Influence of The Inlet Geometry on Operating Parameters of Modernized Industrial Centrifugal Compressors

Wpływ geometrii wlotu na parametry pracy modernizowanych przemysłowych sprężarek promieniowych

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The paper is devoted to an effect the shape of the inlet in the centrifugal compressor under modernization exerts on its performance parameters and an evaluation of the related risk. The state art knowledge experience of the authors concerning centrifugal compressor inlets, and results of the analysis of flow fields upstream of the first stage of the compressor under modernization are presented.
Keywords: centrifugal compressor, modernization, shape of the inlet

Artykuł analizuje wpływ ukształtowania układu wlotowego modernizowanej sprężarki odśrodkowej na parametry jej pracy oraz na powiązane ryzyka. Omówione są przypadki pochodzące zarówno z literatury, jak i z własnego doświadczenia autorów. W części finalnej przedstawiono wyniki symulacji przepływu przez przykładowy układ dolotowy z zastosowaniem pakietu ANSYS CFX.
Słowa kluczowe: sprężarka odśrodkowa, modernizacja, geometria wlotu

Introduction

Compressors, including the process ones (i.e., operating in chemical installations) are characterized by long service life. In Poland, some of them have already been operating for 40 or even 50 years. Such long service often results in modernizations related to alternations in the technical design of the compressor.

In the literature devoted to this topic (e.g., monograph by Lüdtke [12] or papers by Münger and Näf [14] and Connor [4]), a primary reason to carry out modernization of compressors, both radial and axial, is formulated as a necessity to adapt the compressor to continuous operation under variable performance conditions, that is to say, both the pressure ratio and the flow, as well as suction parameters and a chemical composition of the working gas.

The present paper is devoted to an effect the shape of the inlet in the centrifugal compressor under modernization exerts on its performance parameters and an evaluation of the related risk. A longitudinal cross-section of the six-stage centrifugal compressor, subjected to modernization that consisted in an exchange of the whole rotor into a new one of a different geometry on the basis of [9], is shown in Fig. 1. The casing elements remained unaltered. In this case no significant changes in the inlet can be introduced.

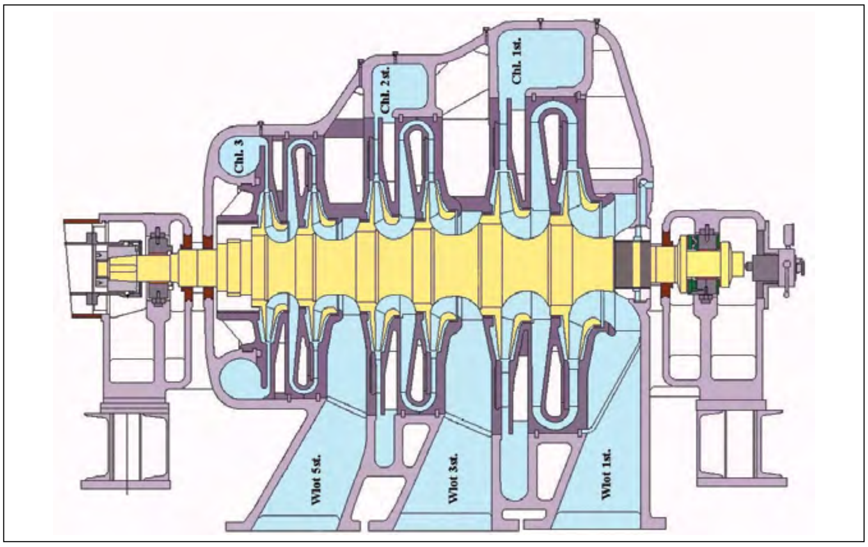

Fig. 1.
Longitudinal cross-section of the centrifugal compressor under modernization, type 6RMY56, manufactured by CKD Praha (Kryłłowicz et al. [9])
Rys.1 Przekrój podłużny modernizowanej sprężarki promieniowej, typ 6RMY56, produkcja CKD Praha (Kryłłowicz i inni [9])

State-of-the-art knowledge in the field of an interaction of inlets in centrifugal compressors

First overall experimental investigations on an effect of nonuniformity in inlet

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parameters on the compressor performance were conducted by Ariga and Kasai in 1982 [1]. The experiment was made on a low-rotational speed compressor with a vaneless diffuser, where disturbances in the working medium upstream the inlet were induced in an artificial way.

