only significant change observed as the compressor approached surge was a reduction in magnitude of velocity along the outer diameter of the impeller inlet [Ref. 4]. This reduction in local flow was significantly greater than the reduction in average flow. The outer ring of the inlet flow field also develops a tangential component in the direction of the impeller rotation as flow rate was reduced. Figure 2-3 [Refs. 4 & 5] is a plot of the inlet flow in the direction towards the impeller measured with a bi-directional pitot type probe along the outer wall of the inlet passage, and shows that the flow along the outer wall reduces rapidly, reaches zero, and begins to flow backwards as surge is

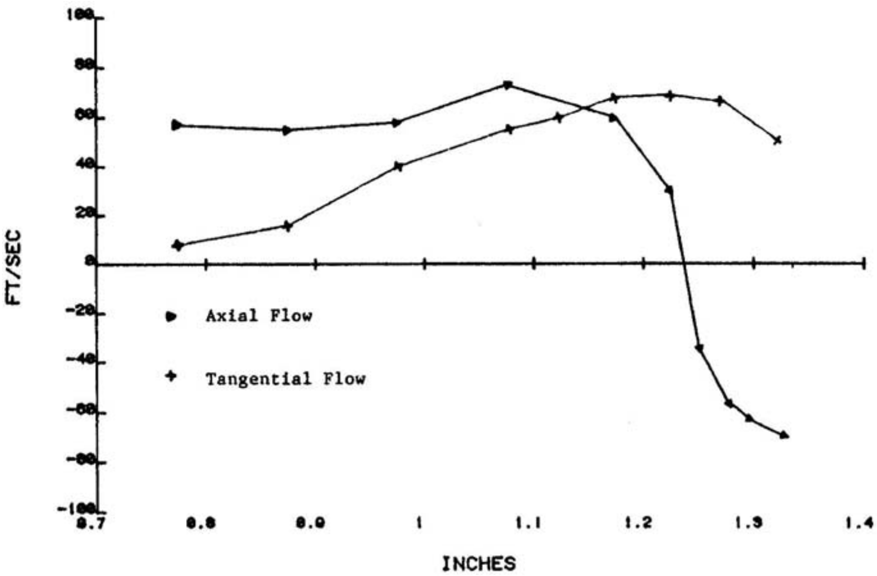


Near the Wall at Impeller Inlet

Figure 2-3. Change in Outer Wall Inlet Flow Signal as a Previous Small Laboratory Compressor Approaches and Reaches Surge

approached. The data shows that the flow re-circulation occurred only in the outer diameter area of the impeller inlet. This phenomenon would be useful as a surge avoidance and control indication if this change can be measured in field compressors. The development of a re-circulating flow pattern at the inlet of a centrifugal compressor impeller is an indication of approaching surge.

Figure 2-4 [Refs. 4 & 5] shows the 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 measured tangential component of the velocity is also shown in the figure where the tangential component is seen to be small at the inner radius and to increase to a large value at the outer radius.



Axial & Tangential Velocity Along the Inlet Wall

Figure 2-4. Velocity Profile of Impeller Inlet Flow in the Axial and Tangential Direct Measured on a Small Laboratory Compressor Near Surge

Since this GMRC research, similar findings of re-circulating flow have been found and published by other researchers. The results of Kammer and Rautenberg's [Ref. 2] work form 1985 is shown in Figure 2-5, where a strong inlet re-circulation is shown and the increase in the tangential velocity component is confirmed. The work of Mizuki and Oosawa [Ref. 3], published in 1992 and reproduced in Figure 2-6, shows that the re-circulation zone increases in size as surge is 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

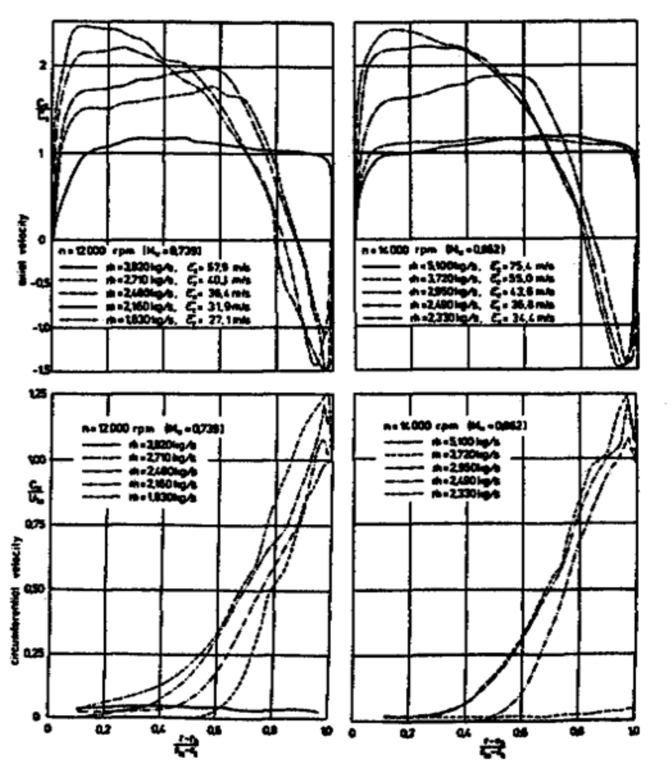


Figure 2-5. Inlet Axial and Tangential Velocity Profiles in a Centrifugal Compressor Approaching Surge as Published by Kammer and Rautenberg [Ref. 2] in 1985

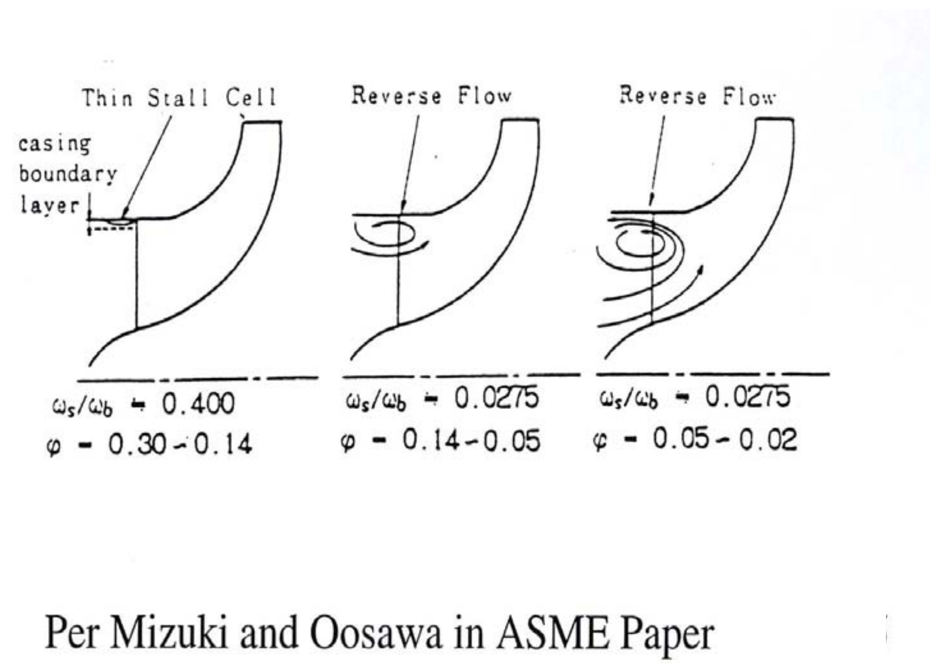


Figure 2-6. Centrifugal Compressor Impeller Inlet Flow Patterns as Flow is Reduced as Determined by Mizuki and Oosawa [Ref. 3] in 1992

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.

3. EXPERIMENTAL

3.1 SPECIFICATION

In order to focus the direct surge development effort, the first step was to identify specifications for a direct surge control system that would benefit the natural gas industry. The overall requirements for the sensor were that it be 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 controller were that it receive the surge signal from a drag probe and incorporate algorithms and flexible processing to produce outputs that prevent a compressor from reaching surge while maintaining an economical surge margin. The full specifications, as reviewed and approved by the industry Oversight Committee, are shown in the following paragraphs.

1. The sensor should have sufficient sensitivity and rangeability to detect changes in flow direction, magnitude, and condition that indicate the nearness of compressor surge. The sensor should respond to flows throughout the compressor's full range of operations and should accurately and effectively measure low, near zero, and reverse local flows. The sensor should measure local flow velocities from less than three feet per second to an upper near surge related limit. The sensor should perform accurately and without drift at low, near zero, and reverse flow conditions over a long life (at least 10 years). The sensor should be insensitive to thermal drift.
2. The sensor should 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 should be compatible with the compressor environment and be robust. An element of redundancy should be incorporated into the sensor system to improve its reliability. The sensor should be such that its failure, mechanically or electrically, will not endanger the compressor in any way. The sensor should be reliable.
3. The sensor should not suffer from flow-induced vibrations under the full potential range of conditions. That is, the sensor's mechanical natural frequencies should be well removed from any vortex shedding frequencies, and, in addition, no severe flow induced phenomena should occur within the normal range of compressor operations.
4. The sensor should 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 should 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 should receive the surge control signal and should 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 should 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 should 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 should 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 should provide for surge control and should 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 should 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 should be tested, evaluated, debugged extensively, and documented with drawings and instruction manuals.

3.2 DESIGN PROCESS

3.2.1 Probe Design and Fabrication

The objective of designing and installing a surge detection probe for any centrifugal compressor was 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 was 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 was a drag probe, which included 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 design process and some guidelines for the design are as follows:

1. Calculate the gas density and flow velocity range in the suction flow near the impeller 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. Modify the above sizing and repeat the calculations until all the requirements are satisfied and the various excitation and response frequencies are mismatched.

5. Design the probe and a probe holder to penetrate or accommodate the outer wall of the impeller inlet channel, such that the drag body is a short distance (from 0.2 to less than 0.6 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. CONAX compression fittings have been used during this research.

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 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 3-1. The probes 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 were mounted on all four sides of the bending beam and the wires were installed through the exit holes, the strain gauges and wires were coated with a protective coating. A tube fitting for retaining the surge probe and threading into the outer wall of the inlet channel was 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 3-2.

Figure 3-1. Photograph of the First Surge Detection Drag Probes Fabricated for a Laboratory Centrifugal Compressor

Figure 3-2. Photograph of a Completed Surge Detection Probe Ready for Installation in a Laboratory Compressor

3.2.2 Probe Calibration and Flow Tests

The first test to characterize the response of surge detection probes was a static bench test in which selected masses (weights) were hung on the probe and the magnitude of the strain was recorded to confirm the probe's sensitivity to force. There were four strain gauges on each probe, and they were wired in pairs so that there were 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 3-3 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 3-4 where the agreement with ideal calculations is good and one of the strain gauges has a lower sensitivity than the others by approximately 3.6%. The sensitivity of one gauge being low was most likely a result of an installation effect. Since the detection of surge was based on a change from the normal flow condition, 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 3-5 where the mechanical natural frequency of 932 Hz was clearly indicated. The mechanical natural frequency of the longer laboratory drag probes was 903 Hz, which was 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 3-6. This data showed the strain at different flow rates for three repeated tests and indicated 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. Except for the uncertainty in strain at very low flows, the response of the long laboratory probe in the 2-4 strain gauge direction was 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 3-7 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 was heated up to approximately 130°F. In Figure 3-7, it appears that the strain for the third test was lower than it should have been. However, the density of the gas was lower at the higher temperature, and when the drag force was calculated and the strain was plotted as a function of force as shown in Figure 3-8, the agreement between the test runs was within 10% or better for all the test points.

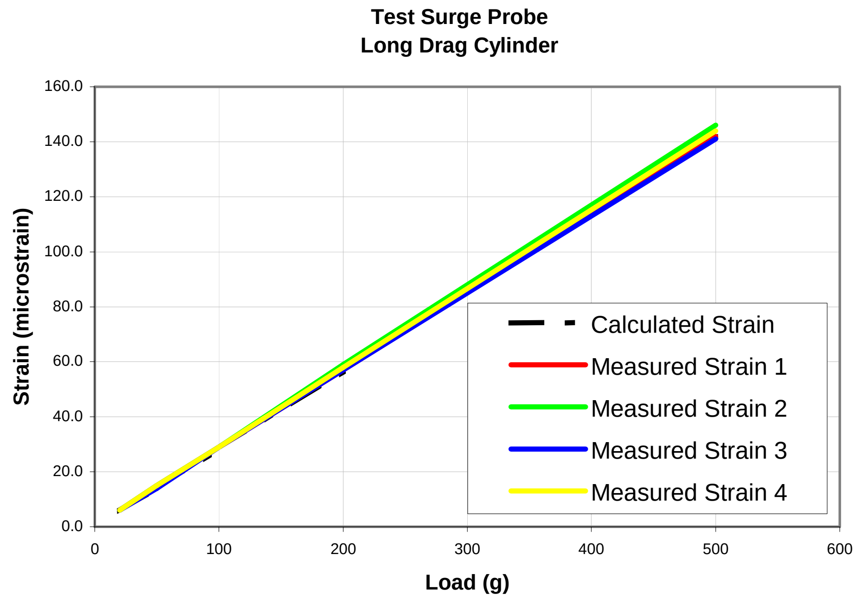


Figure 3-3. Bench Test Results Showing Strain and a Function of Load for All 4 Drag Probe Sensitivity Directions for the Long Probe

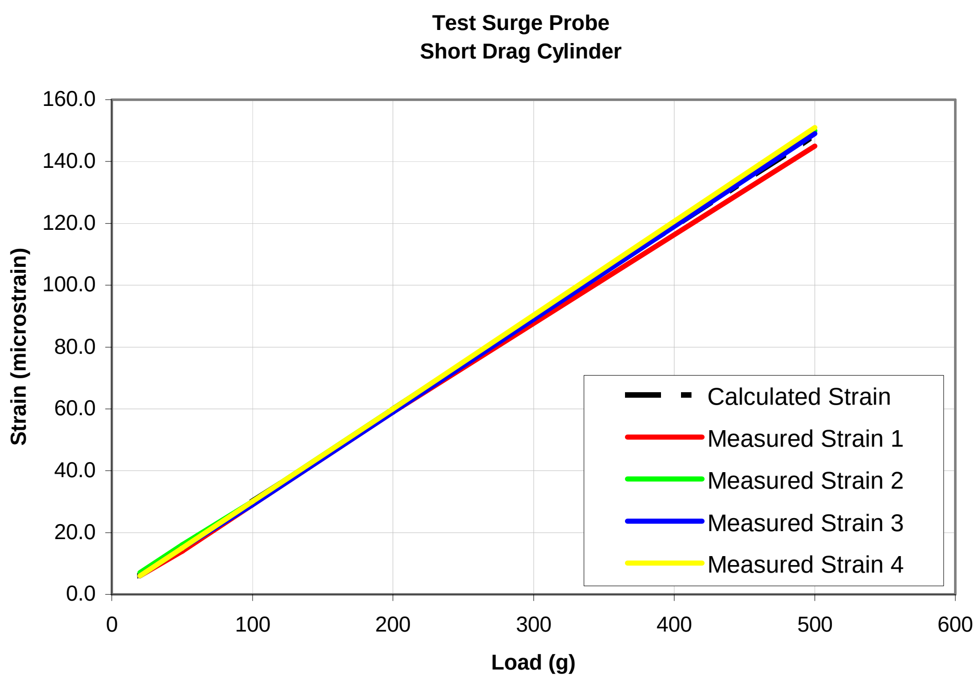


Figure 3-4. Bench Test Results Showing Strain and a Function of Load for All 4 Drag Probe Sensitivity Directions for the Short Probe

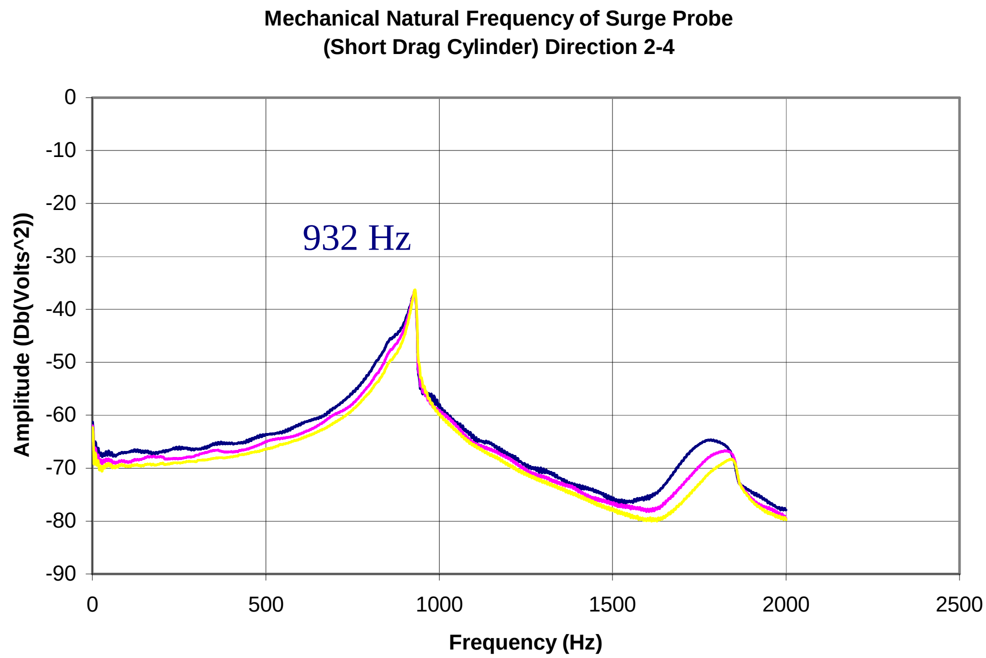


Figure 3-5. Results of Bump Test of Short Laboratory Drag Probe Showing the Mechanical Natural Frequency of the Probe

