CONTROLLING CENTRIFUGAL COMPRESSORS

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Abstract

Various methods of compressor performance control are discussed along with their impact on the aerodynamic efficiency of the compressor. It is emphasized that while the objective of providing the correct discharge pressure at the correct flow can be achieved using the various control methods, some are much more efficient than others and careful evaluation of the impact on compressor efficiency of a proposed compressor control method should be a part of any process plant design. It is also shown that the current API performance guarantees may not adequately address the role that the control method plays in determining which of the various API specifications apply during compressor testing and a new approach is proposed for establishing the performance guarantee specification in the case of suction-throttled compressors.

1. What do we mean by compressor control?

When a compressor vendor or end-user refers to the "method of compressor control", the intent is to describe the method by which the head-flow characteristic of the compressor can be manipulated so as to allow the user to meet various operating requirements. Typical processes that use compressors rarely have only one operating point. It is not unusual for there to be several operating conditions with gas constituents, molecular weights, volume or weight flows that vary widely. To meet all these requirements, a compressor must be "forced" to move in increasing or decreasing flow directions and/or increasing or decreasing head or pressure directions (Figure 1). The methods by which these are achieved are referred to as the method of compressor control.

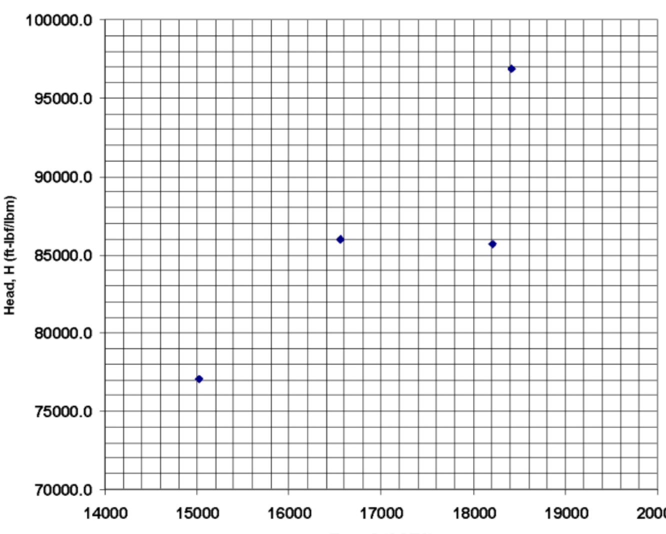

Figure 1: Typical Spread of Compressor Operating points as functions of head vs flow.

2. Methods of compressor control:

The main methods used to control a centrifugal compressor to achieve the objectives outlined above, shown on the API data sheets are:

1. Variable speed drive: This can be achieved using steam turbines, variable speed motors or fixed speed motors coupled to a fluid drive. The speed of the compressor can be changed up or down as needed to meet particular head or pressure requirements.

2. Inlet guide Vanes: These are usually used with fixed speed motors and overwhelmingly with single stage compressors, although, applications involving multi stage compressors are not uncommon. An inlet guide vane relies on the variation in head with inlet swirl velocity theorized by the Euler equations of turbo machinery, to allow the user to control the head produced by a compressor.

3. Inlet suction throttling: This is a method whereby the inlet pressure to a compressor is reduced using a throttle valve; this tends to increase the inlet volume to the compressor (for a fixed inlet weight flow) and also reduces the discharge pressure coming out of the compressor to a predetermined value. It is used mostly with fixed speed motor driven compressors.

4. Cooled bypass: In situations where the compressor is producing the correct discharge pressure, but the amount of flow going on to the process is excessive, the end-user may choose to take the

excess gas, cool it and route it back to compressor inlet, allowing only the amount of gas required for the process to continue on. The compressor may therefore be run at its best efficiency point, or outside of potential surge, while the process plant gets only the amount of gas it needs.

5. Discharge blowoff: In this environmentally conscious era, discharge blowoff, which as its name implies is a compressor control method whereby excess gas is blown off into the atmosphere is most likely to be used only with air compressors. The effect is similar to the cooled bypass except for the savings attributable to the lack of a cooler in the process.

3. Comparing the different method of compressor control:

In the discussion below, the advantages and disadvantages we will consider are limited to the realm of compressor aerodynamic performance. Besides aerodynamic performance, there may clearly be reasons of cost, maintenance, availability of steam or electrical power, environmental considerations etc that may drive the choice of a particular control method over another. These considerations are all valid considerations but are not included in this analysis. Such considerations are clearly site or project specific and are outside the scope of this aerodynamic review.

