CHECK VALVES FOR COMPRESSOR PROTECTION— A USER'S VIEW

by

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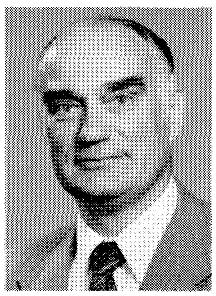

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Subsequently, he spent eighteen years in the Petrochemicals and Plastics Division of Imperial Chemical Industries PLC as a machinery specialist responsible for all aspects of large single stream turbo compressors on world class olefine and other plants. High operating reliability and precision performance assessment have been his major fields of concentration there.

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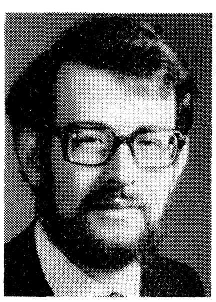

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ABSTRACT

The paper summarizes experience of the results of check valve operation (and maloperation) over a number of years, concentrating on data gathered during the last three years in an olefine plant. Continuous tape recording is employed, and variables such as speed, pressures, valve position and vibration can be made available for study after a machine trip.

Much concern was caused by the discovery of very rapid decelerations in the ethylene and HP process gas compressors. Careful analysis proves that reverse rotation can occur after rapid deceleration, and speeds over 4000 rpm in reverse have been detected.

Observed deceleration rates are compared for typical frictional rundowns and process pressure driven rundown, with and without reverse rotation occurring. The physical details are explained by reference to the tape records.

Calculation of the process energy levels and machine internals are used to show the magnitude of the driving forces and to establish limits to the processes. The concept of limiting runaway speed in reverse is introduced.

Finally, recommendations are made for check valve characteristics and location to minimize the risk of problems due to inefficiency or maloperation.

INTRODUCTION

From time to time, there have been reports of damage to compressors caused by running to overspeed in reverse. The implication is that the protective devices (check valves) have not operated correctly, but the facts are often obscured by the confusion inherent in such incidents. In recent years we have observed at least eleven instances of reverse rotation of compressors. None of the incidents have had destructive consequences. A considerable amount of data has been gathered, and, using some of this, we will attempt to shed light on a subject which many people in the industry view with apprehension.

Two compressors have been involved in the majority of these events. The first is the ethylene compressor with two suctions and one discharge in the same casing, driven by a small, back pressure steam turbine. The second is a single section, HP cracked gas compressor, driven by a larger, extraction-condensing steam turbine. Using chart records obtained by replaying tape recordings of speed, vibration and pressures, we will describe the sequence of events in typical normal and reverse rotation trips. We will also discuss the causes and draw conclusions which may be useful to other compressor system operators and designers.

EARLY EXPERIENCE WITH REVERSE ROTATION

One of the earliest confirmed instances of reverse rotation of a turbo compressor within ICI occurred in early 1966. The machine concerned was the ethylene refrigeration compressor of a 200,000 t/a (450 million lb/yr) ethylene plant. The incident occurred during comprehensive commissioning trials to determine mechanical integrity and response time and to establish criteria for subsequent plant operation under normal and abnormal conditions.

This plant was one of the first of this size in the world. It differed from previous ICI ethylene plants in that the compressors were single stream and that they were all of the centrifugal type. The ethylene compressor comprised two casings, HP and LP, driven by a direct coupled condensing turbine.

In this instance, the compressor, operating at design speed (8,100 rpm), was deliberately tripped. The tachometer needle was observed to momentarily drop to zero before running down "normally" from 4,000 rpm. Our initial reaction was to suspect the instrumentation; however, fortunately, the possibility of reverse rotation was realized and the shaft observed before it stopped rotating. This verified that the shaft was indeed going backwards!

The only condition monitoring device fitted to the machine was at the thrust bearing, where insertion thermocouples were located in the steel backing of the pad adjacent to the babbitt interface. Consequently, before further operation, the following components were examined and actions taken.

Bearings

Both compressors and turbines were fitted with pressure dam radial bearings and back-to-back type thrust bearings; the pivots on the thrust bearings were located centrally. All bearings showed some polishing consistent with reverse rotation, but all were satisfactory for further service.

Check Valves

Only two check valves were fitted, one to each discharge line of the HP and LP casings. They were of the simple flapper type, without counterbalance weight or dampening cylinders. On removal, the flappers were found to move freely throughout their full travel and to seat correctly. However, the valve in the HP discharge was located downstream of a desuperheater exchanger, some distance from the compressor nozzle. This was unsatisfactory, as a considerable volume of gas could flow backward through the compressor even if the check valve closed immediately on tripping. A circuit modification was made to relocate the check valve much closer to the discharge nozzle. Further deliberate tripping showed that the problem had been eliminated.

RELATED CHECK VALVE EXPERIENCE

As a result of this experience, the refrigeration circuits of the first 450,000 t/a (1 billion lb/yr) ethylene plant, already in the final stages of design, were reviewed. The refrigeration systems were more sophisticated than the previous compressor, with several sections within a single casing. Reverse flow protection of such a compressor is more critical than with a single section casing, and the decision was made to provide a measure of redundancy to give higher protection against the possibility of catastrophic reverse overspeed. This involved fitting check valves to each suction connection, in addition to each discharge connection.

This decision was not made lightly. It was recognized that not only was there a high capital cost for a large check valve in the suction line, but also that the operational penalty arising from the pressure drop would be quite significant. However, the penalty was accepted on grounds of safer operation.

