Revamp of Turbocompressors
New life for existing machines
At a Glance

List of symbols and indices used in the formulas and equations.
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>η</td>
<td>-</td>
<td>Efficiency</td>
</tr>
<tr>
<td>κ</td>
<td>-</td>
<td>Isentropic exponent</td>
</tr>
<tr>
<td>$h_{\text{pol}}$</td>
<td>J/kg</td>
<td>Polytropic head</td>
</tr>
<tr>
<td>$\mu_y$</td>
<td>-</td>
<td>Head factor</td>
</tr>
<tr>
<td>π</td>
<td>-</td>
<td>Pressure ratio</td>
</tr>
<tr>
<td>$\rho$</td>
<td>bar</td>
<td>Pressure</td>
</tr>
<tr>
<td>P</td>
<td>W</td>
<td>Power</td>
</tr>
<tr>
<td>T</td>
<td>K</td>
<td>Temperature</td>
</tr>
<tr>
<td>$\phi$</td>
<td>-</td>
<td>Flow coefficient</td>
</tr>
<tr>
<td>$\dot{V}$</td>
<td>m$^3$/s</td>
<td>Volume flow</td>
</tr>
<tr>
<td>$D_2$</td>
<td>m</td>
<td>Outer impeller diameter</td>
</tr>
<tr>
<td>$u_2$</td>
<td>m/s</td>
<td>Circumferential velocity</td>
</tr>
<tr>
<td>$\mu_u$</td>
<td>-</td>
<td>Circumferential mach number</td>
</tr>
<tr>
<td>z</td>
<td>-</td>
<td>Compressibility factor</td>
</tr>
<tr>
<td>$z_{\text{st}}$</td>
<td>-</td>
<td>Number of stages</td>
</tr>
<tr>
<td>R</td>
<td>J/kmol/K</td>
<td>Specific gas constant</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>kg/s</td>
<td>Mass flow</td>
</tr>
</tbody>
</table>

**Indices**

1. Inlet
2. Discharge
int. Internal
st. Stage
st$_1$. Stage 1
st$_2$. Stage 2
st$_n$. Stage n.
c. Compressor overall
pol. Polytropic
corr. Corrected
Foreword

This document provides a brief overview on the revamp of turbomachinery, in particular turbocompressors, taken care of by MAN PrimeServ. Within the MAN PrimeServ product portfolio, engineering studies and execution of revamps and modernisations on all kinds of turbomachinery play a key role and are important and well established services provided to customers. These services have become increasingly important due to advanced process technologies, increasing importance of energy savings and efforts to reduce emission of carbon dioxide.

Turbocompressors belong to the most important components in a process plant where gas is transported or compressed to meet process requirements. Turbocompressors and their drivers are designed and manufactured for decades of uninterrupted operation. Provided they are operated within design specifications and properly maintained and monitored, they meet these demanding requirements.

Process gas compressors are engineered and tailor made to produce the specified gas flow at the required pressures and, in some cases, temperatures for various operating points. Already at the design stage, the mechanical and aerodynamic design engineers have to demonstrate flexibility to fit, depending on the requirements, tailor made or standardised components into a machine which fulfils the expectations of the customer.

There is a high probability that during the lifetime of a turbocompressor train fundamental design criteria such as the technology of the production process, the requirements on gas pressure and throughput or the gas composition etc., will change. Hand in hand with the re-design and modification of the various process components, the revamping or in rare cases even replacing of the compressor train is quite often inevitable in order to obtain increased production goals or to avoid upset of the economics with a reduced plant output.
The way how changes in the performance of a compressor can be achieved can best be demonstrated by explaining the basic thermodynamic design of a centrifugal compressor. The below step-by-step method provides a brief overview of the procedure of designing a new centrifugal compressor.

Non-dimensional parameters, the flow coefficient $\phi$ and the pressure ratio $\pi$ are crucial to determine the main values of a new centrifugal compressor:

**Flow coefficient:**

\[
\phi = \frac{V_1}{D_2^2 \cdot u_2} \quad [-]
\]

The flow coefficient defines the required first stage impeller diameter and therefore the compressor casing frame size. In order to keep the frame size as small as possible, larger values of $\phi$ can be chosen. This results in lower investment costs but slightly decreases the efficiency of the first stage impeller.

