COMPARISON OF TEST MEASUREMENTS TAKEN ON A PIPELINE COMpressor / GAS TURBINE UNIT IN THE WORKSHOP AND AT SITE

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ABSTRACT
The first part of this paper describes the test installation of the gas turbine and the compressor in the workshop, test execution, measuring methods, evaluation and measuring uncertainties.

The second part of this paper describes the site installation, execution of the test under full load conditions on natural gas, measuring methods, evaluation and measuring uncertainties.

The third part of this paper compares both the measurements and the Reynolds number correction which was used for the evaluation of the pipeline compressor test results in the workshop.

INTRODUCTION
Three identical gas turbine/compressor sets were installed in a new gas pipeline station at Olbernhau (Saxony) near the border with the Czech Republic for the expansion of the STEGAL pipeline system belonging to Wintershall Gas GmbH for natural gas transportation from Russia to Germany. The route for this pipeline is shown in Fig. 1. The anagram STEGAL stands for SACHSEN-THÜRINGEN-ERDGAS-LEITUNG (Saxony-Thuringia natural gas pipeline). The gas turbines are equipped with newly developed dry low-NOX combustion chambers with hybrid burners which achieve particularly low emission values [1]. All the gas turbines were subjected to a full load test at the manufacturer's workshop facility. All three compressors (barrel type with dry gas seals) were likewise mechanically tested in the manufacturer's works.

Fig. 2 shows the three machine sets during assembly in the workshop.

One of the three identical compressors was subjected to a thermodynamic test in accordance with VDI 2045 [2]. This was performed using nitrogen as the test gas in a closed loop at reduced speed and a reduced pressure.

FIG. 1 MAP OF STEGAL - PIPELINE

The method according to ICAAMC [3] was employed for correcting the Reynolds number.

Following the commissioning of all three gas turbine/compressor sets, these were subjected to an on-site
performance test under full load conditions in order to confirm attainment of the guarantee values.

The following describes in detail the various test procedures employed on the works test stand and on site with respect to both the gas turbine and the compressor. The results obtained from the works and site tests are compared and correspondingly evaluated.

**PERFORMANCE TEST OF THE GAS TURBINE IN THE WORKSHOP**

All three twin-shaft gas turbines (THM 1304-D, ISO power rating 9.435 MW) equipped with the newly developed dry low-NOx combustion chambers with hybrid burners, were subjected to a full load test in accordance with ISO 2314 [4] at the gas turbine test stand of MAN GHH in Oberhausen/Germany.

Fig. 3 shows a cross section of the turbine. This heavy-duty industrial gas turbine is of modular construction and can be operated with both liquid fuel and gas.

Firing of the combustion chambers in this case is with natural gas fed directly from a high-pressure pipeline system and reduced to a gas inlet pressure of approx. 19 bar upstream of the skid. The operation of safety equipment such as the gas pressure monitoring system, gas detectors, ventilation monitoring system for the test cell to ensure explosion protection, fire fighting equipment and purge systems, is controlled and monitored during each start and stop cycle as well as during the tests to ensure correct functioning by a programmable logic control.

Fig. 4 shows the gas turbine being installed on the test stand.
The basic test arrangement can be seen in Fig. 5.

FIG. 5 TEST ARRANGEMENT OF GAS TURBINE

The power output at the power turbine shaft (coupling) is absorbed by a fluid dynamometer. This dynamometer features a separate speed control system with which the power turbine speed can be held constant. The torque is measured by a precision load cell and a defined lever arm. Employment of this dynamometer arrangement in conjunction with the microprocessor system of the gas turbine enables the measurement of a performance map in which either the gas generator speed can be maintained constant and the power turbine speed adjusted, or alternatively the power turbine speed can be held constant and the gas generator speed varied.

A typical gas turbine performance map is shown in Fig. 6.

FIG. 6 STANDARD ISO PERFORMANCE MAP OF GAS TURBINE THM 1304-D

run, exhaust gas measurements were also carried out. A probe was installed in the exhaust stack to obtain samples for emissions analysis, and these were then fed via a heated conduit to an NO converter. This converter is located immediately adjacent to the sampling point and converts the chemically instable NOx compounds into chemically stable NO. The NO analysis was performed in a downstream, continuous-mode infra-red photometer.

