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# Performance Optimization of A Charge Gas Compressor – A Story of Success



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## **Abstract**

As most of the refineries and petrochemical plants were built 30 or more years ago and an increasing number of plant owners are searching for a new approach to maximize production throughput and reduce operational costs, the low reliability and poor performance of their equipment is a real problem that production managers are facing daily.

This article describes how compressor, steam turbine and heat exchanger equipment maintenance can be improved using effective diagnosis of the root causes that are generating performance degradation.

A case study is presented for the Charge Gas turbo-compressor train of ENI Versalis Ethylene plant located in Priolo, Italy, where recently implemented performance monitoring software “Total Train Performance Advisor” has been successfully used to diagnose compressor fouling due to polymerization.

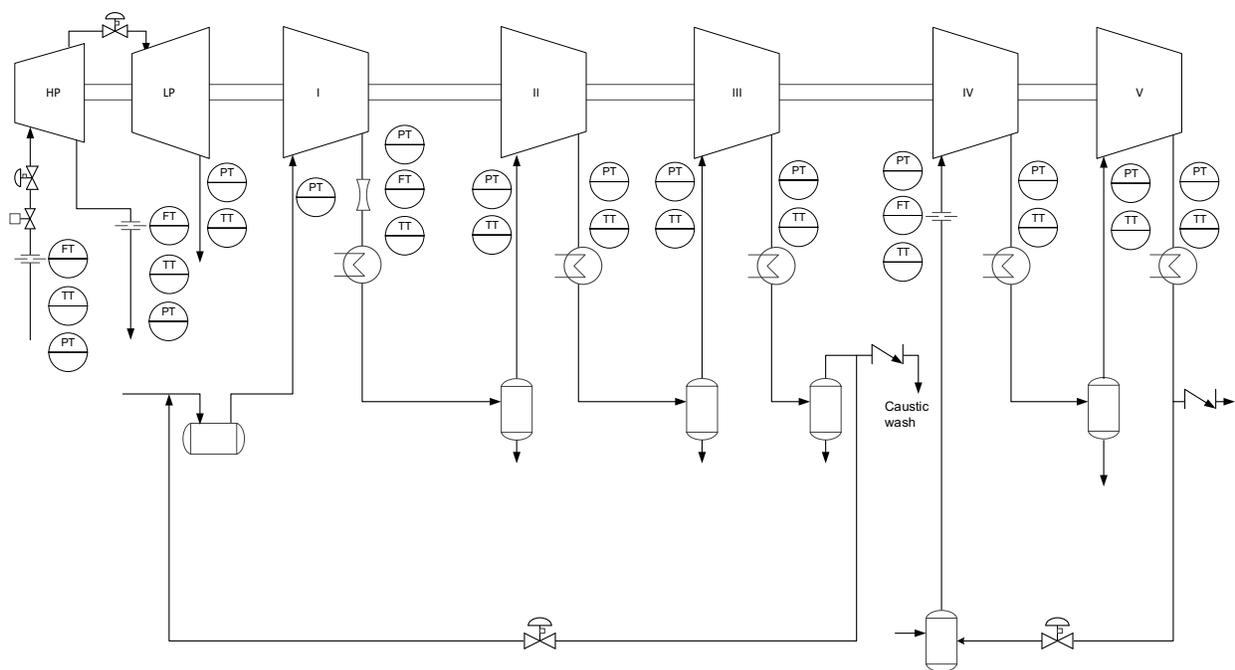
The paper provides an overview of the performance monitoring calculation and demonstrates how such systems can be integrated with the plant DCS and Historian, so that operating personnel can easily monitor Key Performance Indicators, such as steam consumption efficiency, and plan and execute corrective actions.

## Introduction

This paper addresses the issue of the timely or “real time“ detection of the performance degradation of the Charge Gas Compressors (CGCs) in Olefins plants due to fouling, and a means of restoring the performance to nominal levels. Specifically, the paper presents a case study of a CGC fouling at the ENI Versalis Ethylene production facility, located in Priolo, Sicily. A team of turbomachinery experts analyzed the system and applied a Performance Monitoring System (PMS) at Priolo in November of 2017. The system monitors both process and turbine side operation, and comprises a Compressor Performance Advisor (CPA), Turbine Performance Advisor (TPA), and Intercooler Monitor. The supplied PMS identified severe efficiency degradation of the 2<sup>nd</sup> section of the compressor. PMS calculated the extra steam consumption resulting from decreased efficiency and its associated cost, providing justification for maintenance operations. ENI injected a Nalco antifoulant solution program into the 2<sup>nd</sup> section starting in March 2018, which quickly restored efficiency to nearly nominal levels in roughly a period of a few weeks, allowing for record production and lower operating costs.

