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HOW WE ASSESS THE EFFICIENCY OF A COMPRESSION PROCESS: A STATISTICAL METHOD WITH APPLICATION TO STAGING DESIGN AND SELECTION FOR A CENTRIFUGAL COMPRESSOR

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Abstract

Operating efficiency of a gas compressor package can be defined in several ways and it involves not only compressor or driver efficiency at design points, but also the match in speed and power between the two components. Achieving this match becomes especially critical when the driver is a gas turbine where efficiency is inherently dependant on output speed and power, and ambient conditions. A method of analysis is proposed and exemplified on compressor installations experiencing variable operating conditions. The method was used later at the design of an application less sensitive to variations in operating conditions, ensuring predictable distribution for efficiency early in the design phase. In this way the customer was fully advised how the performance will evolve during long time operation.

1 Introduction

The nominal point of a gas compressor installation is specified by the main operating parameters: suction pressure, suction temperature, discharge pressure, flow, gas composition and ambient conditions. This information is usually provided by the customer with the API (American Petroleum Institute) data sheets, together with other required characteristics. It is also customary to evaluate the performance of a compressor installation based on its design point efficiency.

In addition to the nominal point, a number of extreme conditions like min – max suction and discharge pressure and flow can be specified in order to verify the operation at those points. Most of the time the compressor would be accepted based on meeting industry standards at the nominal point while still being satisfactory at the off design points. The compressor efficiency is typically specified at the design point where it reaches its maximum. The gas turbine efficiency is also guaranteed at the nominal power knowing that it will change with load and speed. From the combined performances of the two machines, the overall efficiency of the turbo-compressor package varies significantly at off design points.

The costs incurred in operation are strongly dependent on compressor performance at design point as well as at off design points. From all the operating costs, the gas fuel is one of the most important. Even if the fuel belongs to and is supplied by the user, it would be a mistake not to consider its opportunity cost if sold. In this paper the overall or global efficiency of a gas compressor unit will be evaluated in terms of the fuel efficiency.

2 Distributions and distribution functions

A distribution function is a representation of the *frequency of apparition* of a certain level of magnitude for a parameter relative to its value [1]. A distribution does not represent a time series or an independent to dependent parameter relationship, but tells how often a certain value occurs. Two types of distributions representing two extreme situations are most popular: *normal* (or Gaussian) distribution and *exponential* distribution.

Normal distribution is characteristic to processes where a certain value is most probable and random causes of variability independent of the process determine that value to be smaller or greater than the target, while the process is attempting to maintain the target. Its characteristics are the *mean* and the *standard deviation*. Mean is a measure associated with target value the process attempts to maintain, while standard deviation is a measure of the spread of the process around the mean. An example could be a process attempting to maintain a certain pressure in the system, while actual measured values have a spread around the target value.

Exponential distribution is an extreme situation in which only values on a single side of the target are acceptable, and the system tries to maintain a value as close as possible to the target while not exceeding it. An example could be a system attempting to maintain a peak value like the maximum possible efficiency. There are no values beyond target, and all the values attempt to be as close as possible to the target, however on the lower side.

2.1 Parameters that affect the efficiency

The gas turbine specific fuel consumption and gas compressor efficiency vary with the operating speed, with load (which means flow and pressure ratio) and with ambient conditions. All those parameters are deeply embedded in calculation formulas and not visible at the first look. A good analysis would deliver details and allow the user for an informed decision. Monte Carlo simulation method applied to compressor process is typically based on given statistical data of compressor operation estimated from engineering analysis or from past records. The statistical data include mean and standard deviation for certain parameters, as well as their specific distribution if different from normal. Distributions for parameters like suction pressure, discharge pressure, suction temperature and ambient temperature cannot be influenced by designer and will be considered given data for the process. Other parameters may be calculated for a specific design or they may be required in specifications (ex. standard volume flow).

The influence of each parameter can be summarized considering the following equations for isentropic head and inlet volume flow [2,3]:

$$H_{is} = \frac{k}{k-1} R T_1 Z_1 \left(\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right), \qquad Q_1 = W F_{gas} R \frac{T_1 Z_1}{p_1}, \tag{1}$$

and the equations for compressor required power

$$P_c = \frac{WF_{gas} H_{is}}{\eta_{is}\eta_m} = \frac{1}{\eta_{is}\eta_m} * WF_{gas} \frac{k}{k-1} RT_1 Z_1 \left(\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right) = \frac{1}{\eta_{is}\eta_m} Q_1 p_1 \frac{k}{k-1} \left(\left(\frac{p_2}{p_1} \right)^{\frac{k-1}{k}} - 1 \right) \quad (2)$$

Further analysis and testing for a specific compressor will provide detailed characteristic curves for head and efficiency function of flow and speed [4, 5]:

$$H_{is} = f(Q_1, n), \quad \eta_{is} = f(Q_1, n) \quad (3)$$

Influence of operating parameters on the compression process can be summarized as follows.