In addition, the O2 and CO concentrations were also measured. For conversion of the measured emissions from test conditions to 15 % oxygen the following conversion factor was used:

\[ f_{\text{conv.}} = \frac{21 - 15}{21 - O_{2,\text{meas.}}} \]

O2,meas. [vol %] measured oxygen concentration during test

Fig. 8 shows the emissions map related to 15 % oxygen, as a function of the coupling power at ISO conditions.

PERFORMANCE TEST OF THE COMPRESSOR IN THE WORKSHOP

Only one of the three identical compressor units was subjected to a performance test on the works test stand of MAN GHH in Oberhausen/Germany; the test was carried out in accordance with VDI 2045.

The ISO conditions are defined as follows:
- Ambient pressure: 1.01325 bar (sea level)
- Ambient temperature: 15 °C
- Relative humidity: 60 %
- No losses at inlet and discharge!

In parallel with the performance test and the mechanical test...
M assured emissions of gas turbine THM 1304-10 with Dry Low NOx combustion chambers and continuously controlled dilution airflow.

FIG. 8 EMISSIONS MAP OF GAS TURBINE THM 1304-D

FIG. 7 PERFORMANCE MAP OF GAS TURBINE THM1304-D FOR SPECIFIED CONDITIONS

The design details of the type RV056/04 compressor are as follows:
- Barrel design (vertically split casing)
- Single process stage group comprising four impellers
- Tilting pad journal bearings
- Tandem gas seal
- Diaphragm coupling
- 1st critical speed = 3777 rpm

Fig. 9 shows a cross section of this compressor.

The closed loop test was performed according to the criteria for similarity with nitrogen at a reduced speed and, for power reasons, at a reduced pressure level. The drive power was provided by a variable-speed electric motor via a precision torquemeter and a gear box.

Fig. 10 shows the compressor during the test in the workshop, and

Fig. 11 shows a sketch of the compressor test arrangement.

The inlet conditions and the test speed, were selected to ensure that the compressor operating conditions were similar to those prevailing at site in terms of Mach number and volumetric flow conditions in the region of the guarantee point.
The coupling power was measured using a high-accuracy torquemeter [5] (0.133% full scale, 0.1% repeatability). Two gear rims are mounted on the measuring shaft. When a load is applied, the gear rims rotate relative to one another. This relative rotation is evaluated, with the theoretical rotation by one tooth representing a phase shift of 100%. During the measurement process, the torquemeter shaft temperature is continuously measured. The change in shaft stiffness which occurs with temperature is compensated in an electronic evaluation unit. For each 100 K of temperature rise, the shaft stiffness decreases by approx. 2.45%.

In conformity with ASME PTC-10 [6], a multi-measurement instrumentation arrangement was employed to determine the conditions at the inlet and outlet. For the pressure measurements, pressure transmitters with an accuracy of ±0.1% full scale were used. The temperature measurement was performed with RTDs to an accuracy of ±0.5 K.

The measurement of the electrical variables was performed with a computer-controlled digital voltmeter featuring 13 selectable ranges (15 bit resolution per measuring range). This digital voltmeter also features a high level of voltage noise suppression. The flow measurement was performed with the aid of a metering section (orifice plate with comer taps; ancillary equalizer) in accordance with DIN 1952 (ISO 5167) [7].

The gas capacity, as derived from the measurements taken at the inlet and outlet of the compressor and the gas mass flow obtained from the flow rate measurement, was determined using real gas behaviour relationships according to BWRS (Benedict-Webb-Rubin-Starling). The mechanical losses of the test stand gearbox and of the bearings of the compressor were determined from the temperature rise of the lubricating oil. The power losses of the dry gas seals are negligible.

The radiation losses were determined by surface temperature measurements around the casing.

Two different test methods were employed:
- direct determination of the coupling power by means of the torquemeter and allowance for mechanical and radiation losses;
- indirect determination of the coupling power via the heat balance taking into account the mechanical and radiation losses;

reveal a relatively good level of agreement within the framework of the individual measurement uncertainties.

