

FIELD PERFORMANCE TESTING OF TURBOMACHINERY

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Abstract. *The performance of a great number of turbomachines that can be found in the process industry, especially those in oil refining and petrochemical sector, is critical for the production and economy of these companies. Procedures for field performance testing of these critical machines are deemed necessary. The objective of this paper is the systematization and proposal of field performance tests procedures for turbomachines of process plants in refining and petrochemical industries. These procedures should take into account the specifications for a factory performance test, which can be done more precisely, with the limitations found in the field, in such a way that it should be possible to determine if the performance of the machine is similar to the original. Procedures for field tests of centrifugal pumps centrifugal compressors, steam turbines and gas turbines are presented.*

Keywords: *turbomachinery, field test procedure, refinery and petrochemical industries*

1. Industrial Relevance of Turbomachinery Performance

The relevance of performance evaluation of turbomachines increases every day, as the size of the plants, the profit loss in case of unplanned shutdowns and the need to save energy go up. The present value of the energy consumed (PV) by process turbomachines can be 1 to 5 times the initial cost of the machine (CC) as shown in "Tab. 1" where the ratio of energy cost to initial cost of centrifugal process pumps are indicated as a function of the pump power. These values clearly show the importance of the optimization of the field performance.

Table 1 – Values of energy and initial cost of centrifugal process pumps

POWER (kW)	CAPITAL COST (US\$)	ENERGY CONSUMPTION (US\$/year)	PRESENT VALUE OF ENERGY COST (US\$)	PV/CC
11,1	44.799	8.807	51.497	1,15
41,0	75.331	32.292	188.824	2,51
149,0	126.254	117.426	686.635	5,44

The first step of the machinery performance management is the determination of the actual performance of the installed machine. This paper describes simple procedures for field tests of the process turbomachinery used in the petrochemical and refining industries and discusses some real world examples of field tests.

2. Performance Testing in the Field

Performance testing of turbomachinery on a test bench is a routine activity, done regularly during the manufacturing of new machines. International Standards are available regulating every aspect of these tests, as they are usually connected to contract requirements.

Field testing is not, with the notable exceptions of machines that are too big for the existing test benches, related to contractual requirements regarding acceptance of a new machine, but are more related to the monitoring of the machine behavior in order to allow maintenance planning.

Trend of the outcomes of these field test result in an accurate knowledge of the condition of the machine (see "Fig.1"), so maintenance planning can be done.

The following issues are relevant for the field tests:

- Type of machine to be tested;
- Purpose of the tests and acceptance criteria;
- Test procedure;
- Available instrumentation;
- Fluid properties.

This paper will describe some guidelines on how to deal with the most common situations regarding field testing of turbomachinery.