Suction Pressure is expected to vary within certain limits accepted in the process. It directly influences the inlet volume flow, and indirectly the power required and the operating speed of the compressor at a given power. It usually has a normal or a Weibull distribution and the mean and standard deviation can be estimated from past operating data. The higher the suction pressure, the lower the operating speed and the actual suction volume flow of the compressor.

Discharge Pressure also changes within certain limits and can be modeled with a normal or Weibull distribution with mean and standard deviation estimated from past data. The higher the discharge pressure, the higher the compressor ratio and also the isentropic head the compressor needs to provide. For constant suction pressure and power supplied from the gas turbine, the compressor flow gets lower when the discharge pressure increases.

Suction Temperature increase will increase the actual volume flow and compressor required power for the same mass flow or standard flow. A higher suction temperature is undesirable but sometimes unavoidable and the compressor needs to be sized accordingly. In general a higher suction temperature will move the operation point toward higher volume flow, choke and lower efficiency area. The suction temperature distribution may be uniform or normal.

Isentropic Efficiency has a direct effect on the compressor required power if the other parameters are constant. When the compressor is operated at max gas turbine power, a decrease in isentropic efficiency will be followed by a decrease in flow and an increase in compressor discharge temperature. The compressor will be nearing the surge limit. Distribution of the isentropic efficiency will be an output of the analysis process.

Suction Volume Flow has a direct relationship with the compressor power. Together with compressor speed it is the main parameter that determines head and efficiency. It may be dictated by the process, or will be a result if the turbine driver is operated at max power. In the first situation its distribution is estimated from past data, while in the second situation it is calculated.

2.2 Method of analysis based on global process efficiency

For the present paper, the process efficiency will be related only to the fuel consumption. Different compression processes will be inherently different in nature and operation (suction pressure, flow, speed etc) and the primary driver is also different from application to application. Kurz and Brun [6] define the global (or package) efficiency “in terms of the amount of fuel required to perform the compressor duty with an isentropic compressor”,

$$\eta_{global} = C_1 \frac{WF_{gas} H_{is}}{WF_{fuel} LHV} \qquad (4)$$

(C_1 being a constant accounting for the units used) and further express the global efficiency for a package as a product [2, 3, 6] of the efficiencies on each component:

$$\eta_{global} = \eta_{prim\_driver} \eta_{is} \eta_m \qquad (4a)$$

Due to physical processes in the primary driver and in the gas compressor, such global or process efficiency would always be a function of ambient and operating conditions:

$$\eta_{global} = f(T_0, T_1, p_{baro}, Q_1, p_1, n) \qquad (5)$$

We will introduce here and further use another parameter that we call *Process Specific Heat Value* expressed by the amount of fuel energy (in MJ fuel) for one thousand of Nm3 (MNm3) and one kJ/kg of head rise:

$$\textbf{\textit{Process Specific Heat Value SI}} = \frac{MJ_{prim\_driver\_fuel}}{MNm^3 * H_{is}} \qquad (6)$$

Alternately this can be expressed in Btu per one million of standard cubic ft of gas compressed and one ft-lbf/lbm head rise:

$$\textbf{\textit{Process Specific Heat Value US}} = \frac{Btu_{prim\_driver\_fuel}}{MMSCF * H_{is}} \qquad (6a)$$

Such an indicator has several advantages. First it has a physical meaning and it is directly related to production (standard flow) and performance (pressure ratio, through the calculated H_{is}). Second it allows comparison of different machines, from different manufacturers and with different design specifications, relating directly to the end user’s needs for compressing a certain amount of flow between two different pressures. Third, it can be a required design parameter directly related to the process specification. Fourth, when it is analyzed for a longer period of time or for multiple operating conditions, its distribution can be determined and will provide information not only on the design performance, but also on *how efficiently the machine is utilized and is suitable for the process*. The relationship between the process specific heat value and the global efficiency is inverse proportional,

$$\textbf{\textit{Process Specific Heat Value}} = C_2 / \eta_{global}$$

with C_2 being another a constant accounting for the units used.

3 Application of the method to process analysis and examples

To better exemplify, the method will be applied on two types of examples: first the analysis of an existing compressor over a longer period of time (examples 1 and 2.1 to 2.6), and second in the design phase of a new compressor application (ex. 3).

Note regarding data collection in the examples. All records were from the standard instrumentation of existing packages, under standard calibration and maintenance plan. Data were typically collected daily, at a random time, and are considered covering operating and ambient conditions. The condition of each machine was the one existing at the date of data collection and no consideration for equipment age, normal wear and tear or other adverse condition (like fouling) was made. The results of the analysis are not intended to represent a performance test, but they are considered representative for the operating profile of the installation and for fair comparison between machines.