## CGC System Description

The CGC at Priolo comprises 5 stages and is driven by an extraction steam turbine. Simplified PFD is shown in Fig. 1.



**Fig. 1:** CGC at Priolo – 5 Stages

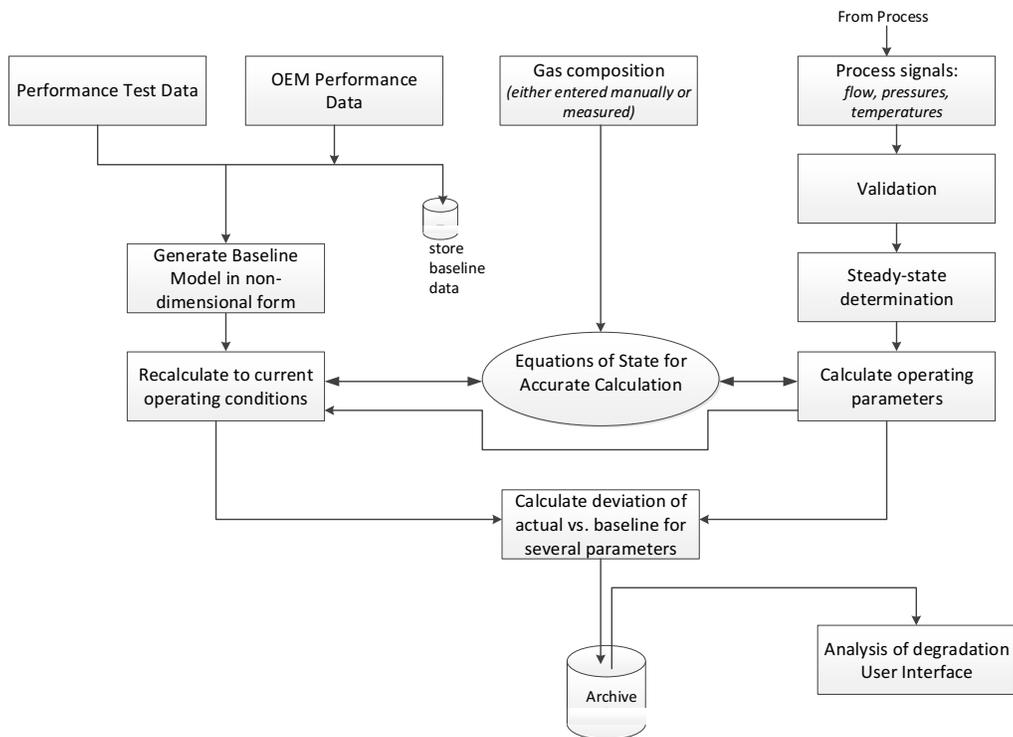
It is well known that CGCs are prone to fouling. The fouling mechanisms include radical polymerization, Diels-Alder condensation, and thermal degradation to coke. These fouling processes show a strong dependence on temperature and can result in depositing a coke-like substance in the compressor and associated equipment, and at times, at adjacent piping and intercoolers.

Fouling due to polymerization and coke formation can severely decrease efficiency of the compression process as well as intercooler heat transfer coefficients. Other causes of CGC unit performance degradation can be attributed to the decrease in the steam turbine efficiency, which may be affected by boiler feedwater quality and steam contamination.

