Compressor System Check Valve Failure Hazards

155f

By
Craig Thompson,
Ralph King
Equistar Chemicals LP
A LyondellBasell company

Prepared for the AIChE Spring National Meeting
San Antonio, Texas
March 22-25, 2010

155f: Compressor System Check Valve Failure Hazards

Craig R. Thompson, Consulting Engineer, Olefins Process Engineering
Craig.Thompson@LyondellBasell.com
Equistar Chemicals LP, a LyondellBasell company, Morris, Illinois

Ralph E. King, Senior Advisor, Mechanical and Discipline Engineering
Ralph.King@LyondellBasell.com
Equistar Chemicals LP, a LyondellBasell company, Channelview, Texas

ABSTRACT

Catastrophic equipment failure due to overpressure can potentially occur in the event of compression system discharge, interstage, and/or suction check valve failure, coincident with compressor shutdown. Depending on system design and application, overpressure values approaching or exceeding 300% of equipment design are possible. Comparatively, for some equipment even limited overpressure can result in catastrophic vessel failure due to brittle fracture. Additional hazards associated with compression system fail-to-check scenarios include risks associated with excessive flare loading and compressor rotor reverse rotation. In the case of an ethylene refrigeration compressor at a typical ethylene plant, rotor reverse rotation can potentially exceed over-speed limits.

This paper summarizes the risk assessment results based on analysis performed on the three primary compression systems within six different ethylene plants. The methodology used to assess associated risks and system dynamics is presented. Alternative methods for mitigating risks are also discussed along with check valve reliability data. An overview of applicable overpressure protection requirements defined in the ASME Boiler and Pressure Vessel Code is provided. This paper will be of interest to anyone that designs or operates multistage compression systems in the chemical, petrochemical or refining industries.

SUMMARY

The major compression systems within a typical ethylene plant include the Process Gas Compression System (PGC), the Propylene Refrigeration System and the Ethylene Refrigeration System. Compression system configurations, relative volumes, design pressures, relief provisions, check valve locations, and other design factors depend on the plant vintage, the technology licensor, feedstock design slate, and other plant-specific design criteria.

For each of these compression systems, check valves are installed at appropriate locations to prevent reverse flow from the high-pressure discharge system to low-pressure interstage and suction systems on compressor shutdown. Often, the design pressure of the low-pressure system is insufficient to prevent overpressure should a check valve fail to close. Overpressure can occur even with check valve closure of 90% or more. Additionally, with instantaneous reverse flow rates potentially as high as two to three times the compressor design flow rate, existing relief capacity is rarely adequate to prevent excessive overpressure. The magnitude of overpressure can potentially exceed 300% of equipment design pressure, i.e., the maximum allowable working pressure (MAWP)*. Overpressure risk scenarios at this magnitude were determined to exist at LyondellBasell's oldest and newest plants. The risk of catastrophic vessel failure depends on the magnitude and duration of overpressure, the vessel mechanical integrity and the vessel materials of construction (metallurgy). A vessel that has **not** been compromised by corrosion, cyclic fatigue, non-compliant alteration, or other deficiencies may not necessarily fail catastrophically, even at pressures in excess of 300% of MAWP [1]. On the other hand, equipment constructed from carbon steel or other ferritic steels can catastrophically fail at very low overpressure due to brittle fracture failure if conditions cross the vessel's minimum allowable temperature (MAT) curve [2]. Brittle fracture failure is **not** strictly a cold-temperature phenomenon. Beyond catastrophic failure risk mitigation, compliance with ASME Boiler and Pressure Vessel Code (referred to as "Code" within this paper) must also be addressed.

In addition to overpressure hazards, if reverse flow is sustained through the compressor case after the compressor rotor speed decays to 0 RPM, rotor reverse rotation may occur. The magnitude of speed reversal, and therefore the probability and extent of resulting mechanical damage, is dependent on several factors. Some of these factors are differential pressure, flow rate, rotor mass, bearing design, and seal design. Reverse rotation of the rotor, or simply "reverse rotation", does not necessarily result in mechanical damage. However, reverse rotation into speed ranges at or near machine criticals can result in catastrophic bearing or seal failures. Specific to ethylene refrigeration compressors, reverse rotation can approach overspeed limits resulting in catastrophic equipment damage and gas release.

Another hazard created by reverse flow conditions is excessive flare system loading, particularly as associated with the process gas compression system. Combined relief of the compressor feed stream and reverse flow stream may exceed the flare tip and/or flare header design flow resulting in high flare header back pressure. Elevated flare header back pressure compromises the capacity of conventional relief valves as well as relief valves with low set pressures. Of particular concern is the impact on the PGC first-stage suction relief valves, which may result in first-stage suction equipment overpressure.

* It must be noted that the percent overpressure compared to the MAWP and potential consequences are for existing equipment built prior to 1998 when the allowed stresses for certain ferritic steels was increased by Code. The overpressure consequences described in this paper are thus based on stresses defined within Code predating 1998.

Industry data indicates that significant check valve failures can be expected at a frequency between 1/10 and 1/100 years [4]. Significant failures involve gross failures and thus exclude valve seat sealing inadequacies which only result in limited leak-by. Analysis of the systems included within the scope of the LyondellBasell study indicate risk of overpressure well in excess that allowed by Code is not uncommon, should a gross check valve failure occur. However, industry data does not indicate catastrophic compression system vessel failures occurring at a frequency that would be expected based on check valve failure frequency statistics, particularly considering that many check valve maintenance programs are potentially inadequate. One explanation is that the frequency of significant (gross) failure is less frequent in ethylene plants than reported by general industry sources. This potential must be accounted for via a sensitivity analysis during the risk assessment process. A second explanation is that overpressure events have most likely occurred; however, these events have not led to significant vessel damage and thus have not been reported within the industry. As previously noted, equipment in good condition subjected to overpressure in excess of 300% of MAWP is unlikely to fail catastrophically unless brittle fracture risks exist. Permanent vessel deformation can be expected at an overpressure of 190% of MAWP. Additionally, overpressure occurs very rapidly and may be limited over a relatively short duration. Consequently, these very short duration overpressure events are not necessarily detected.

Evaluating the probability of occurrence, the magnitude and consequence of a fail-to-check failure; LyondellBasell Industries has concluded that in some cases additional mitigation of the hazard was warranted. The mitigation alternatives listed below are some of the methods evaluated to reduce the risk of a potential fail-to-check incident.