3.1 Example 1

A natural gas compressor package commissioned in 2008 in a gas processing plant was analyzed based on the daily records for a period of 3 years. Its nominal design conditions are the following:

Type: centrifugal, 4 stages, impeller diameter 12”

Driver: gas turbine, ISO shaft power 3.5 MW

The operation during the 3 years was analyzed with a number of 265 records supplied by the customer. In Fig. 1 all operating points recorded were plotted on the H_{is}-Q_1 and η_{is}-Q_1 maps.

The distribution of the following parameters was calculated and plotted: on Fig. 2 process specific heat value in 2008 and in 2009-2010, and on Fig. 3 the compressor only isentropic efficiency.

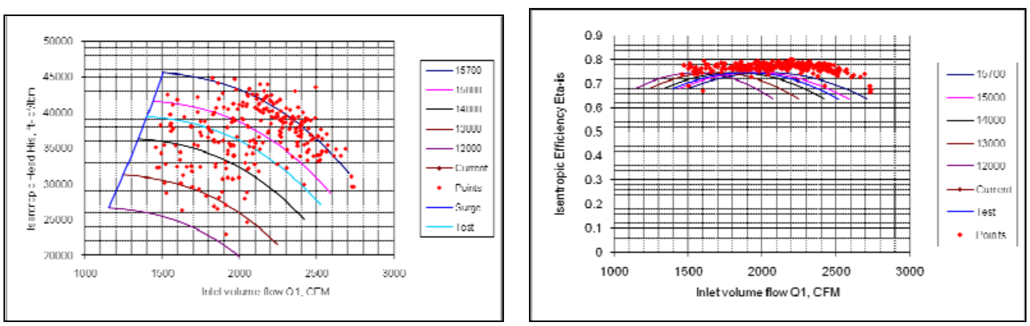


Fig. 1 – Operating points superposed on compressor maps: Had (ft-lbf/lbm) vs. Q_1 (left) and η_{is} (%) vs. Q_1 (right)

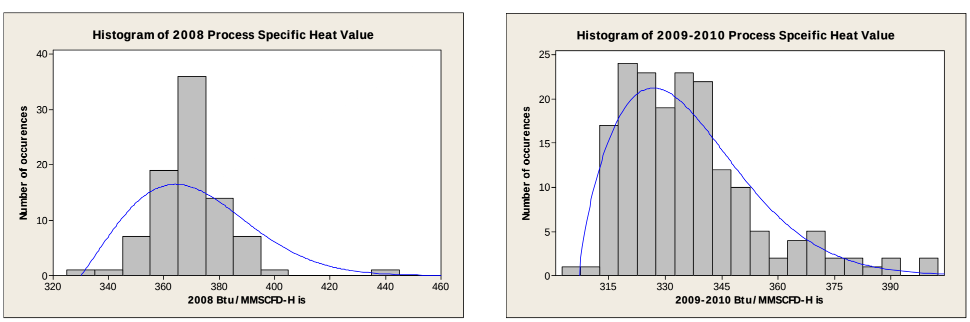


Fig. 2 – Histogram of process specific heat value in 2008(left), and 2009-2010(right)

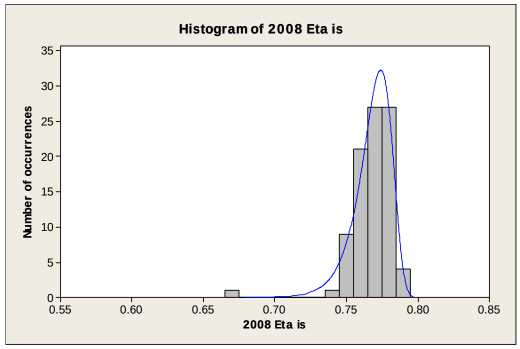


Fig 3 – Histogram with compressor isentropic efficiency η_{is}

The results of this analysis are as follows:

- Fig. 1 (left) shows good correspondence between the compressor map and where the compressor was actually operated, with a concentration of points toward max speed line which is also max power and best specific fuel consumption for the gas turbine. The compressor efficiency chart (right) shows that it operated mainly at max efficiency, which contributed directly to having a good specific heat value for the whole package.
- In 2008 best specific heat value obtained was 330 Btu/1MMSCF-1ft.lbf/lbm isentropic head, with a peak of most common operation at 370 Btu/1MMSCF-1ft.lbf/lbm
- In 2009-2010, the best specific heat value was 310 Btu/1MMSCF-1ft.lbf/lbm iesntropic head, with a peak of most common operation at 330 Btu/1MMSCF-1ft.lbf/lbm and a spread up to 400 Btu/1MMSCF-1ft.lbf/lbm.
- The change indicates better better use of the equipment in 2009-2010, which suggests better match between process and compressor compared with 2008.
- No matter how this equipment was operated, the best efficiency obtained was 310 Btu/1MMSCF-1ft.lbf/lbm
- Tuning the process (pressure and flow) may change the mostly operated value and accordingly improve the average, but not the minimum value which represents what the equipment can do the best
- Compressor isentropic efficiency is situated in the normal range for this class of units (max value 0.79 with most frequent at 0.77)
- The method of analysis can be applied to any type of equipment, and it provides comparable results that allows comparison of processess