A study of different types of check valves was made to determine the optimum type. A spring loaded piston type was found to have the smallest pressure difference ($\triangle$p), plus the additional advantages of a smooth external profile for insulation and no potential leak paths to atmosphere. As a result, this type of valve was specified for all nine connections to the ethylene and propylene compressors. The sizes of these valves ranged from 12 in. to 42 in. (in the propylene primary suction).

Again, as with the earlier, smaller plant, intensive commissioning trials were conducted and the plant brought on line. The painstaking commissioning was rewarded with a lengthy period of steady operation, enabling the performance of many parts of the system to be assessed. These measurements identified an excessively large pressure drop across the 42 in. suction check valve—four to five times that designed. Naturally, no one wished to shut the plant down, although this represented a 10% increase in compressor horsepower. Little could be done on line, for one of the features of the piston type check valve is that there is no external arm to indicate piston position, nor can one attempt to exercise a piston to establish whether it is free to move!

The cause of the high resistance of the valve was established by first inserting a sealed intrascope through a glanded branch at the main flange of the valve. Suspicions were aroused that the valve was not fully open, but, unfortunately, the viewing angle was too awkward to positively establish the piston position. Clear proof was obtained when a very large cobalt 60 RI source (5 Curie) was used to obtain an X-ray cross-sectional photograph of the piston-to-seat relationship. The piston was less than half open! Which was the cause—malfunction, incorrect specification or inadequate design?

The answer was incorrect specification, arising from inexperience. The response time for this valve was chosen to be similar to that of the valve in the discharge line (5-8 sec under test without flow). The response time for these piston type valves to close was dependent on two factors, the return spring and the non-slam orifice sizing. On cross-checking the data, it was found that the sizing was correct. However, the size of the spring was based upon rated plant throughput, and at that time the plant was only equipped with some 80% of the intended furnace capacity. Therefore, for the flow conditions that existed, the spring was far too strong, preventing the valve from opening fully. In retrospect, it was considered not to be necessary for the suction valve to close as rapidly as the discharge valve. Therefore, the problem was resolved by reducing the spring stiffness, thus lengthening the response time, which, as will be seen later, is quite acceptable for normal circumstances. Clearly, when specifying such valves for this type of duty, it is necessary to recognize that the closing time should be consistent with the valve being fully open at approximately 60% of nominal flow.

Tape recordings, made during anti-surge loop optimization and other tests where these machines have been deliberately tripped, have all shown satisfactory check valve operation with normal rundown. This plant has now completed thirteen years of continuous operation, and there have been no known instances of reverse rotation occurring.

RECENT EXPERIENCE WITH 1.3 BILLION LB/YR PLANT

Resulting from this satisfactory experience, the same piston type of valve was chosen for the suction lines of the refrigeration compressors in this new plant. Because the contractor had no previous experience with this type of valve, the valves selected for the discharge were of the typical flapper type, with counterbalance weight and external damper cylinders. The anti-surge loops were similar, although not identical, to those of the earlier plant, and a simplified circuit is shown in

Figure 1. Once again, the commissioning trials were carried out with meticulous attention to detail. On this modern plant, much condition monitoring was fitted, and continuous 42 channel tape recording was implemented. Variables recorded included shaft vibration; axial shaft position; and pressures to suction, discharge and intermediate connections; together with some process temperatures. Extensive records were taken of trip testing, deliberate tripping and all types of conceived malfunction. In addition to the tape recording, continuous multi-pen recorders were used to monitor the speed and other key variables of the refrigeration and process gas compressors on a 24 hr basis. No abnormal rundowns were experienced during the initial testing and commissioning. Three months later, on New Year's Day, when most of the engineering staff were absent, the ethylene refrigeration compressor tripped and ran down to zero speed.

In reviewing the records the following day, a rather sharp, transient kick in the speed trace was noticed. This appeared to be an instrument malfunction. However, because of the incident many years earlier, the tape recordings were re-analyzed to give a more extended (expanded time scale) recording. The speed trace now clearly showed a very rapid deceleration, followed by what appeared to be a bounce back to about half speed (for an example see Figure 5). The rapidity of the deceleration and the apparent rapid acceleration back to 4,000 rpm, approximately, were quite astonishing until it was realized that the speed transducer could not sense direction of rotation, and that, in fact, the compressor was rotating in reverse.

This was confirmed by replaying and photographing the phase mark and the vibration from the x and y probes on a particular bearing. It was very clear that, on the rundown, the x probe was leading the y probe by 90 degrees; whereas after the speed trace had "bounced," the y trace was now leading the x trace, thus very positively confirming reverse rotation. Previous records which had been retained of all trip incidents since commissioning were re-examined, and it was found that there had been a further instance of reverse rotation approximately two weeks prior to the first observed instance.

The shape of the speed curves indicated that there was some limiting effect on the maximum reverse speed. The question was whether higher reverse speeds were likely in the future. Since the first observed trip and reverse rotation was from maximum continuous speed and high load, and hence high discharge pressure, it was felt that much higher reverse speeds were unlikely. With the knowledge gained in the extensive commissioning trials, it was decided that there was no need to shut the plant down, for all indications were that the machine was still operating satisfactorily.

All future trips were carefully scrutinized for reverse rotation, and, at a later date, the HP cracked gas compressor was found to exhibit the same behavior. Table 1 contains some statistics on reverse rotation of these two compressors.

To determine what was occurring, further transducers were added to monitor pressures up- and downstream of each check valve, and to monitor the positions of the anti-surge valves and the turbine operating cylinder. A simplified circuit diagram is shown in Figure 1. Some of the results obtained are discussed in the following sections.