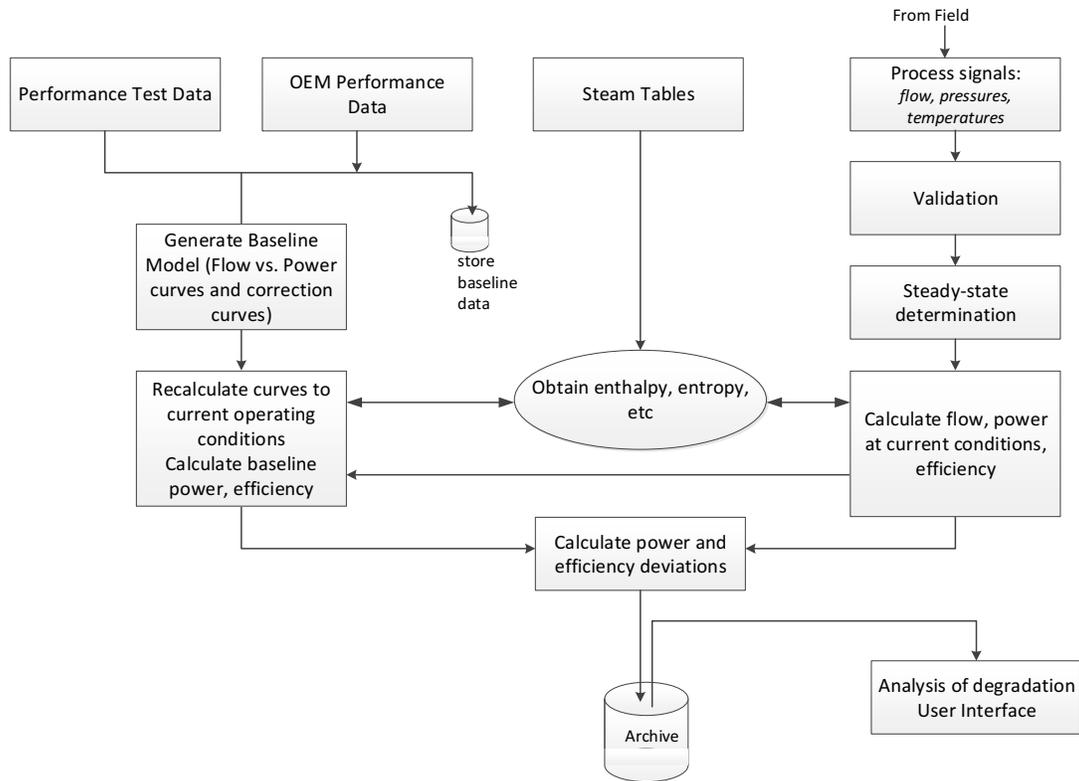
In either case, PMS can identify fouling and evaluate its severity by calculating the corresponding increase in energy consumption and associated costs, providing justification for timely maintenance operations, such as antifoulant injection. Actual data both pre- and post the Nalco antifoulant program, showing the process side efficiency recovery is shown in Section 2.4.

## Performance Monitoring System

PMS works in real-time and compares the actual compressor, intercooler, and turbine performance parameters to their baseline values. The baseline model of the equipment can be obtained from the OEM data, or from field experimental characteristics. The baseline efficiency of the compressor changes as a function of the operating parameters and gas composition. Turbine efficiency calculations depend on steam conditions, rotating speed, and flow. Therefore, accurate estimation of the baseline performance covering a wide range of conditions plays an important role in identifying degradation.



**Fig. 2:** Compressor Performance Advisor Simplified Schematic



**Fig. 3:** Turbine Performance Advisor

CPA calculates process gas properties using the Benedict-Webb-Rubin-Starling (BWRS) equations of state. For this project, the consulting team manually entered gas composition is manually entered into the CPA for each section. Communication link can be used for automatically updating gas composition, if such information is available. TPA utilizes built-in steam tables for enthalpy and entropy as functions of temperature and pressure for accurate estimation of turbine isentropic efficiency. These calculation methods help in avoiding nuisance detection of degradation. In addition, calculations are only carried out only during steady-state conditions and with valid data.

Main calculation results are communicated to the Plant Historian. The system also archives input and output parameters and provides graphical means for viewing their history.

### Operator and Engineering Interface

At Priolo, PMS has been installed as part of the software delivered with the Turbomachinery Control System (TCS). Many of the calculations, performed during the engineering phase of configuring the TCS, such as Surge Line Calculations and Extraction Map calculations, align themselves quite well with PMS calculations, reducing the overall engineering effort in configuring control and monitoring functions. In addition, TCS provides input validation for those measurements connected to the system.

Operators often require the convenience of viewing Compressor and Operating points on their respective performance maps. Such functionality has long been available at Priolo via the TCS, PMS extended this functionality with more accurate calculations and added the baseline models for identifying degradation.

In addition to estimating efficiencies, CPA also estimates baseline values of discharge pressure and temperature, polytropic head, and power consumption for the existing inlet conditions. CPA provides an alarm if monitored parameter exceed deviation threshold. Knowing baseline parameters can assist in identifying drifting measurements.