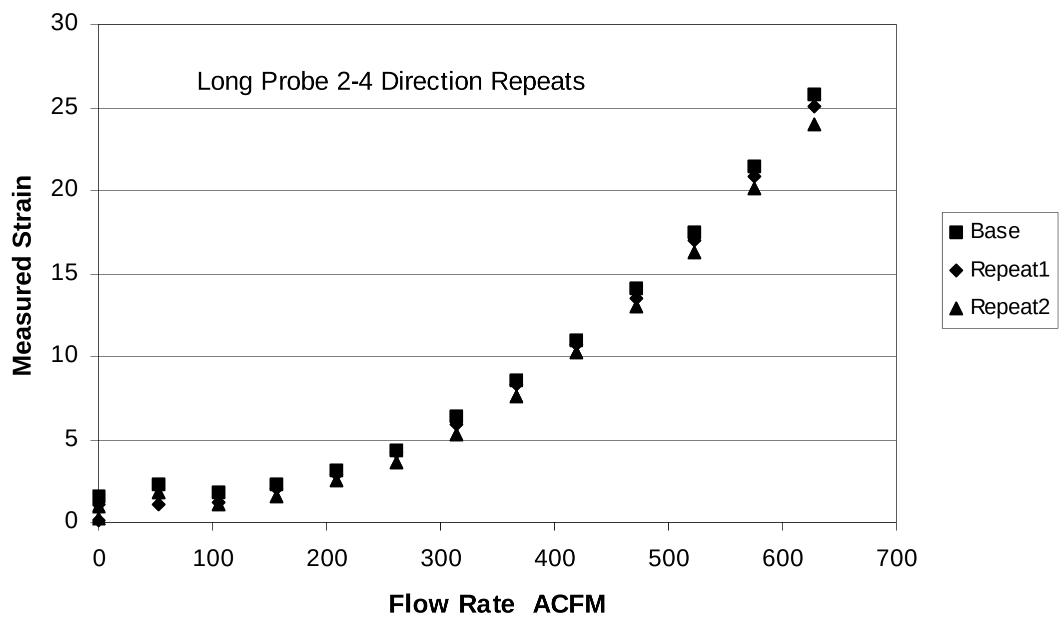


Figure 3-6. Strain as a Function of Flow Rate During Repeated Flow Facility Tests of the Long Laboratory Drag Probe in the 2-4 Direction

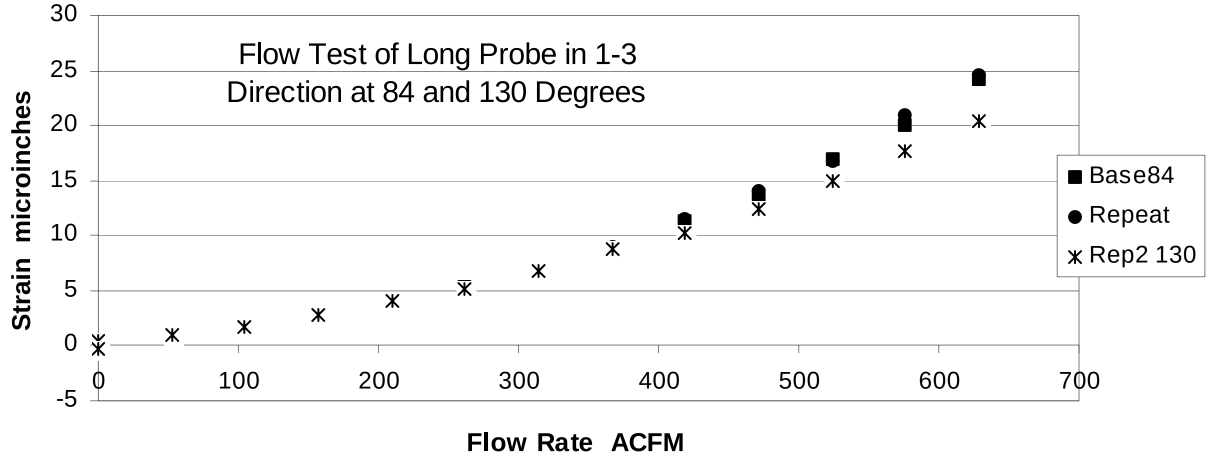


Figure 3-7. Strain as a Function of Flow Rate During Repeated Flow Facility Tests of the Long Laboratory Drag Probe in the 1-3 Direction at Different Temperatures

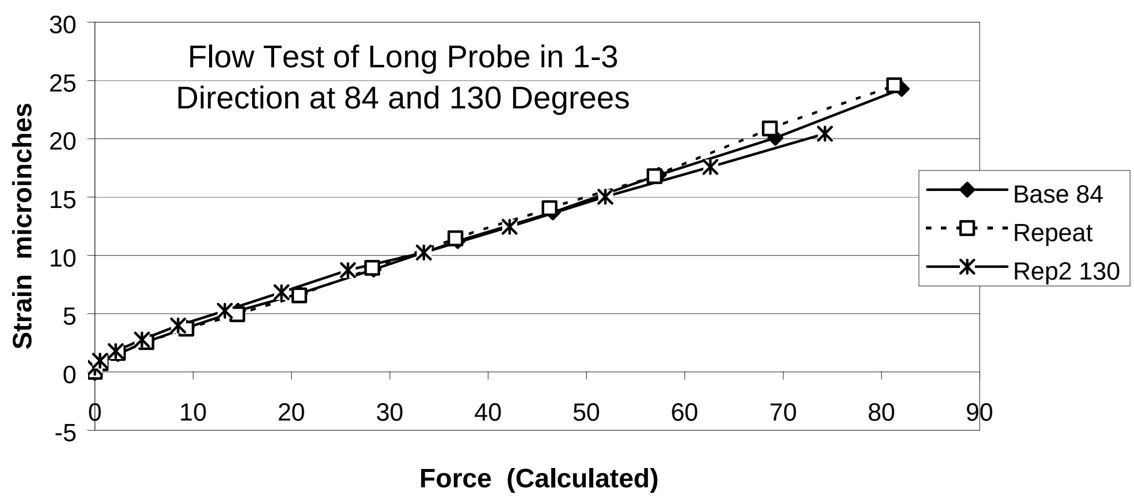


Figure 3-8. Strain as a Function of Calculated Drag Force During Repeated Flow Tests of the Long Laboratory Drag Probe in the 1-3 Direction

A test of the short laboratory probe in the 2-4 strain gauge direction showed an important feature of the drag probes; they are equally sensitive in the reverse flow direction as in the forward flow direction. Figure 3-9 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 directions. When corrected for the applied flow force, the strains from the two tests shown in Figure 3-9 are essentially identical as a function of force as shown in Figure 3-10. The resulting strain sensitivity per unit flow force constitutes a calibration of the probes.

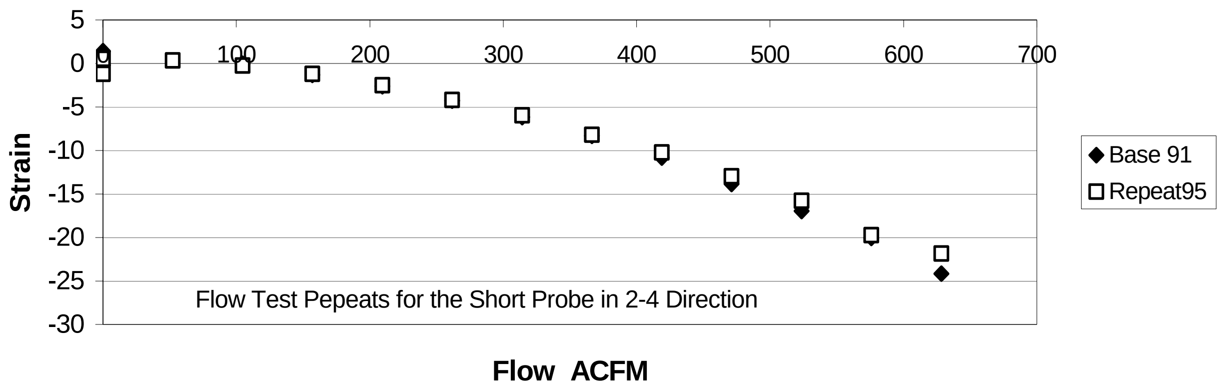


Figure 3-9. Strain as a Function of Flow Rate During Repeated Flow Tests of the Short Laboratory Drag Probe in the 2-4 (Negative) Direction

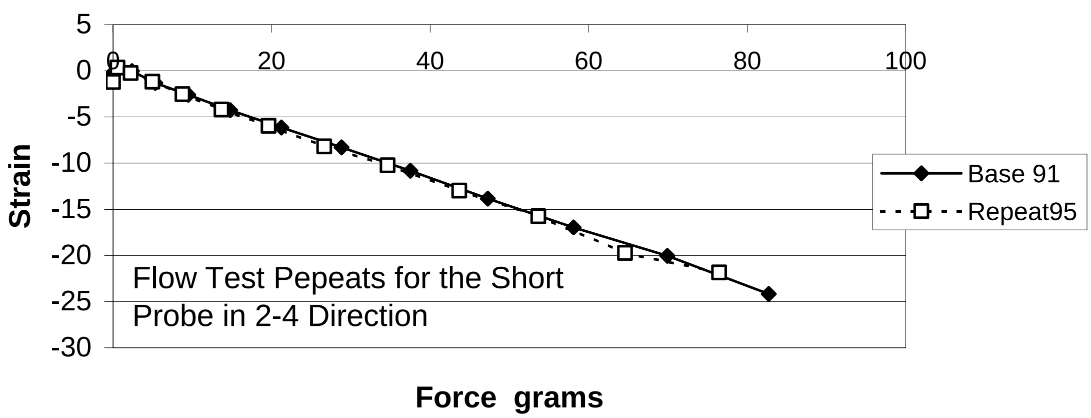


Figure 3-10. Strain as a Function of Calculated Drag Force During Repeated Flow Tests of the Short Laboratory Drag Probe in the 2-4 (Negative) Direction

3.3 LABORATORY TESTS

3.3.1 Two Dimension Test Findings

Once the experimental surge detection probes were bench and flow tested, they were ready for installation in a laboratory compressor. Since the probe calibration and flow test results agreed with design predictions, this type of calibration and particularly the flow testing will not be necessary in the future. Direct surge detection probes can be designed, fabricated, and the functionality and sensitivity checked, and then they can be installed in a compressor and used.

The first laboratory compressor was an older machine with an old style 2D impeller. This machine was typical of pipeline compressors installed before the 1980s. The prototype surge detection probes were installed immediately in front of the impeller inlet.

The result of this series of surge detection test with the drag probes in the SwRI laboratory compressor showed 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 2D nature of the impeller, the narrow inlet channel geometry and the restricted location of the drag probes, the changes are not a clear re-circulation, including a reversal of the axial strain that was expected.

3.3.2 Three Dimension Test Findings

The second laboratory compressor used for this testing was smaller in physical size than the first unit but operated at somewhat higher speeds with a higher flow rate and a similar horsepower. The second laboratory compressor was of a modern design with a 3D impeller and a more open and larger inlet passage that is similar to current pipeline compressors. The results of a low-speed test showing the axial and tangential strain with an indication of the decreasing flow is shown in Figure 3-11. During this test, as flow decreased towards the surge region, the axial strain increased at first and then rapidly decreased and became negative as the tangential strain increased from near zero to significantly positive values. Both the axial and tangential strain levels started to change less rapidly just before surge was reached.

Data from the second test with this modern compressor at a moderate speed showed that the axial strain followed the decreasing flow at first, then increased to a local maximum, and finally dropped rapidly to a negative value as flow was reduced towards surge. The tangential strain during the same approach to surge remained near zero in the normal compressor flow range and then increased significantly as the low flow near surge condition was approached. This second laboratory test result included in Figure 3-12 showed the same pattern of axial and tangential strain changes at a compressor speed of 16,000 rpm as seen in the first test at a lower speed. This test took place during a rapid approach to surge that occurred within a two-minute time frame. The results of a similar test at a higher speed are shown in Figure 3-13, where the increase in axial strain before the rapid decrease was clearly indicated and the change in the tangential strain was also clearly observed. These same changes in strain as flow decreased

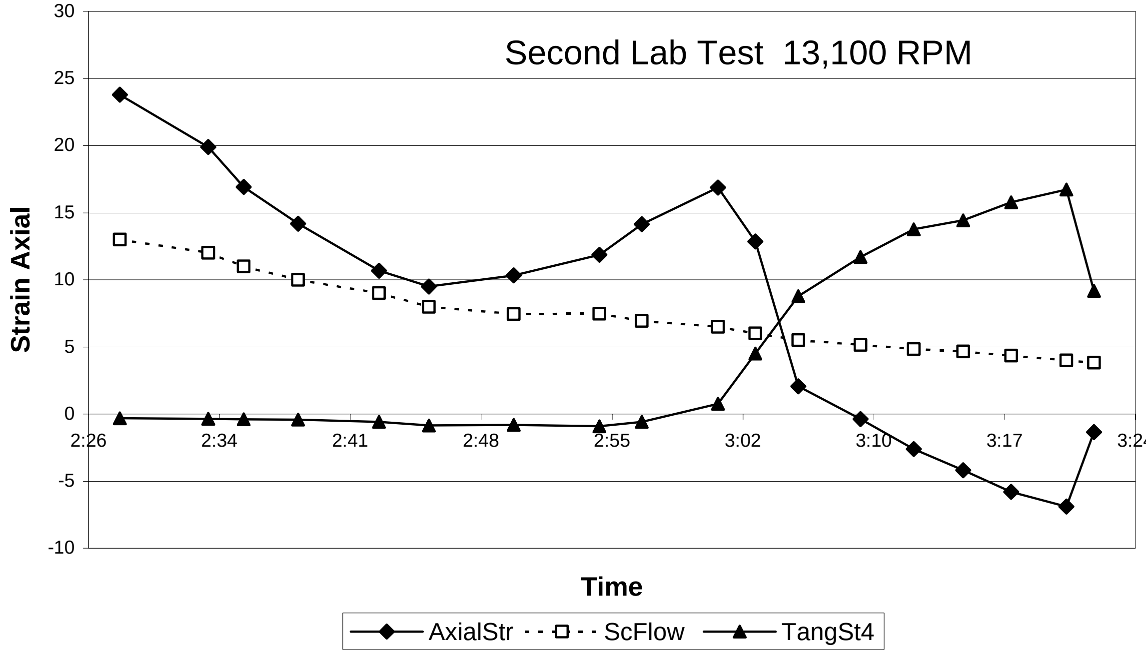


Figure 3-11. Axial and Tangential Strain Changes with Scaled Flow for Low-Speed Operation Leading Up to Surge

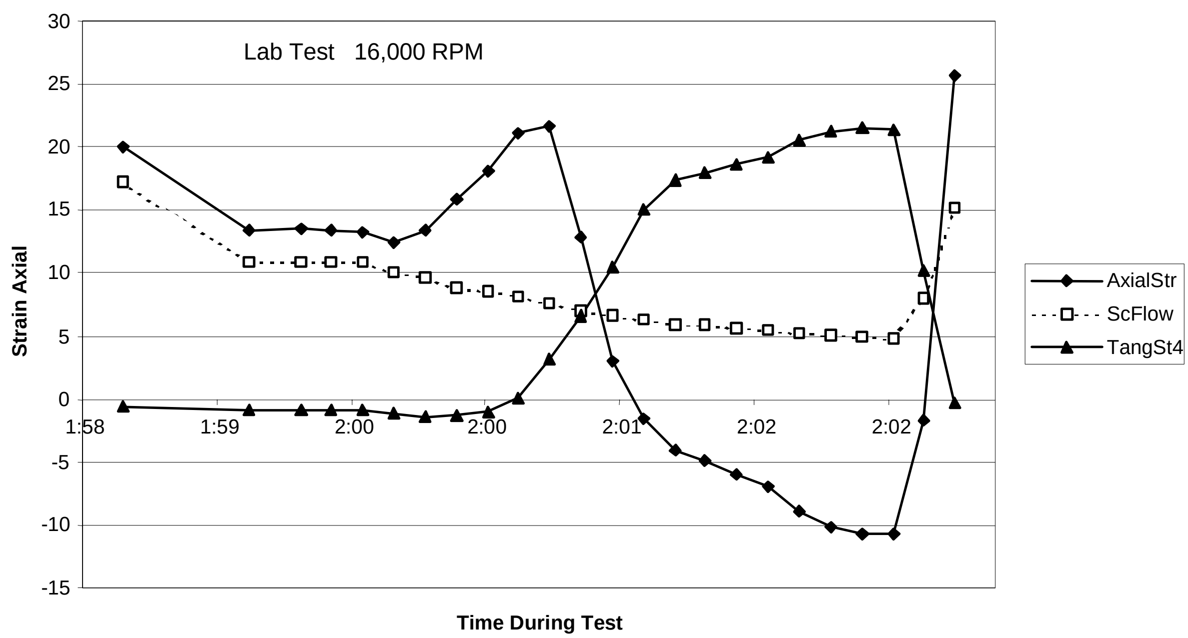


Figure 3-12. Axial and Tangential Strain Changes as Scaled Flow Decreased Towards Surge in a Modern 3D Laboratory Compressor at Moderate Speed

