PREDICTION OF CENTRIFUGAL COMPRESSOR PERFORMANCE AND APPLICATION FOR TEST, SURGE PROTECTION AND MACHINERY DIAGNOSTIC

A new software, Cmap, developed by IPC, is a tool for PC windows based design to perform centrifugal compressor performance evaluations easily and quickly

By Antonio Di Febo, Massimiliano Di Febo, Paganini Pasquale, Antonio Pedone and Gaetano Esposito

INTRODUCTION

One of the main difficulties with centrifugal compressor operative performance analysis comes from the usually expected performances are described in graphic terms. These include discharge pressure, discharge temperature, polytrophic head, efficiencies and absorbed power, related to the design inlet gas conditions, i.e. gas mix composition, inlet pressure and temperature. However, actual compressor performance is strictly related to actual field inlet gas conditions that often are different from design conditions. The comparison of field measurements to expected values become complicated because the expected performances needs to be adjusted to the actual suction field conditions.

The main purpose of this article is to present a method and tools for centrifugal compressor field performance evaluation and comparison to expected design performance. Furthermore, three case studies will be presented, where the method has been applied, for diagnostic purposes, as well as for machine protection. The execution of this evaluation is quite complex and requires calculation algorithms able to predict both machine behavior and thermodynamic gas properties under different operative conditions.

The current practice is based on simplified theories. Often also these theories are completely neglected at plant/machine level, causing a lack of analytical support capability for evaluation of machine health and performance and a consequent lack of support for maintenance decision making. Moreover these simplified theories become progressively more approximate, especially when approaching the high pressure range where errors introduced do not provide enough accurate predictions for more severe applications. In these cases more rigorous calculations and adequate computational tools are necessary to implement the most modern thermodynamic theories and machine models without limiting approximations.

In order to execute this type of analysis, IPC research lab has developed a dedicated software tool based on most recent gas theories and centrifugal machine models.

Cmap is a software tool for PC windows based design to perform centrifugal compressors performance evaluations easily and quickly. Cmap has been built in order to provide centrifugal compressor analysts or industry users, a powerful and easy tool to predict machine performance under different operating conditions.

Cmap also enables some additional tasks useful to compare centrifugal compressor expected/design performance with measured field data. Analysts and engineers that use Cmap will have all quantitative information about machine status necessary to develop diagnostic strategies and maintenance activities.

METHOD

The purpose of this section is to provide a simple explanation of the calculation process used to evaluate compressor performance. As a starting point we can consider that, for a

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centrifugal compressor, performance is strictly linked to the inlet gas conditions. This consideration is valid both to the design performances as for the actual field performance.

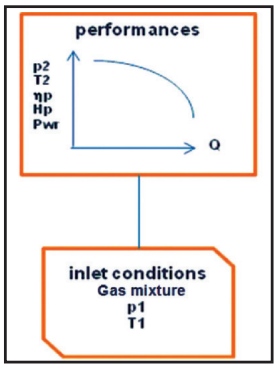


Figure 1. Compressor performance map is valid for specified conditions.

An important observation is that, in general, performance referenced to different inlet conditions cannot be compared. In order to compare performance obtained under different inlet conditions, a necessary step is to adjust and align performance to the same reference inlet conditions.

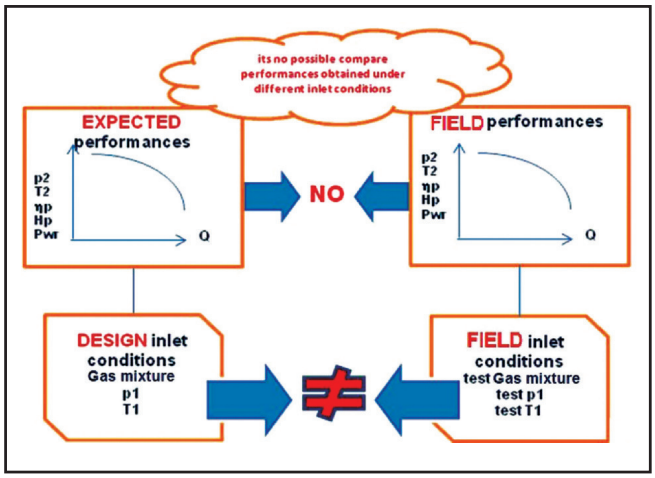


Figure 2. Compressor performances can be compared when/if done with the same inlet conditions.

When attempting to evaluate performance by comparing field measured parameters with expected parameters, the special trick is to remember that one of the two performances needs to be manipulated and adjusted to the inlet conditions of the second one. In brief, inlet conditions need to be homogeneous. (see below sketch for concept flow)

In order to proceed to performance evaluation comparison we've applied this method in the following steps: Step 1, expected performance related to design inlet conditions have been adjusted to field suction conditions. Step 2, field performance is compared to adjusted expected performance.

