PSIG 0913

Modeling Emergency Shutdowns of Centrifugal Compressors.

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ABSTRACT

Emergency shutdowns in compressor stations cause fast transients in the operating conditions. The paper and presentation will address the physics of compressor surge , as well as the physics that have to be modeled to describe the system behavior during these fast transients. Sample calculations are presented.

Surge avoidance generally falls into two classes: Avoidance of surge during normal (slow) process changes and surge avoidance during fast transients, which, far example occur during emergency shutdowns.

In this paper, data from a compressor that surged during an emergency shutdown are presented. The data are analyzed to determine the effects of surge and the rate of deceleration. The issue of the rate of deceleration, in particular for different drivers, is discussed.

A model to simulate shutdown events is developed and possible simplifications are evaluated. The compression system is analyzed, thus verifying the model and the simplifications. The impact of piping geometry, valve sizing, and instrumentation on the results are also covered.

NOMENCLATURE

A	Flow area
c_v	Flow coefficient
C	Compressible valve coefficient
F_p	Piping geometry factor
h	Head
H_{cooler}	Gas cooler heat transfer (W)
J	Inertia
k	Isentropic exponent
k	Constant
K_v	Valve coefficient
L	Pipe length
N	Speed (1/s)
p	Pressure
Q	Volumetric flow
SG	Specific gravity
T	Temperature
TD	Turndown
t	Time
V	Volume
Y	Coefficient
Z	Compressibility factor
α,β,γ	Constants
ρ	Density

Subscripts

avail	Available
compr	Compressor
op	Operating point
surge	At surge
std	At standard conditions
ss	Steady state
v	Valve
1	Compressor inlet
2	Compressor discharge

INTRODUCTION

A centrifugal compressor, operating within a compressor station (Figure 1), will exhibit a stability limit that prevents it from operating at conditions that require a flow lower than the flow at said stability limit (Figure 2). The stability limit is usually referred to as surge limit, because once the compressor operating point crosses the surge limit, the flow through the compressor will reverse. This can cause damages to the compressor. It is important to note that surge is a dynamic system behavior, that follows from the interaction of a centrifugal compressor (expressed by its head-flow characteristic) with the pipes, valves, coolers etc. of the compressor station upstream and downstream of the compressor.

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The possible operating points of a centrifugal gas compressor are limited by maximum and minimum operating speed, maximum available power, choke flow, and stability (surge) limit (Figure 2). Surge, which is the flow reversal within the compressor, accompanied by high fluctuating load on the compressor bearings, has to be avoided to protect the compressor. The usual method for surge avoidance ("anti-surge-control") consists of a recycle loop that can be activated by a fast acting valve ("anti-surge valve") when the control system detects that the compressor approaches its surge limit. Typical control systems use suction and discharge pressure and temperature, together with the inlet flow into the compressor as input to calculate the relative distance ("turndown") of the present operating point to the predicted or measured surge line of the compressor (Figure 2). Turndown is defined by:

$$TD = \frac{Q_{op} - Q_{surge}}{Q_{op}} \Big|_{H_s = const} \quad (1)$$

If the surge margin reaches a preset value (often 10%), the anti-surge valve starts to open, thereby reducing the pressure ratio of the compressor and increasing the flow through the compressor. The situation is complicated by the fact that the surge valve also has to be capable of precisely controlling low. Additionally, some manufacturers place limits on how far into choke (or overload) they allow their compressors to operate.

Properly designed centrifugal compressor systems can provide an extremely large operating range, that is further enhanced by the appropriate use of recycle systems. A properly designed recycle system will allow to keep the compressor online down to zero flow into the system, without upsetting the process during the transition from operating with a closed recycle valve through a fully open valve. A summary on application guidelines for sureg control systems is given by Brun and Nored in [1].

For normal operation, as part of process control, the recycle valve must be sized to allow fine tuned control of the recycle flow. This requires a valve that is properly matched to the compressor map, in order to allow bumpless transfer from recycle to normal operation and vice versa.

During emergency shutdowns, the situation is different from regular, controlled recycle. Emergency shutdowns are initiated to protect the compressor station, and require the immediate shutdown of the compressor and its driver. In gas turbine driven compressors, this is initiated by cutting off the fuel flow into the gas turbine. In electric motor driven stations, the electric power to the motor is turned off.

Parameters such as gas volume capacitance in the recycle path, compressor power train inertia (Figure 3) and the flow capacity of recycle and anti surge valves, the dynamic behavior of these valves and their actuators, as well as check valve dynamic characteristics, are crucial in determining the dynamic behavior of the system.