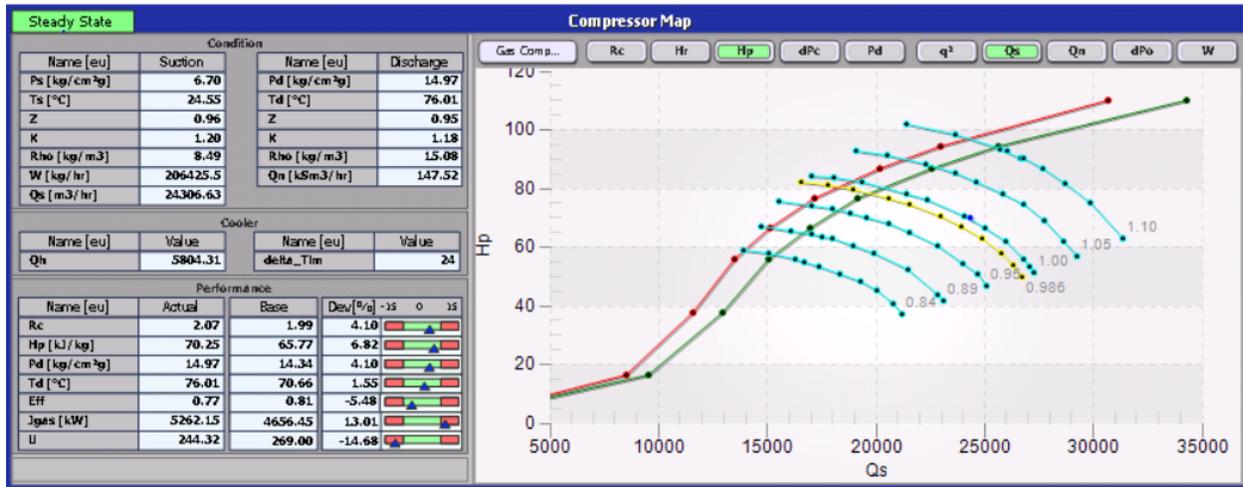


Fig. 4: CPA Typical Operator and Engineering Interface Screen

The compressor map graphic above shows the actual surge and control lines, read from the antisurge controller, superimposed on the OEM characteristics (blue curves). The green performance characteristic is the estimated baseline curve, corresponding to the current suction conditions and rotating speed. The blue “dot” is the current operating point. Under ideal circumstances, it would be located on the green line.

The extraction turbine map shows the actual operating point on the baseline extraction map. The baseline map is dynamic, adjusted based on the steam inlet and outlet conditions.

