Technical aspects of a large size industrial process turbo compressor revamp

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Abstract: This paper describes several modernization aspects of the process 8 MW air compressor and its driver. The main aim of the revamp was to increase internal efficiencies of both the compressor and turbine and increase the load of the compressor without additional power consumption. The original pressure ratio was preserved. To meet these requirements a redesign of the flow path of both machines was necessary. Moreover, the turbine working conditions were changed from extraction-condensing to condensing. In terms of compressor, impeller blade redesign, adaptation of diffusers and Abradable seals were a part of the scope.

The revamp was completed in April 2014 with all the requirements met. This paper covers the general outline and a comparison of the original and modernized machine operating parameters, design and technical assumptions and also describes some problematic issues which occurred during the realization of the project.

Keywords: Process compressor; Turbine; Impeller; Revamp

1 Introduction

Revamp is a common practice in the petrochemical industry. Apart from its relative simplicity (involves no changes or minor changes to existing casing, piping and external devices), it is a reasonable alternative to brand new machine purchase, due to cost and time limitations. These limitations are even more important when it comes to critical machinery, like process compressors, as their performance often determine the production capabilities of the entire plant or installation.

It is clear that production growth strictly depends upon substrate or catalyst availability. Sometimes, if the supplying machines (like air-supplying compressors) have still some margin to increase the working parameters (e.g. operating below max continuous speed), the production can be increased easily without major overhauls. In other cases, where the machine has reached its maximum performance point, some modifications are needed. This involves flow path modernization, but not all elements necessarily require replacement. In the case of process compressors, a several percent increase in flow capacity can be achieved by rotor modifications and only minor changes to the diffusers with no changes to the existing casing.

If the power supply required by the modified compressor is guaranteed by the driver, no further changes are needed. Otherwise, to secure safe compressor work, modifications to the turbine are needed as well, but also do not affect the machine entirely [3, 4, 6–9].

2 Compressor modernisation assumptions

The discussed 8 [MW] compressor (Figure 1) is a 6 stage, 22D bladed (except for stage 1, which is 3D) radial com-

Figure 1. Old compressor rotor.
compressor used as a main source of air in the fluid catalytic cracking process at the refinery. For the purpose of the process, in which hot air is required, no coolers were applied between the stages. The compressor was originally driven directly by a 10 (Curtis + 8 stages) stage extraction-condensing steam turbine in a row configuration (Figure 2). Nominal speed was 3710 RPM with pressure ratio of 4.19.

In this case, the main aim of the compressor revamp was to:

1. Increase internal efficiencies
2. Increase intake load capacity by 10%.
3. Decrease the pressure ratio at the rated point.
4. Reduce power consumption by 800 [kW] (constant rated speed)

The principal aim was to increase the intake capacity by 10%, which corresponded to flow increase from 39.64 [kg/s] (142 701 kg/h) to 43.67 [kg/s] (157 229 kg/h). The original pressure ratio had to be decreased – due to modernized process converter. Satisfying these criteria, as calculations showed, it was still possible to increase the available power margin by reducing compressor power consumption by approx. 800 [kW]. A brief comparison of the original designed, real and modernized parameters is presented in Figure 3.

As the flow path of the original rotor was relatively poor, due to limited calculation methods and available manufacturing technologies at that time, the improvement of the efficiencies by redesigning the channel (hub and tip lines and blade geometry) was relatively easy to achieve. Better shaped leading edges and radius between the blade and hubs and covers, as well as gap minimization between the rotating and stationary elements was also a standard modification working towards better efficiency [1, 2].

Complete calculations of the new rotor blading were performed. Firstly, using 2D computations new meridional sections were designed along with kinematics of the compressor and then with aid of CFD, hub-to-tip blade geometries were produced.

As a result of the design process the inlet and outlet widths of the impellers were increased (Figure 3a). The main changes affected stage 1 and stage 2, which were now both redesigned as 3D, (previously 3D and 2D – respectively), due to its excessive channel widths and significant flow coefficients [4]. Changes to stage 2 are shown in Figure 3b. Stage 1 impeller diameter was increased from 1170 to 1200 [mm]. Other stage diameters were preserved. Inlet blade angles for stage 3–6 were maintained and outlet angles were slightly increased to optimize the overall performance of the compressor. All blades leading edges were moved slightly forward and hub and tip profiles were optimized.

As CFD runs for stage 1 and 2 confirmed, a new capacity of the compressor would reach 43.674 [kg/s]. Due to the changes in diameter the relative Mach number at the outlet of the first stage had increased, but was still at the safe level – no risk of transonic shock.

