Gregory K. McMillan

Centrifugal and Axial COMPRESSOR CONTROL

IRP Instructional Resource Package

STUDENT TEXT

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INSTRUMENT SOCIETY OF AMERICA

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SECTION 1

INTRODUCTION TO THE STUDENT

1-1 COURSE CONTENTS

The cost of machinery damage and process downtime due to compressor surge and overspeed can be from thousands to millions of dollars for large continuous chemical or petrochemical plants. This text demonstrates how to select the proper control schemes and instrumentation for centrifugal and axial compressor throughput and surge control. More material is devoted to surge control because surge control is more difficult and the consequences of poor control are more severe.

Special feedback and open-loop backup control schemes and fast-acting instrumentation are needed to prevent surge due to the unusual nature of this phenomenon. In order to appreciate the special instrument requirements, the distinctive characteristics of centrifugal and axial compressors and the surge phenomenon are described. This text focuses on the recent advancements in the description of surge by E.M. Greitzer (Ref. 15). Simple electrical analogies are used to reinforce the explanation. Simulation program plots using the Greitzer model of surge are used to graphically illustrate the oscillations of pressure and flow that accompany different degrees of severity of surge. Extensive mathematical analysis is avoided. A few simple algebraic equations are presented to help quantify results, but the understanding of such equations is not essential to the selection of the proper control schemes and instrumentation.

The surge feedback control scheme is built around the type of controller set point used. In order to appreciate the advantages of various set points, the relationship of the location of the set point relative to the surge curve and the effect of operating conditions on the shape and location of the surge curve are described. The need for and the design of an open-loop backup scheme in addition to the feedback control scheme are emphasized. The use and integration of process and manual override control schemes without jeopardizing surge protection are also illustrated.

Many of the transmitters, digital controllers, and control valves in use at this writing are not fast enough to prevent surge. This text graphically illustrates the effect of transmitter speed of response on surge detection. The effects of transmitter speed of response, digital controller sample time, and control valve stroking time on the ability of the control scheme to prevent surge are qualitatively and quantitatively described. The modifications of control valve accessories necessary for fast throttling and the maintenance requirements are detailed.

The interaction between throughput and surge control and multiple compressors in parallel or series can be severe enough to render the surge control scheme ineffective or even to drive a compressor into surge. This text describes the detuning and decoupling methods used to reduce interaction.

The computational flexibility and power of modern computers facilitates on-line monitoring of changes in compressor performance and its surge curve. This text describes how computers can be used to predict impending compressor damage and the extent of existing damage by vibration frequency analysis. It also describes how computers can be used to gather pressure and flow measurement data to update the surge curve on a CRT screen.

1-2 AUDIENCE AND PREREQUISITES

This text is directed principally to the instrumentation and process control engineers who design or maintain compressor control systems. Process, mechanical, startup, and sales engineers can also benefit from the perspective gained on the unusual problem of compressor surge and the associated need for special instrumentation. Since instrument maintenance groups are genuinely concerned about the proliferation of different types and models of instrumentation, it is critical that the project team members be familiar enough with surge control to be able to justify the use of special instrumentation. Process and mechanical engineers also need to learn how the compressor dimensions and operating conditions can make surge control more difficult and how the piping design can make surge oscillations more severe.

In order for the reader to understand the physical nature of surge, it is desirable that he or she be familiar with some of the elementary principles of gas flow. The reader should know that a pressure difference is the driving force for gas flow, that gas flow increases with the square root of the pressure drop until critical flow is reached, and that gas pressure in a volume will increase if the mass flow into the volume exceeds the mass flow out of the volume and vice versa. If the reader is also comfortable working with algebraic equations and understands such terms as molecular weight, specific heat, efficiency, and the speed of sound, he or she can use the equations presented to describe the surge oscillations and the surge curve. However, the assimilation of these equations is not essential to understanding the control problem and the control system requirements.