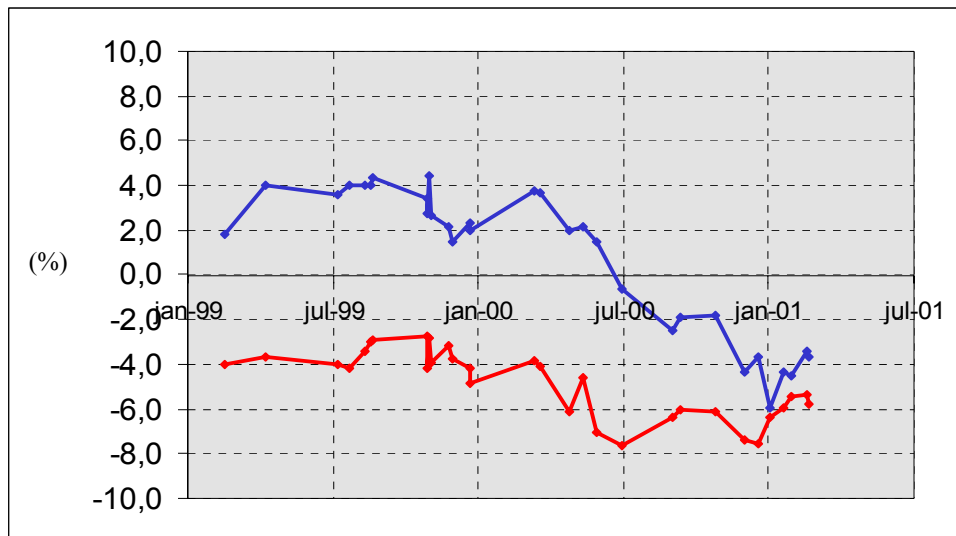


Figure 1. Trend of the difference between actual and original values of head (upper curve) and efficiency (lower curve) of a centrifugal air blower

An important decision to be made is how close to a bench test it is desired that the field test should be. Although it is not usual, a field installation can be designed to comply with the test standards. Nevertheless, the most common situation for a field test is the need for the maximization of the accuracy of the results with the installed hardware.

3. Energy Equation

The basis for turbomachinery performance testing is the energy equation:

$$w = h_{\text{inlet}} - h_{\text{outlet}} + \frac{1}{2} (v_{\text{inlet}}^2 - v_{\text{outlet}}^2) + q \quad (1)$$

Where w is the specific work, h is the specific enthalpy, v is the velocity and q is heat per mas unit. The main task is to measure or calculate the various terms of the equation, so the performance can be evaluated. Most terms can be disregarded and the equation reduced to:

$$w = h_{\text{inlet}} - h_{\text{outlet}} \quad (2)$$

This means that the change in the energy level of the fluid will be equal to the machine shaft work.

4. Test Procedures

Performance evaluation is a comparison of the actual field performance with a reference, usually the bench test results. The operating parameters (pressure, temperature, electric current, etc) have to be reduced to comparable terms (head, efficiency, heat rate, etc), so they can be compared.

The most precise way to do a performance evaluation is complying with the ASME PTC (Performance Test Codes), although this usually cannot be done in the field.

The main differences between a bench and a field test can be summarized by the following:

- Suction and discharge piping layout does not comply with the standard requirements;
- Precision and layout of the process monitoring instrumentation is not the same as the test instrumentation;
- Operating conditions during a field test may not be similar to the original ones, but this condition is easily met in a bench test;
- Operating parameter variation control is usually more difficult in a field test;
- Fluid composition and properties may not be exactly known in a field test.

A valid test can be done, however, even with these difficulties.

4.1. Accuracy, sources of errors and tolerances

The test results are subject to a certain uncertainty. For certain kinds of test, the test procedure and the accuracy of the instruments influence the uncertainty of the test. The easiest way to handle this problem is to obey the existing standards as much as possible.

The main sources of error that affect a field test are:

- Instrument error, both related to calibration, conversion or readings;
- Abnormal operation of the equipment;
- Units conversion errors;
- Uncertainty of the fluid properties;

Test tolerances shall be defined before the test. These tolerances can be different for an acceptance test of a new machine and for a routine test for performance monitoring. The determination of tolerances for the field tests shall consider:

- The trend of the performance;
- The expected time that the machine shall operate before the opportunity for maintenance;
- The cost of energy and the energy expenditure;
- The minimum acceptable performance due to process constraints.

4.2. Field Tests of Centrifugal Pumps

The main objective of a field performance test is to compare the actual performance of the machine with the original one. Performance testing of centrifugal pumps is regulated in ASME PTC 8. This can be done easily by plotting the actual operating point on the original operating curve. In order to determine this point, operating parameters must be measured and actual flow, head and efficiency calculated, according to the following relations:

$$Head = \frac{(P_d - P_s)}{9,81 \cdot \rho} \quad (3)$$

$$W_h = \frac{9,81 \cdot Q \cdot H \cdot \rho}{3600} \quad (4)$$

$$\eta_b = \frac{W_h}{W_{ac}} \quad (5)$$

P_d and P_s : suction and discharge pressure (N/m^2)

ρ : fluid density (kg/m^3)

W_h and W_{ac} : hydraulic and driver power (W)

η_b : pump efficiency

Q : pump flow rate, (m^3/h)

4.2.1. Analysis of a Real Case

The first author has found that 75% of the performance problems observed were due to some kind of problem with the pump. Figure 2 illustrates one of these cases, in which two centrifugal pumps with similar performance were tested (Pump 1 and 2 are the original head and efficiency curves of the pumps). Pump 1 (Test 1 – red squares) presented results quite close to the original ones (head and efficiency). Pump 2 (Test 2) was not able to supply the flow required by the process.

