PERFORMANCE PREDICTION OF CENTRIFUGAL COMPRESSORS DURING THE CONCEPTUAL ENGINEERING PHASE

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ABSTRACT

This paper presents a method for performance prediction of a potential centrifugal compressor, based on the process data at the design point. The procedure is based on a study of the design practice for a number of vendors for the North Sea hydrocarbon processing industry. The study shows that today's compressor vendors tend to follow the classic design rules developed in the early sixties. These design rules can be applied on the process data from a plant simulation to create an imaginary compressor. A mean line prediction method is used to predict the off-design performance over the total operating range of the compressor. A successful prediction depends on the finally chosen compressor being well designed for the given operating point. The procedure, in the form of PC-based programs, has been applied in conceptual studies and modification studies of off-shore compression plants.

NOMENCLATURE

A  flow area
a  ambient
b  blade angle
β  flow angle
Δ  finite difference symbol
y  heat capacity ratio
η  efficiency
μ  dynamic viscosity
ψ  load coefficient
φ  flow coeff. (v, u )
ρ  density
ω  angular velocity

Subscripts

a  ambient
cool  cool coolant
c  compressor
k  critical value
m  meridional
r  radial direction
u  tangential direction
z  axial direction
d  design
1  inlet
2  outlet

1. INTRODUCTION

During the different engineering phases there is a continuous need for determining the operational behavior of projected compression systems. In the later stages of an engineering project a vendor will be selected, and can to some extent support the necessary need for information. In the early conceptual phase this support is not available, and only very crude "rules of thumb" are used to answer questions concerning the behavior of a potential compressor. As a result a concept is selected on a very weak basis and will often have to be

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reworked during the later stages of the project, a very expensive and time consuming process and for the engineers involved, a very frustrating work. The final result is often a sub-optimal concept.

The purpose of this paper is to present a method based on classic design theories that can with high degree of confidence be used to predict the performance of a potential compressor over a wide range of operating conditions, before any vendor data are available.

2. DESIGN AND OFF-DESIGN CALCULATIONS

The aim is to foresee the behavior of a potential compressor, at a stage where compressor performance only exists in the form of a process requirement. The problem is two-fold:

1) The design process; create a geometry such that the design requirements are fulfilled.

2) Off-design; predict the operating range of the imaginary compressor for any given gas composition and inlet condition.

The relationship between the design process and the off-design simulation is shown in Figure 1.

Figure 1 The geometry that is created in the design-process forms the basis for the off-design simulation

2. SIMULATION MODEL

A suitable model of the compressor must be developed before the design and off-design procedures can be determined. The operational behavior of a compressor is too dependent on the actual design to make a generalized compressor map to describe any compressor. This means that a deducive method based on physical laws should form the basis for the compressor model.

The simplest of the deducive simulation methods is the quasi-2-dimensional mean line prediction method. Each impeller forms a control volume which has to satisfy conservation of mass, momentum and energy. Viscous effects are included by the use of empirical correlations.

The flow model for one impeller is determined by the following equations:

\[ \Delta H = \Delta (UV_p) \]  
\[ \dot{m} = \alpha V_p \]  
\[ \eta_p = f(\text{geometry,flow condition}) \]

The overall performance of a multi-impeller compressor is determined by calculating sequentially the impellers in the flow direction.

The performance of an impeller is thus uniquely defined by:

1) Gas data:
   - Inlet mass flow
   - Inlet pressure and temperature
   - Gas composition or gas physical properties

2) Main dimensions:
   - Inlet/outlet diameter
   - Inlet/outlet flow area
   - Inlet/outlet flow angle
   - Rotational speed
   - Polytropic efficiency

Since the final design of the "compressor-to-be-installed" is basically unknown, a more complete model of the imaginary compressor will not necessarily improve the ability of the simulation model to predict the behavior of the finally chosen compressor.

