CENTRIFUGAL COMPRESSORS FOR LARGE AMMONIA PLANTS

Possible failures are investigated with the purpose of preventing them in actual operation

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In recent years the design, manufacture, and operation of turbomachinery have all become considerably more sophisticated. The past four years have witnessed the debut of the centrifugal compressor to the high pressure refining and petrochemical processes.

Centrifugal compressors have long been recognized as being rugged and durable equipment capable of making multi-year runs between plant turnarounds. It is also unforgiving. In brief, the operator should not anticipate satisfactory operation in a partially failed state. High pressure, high speed turbomachinery conducts its own go or no-go inspection. Strangely enough, this characteristic actually enhances the probability of a hazard-free operation.

Take three possible failures

For the purpose of hypothetical evaluation three possible failures are considered here with corresponding design or operating provisions for the prevention of same. Such provisions are intended as examples, rather than exclusive "solutions".

A shaft seal failure in the high pressure case of a syngas unit can be associated with one or more of the following:

1. The supporting seal oil system

2. The seal, shaft, and seal housing assembly

3. Operation of the seal outside the envelope of operating parameters associated with the seal design

Let's look a moment at that part of the seal system usually mounted on a separate console, Figure 1. Put most basically, the system must provide a flow of oil-cooled, filtered and sustained continuously at some positive increment over the gas pressure being sealed. The available hardware for today's high pressure system is surprisingly marginal. Pumps, seal oil filters, and pressure or level contollers require more than a precursory investigation. Standards of design and fabrication of the piping system must be oriented toward high pressure, just as certainly as is the process gas piping. There are some interesting design features that are peculiar to high pressure seal oil systems, e.g., the double manifolded seal with the overhead tank arranged for back-pressure control, Figure 2.

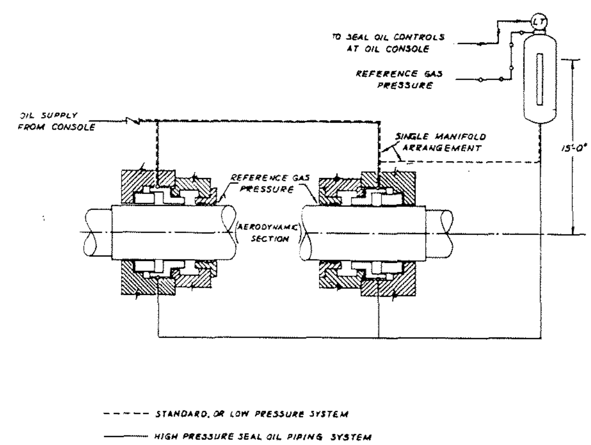

Figure 2. Schematic of double manifold seal oil piping.

Safety engineering, as applied to heavy machinery, anticipates the failure. Assume loss or damage to the bearing material of an outer seal ring. A differential of 2000 lb./sq.in. across an excessive clearance will cause a high oil flow and corresponding high pressure drop at the inlet manifold and seal porting. Comparison of the standard low pressure system helps to explain - by difference - the back pressure control arrangement.

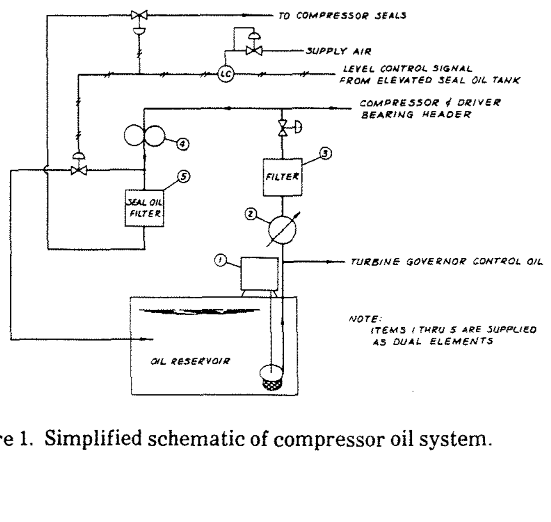

Figure 1. Simplified schematic of compressor oil system.

The seal is vulnerable

A seal assembly is illustrated in Figure 3. It would appear that the seal rings are free to float radially and center on the shaft. Actually during high pressure operation, the thrust load at the radial face tends to lock up the outer ring. This, combined with the necessarily tight clearances, makes the seal especially vulnerable to shaft vibration. Rotor unbalance, operation at or near a critical speed or loss of stability due to oil whirl at the bearings can be the source of a high pressure seal failure.