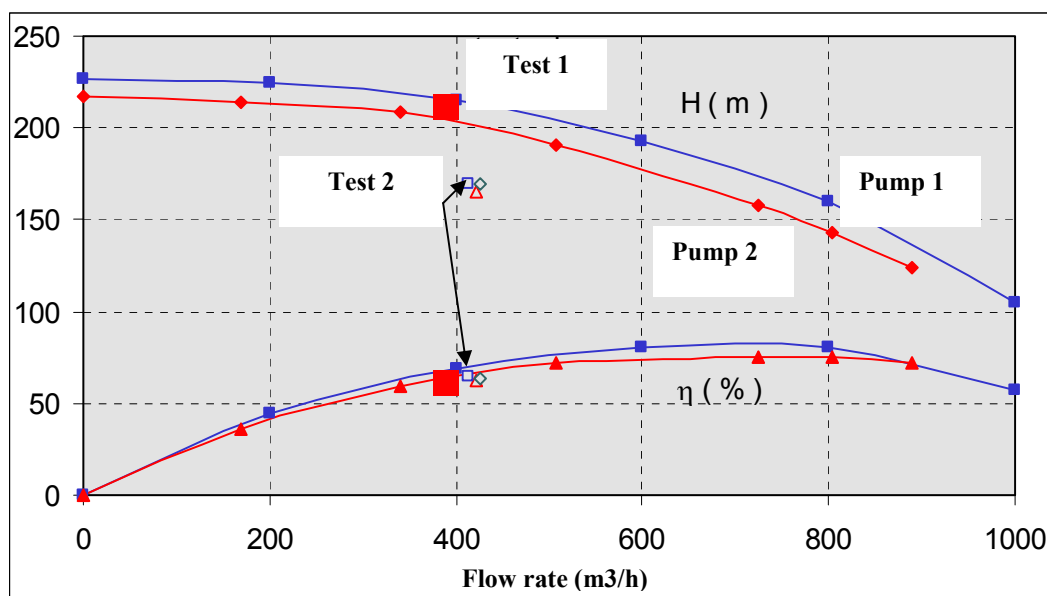


Figure 2. Results of performance tests of a centrifugal pump

It can be seen in “Fig.2” that pump 2 efficiency is quite close to the original efficiency, but developed head is significantly smaller. The reason for this became obvious after the pump was disassembled. The damage observed has been caused by some sort of foreign object which reduced both the diameter and the outlet angle on the impeller blades (see “Fig. 3”). The finishing of the surfaces was not affected; this is the reason why only the developed head has been reduced, as head is related to the square of the diameter and to the cotangent of the outlet angle.

4.3. Field performance tests of centrifugal compressors

The basic ideas behind the performance testing of centrifugal compressors and pumps are the same, but the former can be more complicated due to the compressibility of the gas. This compressibility causes the performance curves of the compressor to be applicable to one set of suction conditions only (pressure, temperature, flow, speed, gas composition), as the actual head is related to the conditions on the discharge of each impeller. Euler’s equation can be simplified, considering that flow is radial on the impeller entrance (Amaral Affonso, 2005).