In order for the reader to understand the control schemes and special instrumentation requirements, it is desirable that he or she be familiar with the structure, terminology, and typical instrument hardware for a pressure and flow control loop. Specifically he or she should know the functional relationship between the controller, the control valve, and the transmitter; know the terms remote-local set point, feedback control, automatic-manual operation, proportional (gain) mode, integral (reset) mode, and sample time; and know the physical differences between diaphragm and piston actuators, rotary and globe control valves, positioners and boosters, and venturi tubes and orifice plates. The structure, terminology, and hardware for ratio control, override, and decoupling are described in the text as the application is developed.

1-3 LEARNING OBJECTIVES

The surge and throughput control loops for a single compressor appear deceptively simple. However, the success of these loops depends upon the engineer's attention to many details, each of which are critically important. These loops are typically protecting a large capital investment in machinery, protecting against a loss of production due to machinery repair or replacement, and determining the efficiency of a large energy user. The overall goal of this text is to instruct the reader on how to properly design and maintain compressor control loops. The specific individual goals necessary to achieve the overall goal are:

- Learn how the compressor characteristics and operating conditions affect the potential for surge, the surge curve, and the surge controller set point.
- Learn how the compressor and piping design affect the frequency and amplitude of the flow and pressure oscillations during surge.
- Learn how the surge cycles cause compressor damage.
- Learn the relative advantages of different types of instruments in detecting surge.
- Learn how fast the approach to surge will be and how fast the flow reversal is at the start of the surge cycle.
- Learn how to generate the surge curve and the set point for the surge controller for different compressors and operating conditions.
- Learn why a backup open loop is needed in addition to the feedback loop for surge control and how to design one.
- Learn how fast the transmitter, the controller, and the control valve must be to prevent surge.
- Learn how to modify and maintain the control valve accessories to meet the stroking speed requirement.
- Learn how the surge and throughput control system designs affect the operating efficiency of the compressor.
- Learn how to assess the severity of interaction between the surge and throughput control loops and how to reduce it.
- Learn how to use a computer to monitor changes in compressor performance for maintenance and advisory control.

1-4 DEFINITION OF TERMS

We frequently take for granted that others understand the terms we use in the special areas of instrumentation and control. However, misunderstandings or incorrect interpretations can become a major obstacle to learning the concepts. To avoid this problem, the definitions of important terms are summarized below.

axial compressor — A dynamic compressor whose internal flow is in the axial direction.

centrifugal compressor — A dynamic compressor whose internal flow is in the radial direction.

compressor characteristic curve — The plot of discharge pressure versus suction volumetric flow for a typical compressor speed or vane position at a specified suction temperature, pressure, and molecular weight. A family of curves is depicted for variable speed or variable vane position compressors.

compressor diffuser — The stationary passage around the compressor impeller where a portion of the velocity pressure is converted to static pressure.

compressor (dynamic) — A compressor that increases the pressure of a gas by first imparting a velocity pressure by rotating blades and then converting it to a static pressure by a diffuser. Dynamic compressors are either centrifugal or axial. If the discharge pressure is less than 10 psig, dynamic compressors are usually called blowers. If the discharge pressure is less than 2 psig, dynamic compressors are usually called fans.

compressor guide vane — Stationary blades at the inlet eye of the impeller that direct the angle of the gas flow into the impeller. The angle of the blades can be adjustable to impart varying amounts of rotation to the gas. This angle is with or against the rotation imparted by the impeller. The adjustable angle varies the capacity and the discharge pressure of the impeller.

compressor impeller — The blades on the rotating compressor shaft that impart the velocity to the entering gas.

compressor map — the compressor characteristic curves and the surge curve at a specified suction temperature, pressure, and molecular weight.

compressor rotor — The rotating element in the computer that includes the compressor impeller and shaft.

compressor stage — each set of compressor blades plus diffuser is a compressor stage. There can be multiple stages within the same housing or there can be a single housing for each stage with a heat exchanger in between.

compressor stall — Unstable flow pattern in a compressor where the forward flow stops in localized regions around the impeller.

compressor stall or surge curve — The curve drawn through the point of zero slope on each compressor characteristic curve. If the operating point is to the right of this curve, compressor operation is stable. If the operating point is to the left of this curve, compressor surge or stall can occur.

compressor surge — Unstable flow pattern in a compressor where the total flow around the impeller alternately stops or flows backwards and then flows forward.

compressor thrust — The axial displacement of the compressor shaft that can occur during surge.