There is though, one effect which must be treated with high degree of precision. Variations in compressibility factor, Z, have a significant effect on the polytropic exponent and the relation between pressure and temperature throughout the compression process. It is absolutely required to have an accurate physical properties model. The author has positive experience with a Soave-Redlich-Kwong (SRK) model from Calsep A/S (1992).
3. THE DESIGN PROCEDURE

The basis for the design is the requirements established in a process plant simulation. For the given gas composition and mass flow, design a compressor such that given outlet pressure is reached.

3.1 Number of impellers

This method has been developed for hydrocarbon processing plant, and the API standard which sets a limit on the maximum 3050 m head per impeller, will be used for determining the number of impellers assuming the same head/stage throughout the compressor.

3.2 Selection of Diameter and Design Speed

Since the purpose of the system is to predict the behavior of a potential compressor before any vendor data is available, a main issue in the design process, is to make sure that the imaginary compressor gets a geometry which is very close to what a vendor would select for the same task.

Two parameters which to a large extent characterize the impeller design is the diameter and the design speed. The selection of these parameters are closely linked to the impeller head and flow. These properties can be combined in two dimensionless design parameters, specific speed, \( n_s \), and specific diameter, \( d_s \).

**Specific speed:**

\[
 n_s = \frac{\omega \sqrt{Q_1}}{\sqrt[3/4]{Y_{ad}}} \quad (4)
\]

**Specific diameter:**

\[
 d_s = \frac{D \cdot Y_{ad}^{1/4}}{\sqrt[3/4]{Q_1}} \quad (5)
\]

where

- \( \omega \): angular speed (rad/s)
- \( D \): impeller diameter (m)
- \( Q_1 \): volumetric inlet flow (m\(^3\)/s)
- \( Y_{ad} \): adiabatic head (J/kg)

Baljé (1962) showed how the performance potential of an impeller design depended on these two parameters. There exists a band of acceptable values of \( n_s \) and \( d_s \). Examination of a number of compressor designs for hydrocarbon processing, shows that the design rules recommended by Baljé are still in use by vendors.

![Figure 2](image-url) Specific speed and specific diameter is distributed along a narrow band for conventional centrifugal compressors.

Most of the compressors examined were of the constant diameter type, which will imply a lower than optimum specific diameter for the high pressure impellers.

The fact that the vendors still follow the classic design rules is the reason that one can succeed in predicting the performance of a potential compressor based only on process data.

If a constant diameter design is selected, one can assume a specific speed between 0.7 and 0.8 and a specific diameter between 3.5 and 4.0 for the first impeller, and thereby determining the absolute speed and diameter for the machine. One should be aware of the relationships between, the specific speed, the specific diameter and the load coefficient:

**Load coefficient:**

\[
 \psi = \frac{Y_{ad}}{\frac{1}{2} U_2^2} = \frac{8}{(n_s d_s)^2} \quad (6)
\]

The load coefficient of a well designed impeller is typically around 1.0. The constant load coefficient determines the shape of the curve formed by the points in Figure 2.

3.2 The Impeller Inlet conditions

The performance of a centrifugal impeller is to a large extent determined by velocity triangle at the impeller outlet. The contribution of inlet swirl is minor, due to the lower rotational speed. For a conventional compressor...
it is reasonable to assume a swirl free inlet. The inlet geometry can be determined by assuming an inlet Mach number and hub/tip ratio.

3.3 Selection of Flow Angles and Impeller Width.

The only remaining design parameter is the outlet flow angle. This can be found by assuming a outlet flow coefficient. A diagram by Eckert and Schnell (1961) indicates that an optimum outlet flow coefficient should be around 0.27 giving an outlet flow angle around 53° for a high efficiency impeller with a load coefficient around 1.0. The determination of the flow angle also determines the flow area and, hence, the impeller width.

3.4 Prediction of Design Efficiency

The efficiency of each impeller is often difficult to predict. According to Balje the efficiency potential for a well designed impeller at high specific speed is around 0.85. At lower specific speed the impeller efficiency (at design) can be as low 0.5. A typical overall polytropic efficiency for a well designed hydrocarbon compressor is around 0.78.