There is an operational error that is so easily done that it bears repeating here. A high pressure seal is by definition intended to operate with high pressure oil. But the oil pressure is normally keyed to the compressor inlet pressure. High speed operation at low pressure will impose a high temperature condition. An external thermometer in the drain piping may actually show a reduced temperature because of the drastically reduced oil flow rate. Limiting instrumentation and or strict operating boundaries are the obvious solution.

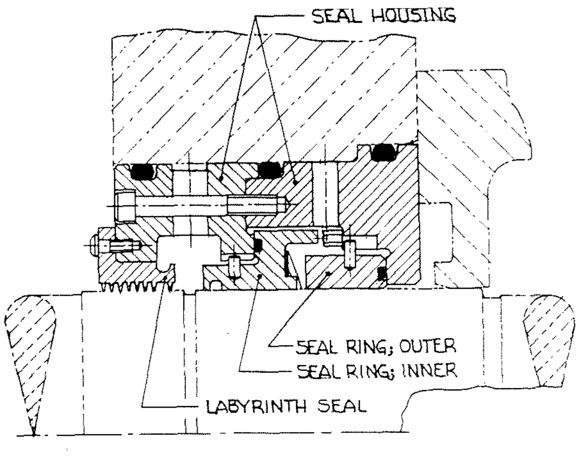

Figure 3. High pressure seal section.

Thrust bearing problems are as old as rotating machinery. Much of past centrifugal compressor design effort has been associated with the calculation of thrust load for accurate balancing and in the development of bigger and better thrust bearings. See Figure 4 for an illustration of relative thrust loads. Much of the problem revolves about interstage labryinth wear. Performance of the low capacity impeller is closely keyed to the interstage labryinth. Thus the compressor may be down for refitting of a labyrinths before dangerously high thrust loads are experienced. Oversizing of the balance piston and use of a high capacity, double-acting thrust bearing are standard design techniques.

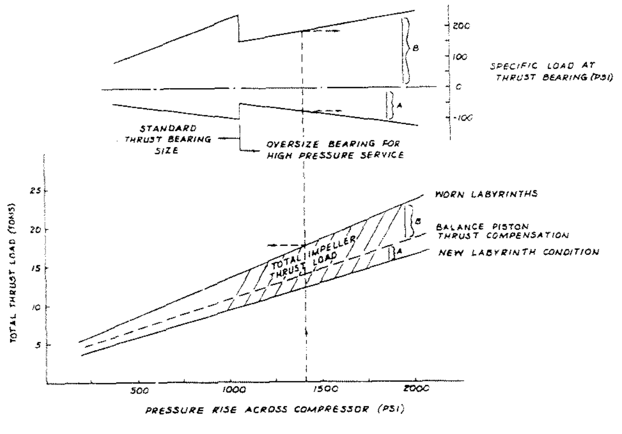

Figure 4. How thrust loading affects compressor.

High thrust load a phenomena

Operationally, the thrust bearing can be vulnerable to unstable flow conditions. High thrust load, repeating and reversing, is a phenomena associated with and the direct result of compressor surging. Compressor surging has caused damage outside the confines of the compressor itself. Check valves, pipe joints and piping supports are unusually stressed during a high-energy-release surge. Expected or test performance curves normally show the expected surge line and controls should be set at least 5% to the right of the surge line. Depending on the application, it may be essential to compensate the CFM computer with the flowing gas density and the controller setpoint can be biased with compressor speed. One refinement that should be in general use is the tuning of the anti-surge recycle valve in its response to the controller. Rapid opening combined with a sharply reduced closure speed will cause the valve to drive open during a period of unstable or cycling operation. On more than one occasion the writer has witnessed the performance of an anti-surge control system that had a slight tendency to respond faster in the closing direction. In each of the cases the controller was unable to re-establish stable conditions once an oscillating input was seen by the flow meter.