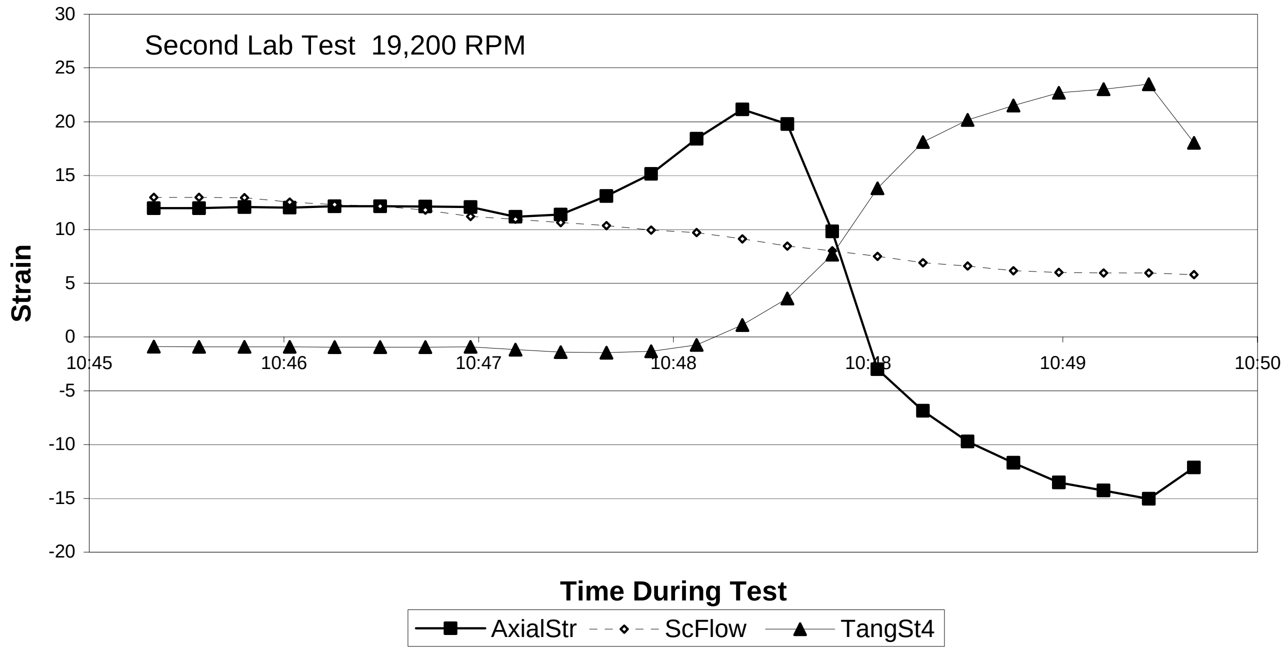


Figure 3-13. Axial and Tangential Strain Changes with Scaled Flow for High-Speed Operation Leading Up to Surge

occurred either slowly with pauses at stable flow conditions or continuously over a short (two-minute) time frame. The changes in strain signals at the highest speeds tested are remarkably similar in characteristic to the other speed results as surge approached. A signal as repeatable as observed during this laboratory testing can be used as an improved surge control.

The flow coefficient, which is the ratio of the inlet volumetric flow to the impeller speed, is a measure of the capacity of the machine per unit speed. Specifically, for this report, the flow coefficient is the inlet volumetric flow rate divided by the compressor speed and the cube of the inlet radius in order to obtain a non-dimensional number. At the design flow and speed, the flow coefficient indicates the size of the compressor. At off-design flow for a particular compressor, the flow coefficient indicates the amount of flow per unit speed by which the operating capacity of the machine has been reduced.

In this experimental data, the flow coefficient appeared to control the conditions at which the surge precursor and surge occurred, as shown in Figure 3-14, which is a plot of the axial strain signal as a function of flow coefficient for three different test speeds. As flow decreased towards surge, the flow coefficient decreased and the axial strain signal moved to the left. While the strain is positive, it is safe to operate the compressor in that condition; however, as the strain level dropped rapidly and became negative, further flow reduction should be prevented in order to avoid surge.

The results for this laboratory test of pre-surge detection are shown on a compressor map in Figure 3-15, where an estimated surge line provided by the manufacturer is included. Using the surge detection probe and well controlled laboratory condition, a significant increase in

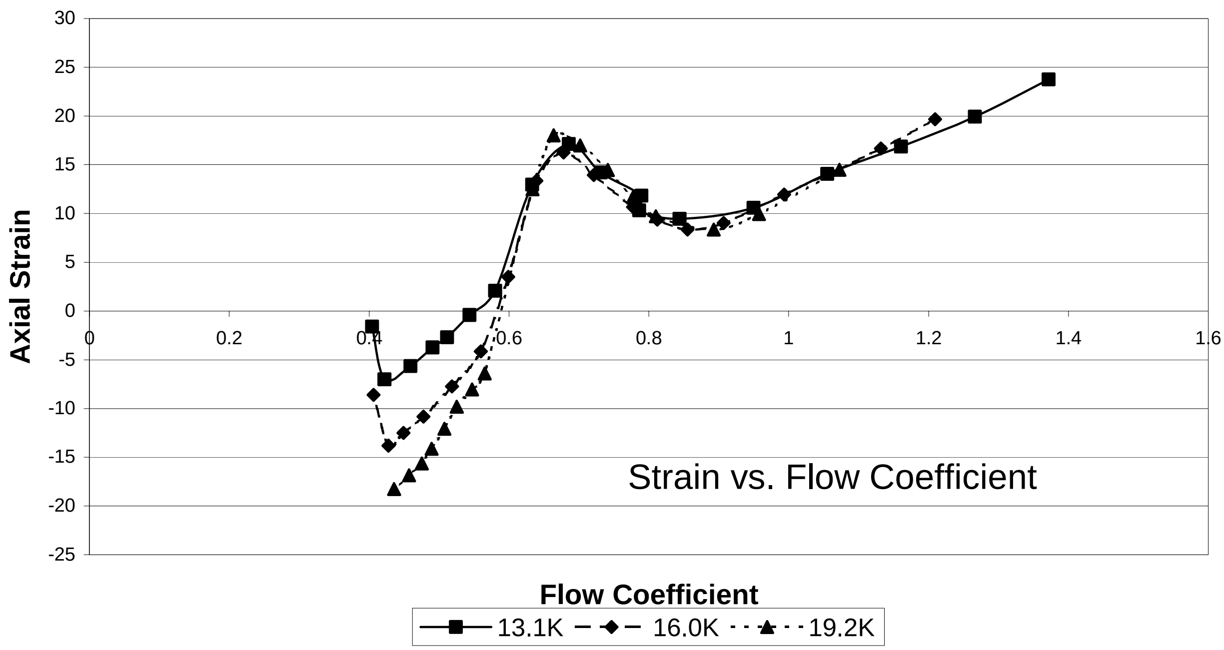


Figure 3-14. Plot of Axial Strain Changes as a Function of Flow Coefficient for Three Different Tests

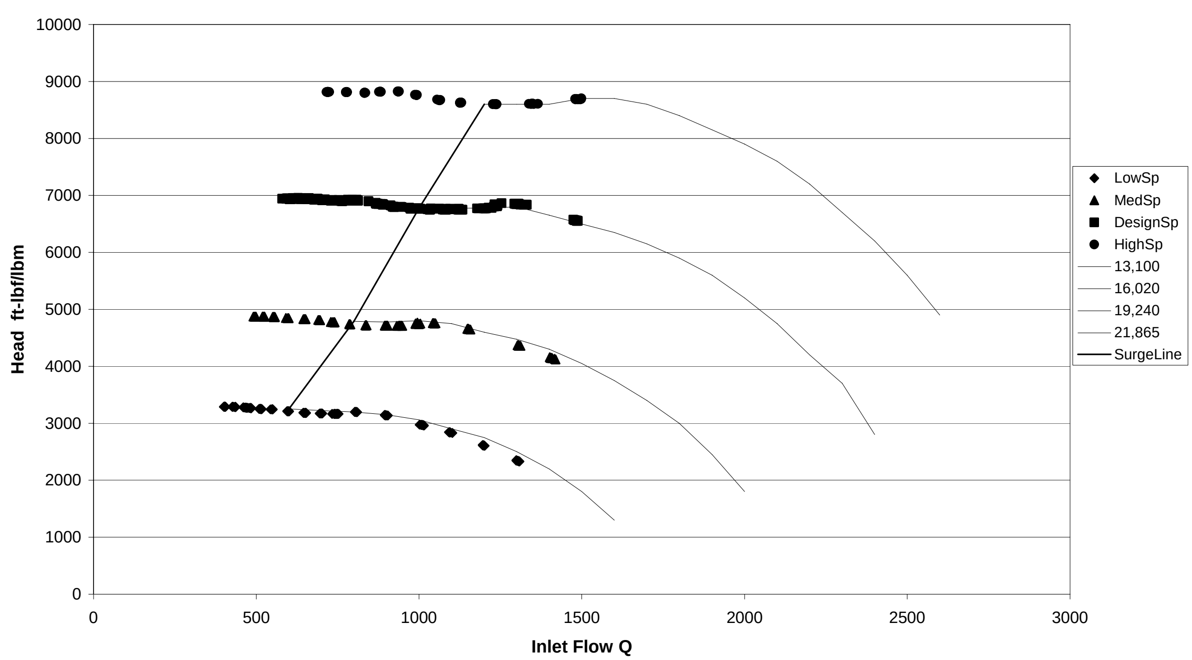


Figure 3-15. Compressor Map for the Second Laboratory Compressor Showing the Operating Range According to the Conventional Surge Line and Using a Surge Probe

operating range was achieved. The term turndown is the percent of the design flow rate by which the flow through a centrifugal compressor can be reduced before surge is reached. The turndown for the second laboratory compressor based on the manufacturer [Ref. 6] estimates was 47%. The turndown based on the surge detection probes was approximately 68%, which is a 21% increase in the operating flow range. Although this is a very large increase in operating range, the increase for pipeline compressors where the test conditions are not as stable and the probe signal may not be as strong may not be as large.

Returning to the strain data as a function of flow coefficient, the shape of the axial strain curves seen in these tests is dependent on the position of the probe relative to the impeller inlet. A series of diagrams showing flow velocity vectors in the area of the drag probe at various flow levels are shown in Figures 3-16 through 3-19. At a normal flow rate, the velocity vectors passed over the drag probe while traveling towards the impeller and the strain was positive. At the first low-flow condition, a small re-circulation zone developed close to the impeller and the outer wall, but did not reach the drag probe. In the area of the drag probe at this low-flow condition, the velocity that was approaching the re-circulation must accelerate to pass over the small re-circulating flow cell, and it is this accelerated flow acting on the drag probe that increased the axial strain, even when the overall flow was decreasing. As the flow was lowered further to the near-surge condition, the re-circulation cell reached the drag probe, the rapid drop in axial strain occurred, and the increase in tangential strain started because it was the re-circulating flow cell that contained the three-dimensional flow patterns, including the tangential flow component. Finally, just before surge, the re-circulating flow cell became so large that flow was acting in the reverse direction on the drag probe (if it was close enough to the impeller), so that the axial strain was negative and the tangential strain was large. At this large re-circulating flow condition, surge was imminent.

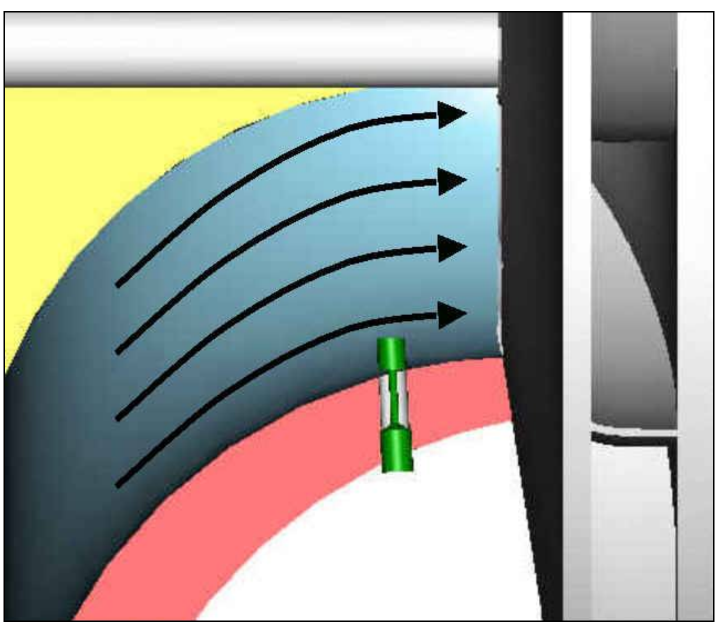


Figure 3-16. Diagram of Flow Vectors at a Normal Flow Condition with the Drag Probe Location Shown

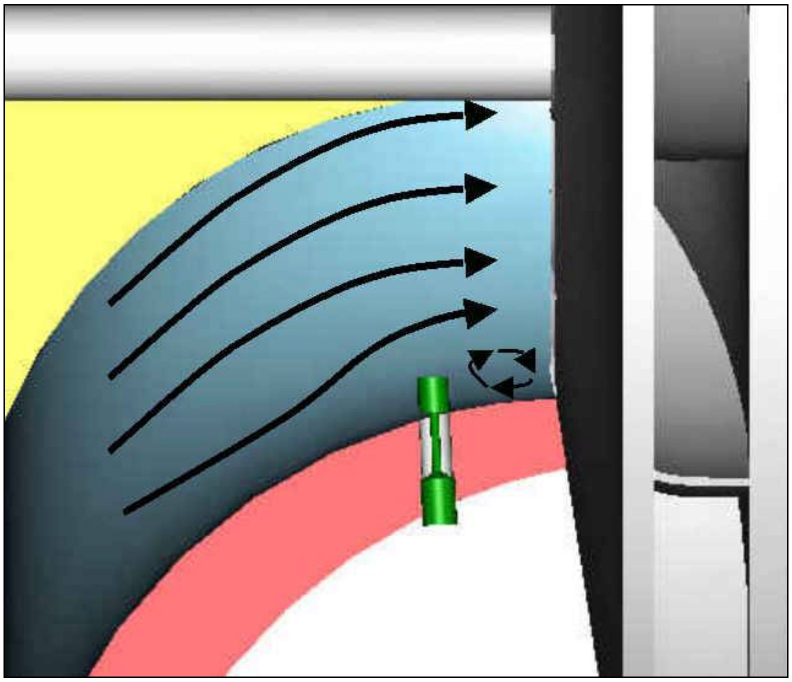


Figure 3-17. Diagram of Flow Vectors at a Low-Flow Condition Showing the First Re-circulation

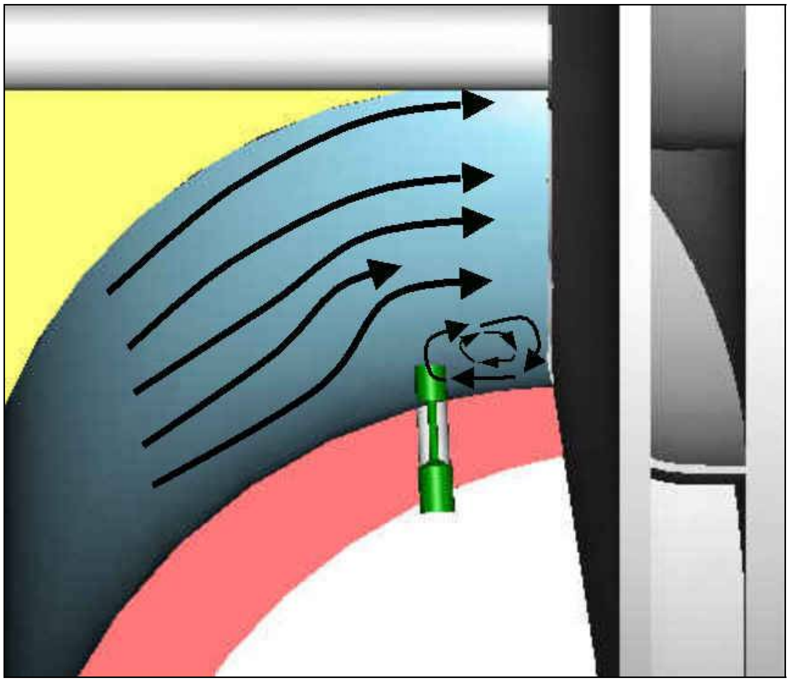


Figure 3-18. Diagram of Flow Vectors at a Near-Surge Condition Showing Re-circulation Just at the Drag Probe