3.a.Variable speed drives: Broadly speaking, and within a reasonable range of relatively low Mach numbers, the non-dimensional characteristic of a compressor is fairly independent of the rotational speed at which the compressor runs. As such, it is relatively easy to run multiple operating conditions at or near the peak efficiency point of the compressor. Figure 2 demonstrates this. All variable speed drive

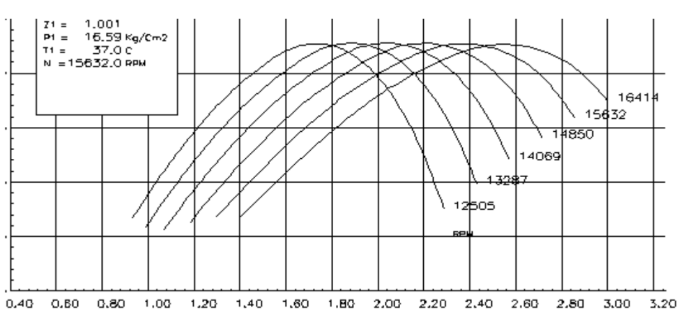

Figure 2: Efficiency characteristic for variable speed drive compressors.

compressors, whether driven by a steam turbine or variable frequency motor or fluid clutch display a similar efficiency characteristic over a wide range of speeds. Of course, as indicated earlier, not all sites may have access to the steam or other items necessary to support this type of driver, but it is clear that compressor designers are more able to optimize the efficiency of the compressor for multiple operating points if the driver is a variable speed driver.

3.b. Inlet Guide Vane Control: In the experience of the writer, variable guide vanes are the most popular compressor control mechanisms for single stage compressors, although as pointed out earlier, they are found on multi stage compressors also. Figure 3 is a plot of typical peak efficiencies of a guide vane controlled single stage compressor as a function of the inlet guide vane setting.

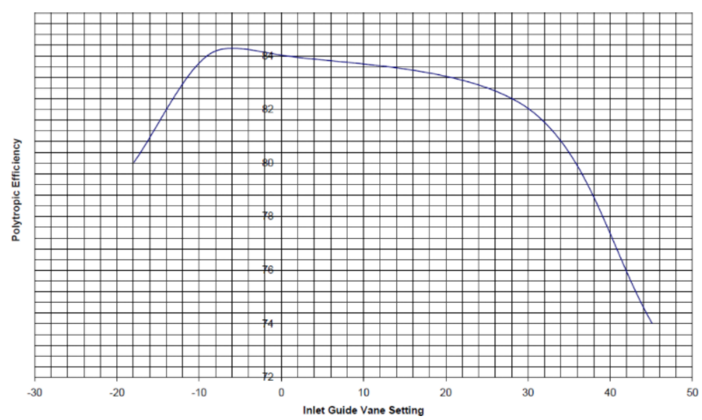

Figure 3: Typical peak efficiencies at different inlet guide vane settings.

Figure 3 indicates that while the compressor controlled by an inlet guide vane can maintain reasonable peak efficiencies up to about 25-30 degrees of guide vane setting, the peak efficiency drops fairly precipitously soon after that, along, of course, with the off-peak values. Note the drop in efficiency at anti-whirl (negative guide vane) settings also. Therefore, while a variable speed compressor might be able to maintain fairly high efficiencies across a wide spectrum of head values using the speed control, such is not the case with a compressor with inlet guide vane controls. Again as with all such comparisons, this may not be a reason to specify a variable speed control in all cases. In a lot of situations, the inefficient guide vane settings may be “Start up” cases which are run very infrequently, perhaps only at the beginning of plant commissioning, so poor efficiencies during such a temporary period may be well worth the cost savings of an inlet guide vane control when compared to a variable speed drive. Aerodynamically speaking though, the variable speed drive is to be preferred over the inlet guide vane control.

3.c. Inlet suction throttling: Figure 4 shows typical non-dimensional test curves for a centrifugal compressor. These curves show the polytropic efficiency, the work input coefficient and the head coefficient. Such curves are the initial curves generated during an aerodynamic performance test of a centrifugal compressor. They are typically generated using a test gas that is different from the process gas. The field performance of the compressor is then derived using these non-dimensional curves.