A comparison to the results for an undisturbed flow showed that the nonuniform inlet velocity profile resulted in a considerable decrease in the impeller wheel efficiency, mainly due to a change in the local angle of attack.

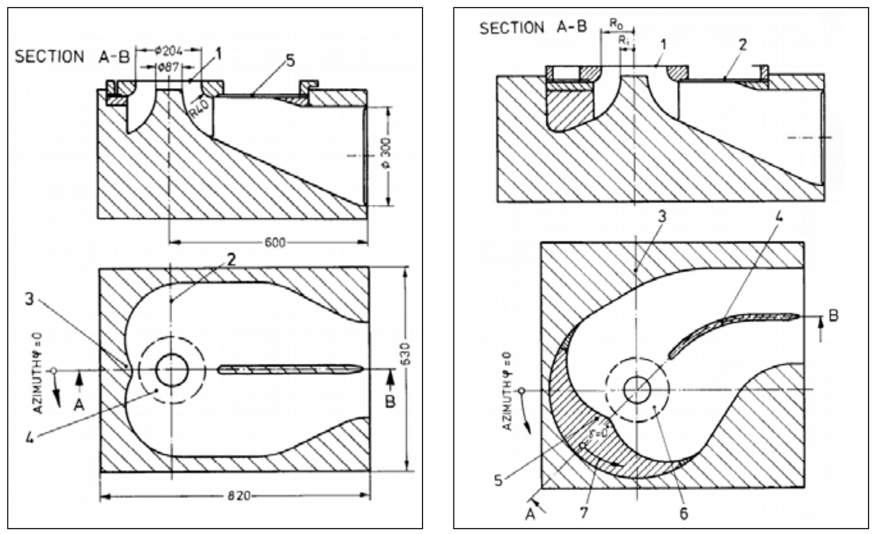
A summary of the investigations on cylindrical inlets to compressor stages with axial-radial wheels is included in the works written by a team headed by Yunbae Kim and Abraham Engeda ([5], [7]). In the first study, both the results of experiments and simulations are included. The second publication is an extension and supplement of the former one, as the commercial TASC-flow code was used (an early version of the present ANSYS CFX package) in the investigations. An impeller wheel of the diameter equal to 5.64 inch (approx. 143.25 mm) and the maximal rotational speed n = 42 900 rpm, and thus the circumferential velocity of the wheel u_2 = 321.77 m/s, was applied in the experiment. Two variants of the inlet were investigated: a "straight inlet" and a "bent inlet". In the figures below, a part of the measurement results is presented: the relative pressure coefficient Ψ/Ψ_{DP} and the relative efficiency η/η_{DP} versus the relative flow coefficient Φ/Φ_{DP}. The symbol DP denotes here the Design Point.

The experimental results [7] showed that the inlet curvature resulted in a significant decrease in the pressure ratio and efficiency of the stage for all three rotational speeds. Thus, a considerable effect of inlet distortions on the stage performance was proven.

Lüdtke [11] conducted comparative tests of two types of radial and axial inlets (Fig. 3), where in both cases different kinds of the inlet pipeline geometry were applied: a straight pipeline of the length equal to 10D and a pipeline with a bent located at the distance of 3D from the casing inlet flange. Thus, a test schedule was very similar to the one used in the experiments conducted by Kim and Engeda. An example of the 3D scheme of a flow pattern for the radial inlet can be seen in Fig.4.

It can be concluded from the results of those investigations that although the symmetrical radial inlet, widely used in compressors due to its simplicity, allows for achieving the stage polytropic efficiency of 84%, however some circumferential nonuniformities in the velocity field occur. It results in deflections from the ideal inlet angle. The bent inlet pipeline deteriorates the situation even more, which was to be expected. On the other hand, the diagonal inlet requires a very careful choice of the geometry to obtain the same performance of the stage as in the former case.