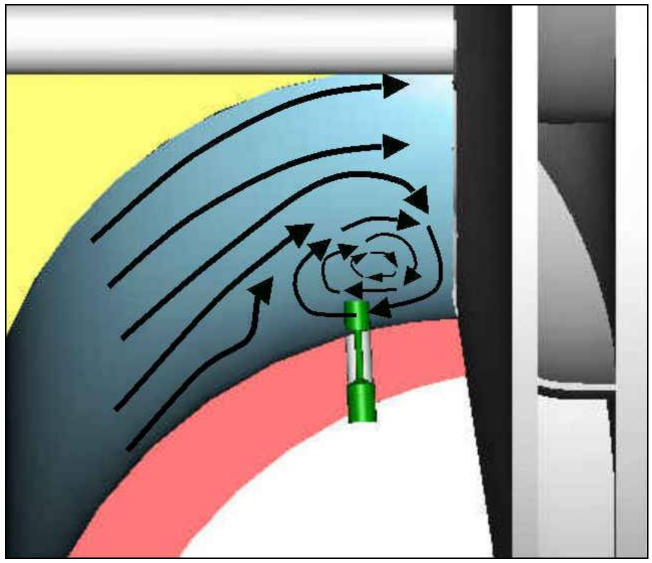


Figure 3-19. Diagram of Flow Vectors at a Surge Flow Condition Showing a Large Re-circulation Zone

3.4 FIELD TEST RESULTS

3.4.1 Probe Installation Experience

One of the lessons learned during this research effort was that installation of drag probes near the impeller inlet eye of a compressor, with secure provisions for the wiring, was not a trivial task. As will be discussed in more detail, the location of the probe in the impeller inlet flow was a critical factor for successful detection of the pre-surge flow re-circulation. Each laboratory installation of a drag probe was successful in terms of suitable locations and wiring reliability. The first field installation, however, resulted in a loss of conductivity in the wiring due to excessive compression of the wires and due to failure of the wire bundle at an unprotected location in the gas flow. The strain gauge wires from the probe to the exterior of the compressor case should be routed through conduit tubing to ensure their protection. At the compressor pressure boundary, sufficiently heavy solid core wires should be used through a fully capable, multi-wire compressor or pressure boundary fitting, such as a CONAX or similar device. It is necessary when installing a surge detection probe to ensure that the location is as close as possible to the face of the impeller at the outer diameter and that the wires are secure, protected, and reliable.

One requirement for the strain gauge circuits in the surge detection application is that they be stable and that the zero flow output be consistent over time. After the surge detection probe was installed in the final field test compressor, an evaluation of the zero drift of the surge detection system was conducted during a period of normal operation. Over a number of days of operation, the output of axial and tangential strain was recorded at random times when the compressor was operating or shutdown. Figure 3-20 shows a plot of the axial and tangential strains recorded as a function of time. When the compressor was in operation, the axial strain was positive and generally larger in magnitude for higher flows. The tangential strain was negative when the compressor was in operation. When the compressor was not in operation, both the axial and tangential strains were near zero. That is, without a flow-induced strain on the surge probe, the strain circuit returned to a balanced condition. There was a small amount of drift (±0.1 volts) in the zero flow strain signals as shown by Figure 3-21, which is a plot of mostly zero flow readings during the period of the check. The variation of the zero flow output was small compared to the magnitude of the flow-induced strain, including the pre-surge signals. The strain output, although not absolutely accurate, was sufficiently stable to indicate the operating flow level, the shutdown conditions, and the unique pre-surge condition.

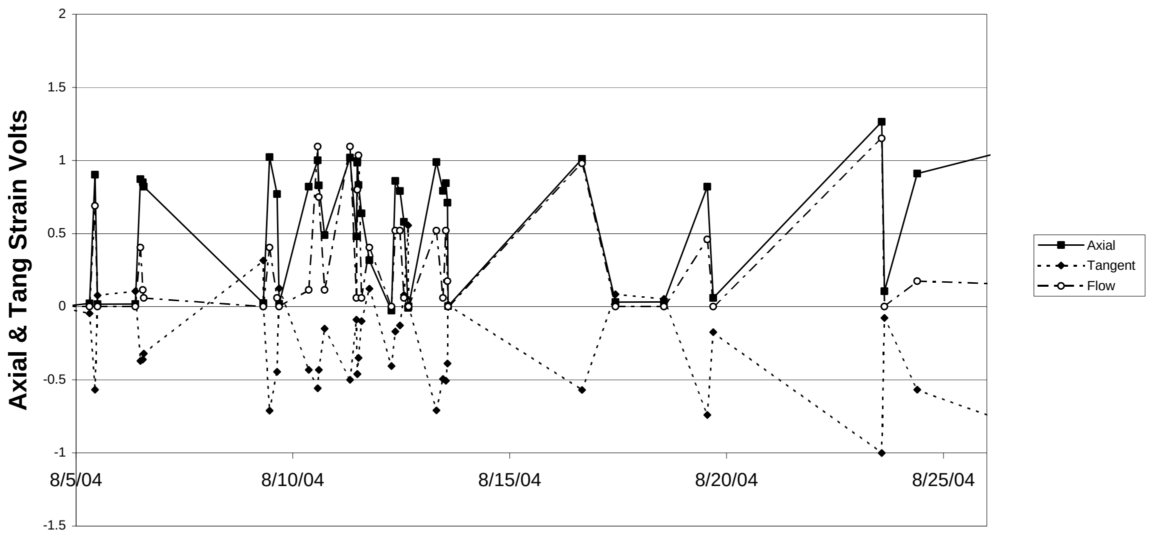


Figure 3-20. Plot of Axial and Tangential Strain Amplifier Output During a Period With and Without Compressor Operations

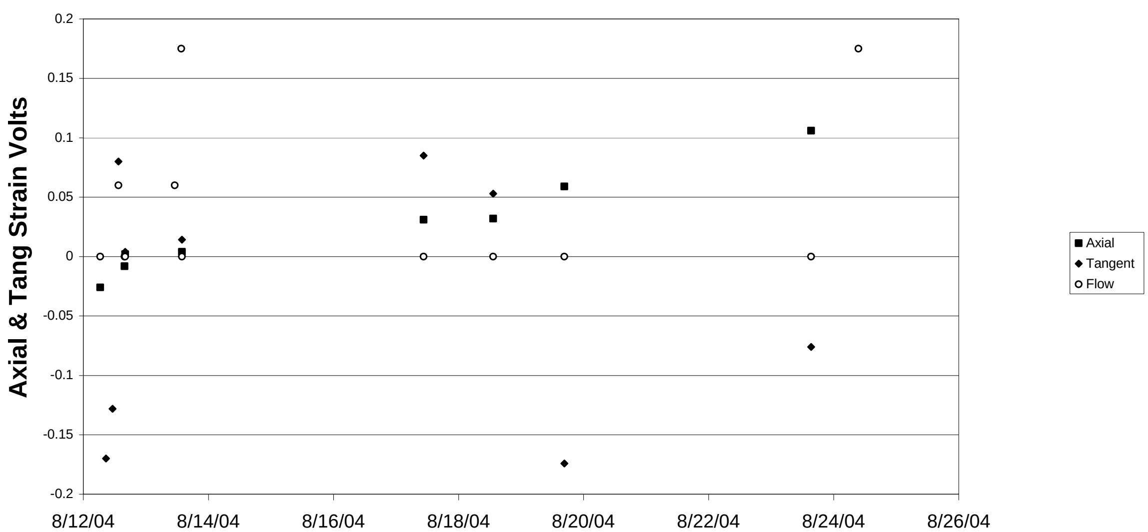


Figure 3-21. Axial and Tangential Strain Amplifier Output During a Period With Mostly Zero Flow Conditions

If the surge probe strain gauge circuit is set to automatically rebalance whenever the compressor is shut down and idle for more than a few minutes, then that would be helpful for long-term stability, accuracy, and reliability of pre-surge detection. It is recommended that future surge probe amplifiers be equipped with auto balance features. It should be clear from the preceding and following data that a pre-surge condition is the unique situation in which the axial strain approaches zero or negative values and the tangential strain becomes zero or positive, while the compressor is in operation.

3.4.2 Field Test of Pre-Surge Detection

The first field compressor was an old style pipeline compressor with a 2-dimensional (2D) impeller. The results from this field test did not show the inlet re-circulation that indicates pre-surge conditions. The first field test did show that the compressor could be operated at a lower flow rate than previously considered the surge control limit without reaching surge. The old centrifugal compressor used for the first field test has a wider operating range than previously used. This first field test also showed that the drag probes are rugged and reliable.

Testing of a direct surge control system has been conducted in the second field compressor, which was a modern three-dimensional (3D) natural gas pipeline style unit. The purpose of this test was to prove pre-surge detection capability, refine the surge controller that was developed, and test surge control algorithms. Some of the results of axial and tangential strain as flow was reduced towards surge are shown in Figures 3-22 and 3-23. In this data, flow decreased as shown by the scaled flow line, axial strain decreased at the same rate as flow, and tangential strain increased during the normal part of the compressor operation. As the flow reached a lower value-approaching surge, the axial strain increased through a local maximum and then decreased slowly while the tangential strain started to increase rapidly. These changes in the flow induced strain, as surge was approached, resulted from the developing re-circulation even before the flow changes reached the probe location and from the tangential components of the flow before surge occurred.

A third test of the strain signals from the direct surge probe is shown in Figure 3-24. The flow rate in Figure 3-24 is indicated by the differential pressure in inches of water while both the axial and tangential strain signals are shown on the same scale in units of micro-strain (micro-inches per inch). The changes in strain signals in Figure 3-24 with the axial strain decreasing with flow, experiencing a local peak, and then slowly decreasing, and the tangential strain slowly increasing until it rapidly increases as surge approaches was the same as in Figures 3-22 and 3-23. The strain signals in these three field test results are slightly different from the strain signals in Figures 3-11, 3-12, and 3-13, which are from the laboratory compressor. The installation of the drag probe in the natural gas compressor, where the probe was overhung from the inlet wall, was somewhat different than in the laboratory compressor where the probe penetrates the inlet wall perpendicular to the flow. There appears to be small differences in the resulting pattern of the strain signals that is dependent on the details of the probe and compressor geometry.

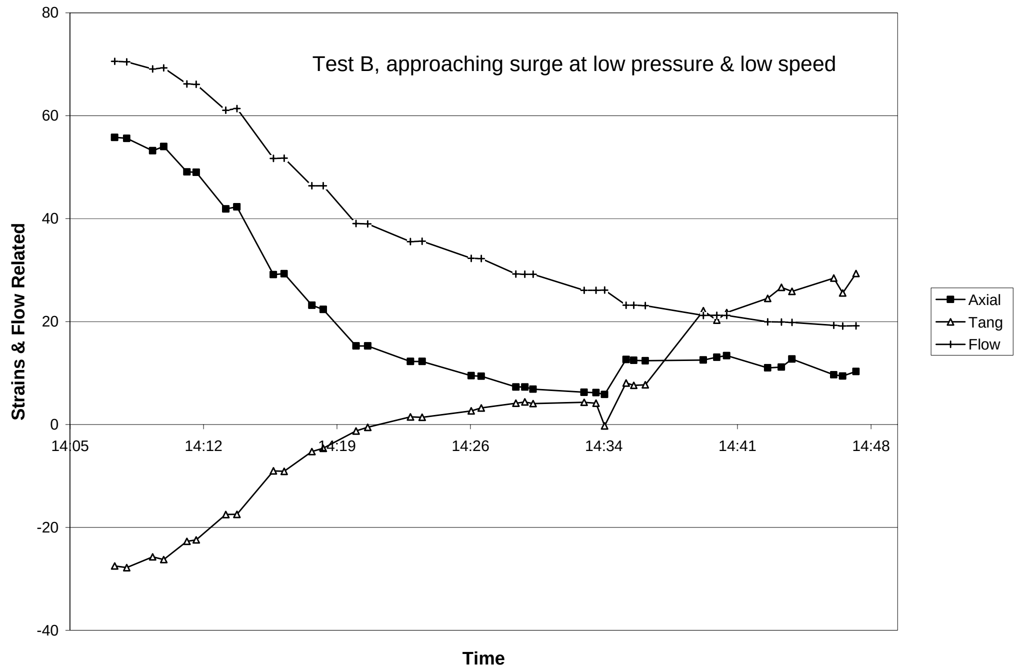


Figure 3-22. Axial and Tangential Strain Changes as Scaled Flow Decreases Towards Surge in a Modern 3D Pipeline Style Compressor, Test B

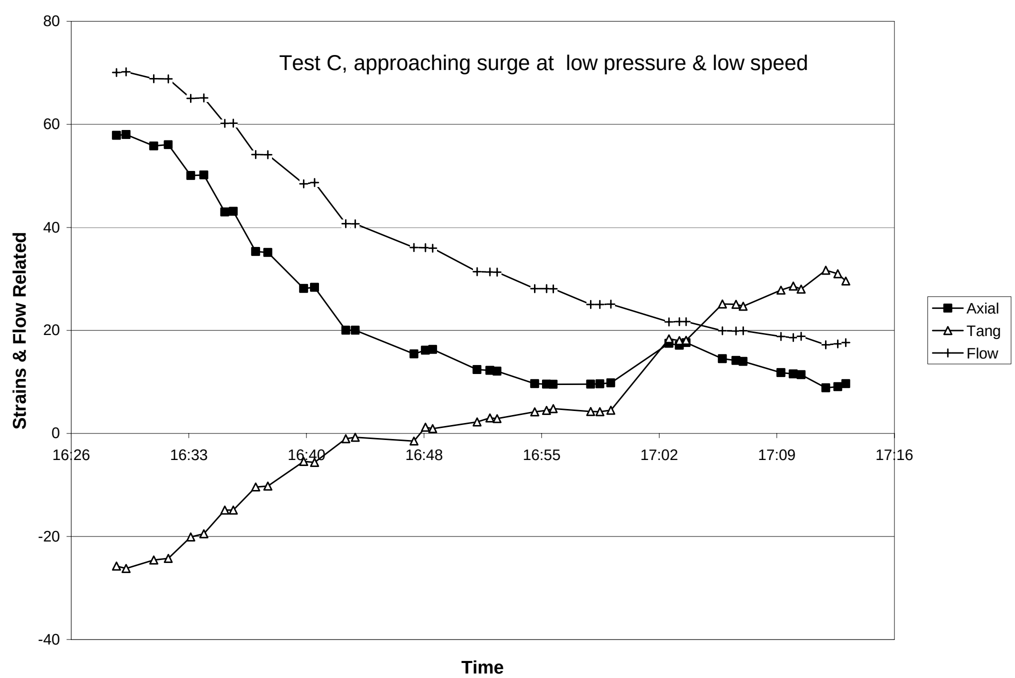


Figure 3-23. Axial and Tangential Strain Changes as Scaled Flow Decreases Towards Surge in a Modern 3D Pipeline Style Compressor, Test C

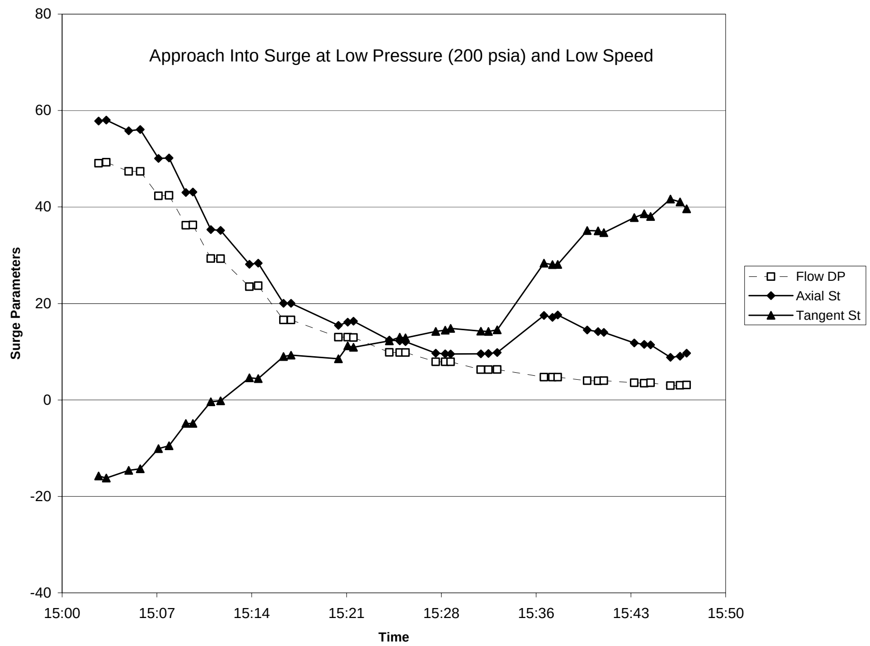


Figure 3-24. Axial and Tangential Strain Changes with Flow DP as Surge is Approached in a Modern 3D Pipeline Style Compressor, Test E

There are many approaches to tracking the relative magnitudes of the axial and tangential strains to determine if surge is approaching, including identifying the rapid decrease in axial strain and the increase in tangential strain, monitoring the slopes or derivatives of these signals, or other combinations of strain changes. The development of an algorithm for identifying approaching surge was developed by listing several possible approaches and considering the advantages and disadvantages of each. With the dependence of the strain signal details on the probe location and alignment in the compressor geometry as shown by the two modern compressor test results, a method that is reliable despite small changes is required. If the difference between the two strain signals, that is, the axial minus the tangential, is taken, then the dependence on particular features of one or the other signal is reduced. During the final field testing, the stability of the strain signals had been sufficient so that a comparison of the values was reliable. Therefore, the difference between the axial wall tangential strain signals is not the only possible method, but it is the method chosen for the current work. If the drop or zero crossing of the axial strain was not significant as a pre-surge indication, then the increase in the tangential component could be used to emphasize the drop in the axial signals. Each compressor can be expected to be slightly different, but with the proper algorithm, the approach of surge will be identifiable. Figure 3-25 shows the difference in the strain signals for each of the six tests in Figures 3-11, 3-12, 3-13, 3-22, 3-23 and 3-24 as a function of a scaled flow rate. The low positive values, zero crossing, and negative values of these signals could easily be used to detect and control the approach of surge. Figure 3-25 is the implementation of the selected difference between axial and tangential strain algorithm, and some selected small, positive, zero, or slightly negative value in Figure 3-25 could be used as a surge control line with a direct surge control system.