One problem which arises in test stand measurements is the precise determination of the radiation losses. In the case in question, comparatively large masses (casing, piping) undergo relatively high temperature changes within short periods of time, caused by changes in the pressure ratio, the suction temperature of the compression fluid and also the ambient conditions (temperature, air movements, etc) during measurement of the performance curves.

The heat exchange with the environment by means of conduction, convection and radiation was considered by using the following method. The surface temperatures of the compressor casing and the piping were measured inside subdivided areas. For further calculation a reference temperature was determined by
weighting the single surface temperatures with the single areas. For computation of the heat transfer the following equation [2]
was used:

\[ \dot{Q} = \alpha \cdot A \cdot \Delta T \]

\[ \alpha = 14 \frac{W}{m^2 \cdot K} \]

\[ A \left[ m^2 \right] \text{surface area} \]

\[ \Delta T \left[ K \right] \text{temperature difference between surface and ambient air} \]

The time required until quasi steady-state conditions occur (i.e. until temperature stability has been achieved), can extend for longer periods in particularly unfavourable circumstances.

From experience, therefore, it was decided that the efficiency calculations for this test should be performed using both the direct power measurements from the torquemeter and the indirect power measurements from the heat balance. This procedure takes into account all the various factors which might influence the result.

The employment of a precision torquemeter has the advantage that the weighted measurement uncertainty according VDI 2045 in respect of the coupling power is smaller than the measurement uncertainty resulting from the heat balance method alone. A comparison of the individual measurement uncertainties as determined according to VDI 2045 reveals the following values:

<table>
<thead>
<tr>
<th>Evaluation process</th>
<th>Heat balance meter value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measurement uncertainty in respect of coupling power</td>
<td>( \pm \ 1.89% \ \pm 0.65% \ \pm 0.62% )</td>
</tr>
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</table>

Unrelated to the evaluation process for the coupling power value, further measurement uncertainties were also ascertained as follows:

- Suction flow volume \( \pm 1.50\% \)
- Pressure ratio \( \pm 0.22\% \)

The specific compression work and the efficiency data were converted using the ICAAMC formulae. In Fig. 12, one operating point during the test is entered in a diagram which shows the permissible range for this conversion process.

Following thermodynamic testing, the compressor successfully underwent a mechanical test in accordance with API 617 [8].

**PERFORMANCE TEST ON SITE**

Fig. 13 shows the complete facility shortly after completion in 1993.

All the main requirements regarding the intended site performance test were taken into account when planning the installation. The main provisions were as follows:

- Two additional nozzles with shutoff valves provided for the static pressure measurements at the inlet and outlet of the compressor. The pressures were measured in straight pipe sections (approx. 3 - 4 x pipe diameter from the flanges of the suction and discharge nozzles)

- Additional orifice metering section in accordance with DIN 1952 (ISO 5167) provided directly at the compressor outlet to ensure that accidental leakage through inadequately sealed branches or valve fittings would not falsify the measurement result.

- Two additional pressure measuring points at the inlet to the gas generator of the gas turbine. These served to

... by independent pressure transmitters with an accuracy of \( \pm 0.1\% \) full scale.

- Two additional thermowells for the temperature measurement at the inlet and outlet of the compressor. These thermowells are located within the immediate vicinity of the pressure sensing points. Temperature measurement was performed using thermocouples with an accuracy of \( \pm 0.5 K \).

An additional orifice metering section in accordance with DIN 1952 (ISO 5167) provided directly at the compressor outlet to ensure that accidental leakage through inadequately sealed branches or valve fittings would not falsify the measurement result.

- Two additional pressure measuring points at the inlet to the gas generator of the gas turbine. These served to
determine the suction pressure loss which is necessary to convert to ISO conditions.

Two additional pressure measuring points at the outlet of the power turbine. These enabled the exhaust gas pressure loss to be determined for conversion to ISO conditions.

Measurement of the electrical values, each of which was averaged out over a longer time period, was performed by a computer-controlled digital voltmeter which, for the measuring range concerned, features a resolution of 30,000 digits. This digital voltmeter also has the advantage of a high level of voltage interference suppression.