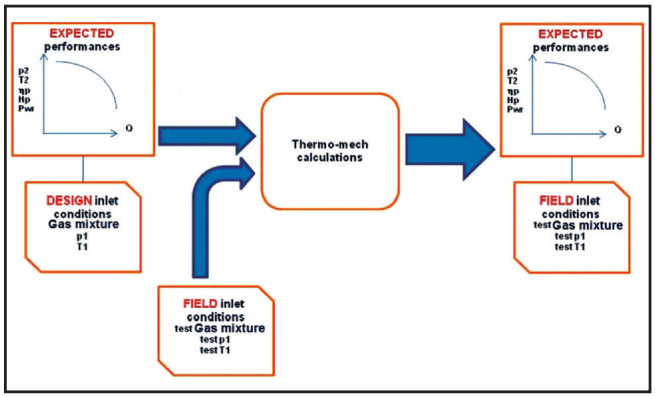


Figure 3. Schematic flow diagram for compressor performance calculations.

At this point we can proceed to the comparison of expected performance to field performance being both referenced to the same inlet conditions.

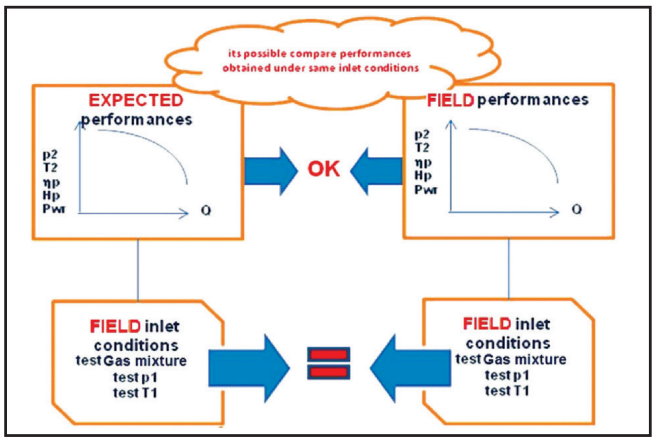


Figure 4. Design performance, adjusted to actual (field/test) performance, can be used as reference for measured performance evaluation.

SOFTWARE

This section describes how to apply the software tool. In order to use Cmap, the starting point is the availability of a centrifugal compressor performance curve, the relevant gas mix composition and thermodynamic conditions (pressure and temperature). Having these input data available, the software will perform all complex calculation in fully automated way and will produce the expected compressor performances for inlet pressures, inlet temperatures and gas mix compositions different from design reference parameters. The software is easy and intuitive, and also will provide automatic storage of data in a project archive for quick consultation and historic trends. Cmap is structured to enable analysts and engineers to build their own machinery database and to maintain records of all executed calculations. Consider the case of availability of centrifugal compressor performance parameters (pressures and temperatures).

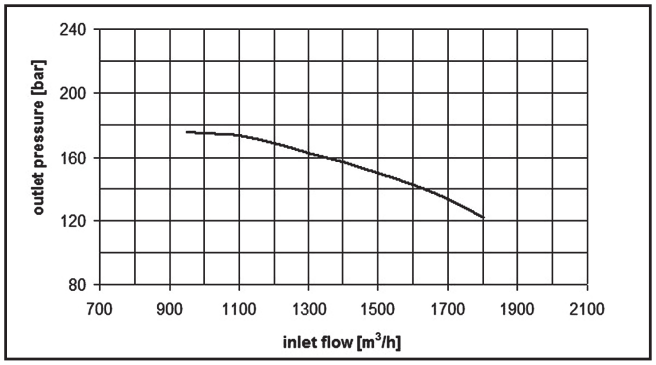


Figure 5. Typical centrifugal compressor performance curve (discharge pressure vs inlet flow) for a fixed speed.

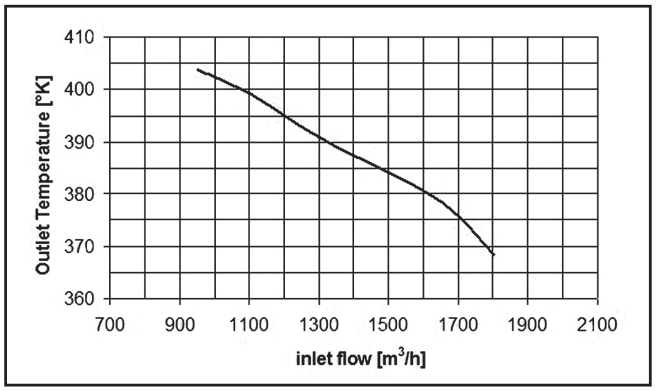


Figure 6. Typical centrifugal compressor performance curve (discharge temperature vs inlet flow) for a fixed speed.

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THESE CURVES ARE VALID IN THE FOLLOWING CONDITIONS	
GAS HANDLED	NATURAL GAS
MOLECOLAR WEIGHT	22.325 [g/mole]
INLET PRESSURE	67.5 [bar]
INLET TEMPERATURE	311.6 [K]
SPEED	11940 RPM

Figure 7. Inlet conditions for centrifugal compressor performance.

