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INTEGRATED CONCEPTUAL DESIGN ENVIRONMENT FOR CENTRIFUGAL COMPRESSORS FLOW PATH DESIGN

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ABSTRACT

A new method for centrifugal and mixed-type compressors flow paths design based on a unique integrated conceptual design environment is presented in this article. At the heart of this new method is the translation of proven, integrated design environments that have been successfully used for axial turbomachinery for many years.

This integrated environment is a seamless and swift processing scheme that incorporates stages aerodynamic analysis and preliminary design/sizing based on the one-dimensional method interactive spatial blade profiling, export of blade geometry to CAD and CFD tools, 3D stress and vibration analysis, and finally, flow modeling..

The design process is demonstrated for a centrifugal compressor design utilizing AxSTREAM software.

NOMENCLATURE AND GLOSSARY

Variables

G	mass flow rate
$H_t = \Delta uc_u/u^2$	load factor
R	reaction
D	diameter
P	pressure
T	temperature
PR	pressure ratio
l	blade length
n	speed of rotation
u	tangential velocity
α	flow angle in absolute frame
eff_tt	isentropic(total-to-total) efficiency

Subscripts

0	at the stage inlet, at the beginning of the process
1,2,3	in the 1,2,3 sections
s	meridian
z	axial
tt	total to total

Superscripts

•	stagnation parameter

INTRODUCTION

Turbomachinery flow path creation using an integrated conceptual design environment (IE) allows the designer to shorten the design development process significantly, thereby decreasing engineering costs and improving productivity. It gives an opportunity to review a large number of variants and design parameters to realize optimum results. [1-4].

This article is devoted to describing new a approach for developing turbomachinery design systems, IE components that work for various turbomachinery design platforms, and new subsystem elements for radial turbomachinery conceptual design.

Initially, IE was developed for axial turbines (mainly steam). Later, it was expanded for gas turbines (especially blade cooling calculations) and axial compressors via plug-in modules. The new challenge designers face today is developing mixed flow machinery. These new design system must be flexible and have the capabilities to develop axial, radial and mixed flow machinery at the same IE. As a result of our research, two main requirements were specified for the system: the majority of modules must be compatible with every type of turbomachinery, and specific modules must be able to run simultaneously (axial and radial turbine, axial and radial compressor). It turned out that invariant modules (project data access, graphical display of information, multi-choice calculation and optimization, import/export, etc. possibilities) can work for any platform. Also, we determined that conceptual design modules for centrifugal and mixed flow machines would fit extremely well within the IE structure prototype that worked for axial machinery. Concurrently this IE approach and integration can be extended to blade (impeller) 3D profiling, stress analysis, and 3D Flow analysis.

In the first section of this article, the architecture of a new integrated system for turbomachinery conceptual design is described. Then, the preliminary design procedures of the radial turbomachinery stages are set forth, followed by 1D calculations on design and on off-design mode. The next part presents 3D blade design procedures, 3D potential flow analysis, and stress and vibration analysis. The concluding section of the article brings all components together by presenting examples of centrifugal compressor design.

INTEGRATED CONCEPTUAL DESIGN ENVIRONMENT

An effective system for turbomachinery flow path conceptual design needs to do the following:

- involve a set of design modules necessary for design procedures under one operating platform (an umbrella, per se) that performs initial sizing and optimization, 1D formulations, and builds blade 3D geometric models that are available for final refinement by means of 3D aerodynamic and stress analysis;
- give the ability to automate multivariate and optimization calculations using embedded models;
- provide flexibility in carrying out interactive design scenarios including rollback, versions support, project integrity, etc
- ensure expandability, scalability, and maintainability;