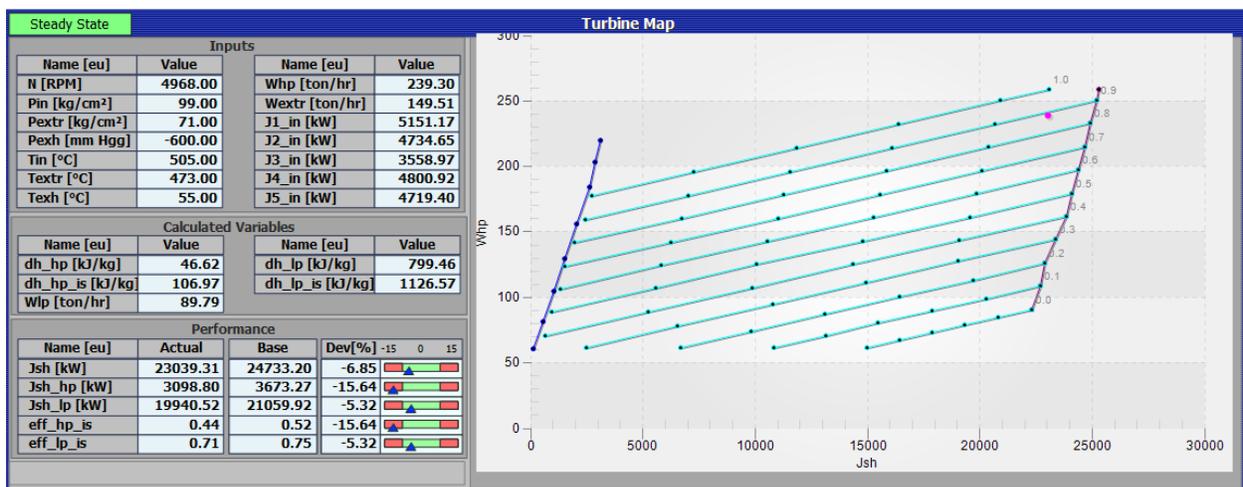


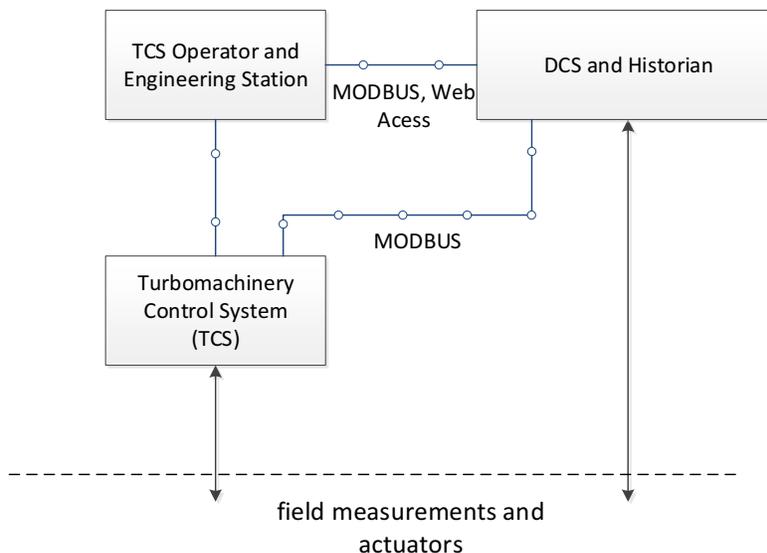
Fig. 5: Extraction Turbine Map

## Communications Between PMS and Other Plant Systems

The communication schematic between Turbomachinery Control System (TCS), TCS Operator and Engineering Workstation, and DCS is shown in Fig. 4. In this case, TCS provides antisurge protection for 1<sup>st</sup> and 5<sup>th</sup> Sections. DCS supplies required measurements for the other sections for PMS calculations via a MODBUS link to the Workstation,

while the Workstation sends back the calculated PMS parameters. Compressor and Turbine map graphics can be accessed by the DCS using web-based communications.

In addition, PMS provides an OPC DA server for OPC communications. A compatible OPC client can access PMS calculations.



**Fig. 6:** Communication Schematic

### Challenges in PMS Installation

Finding and using reliable measurements for the PMS calculations proved to be the biggest challenge in installing and commissioning PMS. PMS is designed to perform calculations for each section. However, only the 1<sup>st</sup> and 4<sup>th</sup> sections have flow measuring devices. PMS approximates the flows through each section by subtracting or adding the liquid flow from the interstage knock-out drums.

### Efficiency Recovery Using the Nalco Program

Nalco Champion started to get involved in the Versalis Priolo compressor fouling beginning 2017. Two observations were made indicating fouling of the compressor: decline of efficiency stage 2 and high vibrations stage 3. To cope with these problems and avoid operational cost going up, prevent unplanned shutdown, Nalco Champion started their multi component anti-foulant program EC3144A on both stages resulting in a quick recovery of the efficiency on stage 2 and reduction of the vibrations on stage 3 including some efficiency increase.

Stage 2 efficiency reduction before treatment/efficiency recovery after treatment:

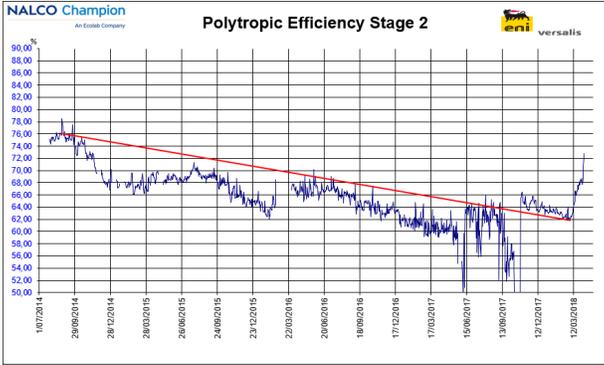


Fig. 7: Over 14 % Efficiency decline Stage 2

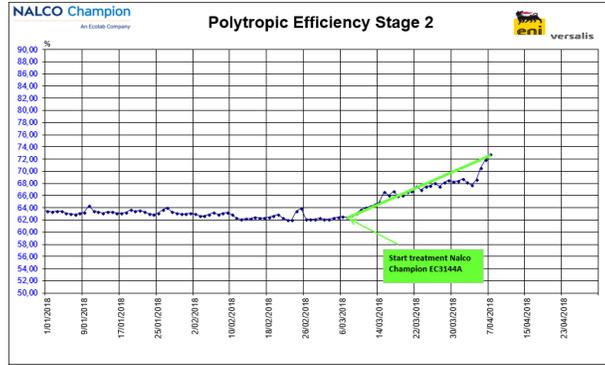


Fig. 8: Over 10 % recovery Efficiency Stage 2

Stage 3 Efficiency increase after treatment/vibration decrease after treatment

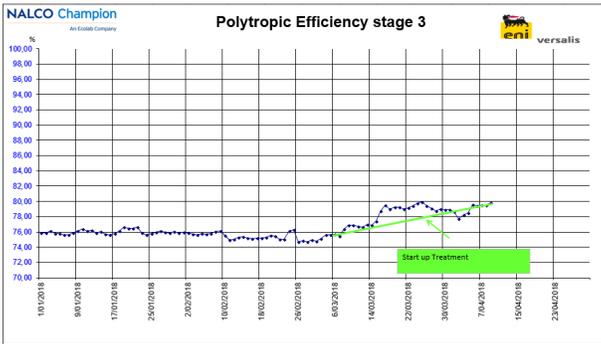


Fig. 9: Over 4 % recovery Efficiency stage 3

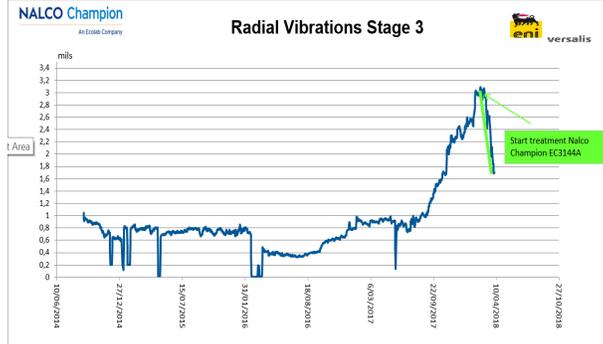


Fig. 10: Vibration decrease stage 3

## Theoretical Background

### Compressor Performance Calculations

In constructing performance model, CPA follows ASME PTC-10 methodology. Difference between static and stagnation pressure are neglected. CPA obtains gas properties from BWRS equations of state. For a single section adiabatic compression the compressor characteristics are approximated as:

$$\psi = f_1(\phi, Ma_N)$$

$$\eta_p = f_2(\phi, Ma_N)$$

$\phi$  (phi) flow coefficient

$\psi$  (psi) head coefficient

$\eta_p$	polytropic efficiency
$Ma_N$	rotational Mach number
$Q_s$	volumetric flow in suction
$H_p$	polytropic head
$N$	rotating speed (rpm)
$Q_{s,o}$	nominal volumetric flow
$N_o$	nominal rotating speed
$H_{p,o}$	nominal polytropic head
$Ma_{N,o}$	rotational Mach number at nominal conditions
$a$	speed of sound at existing conditions
$a_o$	speed of sound at nominal conditions
$f_1, f_2$	piecewise, continuous functions for approximating compressor performance. CPA uses table look-up approach with interpolation for estimating head coefficient and efficiency.

For the purposes of implementation in the CPA, the definitions are:

$$\phi = Const_{phi} \frac{Q_s}{N}$$

$$\psi = Const_{psi} \frac{H_p}{N^2}$$

For normalizing the values relative to design conditions, the constants are calculated as follows:

$$Const_{phi} = \frac{N_o}{Q_{s,o}}$$

$$Const_{psi} = \frac{N_o^2}{H_{p,o}}$$

Rotational Mach number:

$$Ma_N = \frac{\left(\frac{N}{a}\right)}{Ma_{N,o}}$$

$$Ma_{N,o} = \frac{N_o}{a_o}$$

At operating conditions, the polytropic head is calculated as per ASME PTC-10 test code:

$$H_p = P_s v_s \left( \frac{R_c^{\sigma_v} - 1}{\sigma_v} \right)$$

where:

- Ps suction pressure
- Rc compression ratio, Pd/Ps,
- vs specific volume
- v polytropic volume exponent

$$\sigma_v = \frac{n - 1}{n}$$

- n polytropic volume exponent

Polytropic efficiency is calculated as:

$$\eta_p = \frac{H_p}{h_d - h_s}$$

where:

- hs enthalpy in suction
- hd enthalpy in discharge

For estimating discharge conditions from the model, CPA uses the following approximations:

$$\sigma_t = \frac{Z_{avg} R_{gas}}{c_{p,avg}} \left( \frac{1}{\eta_p} + X_{avg} \right)$$

t	polytropic temperature exponent, defined by the equation
	$\left( \frac{T_d}{T_s} \right) = \left( \frac{P_d}{P_s} \right)^{\sigma_t}$
Zavg	compressibility between suction and discharge, averaged over the compression path
Rgas	gas constant
cp,avg	average specific heat at constant pressure
Xavg	compressibility function derived by Schulz, averaged over the compression path

When obtaining baseline performance at given inlet conditions, the discharge conditions are not initially known and are estimated by iterative methods using the above relationships.