3.2 Example 2

A number of 5 compressor units operating in parallel at one of ATMOS Energy compressor stations were analyzed based on the records collected during 2010. The units are of different type as follows:

Unit 1

Type: centrifugal, 2 stages, impeller diameter 12”

Driver: gas turbine, ISO shaft power 3.5 MW

Unit 2

Type: centrifugal, 3 stages, impeller diameter 12”

Driver: gas turbine, ISO shaft power 3.5 MW

Unit 3

Type: centrifugal, 3 stages, impeller diameter 15”

Driver: gas turbine, ISO shaft power 5.2 MW

Unit 4 and 5

Type: reciprocating compressors

Driver: reciprocating engine 4MW

The units operate in parallel as needed, are installed side by side and have a common maintenance and instrument calibration plans, which is supposed to reduce the degree of variability and to offer reliable data for comparison between units. The results are detailed as follow.

Unit 1 (Example 2.1)

The operating points from 2010 were plotted on the chart in Fig. 4. They show a pattern typical to a centrifugal compressor, with a wide variation in inlet volume flow (2000 to over 4000 CFM), as dictated by the process. Fig. 5 shows the distribution of the process specific heat value and compressor isentropic efficiency.

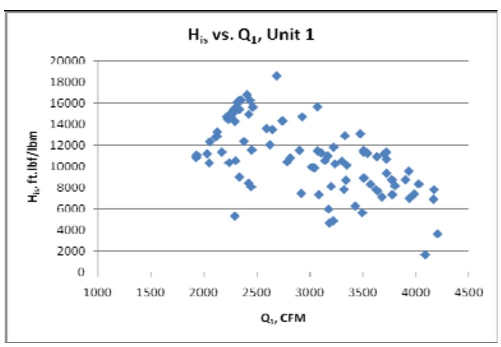
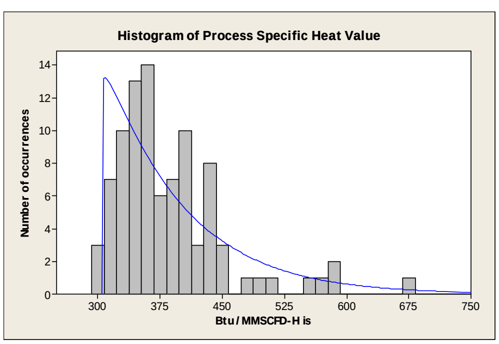


Fig. 4 – Isentropic head vs. inlet volume flow, operating points



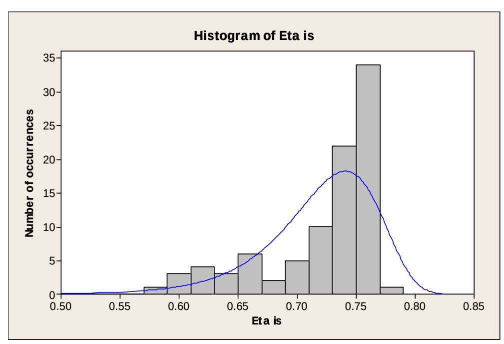


Fig. 5 – Histogram of process specific heat value (left), and of compressor efficiency (right)

The charts indicate the following:

- The best specific heat value this equipment reached in operation was 300 Btu/1MMSCF-1ft.lbf/lbm, and this represents probably the best value the equipment is capable of under normal conditions
- The value at which the equipment was mostly operated was 360 Btu/1MMSCF-1ft.lbf/lbm (20% higher than the best value)
- The maximum spread is up to over 550 Btu/1MMSCF-1ft.lbf/lbm.
- The compressor best efficiency was 0.79, and the value at which was mostly operated was 0.76.

Unit 2 (Example 2.2)

Similar to Unit 1, the operating points from 2010 were plotted on the H_{is}-Q_1 chart in Fig. 6. The pattern is typical to a centrifugal compressor, with a variation in inlet flow between 1600 and 2600 CFM (much smaller range than Unit 1, understandable as this is a different staging).

The distributions for process pecific heat value and for compressor isentropic efficiency are plotted on the charts in Fig. 7.