The input process offers many alternative ways to set up compressor reference data. The most common way is to enter performance curves expressed in terms of delivery pressure vs inlet flow rate, and delivery temperature vs inlet flow rate. As alternative, when available, it is possible to input directly non dimensional data.

Data input process can be digitally executed also, picking data graphically directly on pdf or jpeg files. In order to complete the data input process input of the selected gas mix composition, pressure and temperature.

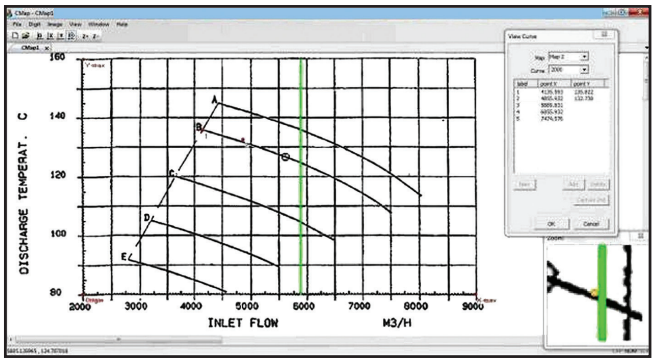


Figure 8. Compressor performance map digitization process (screenshot).

Once finished the project set up, the software has all necessary data to perform calculations. The following step is then to launch the calculation task. To execute this task, only a few seconds are needed to input the new gas mix, and launch the calculation for new inlet pressures an temperatures. When requested the run task also will execute calculations for polytrophic head and efficiency.

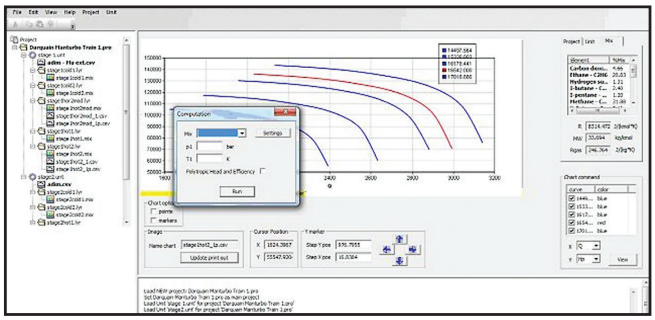


Figure 9. Software calculation output (performance map screenshot).

Once calculations are completed, the software will respond, displaying the new compressor performances map.

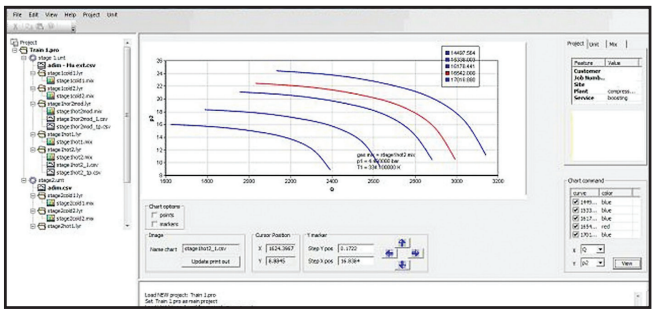


Figure 10. Software calculation output (performance map screenshot).

After this first calculation step a performance analysis can be run that allows comparing field measured data with expected data. This task is useful to the analyst in order to obtain a quantitative evaluation of the compressor running status. Performance analysis provides precious information useful for predictive diagnostic evaluation.

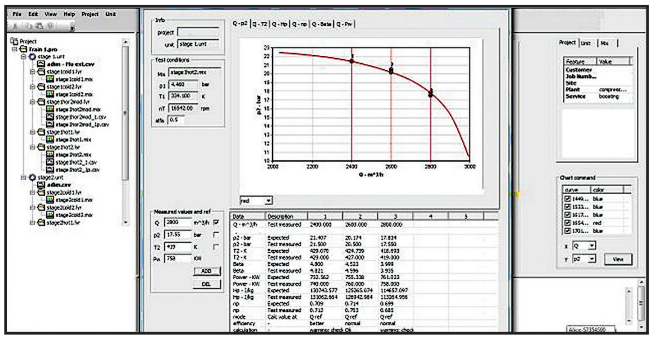


Figure 11. Software analysis tool calculation (screenshot).

CASE STUDIES

In the following paragraphs, three real cases will be presented. In each one, the compressor under study was running under off design inlet conditions, and an analysis with the presented method enables obtaining a performance evaluation and comparison to design values.

Case Study 1: Performance prediction check by comparison to manufacturer's maps.

In this case study, centrifugal compressor performance map for two different inlet conditions of pressure and temperature were available, as shown in the following table.

