DYNAMIC SIMULATION AND TESTING TO ASSESS RUNDOWN SPEED OF A COMPRESSOR

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ABSTRACT

Dynamic simulations are frequently conducted to verify the behavior of compressor stations during transient events. The most difficult situation to analyze is the event of an emergency shutdown, due to the very fast transients. Unfortunately, there is very little documented data that compares simulation results and the actual behavior of the station. We will evaluate available data sets for 3 different studies. The studies include a shutdown of a compressor against a closed recycle valve, a very well documented and published study on the emergency shutdown of a compressor in a test facility, and new data from a compressor station where a major station modification led to a dynamic simulation and subsequent verification. The latter will be described in some detail.
The key finding is, that a difficulty, and a major source of inaccuracy lies in the correct prediction of the speed decay for the compressor.

INTRODUCTION

"The worst case scenario for a compression system is an emergency shutdown (ESD), particularly if the compressor is already operating close to surge when the driver shutdown occurs" (Kurz and White, 2004). When a problem occurs that requires to de-energize the system as fast as possible, the fuel to the gas turbine driver, or the electric power to the electric motor, are instantly shut off. In that situation, the inertia of the drive train will keep the train running, but the running speed is reduced fast. Many simulations assume a loss of 30% speed in the first second. This means, that the compressor quickly losses its capability to produce head. In fact, a loss of 30% speed equates to a 50% loss of head making capability. To prevent the compressor from surging the discharge pressure has to be lowered fast enough, so the system head does not exceed the compressor can produce at its instantaneous speed (Kurz and White, 2004). This requires a fast acting, relatively big valve to recycle gas from the discharge side of the compressor to the suction side, thus reducing the system head (Figure 1).

Surge Avoidance

The compressor map for a speed controlled compressor identifies the possible operating points for the machine. The possible conditions are limited by maximum and minimum operating speed, maximum available power, choke flow, and stability (surge) limit (Figure 2). Surge is the flow reversal within the compressor. It is often accompanied by high fluctuating load on the compressor bearings and should be avoided to protect the compressor. A surge avoidance system ("anti-surge-control") consists of a recycle loop with a fast acting valve ("anti-surge valve"). When the control system detects the compressor approaching its surge limit , the anti surge valve is opened. The control system uses the measured suction and discharge pressure and temperature, together with the inlet flow into the compressor as input to calculate the relative distance ("surge margin" or "turndown") of the present operating point to the predicted or measured surge line of the compressor "(Kurz and White, 2004).
The control system is set up to open the recycle valve at a preset turndown (typically 10%), thus keeping the compressor from crossing the surge line. This is the normal control function of a compressor control system. A well designed system acts without upsetting the process, and can allow the compressor station flow to be reduced down to zero station flow.
Following the description in White et al (2006):

"There are five essentials for successful surge avoidance:

1. A Precise Surge Limit Model: It must predict the surge limit over the applicable range of gas conditions and characteristics.
2. An Appropriate Control Algorithm: It must ensure surge avoidance without unnecessarily upsetting the process,
3. The Right Instrumentation: Instruments must be selected to meet the requirements for speed, range, and accuracy.
4. Recycle Valve Correctly Selected for the Compressor: Valves must fit the compressor. They must be capable of large and rapid, as well a small and slow, changes in capacity.
5. Recycle Valve Correctly Selected for the System Volumes: The valve must be fast enough and large enough to ensure the surge limit is not reached during a shutdown. The piping system is the dominant factor in the overall system response. It must be analyzed and understood. Large volumes will preclude the implementation of a single valve surge avoidance system."

The situation studied in this paper however involves so a called Emergency Shut Down (ESD) . Here, the fuel supply to the gas turbine, or the electric power to the electric motor are instantaneous shut off to protect the equipment, and the compressor station. Of the requirements applying to the surge control system above, items 4 and 5 (valve sized for the compressor, and the system volume), become dominant.