RISK ASSESSMENT STUDY SCOPE

After relief valve assessment studies identified significant overpressure risks on two different compression systems at two different plant locations, LyondellBasell initiated a study to assess overpressure risks at all of its ethylene plants. The study has completed assessments of 23 different compression systems, in seven ethylene plants, designed by four different technology licensors. The oldest plant was designed in 1968 and the newest plant was designed in 1989. The initial focus of the study was limited to overpressure risk assessment, but as the study proceeded, the additional risks of reverse rotation and flare loading were identified and incorporated into the study.

Each plant and compression system is unique with considerable variability in the nature and magnitude of risks identified on each compression system. The findings from this study are directional only and cannot be generally applied to all plants and compression systems. Each compression system in each plant must be specifically analyzed to assess the presence, frequency and magnitude of hazards associated with check valve fail-to-check failures.

RISK ASSESSMENT PROCESS

LyondellBasell employs a two-step process to assess overpressure risks. Initially a static analysis is performed to assess overpressure viability. This “screening” assessment calculates the instantaneous system settle-out pressure as a function of specified check valve performance, i.e., fails versus holds. Settle-out pressures for both single and multiple check valve scenarios are analyzed The impact of flow out of the system via pressure control valves and relief valves is not taken into consideration during this stage of assessment. This requires an accurate calculation of discharge, interstage and suction system vapor volumes including the piping. Piping volume represents anywhere from a small percentage of equipment volume to twice equipment volume, dependent upon the plant design and layout. Therefore, use of simple factors to estimate piping volume rather than more precise calculation methods can be expected to lead to highly erroneous results.

The following is an example of a static analysis report for a process gas compression system:

Analysis - Process Gas Compressor - XYZ Chemical Company

Process Data:		Disc CV+	Disc CV-	5th Suc	4th Suc	Between CVs	3rd Disc.	3rd Suc	2nd Suc	1st Suc
Pressure	Psig	540	540	290	165	165	165	80	40	10
Density	Lbs/Ft3	2.97	2.97	1.57	0.93	0.93	0.93	0.49	0.28	0.13
Volume	Ft3	15,000	900	3,500	2,300	18,000	4,500	6,000	6,000	150,000

Check valve status:			**Evaluation:**	Density	Pres.,	% of	Brittle Frac.
5th Disc	4th Suc	3rd Disc	Stage	Lbs/ft3	Psig	MAWP	Failure Risk ?
Fails	Holds	Holds	5th suction	2.5	476	153%	No
Holds	Holds	Holds	4th suction	1.5	284	162%	No
Fails	Holds	Holds	4th suction	2.5	476	272%	Yes
Fails	Fails	Fails	4th suction	0.5	79	45%	No
Holds	Fails	Holds	Caustic Tower	1.1	197	113%	No
Fails	Fails	Holds	Caustic Tower	1.8	335	191%	Yes
Holds	Fails	Fails	3rd suction	0.2	32	28%	No
Fails	Fails	Fails	3rd suction	0.4	72	63%	No
Holds	Fails	Fails	2nd suction	0.2	32	43%	No
Fails	Fails	Fails	2nd suction	0.4	72	96%	No
Holds	Fails	Fails	1st suction	0.2	32	75%	No
Fails	Fails	Fails	1st suction	0.4	72	167%	Yes

Note: Caustic Tower between 4th suction check valve and 3rd discharge check valve.

This analysis also assesses the risk of equipment brittle fracture failure. This requires the development of minimum allowable temperature (MAT) curves, the procedure for which is detailed in reference #2.

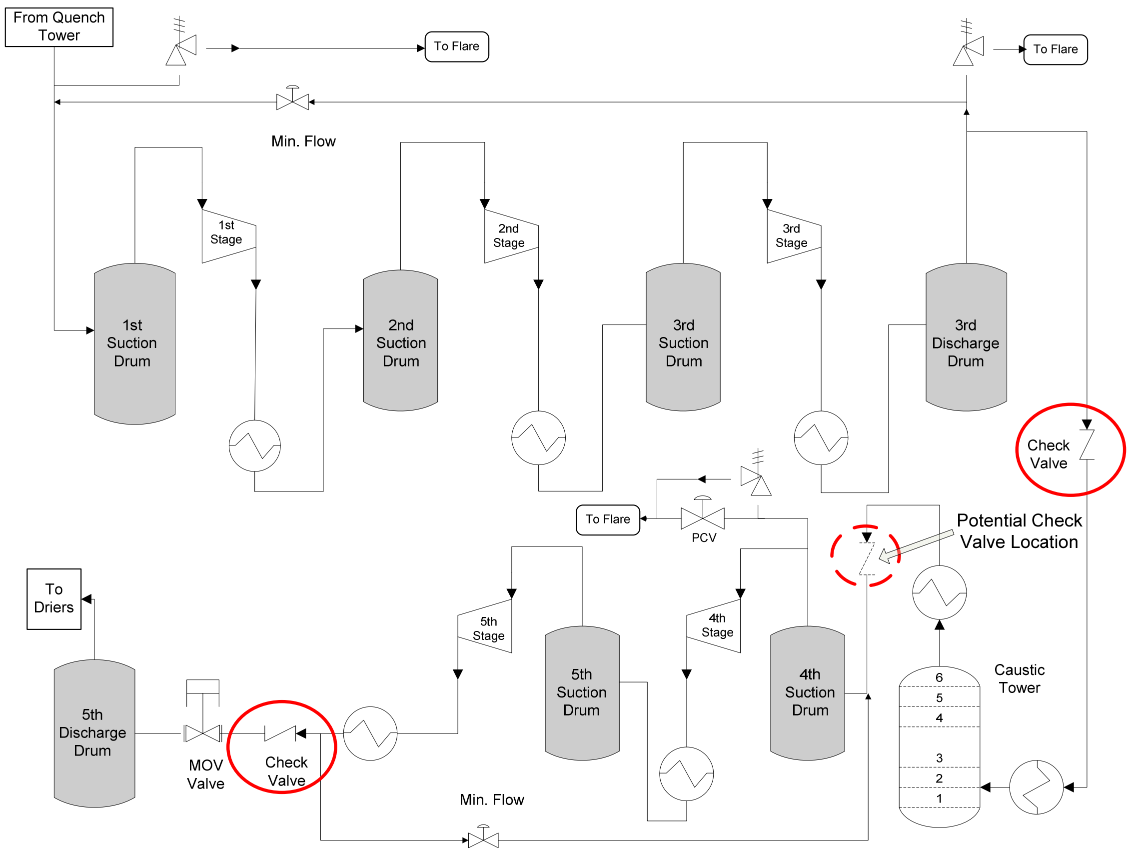
If the static analysis indicates an overpressure risk exists, a more rigorous analysis is performed up to and including dynamic analysis. Dynamic analysis of compressor trip conditions takes into consideration reverse flow rate as impacted by compressor internal geometry, rotor coast-down, as well as interstage piping and equipment flow resistance. Additionally, this analysis includes the impact of relief and vent valve capacity, minimum flow valve capacity and actuation response time, trip valve closure timing (as applicable), and continuing feed in the case of the process gas compressor. System complexity, calculation complexity, and in some cases geometric uncertainties, necessitate various simplifying assumptions and estimates in order to develop the dynamic model. For example, if complete compressor internal component dimensional data is available, internal components are represented as piping components to determine flow resistance. If insufficient dimensional data is available, flow resistance estimates are made based on comparable compressors in equivalent service for which dimensional data is available while making adjustments for differences in volumetric capacity and internal configuration. In either case, the reverse flow rate calculated using this information must be recognized as approximate. Consequently, the overpressure magnitude that is calculated and the effectiveness of analyzed mitigation alternatives, such as increased relief capacity, are approximate.