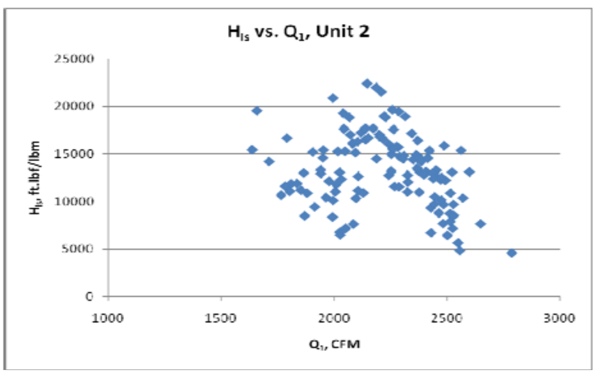


Fig. 6 – Isentropic head vs. inlet volume flow, operating points

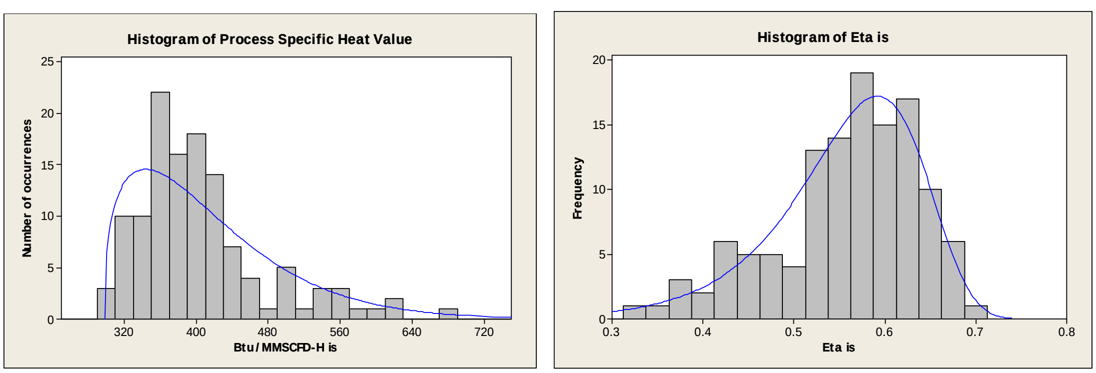


Fig. 7 – Histogram of process specific heat value and of compressor efficiency

The charts indicate the following:

- The best specific heat value this equipment was operated as is 310 Btu/1MMSCF-1ft.lbf/lbm

- The value the equipment was mostly operated was 360-370 Btu/ft.lbf/lbm (almost 20% higher than the best value)
- The spread of values is up to over 600 Btu/1MMSCF-1ft.lbf/lbm
- The compressor best efficiency was 0.75 with most frequent value below 0.7, which may indicate a problem or an opportunity for restaging in order to get the efficiency similar to unit 1. However in a discussion with the customer, they indicated that this compressor was overhauled at the end of 2010 and was found in a very dirty condition. A revaluation was proposed in order to determine if a restaging would be beneficial.

Unit 3 (Example 2.3)

Following the same approach, the operating points from 2010 were plotted on the H_{is}-Q_1 chart in Fig. 8. The pattern looks typical to a centrifugal compressor, with a variation in inlet flow between 2300 and 5000 CFM (much wider than Unit 1 and 2 and understandable as this is a different size compressor).

The distributions for process specific heat value and for compressor isentropic efficiency are plotted on the charts in Fig. 9.

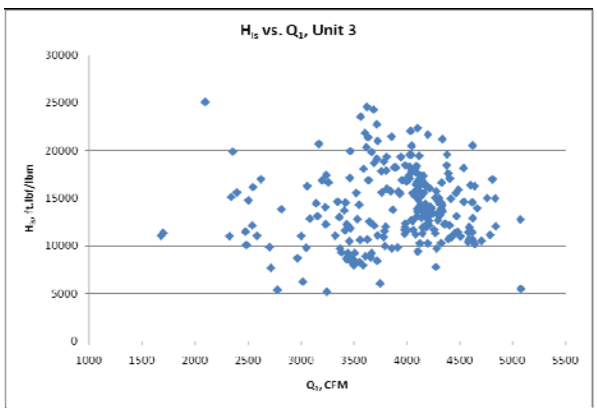


Fig. 8 – Isentropic head vs. inlet volume flow, operating points

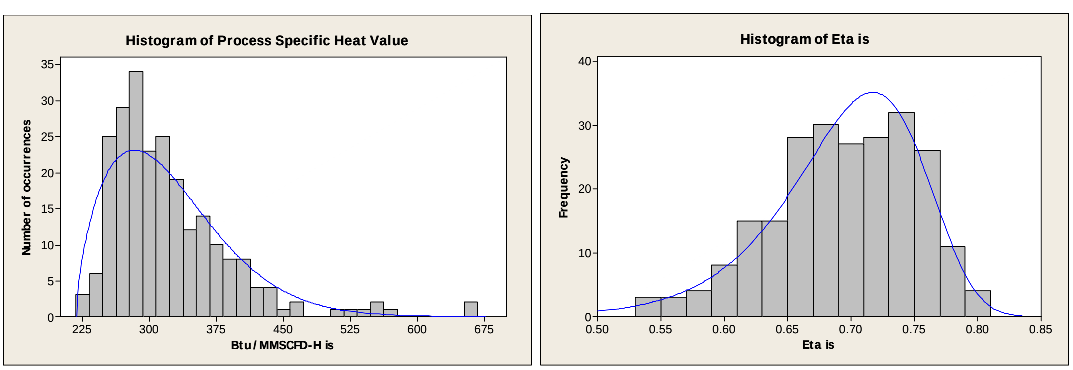


Fig. 9 – Histogram of process specific heat value, and of compressor efficiency

The charts indicate the following:

- The best specific heat value this equipment was operated was 225 Btu/1MMSCF-1ft.lbf/lbm, which is 25% better than Unit 1

- The value the equipment was mostly operated was 285 Btu/ft.lbf/lbm (27% higher than the best value)
- The spread of values is up to 550 Btu/1MMSCF-1ft.lbf/lbm
- The compressor best efficiency is at 0.83, with mostly operated value at 0.74

Unit 4 (Example 2.4)

This is a reciprocating unit, however the analysis will follow the same pattern as the previous ones. The operating points from 2010 were plotted on the H_{is}-Q_1 chart in Fig. 10. The pattern shows a variation in inlet flow between 1300 and 2500 CFM.

The distributions of process specific heat value and of compressor isentropic efficiency are plotted on the charts in Fig. 11.

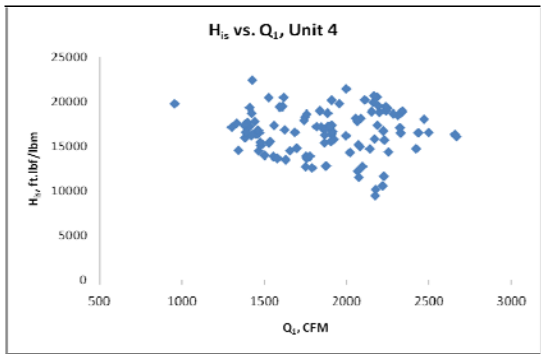
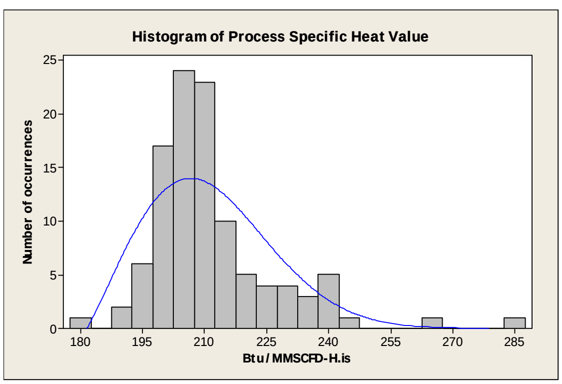


Fig. 10 – Isentropic head vs. inlet volume flow, operating points



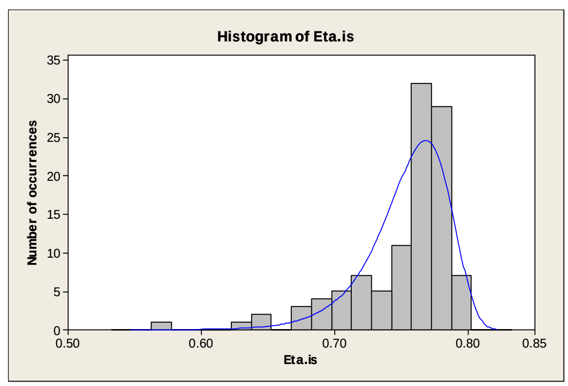


Fig. 11 – Histogram of process specific heat value, and of compressor efficiency

The charts indicate the following;

- The best specific heat value the equipment was operated was 180 Btu/1MMSCF-1ft.lbf/lbm, which is 20% better than Unit 3 and 40% better than Unit 1
- The value the equipment was mostly operated was 205 Btu/ft.lbf/lbm (14% higher than the best value)
- The spread of values is up to 250 Btu/1MMSCF-1ft.lbf/lbm
- The compressor efficiency was max 0.795, with most operated value at 0.765

Unit 5 (Example 2.5)

The operating points from 2010 were plotted on the H_{is}-Q_1 chart in Fig. 12, showing a variation in inlet flow between 1300 and 2500 CFM.

The distributions for process specific heat value and for compressor isentropic efficiency were plotted on the charts in Fig. 13.