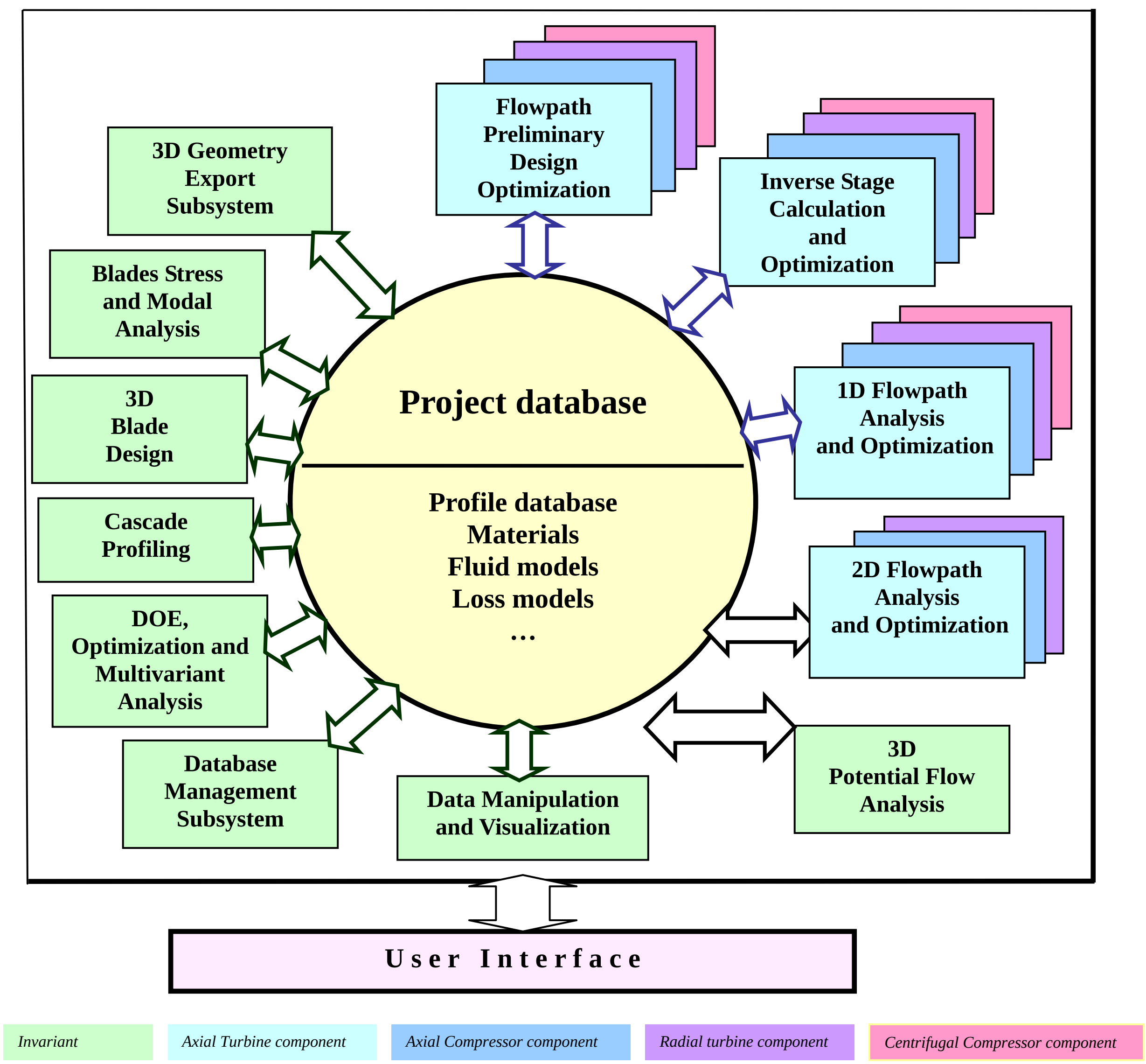


Fig. 1. The architecture of the turbomachinery flowpath integrated conceptual design system

- provides users with convenient mechanisms to input, edit, display data and export data to other systems.

In Fig.1, the architecture of the turbomachinery flow paths integrated conceptual design environment is shown. It can be seen that more than two thirds of the subsystems are invariant (capable of working various platforms while maintaining seamless processing). Thus, after new problem definitions are added into the system by the designer, the program gains total access to existing design subsystems.

CENTRFUGAL COMPRESSOR ANALYSIS

Traditionally, for centrifugal compressor operation analysis for design and off-design points, the verification analysis problem has been performed in 1D formulations.

In verification analysis of a centrifugal compressor, flow is treated as one-dimensional only [6]. Obviously, there is significant 3D flow in a centrifugal compressor. Considering the importance of quick review of design results, a 3D potential flow solver was developed and embedded into the system.

In contrast to 3D viscous flow calculations in a flow path, simplified 1D methods provide a close approximation for experimental data depending on the quality of empirical methods used to determine losses and deviation angles.

Due to the absence of reliable energy loss and flow deviation angles in cascades, there is a significant challenge in developing radial turbomachinery verification calculation algorithms. In practice, [1] the various known construction losses obtained from experimental data are used.

Centrifugal compressor flow path losses and outlet flow deviation angles are estimated based on existing data in literature [5,6]. Though obtained results can be characterized as satisfactory, loss models require further refinement One way of increasing the reliability of results is the use of custom libraries for proven loss models within the integrated design system environment.

The typical approach to solve this task and boundary condition is presented in [6]. The authors of this paper used the same method, so it is not necessary to explain it. As an example, validation of the 1D solver was performed based on experimental data presented in [7].

Flow path of the compressor shown in Fig.2 was studied with both a vaned and a vaneless diffuser. Performance calculations show a close correlation of the obtained efficiency with experimental data for the vaneless diffuser design in a quantitative sense and a close correlation for vaned diffuser design in a qualitative sense (Fig.3). Efficiency levels at 80% rotation speed coincide closely with experimental data as well. (Fig.4, Fig.6). The literature only cites CFD calculations results (Figs.7a,7b,7c,7d) and experimental data at an 80% rotation speed ([9], [10] etc.). These results correlate closely to experimental data at an 80% speed as well.

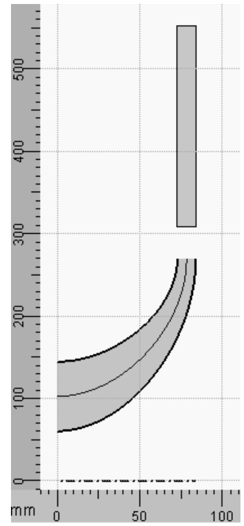


Fig.2. Centrifugal compressor [7] flow path draft

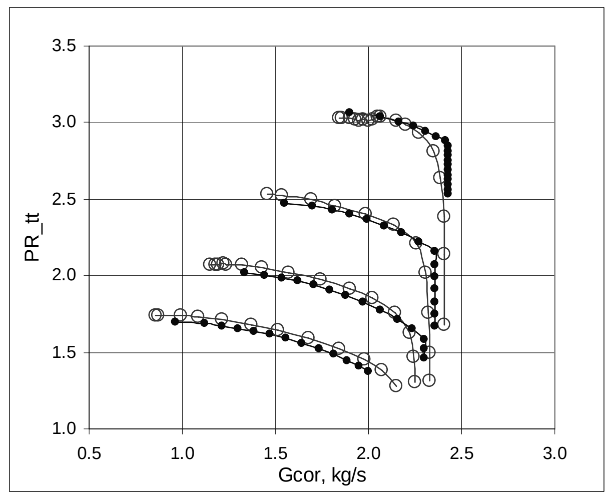


Fig.3. RADIVER compressor performance map with vaneless diffuser. (-o- experiment, -.- calculations). at 80% rotation speed.