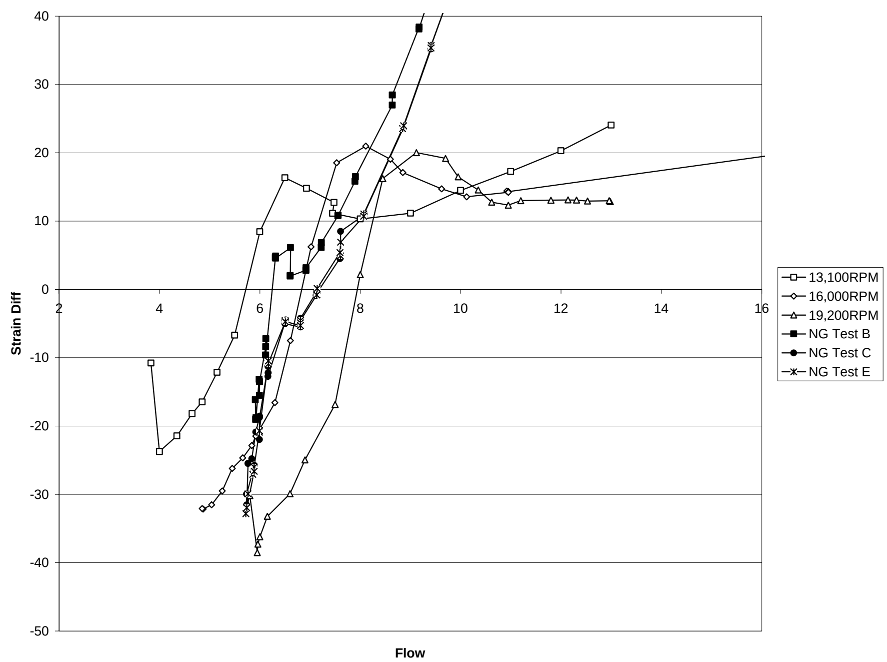


Figure 3-25. Plots of the Axial Minus Tangential Strain Difference for Six Different Tests of Modern Compressors Approaching Surge Showing a Negative Difference Before Surge

3.5 OPERATIONAL TEST RESULTS

3.5.1 Strain Signals Under Different Conditions

Typical plots of the axial and tangential strain as surge was approached in the field compressor, under different operating conditions, are shown in Figures 3-26 through 3-28. The first of these strain plots was recorded during a rapid change from a near surge to an incipient surge condition. Figure 3-26 showed that the axial strain that was already low decreased and that the tangential strain that was already above zero increased as flow rate decreased from a low near surge level to essentially the surge limit at the operating pressure and speed. At the last data point in Figure 3-26, the compressor experienced a surge event at a suction pressure of approximately 200 psia. Figure 3-27 is the second of the strain plots and was recorded at a higher suction pressure of approximately 350 psia and a lower speed of approximately 16,000 rpm compared to Figures 3-22 through 3-24. The flow rate in Figure 3-27 was slowly decreased, but held above the surge level for more than two minutes, and then rapidly decreased for a short period of time. Prior to the decrease in flow in Figure 3-27, the tangential strain was low and the axial strain increased towards a local maximum and became variable. As soon as the flow rate decreased, the axial strain decreased and the tangential strain increased showing the expected indication of surge in which the difference between the axial and tangential strain became negative. The flow rate in Figure 3-27 was immediately increased, by opening the recycle valve, as soon as the definitive indication of surge was observed, and the compressor did not experience a surge event. Figure 3-28 is a strain plot from a high-pressure, high-speed

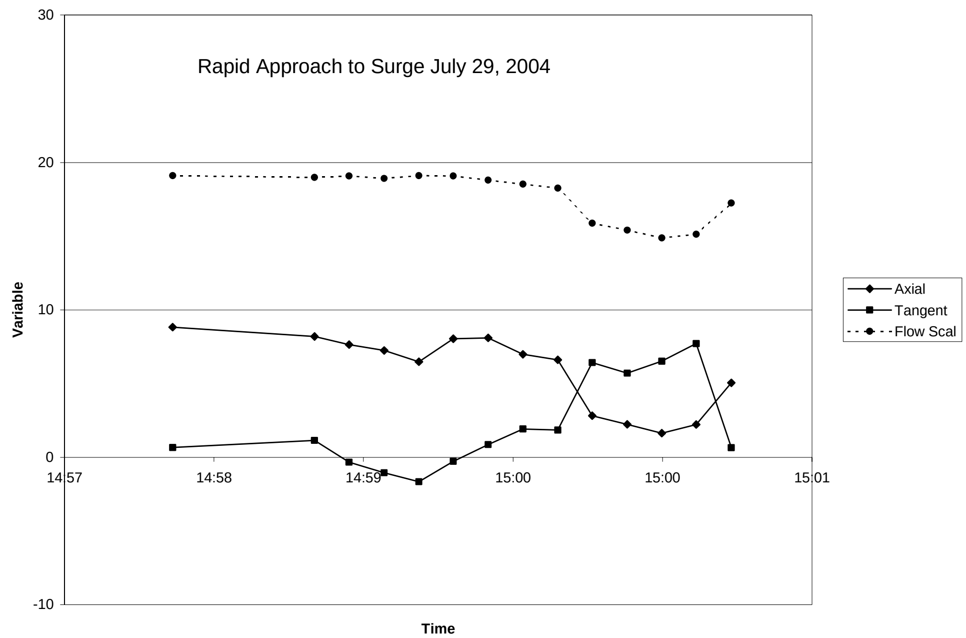


Figure 3-26. Axial and Tangential Strain Changes as Scaled Flow Rate Changed During a Rapid Approach to Surge at a Low Pressure Condition

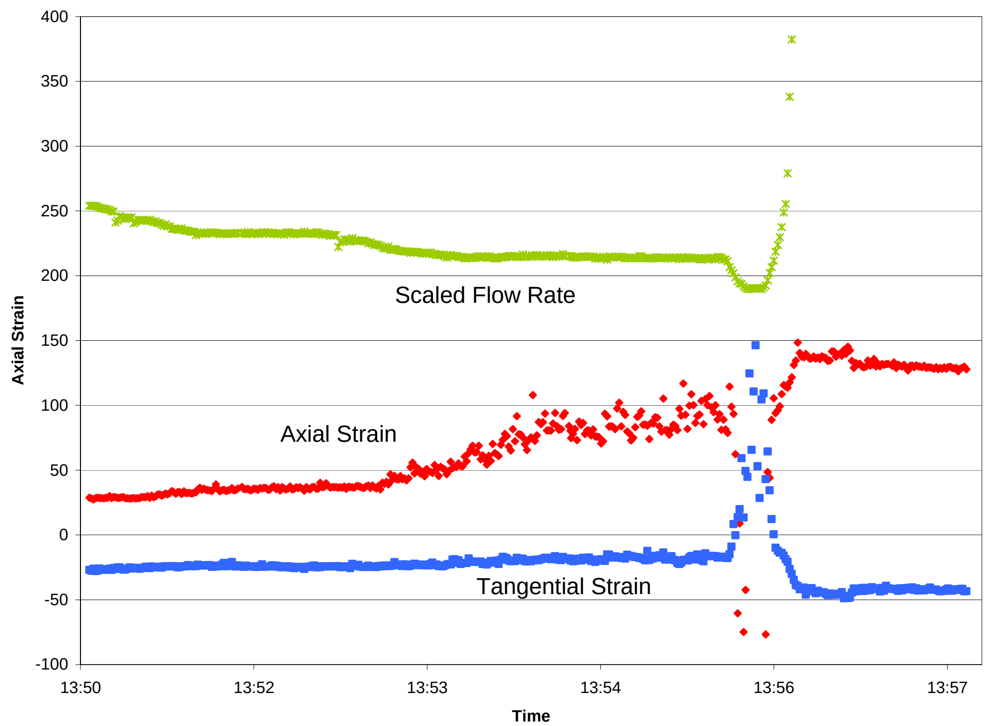


Figure 3-27. Axial and Tangential Strain Changes as Scaled Flow Rate Changed During a Moderate Pressure Approach to Surge

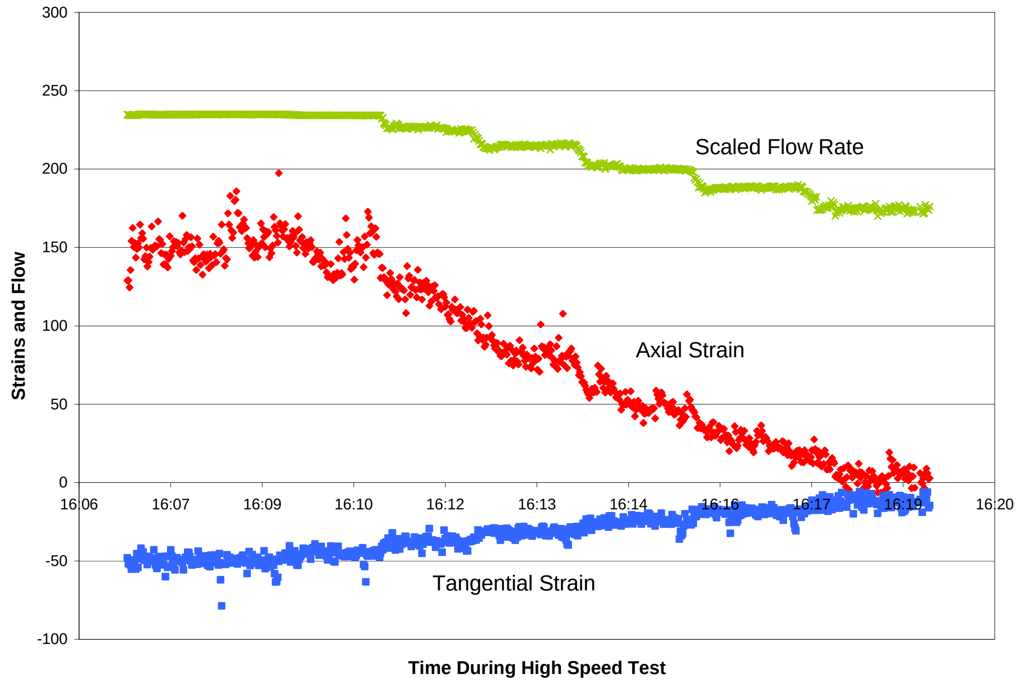


Figure 3-28. Axial and Tangential Strain Changes as Scaled Flow Rate Changed During a High-Pressure, High-Speed Approach to Surge

operational test with a suction pressure of approximately 610 psia and a compressor speed of approximately 19,000 rpm. As the flow rate was decreased, as shown by the scaled flow signal in Figure 3-27, the axial strain decreased and the tangential strain increased. Although the axial strain in Figure 3-28 did reach the level of the tangential strain, the axial strain did not pass through a local maximum or rapidly decrease, the difference between axial and tangential strain did not become negative, and the compressor did not experience a surge condition. Figure 3-28 is the high-pressure, high-speed test data that will be referred to later in the compressor performance evaluation.

The three plots discussed in this section were for operational tests of the field compressor and were for different conditions than for the previous plots but did show the same trends. In every case, the axial strain decreased as flow rate decreased towards surge, demonstrated a local maximum just before surge, and became nearly zero or negative at the surge limit. At the same conditions, the tangential strain increased from a negative value to near zero and to positive values as surge was approached. When the axial and tangential strains crossed or their difference became negative, it was a clear indication that surge was approaching. One additional plot of strain data as a function of flow rate is shown in Figure 3-29. The difference between the axial and tangential strain signals became zero or negative when the compressor was close to surge and this was a fundamental and reliable indication that surge was approaching. Figure 3-29 emphasized the conclusion drawn previously that if the axial and tangential strains both approach zero and the compressor is operating, then it is an indication that the compressor is close to surge. In implementation of the direct surge control technology, a further reduction in

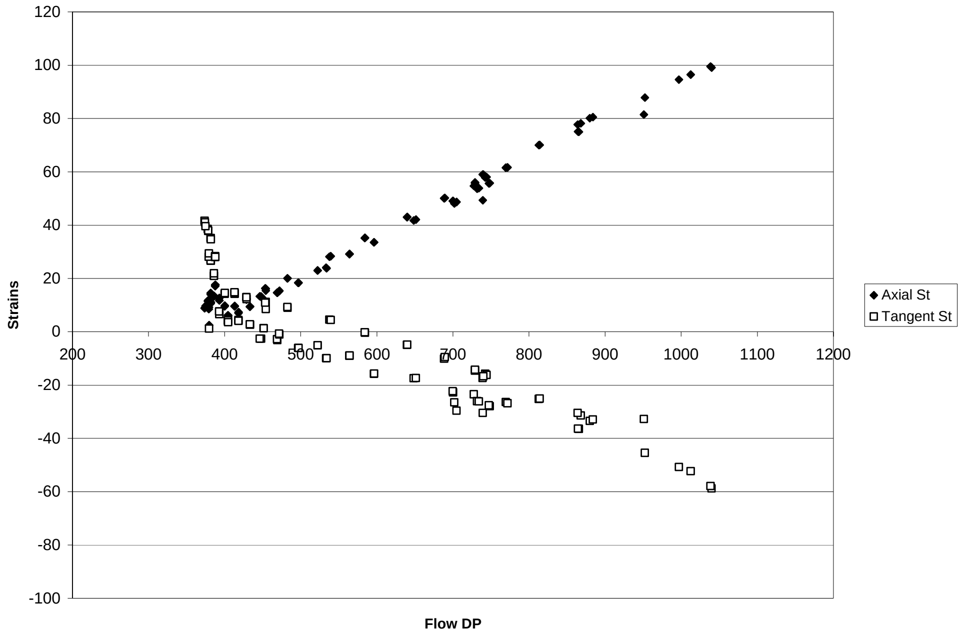


Figure 3-29. Axial and Tangential Strain vs. Approximate Inlet Volume Flow Rate for a Range of Operating Conditions

compressor flow should be prevented through the use of a recycle valve or other means, as soon as the difference in axial and tangential strain signals approaches zero. This is the point at which the surge controller output should be set to open the surge control valve.

3.5.2 Compressor Performance

The compressor performance map for the final field compressor in terms of head vs. inlet volumetric flow rate for several compressor speeds from data provided by the manufacturer is shown in Figure 3-30. An expected surge line, from manufacturer's data, and a surge control line at a 10% higher flow rate are also shown in this figure. The compressor performance map in Figure 3-30 is for a suction pressure of 200 psia with normal pipeline natural gas. Similar performance maps could be prepared for other (higher) suction pressures (or other gases) or the curves could be normalized into a single non-dimensional compressor curve. Figure 3-31 showed a plot of the head coefficient, H/N^2, as a function of the flow coefficient, Q/N, for the field test compressor with the expected surge point shown as the last dot on the left of the plot. From this curve, the expected performance, including the expected surge limit for the compressor at any condition (suction pressure, gas density, and speed) can be predicted.