The result of the design process is the compressor geometry described by its main dimensions of the individual impellers.

4. OFF-DESIGN SIMULATION

The purpose of the off-design simulation is to examine how a compressor performs under varying operational conditions, changing gas compositions or varying inlet temperature or pressure. The problem faced in the design process is now reversed, the geometry is now known and compressor performance is unknown. The flow model applied in the design process remains unchanged, but the algorithm for solving the problem is completely different.

The equations (1), (2) and (3) in the design process forms the basis for the off-design process but the prediction of the off-design efficiency is far more complex and uncertain.

4.1 Prediction of Off-Design Efficiency

It is not the purpose of this paper to review different loss correlations. The experience of the author is that unless very detailed geometrical information is available, one should apply a very simple loss correlation procedure. The off-design efficiency is thus assumed to be a pure function of incidence effects. As a measure of the incidence, the deviation of the inlet flow coefficient will be used. A polynomial relation (7) has been tuned to fit typical hydrocarbon compressors.

\[
\eta_p = \eta_{p,0} + a \left( \frac{\phi_r}{\phi_{r,0}} \right)^2
\]

where:

- \( a = 0.4, \phi < 1.0 \)
- \( a = 0.5, \phi > 1.0 \)

For larger variations in Reynolds number (more than a factor 2), a Reynolds number correction should be made. More details on Reynolds number correction can be found in paper by Casey 1984 and Bakken 1989.

4.2 Range Considerations

The operating range of a centrifugal compressor is limited by surge and "stone wall", or choked conditions.

Surge is a condition where the inertia forces of the flow no longer manage to counteract the pressure gradient over the compressor, and as a result the flow reverses. Stall is on the other hand an instability on a blade channel level caused by boundary layer separation creating a zero-lift situation. In the simulation model it will be assumed that surge will occur at the point where both the inducer and the diffuser stall.

Inducer stall is assumed to be initiated when the incidence angle reaches a critical value, and diffuser stall is assumed to be initiated when the diffuser inlet angle reaches a maximum value (measured from radial axis). Studies of 28 compressors for hydro carbon processing indicates that a good prediction of surge is found if a critical incidence angle of around 9° and a critical diffuser inlet angle of 72° are used.

The maximum flow limit or "stone wall" is reached when the off-design losses causes a rapid drop in the head. For a high Mach number machine, the off-design losses can be amplified by choke losses. In this simulation model "stone wall" is identified when an impeller reaches a minimum polytropic efficiency.

4.3 Results of Off-Design Simulation

The results of these off-design simulations are presented in the form of compressor maps, where the performance parameters are shown as function of inlet mass flow or volumetric flow with rotational speed as a free parameter.

Figure 3 shows a typical example where the discharge pressure is shown as a function of inlet volumetric flow for different speeds. In order to verify the simulation model, test results have been plotted. A good agreement between test and simulation can be observed.
In process plant design it is often important to fix the compressor outlet pressure. Figure 4 shows a graphic presentation where a varying inlet pressure is simulated for a constant discharge pressure.

**Figure 4** Inlet pressure as function of volumetric flow for constant discharge pressure

A more important issue in the design process, is the selection of the design point. There are in some cases reasons to select the design point for the compressor which is different from the design point of the process. A deviation in the design volumetric flow will shift the operating range of the compressor similarly.

In the regions close to surge and stonewall, the off-design efficiency is more sensitive to changes in flow than in the regions around the design point. These regions are thus more uncertain than the rest of the compressor map.

5.1 Verifications

The model has been verified against different vendors performance data in approximately 30 cases with a wide spectra of gas components and capacities.

5.2 Applications

The model was originally developed for conceptual studies, but has in addition to conceptual studies also found applications in modification/revamping studies, optimization of antisurge controllers and recalculation of test results (replacement for PTC-10).
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