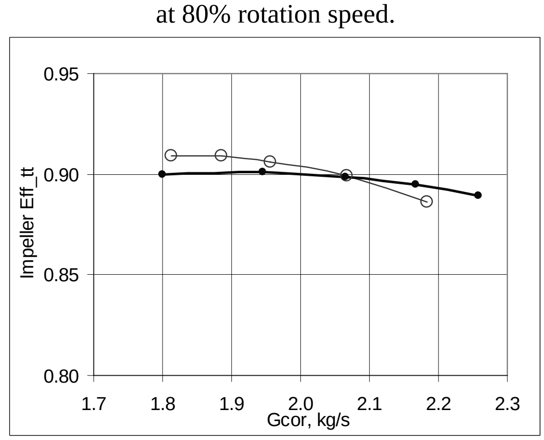


Fig.4. RADIVER compressor total-to-total efficiency map with vaneless diffuser (-o- experiment, -.- calculations) at 80% rotation speed.

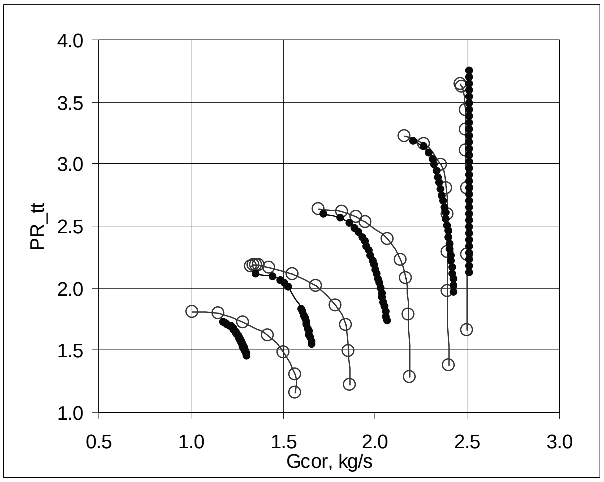


Fig.5. Radiver performance map with vaned diffuser (-o- experiment, -.- calculations).

By means of numbers, rotor rotation speeds are shown in the figure.

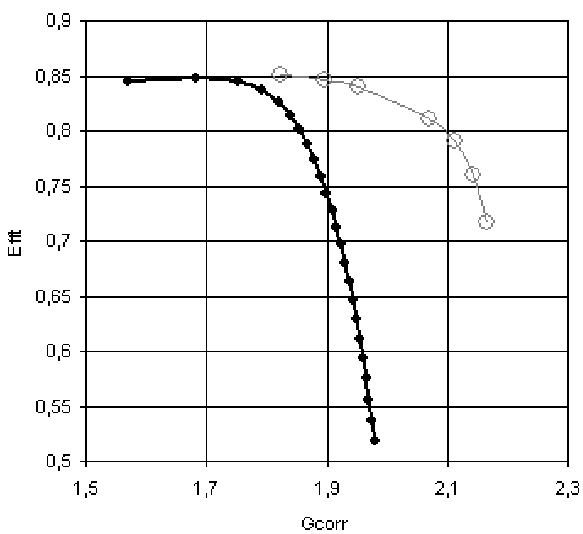


Fig.6. RADIVER compressor total-to-total efficiency map with vaned diffuser (-o- experiment, -.- calculations) at 80% rotation

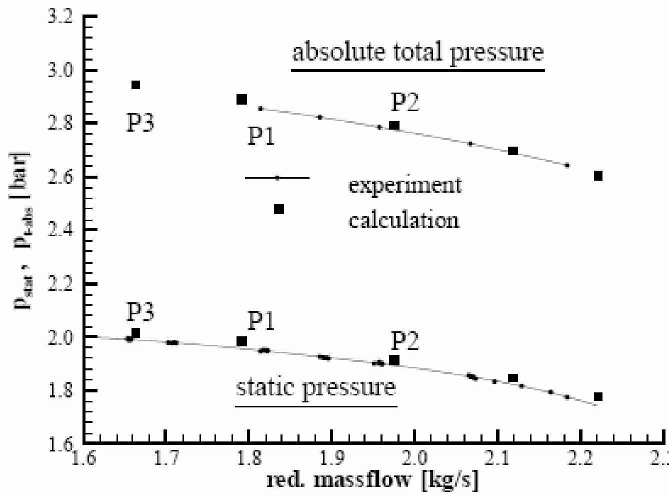


Fig. 7a. Static and absolute total pressure comparison at plane 2m at 80% rotation speed. Experimental data [10], CFD calculations data [7]