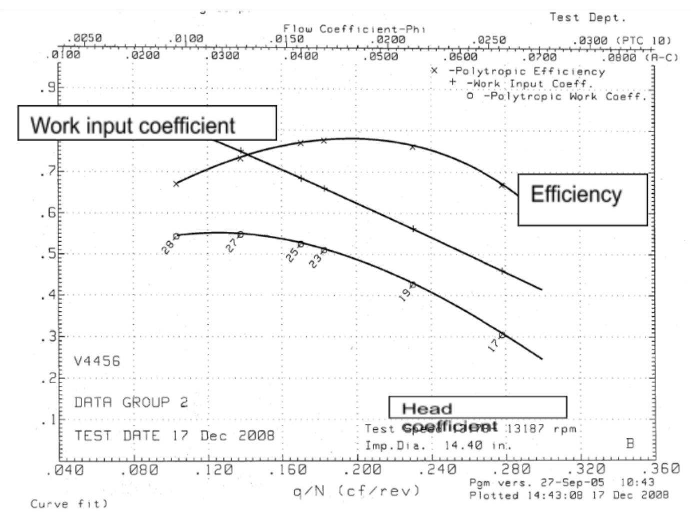

Figure 4: Typical non-dimensional curves for a centrifugal compressor.

Suction throttling implies that a pressure-reducing device of some type will be used ahead of the compressor inlet to reduce the upstream pressure before the inlet of the compressor. Such an approach is a relatively inexpensive way to control a fixed speed compressor. It, however, results in the compressor consuming more power at the suction throttled operating points than is necessary to achieve the head required for the process.

By having to reduce the inlet pressure below what the customer can supply to the compressor to obtain a specified discharge pressure, the head generated at these points is always higher than required, meaning more power is consumed than is strictly necessary.

Depending on where the compressor is operating on its head coefficient curve (Figure 4), suction throttling may result in a small improvement in compressor efficiency (for fixed weight flow processes) and a small decrease in the operating head for some operating points or possibly, a small loss in efficiency but with a corresponding small decrease in operating head at other points. The measured, consumed power may therefore either remain the same or decrease slightly. This is easily understood by a careful review of Figure 4. The peak of the efficiency curve on this plot is at a q/N value of approximately 0.200 in this case. q/N refers to the compressor inlet volume flow (inlet cubic feet per minute) divided by the compressor rotational speed (rpm). It is, in a sense, the same as the flow coefficient, but with the impeller diameter term ignored. Since, for a given compressor, the impeller diameters are fixed, they may be treated as constants in the flow coefficient equation and ignored for purposes of generation of this “non-dimensional” plot.

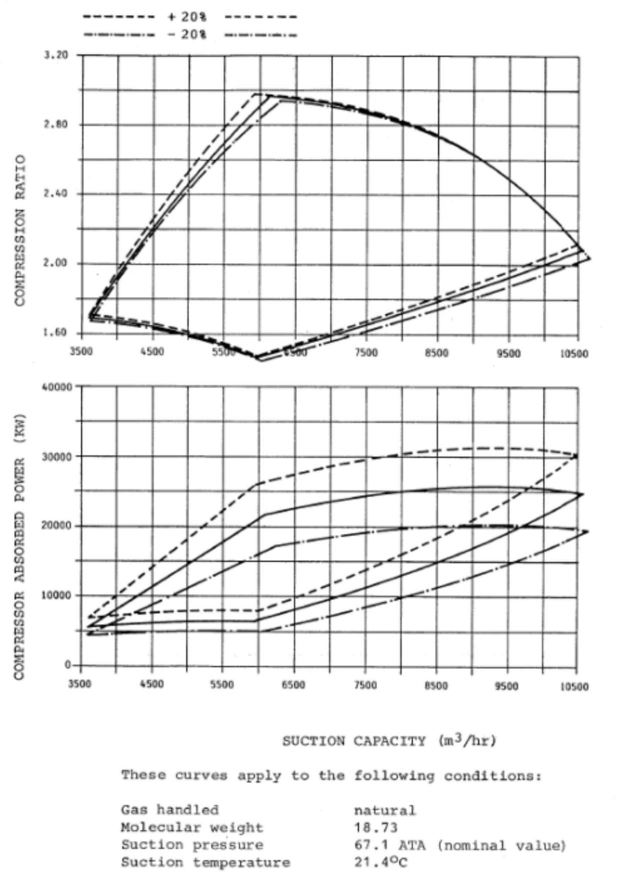

Figure 5: Effect of change of inlet pressure on compressor pressure ratio.