Having obtained the satisfactory blade geometry, the models of the new impellers stress analysis had to be checked. Conducted simulations performed with aid of PTC Mechanica for nominal point (100% nominal speed) and over speed (10% × 115% of nominal speed) brought acceptable stress results for all investigated stages. (API recommended practice 687 - rotor repair, http://www.api.org/events-and-training/api-u-training/api-u-calendar/2012-events/api-standard-687-rotor-repair-training).
Table 1: Modernized compressor theoretical data.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Designed</th>
<th>True</th>
<th>Modernised 3710 rpm</th>
<th>Modernised 3510 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas compressed</td>
<td>Air</td>
<td>Air</td>
<td>Air</td>
<td>Air</td>
</tr>
<tr>
<td>Relative humidity</td>
<td>98%</td>
<td>100%</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td>Mol. wgt. at intake</td>
<td>28.6</td>
<td>28.6</td>
<td>28.6</td>
<td>28.6</td>
</tr>
<tr>
<td>$C_p/C_v$ value @ inlet</td>
<td>1.3975</td>
<td>1.3975</td>
<td>1.3975</td>
<td>1.3975</td>
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<tr>
<td>Individual gas const. $R$</td>
<td>288.6</td>
<td>288.6</td>
<td>288.6</td>
<td>288.6</td>
</tr>
<tr>
<td>Spec. heat at const. press $C_p$</td>
<td>1015</td>
<td>1015</td>
<td>1015</td>
<td>1015</td>
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<tr>
<td>Compressibility factor $Z$ @ inlet</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Compressibility factor $Z$ @ disch.</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Inlet temp. °C</td>
<td>29</td>
<td>20</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Disch. temp. °C</td>
<td>237</td>
<td>214</td>
<td>202</td>
<td>195</td>
</tr>
<tr>
<td>Inlet press kPa ABS</td>
<td>99</td>
<td>100,8</td>
<td>100,8</td>
<td>100,82</td>
</tr>
<tr>
<td>Disch. press kPa ABS</td>
<td>415</td>
<td>361</td>
<td>379</td>
<td>361</td>
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<tr>
<td>Density kg/m³ @ inlet</td>
<td>1,135</td>
<td>1,192</td>
<td>1,191</td>
<td>1,192</td>
</tr>
<tr>
<td>Density kg/m³ @ disch.</td>
<td>2,819</td>
<td>2,568</td>
<td>2,764</td>
<td>2,672</td>
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<tr>
<td>Capacity</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>KG/HR @ inlet</td>
<td>129840</td>
<td>142701</td>
<td>157229</td>
<td>142701</td>
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<tr>
<td>M³/HR @ inlet</td>
<td>114365</td>
<td>119748</td>
<td>131965</td>
<td>119748</td>
</tr>
<tr>
<td>NM³/HR @ inlet</td>
<td>100495</td>
<td>110450</td>
<td>121694</td>
<td>110450</td>
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<tr>
<td>Compressor speed rpm</td>
<td>3710</td>
<td>3500</td>
<td>3710</td>
<td>3500</td>
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<tr>
<td>Maximum drive KW</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>RPM @ driver coupling</td>
<td>3710</td>
<td>3510</td>
<td>3710</td>
<td>3510</td>
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<tr>
<td>KW @ driver coupling</td>
<td>8079</td>
<td>8279</td>
<td>8555</td>
<td>7476</td>
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<tr>
<td>% ΔP @ external loop</td>
<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Polytropic efficiency $\eta_p$</td>
<td>0.778</td>
<td>0.714</td>
<td>0.780</td>
<td>0.775</td>
</tr>
<tr>
<td>Compression ratio $\pi$</td>
<td>4.19</td>
<td>3.58</td>
<td>3.76</td>
<td>3.58</td>
</tr>
<tr>
<td>Internal power kW</td>
<td>7612</td>
<td>7803</td>
<td>8065</td>
<td>7038</td>
</tr>
<tr>
<td>Total internal power kW</td>
<td>7870</td>
<td>8067</td>
<td>8337</td>
<td>7280</td>
</tr>
<tr>
<td>Isothermal efficiency $\eta_i$</td>
<td></td>
<td></td>
<td></td>
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</tbody>
</table>
Figure 3: (a) New (violet) - old (blue) rotor meridional section; (b) Old (left) and new stage 2 impeller.

To adapt the current stationary parts to the new rotor impellers special inserts were installed in the diaphragms; at the inlets of the diffusers and impeller inlet, as can be seen in the Figure 4 and Figure 5. Impeller inlet inserts combined the flow path shaping element with static seals for the impeller inlet. Flow path analysis indicated that the leakages in all stages should be limited as much as possible, hence self-abrasive ‘Abradable’ seals were applied. This allowed for minimizing the seal gap at contact. This was applied in all rotor seals, including the balancing piston [5, 10–13].

The widening the diffuser channel was also necessary to prevent choking in these regions and to optimize the gas flow between impeller and diffuser.

3 **Turbine modernisation assumption**

The turbine revamp aims were as follows:

1. Working conditions change – extraction-condensing to condensing.
2. Flow capacity increase by approx. 8%.
3. Steam supply changes from half to full.
4. Efficiency increase by new stage seals and gland seal application.