The dynamic situation arising is described in detail in Kurz and White(2004). Upon fuel shutoff, the inertia of the rotor system (including the compressor rotor, the power turbine rotor or the rotor of the motor, the inertia of the coupling, and, if present, of a gearbox) keeps the compressor running. Since the compressor still compresses gas, it has to provide the power necessary. Therefore, the rotor will slow down. On the other hand, the check valve on the discharge side will close automatically, so there is a trapped volume of gas in the piping between the compressor discharge, the check valve, and the recycle valve (Fig. 1).

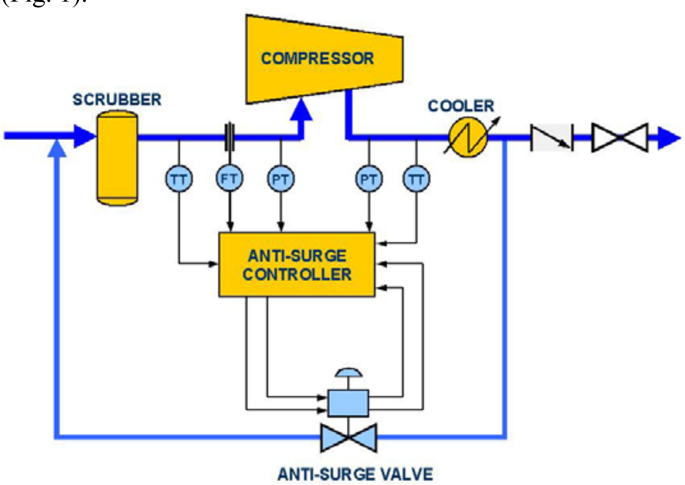

Figure 1. Surge Avoidance System Schematic

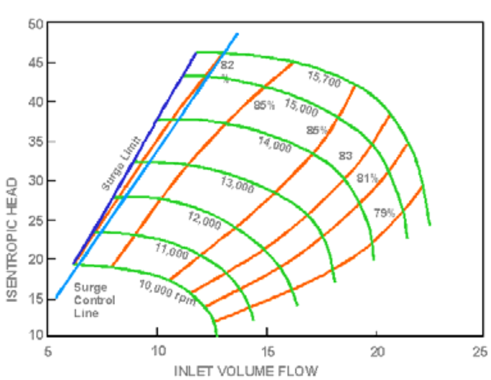

Figure 2. Typical Performance Map for a variable Speed Centrifugal Compressor

For the dynamic behavior of compressors it is important to understand that the operating point of the compressor is determined by the head imposed by the suction and discharge pressure of the system. The compressor, at a given running speed, reacts to the imposed head with a certain amount of flow. The compressor consumes power based on said head and flow (and its efficiency at the operating point.

The pressure at the discharge side depends on the amount of gas trapped in the fixed volume. The compressor adds gas to this volume, while the recycle valve, depending on its instantaneous flow capacity, will reduce the amount of gas in the volume.

Dynamic Simulations

Because the avoidance of surge is important for the safety and reliability of a compressor station, the dynamic behavior of the system during emergency shutdowns is often studied in the planning and design phase of the compressor station. This allows to verify in particular whether the valve size, opening speed and the arrangement of a single or multiple valves, is appropriate. In this phase of the project, it may still be possible to modify the piping layout.

One of the key uncertainties in these simulations is the speed decay of the compressor. To some extent, this is due to the fact that de-energizing the driver does not lead to an instant loss of power to the compressor. In the case of a gas turbine driver, there is a significant amount of power supplied for about 200 to 300 ms after the fuel valve is closed. This is due to the thermal mass of the driver.

More importantly, even the estimate of the compressor speed decay in the absence of driver power is often a problem.