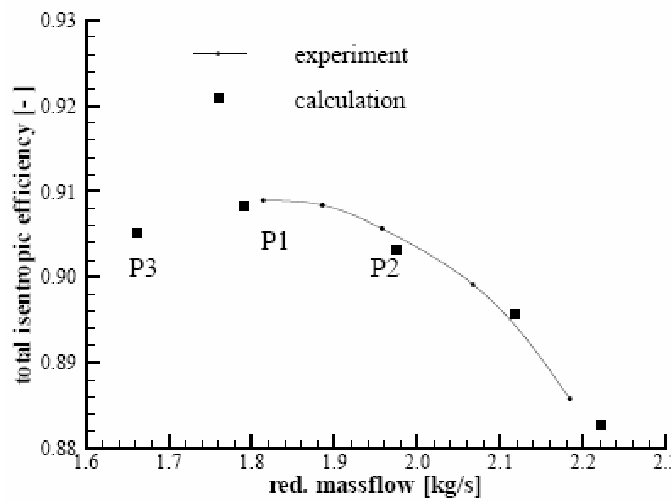


Fig. 7b. Isentropic efficiency comparison at plane 2m at 80% rotation speed. Experimental data [10], CFD calculations data [7]

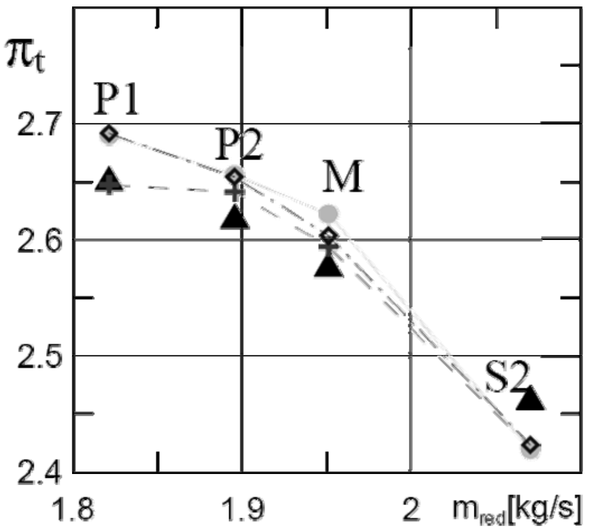


Fig. 7c. Total pressure ratio (πt) and isentropic efficiency (η) comparison (r4/r2=1.14) Experimental data [10], CFD calculations data [8]

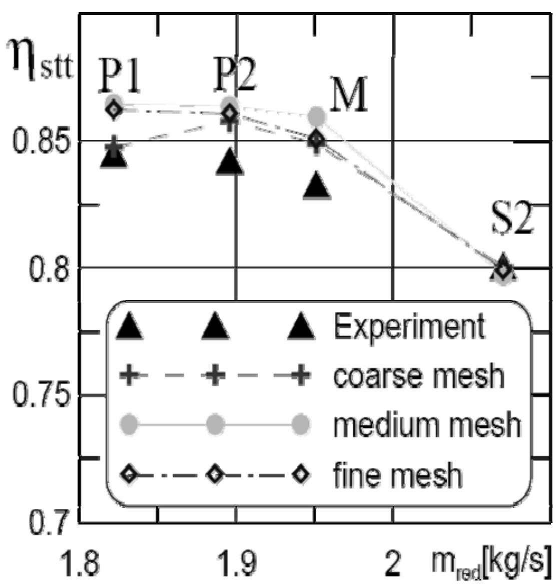


Fig. 7d. Total pressure ratio (πt) and isentropic efficiency (η) comparison (r4/r2=1.14) Experimental data [10], CFD calculations data [8]

CENTRFUGAL COMPRESSOR SIZING

There exists an opinion in the literature that preliminary design of turbomachinery stages may be performed based on experimentally generalized diagrams. For axial turbines, Smith's diagram is often used [9], for radial – Chen and Baines [10].

These diagrams, when compared with solutions generated according to the proposed algorithms of the preliminary design, show that utilization of diagrams will not produce an optimal construction because:

- generalized parameter ranges are not sufficiently large;
- real, current day design solutions may fit outside the accepted ranges of these diagrams that were created several decades ago;
- there is a design results divergence (about 10% at efficiency) at the same generalized parameters.

The new IE approach gives the designer a uniform tool for exercising an optimal parameters search of the axial and radial turbomachines for both turbines and compressors. The essence of the method is in the multiple (sequential) solving of the flow path inverse 1D gas-dynamic calculation problem to find the best solution. See more details [4]. One variant design is performed by assigning proper set of flow paths, with both dimensional (diameter, height) and dimensionless (flow factor) characteristics. An optimal solution search is carried out using quasi random search.

In this paper we are introducing an approach for centrifugal compressors for the first time. For centrifugal compressors, dimensions determination can be assigned: inlet parameters P_1*, T_1*, mass flow rate G, pressure ratio P_3/P_1. It is important to know the diffuser type (vane, vaneless, etc.), Speed of rotation n, rotor diameter at the stage inlet D_1, mass flow rate coefficient c_{1z}/u_1, diameters ratio D_2/D_1, and flow outlet angle at the stage α_3 are variable parameters. Optimal design criteria can be maximum stage efficiency or maximum output (in turbine) or minimum input (in compressor) capacity. If there exists a reliable losses model procedure, the inverse 1D calculation allows for fast generation of solutions with sufficient admission of variables, and the ability to compare design variants to choose the best solution.

Efficiency – flow rate coefficients and typical load diagrams are shown in Fig 8. Point color corresponds to efficiency level. Pale points are outside the range of established constraints for an acceptable solution search such as the maximum outer diameter constraint. The envelope of points, including all "candidates" for an optimal solution, is smooth, but one can see that points delaminate essentially "into the depth." Particularly Fig. 8.b shows that there is an efficiency optimum by load coefficient in the admitted range and efficiency; flow rate coefficient relationship is neutral in the assigned range.