Data from some of the low pressure (200 psia) field testing was plotted on the performance map shown in Figure 3-32. The red data points in Figure 3-32 illustrate the increased performance range of the compressor using the surge detection probe. All of the red data points, except the very lowest flow points, are stable operating conditions without surge. It is clear from the data points in Figure 3-30 that an additional 25.4% of operational range compared to the expected surge line was obtained for this field compressor. Data taken from the

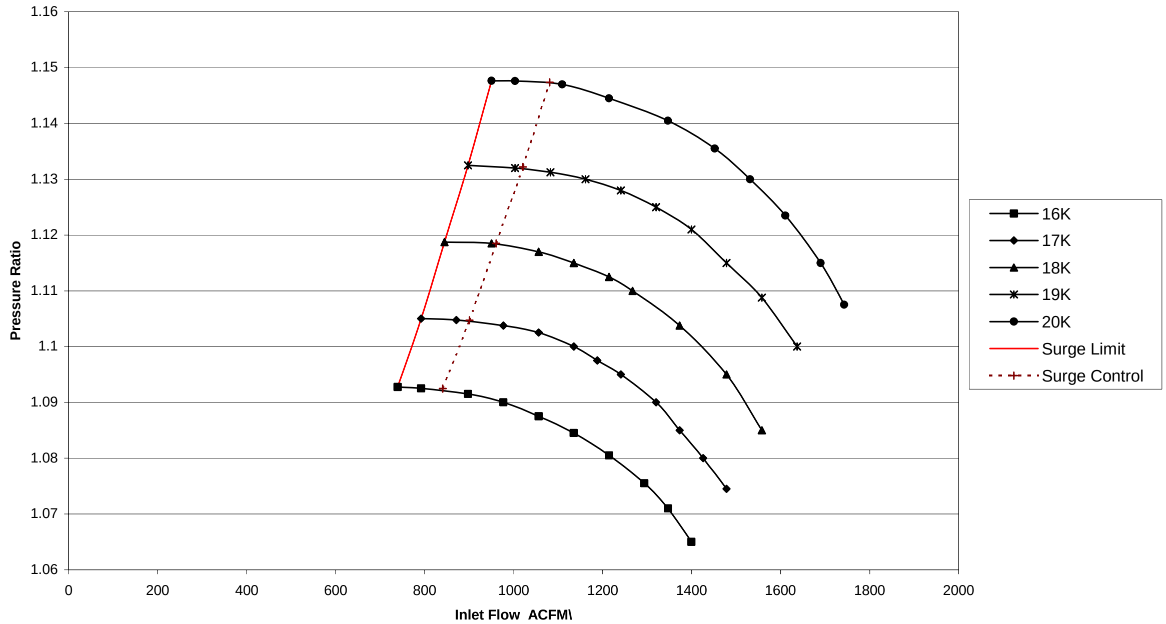


Figure 3-30. Field Compressor Performance Map for Low Pressure (200 psia) Operating Conditions Showing the Surge Limit and the Surge Control Line

high-pressure, high-speed test shown in Figure 3-28 was converted through the head and flow coefficients shown in Figure 3-31 to equivalent conditions for the low pressure performance map and are shown as the green higher speed data points in Figure 3-32. The increased performance range for the high-speed test results is approximately 19.2%. All of the green data points to a stable operating condition. The results indicate that the increase in operating range is less at high speeds than at low speeds. The benefits of direct surge control can be seen in the reduction of the recycle flow necessary to avoid surge in the compressor. The amount of flow difference between the current surge control line (see Figure 3-30) and the new surge control line shown in Figure 3-32 is between 400 and 470 ACFM at the low speed, low pressure operating conditions. The horsepower used to compress the total operating flow of approximately 1,050 ACFM at the low pressure conditions is approximately 460 HP from the driver. Therefore, the power requirement for recycling an extra estimated 430 ACFM, if the previous surge controller was in use, would be approximately 188 HP. Therefore, at an overall driver efficiency of 25% and a fuel cost of $5.00 per MSCF, the direct surge controller would save approximately $2.35 per hour of recycle operation for this small low pressure pipeline compressor. The resulting savings for implementation of the direct surge controller technology in a single large gas transmission pipeline compressor could easily be $50 to $120 per hour for typical low flow operation. Applied to most of the centrifugal compressors in the gas transmission industry, the direct surge control system could save between $50 and $85 million per year in reduced fuel costs.

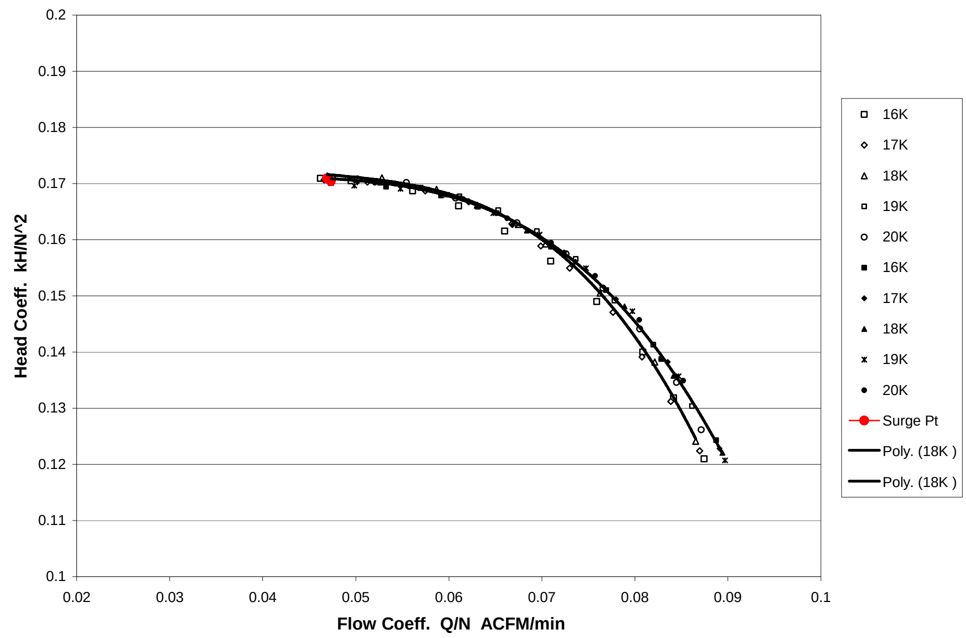


Figure 3-31. Field Compressor Normalized Performance Map for All Operating Conditions Showing the Manufacturer's Expected Surge Point

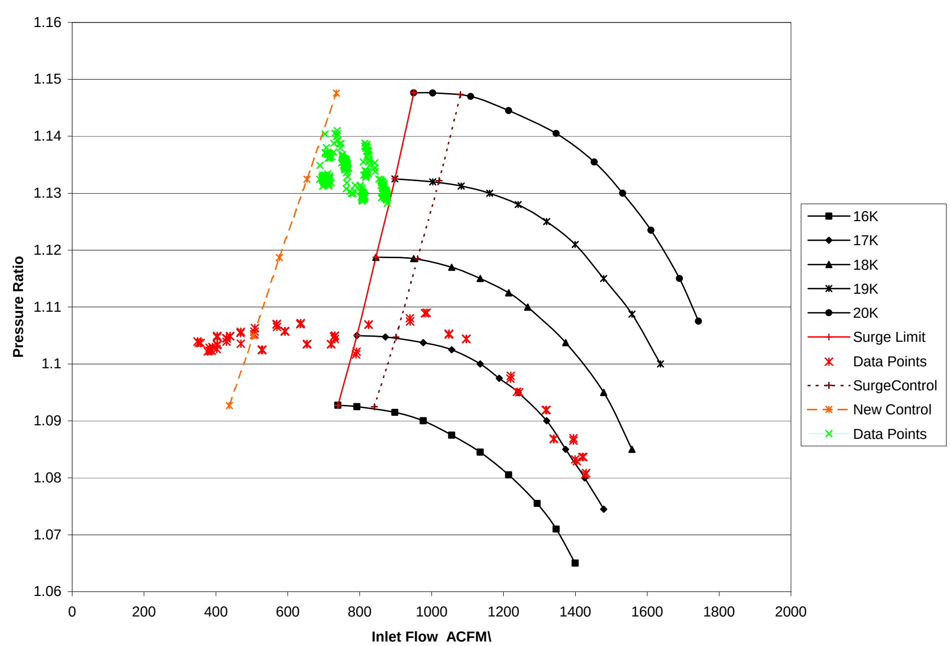


Figure 3-32. Field Compressor Performance Map Showing Operational Surge Test Data and the Suggested New Surge Control Line

4. RESULTS AND DISCUSSION

4.1 CFD MODELING

4.1.1 Modeling Overview

In the experiments conducted for this project, the surge probe was placed near the outer diameter of the flow inlet. This position was selected based upon previous research suggesting where flow re-circulation might occur. An initiative was undertaken to determine if computer modeling could be used to predict where re-circulation might occur and to potentially explain the differences between results with 2D and 3D impellers.

A series of Computational Fluid Dynamics (CFD) models were constructed to determine whether or not such simulations could accurately characterize fluid flow in systems analyzed for this project. CFD utilizes the computational power of modern processors to solve complex fluid dynamics equations over a large discretized region. If the CFD representations in this analysis were found to match the actual flow patterns observed experimentally, the models could serve as a means of predetermining where surge may occur when applying different system parameters. CFD, therefore, could aid in the design of surge control systems by determining where probes might be placed in a compressor assembly to best detect flow re-circulations. These simulations could be useful in explaining discrepancies in experimental results where surge probes did not detect re-circulating flows. The following sections outline the modeling approach, results, and implications of the findings.

A commercially available package, CFDesign 7.0 (Blue Ridge Numerics, Inc.) was used for the CFD analysis in this section. The geometries described in detail below were created in a solid modeling package and then imported into CFDesign. The application of boundary conditions, material and fluid properties, and node meshing were performed in the CFD package.

Three-dimensional solid models of flow geometry were constructed based on dimensions taken from the second compressor used in the field testing for direct surge control development. A cross-sectional view of the flow inlet is shown in Figure 4-1. The figure also shows the relative position of the probe used in the experimental portion of this project.

A 3D model of the inlet was created for the CFD analysis. It should be noted that the models, with the exception of the impeller, are fluid flow volumes and not solid structures. The inlet was artificially extended in the CFD representation as shown in Figure 4-2. The reason for such an arrangement is that a swirl condition could exist around the change in path at the inlet. This non-uniform flow at the location of a boundary condition would prevent the model from converging. The additional straightening section did not affect results in the locations of interest.

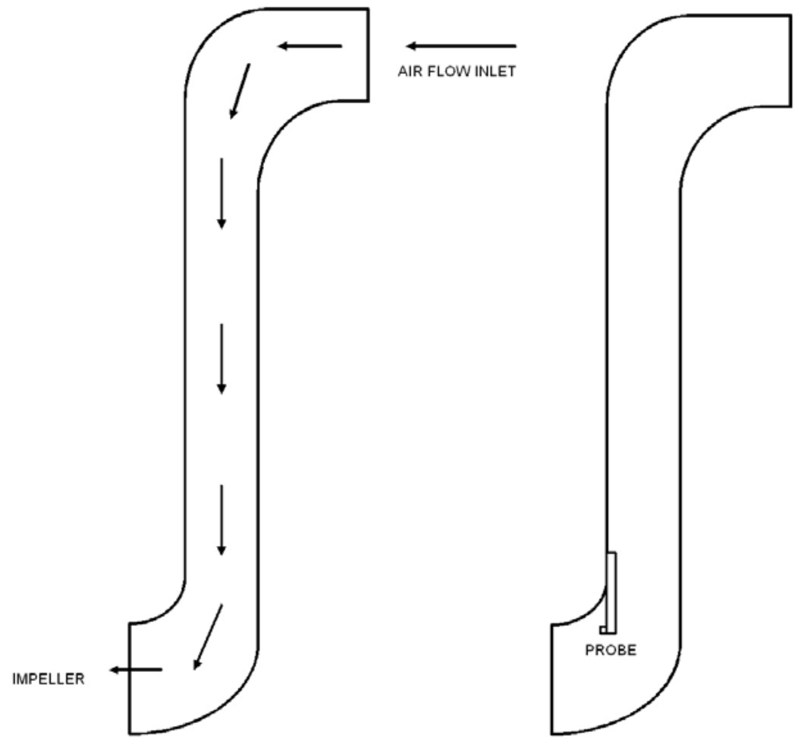


Figure 4-1. Cross-Sections of the Inlet Flow Path, One with the Direct Surge Detection Probe Location Shown

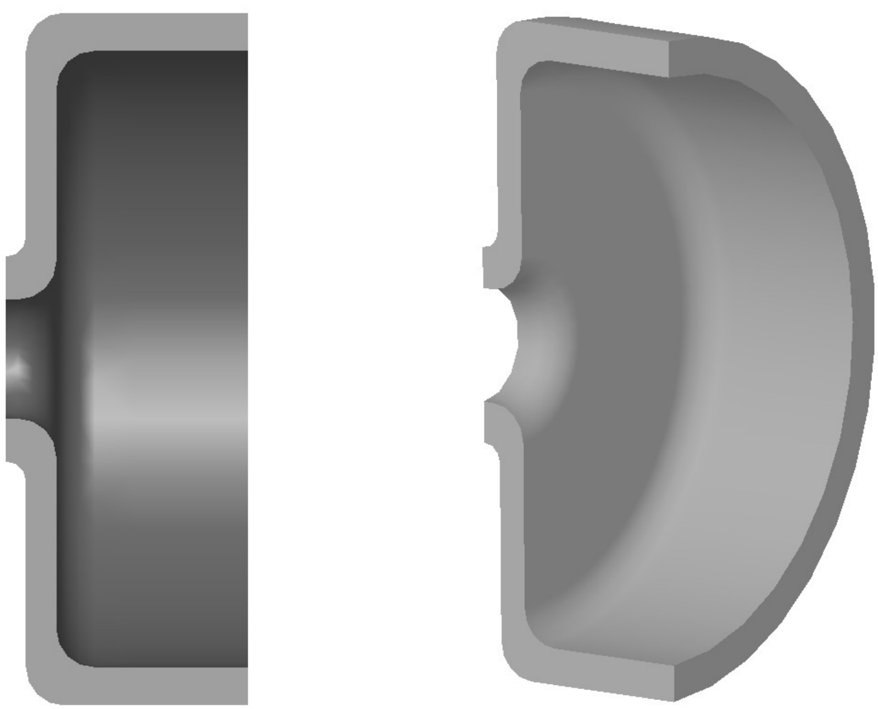


Figure 4-2. Cross-Sectional View of Inlet Flow Path Geometry

The solid model of the impeller wheel was created based on a physical specimen. Two different impeller geometries were created. The first was a full 3D blade geometry in which the blades are flush with the inlet face of the impeller. Such a configuration is depicted in Figure 4-3. The second geometry tested was a 2D blade in which the blades are recessed from the inlet face of the impeller. The 2D impeller model used in this analysis is shown in Figure 4-4.

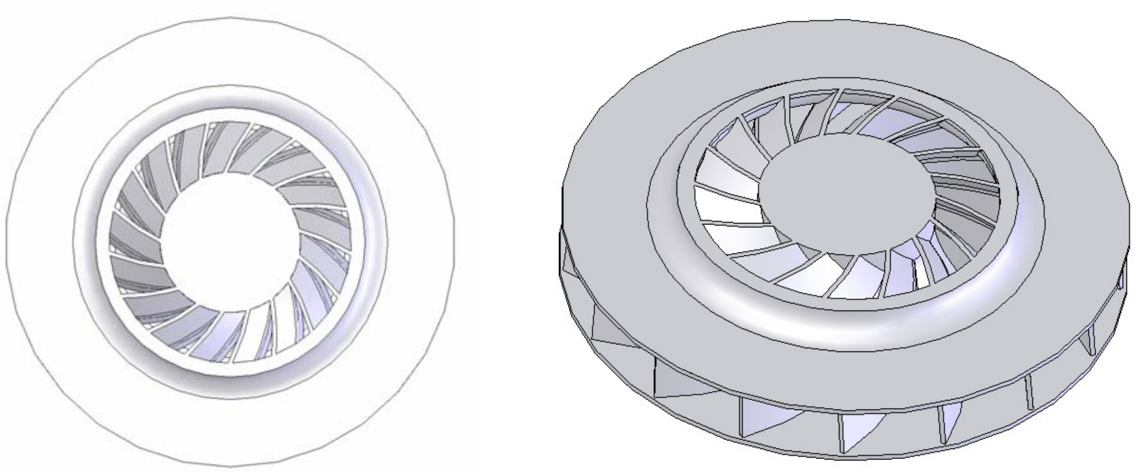


Figure 4-3. Three-Dimensional Impeller Geometry Used for CFD Analysis

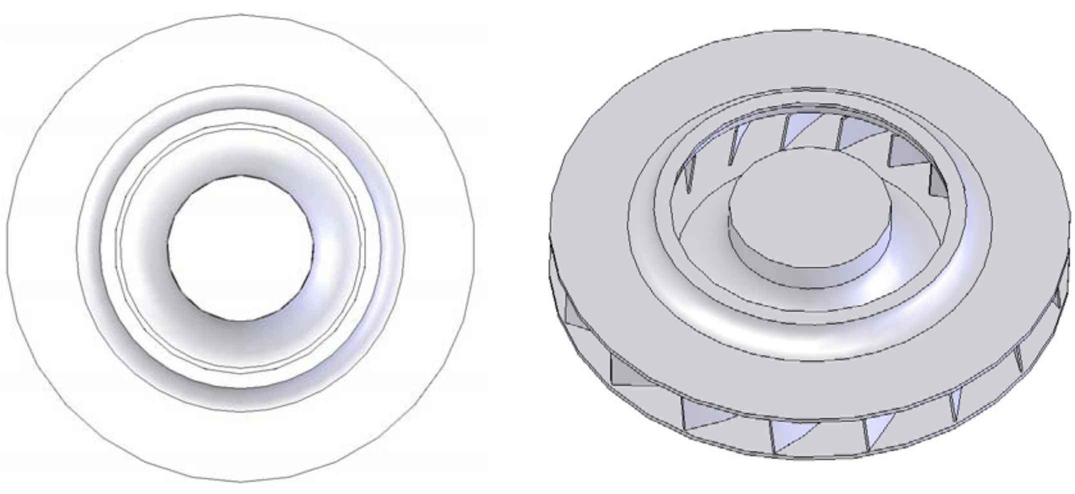


Figure 4-4. Two-Dimensional Impeller Geometry Used for CFD Analysis

As previously mentioned, CFD packages typically analyze fluid volumes and not physical structures. CFDesign 7.0 uses a combination approach to solve systems that contain rotating volumes of fluid. First, a physical representation of the rotating structure is created. In this project, the rotating solid was the impeller geometry described above. Next, a volume of fluid completely surrounding the impeller is created. For this project, a disk with the same outer dimensions as the impeller was created. The CFD package then "cuts out" the solid volume from the fluid volume. Such a representation is shown in Figure 4-5. Both volumes then rotate together in the CFD analysis. The other geometric entity produced for this project was an outlet disk surrounding the outer diameter of the impeller.