## Flow Calculations

Accurate flow calculations are very important in estimating deviation of the operating performance from baseline. PMS calculates flow based on the ISO-5167 standard, using differential pressure, pressure, temperature measurements, flow measuring device geometry, and gas properties from BWRS equations.

## Turbine Calculations

Turbine Performance Advisor (TPA) calculates steam flow, shaft power, and isentropic efficiency of the steam turbine and estimates deviation from the baseline performance. TPA provides graphical interface with real-time plot of baseline steam flow (W) vs. Shaft Power ( $J_{sh}$ ) curve, compensated for the existing steam conditions; and the current operating point. For extraction machines TPA displays multiple curves, each corresponding to a given extraction flow. When the conditions in turbine exhaust are below saturation line, TPA estimates efficiency based on the consumed power, which can be calculated from the parameters of the driven load, such as a compressor or a generator. TPA provides means of correcting baseline data based on rotating speed, steam flow, and steam pressure and temperature.

Along the extraction curves, baseline power is calculated as:

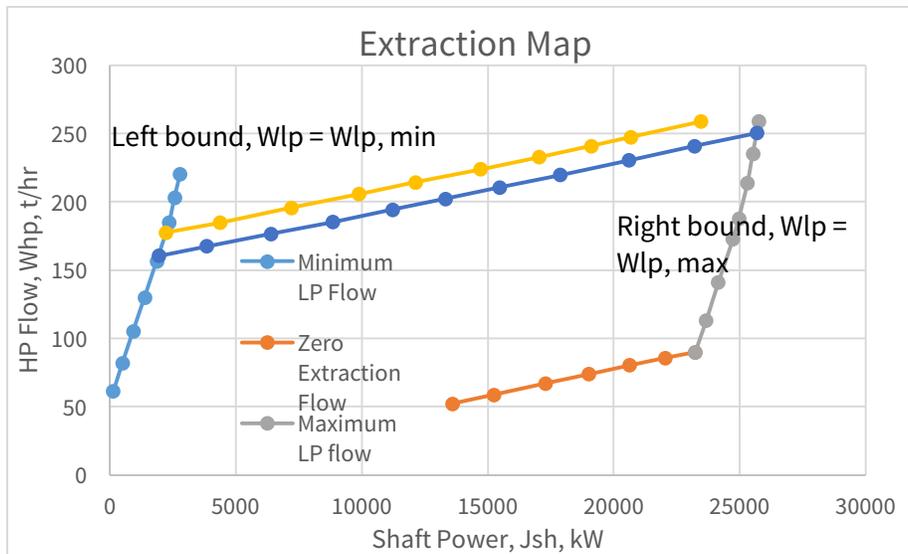
$$J_{sh,HP} = W_{HP} \times dh_{HP,nom} \times corr_{his,HP} \times corr_{n,HP} \times corr_{WHP}$$

$$J_{sh,LP} = W_{LP} \times dh_{LP,nom} \times corr_{his,LP} \times corr_{n,LP} \times corr_{WLP}$$

$$W_{LP} = W_{HP} - W_{extr}$$

$$J_{sh} = J_{sh,HP} + J_{sh,LP}$$

Whp	steam flow through the high pressure stage
Wlp	steam flow through the low pressure stage
Jsh	total power delivered to the shaft
Jsh, hp	hp power
Jsh, lp	lp power
corrWHP	provides correction for changes in isentropic efficiency due to Whp deviation from nominal
corrWLP	provides correction for changes in isentropic efficiency due to Wlp deviation from nominal
corrhis, hp	correction factor for isentropic enthalpy changes between nominal and actual conditions, HP stage
corrhis, lp	correction factor for isentropic enthalpy changes between nominal and actual conditions, LP stage
corrn,HP	correction for changes in isentropic efficiency due to rotating speed, HP stage
corrn,lp	correction for changes in isentropic efficiency due to rotating speed, LP stage



**Fig. 7:** Extraction Map

## Intercooler Calculations

The relationship between the heat transfer and the hot and cold temperature differences is:

$$Q = U_{nom} f_{corr} A \Delta T_{LM}$$

Q	heat transfer (W)
Unom	overall heat transfer coefficient of the exchanger for nominal conditions
fcorr	correction factor for different conditions, such as different flow rates and gas inlet enthalpy. The correction factor as a function of gas flow rate, pressure, and temperature can be obtained experimentally by varying the compressor performance.
A	heat exchanger surface area
TLM	logarithmic mean temperature difference

$$\Delta T_{LM} = \frac{\Delta T_H - \Delta T_C}{\ln \left( \frac{\Delta T_H}{\Delta T_C} \right)}$$