When an emergency shutdown is initiated, the recycle valve, or a dedicated anti surge valve has to be opened as fast as possible to avoid the compressor form surging while it spins down under the train inertia[1]. Conceptually, the problem is as follows: With the driver power cut off, the inertia of the compressor train works against the decelerating force of the gas to be compressed At the same time, the discharge check valve closes.. With the compressor rapidly slowing down, it looses its capability to generate head, and thus to overcome the discharge pressure imposed by the system. The discharge pressure has to be lowered fast by flowing gas through the recycle line. The rate of pressure reduction is determined by the flow through the recycle valve and the volume of gas trapped between the compressor discharge nozzle ,the check valve and the recycle valve. The speed at which the pressure can be relieved of the pressure not only depends on the reaction time of the valve, but also on the time constants imposed by the piping system. The transient behavior of the piping system depends largely on the volumes of gas enclosed by the various components of the piping system, which may include, besides the piping itself, various scrubbers, knockout drums, and coolers. The system boundaries for this study are the first downstream check valve, while the upstream boundary may be either a check valve or an infinite plenum (at constant pressure) Figure 4. The valve arrangement can vary, for example with a single valve in a cooled loop, a dedicated hot bypass valve (for fast transients) in parallel with a cooled recycle valve, or two valves in parallel, with one of them used for slow process control, and both for fast transients (Figure 4). The arrangements described are just a few among many other feasible arrangements (White and Kurz [2])

SURGE PHENOMENON

Figure 2 shows the head-versus-flow characteristic of a typical centrifugal compressor, including the areas of unstable operation. At flows lower than the stability point, the compressor initially shows a reduced capability to generate head with reduced flow, until it experiences reverse flow, that is, the gas now flows from the discharge to the suction side (Figure 5). It must be noted here that the stability point ('surge line') of the system does not automatically coincide with the onset of stall in the compressor, because surge is a system behavior. Thus, the stability limit in question is determined by the interaction of the compressor and the piping system. Many manufacturers will place the surge line on the map at the onset of stall (especially for high pressure compressors) to avoid unduly high vibration levels caused by stall.

[1] Some installations maintain fuel flow to the turbine for 1 to 2 seconds while the recycle valve opens. However, this can generate a safety hazard.

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Once flow reversal occurs, the amount of flow depends on the pressure ratio across the compressor, since in this situation the compressors acts more or less like an orifice. The flow reversal means that the pressure downstream of the compressor is gradually reduced. The speed of pressure reduction depends largely on the size of the volume downstream of the compressor. Once the pressure is reduced sufficiently, the compressor will recover and flow gas again from the suction to the discharge side. Unless action is taken, the events repeat again. Ongoing surge can damage thrust bearings (due to the massive change of thrust loads), seals, and eventually overheat the compressor. Details of the energy transfer from the compressor into the gas are described in [3].

MODELLING THE PIPING-SURGE CONTROL INTERACTION

Design of the piping and valves, together with the selection and the placement of instruments will significantly affect the performance of an anti-surge control system. This is a major issue during the planning stage because the correction of design flaws can be very costly once the equipment is in operation. Typical configurations for recycle systems are outlined in Figure 4. In its simplest form, the system includes a flow-measuring element in the compressor suction, instruments to measure pressures and temperatures at suction and discharge, the compressor, an aftercooler and a discharge check-valve, as well as a recycle line with a control valve, connected upstream of the discharge check valve and compressor flow-measuring device.

The control system monitors the compressor operating parameters, compares them to the surge limit, and opens the recycle valve as necessary to maintain the flow through the compressor at a desired margin from surge. In the event of an ESD, where the fuel to the gas turbine is shut off instantly, the surge valve opens immediately, essentially at the same time the fuel valve is closing.

In a simple system, the boundaries for the gas volume (V) on the discharge side are established by the discharge check valve, compressor, and recycle valve. The volume on the suction side is usually orders of magnitude larger than the discharge volume and, therefore, can be considered infinite. Thus, for the following considerations, the suction pressure remains constant. In a surge avoidance system, a certain amount of the valve's flow capacity will be consumed to recycle the flow through the compressor. Only the remaining capacity is available for de-pressurizing the discharge volume. It must be noted that the check valve is a key feature in determining the dynamic behavior of the system. If the check valve, for whatever reason does not close, or closes very slowly in the event of a shutdown, the huge amount of gas stored downstream the check valve will overwhelm the surge control, or recycle system.