The fuel consumption was determined by means of a vortex flowmeter. The flow rate signal was evaluated, as were the corresponding pressures and temperatures, by a flow computer in which the measurement results were converted to standard conditions \( p = 1.01325 \text{ bar}, t = 0 ^\circ \text{C}, \text{dry} \). This system offers an accuracy of \( \pm 1\% \) full scale.

The fuel and compression gases were continuously analysed by means of a permanently installed, calibrated gas chromatography system.

Fig. 14 shows the configuration of the gas turbine and compressor in the installation without the acoustic enclosure (photo taken during installation).

Prior to commencement of the site testing the first machine set had completed a total of 432 operating hours and 42 starts.

Fig. 15 shows the site test arrangement during the acceptance testing procedure.

The main advantages of this test arrangement as compared with those employed in many other pipeline compressor stations are as follows:

- Independent operation.
- Full coverage of the performance map.
- Individual measuring point setting.
- Stability during the test program.

While the two gas turbine/compressor units are connected to the network, the unit to be tested can be operated separately in a closed loop without influencing the other units.

b) Full coverage of the performance map.

With the arrangement shown in Fig. 15 it was possible to operate the system throughout the complete range of its performance map, i.e. in addition to the relevant performance data in the area of the guarantee point, the choke limit and also the surge line of each speed characteristic could be measured.

c) Individual measuring point setting.

Irrespective of the prevailing pressure level in the pipeline network, the operating points of the unit under test could be individually set.

The suction pressure level could be adjusted both by superfeed injection at the suction end from the pipeline network and by controlled unloading at the discharge.

The operating points along the individual speed characteristics were adjusted by throttling the bypass control valve.

The necessary inlet temperatures were obtained by selective switching of the air coolers. In addition, the speed of the fans could also be varied in a stepwise fashion.

d) Stability during the test program.

Thanks to effective disconnection from the network, the entire system could be regarded as operating at a quasi steady state during the measurement program in relation to pressure, temperature and throughput.

Figs. 16, 17 and 18 show the most important results of the site performance test.

Fig. 16 shows the compressor speed performance map with the polytropic head plotted as a function of suction volume flow and speed. The surge lines were determined in the case of the
performance curves for the three guarantee points G1, G2 and G3. The performance curve at N = 8080 rpm serves to verify the maximum gas turbine output power value.

Fig. 17 shows the coupling power converted to guarantee conditions as a function of suction volume flow and speed.

The measurement uncertainties for these values were likewise determined in accordance with VDI 2045:

- Suction volume flow \( \pm 1.41\% \)
- Pressure ratio \( \pm 0.51\% \)
- Coupling power \( \pm 2.08\% \)

Fig. 18 shows the specific power consumption of the gas turbine and compressor set converted to guarantee conditions as a function of the suction volume flow and speed of the compressor.

The other two identical compressor/gas turbine units were tested in the same way as described above. Results from these units fell within the above mentioned measuring tolerances.

COMPARISON BETWEEN MEASUREMENTS IN THE WORKSHOP AND ON SITE

A direct comparison between the test measurements in the workshop and the results obtained on site is of particular interest to both the operator and the manufacturer. This applies to both the mechanical test results and the thermodynamic measurements of the system. Because the gas turbine, as already described above, was tested in the workshop under conditions which approximated very closely to those prevailing on site, the following comparisons will concentrate on the compressor. As regards the compressor thermodynamics, of particular interest is the
transferability of N₂ closed loop test results in the workshop to the specific site operating conditions. In order to improve the comparability of both sets of measurements, the performance curves were converted to standardised values in accordance with VDI 2045. To this end, non-dimensional values were selected.

Also the running behaviour of the compressor rotor was examined at different gas densities.

a) Gas turbine

Fig. 19 shows the coupling power converted to ISO standard conditions and related to the design value as a function of the heat rate (also converted to ISO conditions and compared to the design value).