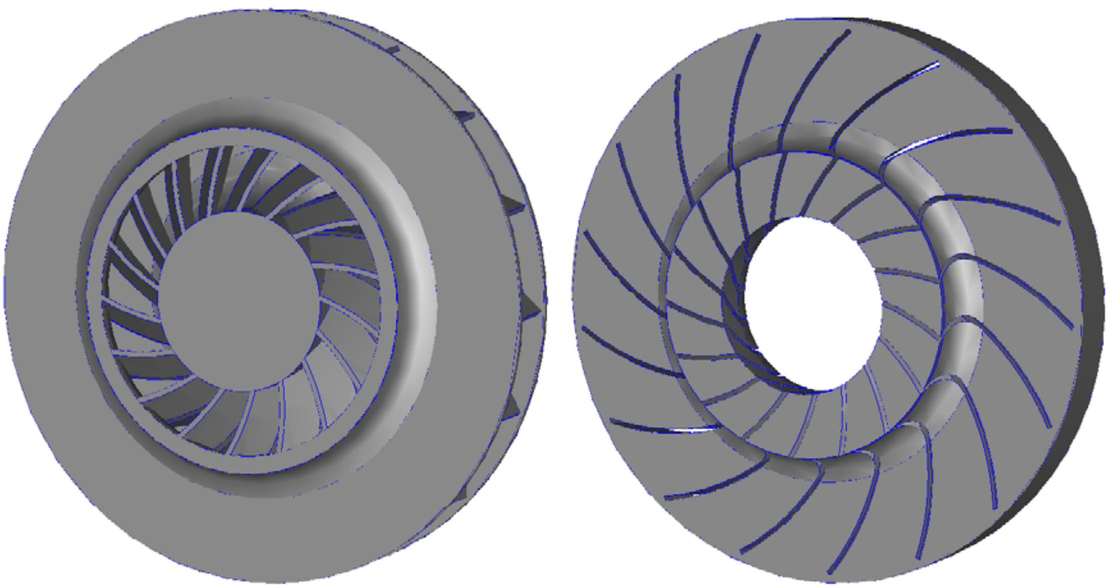


Figure 4-5. CFD Representation of Impeller and Rotating Fluid Volume

Two boundary conditions were given for each model run: an inlet volumetric flow rate and a discharge backpressure. The values used for these conditions were based on measurements from experiments. For all of the simulation runs, the discharge backpressure was held at 221 psia. The change in flow profile prediction was achieved by altering the inlet flow rate. For a compressor that served as the basis for the created model geometry, the design flow rate was 1,036.5 ACFM. A near-surge condition was found experimentally at 842.5 ACFM and surge was detected at inlet flow rates of approximately 684.5 ACFM. The following section provides results from various simulation runs. For all cases presented in this report, the impeller speed was 17,000 rpm.

4.1.2 Modeling Results

The CFD output provides results such as velocity and pressure values at various locations within the modeled geometry. One useful output from the analysis is vector diagrams showing the fluid flow path. These diagrams can be produced at various cross-sections to determine if flow reversal is present and can pinpoint the location of such conditions. Figure 4-6 shows the vector plot of the case with design flow boundary conditions with the 3D blade geometry. The

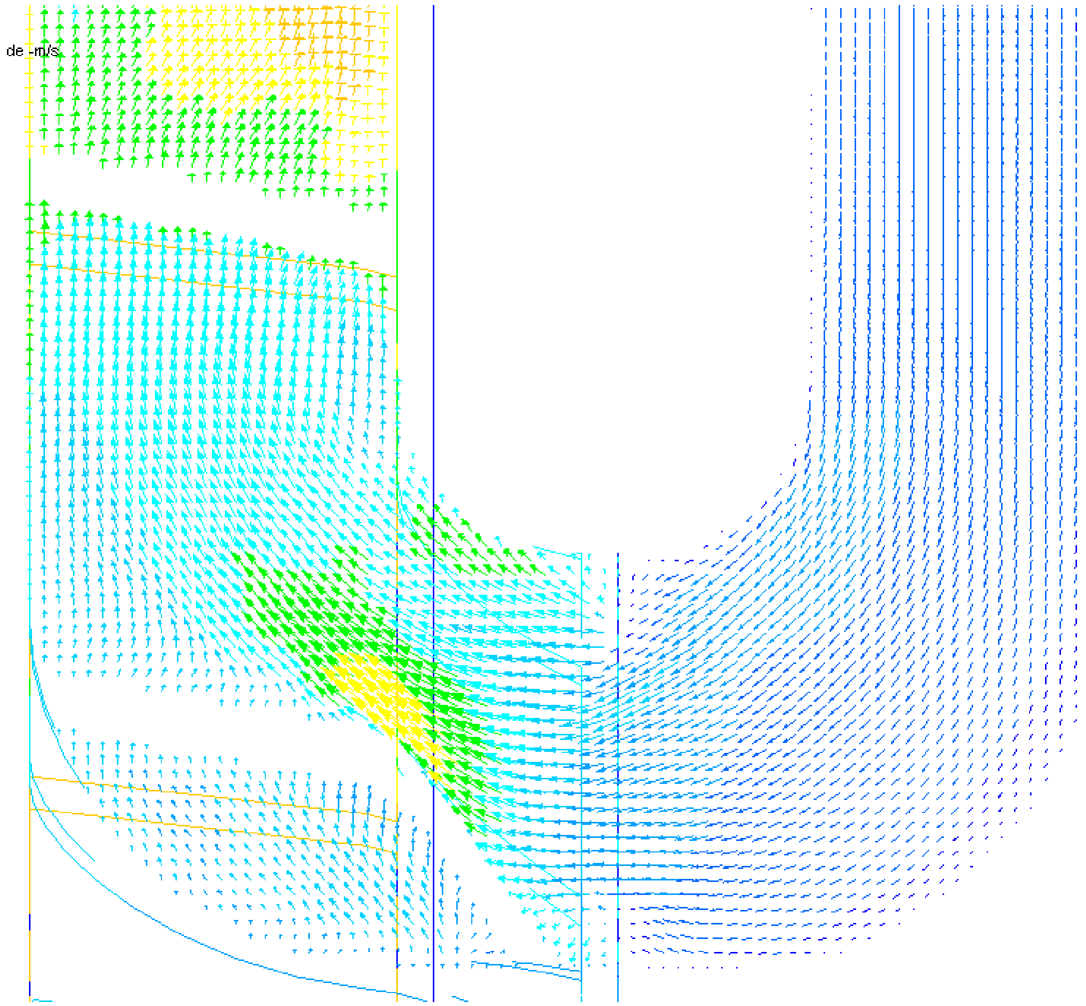


Figure 4-6. Vector Plot for Design Flow (1,036.5 ACFM) Boundary Conditions

voids within the path represent the location of physical obstructions, such as impeller blades. As the diagram depicts, there is no discernable re-circulation of flow. Such an observation is the expected result under these conditions.

The next diagram, Figure 4-7, depicts the flow field for boundary conditions that were experimentally very close to the surge condition with a 3D impeller. The plot clearly shows flow re-circulating back toward the inlet of the impeller. It would be expected that this reversal would be more pronounced at lower inlet volumetric flow rates. Figure 4-8 shows a case with a lower inlet volumetric flow rate. As expected, more flow re-circulation is predicted. The steady CFD analysis used in this modeling cannot predict or simulate the occurrence of surge. In addition to correlating well with test data, the findings also confirm the prediction of the location where the re-circulating flow will occur.

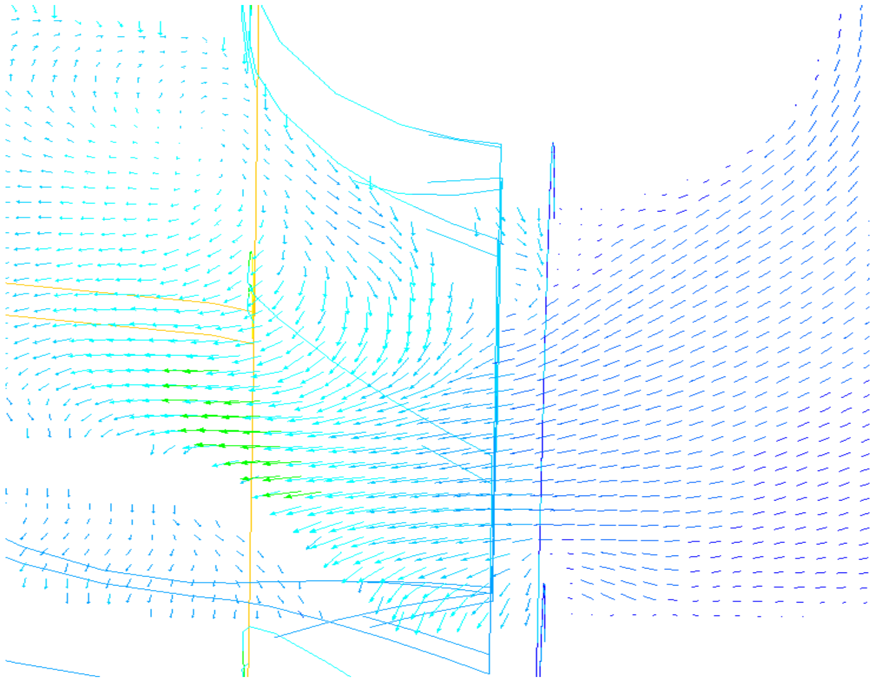


Figure 4-7. Vector Plot with Boundary Conditions at Which Surge is Approximately Expected (684.5 ACFM)

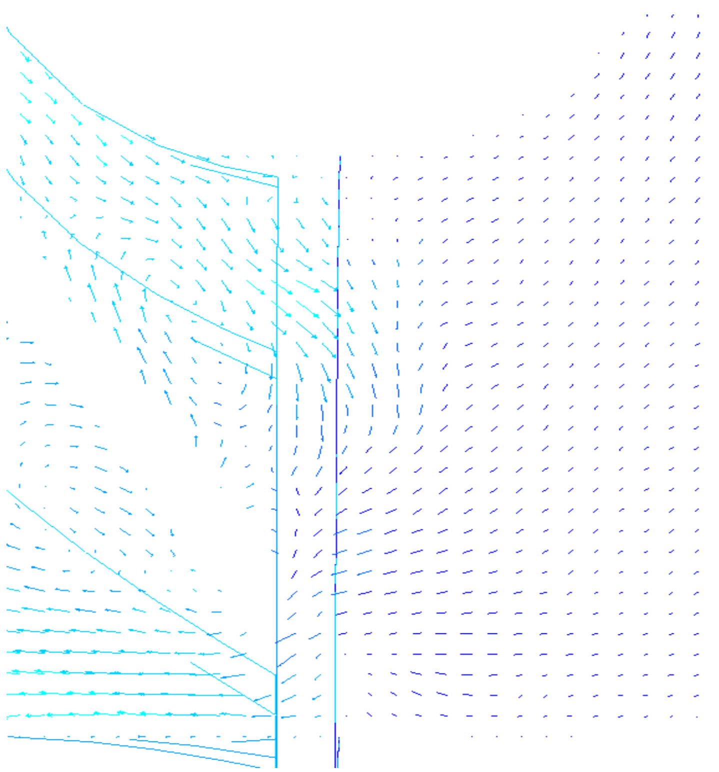


Figure 4-8. Vector Plot with Incipient Surge (Lower Flow) Boundary Conditions

A significant point of interest illuminated by laboratory experiments is the location of flow re-circulation in a particular compressor geometry. Previous research showed that for a standard 3D blade geometry, surge was likely to occur near the outer diameter of the flow inlet passage. Such a prediction was confirmed in tests performed for this project. However, surge probes did not detect backflow for 2D blade geometries. One theory to explain such a discrepancy is that re-circulation may have occurred in a location other than the one at which the probe was placed. A series of CFD simulations were performed for the 2D geometry previously depicted in Figure 4-4. For the design flow cases, no re-circulation was seen in the model as expected. The results for an approximate surge condition flow rate of 684.5 ACFM are shown in Figure 4-9, which indicates that re-circulation occurs in a position recessed from the impeller inlet (at the blade entrance). This revelation would lend credence to the theory that probe positioning may have been responsible for differences in data between compressors.

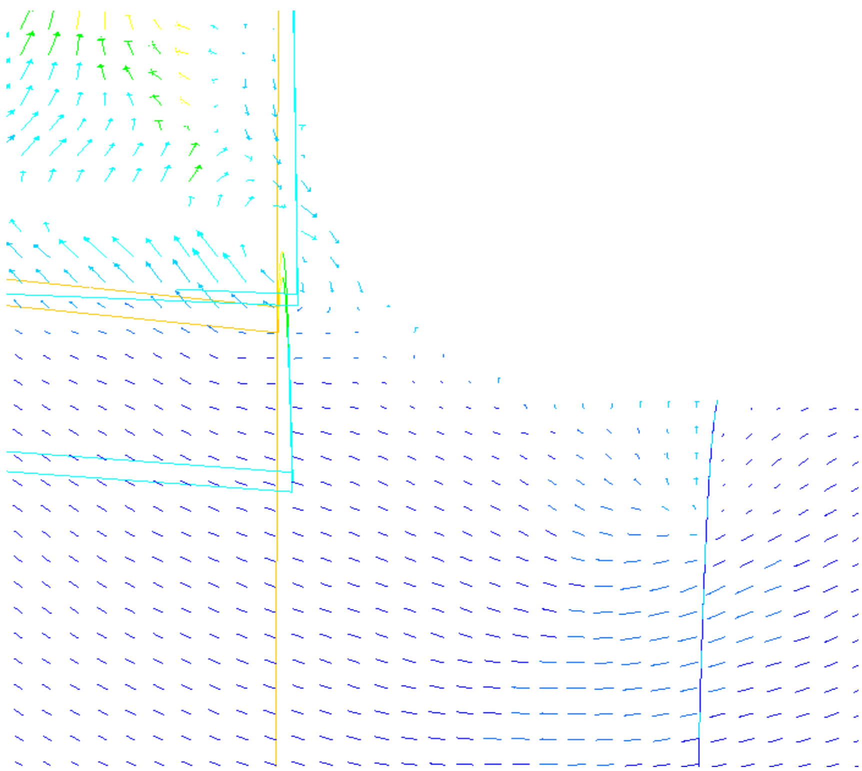


Figure 4-9. Vector Plot Showing Location of the Re-circulation for an Approximate Surge Flow Condition in a 2D Blade Geometry Impeller

The results of the CFD analysis reflect experimental observations found for the 3D impeller. Future work in surge analysis might benefit from the use of CFD. One example of a utilization of such a computer tool would be using CFD to determine where a probe should be placed to most accurately detect the re-circulating flows. Additionally, the models would aid designers in the selection of flow conditions under which normal operation should be sustainable.

The CFD analysis of the 2D impeller predicted that flow re-circulation occurred inside the impeller and not at the inlet-impeller interface. In the experiments performed for this project, the surge probe was not placed inside the impeller. This placement may explain why surge patterns were not observed at relatively low inlet flow rates for 2D impellers.

4.2 STEPS IN THE APPLICATION OF DIRECT SURGE CONTROL

4.2.1 Designing a Direct Surge Control Probe

The steps necessary to retrofit an existing compressor with a direct surge control system have been defined and developed during this project. Installation of a direct surge control system in a new compressor during the design and manufacturing process would be a relatively easy subset of these steps.