**Overall pressure ratio:**

\[
\pi = \frac{p_2}{p_1} \quad [-]
\]

The overall pressure ratio determines the number of stages required for the compressor. Each stage consists of an impeller, a diffuser and a return channel. Conventional centrifugal compressors consist of two to eight stages, tailor made exceptions are available as well.

One of the most important aerodynamic limitations is the circumferential mach number which is defined as follows:

\[
Mu_2 = \frac{u_2}{\sqrt{k_1 \cdot z \cdot R \cdot T_1}} \quad [-]
\]

The overall performance is derived by the stage stacking method which allows each individual stage to be treated in a straightforward manner.
Step-by-step method of designing a centrifugal compressor:

The basic design tools use stage characteristics which show the efficiency and the head factor of an impeller as functions of a flow factor (Figure 2).

Choosing the flow factor pinpoints the design point of a particular stage. By knowing the required design suction volume flow, the necessary area of the first stage impeller can be calculated as follows:

\[ \phi_{sl1} = \frac{V_1}{D_2^2 \cdot u_2} \rightarrow D_2 = \sqrt[\phi_{sl1} \cdot u_2]}{V_1} \quad [m] \]

The required number of stages is defined by the non-dimensional head factor \( \mu_y \).

The head factor is a measured property of the stage and depends on impeller type, flow coefficient, width of stage and circumferential mach number. Using this equation, the polytropic head of a particular stage can be calculated:

\[ \mu_y = \frac{h_{pol,sl1}}{u_2^2} \rightarrow h_{pol,sl1} = \mu_y \cdot u_2^2 \quad [J/kg] \]

As a simplification, the assumption is made that all impellers utilized in this compressor are of the same diameter and are referring to the same stage characteristics (Fig. 2). Therefore the below relationship can be established:

\[ h_{pol,C} = \sum_{z_{sl}} (\mu_y \cdot u_2^2) = z_{sl} \cdot \mu_y \cdot u_2^2 \quad [J/kg] \]
Using equation (4), the corrected $\phi_{st1,corr}$ can be calculated:

$$\phi_{st1} = \frac{V_1}{D_2^2 \cdot u_{2,corr}}$$  
[-]

(The overall polytropic efficiency has to be assumed for this first calculation)

Equations (6) and (7) are equal and can therefore be equated with each other. The new equation allows determining the required number of stages:

$$Z_{st} \cdot \mu_y \cdot u_2^2 = \frac{\kappa \cdot \eta_{pol,C}}{\kappa - 1} \cdot z_1 \cdot R \cdot \frac{T_1}{\pi} \left( \frac{Z_{st}}{Z_{st,1}} - 1 \right)$$  
[-]

The result of equation (9) will probably not be an integer and has therefore be rounded up to the next higher natural number, thus causing deviations from the preliminarily determined values $\phi_{st1}, u_2$ and $h_{pol, st1}$. These values have to be rectified:

$$u_{2,corr} = \sqrt{\frac{h_{pol,C}}{\mu_y \cdot Z_{st}}}$$  
[M/s]

$$\rho_{2, st1} = \rho_1 \cdot \pi_{st}$$  
[bara]
The discharge temperature of the first stage can be calculated by:

$$T_{2,st1} = T_1 \cdot \left( \frac{p_{2,st1}}{p_1} \right)^{\frac{k-1}{k}} \quad [K]$$

The relevant thermodynamic data of the stage are thus known. Owing to the assumption that every stage of this compressor complies with the stage characteristics shown in Fig. 2, the above step-by-step method can be used for defining the thermodynamical values for the rear stages as well. The density change across the first stage determines the suction volume flow of the next stage downstream, which can now be designed in the prescribed manner:

$$\dot{V}_{1,st2} = \frac{z \cdot R \cdot T_{2,st1} \cdot \rho_1}{p_{2,st1}} \quad [m^3/s]$$

The overall internal power of a compressor is obtained by adding up internal power of the individual stages.

$$P_{int,C} = \sum_{st} P_{int,st} = \sum_{st} \left( \frac{\dot{m} \cdot h_{pol,st}}{\eta_{pol,st}} \right) \quad [W]$$

The overall polytropic efficiency is obtained by adding up the individual stage efficiency multiplying the individual stage power.

$$\eta_{pol,C} = \frac{P_{st1} \cdot \eta_{pol,st1} + P_{st2} \cdot \eta_{pol,st2} + ... + P_{stn} \cdot \eta_{pol,stin}}{P_{int,C}} \quad [-]$$

The overall head of a compressor is obtained by adding up head rise of the individual stages or by inserting the overall conditions into equation (7).