Similar results were obtained by Flathers and Bache [6] and by Koch, Chow and Hutchinson [10]. Both teams of researchers conducted independently experiments that consisted in measurements of velocity distributions in inlets to multi-stage compressors (the inlets were of a radial type). The inlet models under testing were made in various scales: Flathers' team investigated a 1:2 model, whereas Koch's group tested a 1:4 model. Next, comparative numerical investigations were carried out. That comparison showed a sufficient agreement



1 MEASURING PLANE (IMPELLER EYE, ANNULUS)
2 PARTING JOINT
3 SHAPED FLOW SPLITTER
4 MEASURING PLANE
5 COVER PLATE (PLEXIGLAS)

1 MEASURING PLANE (IMPELLER EYE,ANNULUS)
2 COVER PLATE (PLEXIGLAS)
3 PARTING JOINT
4 FLOW DIVIDING RIB (ADJUSTABLE)
5 SHAPED FLOW SPLITTER (ADJUSTABLE)
6 MEASURING PLANE
7 SHAPED FLOW SPLITTER ANGLE

Fig. 3.
Topology of the radial and diagonal inlet according to Lüdtke [11]
Rys. 3 Topologia wlotu promieniowego oraz diagonalnego wg Lüdtkego [11]

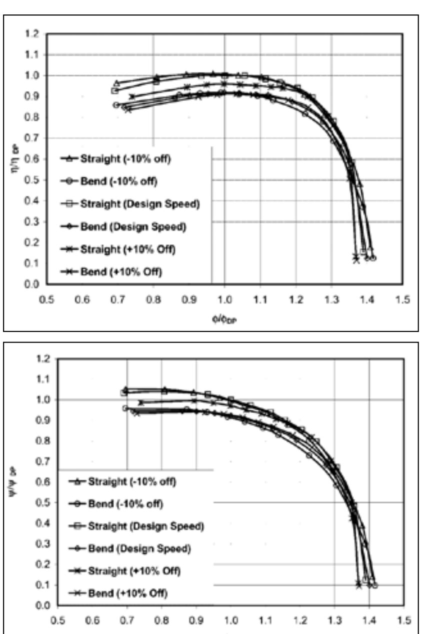


Fig. 2.
Results of measurements by Kim and Engeda: two variants of inlets investigated at three rotational speeds [7]
Rys. 2. Wyniki pomiarów Y. Kima i A. Engedy dwu wariantów dolotu przy trzech prędkościach obrotowych [7]

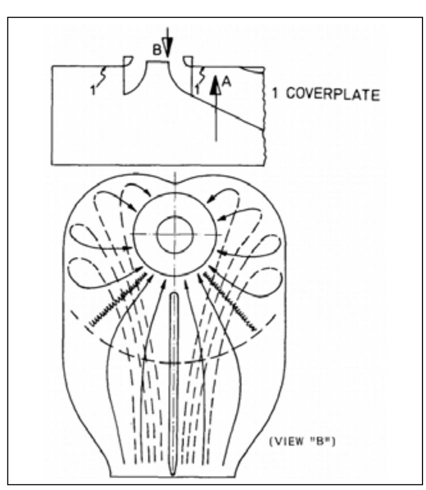


Fig. 4.
Example of a flow pattern scheme in the radial inlet of the centrifugal compressor according to Lüdtke [11]
Rys. 4 Przykładowy schemat linii prądu we wlocie promieniowym sprężarki promieniowej według Luedtkego [11]

between the experiment and the CFD calculations. Moreover, Flathers investigated an inlet with a straight suction pipeline only, whereas Koch, like Kim [7], tested three configurations of the pipeline: straight, bent forward and backward. On the basis of analysis of the results obtained, Koch found that:

- losses at the inlet were a function of the Mach number,
- flow patterns were independent of the Mach number,
- nonuniformities in the velocity fields due to the pipeline bent were eliminated in the inlet chamber – they did not affect the velocity profile upstream the impeller wheel.