A third area of interest is the impeller, the very heart of a centrifugal compressor. There are the obvious and the very subtle design errors that can precipitate a failure, especially in the more exotic, high performance impellers. But I would like to bring to your attention an operational problem that you may well have an opportunity to grapple with. One of the features that has so popularized the use of the centrifugal in the high pressure processes, especially in recycle services, is its apparent ability to shrug off liquid ingestion--within a reasonable tolerance. Obviously, a heavy slug of liquid would be sudden death to such a high speed machine, but its ability to handle the 1% or 2% liquid phase, that escapes the separator, is generally acknowledged and even taken for granted. The quantity of liquid may increase such that, where there was once a laminar, creeping film along the impeller vanes, there is now a turbulent high velocity channel. Under this circumstance, erosion takes place and, if the condition persists, a complete and sudden impeller failure will eventually take place. Figure 5 illustrates three different types of impellers:

A. The all welded construction

B. Milled with the cover or shroud riveted and

C. The fabricated impeller employing short rivets at both the disc and the cover and blades or vanes of characteristic Z section

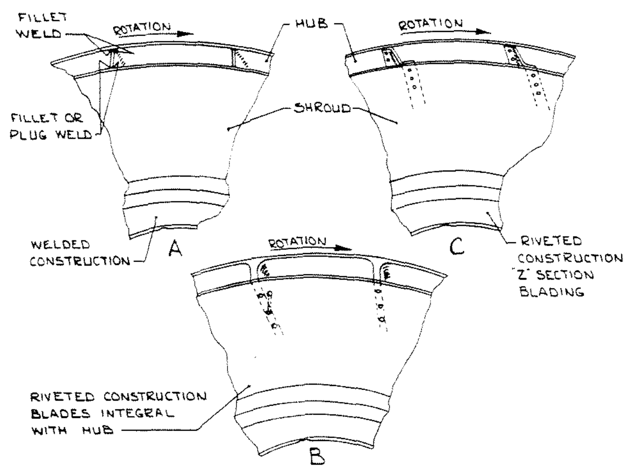

Figure 5. Impeller flow section characteristics.

Erosion rates can produce failure

The third or fabricated type presents a trapezoidal gas flow section and, while this works well with gas, any of the other two phases will naturally flow along a well defined line where the disc and blade join. Thus for a given flow rate the effective hydraulic radius of the liquid stream is increased. The radial velocity of the liquid and or entrained solids will more nearly approach that of the gas phase. Erosion rates have been experienced such as to produce a failure within a period as short as 16 months. It is apparent that the first two will tend to distribute a high density phase evenly across the face of the blade. Thus the flow pattern is reduced to a laminar film of low velocity relative to the blade surface and the erosion rate should be negligible within the normal separator tolerances.

Any discussion of machinery and its attendant ills, even if hypothetical, runs the risk of creating a somewhat negative attitude. So much genuinely successful experience has been accumulated in recent years that I can be confident that the reader cannot be so affected. I have brought to your attention three classic problems that have in the past, and may again, come to the attention of process plant safety engineers, as well as maintenance personnel. The solutions, as is so often the case, are a combination of design and operational technique.

Discussion

Q. In one of your opening statements, did you imply that because of the fact that this compressor operates in what you call a go or no-go condition, does this mean it was safer to operate than the standard machine, or something like that? Or did I not get the point?

Hile: What I tried to say was that the first time that a fault appeared in a centrifugal compressor, normally the result is that you have a shutdown looking at you. It is pretty hard to run the machine with as much as a single fault.

Q. Yes, but supposing the fault occurs when it is in operation.

Hile: It is possible that the fault will produce a hazard.

Q. Then it is not so safe?

Hile: I couldn't deny that. I think the point here is that during a start-up procedure, any faults within the machine often will show itself even before the plant is on the line. Those of you who have started centrifugal compressors in large process plants may remember that the debugging operation sometimes draws itself out almost interminably, but once on the line and running, the thing seems to make up for itself. This is what I mean by the fact that the machine seems to be so unforgiving. It is very intolerant of anything wrong with it fundamentally.

Q. I think there is a question of misunderstanding between the two of us. I just can't quite agree with that.

Q. Have you encountered stress corrosion cracking fatigue, corrosion fatigue, or hydrogen damage as a phenomena that has impaired the reliability of your rotating equipment in the various facilities of which you have experienced?

Hile: I think that I could say yes. This is not an unknown phenomena in a centrifugal compressor.

Q. But can't you by anticipating, and using materials which are appropriate for the service, avoid the problem?

Hile: Yes. For example, if we have a known situation where there is a high hydrogen sulfide, especially a wet hydrogen sulfide mixture coming through the compressor, there is a way that we can handle this.

Thomas Kuivenen (Cooper-Bessemer Co.): In cases of highly corrosive gases, particularly in the sulfide families, we will use the 1704 ph steels for the impeller construction.