The worst case scenario for a surge control system is an ESD, particularly if the compressor is already operating close to surge when the engine shutdown occurs[2]. With the initiation of shutdown, the compressor will decelerate rapidly under the influence of the fluid forces counteracted by the inertia of the rotor system. A 30% loss in speed equates to approximately a loss in head of about 50%. The valve must, therefore, reduce the head across the compressor by about half in the same time as the compressor loses 30% of its speed.

The larger the volumes are in the system, the longer it will take to equalize the pressures. Obviously, the larger and faster the valve, the better its potential to avoid surge. However, the larger the valve, the poorer its controllability at partial recycle. The faster the valve can be opened, the more flow can pass through it. There are, however, limits to the valve opening speed, dictated by the need to control intermediate positions of the valve, as well as by practical limits to the power of the actuator. The situation may be improved by using a valve that is only boosted to open, thus combining high opening speed for surge avoidance with the capability to avoid oscillations by slow closing.

If the discharge volume is too large and the recycle valve cannot be designed to avoid surge, an additional short recycle loop (hot recycle valve) may be considered, where the recycle loop does not include the aftercooler.

While the behavior of the piping system can thus be predicted quite accurately, the question about the rate of deceleration for the compressor remains. It is possible to calculate the power consumption for a number of potential steady-state operating points. The operating points are imposed by the pressure in the discharge volume, which dictates the head of the compressor. For a given speed, this determines the flow that the compressor feeds into the discharge.

In a simple system as described above, mass and momentum balance have to be maintained (Sentz [4] and Wachter and Rohne [5]). The valve can be described by its flow as a function of the pressure differential, with p_v the pressure just upstream of the valve:

$$Q_v = \sqrt{\frac{1}{K_v}(p_v - p_1)} \tag{2}$$

unless the flow across the valve is choked, thus:

$$Q_v = C\sqrt{\frac{p_v}{\rho_v}} \tag{3}$$

while mass and energy balance yield:

[2] Similar considerations are to be made for the trip of an electric motor driver. The main difference is the different inertia of the motor (and the gearbox).

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$$\frac{dp_v}{dt} = \frac{k}{V}\left[p_2 Q - p_v Q_v - \frac{k-1}{k} H_{cooler} \right] \tag{4}$$

and the momentum balance:

$$\frac{d(p_2 Q)}{dt} = \frac{p_2 - p_v}{L}\left[ART_2 - \frac{p_2 Q^2}{A p_v} \right] \tag{5}$$

where p_2 is a function of the compressor operating point, expressed by:

$$\frac{h}{N^2} = \alpha\left(\frac{Q}{N}\right)^2 + \beta\frac{Q}{N} + \gamma \tag{6}$$

Based on the relationship in Eq. 6, p_2 is calculated from p_1, T_1 and the gas composition using an equation of state.

The above relationship can be used for any positive flow. If the compressor exhibits reverse flow, it can be modeled as an orifice with:

$$Q = -\sqrt{\frac{1}{K_{v,compr}}(p_2 - p_1)} \tag{7}$$

where Kv,compr describes the flow resistance of the compressor against the reverse flow. It should be noted that this somewhat crude formulation suffices for the present study because we want to determine whether the compressor will go into surge at ESD or not. The post-surge behavior is, thus, not important and is only introduced to keep the numerical model stable.

The behavior of the compressor during ESD is governed by two effects. The inertia of the system consisting of the compressor, coupling and power turbine (and gearbox where applicable) is counteracted by the torque (T) that the fluid imposes on the compressor (mechanical losses are neglected) for any given speed. The balance of forces thus yields:

$$T = -2\pi \cdot J \cdot \frac{dN}{dt} \tag{8}$$

Knowing the inertia (J) of the system and measuring the speed variation with time during rundown yields the torque and, thus, the power transferred to the gas:

$$P = T \cdot N \cdot 2\pi = -(2\pi)^2 \cdot J \cdot N \cdot \frac{dN}{dt} \tag{9}$$

If the rundown would follow through similar operating points, then $P \sim N^3$, which would lead to a rundown behavior of:

$$\frac{dN}{dt} = \frac{k}{J}\frac{N^2}{(2\pi)^2} \rightarrow \int N^{-2} dN = \frac{k}{J(2\pi)^2}\int dt + c \rightarrow N(t) = -\frac{1}{-\frac{k}{J(2\pi)^2}t - \frac{1}{N_{t=0}}} \tag{10}$$

Regarding the proportionality factor (k) for power and speed, this factor is fairly constant, no matter where on the operating map the rundown event starts. Thus, the rate of deceleration, which is approximately determined by the inertia and the proportionality factor, is fairly independent of the operating point of the compressor when the shutdown occurred; i.e., the time constant (dN/dt(t=0)) for the rundown event is proportional to k/J. However, the higher the surge margin is at the moment of the trip, the more head increase can be achieved by the compressor at constant speed.