Suction throttling, by itself, does not reduce the head generated by a compressor. The reduction in inlet pressure that results in a lower discharge pressure does not necessarily imply a reduction in head. Figure 5 demonstrates that changing the inlet (suction) pressure by +/- 20% has no effect on the compressor pressure ratio, besides some small effects on the range of the compressor. Since the head is proportional to the pressure ratio, the constancy of the head value should come as no surprise.

The changes in head that may be observed as a result of suction throttling are observed only for fixed weight flow processes. They are explainable by the fact that, for a given weight flow, the corresponding inlet volume to the compressor is inversely proportional to the inlet pressure. Upon suction throttling, the inlet volume flow to the compressor increases. This moves the operating point on the non-dimensional curve shown in Figure 4 further to the right. Due to the fact that the head curve decreases with flow, it implies that this increase in flow due to suction throttling will result in some head decrease for the compressor. How much of a decrease depends purely on how much the volume flow increases as a result of the decrease in inlet pressure. The impact on efficiency is obvious. If, in moving its operating point to the right, the compressor is now operating nearer the peak of the efficiency curve, there may be both a gain in efficiency and a decrease in head resulting in some decrease in measured power consumption. If the new operating point is moved enough to the right that the compressor now sits at a lower efficiency point than before, then it is unlikely that the power consumption would improve. Since the head is lowered along with the compressor efficiency, then depending on the relative changes in these two parameters, there may be no change in overall power consumption. Suction throttling, therefore cannot be expected to improve power consumption in all cases. Indeed, test results for the numerous compressors tested at GE Oil & Gas (North America) show that in most cases where there is some power consumption improvement, there is only a very slight decrease, often less than 1%.

3.d. Cooled Bypass: It cannot be over-emphasized that a compressor exists only to provide a fluid with certain thermodynamic properties to a process downstream of the compressor. What these properties are coming out of the compressor and what the downstream processes require may not always be in tune. Taking some volume of gas that may be more than the downstream process needs and routing it back to compressor inlet can at times ameliorate this mismatch. There is only one problem with this approach. The gas coming out of the discharge nozzle of the compressor is at a much higher temperature than that at the inlet. If these two were mixed without cooling the discharge gas, the discharge temperature would rise further and a vicious cycle would be set up leading to intolerably high temperatures in the compressor. All compressors have their maximum allowed operating temperatures beyond which O-rings and other items would begin to fail resulting in a possible shut down of the compressor when design temperature limits are exceeded. To avoid this, it is desirable to cool that portion of the discharge gas that is being routed back to compressor inlet, hence the term "cooled bypass".

This type of control, of course requires a cooler on site, which adds to plant costs, but it is an acceptable form of compressor control. From a performance point of view, it is also wasteful of energy. Compressor power consumption is proportional to the weight flow through the compressor. Cooled by-pass by its very nature implies that far more gas is being compressed than is needed by the downstream process. End-users, of course, are responsible for the overall economics of their operation and there may be very good reasons why this is an acceptable mode of operation. It may be a temporary mode, a start up mode or some other acceptable mode of operation. The discussion here only addresses the aerodynamics of these various modes of operation. The end-user presumably knows what is best for the overall economics of the plant.

3.e. Discharge Blow off: Discharge blow off as its name implies, means that excess gas at the discharge end of the compressor is blown off, presumably to atmosphere. Clearly, as stated earlier, this type of compressor control can only be applied to air compressors. They do have the advantage though that the need for a cooler is eliminated. The power wasting aspect of this type of compressor control is the same as for the cooled bypass, if one ignores any power consumed by the cooler.

At the end of the day, a compressor, no matter how it is controlled, must pass a performance test usually at the vendor's test stand. The API Standard 617, currently in its seventh edition, for compressors built to the standards of the American Petroleum Institute, governs such tests. The application and interpretation of these standards can be the difference between whether a compressor leaves the test stand and heads for the customer site quickly and starts a productive life or is subjected to minor or major modifications and additional testing. It may appear on the surface that the interpretation of these standards ought to be straightforward, but in some cases, this is not the case. In addition to the choice of method of control, it is extremely important that both vendor and customer have a clear understanding as to what makes for a good compressor, and how the test specifications are interpreted to determine this so that on the day the compressor is tested, both parties are on the same wavelength.