THE EXPERIMENTAL EVIDENCE—ETHYLENE COMPRESSOR

This compressor consists of two sections in a single casing, with each section having four impellers (Figure 1). There are five check valves, three around the compressor and a further two (in series) separating most of the high pressure portion of

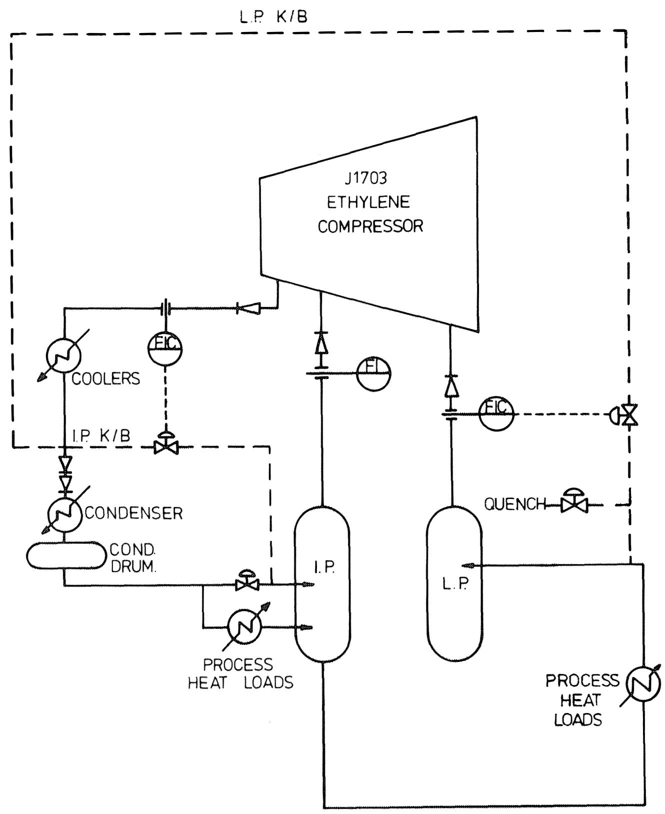

Figure 1. Simplified P and I Line Diagram.

the circuit from the machine and its anti-surge piping.

The results of the trips are shown in Figures 2-7. These charts have been obtained by replaying tape recordings. Speed is shown on every chart (usually 8,000 rpm before trip, falling rapidly thereafter), and it is a common factor enabling cross-reference between different charts of the same event.

Up to seven pressure traces are shown. Six of these (identified as 1-6 on the charts) are from pressure transducers on either side of the three compressor check valves. The seventh pressure trace (where shown) is from a transducer placed downstream of the two check valves in series. The pressures on either side of the machine check valves are vital in understanding what was happening. When a check valve is open and the flow is steady, the pressures on either side of the valve will run parallel; e.g., Traces 3 and 4, Figure 2. Any offset between the traces will be due to pressure drop, particularly if there is other flow resistance between the measuring positions, or transducer zero error. When the valve closes, the pressures diverge, either dramatically, such as Traces 1 and 2, Figure 2, or more gradually. In order to give a complete picture, all six traces must be shown on the same chart. As a consequence, the charts which show the full pressure range (Figures 2, 4 and 6) are condensed, and, in each case, a second chart (Figures 3, 5 and 7) is presented on an enlarged scale to allow closer study of the detail and to aid understanding of the complete record. A detailed description of the sequence of events, for both a normal rundown and a reverse rotation, follows.

Consider first a typical "normal" rundown, where there is no reverse rotation: the record of 5 April 1980 (Figures 2 and 3). Note that the speed decreases much more rapidly than the turbine solo (21 sec, compared with 95 sec from 7000-1000 rpm). The speed decrease is approximately linear for about 0.7 sec, and then follows a nearly exponential form. From Pressure

Table 1. Reverse Rotation Data.

Date	Type of Rundown	Deceleration Time (sec) 7000-1000 rpm	Max Reverse Speed (rpm)	Check Valve Closure Time (sec) Discharge	HP Suction	LP Suction	Time for Discharge Check Valve to Reopen (sec)
J1703 ETHYLENE COMPRESSOR							
27 Jul 79	Turbine Solo	95	-	-	-	-	-
15 Dec 79	Reverse Rotation	6	3190	*	*	*	*
1 Jan 80	Reverse Rotation	4	3750	*	*	*	*
15 Jan 80	Reverse Rotation	5	3400	*	*	*	*
5 Mar 80	Reverse Rotation	6.9	3480	*	0	78	30
5 Apr 80	Normal Rundown	21	-	0.5	0	0	10
31 Mar 80	Reverse Rotation	6.9	3760	0	0	25	20
18 Jun 82	Reverse Rotation	3.9	4140	7	0	∞	13
J1701 HP CRACKED GAS COMPRESSOR				Comments			
21 Jan 81	Reverse Rotation	21	1200				
21 Mar 81	Reverse Rotation	15.6	4750	Speed oscillating 7500-9000 rpm before trip			
10 Apr 81	Reverse Rotation	18.6	1600	Deliberate trip to check valve operation			
10 May 83	Normal Rundown	>120	-	Different valve fitted October 1981			

*Data not available

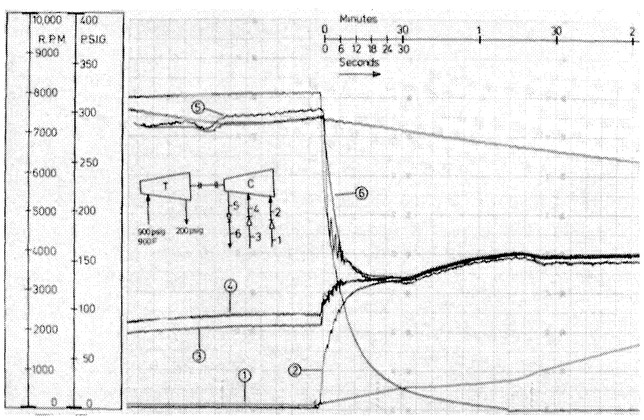

Figure 2. Speed and Pressure Traces, Ethylene Compressor, 5 April 1980.