From this complete model, some simplifications can be derived, based on the type of questions that need to be answered. Obviously, for relatively short pipes, with limited volume (such as the systems desired for recycle lines), the pressure at the valve and the pressure at compressor discharge will not be considerably different. For situations like this, the heat transfer can also be neglected. The set of equations then is reduced to:

$$\frac{dp_2}{dt} = \frac{k \cdot p_2}{V}\left[Q - Q_v\right] \tag{11}$$

This means that the discharge pressure change depends on the capability of the valve to release flow at a higher rate than the flow coming from the compressor. It also shows that the pressure reduction for a given valve will be slower for larger pipe volumes V (Figure 6).

The model described above, which contains and accounts for all physical features of the discharge system, can be simplified even further to determine whether the combination of discharge volume and valve size can prevent the compressor from surge during an ESD. Thus, it allows the two important design parameters to be easily varied to avoid surge during ESD. The surge valve size and opening speed can be increased for a given discharge volume or the maximum allowable discharge volume for a given configuration of valves and compressor characteristic can be limited.

A conceptual problem occurs when simulating gas turbine applications. After the fuel is shut off, the gas turbine will still produce power due to its thermal inertia. Data in Figure 7 shows that the power turbine loses speed very slowly for about the first 300ms, and only then is decelerated rapidly by the gas forces counteracting the inertia. Data of this type can only be determined experimentally, therefore Eq. 10 is a useful tool in the absence of such information, because here, only the more readily available train inertia is required.

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The simplified model avoids the calculation of the instant compressor speed by a prescribed known deceleration rate (for example the behavior displayed in Figure 7). The deceleration in Figure 7 is based on test data, and allows to describe the initial speed reduction of a gas turbine better. The data indicates a loss of 30% speed within the first second after the fuel supply is shut off. This is,similar to data from Bakken et. al. [6], where the gas turbine driven configurations lost about 20-to-25% speed in the first second, while the electric motor driven configuration lost 30% speed in the first second As a result of the loss of 30% speed, the head the compressor can produce at the surge line is about 50% lower than at the initial speed, if the fan law is applied Obviously, the operating point of the compressor at the nitiation of the ESD plays a role, too.

Any ESD is initiated by the control system. As a result of various delays in the system (fuel valve to shut completely, hot pressurized gas supply to the power turbine seizes, opening time of the recycle valve), ESD data show that the surge control valve reaches full open and the beginning of deceleration of the power turbine / compressor are considered to happen simultaneously. This is the starting time (T_0) for the model.

Usually, the suction volume is more than three orders of magnitude greater than the discharge volume and is therefore considered at a constant pressure. The general idea is now to consider only the mass flow into the piping volume (from the compressor) and the mass flow leaving this volume through the recycle valve. Since the gas mass in the piping volume determines the density and, thus, the pressure in the gas, we can for any instant see whether the head required to deliver gas at the pressure in the pipe volume exceeds the maximum head that the compressor can produce at this instant. Only if the compressor is always capable of making more head than required can surge be avoided.

A further simplification can be made by splitting the flow coefficient of the recycle valve (c_v) into a part that is necessary to release the flow at the steady-state operating point of the compressor ($c_{v,ss}$) and the part that is actually available to reduce the pressure in the piping volume ($c_{v,avail}$).

The first stream and, thus, $c_{v,ss}$ of the valve necessary to cover it are known. Also known is the c_v rating of the valve. Thus, the flow portion that can effectively reduce the backpressure is the determined by the difference:

$$c_{v,avail} = c_v - c_{v,ss} \tag{12}$$

The model is run at constant temperature. Most of the compressor systems modeled contain aftercoolers. The thermal capacity of the cooler and the piping are much larger than the thermal capacity of the gas; thus, the gas temperature changes are negligible within the first second.
The rate of flow through the valve is calculated with the standard ISA method [7][3]:

$$Q_{std} = 1360 \cdot F_p \cdot c_v \cdot Y \cdot \left[\frac{dp}{p_2} \cdot \frac{1}{SG \cdot T_2 \cdot Z_2} \right]^{0.5} \tag{13}$$

The compressibility is calculated with the Redlich-Kwong equation of state. The flow calculated above in each step of the iteration is then subtracted from the gas contained in the discharge and a new pressure in the pipe volume is calculated. Depending on whether the system has achieved a reduction of head of at least 56% in the first second, the volume of the system is either increased or reduced by a small margin. Thus, the calculation yields the maximum allowable piping volume for the set parameters that will not cause surge at ESD.