![Fig. 19 PERFORMANCE MAP OF GAS TURBINE. COMPARISON BETWEEN TEST IN THE WORKSHOP AND AT SITE.](image)

As is apparent from the diagram, the workshop results show good agreement with the on-site test results. The discrepancy in the heat rate at constant coupling power in the area of the main guarantee point represents a deviation of no more than ± 0.5%. This variation is due firstly to the different measuring processes employed in the workshop and on site; secondly it must also be remembered that the gas turbine at the time of on-site testing had already completed 432 operating hours and 42 starts. Moreover, wash-cleaning of the gas generator prior to the site testing was not possible for operational reasons.

b) Compressor thermodynamics

Fig. 20 shows in a direct comparison the polytropic work coefficient as determined in the workshop and on site as a function of the flow coefficient at design speed. The two performance curves virtually coincide and exhibit excellent agreement.

![Fig. 20 PERFORMANCE MAP #3 OF COMPRESSOR. COMPARISON BETWEEN TEST IN THE WORKSHOP AND AT SITE.](image)

Fig. 21 shows the polytropic efficiencies as revealed by the workshop and on-site tests as a function of the volume flow coefficient at design speed. The level of the efficiencies measured in the shop tests, which were appropriately corrected in accordance with the ICAAMC method, is slightly lower than the efficiencies measured on site.

The ICAAMC process is based on the analogy with turbulent flow in technically rough conduits, and therefore takes into account the relative roughnesses. It is therefore important for the overall conversion calculation that a representative relative wall roughness for the entire compressor can be determined, i.e. a representative value which takes into account the surface conditions of the impeller gas passages, deflector vanes, diffusers, volutes and nozzles.

The representative average roughness was determined using the equation:

\[
R_a = \frac{1}{2} \cdot R_{a,i} + \frac{1}{3} \cdot R_{a,d} + \frac{1}{6} \cdot R_{a,v}
\]

- \(R_a\) [µm] representative average roughness (overall)
- \(R_{a,i}\) [µm] average roughness (impeller)
For the above mentioned test evaluations of the compressor a determined average roughness of \( R_a = 2 \) \( \mu \)m was considered.

As already mentioned above, the measurement uncertainties were determined in accordance with VDI 2045. In Fig. 22, the relevant measurement uncertainties for two measuring points are plotted in the area of the design point. As can be seen from this diagram, the two measurement uncertainty fields intersect with a considerable overlap. It can be concluded from the graph, therefore, that the two measurements are mutually plausible.

These measurement uncertainties are distinctly smaller than the power tolerances of 4% according to API 617 and ISO 10439 basing on the API standard [9].

c) Compressor mechanical performance

Fig. 23 shows the frequency spectra recorded by a shaft vibration sensor at the drive end, providing a comparison between the shop test run under partial load conditions and the full load test performed on site. From a total of four sensors (two in each case at the drive end and the non-driven end, offset at 90°), that sensor with the highest values was selected for the purpose of providing a clearer picture.
It is apparent from the direct comparison between the spectra that the compressor rotor exhibits a stable running behaviour across a wide range of gas densities, i.e. no dominant subsynchronous interference frequencies occur in relation to the nominal frequency.

SUMMARY
Three identical gas turbine/compressor units were tested in the workshop and on site. All the gas turbines were tested under full load conditions both in the shop and after final installation. One of the three identical compressors was subjected to a performance test in the workshop. This test was performed in a closed loop with nitrogen at a reduced speed and reduced pressure. Conversion of the specific compression work and efficiency data was carried out using ICAAMC formulae, which validations have been investigated in different ASME publications, i.e. in [3].

The pipeline station was designed, inter alia, in order to ensure optimum measuring conditions for the site performance test. As a result, the entire performance range could be tested under original conditions.

Comparison between the measurement results achieved in the workshop and those obtained on site reveal a very good level of agreement with regard to the compressor performance map.

In spite of the different instrumentation, and variations in the measuring and evaluation processes applied during the workshop and site tests, there is relatively good agreement on the measured compressor efficiencies after consideration of existing measurement uncertainties. Using the above-described example, it was demonstrated that, by employing a precise measurement regime and through the application of the ICAAMC Reynolds No. conversion process, very good guarantee comparisons can be obtained from the workshop test stand as compared to the performance at original site conditions.

A comparison of the frequency spectra likewise shows that the behaviour of the compressor in terms of its shaft vibrations is excellent both in the workshop and on site.

REFERENCES


