Abstract
This paper presents the concept of a centrifugal compressor stage with so-called tandem impeller blading. This type of blading is expected to increase the integral parameters (total pressure ratio, isentropic efficiency, mass flow parameter) of a conventional stage (baseline stage) without modifying its outer dimensions, by interrupting the impeller blades transversely and designing the inducer part anew, treating it as a separate vane. This should lead to a reduction of the size of the low-momentum fluid zone at the impeller exit.

In the paper, two different design concepts of tandem blading are presented. The first concept splits the main blade cascade into two separate cascades after the inducer and rotates them against each other by such an angle in order to achieve the best possible direction of the wake flowing downstream of the inducer part of the blading.

The second concept concerns a redesign of the inlet parts of every second splitter blade in order to obtain equal portions of the mass flow in each exducer channel.

In the paper, the assumed effects of these modifications of the designed impeller are presented and compared with the results obtained by CFD calculation. These results are also compared with the parameters of the baseline stage.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>k</td>
<td>Turbulent kinetic energy</td>
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<tr>
<td>m</td>
<td>Mass flow</td>
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<td>p</td>
<td>Pressure</td>
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<td>Q</td>
<td>Mass flow parameter</td>
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<td>T</td>
<td>Temperature</td>
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<td>y⁺</td>
<td>Dimensionless distance from the wall</td>
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<td>ε</td>
<td>Turbulent eddy dissipation</td>
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<tr>
<td>ηᵢₙ</td>
<td>Stage isentropic efficiency</td>
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<td>π</td>
<td>Stage total pressure ratio</td>
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<tr>
<td>ω</td>
<td>Specific rate of turbulent kinetic energy dissipation</td>
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Subscripts

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
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<tr>
<td>in</td>
<td>Stage inlet</td>
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<tr>
<td>tot</td>
<td>Total pressure/temperature</td>
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Introduction
Centrifugal compressors are often used in small-sized turboprop, turboshaft and propfan engines as the last compressor stages. Their application in these types of engines brings several advantages. A centrifugal compressor stage usually achieves a substantially higher pressure ratio compared with an axial stage with the same mass flow rate. Therefore, a centrifugal compressor covers much less axial space than four or five axial stages which have the same pressure ratio. Moreover, small centrifugal compressor stages often achieve a greater isentropic efficiency than axial stages when used as last stages. This is the case of smaller engines. Their lower mass flow rates, together with decreasing chord lengths in the last axial compressor stages, lead to lower values of Reynolds number. This means that viscous forces play a more significant role in the flow field, thus reducing the efficiency of such an axial stage. The use of a centrifugal stage, on the other hand, does not affect the Reynolds number negatively.

Up till now, centrifugal compressors have been developed into a relatively mature state. Based on previous experience of the authors of the paper in
the development of centrifugal compressors made in collaboration with the Aerospace Research and Test Establishment (VZLU), Walter Aero Engines Company and Howden CKD Compressors (all based in Prague), the concept of a new innovative design of a centrifugal compressor stage is presented in order to improve its performance characteristics even more.

One of the most important factors which influence the overall characteristics of a centrifugal compressor is the three-dimensional nature of the flow with a strong action of transversal pressure gradients and turbulence. Since the impeller channel is relatively long, a significant influence of the growing boundary layers is present. The chief idea of the new design approach is focused on such an arrangement of the existing geometry of the impeller blades which will simultaneously reduce aerodynamic losses and improve the flow stability.

The new compressor stage design maintains the shape of the impeller meridional section and the number of both main blades and splitter blades as well as the outlet angle of the splitter blades. It only affects the channel length by splitting the main blades transversely and thus interrupting the growth of the boundary layer. This leads to a reduction of the size of the wake region at the impeller outlet which positively affects the diffuser flow.

This new approach to the design of centrifugal compressors, presented in this paper, will allow certain types of engines to work with increased efficiency and thus improve their fuel consumption.

Moreover this design concept can be applied to existing conventional centrifugal compressor stages by redesigning the impeller blading while keeping its outer dimensions. Thus the existing impeller can simply be replaced. This is, actually, our case.

The compressor stage which forms a baseline to our tandem-bladed stages consists of an impeller with axial inlet and radial outlet with a backsweep of 30° and a diameter of 360 mm. The impeller has 16 main blades and 16 splitter blades placed halfway across the exducer channels. A pipe diffuser with 15 channels is placed downstream of the impeller. The design point total pressure ratio of the stage is 4.143, the mass flow rate is 7.5505 kg.s⁻¹ at 25,000 rpm. The measured design point isentropic efficiency is 80.8%.

**Design Concept No.1**

The first design concept splits the impeller blades transversely into two parts, with the splitting plane lying at the leading edges of the splitter blades. The inducer part of the blades is thus formed by an axial blade vane while the exducer part consists of the remaining radial parts of the main blades, and the existing splitter blades.

This concept was already examined 10 years ago by the Institute of Aerospace Engineering, Faculty of Mechanical Engineering, CTU in Prague, and presented at the ISABE 2005 conference in München, Germany [8].
The baseline conventional compressor stage used in 2005, different from the one used today, had originally been developed for the M602 turboprop engine by Walter Engines (now GE Aviation Czech) and VZLÚ (Aerospace Research and Test Establishment). The compressor of this two-shaft engine consisted of two subsequent centrifugal stages (a low-pressure and a high-pressure compressor).