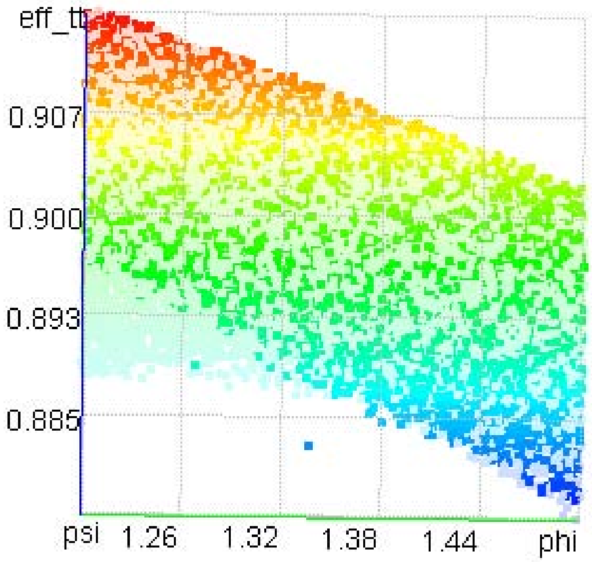


Figure 8.a

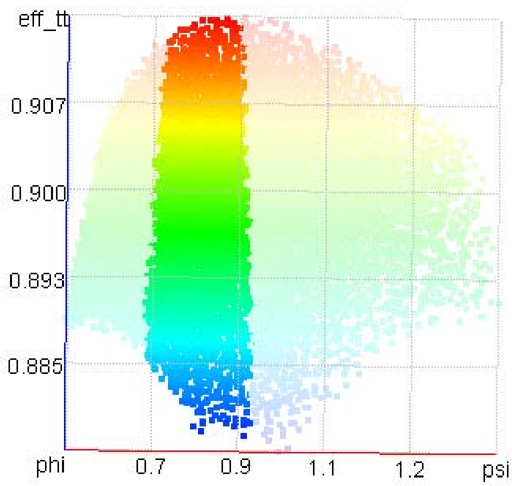


Figure 8.b

Fig.8. Estimated diagrams for centrifugal compressor efficiency (phi – flow coefficient, psi – work coefficient)
Source: AxSTREAM.

3D BLADES DESIGN

Approaches to blade profiles are based on a definition of hub/shroud meridional trajectories with quadratic Bezier-curves with linear extensions. Intermediate stations (trajectories) are created as a channel/sub-channel equidistant curve. (Fig. 9)

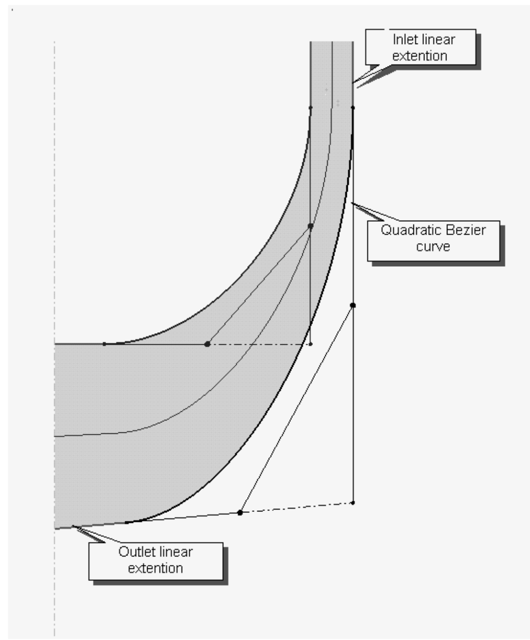


Fig.9. Meridional projection

The frontal camber map is built as an m'-theta plane. The angles are preserved on the leading and trailing edges while intermediate points are distributed along the quadratic Bezier-curve (Fig.10).

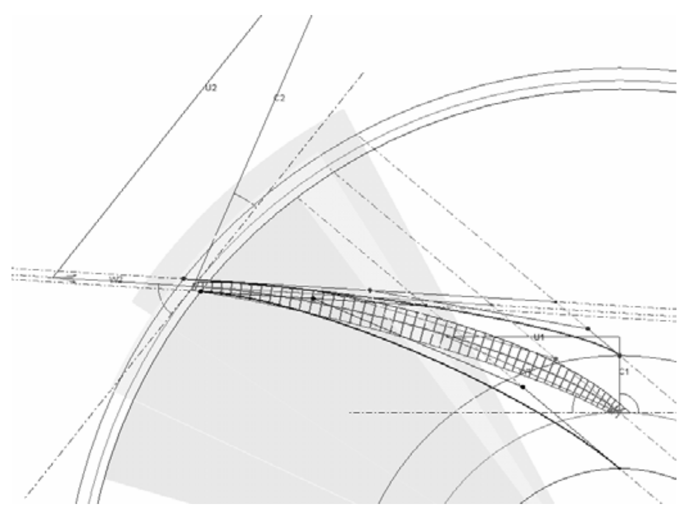


Fig.10. Frontal m'-theta plane

Spatial shape of blade camber surface is defined with conformal mapping of meridional trajectories on the m'-theta plane. Camber line spatial point P is defined with coordinates (x, R, Theta) where x and R are its coordinates appropriative on meridional view. The theta coordinate is defined as the intersection of the camber map on the m'-theta plane and the circular arc with the radius equal the distance to point on meridional projection along the curve.

Operational/manufacturing issues are also considered in a blade design. The availability of radial elements is critically important for high-speed wheels.

Stacking of 3D sections is controlled with the angular (theta) offset of LE points of hub, shroud, and any intermediate sections.