Figure 3. Impeller of the damaged pump

The usual set of performance curves, where head and efficiency are plotted against suction flow, is an approximation of the real performance, if the suction conditions are not exactly the same. The main condition for this is that gas compressibility should be the same in both cases, so discharge flow will not be affected. The application of the continuity condition for the gas flow inside the impeller results (Eq. (6)):

$$Q_d = Q_s \cdot \frac{\rho_s}{\rho_d} \quad (6)$$

This equation shows that the discharge flow rate (Q_d) is different from the inlet flow rate (Q_s), because of the change in the density of the fluid (ρ), and it can be concluded that the energy transfer and the performance curves will be the same in different operating conditions only if the compressibility of the gas is maintained the same.

4.3.1. Similarity conditions

Although head and efficiency can be calculated for any operating condition, the comparison of the calculated values with the original performance data can be done only if the operating conditions are similar.

This means that the gas flow at test conditions needs geometric, kinematic and dynamic similarity with the gas flow at the test conditions. The similarity conditions for the test have been set in the test codes (ASME PTC 10) and are summarized below:

- Specific volume ratio shall be within 5%;
- Flow coefficient shall be within 4%;
- Mach number shall be +7%, -4% for machines with Mach number greater than 0,8;
- Reynolds number shall be in the range from 0,1 to 10 times the original one.

4.3.2. Sources of error

Besides the possible errors mentioned above, the influence of errors on the measured variables have a greater influence on the calculated values than on the preceding case. Some examples are described in "Tab. 2" and "Fig. 4", where it can be seen the error on the calculated variable introduced by a 1% error on the measured variable. The example has been calculated for the compression of propane from 2,0 kgf/cm² to 10,0 kgf/cm² gage, suction temperature 40,6 °C.

Table 2. Influence of the temperature measurement error in the politropic efficiency

T_d (°C)	Politropic Efficiency
99	79,60
100	77,99
101	76,48

The non-linearity on the method used increases the errors on the calculated variables.

The method used to calculate head and efficiency is due to Schultz, and is described in PTC 10.

4.3.3. Analysis of a real case

A centrifugal compressor operating on a refinery process has shown deterioration on the performance, as shown in “Fig. 4” (red squares are field operating points).

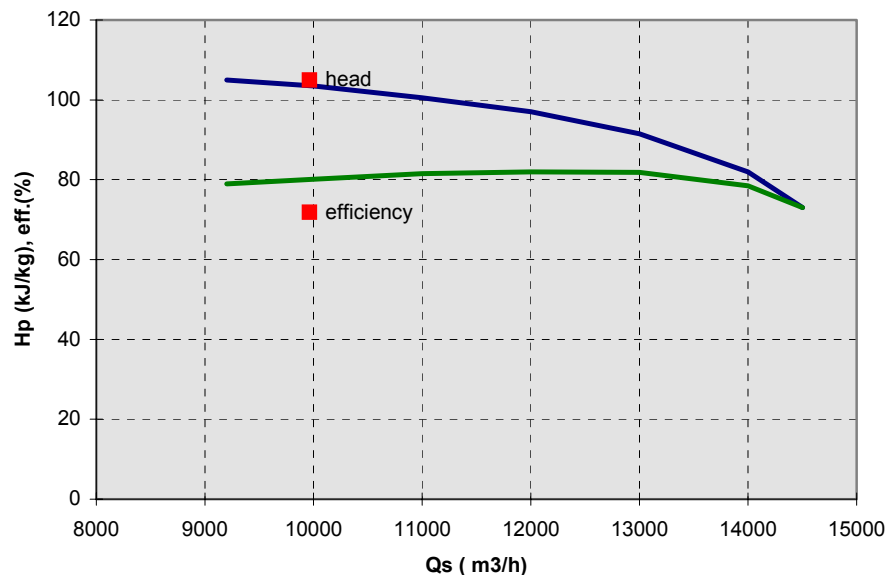


Figure 4. The reduction on the performance of a centrifugal compressor

After opening the compressor casing, a heavy deposit caused by the polymerization of some components of the gas has been found (see “Fig. 5”).