The fundamental idea of this concept of tandem blading is to control the formation of boundary layers on the impeller blade surfaces. After the inducer parts of the blades, their growth is interrupted and the resulting wake flow is directed into the middle of the the exducer channel (Figure 4). This should result in a reduction of the size of the wake region at the impeller exit which, in turn, leads to an increase of the compressor stage isentropic efficiency. This assumption has already been confirmed by preliminary calculations presented in [8]. These calculations showed that a slightly higher isentropic efficiency was achieved by the tandem-bladed stage (see Figure 5). The calculations of the relative velocity pattern also showed the positive effect of directing the wake downstream of the inducer part of the blades into the exducer channels (Figure 4).

However, no further steps were made in improving the tandem-bladed impeller geometry. The concept was therefore not optimized to achieve the best possible parameters.

In the present project, the development of this design concept was started anew. After the baseline compressor stage computational model was completed and validated using experimental results.
(described in [9]), the first newly designed tandem-bladed impeller was made using the original blade shapes and just interrupting them after the inducer.

This stage, however, did not achieve satisfactory results. All the integral parameters examined (the total pressure ratio, mass flow parameter, and isentropic efficiency) were significantly lower than the baseline stage had. This was caused by large areas of flow separation which appeared at the splitter blade suction sides and also major shock waves followed by flow separation at the main blade suction sides (see Figure 7). This concept could not function because of the curvature of the inducer blades which was higher compared with usual axial compressor vanes, especially at the blade tips.

Afterwards, when this arrangement was not found to be satisfying, a design process leading towards finding an optimum axial inducer vane was started. The newly designed axial vane consisted of blades with circular camberlines designed according to a method used in the design of axial compressors presented in [7].

Two tandem-bladed stages have been tested so far. They are denoted Stage A (with variations A1, A2, A3) and Stage B. Their geometry and results achieved are described in [9]. Here, only the best results achieved so far by this design concept will be presented.

The computations made until now have shown that the total pressure ratio can be increased by introducing tandem blading. This is evident from the results of computations with stage B using an axial vane with a total pressure ratio of 1.2. Its mass flow parameter is the same as that of the baseline stage. The total pressure ratio is, along the entire performance curve, about 0.2 higher than that of the baseline stage which is a difference of about 5%. This means that by using this stage the overall size of an aeronautical engine can be reduced.

The isentropic efficiency is, in the best case, about 1% higher than that of the baseline stage. Thus, if this stage was used, the fuel consumption of the aeronautical engine would decrease.

This effect, however, strongly depends on the circumferential position of the splitter blades. The effect was only examined in case of Stage A. The computations of Stage B are still underway.

**Design Concept No.2**

The second design concept also involves splitting the impeller blades transversely into two parts, just as in the first case. Moreover, the splitter blades which are placed downstream of the axial vane now consist of two different sets, one half of them have...
a different leading edge stagger angle than the other half (see Figure 6). This arrangement should assure equal mass flows inside the two sets of channels between the splitter blades. When concept No. 1 was used, the ratio of mass flows between the two sets of channels was, for example, 53.4/46.6 percent in Stage A1 design point. So there is a 6.8% difference between the mass flows in the exducer channels.

It is expected that the arrangement with equal mass flows will lead to a more equal distribution of velocity patterns between the two sets of channels at the impeller exit. The papers studied (e.g. [12]) claim that this equal distribution helps to reduce mixing losses inside the diffuser. Moreover, since the total pressure losses are proportional to the second power of mass flow, these losses inside the impeller channels should also be more evenly distributed, and their sum may even be reduced.

Again, the computations were made with two different arrangements of the axial vane. One was designed for a total pressure ratio of 1.1 (Stage C), the other for 1.2 (Stage D). For each of these arrangements, the suitable geometry of exducer channels was found after several iterative steps.

The calculations made so far have not shown any improvement of the integral parameters compared with the baseline stage, the exducer blades of which (incl. splitter blades) had the same camberlines.

The radial velocity pattern at the impeller outlet of Stage D (Figure 14) shows that the distribution of the velocity profiles between the two sets of channels is still rather uneven. The eddy viscosity distribution at 75% span of Stage D impeller...
(Figure 13) also shows a significant unevenness of pressure loss production between the two sets. In the Mach number distribution at 50% span of Stage D impeller (Figure 12), a flow separation followed by a larger wake zone in one set of channels can be seen.

The effect of increasing the axial blade curvature can also be observed with this type of stages. Stage C, since it has an axial vane designed for a total pressure ratio of 1.1, does not even achieve the values of mass flow parameter of the baseline stage, the difference was about the same as in the case of Stage A (approx. 0.07 units). The mass flow parameters achieved by Stage D are the same as in the baseline stage.

However, no improvement of the total pressure ratio has been observed. Both Stage C and Stage D have the same, or slightly smaller, pressure ratios compared with the baseline stage. Their isentropic efficiency is by about 2.0% lower than that of the baseline stage.

Therefore, this design concept has so far not proved to be promising, and requires further optimization in order to meet the expectations presented above.

Conclusions

In this paper, the philosophy of tandem-bladed centrifugal compressor stages was introduced. Two different design concepts were presented and, based on computational results obtained so far, the predictions of their behaviour were compared with their actual performance.

While the tandem-bladed stage with all splitter blades identical (Concept No.1) has shown its potential for improving the integral parameters of the stage while preserving its outer dimensions, the stage with evenly distributed mass flow in splitter blade channels (Concept No.2) has not yet been found to be promising.

However, both concepts are still being investigated in order to find a geometry optimized for the best isentropic efficiency while at least maintaining the same total pressure ratio and the same mass flow parameter.

Figure 14 Contours of radial velocity at Stage D impeller exit
References