D1: Design 1 Condition	D2: Design 2 Condition
• Discharge pressures and temperatures map for different running speeds • Efficiency and polytropic head map for different running speeds.	• Discharge pressures and temperatures map for different running speeds • Efficiency and polytropic head map for different running speeds.
• Gas Mixture 1	• Gas Mixture 1
• Pressure 1: 40.2 bara	• Pressure 2: 53.2 bara
• Temperature 1: 50°C	• Temperature 2: 37.8°C

Table 1

Starting from the curves of the D1 condition, the expected performance curves in the D2 condition have been calculated using the software Cmap. The resulting curves, D2calc, have been compared to the D2 curves available as input. The following figures compare the D2calc (x points), obtained with the software Cmap, versus D2 (continuous line) performance curves.

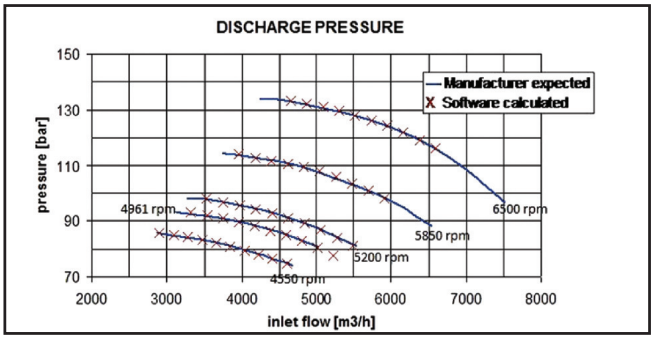


Figure 12. Centrifugal compressor design performance compared to calculated performance (discharge pressure).

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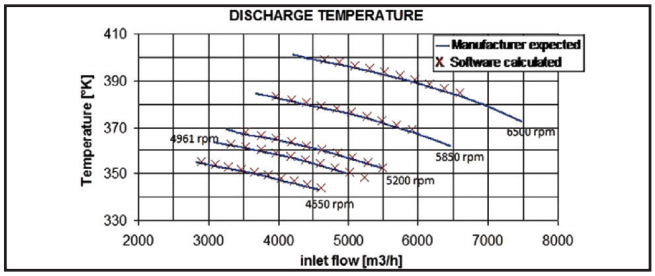


Figure 13. Centrifugal compressor design performance compared to calculated performances (discharge temperature).

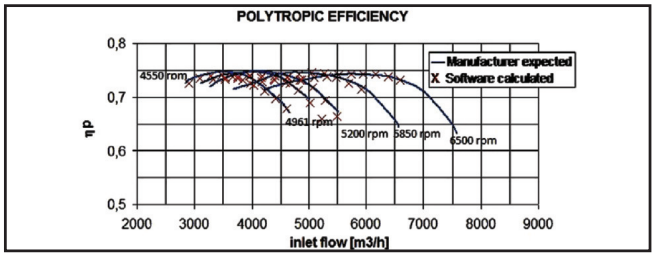


Figure 14. Centrifugal compressor design performance compared to calculated performances (polytropic efficiency).

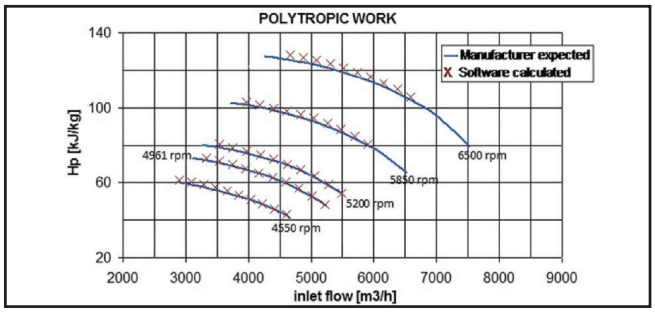


Figure 15. Centrifugal compressor design performance compared to calculated performance (polytropic work).

In the previous figures. note how the curves almost overlap; the maximum error calculated is less than 0.2 % (polytropic work at maximum speed). Measured errors have been considered tolerable for the purpose of the evaluation of compressor field performance. This case study confirms the capability of the proposed method to predict compressor performances aligned with design performance.

Case Study 2: Field Test

This second case study is aimed at evaluating compressor field performance when operating with inlet conditions different from design conditions.

Test layout

In the following figure there is a sketch of the test loop that indicates the position and types of installed instrumentation.

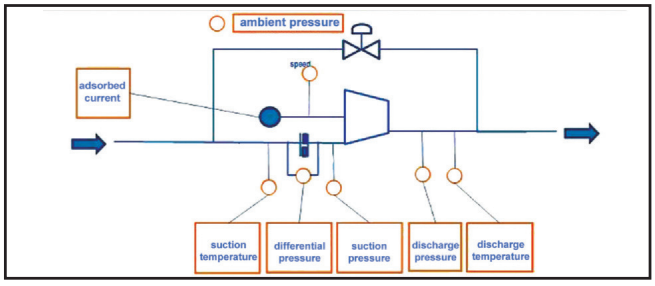


Figure 16. Test loop sketch.