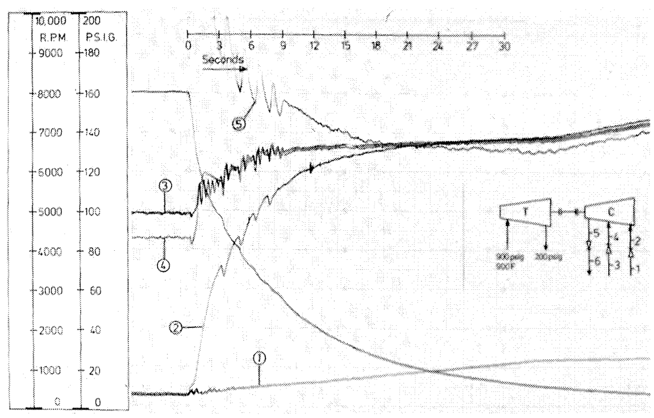

Figure 3. Speed and Pressure Traces, Ethylene Compressor, 5 April 1980 (Enlarged).

Traces 1 and 2 (Figure 3), it can be seen that the LP suction check valve closes almost immediately, as does the IP suction check valve (Traces 3 and 4, Figure 3). The discharge check valve closes after about 0.5 sec (where Traces 5 and 6 diverge, Figure 2). The IP suction valve re-opens at 3.5 sec after trip, when the pressure equalizes across it; from then on, the pressures on either side are essentially identical, and the valve remains open. The pressure ratio across the machine reduces rapidly, and both sections of the machine surge, as shown by the pressure swings on Traces 2, 4 and 5, Figure 3. The discharge check valve re-opens about 10 sec after the trip, when the external pressure has been reduced by the action of the kickback valves. Surging then ceases, as a small forward flow is established, and frictional decay follows with a $k \simeq .012$. (See Appendix for a definition of the deceleration rate, k).

A typical reverse rotation is shown in Figures 4 and 5—the record of 31 March 1980. The speed reduces even more rapidly, 6.9 sec from 7000-1000 rpm. The speed curve (Figure 5) passes through zero at 9 sec after trip. Because the speed detector is unable to sense direction of rotation, the reverse speed that follows is shown reflected about the zero speed line. The kick in the speed trace near zero rpm is also due to the speed detector. The speed curve is very nearly linear through to -1500 rpm. The discharge valve closes almost immediately (Traces 5 and 6, Figure 4), as does the IP suction valve (Traces 3 and 4, Figure 5). However, the LP suction valve does not close for about 25 sec (Traces 1 and 2, Figure 5). The IP suction valve re-opens at 3 sec after trip, when the pressure inside the machine has fallen sufficiently. This allows flow *into* the IP suction and *out of* the LP suction, and, as the compressor passes into the reverse rotation regime, the first (LP) section of the compressor works as a radial inflow turbine, driving the string backwards. At the same time (from 9 sec after trip), the second (HP) section of the compressor behaves as a forward vaned compressor, surging as it attempts to compress gas from the IP suction through to the discharge (Traces 3, 4 and 5, Figure 5). At 20 sec after trip, the delivery check valve opens, allowing forward flow from the HP section of the compressor; and, for the first time, the reverse rotational speed starts to decrease because work is being done by this section of the compressor. At 25 sec after trip, the LP suction check valve opens (Trace 2, Figure 5). This removes the driving force for reverse rotation by preventing reverse flow through the LP section of the compressor, and frictional speed decay begins.

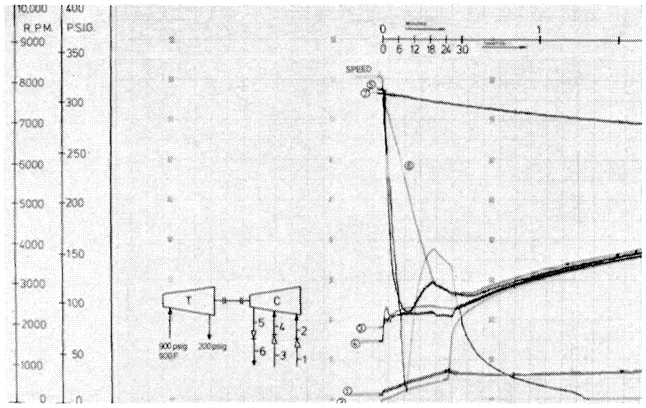

Figure 4. Speed and Pressure Traces, Ethylene Compressor, 31 March 1980.

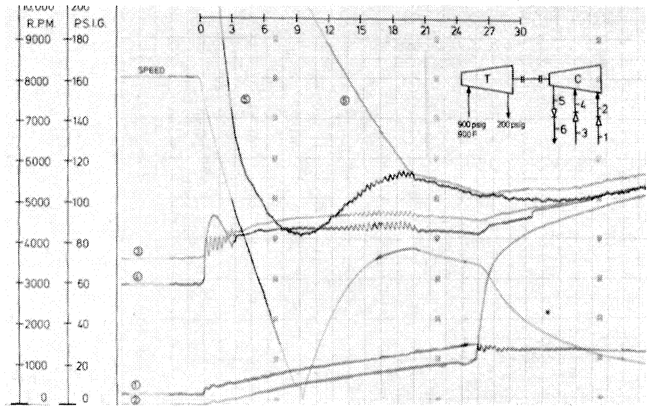

Figure 5. Speed and Pressure Traces, Ethylene Compressor, 31 March 1980 (Enlarged).