Experience concerning inlets to centrifugal compressors gathered at the Institute of Turbomachinery, TUL Major Headings

The designs of inlets to industrial centrifugal compressors developed at the Institute of Turbomachinery, TUL, were based on the Russian literature, in particular Nikitin [15]. Satisfactory results were attained, which is testified by the results of measurements carried out on a 10 P8 compressor prototype [17]. Some selected results of these tests are shown in Fig. 5.

number range under investigation. Figure 5 shows a radial distribution of the axial component and the flow angle αO averaged for both probes. It turned out that the scattering of the axial component c0, max to c0, min did not exceed the value of 1.08 and the maximal swirl was 6 degrees. The

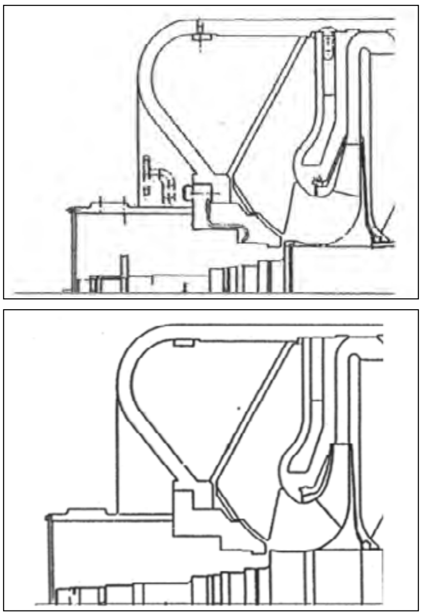


Fig. 6.
Inlet part of the propylene refrigerating compressor before and after modernization
Rys. 6 Część wlotowa chłodniczej sprężarki propylenu przed i po modernizacji

one and a new 3D is depicted in Fig.6. An aerodynamic structure of the 3D wheel was developed on the basis of literature recommendations (van den Braembussche [3] and Lüdtke [11]). A CFD technique was applied, i.e., the k-ε turbulence model was used, the computational grid covering a part of the inlet, the wheel and the return bend had 6.24 million nodes.

The results of acceptance tests turned out to be disappointing, which can be seen clearly in Fig. 7. A very low efficiency was obtained (the internal stage efficiency was $\eta i = 0.67$) and the pressure ratio was even lower than the one assumed in the pessimistic variant. The parameters of the remaining part of the compressor, including the second stage with a 3D impeller wheel, were satisfactory.

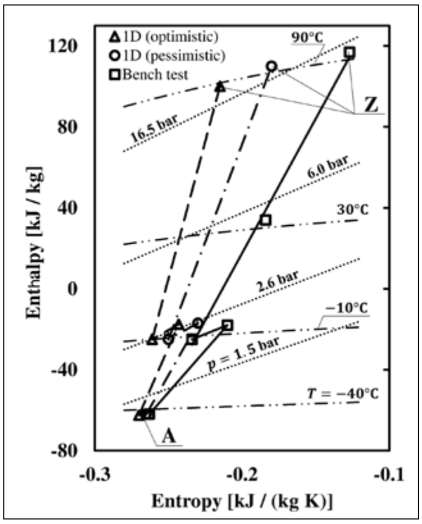


Fig. 7.
Results of acceptance tests of the polypropylene compressor (according to [8])
Rys. 7 Wyniki badań odbiorczych sprężarki propylenu (wg [8])