With the installed field instrumentation the following parameters can be measured: ambient pressure, flow orifice differential pressure, suction pressure, suction temperature, discharge pressure, discharge temperature, speed, absorbed current. Furthermore, in order to provide a complete reference for measured performance, also, necessary is to specify the operating gas mix composition.

Design performances

In order to provide an evaluation of field performance, an important necessity is to have reference performance obtained with design inlet conditions. In these case studies, the following design performance graphs were available as reference data.

Discharge pressure vs suction flow

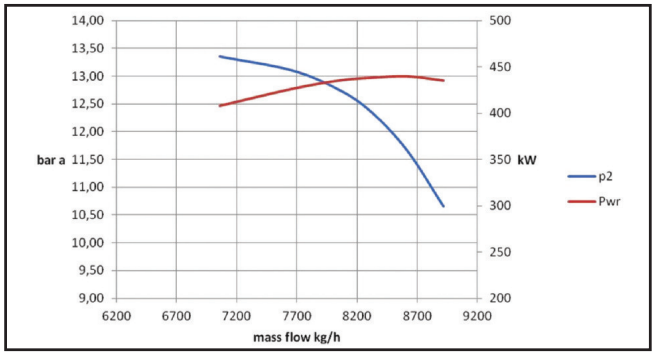


Figure 17. Case Study 2: Compressor design performance curve.

Polytropic head and efficiency curves

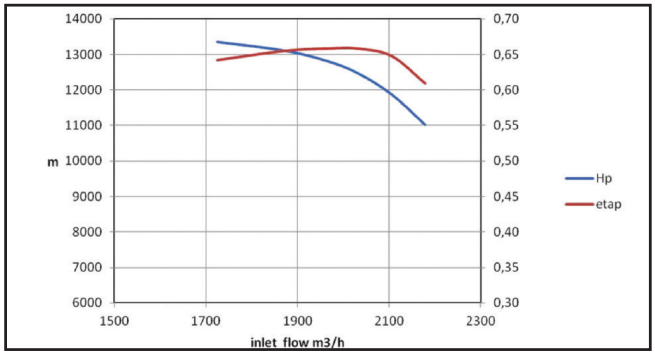


Figure 18. Case Study 2: Compressor design performance curve.

Design reference inlet conditions:
Design gas mix
Inlet pressure: 3.3 bara
Inlet temperature: 21 °C
Speed: 22401 rpm.

Design Gas analysis

Gas mix composition has been provided through a gas analysis:

Test performances

From installed measuring devices, the following test data have been obtained:

Table 2

	Design gas mix
Symbol	% mol
H_2O	0.83
CO_2	1.33
N_2	0.93
CH_4	51.58
C_2H_6	15.33
C_3H_8	15.62
i-C_4H_{10}	3.56
n-C_4H_{10}	6.82
i-C_5H_{12}	1.88
n-C_5H_{12}	1.30
C_9H_{20}	0.57
C_7H_16	0.18
C_8H_{18}	0.06
$C_{10}H_{22}$	0.01
Total	100
M (g/mole)	30.09

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Test Point		A
Mix		Test Gas Mix
p amb	mbar a	1003,6
n	RPM	22401
p1	bar a	3,76
T1	°C	25,22
p2	bar a	10,52
T2	°C	112
I	A	52.0
Differential Pressure	bar	0.168

Table 3

Test gas analysis

During the test, a sample of the operative gas mix has been analyzed with the following results:

	Test gas mix
Symbol	% mol
Nitrogen	0.710
Carbon dioxide	0.499
Methane	63.696
Ethane	17.277
Propane	11.367
i-butane	1.185
n-butane	3.225
i-pentane	0.577
n-pentane	0.692
Hexanes (C_6)	0.390
Heptane (C_7)	0.212
Benzene	0.017
Octanes (C_8)	0.090
Toluene	0.010
Nonanes (C_9)	0.031
Decanes (C_{10})	0.011
Undecanes (C_{11})	0.006
Dodecanes (C_{12})	0.005
Total	100
M (g/mole)	25.07

Table 4

Test point A analysis

Using the data measured during the test point A, the following performances have been calculated:

			Test point A
Calculated Values	Inlet density	kg/m³	3,86
	Q (*)	m³/h	2141
	Hp	m	11536
	etap	%	63.0
	Pw (gasdyn)	kW	409

Table 5. *(*) see note (1) on calculated suction flow.*

Performances test evaluation and comparison to expected

At this point superimposing the measured and calculated values to the design performances adjusted to inlet test conditions is possible. The estimated operative point is then reported in the following plots.