In order to obtain an optimal compressor design for changed compressor conditions, countless loops of iteration are required. Hence special internally developed computer programmes are supporting the applications engineers. The performance of individual stage characteristics can be well predicted analytically which can accelerate the compressor design process substantially but cannot substitute the fastidious interpretation work done by applications engineers. Despite the nowadays available computer programmes, the influence of mach and Reynolds numbers as well as tip clearances and impeller geometry still require comparison with experimentally gained figures. The final design documents are still compared with test results on research compressor which are obtained in decades of experience.

This information is not only needed for the initial dimensioning of compressor stages, it is also essential to assess and predict a revamp of a compressor.
Research to improve aerodynamic efficiency of rotating equipment is a continuous task within the research and development departments at MAN Diesel & Turbo. It is an essential task for any compressor manufacturer to stay competitive in a market of sometimes extremely high requirements on efficiency and optimized specific power consumption.

In some industries with very high power consumption, continuous improving of the machine performance and efficiency is a key factor for business success. That forces operators to look for continuous improvement of their operational expenditures (OPEX). Upgrading the compressors with the latest high-efficiency impellers, blades or sealing technologies may offer such improvement. Whenever extensive reconditioning of a compressor or the replacement of rotors and stator parts is planned, a simultaneous upgrade to the latest technologies may be of benefit.

Due to the low number of impellers utilised in a centrifugal compressor (usually 2 to 8 impellers), removing, replacing or adapting of any impeller has a vast influence on the compressor’s performance. Instead of completely removing impellers, replacement with impeller types of different flow capacity is the common way to adapt the compressor to a new demand. In order to not disturb the matching of the remaining stages, a new impeller in conjunction with the corresponding diffuser should produce a pressure rise which provides a similar suction volume to the next impeller downstream. The effect of this measure is identical to the one obtained by varying the speed.

Some general revamp options:
More flow, an increase in pressure ratio or lower molecular weight cause higher power demand. These cases mostly require an investigation of the complete compressor train. In general, following revamp options can be considered:
- Speed increase or speed decrease
- Change of stationary parts
- Replacement or modification of impellers
- Re-defining of the number of stages

Older compressors (built before the 1980’s) can be revamped with new impellers and blades to improve the efficiency. This is mainly asked for motor driven compressors. Following revamps can be taken into account:
- Replacement of impellers
- Removal of impellers

All these modifications can be combined. Every modification is different and specifically designed to meet the customer’s demands. MAN PrimeServ is able to modify also turbomachinery of other OEM’s.
Revamp Process

MAN PrimeServ follows a defined business process for revamp projects. This process includes the main steps outlined below which are followed in most cases:
- Definition of customer requirement
- Feasibility study
- Budget quotation
- Engineering study
- Quotation
- Project execution
- Project implementation

Definition of customer requirement
The customer is requested to clearly define his requirements by submitting a process datasheet or a specification. Clear definition of the requirements is essential to avoid any misunderstanding and to enable MAN PrimeServ to present a revamp proposal in time and in line with our customer’s expectation.

Feasibility study
MAN PrimeServ carries out a feasibility study based on the received information. This study includes collection and review of the known machine history, design data and applicable limits.

A thermodynamic concept is worked out. The preliminary impeller and diffuser layout as well as the speed are defined. In close co-operation with mechanical engineers the proposed layout is checked for feasibility and corrected if and as required.

The agreed preliminary layout is used to define the basic scope of the required modification to the existing unit.

Budget quotation
Based on the findings and conclusions of the feasibility study a cost and delivery time estimate is worked out and discussed with the customer.

Engineering study
An engineering study serves to further adjust the basic layout and modifications proposed with the feasibility study and to investigate effects and consequences in more detail. The benefit of the engineering study is the fact that the scope of modification, the impact on auxiliary systems and controls and if required also the approx. downtime for implementation are defined. Such study considerably reduces the risk of encountering unwanted surprises for both the customer and MAN PrimeServ.

The thermodynamic layout is finalised by the MAN PrimeServ engineering department, taking comments by the customer on the basic layout into account. Possible impacts on thrust forces, journal and axial bearings as well as couplings are investigated. A draft version of the compressor cross sectional drawing for the new layout is usually developed. This draft is also used by the rotor dynamic specialists to check the lateral behaviour of the modified rotor.