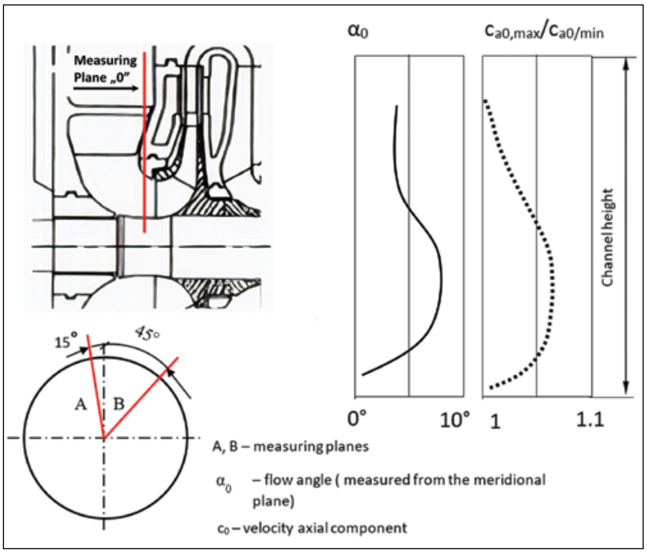


Fig. 5.
Results of measurements of the working medium velocity on the "0" inlet plane of the first stage wheel in the 10 P8 compressor (according to [16])
Rys. 5 Wyniki pomiarów prędkości czynnika w powierzchni wlotowej „0" koła wirnikowego pierwszego stopnia sprężarki 10 P8 (na podstawie [16])

Measurements of the gas velocity profile were conducted in two meridional planes, distanced from the vertical plane by 15°and 45°, correspondingly. A three-hole "cobra-type" probe, which moved vertically, was used. Because of the type of the probe applied, the measurements did not cover the radial component of velocity. The insensitivity of the probe to the inflow direction checked during calibration was equal to +/ − 3 degrees for the Mach

flow channel upstream of the edge of the wheel blade is confusor-like (8% medium acceleration), which additionally provides good conditions for an inflow on the blades.

In the propylene refrigerating compressor (see [8] and [16]) under modernization, a 3D-type wheel of the first stage was used, which was justified by a high flow coefficient $\phi = 0.081$. A comparison of the two design variants, i.e., the original 2D

While designing the 3D wheel, it was assumed that the working medium acceleration between control plane 0 and 1n would be very low, i.e., equal to 1.03 (see

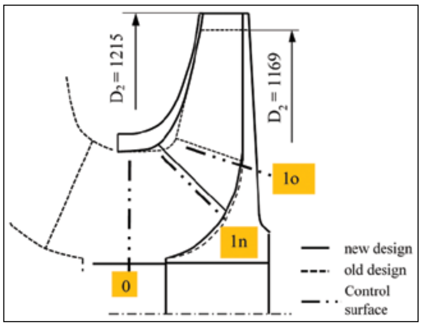


Fig. 8
Comparison of the impeller wheel geometry in the propylene compressor in the original and modernized version
Rys. 8 Porównanie geometrii koła wirnikowego sprężarki propylenu w wersji oryginalnej i zmodernizowanej

Fig.8). It turned out, however, that it was not enough. In the old design, that acceleration was 1.14 (between control planes 0 and 1o). Consequently, when a uniform velocity profile was assumed at the inlet (on plane 0 in Fig. 8), a different angle of attack along the blade leading edge was obtained. As a result, the wheel efficiency was very low.

To sum up, a failure resulted from the fact that actual boundary conditions were not accounted for in the CFD computations.

Analysis of the flow field upstream of the first stage in the compressor under modernization

The modernization of the 6RMY56 compressor was carried out on the basis of the procedure used successfully at the Institute of Turbomachinery, TUL (Kryłłowicz [8]). This procedure involves three stages, namely:

- tests of the compressor and a concept of its modernization,
- 1D calculations of a new flow system,
- verification calculations based on the CFD method.

In the case of the compressor under modernization, it was not possible to change the inlet geometry as the outer walls were defined by cast walls of the casing. The modernized impeller wheel of the first stage had the same outer diameter D2 = 562 mm, but differed significantly as regards the blading geometry. Blade angles, blade profile and thickness were altered. Table 1 presents the most important parameters of the original and modernized first stage wheel:

Table 1. Important parameters of the original and modernized first stage

Value	Unit	Original	Modified
Outer diameter D_2	mm	562	562
Blade exit angle β_2^*	°	45	60
Blade inlet angle	°	27	30
1-st stage pressure ratio		1.62	1.67*/1.72**
1-st section pressure ratio		2.485	2.59 / 2.69**

* Design
** test results

To minimize an error risk, simulation computations of the whole inlet were conducted, starting from a suction throttle valve up to the inlet plane of the first impeller wheel (see Fig. 9). It was necessary to consider the valve as it generates a strong vortex of the Rankine type.