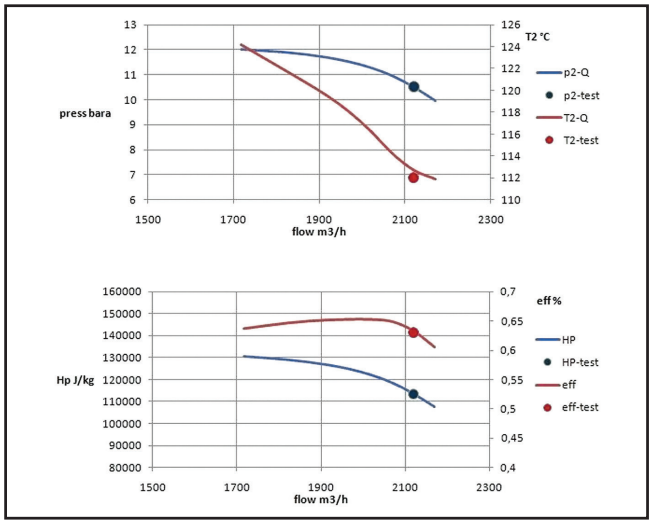


Figure 19. Field measured performance superimposed on expected compressor performance curves.

Note on suction flow

The flow calculation has been developed by IPC software for orifice calculation according to ISO 5167-2.

For the test point A, the flow rate corresponding to the ?p reading (0.168 bar) is:

$$Q = Q_{max} x \sqrt{\frac{\Delta p}{\Delta p\,max}} = 2611.89 x \sqrt{\frac{0.168}{0.250}} = 2141 \ m^3/h$$

Notes on Temperature readings

Temperature readings seem to be a little bit lower than the effective discharge temperature. This fact could be caused by the discharge temperature sensor design. It introduces an error that could cause temperature underestimation of about 3-4 degrees. This is a very important observation that needs to be investigated correctly in order to develop more reliable industrial temperature sensors for performance test purposes. In this case study a corrected discharge temperature of about 112°C has been considered and consequent compressor calculations give a polytropic efficiency of 63% and a polytropic head of 11,627 m aligned with expected performances.

Notes on Absorbed power

With reference to the table 3, calculating the electrical absorbed power using the line current measured value:

$$P_{Wel} = \sqrt{3} \text{ x } 5000 \text{ x } 52 \text{ x } \cos\varphi = 430 \text{ kW}$$

Considering the measured thermodynamic parameters, calculating the gas dynamic absorbed power:

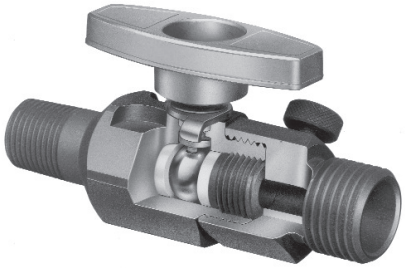
$$P_{gasdyn} = 409 \text{ kW}$$

Considering the electrical motor efficiency at 98%, and assuming the mechanical and volumetric compressor efficiency at 99%:

$$P_{Wel} = \frac{P_{gasdyn}}{\left[0.99^2 \text{ x } 0.98\right]} = 426 \text{ kW}$$

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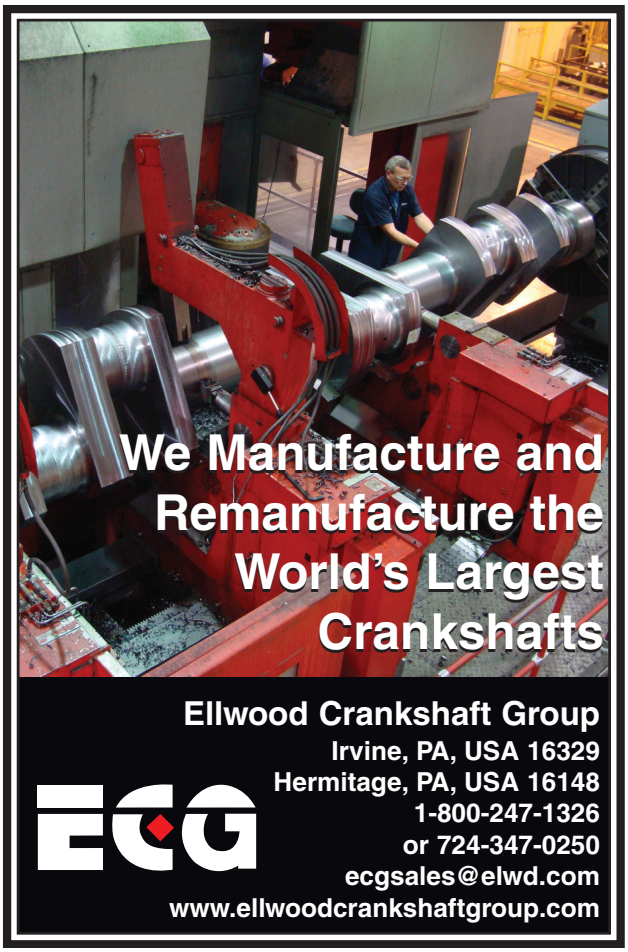
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The electrical power value calculated through the line current is then very near to the electrical absorbed power calculated through compressor performance.