The frequently used approach is to use a fixed decay rate for the compressor. Obviously, the faster the speed decay, the more difficult is the task for the recycle valve to keep the compressor form surging. A decay rate of 30% speed loss per second is often used in simulations. This rate has proven to be rather conservative. While this is generally comforting, it may also force systems to be more complex and expensive than necessary.

Another issue that also should be considered comes from the assumption of the compressor operating point at the moment the initiation of the shutdown. If that point is close to the surge line, the system has to react very fast to avoid surge (Moore et al, 2009). It is thus important to select the starting points for the study so that they reflect realistic, actually achievable operating conditions.

The governing equations consider the following:

-Mass conservation in the piping system

-Torque balance of the rotor system

-Friction losses in the piping

-wave propagation in the pipe.

-Flow through the valve (based on the instantaneous valve flow area, and the pressure ratio over the valve)

Regarding speed decay, the situation during rundown is as follows: The inertia of the rotor keeps the rotor running, while the power the impeller transfers to the gas slows the compressor down. In other words (Kurz and White , 2004):

$$P = T \cdot N \cdot 2\pi = -(2\pi)^2 \cdot J \cdot N \cdot \frac{dN}{dt} \quad (1)$$

If the rundown would follow through operating points along the fan law, then P=k N^3, which would lead to a rundown behavior of:

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

The time constant (dN/dt(t=0)) for the rundown event is proportional to k N^2(t=0)/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."

Dimensional analysis suggests a parameter K related to deceleration (Moore et al,2009):

$$K = \frac{P}{J(2\pi N)^3} \quad (3)$$

This indicates that at a given speed the speed decay rate will be slower for a large compressor (large inertia J) consuming a small amount of power, than for a small compressor consuming a high amount of toque. Thus , a large value of K should lead to a high rate of deceleration. The compressor trains in the three studies presented represent a quite large range of K, from about $1 \cdot 10^{-4}$ to about $4 \cdot 10^{-4}$. Therefore, one would expect some differences in the speed decay between the machines. However, for many applications , especially for compressors directly driven by gas turbine, the dominant inertia comes from the power turbine. For a typical 15,000 hp gas turbine driving a 2 stage pipeline compressor, the power turbine inertia may be 10 to 15 times higher than the compressor inertia.

DATA

To verify simulations, we are analyzing data from three tests, with the goal to get a better estimate of reasonable deceleration rates.

STUDY 1

Figure 3 shows test data from a test where the recycle valve did not open. The compressor was a two stage pipeline compressor, driven by a 10,000hp two shaft gas turbine. During commissioning, it was running at part load and about 60% speed when an emergency shutdown was initiated. The anti surge valve did not receive the signal to open, and thus stayed closed during the entire run down phase. The compressor started to decelerate, and subsequently surged repeatedly. Because the compressor was instrumented for a performance test, operational data as well as radial vibration data was sampled at a high rate. The vibration data was used to identify the time periods the machine went into surge. We can identify two time periods where the compressor actually surged (Kurz and White,2004).

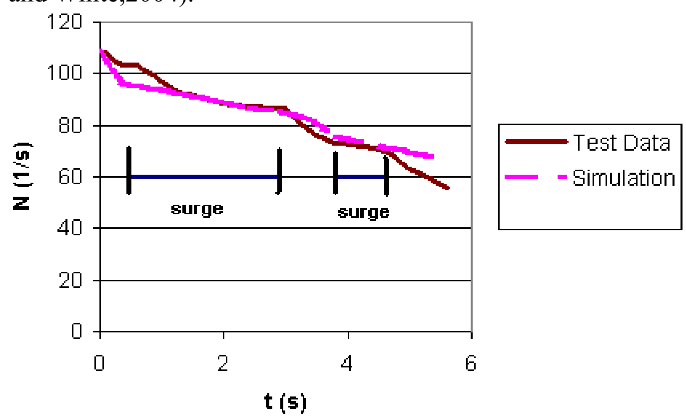

Figure 3:Emergency shutdown against closed recycle valve – Test data versus simulation with complete model. Time spans in surge based on vibration data from test (Kurz and White, 2004).