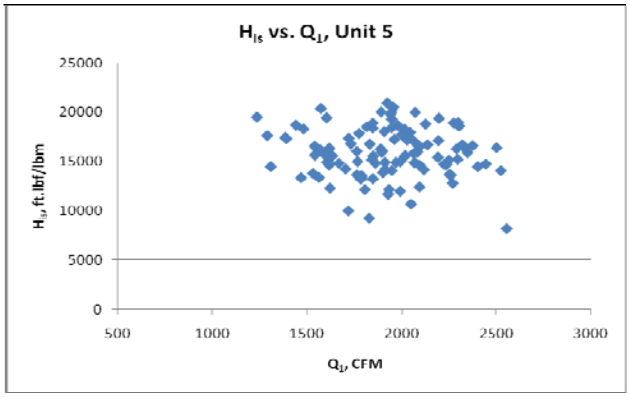


Fig. 12 – Isentropic head vs. inlet volume flow, operating points

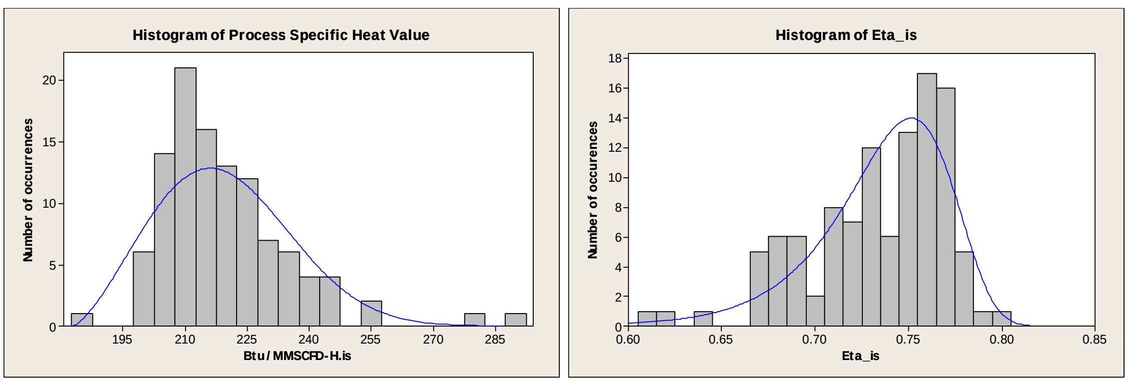


Fig. 13 – Histogram of process specific heat value, and of compressor efficiency

The charts indicate the following:

- The best specific heat value this equipment was operated was 180 Btu/1MMSCFD-1ft.lbf/lbm (same as Unit 4)
- The value the equipment was mostly operated was 210 Btu/ft.lbf/lbm (16% higher than the best value)
- The spread of values was up to 255 Btu/1MMSCF-1ft.lbf/lbm
- The compressor best efficiency is 0.80 and mostly operated value at 0.76

3 Discussion

The method proposed and exemplified introduces a new parameter – the process specific heat value - with a certain advantage in quantifying the energy consumption per unit of flow compressed and head rise created by a compressor. The most powerful interpretation is obtained when the results are plotted on a histogram rather than presented as an average because in this way they show three important metrics: what the equipment did best, where the equipment is operated most of the time, and what is the spread in fuel efficiency (best to worst) that the package experienced during the analyzed period of time.

The first value – what the equipment did best – is the minimum value seen on histogram of the specific heat value. It can be set as a target for future operation, and it helps understand how – if possible – tune the process operation in terms of pressure and flow in order to get the best advantage of the fuel consumed. The second value – where the equipment was operated most of the time – is the median value seen on the histogram and is close to the average value measured and reported currently. It is a good indicator for periods of long term operation and although it will not give an indication of how the process could be improved, when compared with the design point of the unit it will help understand if there will be an improvement if restaging that compressor. The third value – the spread – gives an indication for the variability of the process and will help understand if and how it can be reduced, being known that processes with wider variability are less efficient that the ones with reduced variability. A fourth indication is given by the shape of the distribution plot. It is strongly desired to have the distribution of specific heat value parameter strongly skewed, with the process operating at or close to best efficiency most of the time.

4 Application of the method during design of a compressor application.

In the following example the method was applied while in the design phase, with assumed distributions for input parameters – suction and discharge pressure, flow, ambient conditions – with the purpose to select the best design that will accommodate the diversity of conditions, and of predicting the specific heat value distribution.

Example 3

A compressor package ordered in 2007 was analyzed using this method in the design phase. The nominal conditions specified by the customer are the following:

Suction pressure: 654.7 psia

Discharge pressure: 960.7 psia

Flow: 41 MMSCFD

Suction temperature: 110 deg. F

Ambient temperature: 95 deg. F

Compressor type: centrifugal, 4 stages, impeller dia. 7”

Driver: gas turbine, ISO shaft power 0.8 MW

In addition, a number of 250 different verification points were provided by the customer to analyze the off design operation of the compressor package.

The approach was to consider an initial design for which a model was developed both for compressor and the gas turbine driver, and then running the model for all 250 different operating conditions. The model included estimated performance curves for compressor and gas turbine. The resulted specific heat value distribution and arrangement of points on the compressor map were used for improving the design and, and a second analysis which is presented here followed.