Additionally, TE points of shroud, and any intermediate sections also may have an angular (theta) offset.

Blade design procedure allows the use of rounded (circular) edges and truncated ones (cut-off design).

A centrifugal compressor design example is shown in Fig.11.

The embedded capability to export system cascade geometry into popular CAD formats and CFD packages, permitting easy traversing all the way from preliminary design to 3D gas-dynamic and stress analysis of the wheel .

Designing a wheel with splitters often improves the performance of centrifugal compressors. New design tools allows the use of "dependent" splitters (the splitters and blades are located with even angular pitch) and "independent" splitters when splitter's leading edge is located in arbitrary angular offset (Fig.12).

The splitter camber curvature and LE/TE position of independent splitter may be changed with click-and-drag of control points as may also be done for main blade.

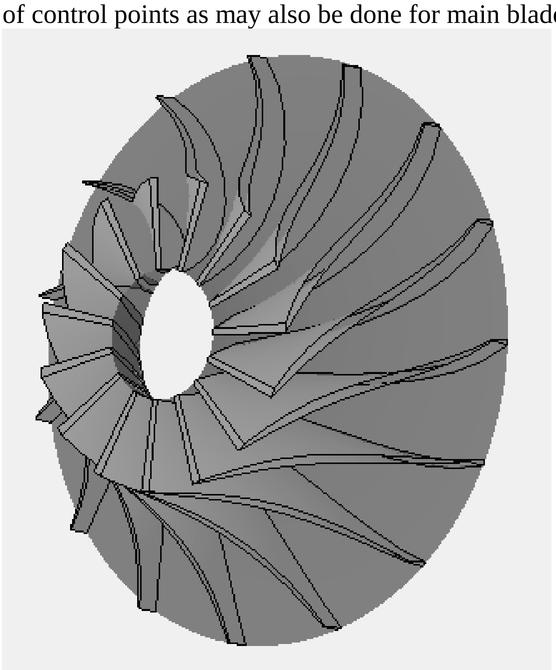


Fig.11. Designed centrifugal compressor blade

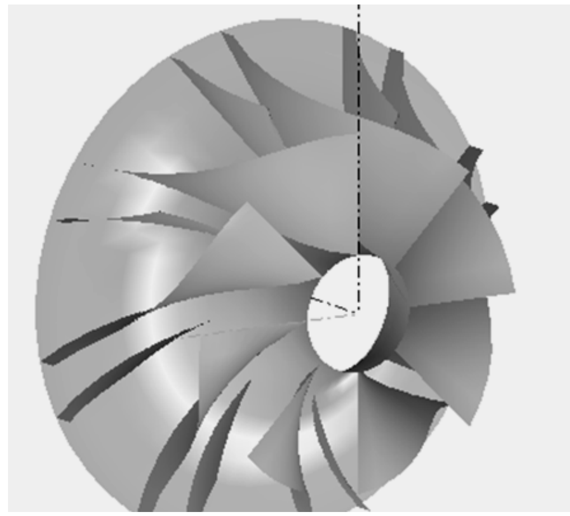


Fig.12. Wheel with independent splitters

3D STRESS ESTIMATION

Aerodynamics and strength requirements lay down additional criteria to meet design objectives. That is why strength and stress estimations should be performed during the early stages of the design process. Historically, detailed stress calculations have been very time consuming, slowing down the process of obtaining an optimal project solution. New design tools provide real time stress analysis in seconds.

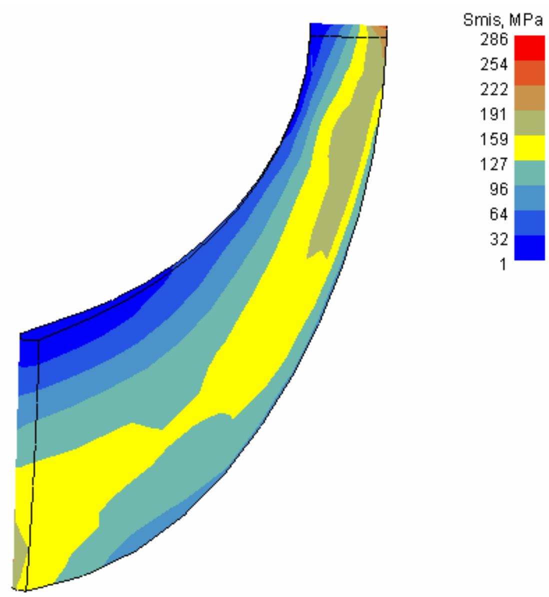


Fig. 13. Stress (von Mises) in the centrifugal compressor blade.

Use of beam theory for radial machine blades is impossible because blade deformations do not conform to the hypothesis of linear distribution of displacements on a cross-section. In spite of the fact that shell theory is more comprehensive to a blade's deformation character, it does not allow one to perform specified calculations and does not agree with the three-dimensional body. That is why FEM was chosen for stress and vibration analysis. There are more details regarding this approach in "The Finite Element Method" [11]. With a minimum number of quadratic brick finite elements, the results are similar to those from calculations obtained with rods and shells theories. If the number of finite elements are increased, higher levels of calculation accuracy will be obtained.

Early in the design process, the aim of stress analysis is not to determine the true stress field but to identify the best project variant among the possible solutions that match the aerodynamic criteria. As the final project nears completion, accuracy must be increased. To ensure that hub and rim are included in the calculations (taking into account rounded radiuses), the performed calculations can be made using one symmetric airfoil or the whole rotor.