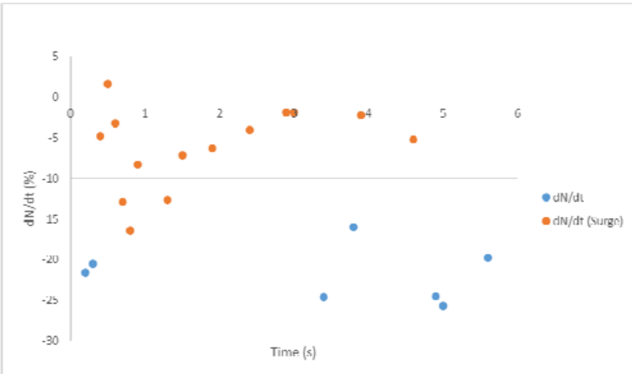

Figure 4: Speed decay dN/dt for data in Figure 3

These segments show a distinctly slower speed decay, consistent with the fact that the compressor consumes less power during reverse flow situations (Aust, 1988). In segments where the compressor does not surge, the speed decay was generally between 20 and 25 % speed per second, with an average of 21.8%.

STUDY 2

Moore et all (2009) presented data from a dedicated test to identify the behavior of a compressor station during an emergency shutdown, which was subsequently analyzed further by Blieske et al (2010). The test bed used a 1500p gas turbine driver, powering a small single stage centrifugal compressor with a single, 7.5 in high flow impeller, running at about 20000rpm. For the test, the absorbed power in put was about 200hp.

The test was conducted to generate a reference data set to calibrate dynamic simulation software.Data acquisition includes use of a 24-bit A/D high frequency instrument with 16 channels of parallel sampling.The compressor speed was measured through a magnetic pick-up sensor located on the shaft of the power turbine of the driver. The frequency signal was converted to a voltage signal by using a frequency-to-voltage converter. Figure 5 shows typical test data: The traces of the operating points of the compressor during ESD events form different starting points are superimposed to the compressor map. It shows the operating points initially moving towards the surge line. Eventually, the recycle valve allows the discharge pressure to drop enough so the operating point starts to move away from the surge line.

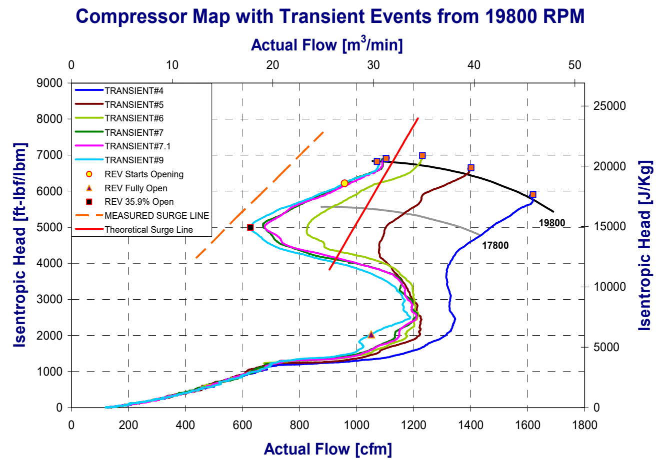

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Figure 5. Transient Shutdown Loci Measurements for 19,800 rpm (Moore et al , 2009)