Final comments

The analysis preformed indicates that the compressor is working with performance aligned with the expected performance. The compressor efficiency, in the test point, A is about 63% and is congruent with the design efficiency adjusted to test conditions. The compressor delivery pressure and polytropic head also are aligned with expected design values.

This case study shows that the centrifugal compressor performance analysis tool can support the evaluation of field compressor operating conditions also providing a quantitative indication of the machine's health status, through the evaluation of field performanc and comparison to expected performance.

Case Study 3: Anti-surge protection and diagnostics

An important application of these proposed methods is with surge protection systems and for diagnostic purposes.

Surge consists of a rapid oscillation of the mass flow, intensified by the compressor, accompanied by a characteristic noise and high vibrations. A surge is an abnormal operative condition, that can have a destructive nature, and could cause, when the machine remains in this condition without adequate protection, damage or failure of the compressor. Consequently, the operator, of necessity, must be able to diagnose quickly eventual incoming surge conditions in order to take the appropriate actions and without incurring conservative reductions of the operating range.

Actually, the state-of-the-art for anti-surge systems provides protection action consisting of opening, either partially or totally, a special control valve located on a line that recycles the cooled gas from the discharge to the compressor suction. In this way, the control system reduces the overall resistance and increases the flow, moving the compressor operating point to the right of the characteristic curve corresponding to the operating speed. Figure 20 shows an example of characteristic curves of a compressor, drawn on a plan having as axes the volume flow rate and the discharge pressure. Also shown in Figure 20 is the surge area. (SSL). Figure 21 shows a typical plant layout with a suction recirculation line that includes a recycle control valve (anti-surge valve).

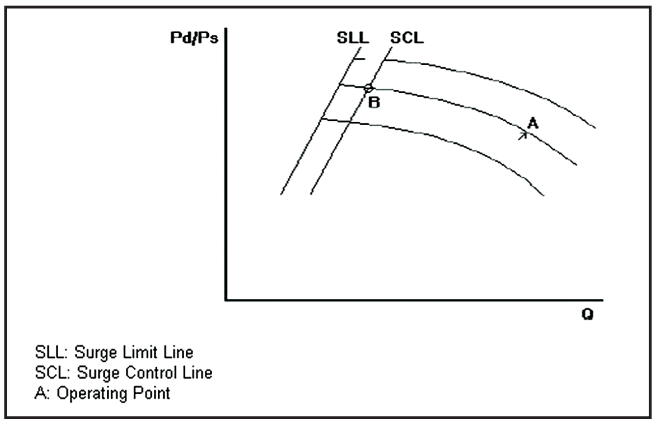
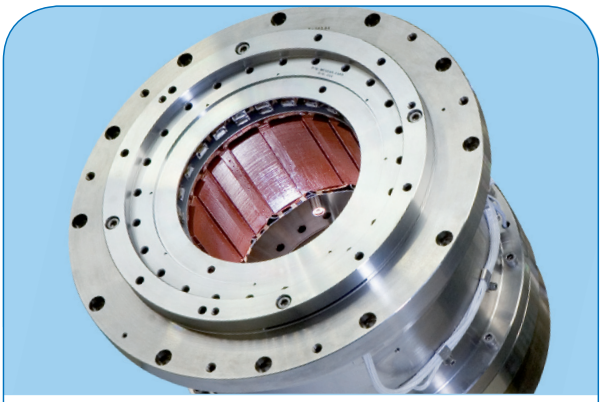


Figure 20. Example of characteristic curves of a compressor, drawn on a plan having as axes the volume flow rate and the discharge pressure also showing the surge area (SSL).

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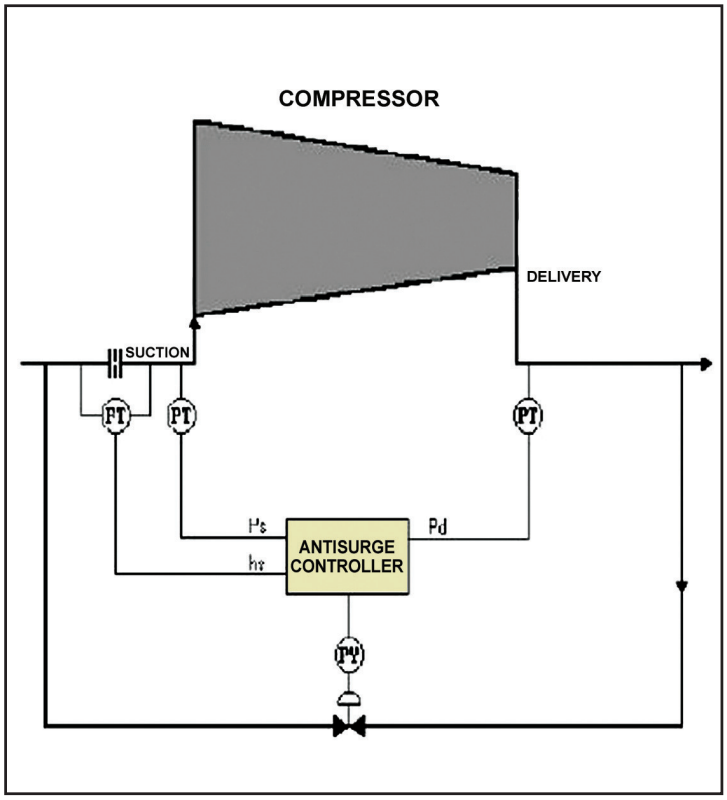


Figure 21. Typical plant layout with a suction recirculation line that includes a re-cycle control valve (anti-surge valve).