PROCESS GAS COMPRESSION SYSTEM HAZARDS

The process gas compression system configuration depends on plant design basis, technology licensor and plant vintage. All systems analyzed include either four or five stage compressors. Compressors have either a single driver or two drivers. None of the systems analyzed included front-end Depropanizer plant configurations; this configuration is not specifically addressed within this paper. However, the study methodology and concept is also applicable to other process gas compressor configurations. This paper addresses two basic system configurations relative to check valve application.

In the predominant configuration, check valves segregate minimum flow loops, i.e., there is a check valve located just downstream of each minimum flow source tie-in. Minimally one check valve is located near the final stage discharge and one check valve is located upstream of the caustic tower which is normally located at the third stage discharge. In some cases an additional check valve is located downstream of the caustic tower. The process gas compression systems at LyondellBasell are designed with anywhere from one to three minimum flow loops; the majority are designed as two minimum flow loops. Typically, the low-stage minimum flow loop encompasses stages one through three and the high pressure minimum flow loop encompasses the final stage (four-stage compressor) or stages (five-stage compressor).

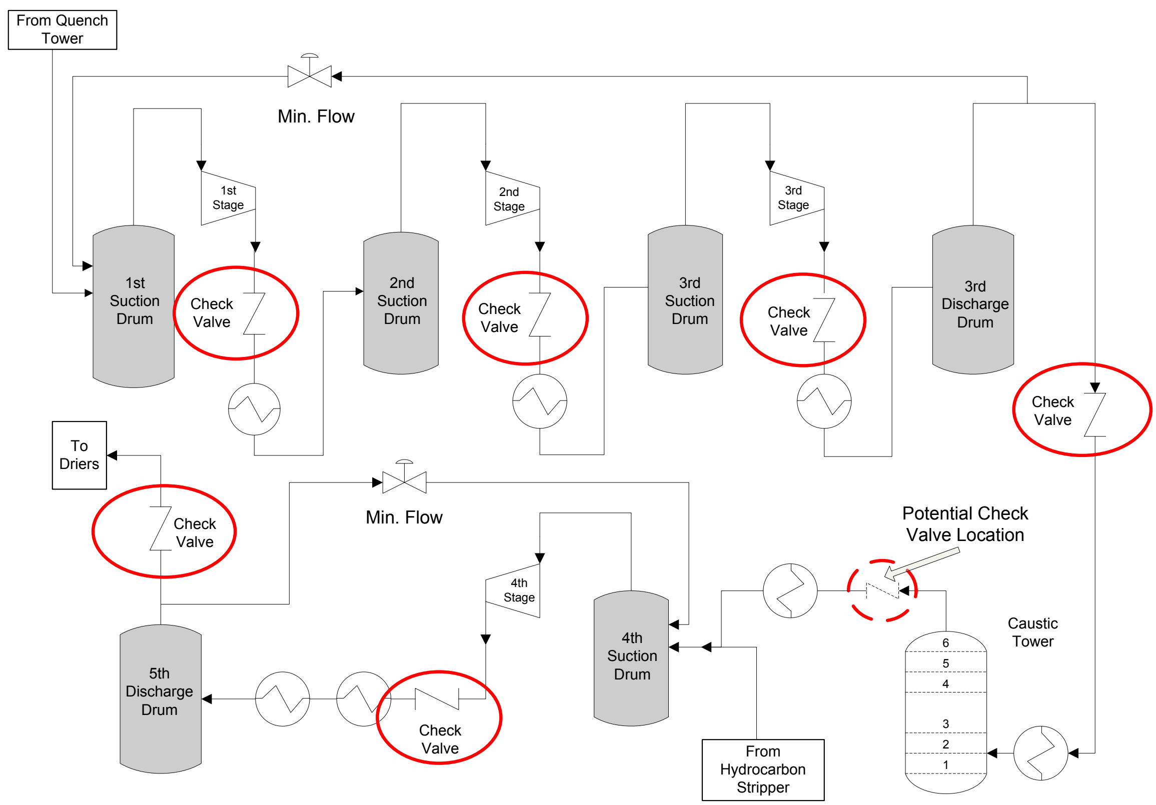
A simplified flow sheet representing the predominant system configuration is as follows:



With this configuration, overpressure risks predominantly occur at the fourth-stage suction and at the first-stage suction, and are dependent on single or dual check valve failures. If the compressor's discharge check valve fails, the large vapor volume of the process dryers and chilling train rapidly flows back to the process gas compression system. Pressures approach equalization within one to three minutes if unrestricted by back-flow preventers (restricted only by piping and equipment hydraulics). Fourth stage suction overpressure can be expected in excess of 150% of equipment MAWP and possibly approaching 200% of MAWP. In this study, fourth-stage overpressure determined by dynamic analysis ranged from 60% to 90% of the overpressure determined by static analysis (settle-out pressures). The large variation is due to differences on fourth-stage equipment volume and relieving capacity. If affected equipment includes the caustic system, equipment failure with limited overpressure is a concern if cracks are present due to caustic stress corrosion cracking. First stage overpressure can occur either due to combined check valve failures (high pressure discharge and third discharge) or due to excessive flare header back pressure dependent on fourth-stage suction venting and/or relieving capacity. Reverse flow rates to fourth-stage suction can exceed the compressor's design flow rate. This flow combined with first-stage suction relief load (compressor feed flow) creates significant backpressure at the first-stage suction relief valve, thus compromising relief capacity. Potential first-stage suction overpressure can only be determined with reasonable accuracy via dynamics analysis. This analysis needs to take into consideration continuing furnace effluent flow, reverse flow through the compressor, flow through the minimum flow line, relieving capacity and flare header back-pressure. Flare header back pressure must also be assessed dynamically taking into consideration interstage and suction flare vent and relief valves.

Within the other process gas compression system configuration, a check valve is located at the discharge of every compression stage regardless of minimum flow loop configuration. In either configuration, a check valve is located upstream of the caustic tower and in some cases downstream of the caustic tower. Some plants have added a check valve downstream of the caustic tower due to the risk of tower tray damage attributed to compressor surge. This increases the potential magnitude of fourth-stage suction overpressure, since the caustic tower no longer serves as a reservoir in the event of a discharge check valve failure.

A simplified flow sheet representing the second process gas compressor design configuration is as follows:



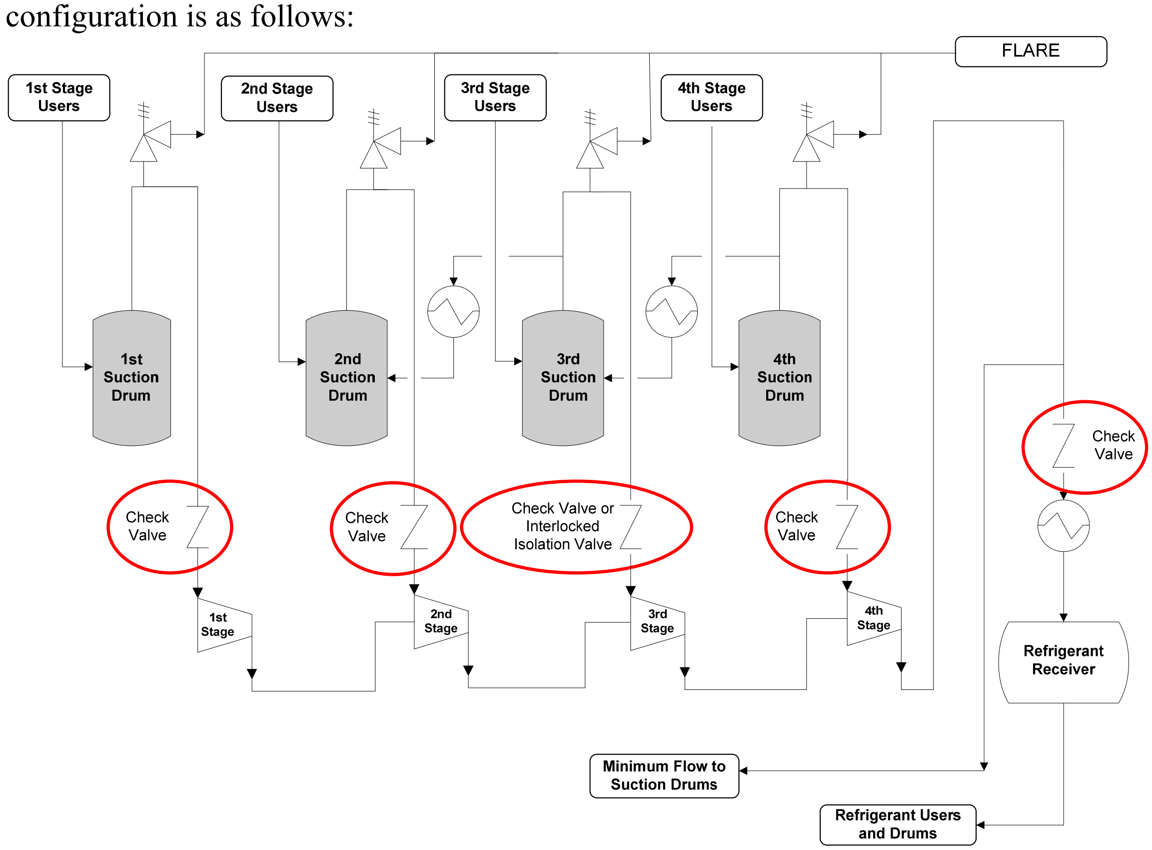
With check valves located at the discharge of each stage, overpressure hazards can exist at each stage for either individual or multiple check valve failure scenarios. Typically interstage relief capacity is relatively small, sized for a fire case scenario. Check valve failure can result in overpressure as high as 300% of MAWP.

Following the compressor trip, compressor speed declines rapidly initially, with the speed decay rate slowing as system pressures equalize. In the event of a discharge check valve failure, differential pressure across the compressor is sustained for a period of several minutes. During this period the substantial inventory in the Process Dryers and the Chilling Train flow back through the compressor case. The rapid speed decay rate is sustained with rotor speed reaching 0 RPM within 30 to 60 seconds. With substantial reverse flow remaining once the compressor speed reaches 0 RPM, rotor rotation will reverse. Due to the large mass of the compressor and turbine rotor, rotation speeds will be limited but can reach critical speed and remain in the critical speed range for several minutes. This presents a risk of potential bearing and seal damage. This risk also is present if the third-stage discharge check valve fails, allowing the large volume of the caustic tower to depressure back through the compressor. The magnitude and duration of reverse rotation is impacted by low-stage minimum flow valve response and capacity.

PROPYLENE REFRIGERATION SYSTEM HAZARDS

Since the volume of the Propylene Refrigeration System suction systems are relatively large compared to the volume of the discharge system, the magnitude of overpressure in the event of check valve failure is normally limited. However, brittle fracture failure hazards at limited overpressure are not uncommon within propylene refrigeration systems. Possible overpressure is dependent on failure of two isolation devices, the discharge check valve and the suction isolation device, which can be either a check valve or an automated trip valve.