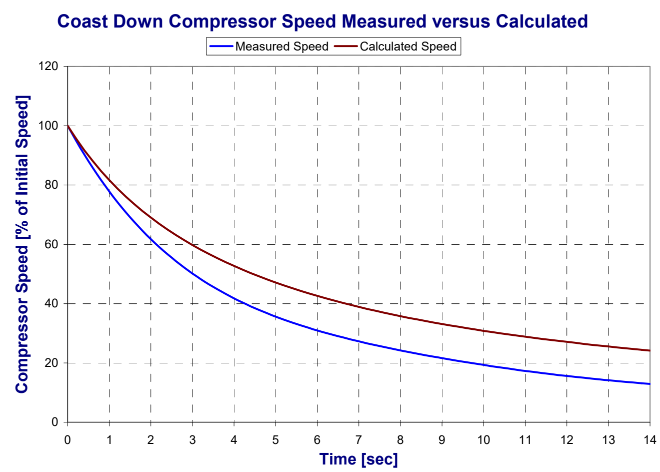

Figure 6. Measured vs. Calculated Speed Decay (Moore et al, 2009)

Using the approximation in Kurz (2004), and assuming all operating points of the compressor follow the fan law, the compressor power may be expressed as outlined in Equation 2 above:

$$P_{compr}(t) = P_o \left(\frac{N}{N_o} \right)^3 \qquad (4)$$

with P_o and N_o the steady-state power and speed at the initial operating point, respectively. This allows the use of Eqn 4 above, and the results are compared with the measured results in Fig 6.

For both the as tested and calculated data we see an initial speed decay of about 20% per second, with the calculation slightly under estimating the actual speed decay.

STUDY 3

In a study by Zwerver et al (2017), a dynamic transient analysis was performed to model an existing compressor station that was to undergo a major engineering change that included piping, control valve, and compressor drive train modifications.

The compressor station consists of three turbo driven centrifugal compressor units operating in parallel. Through the turbine driver modifications, the compressors had increased capacity due to an increase in available power. In order to insure safe system operation with the new modifications, the end user expressed interest in modeling the compressor system with the intended modifications and simulate various transient events that included emergency shutdown, compressor start up, compressor normal shut down, and upset scenarios to model the modified hydraulic behavior and response of the system. The primary concern of the collective system modifications was the need to maintain adequate surge protection during different normal operation and upset conditions. The simulations were performed with a transient hydraulic modeling software that utilizes compressor performance curves, anti-surge control logic, PID controls, valve characteristics, mainline piping, and compressor station operating sequences. Following the transient simulation study, the actual dynamic behavior of the system was measured through field testing. Thus, this study will provide a rare example of where simulation analysis and field test data can be compared.

The end user was conducting a major engineering project to their Netherlands Wieringermeer Compressor station that included rerouting of the recycle piping , adding compressor suction flow elements, and changing the process requirements. This was done particularly to accommodate the increased flow capacity of the system. Of particular importance for this discussion is the recycle loop from each compressor discharge to common compressor suction header, and the anti surge valves for each compressor. This constitutes a typical arrangement found in many compressor stations (Figure 7).

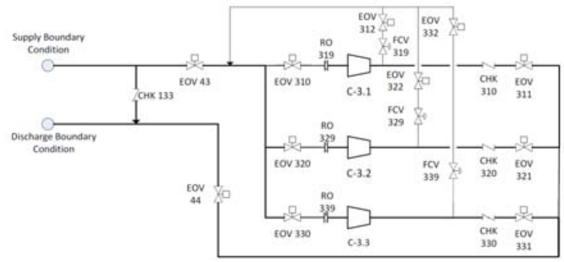

Figure 7: Compressor Station General Schematic (Zwerver et al, 2017)

The compressor station operates with three parallel centrifugal compressors (C-3.1, C-3.2, and C-3.3). The existing compressor section of each of the compressor trains remained unchanged with the original compressors, while the gas turbine drivers were upgraded to 4700hp machines, with lean-premix combustion systems. Through this modification which increased the available power, the compressor train was projected to increase its capacity to approximately 1,000,000 Nm3/hr per unit. The main purpose of the system dynamics analysis is to ensure that the design of the anti-surge control system is adequate to prevent the compressors from experiencing a surge event.