Actual centrifugal compressor protection systems are based on simplified algorithms that use basically the inlet flow, suction and discharge pressures and temperatures. These actual algorithms, are based on a basic surge locus equation obtained under simplified hypothesis such as the ideal gas law, validity of hydraulic affinity laws, reference gas mix and inlet conditions.

These algorithms are, consequently, completly dependent on the suction thermodynamic parameters (pressure, temperature and gas mix composition). The protection action is then rigidly linked to design suction parameters and remains valid only for the design specified gas mix. Consequently, these systems provide a limited surge protection valid within a small operative range around the design conditions. Compressor protection is not correctly active when inlet conditions move to off design values, especially with sensible changes of gas mix composition. Consequently, the protection of these algorithms is partial and limited. When the compressor is operated outside the design conditions, the protection action may become inefficient and potentially can damage the compressor.

Many cases of catastophic compressor failures can be pointed out that are cause by antisurge systems lack of protection. This is because they are technically dependent on the gas mixture and on the thermodynamic suction conditions (temperature and pressure). The characteristic curves in Figure 20 are, infact, valid only for certain inlet gas conditions, and vary in complex ways as functions of pressure, temperature and gas mixture composition. The proposed method for prediction of centrifugal performaces can be profitably used to predict centrifugal compressor overall performances in all inlet conditions and then to detect surge points in all operating conditions. IPC has embedded thermodynamic and compressor symulation algorithms in a new advanced anti-surge sytem, aimed to obtain an integrated control system able to provide overall protection.

These innovative antisurge systems will enable overcoming

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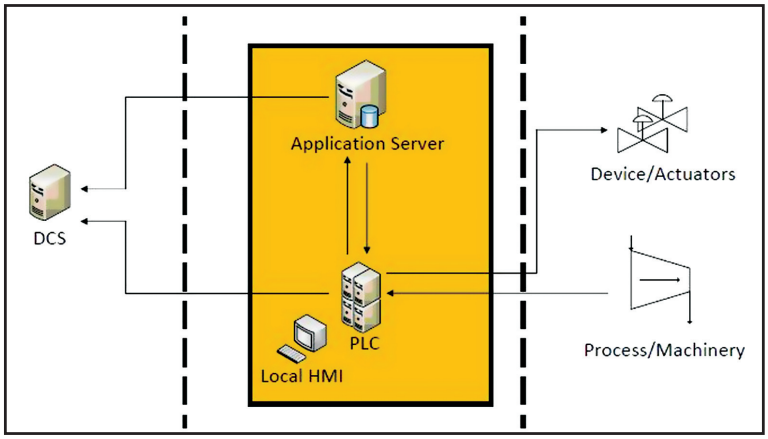


Figure 22. Innovative anti-surge protection system architecture.

the actual protection system limitations and achieve optimal surge protection in every operating condition. Protection also will be effective in cases of gas molecular weight changes. In fact, in these cases, surge points will be detected without approximations or simplifying assumptions, but taking advantage of most recent theories for real gases and thermodynamic modeling of compressors presently available.

Application of these proposed methods provides a new kind of diagnostic capabilities as well. The software is able to automatically store data and to provide a real time indication of the compressor efficiency and comparison with expected data. This is a powerfull feature that provides continuous indication about how much the machine behaviour is aligned with design expectations. These evaluations then can be used for historic trends and to build records of the machine status during its operating period. Collected data will be useful to support predictive maintenance decisions and operating plans.

CONCLUSIONS

Methods proposed and described in this article can be used to: 1. Predict the performances of a centrifugal compressor under varying thermodynamic conditions of the inlet gas. The prediction of compressor performances is accurate even at high pressures, where the ideal gas theory commonly used introduces considerable errors. 2. Analyze the performance of the compressors during operation and compare them with expected data provided by the manufacturer. 3. Implement advanced protection from surge, overcoming limits of the current technology. 4. Provide useful indications of the health of the compressor (diagnostics) based on the capability to analyze the performance and efficiency of the machine in a simple and immediate way. 5. Support decisions and planning of predictive maintenance and activities. 6. The methods and software tools presented have been validated by comparison with much compressor technical data, and are in agreement with all the available scientific papers on thermodynamic gas theories developed in the hydrocarbon research field.

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