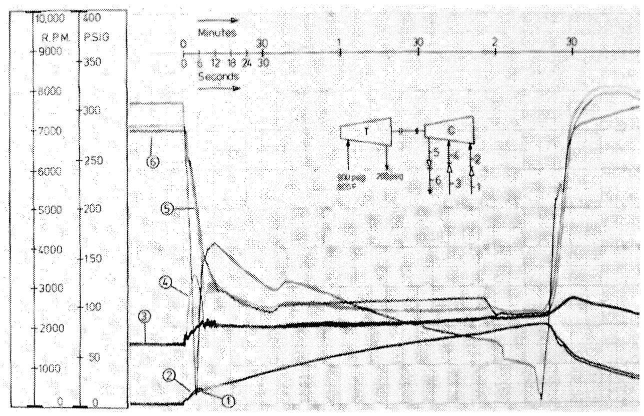

Figure 6. Speed and Pressure Traces, Ethylene Compressor, 18 June 1982.

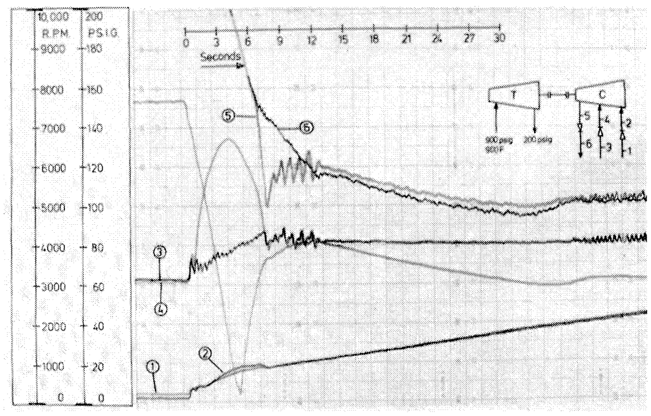

Figure 7. Speed and Pressure Traces, Ethylene Compressor, 18 June 1982 (Enlarged).

Note that the rate of decay ($k \simeq .011$) of the reverse rotation is the same as that of forward rotation ($k \simeq .012$).

In late 1981, all check valves were inspected and found to be free to move by hand. This was not the end of the problem, however, as Figures 6, 7 and 8 show. On this occasion, the LP suction check valve did not close throughout the duration of the trip, and the discharge check valve also failed to close for seven seconds. As a result, this was the most rapid deceleration to date. Figure 8 shows shaft vibration from the drive end of the compressor. The peaks in vibration between 8 and 13 seconds after trip are associated with the surging of the HP section of the compressor prior to the re-opening of the discharge check valve. Despite exceeding the (forward) critical speed in reverse, the shaft system still appears to be stable.

Why the LP suction valve malfunctions is unknown. There are three hypotheses, all associated with the low operating temperature:

- Loss of internal clearance
- Inadequate bearing material
- Frozen compressor lubricating oil

THE EXPERIMENTAL EVIDENCE— HP CRACKED GAS COMPRESSOR

This compressor has four impellers in a single section casing. In a normal trip and rundown, this machine takes >120 seconds to slow from 7000 to 1000 rpm, and the speed curve is approximately exponential, with $k \simeq .003$. This should be compared with $k \simeq .013$ for the ethylene compressor (normal rundown). The major differences are the greater inertia and lower windage of the condensing turbine which drives the HP cracked gas compressor. In the trip with reverse rotation shown in Figure 9, the speed curve is nearly linear to −3000 rpm, and the duration from 7000 to 1000 rpm is 15.6 sec.

No pressure traces have been recorded on this compressor, but it is reasonable to assume that the discharge check valve took longer to close on 21 March 1981 (the case illustrated in Figures 9 and 10) than in the other two recorded instances of reverse rotation on this machine.

Figure 9 shows the speed, total and filtered (1×) vibration and phase angle from the turbine HP bearing (H) probe. The turbine first critical is shown clearly three times: in forward, reverse, and reverse rotation, in rapid succession! Also shown is the 180 degrees phase shift in the vibration signal as the speed passes through zero, a clear demonstration of the reversal of rotation.

Figure 10 gives a beautiful example of one of the possible unpleasant consequences of reverse rotation—whirl. This figure shows one channel of vibration from each end of the compressor, total and filtered (1×). Prior to trip, the effect of torque centering and unlocking of the coupling on the compressor drive end vibration can be seen as the speed oscillates over a 1500 rpm range. At just below peak speed in reverse, the compressor rotor starts to whirl, and whirl continues until reverse rotation has decayed almost to zero. The frequency of this non-synchronous vibration is about 0.45 × rotational speed throughout.