A coast down analysis was performed to estimate the compressor train inertia based on the steady state compressor operating conditions (speed and compressor power utilized) and actual coast down data. With the operating conditions fixed, the provided coast down data can be matched to provide an estimate of the compressor train inertia that is active during a compressor coast down. The estimated inertia can then be utilized as a fixed parameter for other operating conditions (speed and compressor power utilized) to predict the compressor coast down trend for any steady state operating condition prior to an emergency shutdown. The coast down field data was provided by the end user with an emergency shutdown from an initial 83.3% maximum compressor speed. Figure 8 shows the comparison of the end users field data against the compressor manufacturers expected coast down trend. The OEM’s expected coast down is more aggressive in its inertia speed decay making it more difficult for the anti-surge system to reduce compressor head and increase suction flow during the first seconds of the transient, which are the most critical. The OEMs expected coast down was initially used in the ESD simulations to model the worst case scenario.

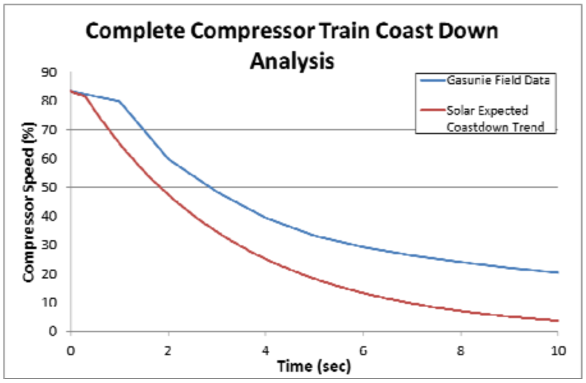

Figure 8: Coast down trend comparison (Zwerver et al 2017)

The event sequencing for the emergency shutdown was developed from data provided by the end user. The shutdown sequence controls various valves during predetermined elapsed time after the event or after a operating condition has been achieved, as shown in Table 1. The duration of the valve events was based on the stroke time of the actuating valve. While the duration of the compressor coastdown is approximately 120 seconds the most critical and probable surge occurance would occur at the beginning of the event within the first seconds. This is due to the energy balance that the compressor, anti-surge valve, and the compressor piping experience during the transient event. Between the volumes upstream and downstream of the compressor exists a high pressure differential, while the compressor speed decay reduces the energy that the compressor is able to impart of the gas flow. If the anti-surge valve is not able to reduce the piping energy state to maintain forward flow the compressor will be overcome by the piping system and experience surge. In any case, any surge event at the low speeds and pressure differentials as they exist after the first few seconds (Fig. 8) would not be violent enough to be of concern for the integrity of the system.

The details of the valves, the valve maximum Cv, valve cage trim, and the stroke time, were derived based on the available valve data sheets. In some instances, typical values were used particularly for main isolation valves. The dynamic behavior of the ASV was specified by the OEM based on port size. Equation 5 below defines the time the valve needs to open to 63.2%, including a 100ms delay to de-energize the ASV at the start of the ESD.

$$T_{63.2} = 100 + 100\sqrt{in\ of\ port\ size} \quad (5)$$

The geometry of the piping system and the recycle loop were based on the isometrics provided by the end user, thus defining length, outside and inside diameter. By representing the physical piping layout with 1-D piping blocks that capture the dimensions of the pipe the model is able to predict time delays of the recycle flow from the anti-surge valve to the compressor suction nozzle. This project provided a unique opportunity to compare predictions with the actual equipment behavior and simulation results. Figure 9 is used to compare the measured speed decay during a shutdown with different assumptions (20% speed decay per second and 30% speed decay per second). Shutdown was initiated at 10569 seconds.