4.4. Field Tests of Steam Turbines

Unlike pumps and compressors, turbines retrieve energy from the fluid flow. The most common method to make a field performance test of a steam turbine is the enthalpy drop test, when the operating conditions are measured and calculate the isentropic efficiency by dividing the real enthalpy drop by the isentropic enthalpy drop. Steam turbine testing is regulated by ASME PTC 6. Fluid properties are calculated according to the International Association for the Properties of Water and Steam (IAPWF – IF97).

This simplification is useful for backpressure steam turbines. Condensation steam turbines are best evaluated by the determination of the steam rate, the ratio between produced power and steam consumption. This is especially useful for turbines with extractions, reheating, etc, situations where it is not very easy to define the isentropic efficiency for the whole turbine.

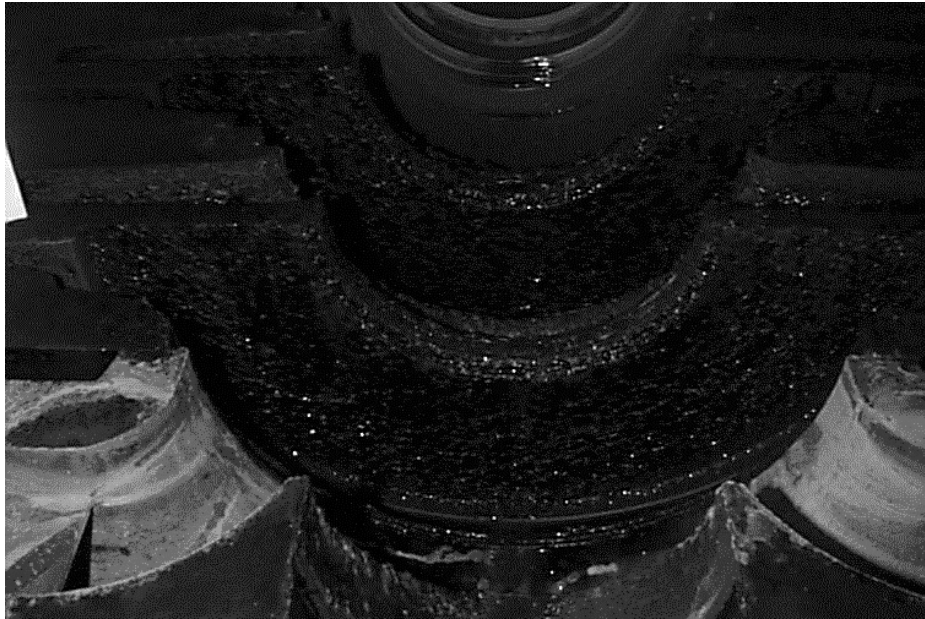


Figure 5. Heavy incrustation inside the compressor was responsible for the loss of efficiency

Again, the non linearity of the process causes small measurement errors that result in large errors on the calculated results, as shown in “Tab. 3”.

Table 3. Influence of the steam temperature measurement error on turbine isentropic efficiency.

	Turbine admission	Turbine outlet	Actual turbine outlet (error 1% T)	Isentropic turbine outlet
P (kPa gage)	11143	4255	4255	4255
T (°C)	520,0	395,3	399,3	366
h (kJ/kg)	3414	3499	3208	3127
η_{iso}		0,750	0,716	

Actual testing shall be done with the same conditions as those of design, as far as possible. Similarity conditions for the steam flow have not been determined. Small deviations are accounted for using corrections supplied by the turbine manufacturer.

4.4.1. Real case analysis

A high speed turbine driving an electricity generator has been tested by the first author, according to the enthalpy drop method. The results are given in “Tab. 4” and “Fig. 6” (red square).