A simplified flow sheet representing a common Propylene Refrigeration System configuration is as follows:



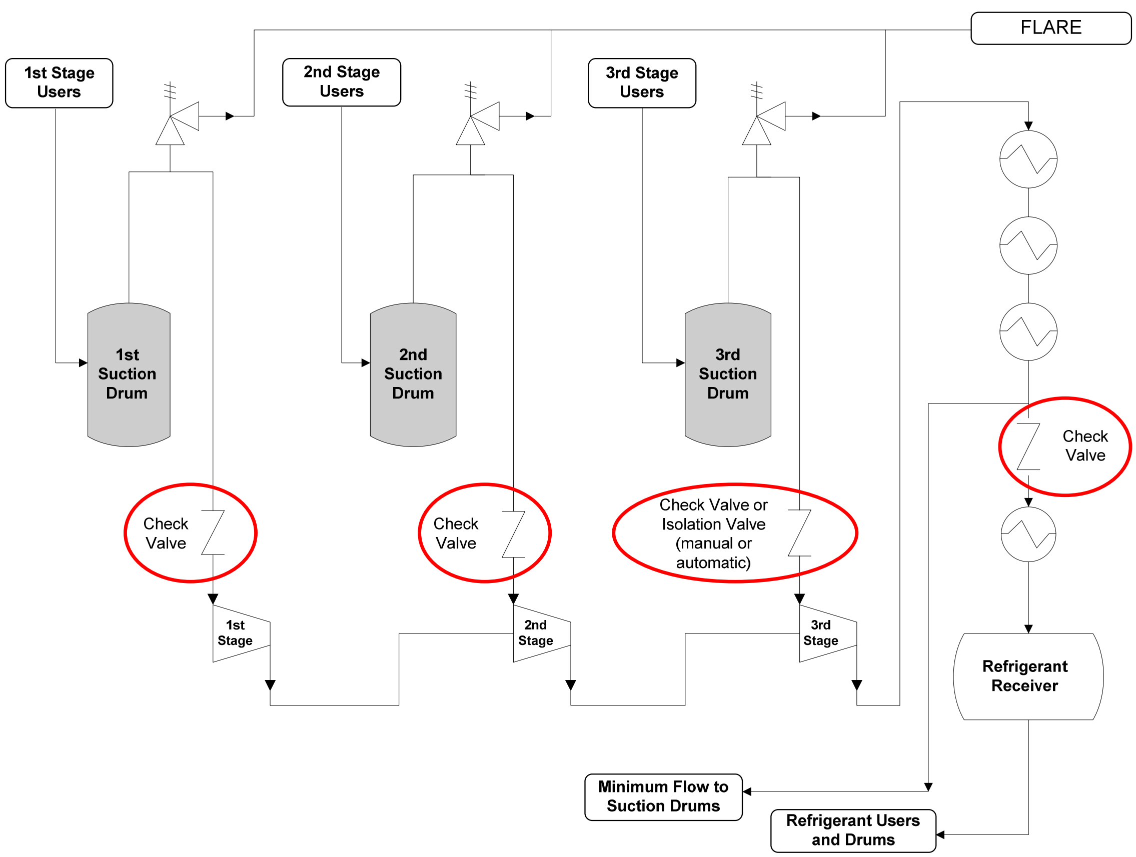
Limited interstage relief capacity has a relatively small impact on the magnitude of overpressure. However, due to the large 1st stage minimum flow valve capacity and large first-stage suction volume, low stage minimum flow valve response does have a significant impact on possible overpressure. Note, not all propylene refrigeration systems are designed with suction check valves or trip valves.

Compressor reverse rotation risks with the propylene refrigeration system are limited. Under normal trip conditions the compressor's coast-down rate can initially be very rapid (for 10-20 seconds following trip). The large first-stage minimum flow valve capacity rapidly deinventories the lower volume discharge system to the much larger volume first-stage suction system. Pressure ratios decay rapidly enough to maintain conditions to the right of the compressor's surge line on each stage, i.e., forward flow through the compressor continues, albeit at rapidly declining rates. Thus the compressor continues to perform work which consumes inertial energy. This causes rapid rotor deceleration until pressures approach equalization. In the event that the discharge check valve fails to close, but the suction isolation performs properly, the discharge system depressures at a slower rate and conditions move to the left of the surge line for multiple stages. Reverse flow through the compressor allows the compressor case to pressure up, reestablishing conditions to the right of the surge line and thus reestablishing forward flow. Then, as forward flow deinventories the compressor case, conditions again move to the left of the surge line. Subsequently, the compressor continues to rapidly cycle through forward (compression) and reverse flow conditions (surge). At comparable pressure ratios, reverse flow conditions consume less inertial energy than forward flow conditions, causing an extension of coast-down duration. Additionally, without forward flow sustained (which must also flow through the minimum flow valve), the rate of discharge system depressurization may not be significantly impacted when compared to conditions with a properly functioning discharge check valve. In the event of a suction isolation failure, with or without discharge check valve failure, reverse flow through the compressor is sustained. Compressor coast-down duration is further extended and discharge system pressure reduces rapidly with system deinventoried through minimum flow valves and the failed suction isolation.

Propylene refrigeration compressor rotation reversal is possible. However, if it occurs it will most likely be limited to speeds below critical. Bearing damage is unlikely under this low speed, low load condition but seal damage is possible depending on the seal design. Reverse rotation is unlikely to occur with the first-stage minimum flow valve open. If the first-stage minimum flow valve is closed due to failure or by design (designed to close on compressor trip/high stage isolation interlock), reverse rotation could occur in the event of dual isolation failures. However, this only occurs if one check valve limits flow sufficiently, allowing reverse flow to continue after the rotor speed decays to 0 RPM. The low reverse rotation risk is unique to the propylene refrigeration system due to the combination of large rotor mass (rotor inertia on trip), the relative volumes of discharge versus suction systems and the large first-stage minimum flow valve capacity. This situation is not necessarily applicable to all propylene refrigeration system designs.