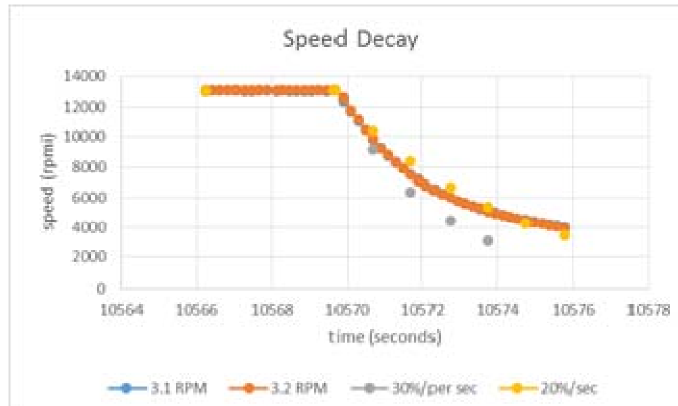

Figure 9: Speed decay. Test data for compressor 3.1 and 3.2, compared with different assumptions on decay.

The speed decay is a crucial factor in the dynamic behavior of compressors during shutdown events. A common assumption (Kurz and White, 2004) of a decay of 30% speed per second was initially used in the simulation. As can be seen , the actual speed decay is closer to 20% speed per second. In other words, the simulation used a more conservative approach.

Figures 10 and 11 show the simulated operating trends in the compressor map, comparing the simulation with field data. The behavior actually looks qualitatively similar to the results presented in study 2: The locus of the operating points first moves rapidly to lower flows, and closer to surge, until the anti-surge valve starts to effectively relief the discharge pressure.

Table 1: Valve Timing During Compressor ESD

Sequence of Events	Start Time [s]	Duration [s]
Steady-state period	0	300
Coast down begins	300	120
Recycle valve opens	300	2
Close compressor discharge block valve	304	30
Close compressor suction block valve	NPT < 5%	33.5
Close compressor recycle block valve	NPT < 5%	22.5

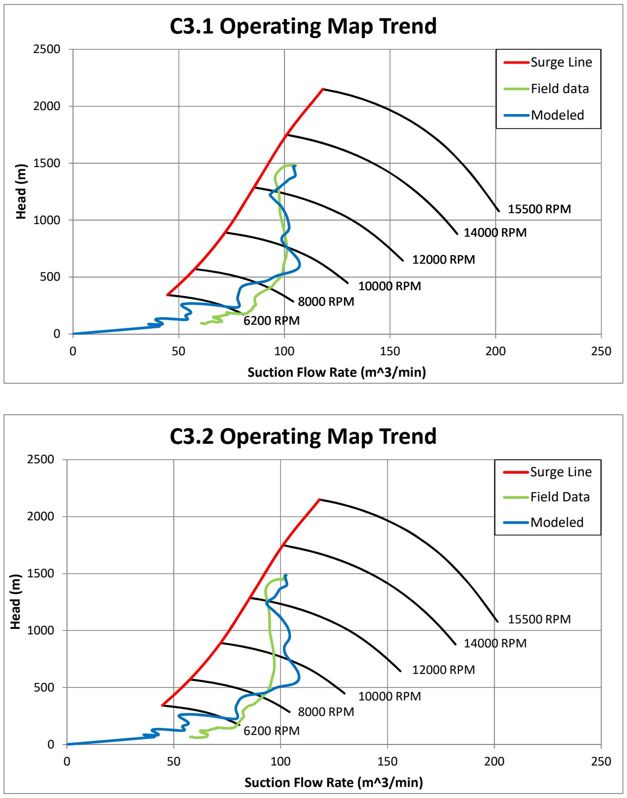

Figure 10: Simulated operating lines in the compressor map for coastdown after ESD, comparing the simulation with field data for compressors 3.1 and 3.2. (Zwerver et al, 2017)

These simulations make use of field observations that are often not available for the initial simulations. In this particular case, increased resistance in the recycle line was observed in the field data. This is either caused by fitting resistance (such as reducers and elbows, not in the original model) or due to a possible a sonic condition at the recycle valve smallest flow area. Both effects reduce the actual flow through the recycle line and limit recovery flow needed for surge avoidance. After this correction, the simulated run down curve matches very closely the observed run down curve in the test. The crucial feature in such a simulation is the minimum surge margin during run down, in other words, the closest approach of the operating point to the surge line. The simulation matches the test data well. The anti-surge system avoids surge of the compressors under the tested conditions.