Table 4. Operating conditions

P_{adm} (kPa gage)	11751
T_{adm} (°C)	519,9
$P_{ex Man}$ (kPa)	4954
T_{adm} (°C)	418,3
Q_v (t/h)	91,0
h_1 (kJ/kg)	3409
s_1 (kJ/kgK)	6,58
h_2 (kJ/kg)	3243
h_{2s} (kJ/kg)	3149
heat rate (kg steam/kWh)	21,6
Power (kW)	4211
η_{iso} (%)	63,9%

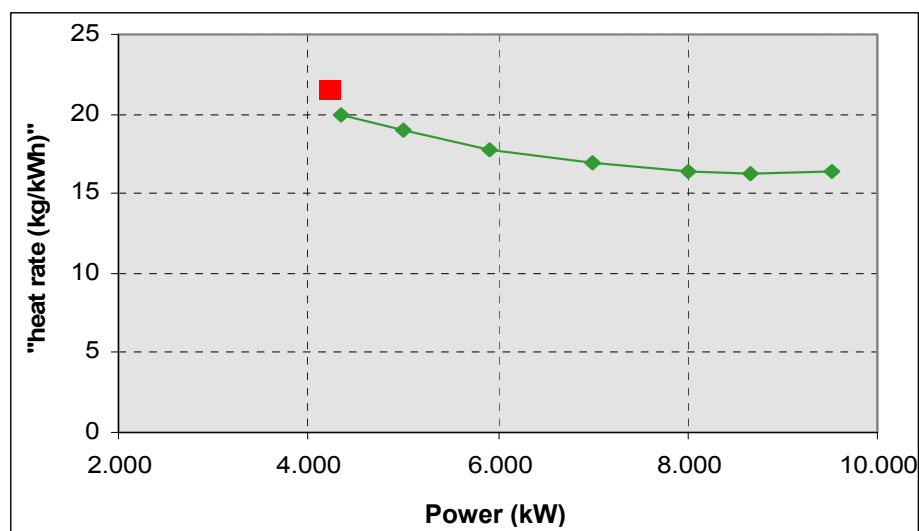


Figure 6. Comparison of the actual (red square) with the original heat rate of a steam turbine.

4.5. Field Tests of Gas Turbines

Gas turbines are amongst the most complex turbomachinery types that can be found, as it consists of a compressor, a burner and a turbine, all of them operating together.

Field testing of gas turbines can be an overall performance test, where only the overall heat rate is determined (as described in PTC 22) or a complete individual test of all the components, assessing their performance. The first approach can be used for an acceptance test and the latter for a diagnostic test to find out the reasons for an inadequate performance of the train.

The diagnostic testing can be done as described bellow, after collection of the actual operating parameters and original performance data:

- ✓ Determination of the chemical composition of the intake air;
- ✓ Calculation of the thermodynamic properties of the gas inside the machine;
- ✓ Calculation of the isentropic efficiency of compressor, turbine and overall;
- ✓ Comparison of calculated shaft power with load shaft power, as a consistency check.

Determination of thermodynamic properties is done with the help of measurement of temperature and pressure at various points on the cycle and determination of gas composition on the turbine exhaust. Any fluid properties code can be used.

When air flow at the inlet of the compressor is known, turbine outlet power W_{tg} can be calculated as:

$$W_{tg} = W_t - W_{cp} \quad (7)$$

W_t is the turbine generated power and W_{cp} is the compressor demanded power. Knowing air and combustion gases flow rate and the specific enthalpies at inlet and outlet sections of the compressor and turbine, it is possible to determine W_{tg} .

When air flow rate at the inlet of the compressor is not known, turbine outlet power is calculated from the heat input and turbine efficiency:

$$\eta_{conj} = \frac{W_t - W_{cp}}{W_t - W_{cp} + Q_{rej}} \quad (8)$$

Q_{rej} is the rejected heat rate between gas turbine outlet and the stack outlet. The net power (W_{conj}) is:

$$W_{conj} = \dot{m}_{comb} \cdot PCI \cdot \eta_{conj} \quad (9)$$

Two gas turbines performance studies can be found in Amaral Affonso (2005).

5. Conclusion

The main conclusion of this paper is that it is possible to perform field performance tests of turbomachinery and obtain results that are accurate enough for routine decisions regarding operation and maintenance of the machines. The field test procedures outlined should not be used for an acceptance testing.

6. References

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