The inlet volute geometry was obtained from the technical documentation, whereas the geometry of the inlet pipeline was measured directly on the compressor.

The main goals of the CFD computations (ANSYS CFX code) of the inlet were:

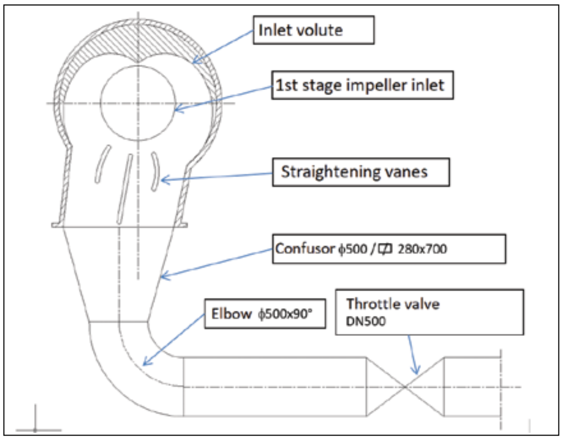


Fig. 9
Inlet to the CKD compressor first stage with suction pipeline elements (according to [9])
Rys. 9 Kanał wlotowy do pierwszego stopnia sprężarki CKD wraz z elementami rurociągu ssawnego (wg [9])

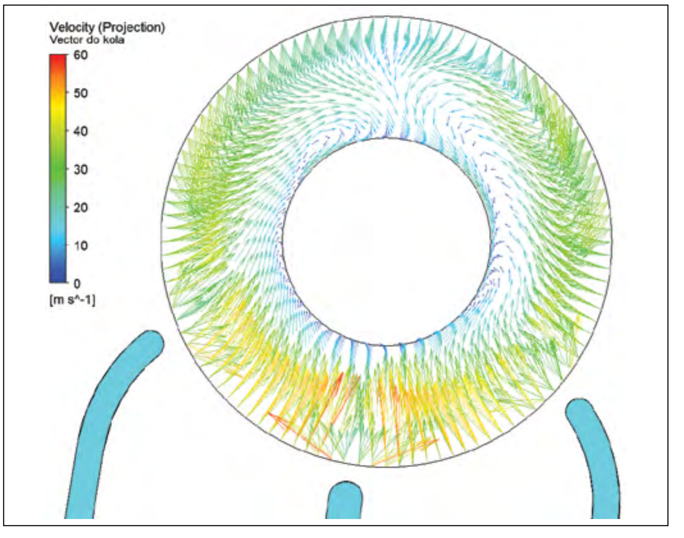


Fig. 10
Distribution of the radial velocity component at the inlet to the first stage of the modernized compressor (according to [9])
Rys. 10 Rozkład składowej promieniowej prędkości na wlocie do pierwszego stopnia modernizowanej sprężarki (wg [9])

- to check a velocity distribution at the inlet to the impeller wheel and to estimate a magnitude of a possible inlet swirl,
- to determine the boundary conditions at the inlet to the first stage in order to conduct a flow analysis in the whole first compression section.

Figure 10 shows a distribution of the circumferential component at the inlet to the first stage impeller wheel, whereas the next figure presents a distribution of the axial-radial component, obtained as a result of the CFD computations. Mean values of the velocity at the inlet are equal to, respectively:

- circumferential component – 1.03 m/s,
- radial component – 19.0 m/s,
- axial component – 65 m/s.

The test results of the compressor (Błaszczyk et al. [2]) confirmed that the inlet provided good inflow conditions on the blades of the first stage. The compressor (and three identical machines) are operating now in a continuous regime in one of Polish power plants.