System engineers check the impact of the proposed modification on the auxiliary systems such as lube oil, seal oil, dry gas seal or condensate release. The control systems, especially the anti-surge control equipment are reviewed to identify required adaptations. An upgrade of the control systems may also be proposed as part of a major compressor revamp.

Final datasheets and performance curves are developed. The applicable performance guarantees will be provided with the quotation if requested.

The results and findings of the study as well as any recommendations are summarised in an engineering study report.
Quotation
The results of the engineering study are usually detailed enough to define the scope of work and supply sufficiently to permit submission of a firm quotation for delivery of engineering services and hardware required for the project. Both documents, the quotation and the engineering study report form the basis for further discussion and clarification in co-operation with the customer. It is essential at this stage to come to a common understanding of all aspects and requirements of the project.

Project execution
It is essential for a smooth and successful start of the project execution that the customer purchase order is accompanied by clear and final project specifications. The basic layout proposed with the engineering study is finally adjusted if and as required to meet the specifications.

Project implementation
Although of major interest and most likely the most important step from a production point of view only very few important items are mentioned here. The reason is that site work depends strongly on the particularities of an installation, the scope of work, accessibility, crane capacities etc.
As common important items for all installations a good planning of the work and human resources, availability of material and spare parts as well as completeness and good condition of the tools are required.
Example
Revamping of a FPSO

Plant description
The floating production, storage and offloading (FPSO) platform was built in 1997 equipped with two compressor trains. Each train consists of an electric motor drive, gearbox and two barrel type centrifugal compressors.
The reason for revamp in 2005 was to increase the production capacity in order to maintain production rates from existing wells. The target production rate was 90 MMSCFD for the Low Pressure (LP) Compressor and 85 MMSCFD for the High Pressure (HP) Compressor. The current compressors were not capable to produce the required gas production rates. LP as well as HP compressors need to be revamped to the new requirements.

Operation Point

<table>
<thead>
<tr>
<th>Medium</th>
<th>LP</th>
<th>HP</th>
<th>LP</th>
<th>HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molecular weight [kg/kmol]</td>
<td>25.45</td>
<td>26.91</td>
<td>21.07</td>
<td>20.89</td>
</tr>
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<table>
<thead>
<tr>
<th>Inlet condition</th>
<th>Original 1997</th>
<th>After revamp 2005</th>
</tr>
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<tbody>
<tr>
<td>Volume flow [MMSCFD]</td>
<td>70.5</td>
<td>90</td>
</tr>
<tr>
<td>Weight flow [kg/h]</td>
<td>89,530</td>
<td>94,361</td>
</tr>
<tr>
<td>Volume flow [m³/h]</td>
<td>5,144</td>
<td>6,214</td>
</tr>
<tr>
<td>Pressure p₁ [bara]</td>
<td>15.8</td>
<td>17</td>
</tr>
<tr>
<td>Temperature T₁ [°C]</td>
<td>26.8</td>
<td>25.7</td>
</tr>
</tbody>
</table>

| Discharge condition | | |
|---------------------|-------------------|
| Pressure p₂ [bara] | 66 | 66.4 |
| Temperature T₂ [°C] | 144 | 144 |

* MMSCFD – Million standard cubic feet per day based on standard conditions at 60 °F and 14.7 psia
Uprating study

At first an engineering study was done in order to investigate effects and consequences in more detail.

Our investigations showed that the LP compressor requires a complete new rotor and internal parts. The polytropic head increases due to the lower molecular weight. In order to handle this increase the diameter of the impeller was designed with a larger diameter which results in a higher circumferential speed. A higher circumferential speed and larger diameter means that the higher volume flowrate can be handled with the new design since the flow coefficient remains nearly the same.

The HP compressor requires a complete new rotor and internal parts as well, for the same reason as the LP compressor. The selected impeller sizes are the maximum dimensions that can be built into the existing casing frame size.

The power consumption for the specified operating conditions is higher compared to the original design. Actually the power rating of the motor and gearbox is exceeded.

The number of stages for the LP and HP compressor remain the same.

Modification required

The modifications required to adapt the compressor to the new process requirements are done by replacing the entire compressor cartridge. This decision was made for the reason that the downtime should be minimum.

Basically a new rotor with different dimensions and types of impellers is required. Adaptations and modifications to the inner casing and the internal parts such as channel walls and diffusers are required due to the new rotor design.

Conclusion

The modified compressor was commissioned in 2006. An on site performance test was done, the compressor run as predicted.