TEST DATA

Data were available for a situation where the recycle valve failed to open at a shutdown situation. The complete model was run against the data and the results are shown in Figure 8. The model predicts initially a faster deceleration than indicated by the test data. This is likely due to some residual power provided by the power turbine even after the fuel supply is shut off, as mentioned above. Since the purpose of the calculations will be to determine the capability of a recycle system to avoid surge, this deviation is acceptable. The onset of surge is predicted quite accurately. Interestingly enough, even the post-surge behavior is captured quite well, despite the fact that no particular effort was made to optimize the compressor characteristics for operating points at lower than surge flow and reverse flow. Both data and model show the characteristic flattening of the speed line in surge, largely due to the fact that the impeller absorbs less power in these situations. The presence of surge in the data was determined by the analysis of vibration data taken at the compressor bearings.

SIMULATION

The model described earlier can be used in a software simulation program to rapidly evaluate whether a selected valve is sized correctly for the piping volume.

The model can also iteratively determine the maximum allowable discharge volume for a given valve configuration. This is important, because the valve size can be determined early in the project phase. With a known valve configuration, the station designer can be provided with the maximum volume of piping and coolers between the compressor and the check valve, that allows the system to avoid surge during an ESD.

[3] Qstd is the standard flow. Fp is the piping geometry factor. It is usually not known and can be assumed to be 1. The pressure is assumed to be constant in the entire pipe volume. It is thus the same just upstream of the valve and at the discharge pressure of the compressor.

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The calculation requires specification of the head-flow-speed relationship of the compressor, and the definition of the surge line as a function of either compressor speed, compressor head or compressor flow. Further, the valve needs to be described by its maximum capacity (Cv), as well as by its capacity as a function of valve travel and the opening behavior, including the delay. The discharge check valve is assumed to be closed as soon as the recycle flow exceeds the compressor flow, i.e., once the depressurization begins. The calculation procedure is started by initiating the deceleration of the train and the valve opening. For each time step, the compressor head and flow, based on speed and system pressures, and the flow through the valve, based on system pressures and valve opening, are calculated. The mass of the gas trapped between the recycle valve and compressor discharge is subsequently determined, yielding a new discharge pressure. If surge occurs, i.e., if the flow drops below the flow at the surge line, the backwards flow through the compressor is assumed to increase with time in surge, with a recovery once the required head drops 1% below the head at surge. The modeling of the backwards flow is not critical, and is only made to avoid numerical instabilities, because the only information that is expected from the model is whether or not the compressor will surge for the given configuration.

Figures 7 shows the deceleration of the engine's power turbine, and thus the driven compressor, following an ESD. Also shown is the response of the recycle valve.

Figures 9, 10, and 11 show typical results of these simulations. In Figure 15, the discharge volume is small enough, and while the actual flow of the compressor approaches the minimum allowable flow (surge flow) at about 500ms after the initiation of the ESD, so surge can be avoided. In Figure 10 the compressor surges about 700ms after the initiation of the ESD. For this configuration, either the valve size has to be increased, or the discharge volume has to be reduced, to avoid compressor surge during an ESD. In Figure 11, the system is severely under-designed and will require significant changes including the possible addition of another valve in a hot bypass mode

OTHER ISSUES

In the present study, we assumed that the check valve downstream of the compressor is closed immediately, so that no additional flow enters the system from downstream the check valve. This may not always be true. Check valves with anti-slam trim, for example, essentially cause a delayed closing of the check valve. The flow through an open or leaking check valve can overwhelm the capability of the surge valve to reduce the pressure on the discharge side of the compressor [8].

The other issue (which the simplified model presented here didn't have to address) is that pressure waves often are assumed to be linear waves. Linearizations are admissible for small pressure disturbances, however, massive pressure disturbances are not readily described by linear wave equations [9].