1. Determine if the compressor has a modern 3D impeller with blades at the front face of the impeller. If the impeller has 2D blades in which the leading edge of the blades is recessed from the impeller face, the current approach is not appropriate.
2. Calculate the flow velocity range and gas density range from the performance map and operating conditions of the compressor. The inlet geometry of the impeller, including the inlet face smallest (hub) and largest (outer blade tip) diameters is required for this calculation.
3. Size the drag body that will be located along the outer wall of the inlet passage. The drag body should extend into the inlet channel by considerably more than the boundary layer thickness, but by no more than a quarter of the channel height (distance from inner to outer radius). A size of 8 to 15% of the channel height is a good starting point, with the understanding that in small passages a height of more than 20% may be necessary, while in large channels, 5% may be sufficient. The total cross-sectional area of the probe should be no more than 2% of the total inlet flow area, and the diameter of the drag body can be set from this and from practical considerations of penetration hole diameter, other geometry considerations, and manufacturable sizes. If the impeller inlet has several individual channels or passages between inlet guide vanes, then the probe should not be so large as to excessively block an individual channel or passage. In general, the height and width of the drag body should be of the same order of magnitude.
4. Calculate the force on the drag probe from the drag coefficient times the velocity pressure and cross-sectional area of the probe. The velocity pressure in a flow is the density times the velocity squared divided by two times the gravitational constant (e.g., $\rho V^2/2g$). The drag coefficient for a circular cylinder, the shape used for these drag probes, is approximately 0.5 at the high Reynolds numbers of inlet flows. Force on the drag probe is the velocity pressure times the cross-sectional area times the drag coefficient.
5. The probe's bending beam width and length are then determined. The bending beam will generally have a square cross-section, and the width can be wider for strength and narrower for sensitivity, but must accommodate the strain gauges to be

used. The bending beam does not need to be exposed to the flow, but must be attached to the support or restraining piece.

6. The strain expected from the strain gauges placed on the bending beam is calculated. Strain gauges are used in pairs (on opposite sides) at the support end of the bending beam to generate the maximum strain per unit force and to produce a null strain due to thermal expansion or contraction.

7. Calculate the mechanical natural frequency of the probe on the end of the bending beam and calculate the vortex shedding frequencies due to flow over the drag body. Check that the mechanical natural frequency does not correspond to the compressor running speeds or any of several higher orders of compressor speed such as blade passing frequencies. This step also confirms that the vortex shedding frequencies are above the expected surge and sub-synchronous stall frequencies of a few to several Hz and below the mechanical natural frequency of the probe over the full range of expected flows.

8. Design the probe holder or support to attach the probe to the compressor's stationary parts, to position the drag body along the outer wall of the inlet passage as close as practical to the impeller, to allow for the length of the bending beam, and to retain the probe and provide a rigid support. The location of the probe with respect to the expected re-circulation can be checked at this point by conducting a CFD analysis of the target compressor inlet. The other part of the physical design, which must be considered at this stage, is a means to allow the signal wires to exit the compressor.

9. Checking the results of the design so that the probe is not too large for the inlet passage, experiences a reasonable force at near-surge flow conditions, will not fail or deform at extreme flow conditions, will not vibrate due to compressor or flow excitation, is located correctly (per experience and the CFD results) along the outer wall of the inlet passage near the impeller, and can be supported firmly in the compressor. If these conditions are not met, repeat the design process and adjust the factors to improve the flow sensor's parameters and overall performance. The above design steps are iterative and several adjustments to the size, the bending beam, the supports, and even the location, may be required until the probe is strong, sensitive, and vibration-free. In general, a drag probe is an adaptable and versatile device that can be designed for the specific compressor geometry and operating conditions. A drag type direct surge probe, which resulted from this design process, is shown in Figure 4-10.

10. The final step in the design of the probe involves arranging for signal wires to pass from the probe to the outside of the compressor case where the signals can be monitored. Arrangements 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 invalidate the hydro test and pressure rating of the compressor case is an essential design step. The probe is located on the suction side of the impeller, and signal wires may have to be routed through an internal division wall, a flow guide, or other internal compressor structure, but will generally be in the suction cavity of the compressor. The signal wires may

Figure 4-1(Bending

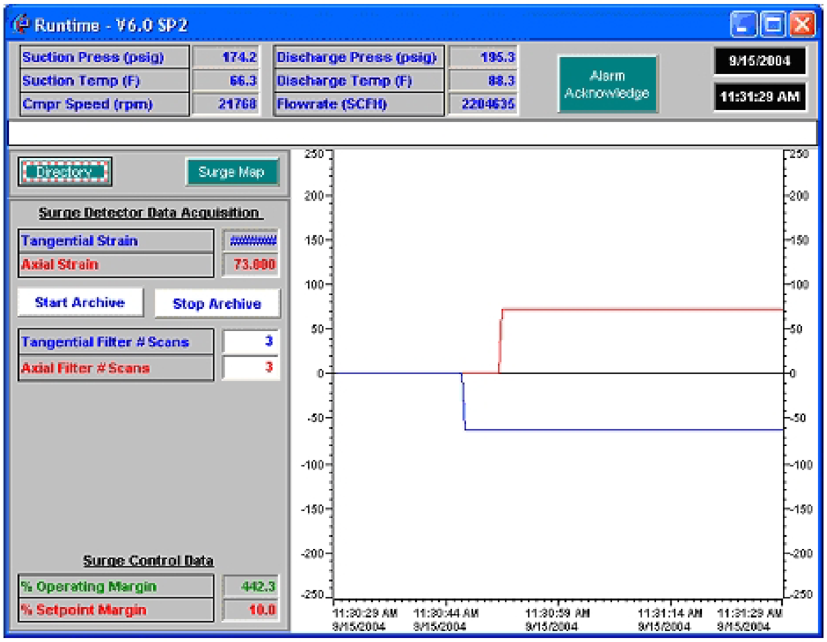


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Figure 4-11. Surge Detection Display from Prototype Direct Surge Controller

penetrate a drain plug or an existing threaded pressure tap, a flanged pressure cover or plate used for a control device penetration, or as a final option can be routed into the suction piping and through a pipe pressure tap. The strain gauges and wires on the probe are coated for protection, and the wires are run through well-secured conduits or metal tubing for protection. The final wiring design involves many details unique to each model of compressor, but if all of the issues are considered, can result in an effective, reliable, and practical means of connecting the surge probe to an external bridge amplifier and signal monitoring circuit.

4.2.2 Designing a Direct Surge Controller

The steps in the application of a direct surge control continue as follows:

11. Design and select the surge controller, which can be an adaptation of a current technology controller that is able to monitor the compressor's suction and discharge pressures and temperatures, and the compressor speed and flow signals. The surge probe signals that the controller must monitor are two voltages from bridge type amplifiers that are usually in the range of ±5 volts and represent the total axial and tangential strains. Strain gauge outputs contain high frequency information and internal compressor flows can change rapidly; however, a controller sample rate of 100 to 200 Hz should be adequate to follow the pre-surge flow changes and control the compressor. Independent of the other data, the direct surge controller should be able to sample, filter, or short average, and process and compare the two strain signals rapidly. A view of the surge detection display for the prototype direct surge controller is shown in Figure 4-11.

12. Refinement or tuning of the control algorithm is required to control a compressor's recycle valve from the direct surge probe signals. This is the commission or setup phase for implementation of a direct surge control system. If the difference between the two strain signals is taken as shown in Figure 3-25, then small positive values (5 micro-strain) or a zero crossing could be used to trigger initial action (recycle valve opening) and a stable negative value (-5 micro-strain) could be taken as the minimum recycle, most efficient, operating point when surge control is needed. An output to start opening the recycle valve at a selected small positive value of the difference in strain should be automatic and fast acting. A finer control could then be used to slowly close the recycle valve until the strain difference signal is near zero. This is one option for direct surge control and should be applicable to essentially all modern compressors. Other algorithms are possible and could be considered.

Some of the details of the controller and the implementation of direct surge control used during the final field and operational testing are described in the following paragraphs. The strain gauge wires from the surge detection drag probe are connected to a two half bridge amplifiers and the voltage output from the amplifiers, one for the axial and one for the tangential strain, are monitored by the controller. The prototype surge controller is a Class I, Division 2, Group D system that is wall mounted in a NEMA 12 panel. This PLC controller includes a power supply, 16 digital, 8 analog, and 8 RTD inputs, 8 relays and 4 analog outputs, and MODBUS communication. In addition to the strain signals, the surge controller accepts the suction and discharge pressures, suction flow DP (from an orifice), compressor speed, and suction and discharge temperatures as inputs. A wiring diagram of the surge controller input, prior to final modifications, is shown in Figure 4-12. The controller calculates and monitors the compressor flow, speed, and pressure ratio so that the operating point can be tracked on a compressor performance map. A traditional surge control routine based on the flow rate compared to the surge control line on the performance map operates in the background. The controller is capable of interfacing with the compressor controls to provide unit starts, stops, and other operating modes, although this was not implemented during the field testing.

As the primary and new surge control, the strain signals are displayed on a screen with the generally positive axial strain shown in red and the generally negative tangential strain shown in blue as in Figure 4-11. The strain gauge signals are sampled at a rate of approximately 80 to 100 samples per second and then are averaged a selectable number of times, displayed on the output screen, and recorded into memory. The average process is effectively a signal filter and can be adjusted between 1 to 10 averaged samples per output data point. Averaging three samples was used for most of the field and operational testing as this provides a relatively smooth but responsive output. When logging is requested, all of the input data, including strains, pressures, temperatures, speed, and calculated flows, are stored in files on the computer disk and can be downloaded following the test. The algorithm implemented in the surge controller at the end of the field testing is based on the difference between the axial and tangential strains. When the two signals approach or cross each other near zero as indicated by a small or negative difference (and the compressor is operating), then the unit is near surge. A further increase in the tangential, particularly with a decrease in the axial strain, indicates that surge is imminent.

The controller has output that could be used to control the compressor recycle valve, although this was not implemented during this development testing. The output signal would normally be in the form of a 4-20 milliamps signal but could also be voltage or digital signals. A selected value of the difference between the axial and tangential strain, such as 5 or 10 micro strain, could be selected as the value if reached over two or more repeated average reading would cause the recycle valve to start to open. The controller could hold the valve just open enough to maintain the small positive difference value or even to trend towards a zero difference value.

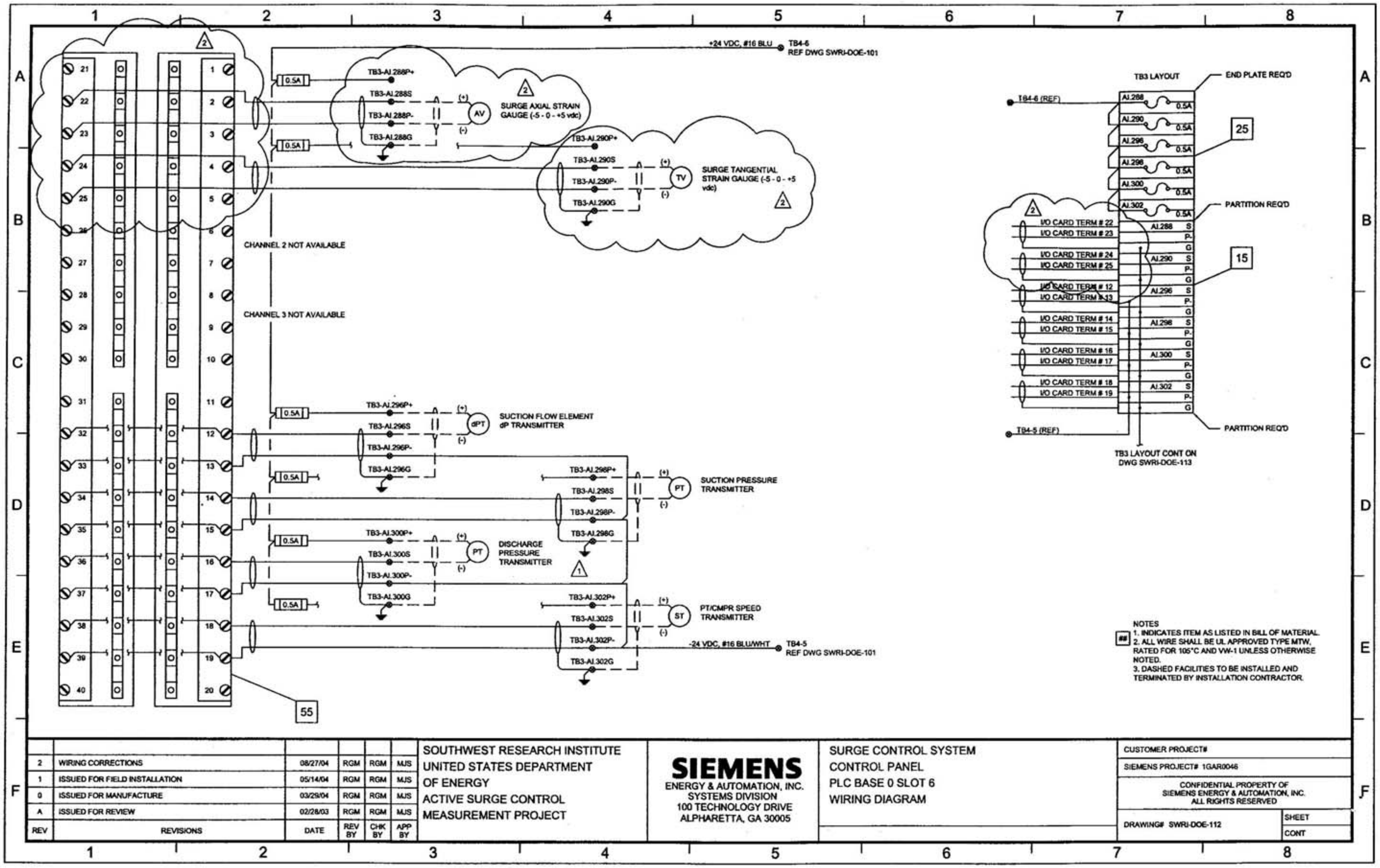

Figure 4-12. Wiring Diagram of Surge Controller Input

5. CONCLUSIONS

The following conclusions result from the industry's need for increased flexibility for turbo-compressor operations and from research that has been conducted on pre-surge detection and direct surge control for pipeline centrifugal compressors.

1. The motivations for a direct surge control system are to refine surge control, safely and properly reduce surge margins, increase operating range and flexibility, and in a safe and reliable manner increase efficiency, save energy, and reduce operating costs.

2. Surge is a potentially damaging flow instability that limits the low-flow operation of centrifugal compressors and is usually avoided by recycling flow to maintain an arbitrary minimum flow above a surge margin. This can be wasteful when more flow is recycled than necessary and is costly in terms of energy and fuel when the compressor operation is maintained above the near-surge conditions at which the compressor can safely operate without recycled flow.

3. The background for this research is previous research sponsored by GMRC that identified an impeller inlet flow pattern (re-circulation) that indicates the approach of surge in centrifugal compressors and a method to detect that flow change.

4. Detailed specifications for surge detection probes and surge controllers 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.

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, the strain sensitivity needed at minimum flow, the mechanical natural frequency, the excitation frequencies so that the sensor will not fail dynamically, 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. It has been shown that functionality and sensitivity checks are necessary in the future but that new probes will not need to be flow tested or calibrated in detail.

7. Both early laboratory and the first field test showed that the current approach to the direct surge control probe location requires a 3D impeller with the blades at the face of the impeller. With the current probe design and location, 2D impellers do not provide sufficient re-circulation at the impeller inlet for adequate direct surge control.

8. Testing in a 3D laboratory compressor showed strong re-circulating flow signals with axial strains, which decreased with flow, passed through a local maximum,

and then decreased rapidly as surge approached. The tangential strain in the laboratory compressor remained low at flows away from surge and then increased rapidly as surge approached. These results were consistent and distinctive indications of approaching surge.

9. The field testing results showed that the strain output is sufficiently stable to provide comparative results over extended periods of time, that sound reliable wiring connections can be made through the pressure case of the compressor, and that the re-circulation flow signals are similar to the laboratory results. The axial strain decreases and the tangential strain increases as surge approaches and when both values are near zero or cross with their difference becoming a small value, then it is a strong indication that surge is imminent.

10. The operational test results showed that a significant (19% to 25%) increase in low flow operating range and flexibility of the compressor could be obtained through use of the direct surge control technology. The potential savings from this increase in operating range is approximately 10 to 24 MSCF or typically $50 to $120 per hour of operation per pipeline compressor to which the direct surge control technology is applied. An industry wide saving in fuel of more than $50 million per year is possible.

11. Computational Fluid Dynamics (CFD) modeling of a representation of the field compressor showed that the flow re-circulation does occur as expected in the outer diameter portion of the impeller inlet at low flow rates. A 2D model of the same overall compressor configuration showed that the re-circulation occurred at the blade tips and not at the impeller inlet when the blades were recessed from the impeller inlet. The development of this modern CFD technique can be used to help place surge detection probes in the proper locations.

12. A step-by step design procedure for direct surge control probes and controllers has been defined by this research and is enumerated in this report. Although a probe design will have to be completed for each new compressor model and pressure or flow range, the results will be similar for compressors of a given type and size. Direct Surge Controllers, once implemented and tuned for each compressor condition, will be applicable to all direct surge installations.

6. REFERENCES

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