compressor vibration — the radial oscillation of the compressor shaft that can occur during surge.

controller proportional band — The mode that changes the controller output by an amount proportional to the change in error. The proportional band is the percent change in error necessary to cause a full-scale change in controller output. Proportional band is the inverse of gain multiplied by 100.

controller rate — The mode that changes the controller output by an amount proportional to the derivative of the error. The derivative time is that time required for the proportional band contribution to equal the derivative (rate) mode contribution for a ramp error.

controller reset — The mode that changes the controller output by an amount proportional to the integral of the error. The integral time is that time required for the integral (reset) mode to equal (repeat) the proportional band contribution for a constant error. Most controllers use the inverse of integral time so that the reset setting units are repeats per minute.

controller reset windup — The condition of controller output when the reset contribution to the controller output exceeds the output change of the controller. The controller output is at the upper or lower extremity of its range and will not change until the measurement crosses set point. Most anti-reset windup options for controllers limit the reset contribution so that it plus the proportional band contribution does not exceed an adjustable upper and lower output limit.

steady-state gain — The final change in output divided by the change in input (all the oscillations have died out). It is the slope of the plot of the steady-state response versus input. If the plot is a straight line, the gain is linear (slope is constant). If the plot is a curve, the gain is nonlinear (slope varies with operating point).

steady-state response — The final value of an output for a given input (all oscillations have died out). The compressor characteristic curve is a plot of the steady-state response of compressor discharge pressure for a given suction flow.

time constant — The time required for the output to reach 63 percent of its final value with an exponentially decreasing slope.

time delay dead time — The time required for an output to start to change after an input change.

transient response — The value of an output as it varies with time after an input change. The oscillations of compressor discharge pressure and suction flow during surge are transient responses.

valve positioner — A proportional-only pneumatic controller mounted on a control valve whose measurement is valve position and whose set point is the output of a process controller or manual loader (via an I/P transducer for an electronic loop). The gain of this position feedback controller is typically greater than 100 (the proportional band is less than 1 percent).

volume booster — A pneumatic relay (usually 1:1 — the change in output signal is equal to the change in input signal) that has a much greater air flow capacity than positioners or I/P transducers. The greater air flow capacity increases the speed of the control valve stroke. How fast the pressure in the actuator volume tracks the penumatic signal depends on the supply and exhaust flow capacity of the booster and the size of the actuator.

Important new terms such as *surge* and *stall* will be defined in greater detail in subsequent sections. The first time an important new term is introduced, it will be italicized for reference and emphasis.

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SECTION **2**

DESCRIPTION OF COMPRESSORS

2-1 GENERAL TYPES

The two general types of compressors are positive displacement and dynamic. The positive displacement compressor increases the gas pressure by confinement within a closed space. *Reciprocating compressors* are positive displacement compressors where the closed space in a cylinder is decreased by a piston to compress the gas.

Figure 2-1 shows the steps in the compression cycle of a reciprocating compressor. In step 1 the piston is completely withdrawn from the cylinder. Since the suction pressure is greater than the pressure in the empty cylinder, the inlet valve opens and the gas fills the cylinder until the cylinder pressure equals the suction pressure. In step 2 the piston has partially stroked and reached a position where the cylinder pressure is greater than the suction pressure but less than the discharge pressure. Both the inlet and outlet valves are closed so that there is no suction or discharge flow of the gas. In step 3 the piston has fully stroked and compressed the gas enough to cause the cylinder pressure to exceed the discharge pressure. The outlet valve opens and the gas flows out of the chamber. Notice that at the fully stroked position of the piston there is a clearance volume. If this clearance volume is increased, the capacity of the compressor is decreased. The capacity of reciprocating compressors can also be decreased by decreasing the number of cylinders in service or the speed of the cycle.The capacity cannot be decreased by throttling the discharge flow or the suction flow with a control valve. The suction volumetric flow at suction conditions (actual cubic feet per minute or acfm) for a reciprocating compressor is essentially independent of gas composition or gas pressure within its design limits. Relief valves are installed on the discharge of reciprocating compressors because the discharge pressure can rise and exceed the pressure rating of downstream equipment if a downstream valve is closed.