ETHYLENE REFRIGERATION SYSTEM HAZARDS

The majority of the ethylene refrigeration systems evaluated were conventional three stage compression systems. LyondellBasell's ethylene plants located in the United States also includes two two-stage ethylene refrigeration systems and one four-stage ethylene refrigeration system. The latter includes a heat pumped ethylene fractionator integrated into the compression system's fourth-stage. All systems operate with discharge pressures of 250-325 psig, with the exception of two systems that operate at discharge pressures of 375-450 psig. All systems include check valves on the first and second-stage suction lines and most include third-stage isolation, i.e., either a check valves or a trip valve. Two systems include no third-stage suction back flow prevention devices. Some industry ethylene refrigeration systems include no suction isolation (check valves or interlocked trip valves). A simplified flow sheet representing the predominant Ethylene Refrigeration System configuration is as follows:



Overpressure risks can exist on all suction systems in the event of single check valve failures as well as dual check valve failures. Single check valve failures can result in pressures approaching 200% of equipment MAWP and dual check valve failures can result in pressures approaching 300% of equipment MAWP. Overpressure determined by dynamic analysis is typically within a few percent of overpressure determined by static analysis. Pressure equalization occurs very quickly due to the rapid deceleration of the compressor rotor, i.e., significant flow restriction caused by the spinning rotor occurs only during a few seconds. There is insufficient suction system relief capacity to cause any significant impact on settle-out pressures over this short duration. For compression systems operating with discharge pressure near 300 psig, overpressure magnitude is typically limited as long as the suction equipment design pressure is at least 150 psig; however, this depends on relative system volumes. Ethylene refrigeration system suction equipment is frequently constructed from low-temperature carbon steel alloy with a minimum design temperature of -155°F. Consequently, with first-stage suction equipment operating near -150°F, carbon steel alloy equipment is at risk of brittle fracture failure even with moderate overpressure. Check valve failure scenarios result in rapid pressurization of suction equipment with equipment metal temperatures remaining near operating temperature long after peak pressure is reached.

The post-trip coast-down duration of the ethylene refrigeration compressor and turbine rotor assembly is short, typically dropping below 1000 RPM within less than 10 seconds. After this short duration, discharge pressure remains high as does differential pressure across the compressor case. Due to the low mass of the rotating assembly, reverse flow conditions can result in rotor reverse rotation to very high speeds. Speeds in excess of overspeed limits are possible. At these speeds, catastrophic equipment failures have been known to occur, potentially resulting in a gas release and/or fire. The authors are familiar with multiple ethylene refrigeration system rotor rotation reversal incidents, the majority of which have resulted in no mechanical damage. However, one incident resulted in reverse speeds beyond operating speed which led to major mechanical damage and a fire.

CHECK VALVE HYDRAULICS

Until the valve's opening is significantly restricted, a check valve provides very limited flow resistance. Examples of system overpressure as a function of check valve opening are as follows:

Process gas compressor examples:

Plant A

Check Valve Flow Area % of maximum	Equipment MAWP %
100	140
33	138
10	130
5	111

Plant B

Check Valve Flow Area % of maximum	Equipment MAWP %
100	164
33	163
10	155
5	141

Ethylene refrigeration compression system example:

Check Valve Flow Area % of maximum	3rd Suction MAWP %	2nd Suction MAWP %	1st Suction MAWP %
100	209	166	127
50	207	163	124
33	204	158	120
20	199	146	112
15	193	134	106
10	169	124	101
5	119	111	101

A gross check valve failure is not necessary to create a significant overpressure risk in many applications. Hazards can occur due to delayed check valve response or limited travel which may be caused by bearing degradation, excessive dampening system resistance, fouling or other factors.

CHECK VALVE SELECTION AND RELIABILITY

Check valve reliability is a function of design, application/service, installation, maintenance and operation. In centrifugal compressor service, check valves are at risk of failure due to compressor surge, with compressor surge providing a common mode failure mechanism which can result in multiple check valve failures. Swing type check valves in particular are at risk of damage during a surge event, due to forces applied to the disc and seat as the check valve rapidly cycles from full open to full close during surge, even with dampening provisions. Additionally, external dampeners used on swing type check valves to limit forces during rapid valve closure can fail and compromise check valve performance. Dual plate (wafer) type check valves and axial (nozzle) type check valves are at reduced risk of damage during surge due to “non-slam” characteristics [3] accomplished without the use of external dampeners. However, in process gas compressor applications, dependent on check valve location and plant operating experience, the potential impact of fouling on check valve performance needs to be taken into consideration when evaluating check valve design alternatives.

Particularly in low pressure applications in which even small changes in pressure drop can create significant economic penalties, extreme care must be applied when specifying and selecting check valves. Process conditions must be specified over the full range of operating flows. Additionally, the sensitivity of valve performance and pressure drop relative to piping design must be fully understood. This is particularly true of the axial type check valve. In pressure drop sensitive applications, the basis for the check valve supplier’s pressure drop data must be understood and appropriately challenged. In these applications, check valve bench testing to validate pressure drop curves should be given consideration.

Check valve maintenance is an obvious factor impacting performance; however, it is frequently neglected or inadequate. Often, at most, check valves are merely cleaned and visually inspected. The authors are aware of the gross failure of four separate check valves in a process gas compression system, the cause of which was primarily attributable to maintenance inadequacies. Other influencing factors were fouling and material selection. Risk reduction claims dependent on proper check valve functionality should only be claimed for properly designed, selected and maintained check valves. Critical service check valves should be subject to inspection, refurbishing and testing during every major turnaround.

Industry data on check valve reliability [4] independent of check valve type, application and maintenance practices indicates failure rates no better than 1/100 years with an average failure frequency rate of 1/52 years and a failure frequency range between 1/17 and 1/394 years. Nuclear industry check valve failure rates [5, 6, 7] are comparable as follows:

- Significant failure frequency range = 1/63 years to 1/438 years
- Average significant failure frequency for swing check valves = 1/80 years
- Average significant failure frequency for double plate check valves = 1/100 years

Significant failures are defined as failures involving detached or broken components, restricted motion failures, valves stuck open and valves stuck closed. This does not include sealing deficiencies resulting in leak-by. Certain ethylene manufacturing process and application factors detrimentally impact check valve reliability. These factors include surge risks and, in the case of the process gas compression system, corrosion and fouling risks. This needs to be taken into consideration when assessing check valve availability. Common mode failure risks associated with surge induced damage should be taken into consideration when evaluating risks dependent on multiple check valve failures, as risk may only be marginally reduced by a second check valve.