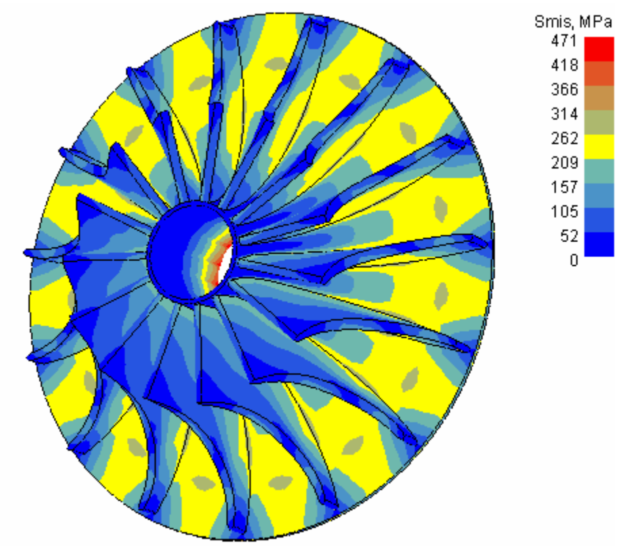


Fig. 14. Stress (von Mises) in the compressor's blades and hub.

Single blade (Fig. 13) and bladed disk (Fig.14) von Mises stress calculation results comparison shows that stresses calculated for single blade differ from blade in bladed disk up to 30%. In comparison, the first calculation requires a fraction of a second as compared to the second one that takes a few minutes in normal formats .

In Fig.15 the first 50 natural frequencies of bladed disk are shown. Frequencies thickening near 5000 and 9000Hz correspond to blade forms of natural vibrations. Horizontal lines f1 – f4 show first 4 frequencies of a single blade.

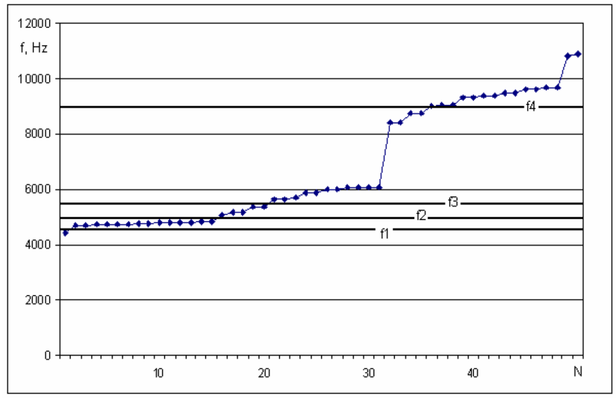


Fig. 15. The first 50 natural frequencies of the compressor bladed disk

3D POTENTIAL FLOW ANALYSIS

3D potential flow analysis [12] is performed to analyze gas field velocities in the channels to determine aerodynamic forces on compressor blades. It is assumed that the flow is potential in a fixed coordinate system. Viscosity absence, external forces potentiality, and channel inlet gas vorticity absence are sufficient conditions to determine flow potentiality.

A flow potential is defined via FEA. The pressure field can be defined from the Cauchy-Lagrange integral. Gas compressibility is taken into account iteratively by the way of flow potential determination with non-uniform density. The iterative process is convergent for subsonic flows.

Fig. 16 shows the relative velocities field of the centrifugal compressor channel, calculated according to the gas potential flow model. The model allows one to determine flow degree of irregularity in the channel.

In Fig. 17 static pressure field calculated according to the same model is shown. This field is used to estimate bladed disk strength.

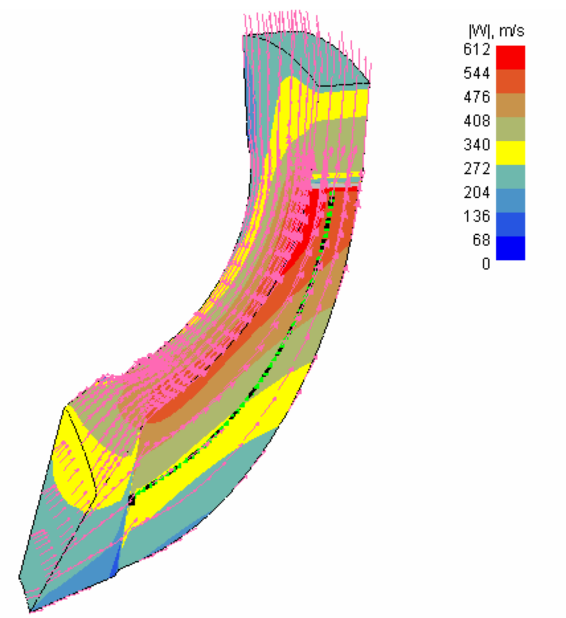


Fig. 16. Field of the relative velocity of the compressor channel.

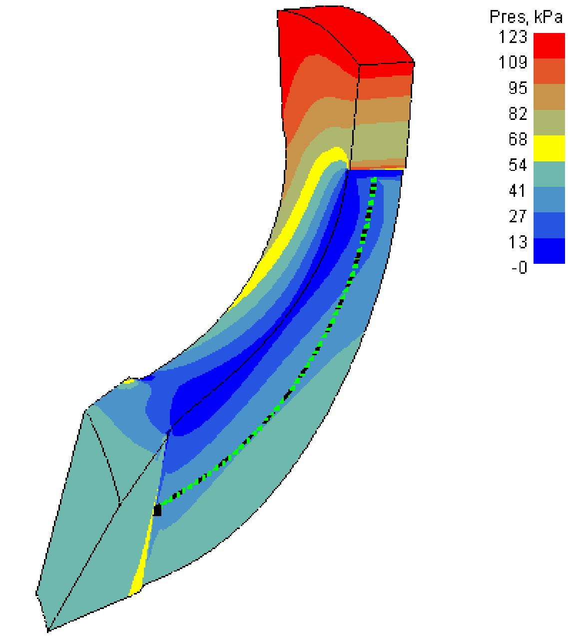


Fig. 17. Field of the static pressure in the compressor channel.