CONCLUSIONS

A model to simulate shutdown events was developed and used to define simpler rules that help with proper sizing of upstream and downstream piping systems, as well as the necessary control elements. The compression system is analyzed, thus verifying the model and the simplifications. The model coincides well with the data, particularly with regards to proper prediction of surge events. The inaccuracies and limitations inherent in the current model are only problematic if the entire rundown process needs to be described. The goal in this paper, to determine whether a system will surge during ESD situations, can be achieved with either version of the simulation.

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FIGURES

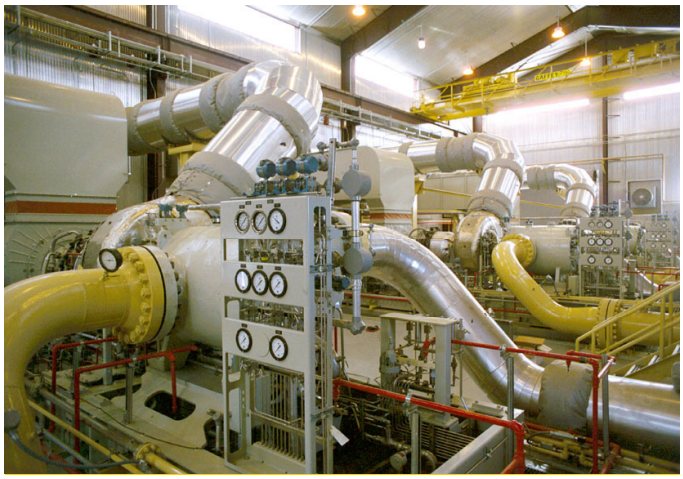

Figure 1- Compressor station with 3 Centrifugal Compressors Driven by Gas Turbines.

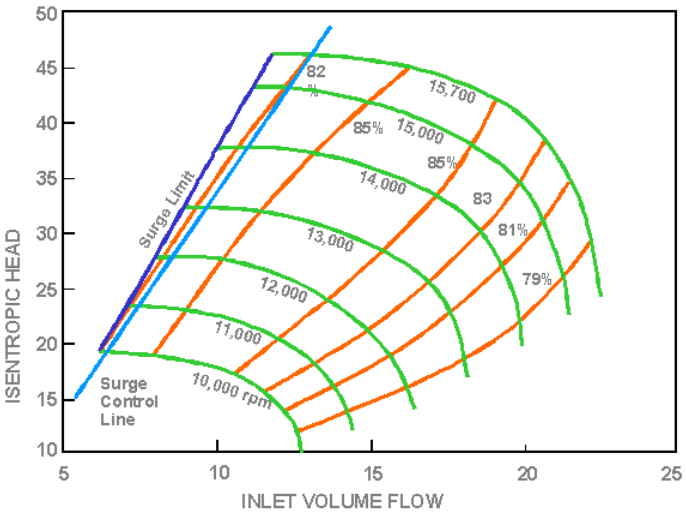

Figure 2- Typical compressor map

Figure 3- Rotor System , consisting of the power turbine, the coupling and the compressor rotor.