For this compressor, there is a clear-cut reason for the

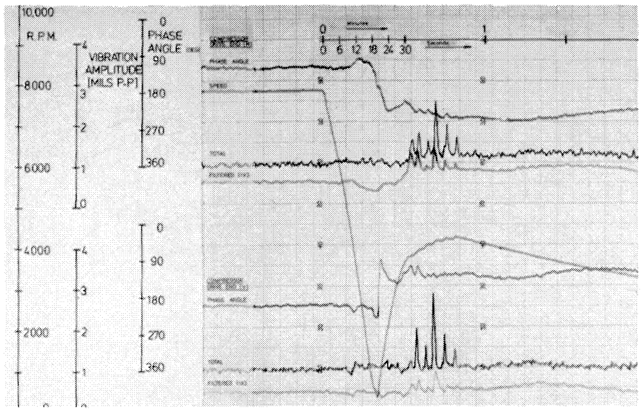

Figure 8. Speed and Vibration Traces, Ethylene Compressor, 18 June 1982.

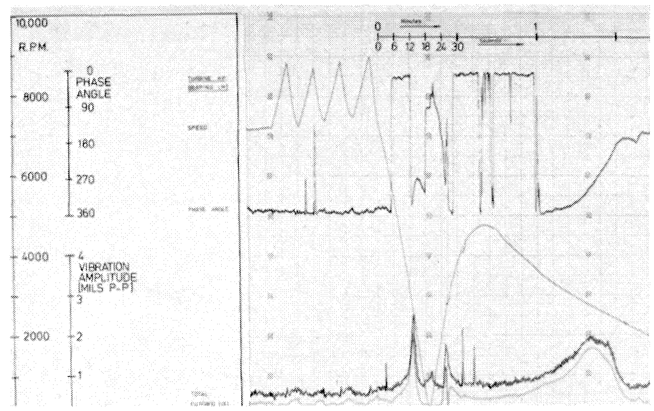

Figure 9. Speed and Vibration Traces, High Pressure Cracked Gas Compressor, 21 March 1981.

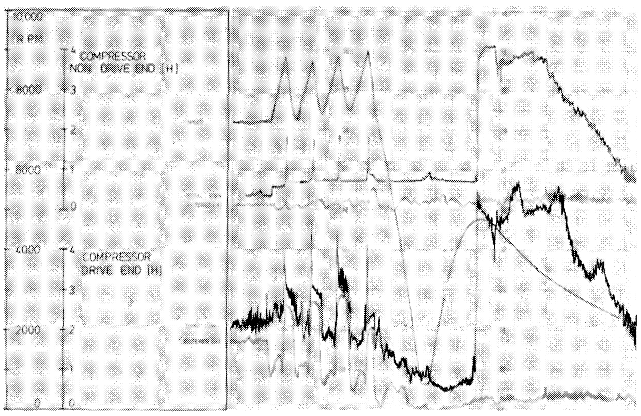

Figure 10. Speed and Vibration Traces, High Pressure Cracked Gas Compressor, 21 March 1981.

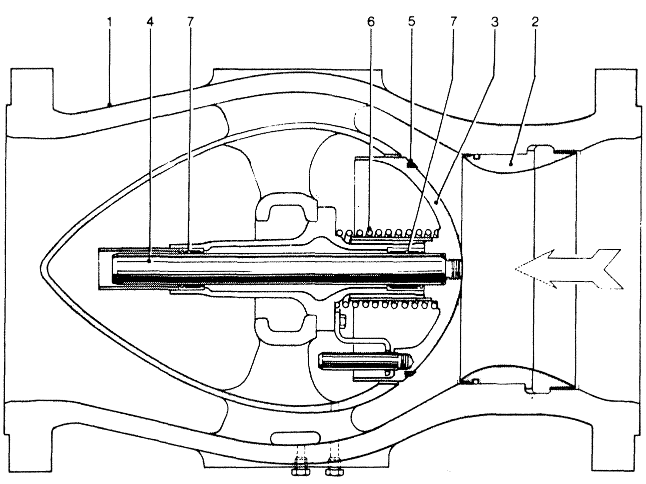

Figure 11. Cross Section of Piston-Type Check Valve (Valve Open).

malfunctioning of the discharge check valve. This is a piston (low pressure drop) valve (Figure 11). When the valve was inspected at a turnaround in 1981, the space around the sliding shaft between the internal bearings was found to be full of polymer. This resulted in considerable friction, resisting correct valve operation. The assumption of the design contractor that this was a "clean duty" was obviously incorrect. A dual flapper type valve was immediately purchased and installed, and no further problems have occurred.

LIMITING SPEED IN REVERSE ROTATION

One clear suggestion from Figures 5, 8 and 9 is that the speed in reverse is becoming asymptotic to a maximum value. This leads us to look for the physical basis of such a maximum. The velocity of gas flowing backwards through the stationary passages of the compressor will be limited at the diffuser throat to the speed of sound. With vaneless diffusers and radial inlet guide vanes, no swirl should be acquired by gas flowing back into the compressor from the discharge or an intermediate suction pipe. Therefore, the backward flow will be unable to accelerate the rotor beyond zero incidence (Figure 12); and, hence, depending on the temperature (and therefore speed of sound) of the gas and the impeller outlet angle, a maximum speed in reverse can be calculated.

$$N_{max} = \frac{720 \text{ a } \tan \alpha}{\pi \text{ d}} \tag{1}$$

where

a = speed of sound (ft/sec)

d = diameter of the wheel (in.)

In the case of the ethylene compressor using a = 925 ft/sec (appropriate to gas coming from the IP suction) and α = 30°, we obtain N_{max} = 5550 rpm, which seems a reasonable upper limit, based on results to date!

For the high pressure cracked gas compressor we calculate

$$N_{max} = 6660 \text{ rpm}$$

Obviously, the higher the value of α (i.e., the more the backward lean), the higher the maximum reverse rotational speed can be.