CENTRIFUGAL COMPRESSOR DESIGN EXAMPLES

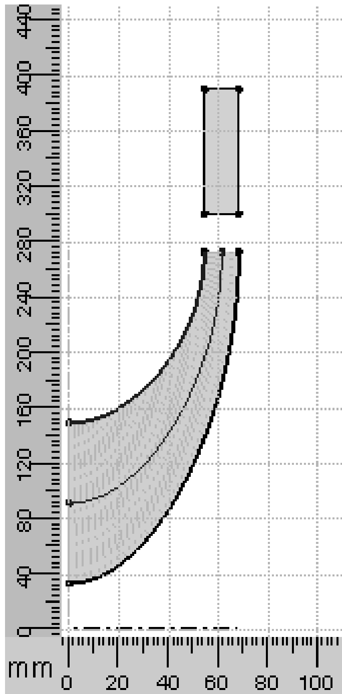
Parameters for CC with vane diffuser design are:

$P_1^* = 60$ KPa,
$T_1^* = 300$ K,
$n = 25000 \ldots 35000$ rpm
$G = 1.5$ kg/s
$P_3/P_1 = 3.5$
$D_1 = 90 \ldots 110$ mm
$c_{1z}/u = 1...1.5$
$D_2/D_1 = 2.0...3.0$
$\alpha_3 = 80^o...100^o$
$D4/D3=1.3$

Optimal stage compressor flowpath with vaned diffuser draft is shown on the Fig.18a, velocity triangles are on the Fig.18b.

The following parameters are received in an optimal point:

$n = 29000$ rpm,
$c_{1z}/u_1 = 1.01$,
$H_t = 0.86$,
$B2 = 95.8$.
$Eff=91.2\%$



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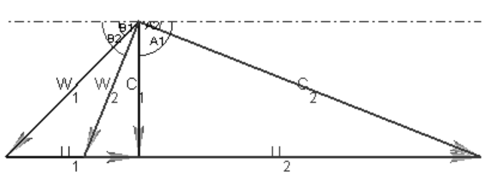

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Fig.18. Velocity triangles of the centrifugal compressor rotor with vaned diffuser.

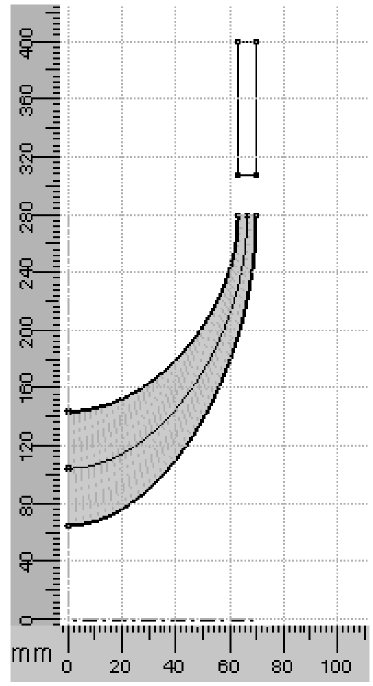
Parameters for CC with vane diffuser design are the same.

Optimal stage compressor flow path with vaneless diffuser draft is shown in the Fig.19a, velocity triangles are on the Fig.19b.

The following parameters are received in an optimal point:

$n = 35000$ rpm,
$c_{1z}/u_1 = 1.03$,
$H_t = 0.86$,
$B2 = 83.0$
$Eff=85.7\%$

Results presented in the article are obtained using turbomachinery flow path conceptual design suite AxSTREAM [2].



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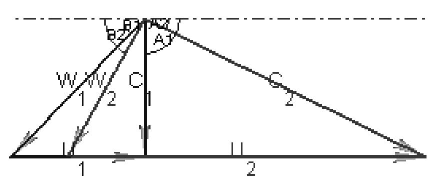

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Fig.19. Velocity triangles of the centrifugal compressor rotor with vaneless diffuser.

CONCLUSIONS

This article introduces a new approach to the development of turbomachinery design systems. The approach gives the designer the opportunity to design axial, radial and mixed flow turbomachinery using the same tool.

Features of radial turbomachinery integration into the turbomachinery conceptual design software environment have been examined in this article. The benefits of this approach are unification of the theory, methods, design procedures, and total functionality for different types of machines with fast calculations to improve human productivity.

Through a single interface with embedded (plug-in) modules, the design process successfully traverses the way from preliminary design estimates through detailed optimization via multivariate calculations. The final phase culminates with the development of three-dimensional blades models with FEA stress and CFD flow analysis at minimum time.

Work in the IE also gives other useful opportunities such as optimization and takes into account weight and strength criteria, in addition to mixed type flow path analysis.

The IE incorporates an algorithm of the radial turbomachinery rotor, stress, and modal analysis that can be easily adapted to meet the needs of preliminary design in terms of accuracy and speed of results.

These new theories and tools will substantially increase the efficiency and reliability of radical and mixed flow turbomachinery.

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