TH	temperature difference between hot and cold fluid at the gas inlet side of the heat exchanger
Tc	temperature difference between hot and cold fluid at the gas outlet side of the heat exchanges

The heat transfer can be calculated from cracked gas conditions upstream and downstream of the cooler:

$$Q = W_{gas} (h_{gas,in} - h_{gas,out})$$

Wgas	flow rate of the cracked gas, estimated to be same as through the corresponding section
hgas,in	inlet enthalpy of the cracked gas, calculated using BWR equations of state as function of P, T, and gas composition
hgas,out	outlet enthalpy of the cracked gas

The actual overall heat transfer coefficient can be calculated as:

$$U_{act} = \frac{W_{gas}(h_{gas,in} - h_{gas,out})}{f_{corr}A\Delta T_{LM}}$$

The module will calculate the deviation of the actual heat transfer coefficient from nominal:

$$d_U = \frac{U_{act}}{U_{nom}}$$

## Calculation of The Excess Energy Costs

At Priolo, the additional costs of operating the machine with lower efficiencies are a function of the quantity of the external medium pressure steam required to run the LP turbine at the power levels demanded by the process. The HP turbine uses the high pressure steam produced by the pyrolysis process. However, the quantity of this steam is not sufficient to satisfy plant power requirements. Priolo purchases additional medium pressure steam from an external source. The cost of this “extra” steam is a major contributor to the additional energy costs.

Additional flow through the LP section required for the additional power, due to decreased efficiency, neglecting mechanical losses and changes in turbine efficiency:

$$\Delta W_{LP} = \frac{\sum_1^5 J_{i,actual} - \sum_1^5 J_{i,baseline}}{\Delta h_{LP}}$$

$\Delta h_{LP}$

enthalpy drop across the LP section at existing conditions

$\sum_1^5 J_{i,actual} - \sum_1^5 J_{i,baseline}$

difference between the sum of the actual power consumed by each stage, and the sum of baseline power consumption

The initial situation at Priolo was as follows:

	Section 1		Section 2		Section 3		Section 4		Section 5	
	Act	Base								
Ps (kg/cm2)	0.56		1.97		4.03		6.69		14.48	
Pd (kg/cm2)	2.35	2.03	4.44	5.87	8.4	8.91	14.96	14.25	34.6	33.49
Ts (deg C)	32.97		29.13		29.37		24.67		28.94	
Td (deg C)	83.07	75.52	77.95	78.35	71.11	70.95	76.13	70.64	86.93	83.59
Rc	2.13	1.93	1.92	2.3	1.86	1.96	2.07	1.98	2.3	2.23
Hp (kJ/kg)	74.43	63.98	54.62	74.67	56.59	61.3	70.43	65.47	76.79	73.5
W (kg/hr)	238,346		225,141		205,588		206,882		194,651	
pol. Eff.	0.79	0.81	0.60	0.85	0.75	0.82	0.77	0.81	0.78	0.79
Jgas (kW)	6237	5259	5680	5599	4310	4245	5274	4655	5317	5003
J delta (kW)	978		81		65		619		314	

The extra power consumption resulted mainly from extra power required to compensate for the poor condition of 2<sup>nd</sup> section. The efficiency degradation was accompanied by the degradation in the ability to deliver required pressure ratio, so that other sections had to expend more power.

Sum delta J (kW)	2057
dh_lp (kJ/kg)	844
t/hr, extra steam	8.774
Total cost (EU)	<b>2,305,790</b>
hrs/year	8760
Euro/ton	30

## Conclusion

This paper presented a case of the Charge Gas turbo-compressor train of ENI Versalis Ethylene plant located in Priolo, Italy, where a team of turbomachinery experts successfully used performance monitoring software to diagnose compressor fouling due to polymerization. The team consulted with the facilities engineering and operating staff to develop and implement a Performance Monitoring System. The solution used standard software products to monitor and integrate with plant systems to ensure a technical solution that would work in the operations environment.

The paper provides an overview of the performance monitoring calculation and demonstrates how such systems can be integrated with the plant DCS and Historian, so that operating personnel can easily monitor Key Performance Indicators, such as steam consumption efficiency, and plan and execute corrective actions. Implementation of these systems by knowledgeable experts have resulted in real-world savings of roughly Euro 2.3 million. The method and tools are easily repeatable.