COMPRESSOR AND TURBINE CHECK VALVE FAILURE INCIDENTS

Check valve failure documentation, as well as individual experience with check valve failures, is relatively extensive. However, there is very little documentation of overpressure incidents specifically associated with check valve failure. Undoubtedly such incidents have occurred when one takes into consideration check valve failure statistics. However, unless brittle fracture failure is a factor, uncompromised equipment can be subjected to substantial overpressure without catastrophic failure. Damage from overpressure as high as 300% of equipment design pressure may potentially be limited to vessel deformation and possibly a small leak [1]. Incidents of this nature that result in limited vessel damage typically go unreported within the industry. The fact that much of the industry equipment subjected to significant overpressure will not fail catastrophically, neither achieves Code compliance nor ensures that catastrophic failure will not occur. Catastrophic vessel failure can occur at limited overpressure if conditions are at risk of brittle fracture failure or if equipment integrity has been compromised by corrosion, active cracking mechanisms (e.g. caustic induced stress corrosion cracking), fatigue or non-compliant modification.

Some check valve failure incidents associated with compressors, turbines or pumps that have resulted in equipment damage are as follows:

- Process Gas Compressor Service - Failure of three individual check valves results in compressor reverse rotation in the critical region for three minutes. The primary failure factor is inadequate maintenance with fouling and construction material incompatibility as secondary failure factors. A fourth check valve also failed.
- Ethylene Refrigeration Service – Check valve(s) failure results in reverse rotation with speeds beyond 10,000 RPM, with resulting mechanical damage and fire.
- Propylene Refrigeration Service – Surge event results in check valve internal component fracture and compressor damage. An eleven-day plant shutdown is required to implement repairs.
- Refinery FCUU Compressor – Check valve failure, preceded by a surge event, results in reverse rotation >4500 RPM with mechanical damage and fire.
- Ethylene Refrigeration Service – Multiple reverse rotation events on two separate compressors with speeds up to 6000 RPM. The specific cause has not been identified. No mechanical damage resulted.
- Cooling Water Pump – Discharge check valve failure causes reverse rotation resulting in driver overspeed and subsequent catastrophic failure of the turbine. Steam header damage resulted in a plant shutdown.
- Refinery Hydrotreater Charge Pump – Failure of multiple back flow prevention devices (series check valves and SIL 3 isolation interlock) results in reverse rotation, mechanical damage and fire. Operator intervention prevented catastrophic vessel failure [8].

CODES AND STANDARDS

Requirements to address equipment overpressure risks are governed by ASME Boiler & Pressure Vessel Code, Section VIII, Divisions 1 and 2. Allowable overpressure is limited to 110% of equipment MAWP for equipment protected with a single relief valve and 116% of equipment MAWP for equipment protected with dual relief for scenarios other than fire exposure. Part UG-125 states that it is the user's responsibility to identify all potential overpressure scenarios and the overpressure protection methodology to be used. Part UG-140 addresses overpressure by system design including use of interlocks in lieu of relief valves to mitigate overpressure hazards. UG-140 requires that the following conditions be met in order to utilize overpressure protection by system design:

- The system cannot be exclusively in air, water or steam service.
- The user is responsible for defining and providing protection from overpressure by the system design. Acceptance of the overpressure system design by the jurisdiction may be required.
- The user shall conduct a detailed analysis to identify and examine all scenarios that could result in an overpressure condition and the magnitude of overpressure.

  - "Causes of Overpressure" as described in ANSI/API 521 Pressure-Relieving and Depressuring Systems shall be considered.
  - Detailed PHA using a multidisciplinary team experienced in utilizing PHA methods must be conducted.
- The overpressure scenario must be readily apparent so that operators or protective instrumentation can take corrective action to prevent operation above MAWP at the coincident temperature.
- No credible overpressure scenario in which the pressure exceeds 116% of the MAWP shall exceed the test pressure.
- The results of the PHA shall be documented and signed by the individual in responsible charge of the management of the operation of the vessel.

UG-140 references WRC Bulletin 498 "Guidance on the Application of Code Case 2211 – Overpressure Protection by Systems Design" [9] for direction in defining credible overpressure events and performing scenario analysis. WRC 498 presents a method for defining credible overpressure scenarios which is comparable to typical industry risk classification procedures which typically mitigate catastrophic hazards to a frequency of 10E-05 or less.

Part 9 of Section VIII, Division 2 as well as UG-140 also directs the user to ANSI/API Standard 521 [10] for possible guidance assessing and defining all applicable overpressure scenarios. Specific to check valves, ANSI/API Standard 521 provides the following guidance:

- Single check valves:
  - States that "a single check valve is not always an effective means for preventing overpressure by reverse flow from a high-pressure source."
  - States that "Overpressure protection shall be provided for a single check-valve latent failure (e.g. stuck open or broken flapper)."
  - Relief valve sizing is based on a full open check valve.
  - Even if a check valve failure is considered unlikely, relief protection should be provided if the maximum normal operating pressure of the high-pressure system is greater than the upstream equipment's hydrotest pressure.

- Series back flow prevention:
  - States that experience has shown that two properly maintained back-flow prevention devices in series are sufficient to eliminate significant reverse flow.
  - If reliability of series check valve cannot be assured, then the quantity of back-flow leakage depends on the type of check valve, the fouling nature of the fluid and other system considerations.
  - It is the responsibility of the user to determine the appropriate technique for estimating reverse flow.
  - Where no specific experience or company guidelines exist, reverse flow can be estimated by representing the check valve as a single orifice with diameter equivalent to one-tenth the diameter of the largest check valve.

The following needs to be considered when applying the guidelines provided in ANSI/API Standard 521:

- ✓ With compression systems, there is a common mode failure risk of surge induced check valve damage. This needs to be considered when assessing the reliability of series check valves.
- ✓ Check valve failure are typically covert (latent) failures and this is particularly applicable to series check valves. Operation with a failed check valve can occur over a number of years without detection. Field visual inspection, even with the valve removed during shutdown, may be insufficient to detect compromising failures.
- ✓ ANSI/API Standard 521 guidance regarding double jeopardy exceptions for relief valve sizing scenarios is based on a philosophy of mitigating risks to a frequency no less often than 1/100 years. This compares with standard industry practice of mitigating catastrophic hazards to frequency of 1/10,000 to 1/100,000 years or less.
- ✓ Mitigating risks in compliance with ANSI/API Standard 521 does not necessarily mitigate risk in compliance with corporate risk standards nor does it necessarily achieve Code compliance.