The principal turbine revamp target was to change turbine working conditions from extracting-condensing to condensing, as the extraction was never used at this particular site. Hence, the existing steam box of the extraction valve was generating unnecessary losses – it was decided...
The principal turbine revamp target was to change turbine working conditions from extracting condensing to condensing, as this operation was the most time consuming, mostly due to the size of the diaphragms. After disassembled, new locks for compressor inserts were machined in the diaphragms. After being assembled in the lock each insert itself was machined, so it formed a smooth surface with the own. The complete compressor and turbine rotors, diffuser inserts, was designed in front of R1 stage for better flow directing.

Stage 1 diaphragm with inserts assembled.

Additionally, the steam supply arch before the stage R1 was originally only at a half circumference. The design was changed to a full-circumference supply. Also, a set of inlet guide vanes was designed in front of R1 stage for better flow directing.

to remove it and close the inlet. The secondary target was to increase the flow capacity from 11.66 [kg/s] (42 [t/h]) to 12.77 [kg/s] (46 [t/h]) to allow a safe driving power capacity margin.

Figure 4: Adapting inserts (green).

Figure 5: Stage 1 diaphragm with inserts assembled.

Figure 6: New rotor installed in casing.
Figure 7: Test data at performance map.
In Figure 8 the results of the power performance test are shown. The red curve shows the steam consumption for the old rotor while the blue curve for the new one. The main revamp target was to achieve at least \(1.15 \text{[kg/s]} (4140 \text{[kg/h]})\) steam saving, which was fully achieved. The real steam consumption has decreased by more than 10% of the original. The test proved that the compressor load can be increased by 10% without additional power consumption – the maximum steam consumption was preserved at \(11.9 \text{[kg/s]} (43180 \text{[kg/h]})\). With increased turbine flow up to \(12.77 \text{[kg/s]} (46000 \text{[kg/h]})\) it is possible to overload the compressor for more than 15% of original rated point.

### 4 Realisation of the project and test runs

The scheduled overhaul duration of the plant was as tight as 30 days, so the majority of the work was done before the shutdown. The complete compressor and turbine rotors, diffuser inserts, ‘Abradable’ seals turbine guide vanes and new steam box were ready, with material surplus on base surfaces, allowing for adjustments of axial position, if required. Once the machines were cooled down and opened, checking measurements were performed and elements to be modified were disassembled. New locks for compressor inserts were machined in the diaphragms. After being assembled in the lock each insert itself was machined, so it formed a smooth surface with the diaphragms.

This operation was the most time-consuming, mostly due to the size of the diaphragms. After the installation of the new rotor (Figure 6), the run tests took place and some results can be seen in Figures 7 and 8. In Figure 7 the results of compressor anti-surge test are presented. Due to the completely redesigned compressor and new performance characteristics the surge line of the compressor has changed. It was necessary to check the real surge line and re-adjust the anti-surge system. As can be seen on the performance map a real surge line is located on the left-hand side of the calculated one and compressor is fully operational in the predicted range. It also shows a secure margin of compression of the compressor – the real performance lines are steeper than calculated in the low range capacity.

In Figure 8 the results of the power performance test are shown. The red curve shows the steam consumption...
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allowed for safe milling tool operation down to the bottom of the flow channel of the impellers. These parts were balanced and mounted on the shaft separately with shrink fit. To match the blades between the elements, positioning pins were applied.

### 5 Troubleshooting

All modified impellers were manufactured in the following technology pattern; hub and cover machined with partial blades and joined by welding in half blade height.

It was clear that stage 1 would be the most complex element to manufacture, due to its significant outside diameter, hub height and narrow and deep blade-to-blade channel, which imposed severe limitations for the machining tools. With the technology pattern applied to other impellers it was impossible to apply it to stage 1. The solution to this was to manufacture the impeller as 2 separate parts. Part 1, inlet – BLISK technology, part 2, outlet - welded in half blade height (Figure 9). This allowed for safe milling tool operation down to the bottom of the flow channel of the impellers. These parts were balanced and mounted on the shaft separately with shrink fit. To match the blades between the elements, positioning pins were applied.

### 6 Conclusions

Within the described scope of the modernization, the following major improvements were made:

- **Compressor**
  - Increased intake load capacity (39.64 [kg/s] → 43.67 [kg/s])
  - Decreased pressure ratio (4.19 → 3.76)
  - Reduced power consumption (8279 kW → 7476 W)

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**Figure 9:** New stage 1 impeller manufacturing technology.
Turbine:
- Flow capacity increase (11.99 kg/s → 12.77 kg/s)
- Working conditions change (extraction → condensing)

Based upon the project it can be seen that with relatively small changes significant goals can be achieved within a short overhaul time, not longer than a standard maintenance check, in which both stationary elements and the rotor has to be inspected outside the machine.

All planned work was carried out. Test run results were consistent with the requirements and assumptions. As usual during such projects some problems may occur, and so it was in the described case. However, with good planning and simultaneous revision of the work scope, responding to the occurring difficulties, these problems can be fixed within the available time limits.

The project has once again proved the feasibility of revamps in petrochemical facilities.

References