Non-radial guide vanes should have only a small effect, because inlet guide vanes do not affect the flow at the tip of the last impeller in a section. A vaned diffuser would invalidate Equation 1. The reverse speed limit described by Equation 1 depends on the unloading effect as the compressor rotates faster and faster in reverse, and on there not being supersonic gas velocities in the stationary passages. Another effect which could limit reverse speed is pressure equalization, which will reduce the driving force for reverse rotation and ultimately may allow forward flow to remove energy from the reverse rotating rotor. Pressure equalization takes place both through the compressor (if the check valves remain open) and through the anti-surge (recycle) valves.

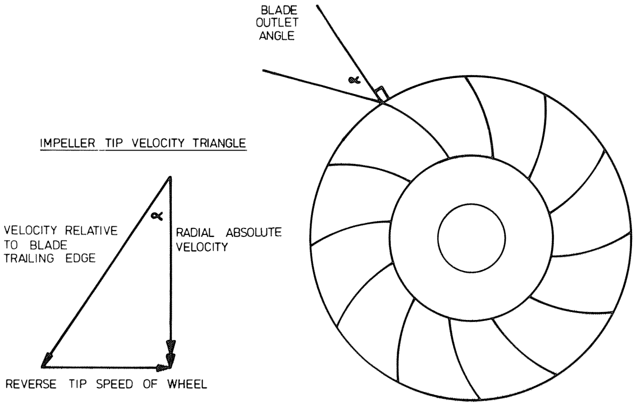

Figure 12. Impeller Tip Velocity Triangle.

Traces 2 and 4 of Figure 5 show that the driving force for reverse rotation, namely the pressure differential across the LP section of the compressor, has not reduced significantly by the time the acceleration is beginning to reduce, at 15 sec after trip. This demonstrates that other, limiting effects are involved, such as the stalled torque exerted by the second (HP) section of the compressor and the reduction in driving torque implicit in Equation 1.

ENERGY LEVELS AND CHECK VALVE PLANNING IN REFRIGERATION CIRCUITS

A brief consideration of energy levels serves to show the importance of the check valves in separating sections of the refrigeration system. This is illustrated with reference to the ethylene refrigeration system discussed in this paper (Figure 1).

Kinetic energy of rotor system = 4600 BTU

Energy in steam between turbine and trip valves = 900 BTU

Energy contained in the gas between the compressor and the check valves:

Discharge gas relative to IP suction = 2500 BTU

IP suction gas relative to LP suction = 104 BTU

Thus, if the check valves work correctly, there should be no possibility of the compressor running backwards.

By considering the consequences of a single check valve not operating, we construct the following table for our system.

Case	*Check Valve Not Operating*	*Result*
1	Discharge	Normal rundown
2	IP suction	Normal rundown (more rapid deceleration initially)
3	LP suction	Reverse rotation

By considering the consequences of two check valves not operating, we construct a second table.

Case	*Check Valves Not Operating*	*Result*
4	Discharge and IP suction	Not experienced, but expect reverse rotation, similar to Case 5 except pressures should equalize after about 12 sec
5	Discharge and LP suction	Reverse rotation (most severe case experienced)
6	IP suction and LP suction	Reverse rotation (not significantly different from LP suction valve failing on its own)

From these tables, we can see the critical importance of the LP suction valve in preventing reverse rotation. If the LP suction check valve does not function, all the energy contained in the discharge and IP suction systems is available to be driven through the LP section of the compressor. This energy content is considerably in excess of the kinetic energy of the rotor in normal operation.

The discharge check valve is important in reducing the rate of deceleration of the rotor; but, with the two additional valves in series upstream of the condenser, the compressor is protected from the bulk of the high pressure system. The IP suction valve performs a useful purpose only when the discharge check valve fails. This can be regarded as the "extra insurance" valve.

DISCUSSION AND CONCLUSIONS

The initial decision that the plant would continue running is seen to be correct. If the mechanical design can tolerate reverse rotation, and if the limiting speed in reverse is below the maximum continuous speed (Equation 1), the chances of a disastrous failure are small. If the limiting speed in reverse is above the maximum continuous speed, greater care is required. Two check valves in series or emergency slam valves may be needed.

Check valves for machine protection should be placed as near to the compressor as practical, and on no account should a vessel of any size come between the compressor and the valve. Double check valves in series are desirable, to segregate the large volumes of high pressure gas from the anti-surge loop.

For multi-section compressors, decide where to place check valves by studying the consequences of their not being present, or not operating. The additional protection of a side-stream check valve is worthwhile, but two in series in the discharge and LP suction may be more effective. No type of check valve is infallible. Piston rods and flapper spindles may seize, and debris of any type can prevent closure.

The aerodynamic design of the internal piston type valves makes good economic sense, and our experience in general is good. However, these valves should not be used for duties which might not be clean. Spring rates must be chosen so that they are fully open at a maximum of 60% of design flow.

Offset flapper type valves with external weighted arms can represent a personnel hazard when operating, even when fitted with dampers. Our experience shows that weighted arms are not always capable of withstanding inertia forces in transient operation (e.g., surge). Designers should assume that reverse rotation will occur and design accordingly. If screwed threads are specified (self tightening in normal rotation), they must be positively locked in order to prevent slackening in reverse rotation.

Journal bearings and thrust bearings should be designed with reverse rotation in mind; for example, with directly lubricated thrust pads, an auxiliary jet might be a prudent precaution. Pressure dam bearings may become unstable in reverse rotation, or critical speeds may be lowered—can this be tolerated?