COMPRESSOR OVERPRESSURE RISK MITIGATION ALTERNATIVES

Various alternative approaches exist for mitigating compressor overpressure hazards and reverse rotation hazards. Determining the appropriate approach depends on various factors including system characteristics, application specifics, hazard frequency assessment, and risk mitigation requirements, e.g., the number of independent protection layers (IPLs) required to mitigate the hazard within corporate risk mitigation guidelines. Possible alternatives and associated advantages and disadvantages are as follows:

<u>Series Check Valves</u> – Following ANSI/API 521 guidelines, install series check valve in applications where a single check valve failure can result in an overpressure or rotor reverse rotation hazard.

- ➢ Pros:
  - ✓ Low-cost alternative.
  - ✓ Mitigates overpressure, flare loading and reverse rotation hazards.
- ➢ Cons:
  - ✓ Projected catastrophic failure frequency higher than allowed by industry risk mitigation standards.
  - ✓ Check valves failures are latent and detected only via inspection during turnaround or as a result on an incident.
  - ✓ Due to common mode failure risks associated with compressor surge, operating history needs to be considered and addition of or improvements to anti-surge controls may be necessary.

Equipment Replacement – Install replacement equipment designed for the maximum possible pressure. Upgrade metallurgy to address brittle fracture failure risks as applicable.

- Pros:
  - ✓ Completely mitigates overpressure risks.
- Cons:
  - ✓ Typically the highest cost alternative.
  - ✓ Doesn't address reverse rotation risks.

Increased Relief Capacity – Increase relieving capacity sufficiently to limit overpressure magnitude within Code allowable.

- Pros:
  - ✓ Potentially low-cost alternative.
  - ✓ Achieves Code compliance.
- Cons:
  - ✓ Determining adequate relief capacity subject to significant calculation uncertainty. Due to this uncertainty, appropriate conservatism needs to be considered when defining relief valve capacity requirements.
  - ✓ Can result in excessive flare loading which can then compromise relieving capacity of other relief valves, in particular process gas compressor first-stage suction relief valves.
  - ✓ Doesn't address reverse rotation risks.

Isolation Interlock – Install compressor discharge and/or interstage isolation valve to trip closed on compressor shutdown supplementing check valve back-flow prevention.

- Pros:
  - ✓ Typically reduced cost versus equipment replacement.
  - ✓ Mitigates overpressure, flare loading and reverse rotation hazards.
  - ✓ Can achieve Code compliance per UG-140 if properly designed.
- Cons:
  - ✓ Trip valve closure timing requirements are subject to uncertainty. Rapid closure frequently required, particularly on ethylene refrigeration systems.
  - ✓ Risk of inadvertent isolation valve closure while compressor is running (process upset consequences and compressor surge risks, may necessitate compressor trip on closure detection).
  - ✓ Can necessitate additional costs to upgrade minimum flow controls/valves and trip detection instrumentation.
  - ✓ Larger, SIL 3 applications are costly if required.

Frequently, a solution composed of a combination of the above alternatives will prove to be the most cost effective approach.

CONCLUSIONS

Industry data supports expected check valve failure frequencies of between 1/10 years and 1/100 years. Lack of adequate maintenance and testing, fouling, corrosion and compressor surge are factors that can negatively impact check valve reliability in compression systems. Check valve failure can go undetected for many years even with an appropriate maintenance program that involves check valve inspection and refurbishing during all major turnarounds. Check valve failure scenarios can result in equipment overpressure in excess of 300% MAWP, excessive flare loading conditions, and compressor/turbine reverse rotation that can result in major mechanical damage as well as a gas release and fire. Simple settle-out calculations can be used to identify at-risk systems. Where hazards exist, check valves should be classified as safety critical devices with appropriate maintenance programs implemented involving shop inspection, refurbishing and testing of check valves during every major turnaround. However, additional protective measures are frequently necessary to mitigate risks in order to comply with company risk mitigation standards as well as to achieve compliance with the ASME Boiler and Pressure Vessel Code. Where additional risk mitigation is needed, a number of alternatives exist to mitigate hazards, each with their associated advantages and disadvantages.

REFERENCES

1. Williams, J. P., "Reliability for Safety Instrumented Systems," *Chemical Engineering Progress*, pp. 27-32 (September 2004)
2. King, R. E., "Auto-Refrigeration/Brittle Fracture Analysis of Existing Olefins Plants--Translation of Lessons Learned to Other Processes," Journal of Hazardous Materials, Vol. 142, Issue 3, pp.608-617 (11 April 2007).
3. "Dynamic Characteristics of Check Valves," Val-Matic Valve and Manufacturing Corporation (2003). Access via: www.valmatic.com
4. "Guidelines for Process Equipment Reliability Data", New York, AIChE Center for Chemical Process Safety. (1989). Access via: http://www.saunalahti.fi/~ility/)
5. McElhaney, K. L., "Failure Modes and Causes for Swing and Lift Type Check Valves," prepared by the Oak Ridge National Laboratory (ORNL) for the U.S. Nuclear Regulatory Commission (1997).
6. Hart, K., McElhaney, K. L., Casada, D. A., "Efforts by the Nuclear Industry to Evaluate Check Valve Failures", presented at the *Third NRC/ASME Valve and Pump Symposium,* Washington, D.C., July 1994.
7. Hart, K., "The Use of Check Valve Performance Data to Support New Concepts (PRA, Condition Monitoring) for Check Valve Programs".
8. "Safety Alert: Failure of the Backflow Protection System in a Hydrotreater." Mary Kay O'Conner Process Safety Center, Centerline Vol. 8, No. 2, pp. 14-16 (2004)
9. Sims, R. J., Yeich, W. G., "Guidance on the Application of Code Case 2211 – Overpressure Protection by System Design," *WRC Bulletin 498,* (Jan. 2005)
10. American Petroleum Institute, "ANSI/API Recommended Practice 521, Pressure Relieving and Depressuring Systems," 5th Edition (Jan. 2007).