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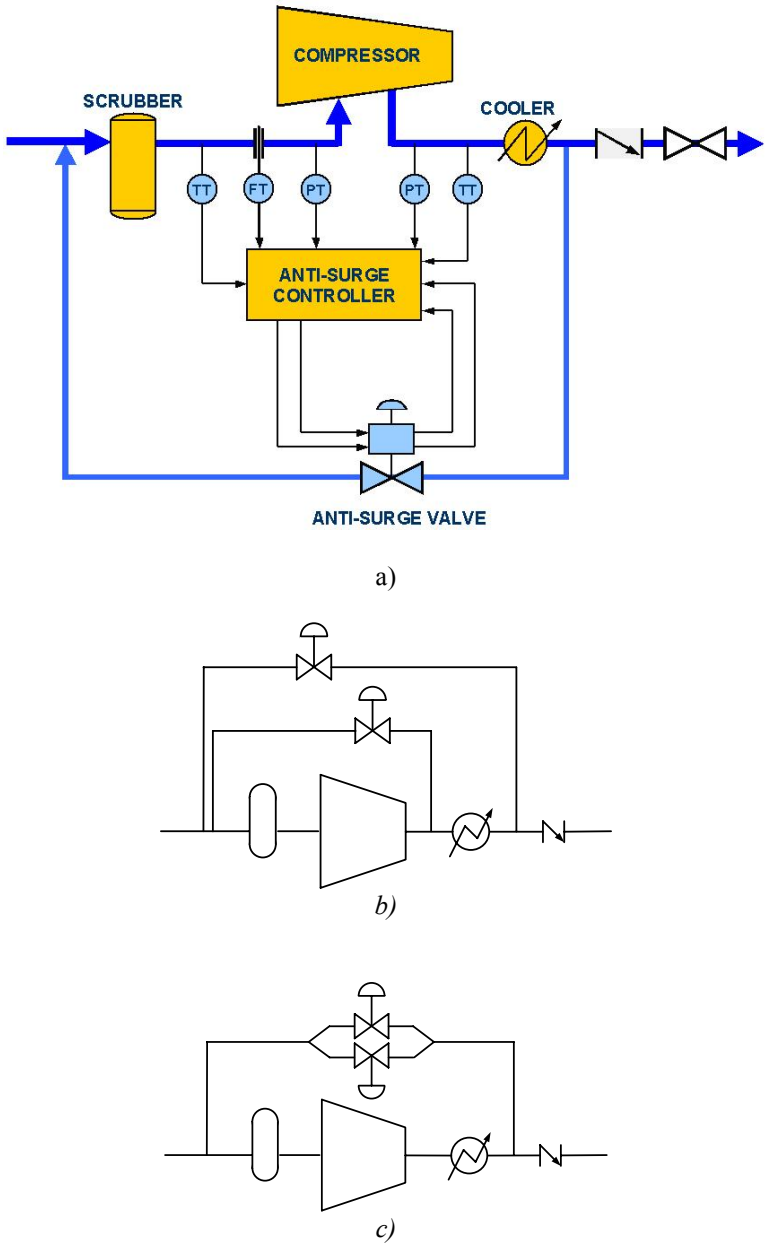

Figure 4 - Anti-surge and recycle system: a) cooled recycle with single valve, b)hot and cold recycle valve arrangement, c)parallel recycle valves.

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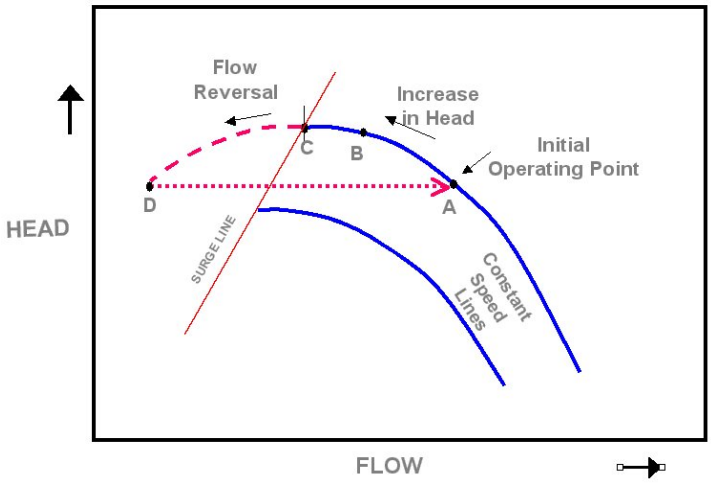

Figure 5 - Simplified surge cycle

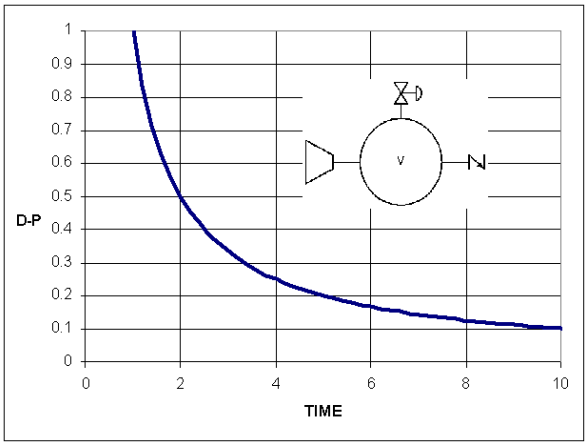

Figure 6 - Simplified System Model for Surge Control System

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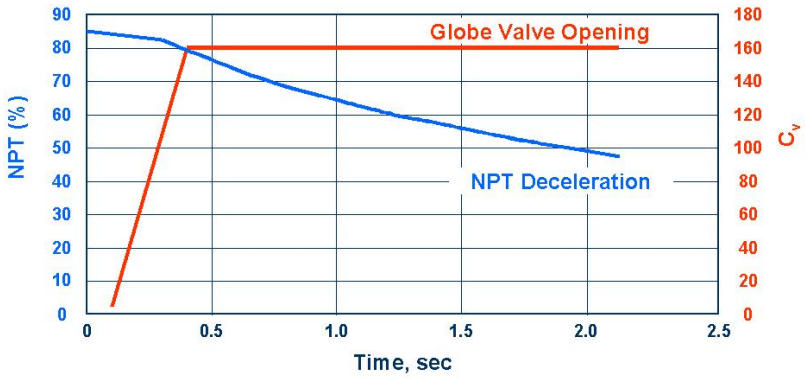

Figure 7 - Valve Opening and Train Deceleration.