An interesting phenomena exists here. To design a high speed centrifugal compressor impeller does not permit the usual selectivity of materials that you have in designing other things in a plant. It is impossible, for example, to use any of the 18 and 8 stainless steel families because the yield strength is so reduced that the designer cannot get enough speed in the machines to be able to pump anything practical so you are confined.

We have also looked into some constructions but had not made any yet considering the straight chromium steels.

Q. Would you consider air a corrosive medium?

Hile: No.

Q. Would you comment on the possibility of having corrosion or erosion of the recycle wheel of a syngas compressor due to the possible presence of ammonium carbamate?

Hile: Our limited experience in this line has not revealed anything of this nature. However, there has been a lot of prognostication that eventually the centrifugal compressor would suffer some from carbamate. I guess all I can say is we are watching for it.

Q. Has Cooper-Bessemer taken any steps to design for a specific separation of the re-cycle flow, an isolated wheel, or something like that.

Hile: The re-cycle wheel has been incorporated into the third stage, or high stage of the syngas machine, and this, of course, is typical. Towards avoiding carbamate problems, a 4-nozzle case has been developed with the re-cycle impeller turned around, in effect, facing the reverse direction of the third stage of feed-gas impellers. The third stage discharge of the feed-gas compressor, then, is taken out and cooled. Hopefully the stream is sufficiently cleaned up that by the time it comes into the re-cycle impeller, we will have no carbamate problems. This has been given quite a bit of treatment by a number of contractors. There is considerable diversity of opinion, however. I talked to several contractors who absolutely disavow any consideration here. They feel this problem has gone away. I can't honestly speak to this thing. I don't know what the basis of their opinion is.

Q. I was just wondering, because I guess it is obvious that this is something that has to be specifically considered in conjunction with design.

Q. What maximum discharge pressure can you practically reach now with the centrifugal machine? When I say practical, I mean what would you be willing to install?

Hile: I think this depends on the flowing rate. If we were talking about a 600/ton day plant, the flow co-efficient that we experience in the last impeller of the feed-gas section, is quite low. It represents about the minimum we can get to if you establish converter level something on the order of 2,500 lbs. As we go up now in tonnage rate the pressure level we can reach is obviously higher because the total volume of the gas continues to increase, despite the fact that the specific volume is going down.

I would judge, if we can talk in terms of a 2,000 ton/day plant, the 4,000 to 5,000 lb. level would be quite reachable.

Clark (ICI England): You may be interested in our experience when setting up a line of three 350 lb/sq.in. multistage air compressors. The first ran about 1100 hours before misbehaving: when opened up the 4th stage impeller blades were found to have broken loose and were plastered into the diffusers. No. 2 was then tackled, and ran for 1200 hours and then failed in exactly the same way. No. 3 ran about 1250 hours and was then opened up and fortunately only one blade had broken out and it was possible to determine the cause.

The compressors had aerofoil section blades with integral rivets machined on both ends. The rivets were failing by a combination of fatigue and hydrogen embrittlement - stress corrosion cracking from the slightly acid condensate from the air. The interstage cooling was such that stages 1-3 were dry but stage 4 was wet. To overcome it we assembled the replacement impellers, by coating the internal surfaces of the backplate and the shroud with epoxy resin, inserting the blades, putting on the shroud and rivetting it up while the epoxy was still mobile, and in this way we were able to get a complete seal around the blade roots, the places where corrosion, fatigue etc. might happen. It was interesting that whereas normally when you over-speed test such impellers, you expect to see one or two rivet heads showing a slight sign of moving, with the epoxy there was so much additional strength from the glueing, that there wasn't the faintest sign of any movement on any rivet. We consider this a very satisfactory procedure, and will use it generally in future.

Q. Didn't you have some rather better than average stainless steel from a yield point of view?

Clark: The ones you are thinking of were a different set of air compressors built up by welding together various items of heat treated 1% Cr-Mo steel. The original impellers cracked along the H.A.Z. of the weld: there was much argument as to whether post-weld heat treatment should have been applied. The material was changed to a steel (FV520B) somewhat similar to 17-4 PH. The only thing that went wrong with this steel was that cracks at welds were detected magnetically on inspection after a period of running and it took us a little time to realize that they were not cracks but only an indication of the formation of a thin layer of non-magnetic austenite as a result of the heat of welding.