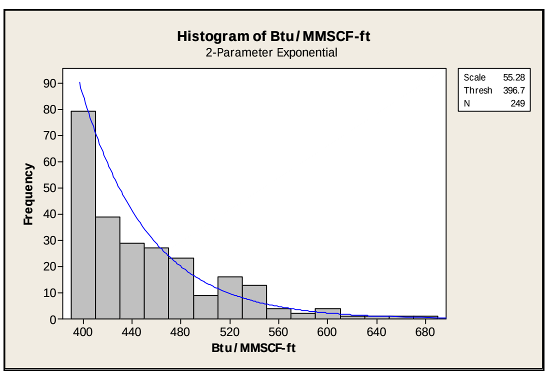
4.1 Input/output Parameters

Suction pressure and discharge pressure are input parameters and the customer provided historical records with a range between 590 and 750 psia for suction pressure, and 810 to 990 psia for the discharge. The resulted pressure ratio ranged from 1.22 to 1.56. Similarly, the input values for suction temperature and for ambient temperature were specified for each of the 250 verification points. Using these data and the model, a simulation was then performed. In this simulation the assumption was that the equipment will run at max power and the gas flow will be the max the equipment can produce. For each operating point, the model predicted the following:

- max available power from the driver,
- process gas flow through compressor
- fuel flow for the driver

4.2 Discussion on Process Efficiency

The ultimate parameter calculated at each point in this analysis is the *process specific heat value* as previously defined and plotted as histogram in Figure 14 (left). Its distribution is exponential and is strongly skewed, which is what was desired. The best value for this parameter is 400 Btu/1MMSCF-1ft.lbf/lbm and this is also the value at which the equipment is operated most of the time. Its spread ranges up to 600 Btu/MMSCF-ft.lbf/lbm. As opposed to a single nominal value or average value, a histogram approach offers more elements for understanding the process, optimizing it and in general making better decisions.



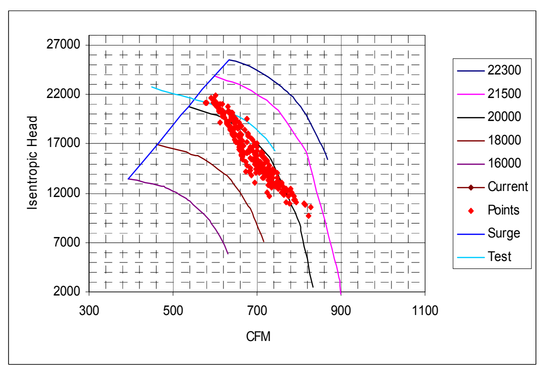


Figure 14 – Compression Process Overall Efficiency (left) and Compressor Map Operating Points (right)

The compressor map in Figure 14 (right) shows all analyzed operating points. It can be seen the wide spread of points and this is the cause for which many are out of max efficiency. However the compressor unit is able to accommodate all.

5 Conclusions

A number of 7 compressor packages were analyzed and the results are summarized in the Table 1. The analysis allows the comparison of different compressor units among the entire range of operation regardless of size, design point, type and manufacturer, reflecting ultimately the usage of fuel burned and the

Table 1

Btu/1MMS CFD-1ft.lbf/lbm	Centrif., 4 stages 12" dia., gas turbine 3.5 MW	Centrif., 2 stages 12" dia., gas turbine 3.5 MW	Centrif., 3 stages 12" dia., gas turbine 3.5 MW	Centrif., 3 stages 15" dia., gas turbine 5.5 MW	Recip, 6 cyl, 4 MW	Recip, 6 cyl., 4 MW	Centrif., 4 stages 7" dia., gas turbine 0.8 MW
Best value	**310**	**300**	**310**	**225**	**180**	**180**	**400**
Mostly seen	**330**	**360**	**365**	**285**	**205**	**210**	**400**
Max value	**400**	**600**	**640**	**550**	**250**	**255**	**620**

value created for the customer. The element of analysis is the histogram of process specific heat over a period of time. The left most value in the histogram is the minimum value for the specific process heat, which is a direct measurement of equipment best performance (ranging from 180 to 400 Btu/MMSCF-ft.lbf/lbm in Table 1). The spread in the histogram shows how much variability is in the process and accordingly how well the machine is used in the process. Notably, unit in example 1 which operates in a gas processing plant has less variability than quasi-similar units from examples 2.1, 2.2 and 2.3 operating in a gas compressor plant, due to a significantly more stable process.

In general the operating points have a relatively wide spread and by narrowing this variability, based on understanding how the machine is providing best value, the operating efficiency can be further improved. A better control with a philosophy of reduced variability and better targets for the process may help in reducing the spread and better concentrate the points in the area of max efficiency.

The examples suggest that an evaluation based on mean value would be of less use because it would hide how much the efficiency drops at off design points, and would not show opportunities for process improvement. Applying the analysis in design phase in example 3 allowed retuning of design for covering all operating points requested, and for having the histogram of specific heat value skewed with most frequent operating points coinciding also with the point of max fuel efficiency.

6 Nomenclature

P_c compressor power, kW

WF weight flow rate, kg/sec

MW gas molecular weight

H_{is} isentropic head rise, kJ/kg

Q_1 suction actual volume flow, m^3/s

T_1 absolute suction temperature, deg. K

Z_1 gas compressibility at suction conditions

p_1 absolute suction pressure, bar

p_2 absolute discharge pressure, bar

k specific heat ratio

R Gas Constant, KJ/Kg.K

η_{is} compressor isentropic efficiency

η_m mechanical efficiency

MMSCFD, SCFM volume standard flow per day and per minute

n compressor operating speed

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