Another feature was verified in the simulation. There is usually a delay between the open signal reaching the valve and this delay significantly affects the outcome of the study. The system prevents transient surge at shutdown, however, if the delay is increased to 100ms, the operating point almost reaches surge, while a 300ms delay would cause the compressor to surge. It is therefore crucial to specify maximum delay times when specifying the recycle valves (Figure 11).

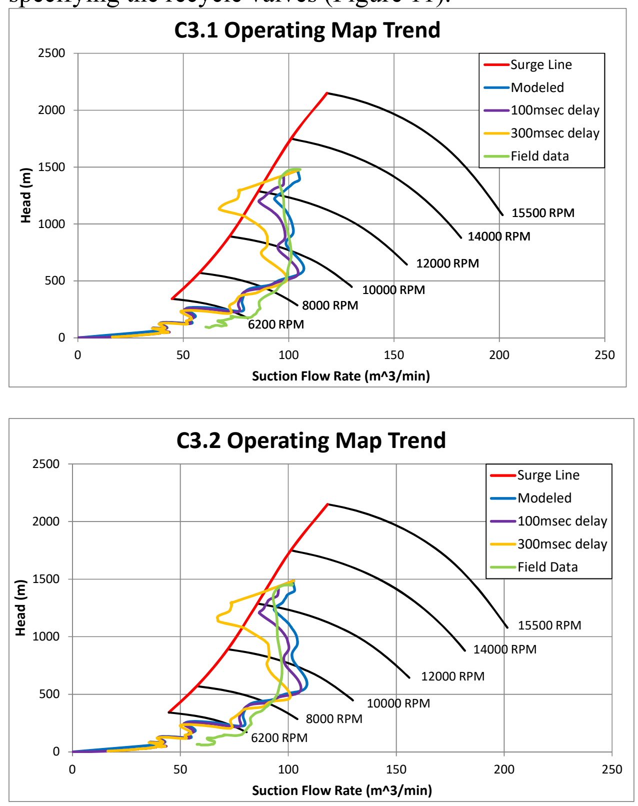

Figure 11: Additional simulations with ASV delays of 100msec and 300msec superimposed.(Zwerver et al., 2017)

Another assumption in these simulations is the almost instant closing of the check valve downstream of the compressor (chk310,320 and 330 in Figure 7). If this action is delayed, either due to malfunction of the check valve, or due to a built in damping feature, the chances of surge during a shutdown increase.

CONCLUSIONS

Data from three separate studies, for different compressor sizes, station layout, and power levels were used to evaluate the speed decay during compressor emergency shutdown. The rate of speed reduction is a crucial parameter for shut down simulations. Evaluating the data reveals that the speed decay rate seems to be consistently fairly close to 20% of speed per second during the initial phase of the shutdown event. The measured speed decay also generally compares reasonably well with the decay rates from simulations where the absorbed compressor power is balanced with the rotor system inertia. Many simulations use a constant decay rate of 30% per second.

This is a more pessimistic assumption, often leading to the requirement for additional valves or larger valves in the project phase.
Furthermore, a detailed study indicates the sensitivity to assumptions about pressure losses in the pipe, and in particular, the speed decay of the compressor after a shutdown. It also emphasizes the importance of short opening delay times for anti-surge valves used to prevent compressor surge in an emergency shutdown.

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NOMENCLATURE

J	= Inertia	$(M\ L^2)$
K	= Deceleration Parameter	(-)
N	= Speed	(T^{-1})
P	= Power	(F L/T)
T	= Torque	(F L)
t	= Time	(T)

ASV=Anti Surge Valve
ESD = Emergency Shut Down
NPT=Power Turbine Speed