4. The impact of the method of compressor control on acceptance testing:

Under section (2.1.1.3), API Standard 617, 7th Edition states: "The compressor shall be designed to deliver normal head at the normal inlet volumetric flow without negative tolerance. The power at the normal operating point shall not exceed 104% of the predicted value".

Under 4.3.3.1.2, it further adds: "For variable speed machines, head and capacity shall have zero negative tolerance at the normal operating point (or other point as specified), and the power at this point shall not exceed 104% of the vendor predicted shaft power value. This tolerance shall be inclusive of all test tolerances."

Finally, section 4.3.3.1.4 states, "For constant-speed compressors, the capacity shall be as specified in 4.3.3.1.2. The head shall be within the range of 100%-105% of the normal head. The horsepower based on measured head at normal capacity, shall not exceed 107% of the value at the specified normal operating point. If the power required at this point exceeds 107%, excess head may be removed by trimming impellers at the purchaser's option.

These specifications appear to be quite clear, except that in all cases, they address the type of driver, but leave out the impact of the method of compressor control on how a compressor can be judged fairly as to whether it is meeting the intent of these specifications or not.

Let us begin by looking at the case of a constant speed compressor controlled by an inlet guide vane. With a guide vane, it is possible to control the discharge pressure (or head), so that there is zero excess head coming out of the compressor. In such a case, a fixed speed compressor controlled with inlet guide vanes, is more like a variable speed compressor in the sense that, its performance must be judged at the guide vane setting where the guide vane produces zero excess head and zero negative tolerance on flow. The power consumed must not, under such a scenario, exceed 104%. For this situation, therefore, even though the compressor is a fixed speed compressor, the head guarantee (100%-105%) and the power guarantee of 107% make no sense and are not in tune with spirit and intent of the API specifications. For this situation, the correct guarantee consistent with the spirit and intent of the API specifications is that at zero excess head set by the guide vanes, there should be no negative tolerance on flow and the maximum power consumption should not exceed 104%. This is the method most compressor vendors and end-users have used to interpret the results of compressor tests where inlet guide vanes are the method of control. Strictly speaking, if one were to go by the API specifications, the fixed speed guarantee (105% excess head, 107% excess power) ought to apply. In short the API sanctioned method is widely ignored and, clearly, for good reason.

The second scenario we consider here is the case where a fixed speed compressor is controlled using inlet suction throttling. In the vast majority of such applications, the compressor is in a fixed-speed operating mode. The customer supplies a maximum or not-to-exceed inlet pressure for all the operating points and vendors are asked to determine the inlet pressures at which each condition must be operated in order to meet a specified discharge pressure (usually a constant value). These vendor-predicted inlet pressures are estimated using the vendor's prediction codes and the polytropic head consistent with these predicted inlet pressures are entered on the API data sheets.

It is of the utmost importance to understand what this means. Generally speaking, these predicted inlet pressures, and the corresponding predicted head values are "meaningless" numbers. Whether they are met on test within a particular tolerance or not is more a reflection of the quality of the vendor's predictive software or perhaps the quality of the test, than any measure of whether the compressor can meet the customer's process conditions or not. So long as on test, the measured inlet pressure is below the inlet pressure specified by the customer, and the measured power meets the 107% power guarantee requirement of API 617, the compressor is and should be considered as acceptable to the end user. The concept of excess head in this case, as a measure of compressor performance, has no logical basis.

The reasoning here is quite simple, but can be obscure to some who may not have spent a lot of time thinking about this issue. The customer's only requirement is that at an inlet pressure not exceeding the customer-supplied value, the compressor must, at the indicated flow, produce the desired discharge pressure. In theory then, the head required by the customer, based on the supplied inlet pressure is the minimum head that will satisfy the process conditions. The head values (mostly higher) calculated from the suction throttled, and therefore lower inlet pressures, are relevant only so long as they do not lead to excessive (i.e. greater than 107%) power consumption at the guarantee point. In short, comparing the measured head at test with that shown on the data sheet is a meaningless exercise that has no relevance to the customer's process or the ability of the compressor to perform as required, again, so long as the 107% maximum power guarantee is met. In may in fact be argued that, for suction throttled compressors, the performance guarantee should read as follows:

"The head produced by the compressor with no negative tolerance on flow at the guarantee point must, at a minimum, equal the head determined based on the customer supplied maximum inlet pressure and customer supplied discharge pressure. The corresponding power consumed at head values equal to or exceeding this value must not exceed 107% of that shown on the data sheet. "

The irrelevance of the exact value of the measured inlet pressure (and the corresponding measured polytropic head) when suction throttling is the method of control can be demonstrated even more pointedly by the fact that, assuming that the measured inlet pressure is slightly higher than the predicted value (but still below the customer supplied value), it might appear as if the measured compressor head tolerance is negative, compared to that shown on the API data sheet. Conversely, if the measured inlet pressure is much lower than the predicted value, the measured head might appear to be excessive, again compared to that shown on the data sheet, especially if it exceeds the 105% value. It is obvious that in neither case is a remedial action required so long as the power consumption meets the 107% value. Trimming impellers in the latter case of supposed excess head is not called for, since the supposed excess head is nothing more than a mirage, nor is there a need to use bigger impellers or an increase in speed in the former case, since under both circumstances the compressor will provide the customer with the operability required, again, so long as the power consumption does not exceed the 107% value.

As stated earlier, the interpretation of API 617 as regards compressors with suction throttling can be confusing. I hope this article has helped some in clarifying this issue. It must be pointed out that this is not a "get out of jail free card" for compressor vendors since the power guarantee must still be met no matter how low the suction throttled inlet pressure has to go to meet the customer required discharge pressure value. If the power consumption exceeds 107% of the guaranteed value, then trimming of some or all of the impellers may be required. The intent behind such trimming, however, must be to bring the power level down to or below the 107% level. It should not be to attempt to meet the 105% head value shown on the data sheets based on the same arguments advanced earlier. A customer is free, however, to ask for trimming to a lower power level, at his option, but it would seem to the writer that under such circumstances, the customer ought to bear the costs of such trimming.

There are certain conditions such as refrigeration processes where the customer may require and impose a limit on how low the suction throttled inlet pressure can go, in order to avoid sub-atmospheric inlet conditions for example. Such lower limits on the suction throttled inlet pressure ought to be discussed as part of the contracting process so all parties are aware of any such limitations. At GE Oil & Gas (North America), we routinely insert the following statement in our "Comments and Exceptions" or on the API data sheet to address this issue.

"If suction throttling is used as a means of control, then (a), the inlet pressures stated above are only a prediction and the actual value will be equal to or less than the Buyer's specified value when determined from test results, and (b) the head shown above is based on the predicted inlet pressure and the final head will be based on the actual inlet pressure determined from test results.

For constant speed compressors with suction throttling, the actual head may exceed 105% at the certified point so long as the power does not exceed 107% of the predicted value for the certified point if suction throttling allows the discharge pressure to be met."

The writer believes that such a statement meets the spirit and intent of the relevant API 617 sections by filling in a gap in that document, as it currently exists while providing customers with good equipment that meets their process and control needs.

Testing for the other modes of control have generated few heated discussions in the writer's experience, unlike the suction throttled case. Since the compressor is mostly operating at its design flow in these other cases, and in either fixed speed or variable speed mode, the relevant API sections are well addressed in the specifications.

5. Conclusions:

The choice of the method for controlling a centrifugal compressor has a very significant impact on the operating efficiency of the compressor and the method of determining acceptability of the compressor during performance testing. From a point of view purely of compressor efficiency, variable speed operation appears to be the preferred mode of operation, followed by inlet guide vane control. For compressors with minor variations in head across a wide spectrum of operating points, inlet guide vanes are a relatively efficient and comparatively inexpensive method for controlling the compressor when feasible. Other methods such as suction throttling, cooled bypass or discharge blow off are quite wasteful of energy, but may be acceptable depending on the combination of installation and operating costs. It is recommended that customers, when possible, work with OEM's to evaluate the long-term costs and benefits of the various modes of control before an order is placed.

6. References:

1. Internal Test Report, GE Oil & Gas

2. Internal Test Report, GE Oil & Gas

3. American Petroleum Institute. "Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry API Standard 617, 7th Edition, July 2002."

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