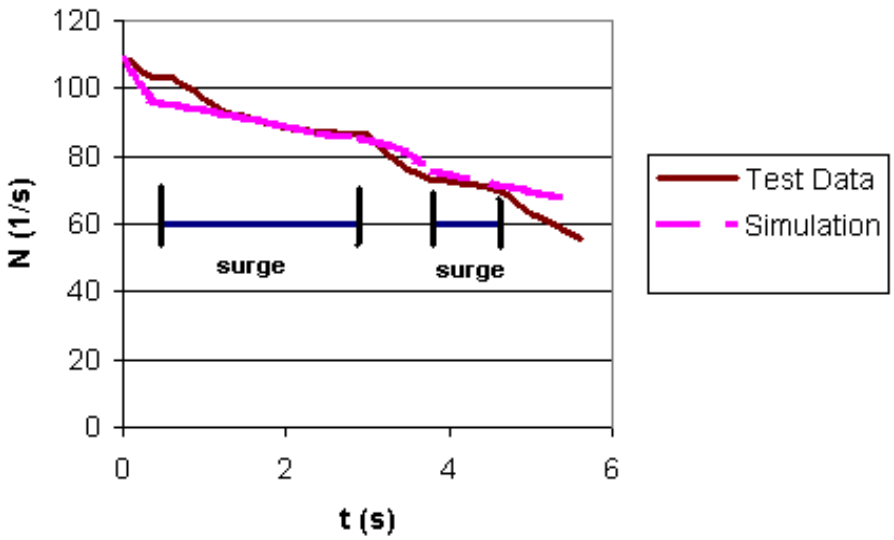

Figure 8 - Emergency shutdown against closed recycle valve – Test data versus simulation with complete model. Time spans in surge based on vibration data from test.

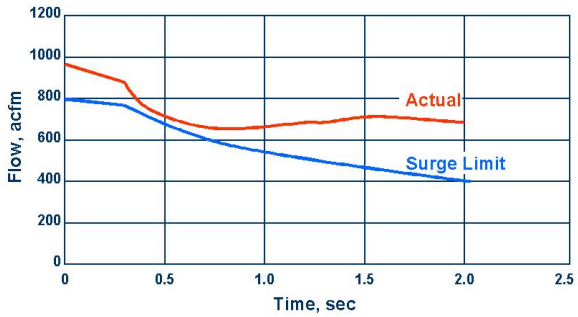

Figure 9 - Actual Flow And Flow At The Surge Line During ESD – Surge Avoided

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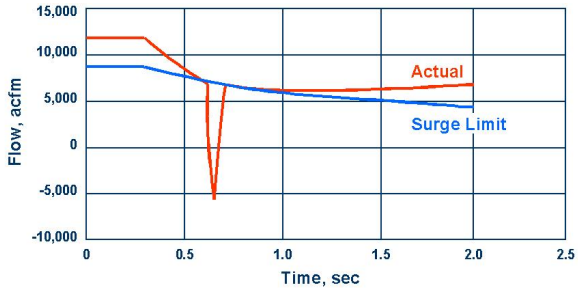

Figure 10 - Actual Flow And Flow At The Surge Line During ESD. Compressor - Surge At 0.7 sec.

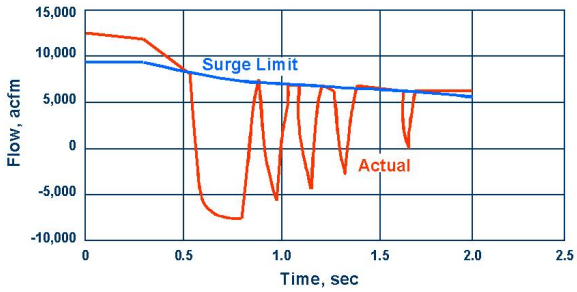

Figure 11 - Actual Flow And Flow At The Surge Line During ESD. – Multiple Surges.

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