A shaft driven oil pump would require the auxiliary to be initiated by pressure, *not* shaft speed.

Engagement of a barring motor (turning gear) while a turbine shaft is rotating backwards could result in automatic lock-in of the drive and in motor overspeed.

Anti-surge valve operation must be a compromise between letting material down from high to low pressure (thus reducing the chances of reverse rotation) and maintaining pressure differentials as near to pre-trip values as possible, in order to facilitate a rapid recovery. In the case of the refrigeration circuits, the anti-surge valves are driven closed after 25 sec. For the valve connecting the discharge to the IP suction, this has no effect, as pressures have equalized by that time. For the valve connecting the discharge to the LP suction, this serves to minimize plant disturbance, especially if the LP suction check valve operates correctly.

It is hoped that the presentation of these experiences will provoke some debate and sharing of data on this rather neglected, but important, part of compressor protection.

APPENDIX: MODELS OF THE DECELERATION PROCESS

Two simple models seem useful. The first is the "frictional decay" associated with bearings, windage, etc. Assuming that drag is proportional to speed, we obtain

$$w = w_o\, e^{-kt} \tag{A1}$$

where

- w = rotational speed
- w_o = initial speed
- k = a constant
- t = time

This exponential decay of speed describes well what happens to a solo turbine (Figure 13), where the value of k obtained would be about 0.004 sec^{-1}.

The second model is that of "constant torque." The idea here is that when a compressor trips, it stalls, due to the immediate reduction of speed. At this time, the compressor develops a torque—the stalled torque—which is a significant fraction of the normal torque. This torque exists because of the differential pressure applied across the compressor by the process, and it continues throughout the deceleration as long as the differential pressure is applied. If we assume the torque is constant, as it appears to be from the linearity of the speed traces (see, for example, Figure 14, where the speed curve has been extended through the origin), we obtain

$$w = w_o - \frac{Tt}{J} \tag{A2}$$

where

- T = stalled torque
- J = rotational inertia of the shaft system

In the case of the ethylene compressor discussed in this paper, a stalled torque equal to that developed in normal operation close to surge would result in the compressor being decelerated from 7000 rpm to 1000 rpm in about 3.3 sec. It can be seen from Table 1 that some of the deceleration times obtained imply very large fractions of this torque! For the high pressure cracked gas compressor, a similar calculation gives a deceleration from 7000 to 1000 rpm in about 8 sec, with the major difference being the greater inertia of the turbine.

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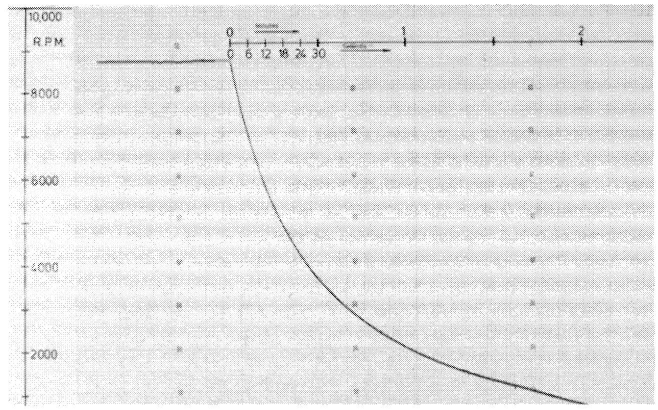

Figure 13. Speed Trace, Ethylene Compressor Turbine Solo, 27 July 1979.

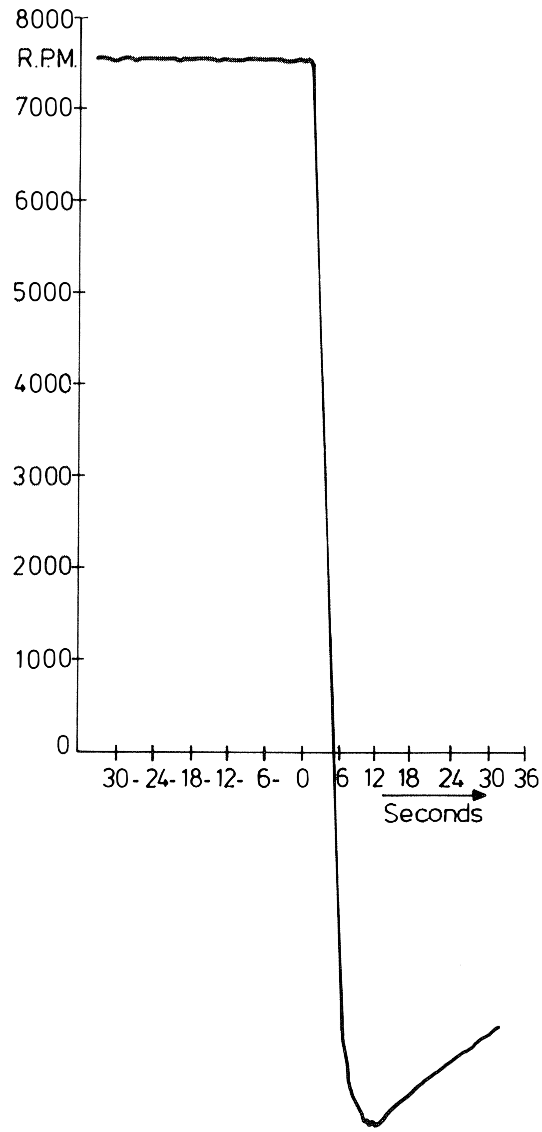

Figure 14. Speed Trace, Ethylene Compressor, 18 June 1982.