Conclusions

Within the last 20-years, thirteen modernizations of centrifugal compressors,

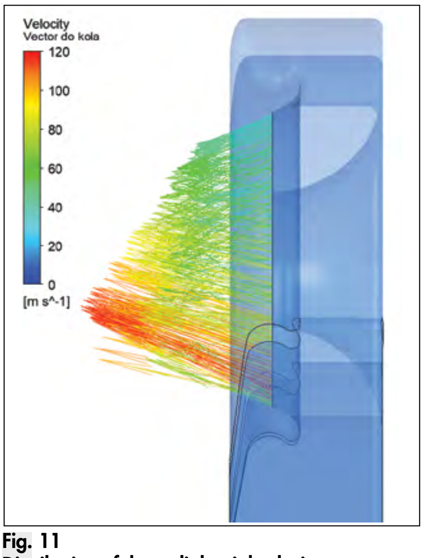


Fig. 11
Distribution of the radial-axial velocity component at the inlet to the modernized compressor (according to [9])
Rys. 11 Rozkład składowej promieniowo-osiowej prędkości na wlocie do modernizowanej sprężarki wg [9]

mostly the process ones, were successfully conducted at the Institute of Turbomachinery, TUL (Kryłłowicz [8]). Both classical 2D impeller wheels and the modern 3D ones have been applied. This set of data has allowed us to generalize the risk of

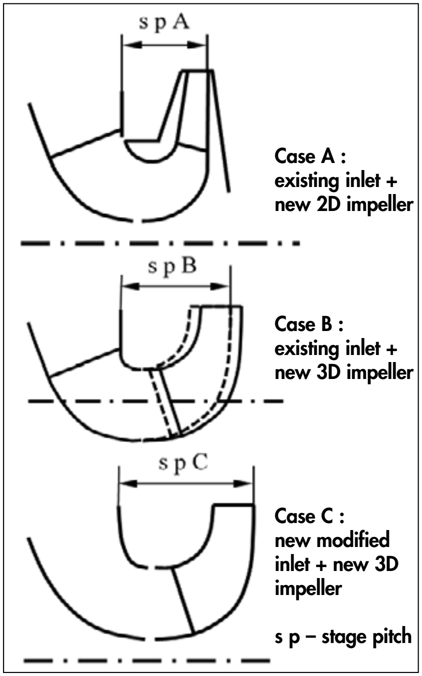


Fig. 12
Three designs of the first stage inlet and impeller wheel of the centrifugal compressor under modernization
Rys. 12 Trzy przypadki konstrukcyjne wlotu oraz koła wirnikowego pierwszego stopnia modernizowanej sprężarki promieniowej

a lack of compatibility of the first stage impeller wheel and the inlet. In Fig.12, schemes of three configurations of the inlet part of the centrifugal compressor under modernization are presented.

Case A refers to an interaction of a new 2D wheel with the existing inlet. As the channel upstream of the inlet edges of blades is a confusor, then a risk of an improper inflow, and thus low performance of the stage, is slight. In case B that refers to the existing inlet and a new impeller wheel, an impact inflow on the blades can occur. Due to a relatively long inlet edge and a low acceleration in the channel upstream of the wheel, a risk of improper operation of the wheel is significant. On the other hand, in case C, we are able to adapt the inlet in such a way as to ensure proper angles of attack along the whole length of the inlet edge.

However, in practice, a possibility to shape arbitrarily the inlet is limited in many instances. Moreover, an application of highly efficient 3D wheels requires a sufficiently high axial stage pitch (marked as sp in Fig. 12). For example, a dashed line is used to represent an outline of the 3D wheel, whose axial position, i.e., the stage pitch sp, results from the dimensions of the existing casing diaphragm. As a matter of fact, an increase in sp enables a rational shaping of the wheel. It means that:

- case A poses no aerodynamic risk,
- case B poses an aerodynamic risk to a serious degree,
- case C poses a limited aerodynamic risk.

Hence, during the modernization of the compressor, an analysis of the compressor inlet geometry should be always conducted (preferable with the CFD method) and a degree of the resulting risk ought to be assessed.

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