

Methodology of impeller curved vanes modelling in 2D axisymmetric stress analysis

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Abstract

The purpose of this paper is to propose a more suitable methodology of impeller vanes modelling in axisymmetric 2D models which improves the results but does not increase the computational complexity of the task. For this purpose, two different approaches have been tested which led to a significant improvement in 2D axisymmetric model results. The first approach directly models the spatial curvature of the vanes using shell elements which are connected to axisymmetric disk elements using coupled degrees of freedom. The second approach modifies the commonly used methodology which simulates the vanes using plane elements under the plane stress conditions. The influence of spatial curvature of the vanes is then empirically considered by reducing the Young's modulus of elasticity of vanes material. In addition, this approach has been extended using orthotropic material that allows material properties to be changed in certain directions. Advantages and disadvantages of these approaches are reported.

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1. Introduction

Even today the turboprop engine plays an important role in the field of aviation, where high flight speeds are not required. In this field the turboprop achieves greater efficiency and therefore lower fuel consumption than conventional jet engines. This type of propulsion is therefore ideal for small slower airplanes such as those used in agriculture, military transport and private or recreational flights. GE Aviation is currently developing a completely new turboprop engine called Catalyst shown in Fig. 1. GE's Catalyst engine is the first all-new, clean-sheet engine for the business and general aviation turboprop market in more than 30 years. Czech Technical University in Prague (CTU) is partnering with GE Aviation in the development of the new powerplant. More GE Catalyst engines are being assembled which will be tested in the CTU Prague test facility.

Development of a new turboprop engine is a challenging and complex process that affects many professional disciplines. At present, with the constant development of computers, great emphasis is placed on the use of computational models that make the development process considerably more efficient. One of the key models in this area is the structural model of the core rotor which among other things is used to calculate low-cycle fatigue (LCF) or to determine boundary conditions for more detailed models of selected components. The core rotor model usually consists of dozens of complicated pieces. Due to contacts and large deformations, it is a strongly non-linear analysis. Moreover, hundreds of load cases defined by the flight mission are

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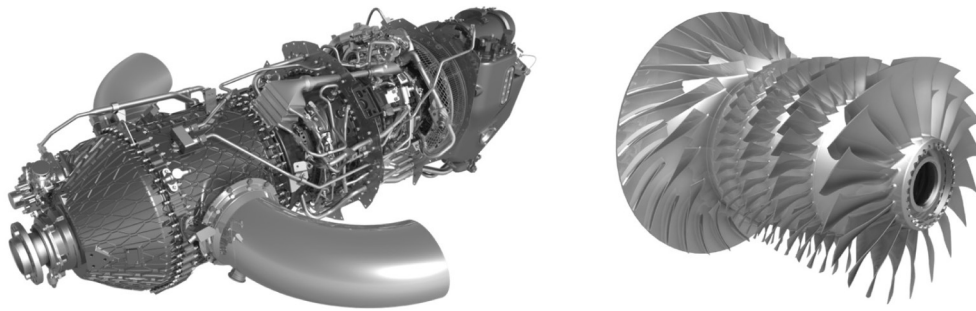


Fig. 1. GE Aviation turboprop engine Catalyst

analysed. For these reasons even with the current computing capabilities of modern computers, it is not convenient to analyse such task as full 3D model. It is not just a problem of long computational time, but especially of the huge amount of data that needs to be further processed and archived. For this reason, axisymmetric 2D models are often used for rotors. The advantage is a considerable reduction of computational time and stored result data size. However, this approach involves the problem of non-axisymmetric features modelling. Non-axisymmetric features in 2D axisymmetric finite element models need to be modelled to provide accurate component mass, stiffness and hoop carrying capability. These features are for example bolt holes, air holes, scallops, disk posts, dovetail slots, blade shanks, blade airfoils, bolts, nuts, retaining rings, balance cuts, splines, counterbores or curvics. It is very important to simulate a 3D feature in 2D as effectively as possible to ensure consistency between 3D/2D models and provide for a valid 2D model. The selected non-axisymmetric features are then analysed using detailed 3D models which use boundary conditions from a global 2D axisymmetric model of rotor.

In small turboprop engines a high-pressure compressor often consists of a combination of axial and radial compressors. In the case of the Catalyst it consists of four stages of axial and one stage of radial compressor as shown in Fig. 1.

In the case of the impeller inconsiderable differences between 2D and 3D results were observed. It has been found that the commonly used methodology of modelling impeller vanes in the axisymmetric model gives acceptable results in the case of a straight vanes impeller, but in the case of curved vanes the ability of the methodology to approximate reality is substantially reduced. Curved vanes modelled by commonly used methodology have higher stiffness compared to reality. It produces differences in stress values even in locations that should no longer be affected by vanes stress peaking.

The purpose of this paper is to propose a more suitable methodology of impeller vanes modelling in axisymmetric 2D models which improves the results but does not increase the computational complexity of the task. For this purpose, two different approaches have been tested in the software Ansys [1] that leads to improvement in 2D axisymmetric model results. The first approach directly models the spatial curvature of the vanes using shell elements which are connected to axisymmetric disk elements using coupled degrees of freedom. In this way, the real behaviour of the vanes is described more precisely at the cost of a slight increase in the computational complexity of the analysis. The second approach modifies the commonly used methodology that simulates the vanes using plane elements under the plane stress conditions. The influence of spatial curvature of the vanes is then empirically considered by reducing the Young's modulus of elasticity of vanes material. In addition, this approach has been extended using orthotropic material that allows material properties to be changed in certain directions.

This method does not increase the computational complexity of the analysis, but it is a less robust purely empirical approach that requires parallel 3D analysis to successfully calibrate the material properties of the vanes.

All these modelling methodologies have been tested and verified on a Catalyst rotor model for real ADP (aero design point) loading conditions. However, because it is a highly sensitive data subject to strict confidentiality, a completely fictive geometry of the impeller has been created for this paper. This geometry was created only to demonstrate the curved vanes modelling problem. Therefore, it does not meet any design, aerodynamic or for example stress requirements. The geometry of the fictive impeller is shown in Fig. 2.

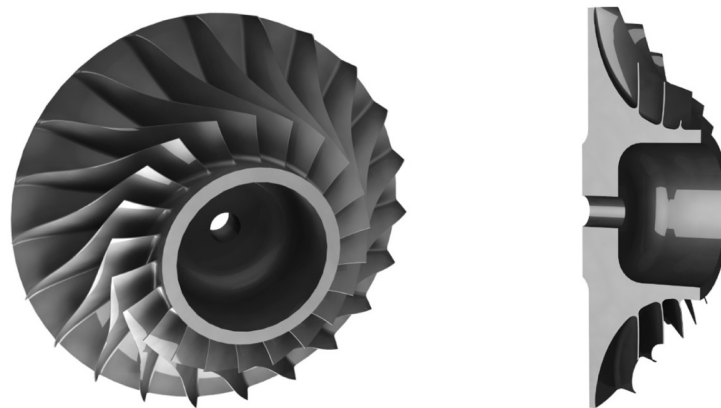


Fig. 2. Fictive geometry of the impeller with curved vanes

2. Commonly used methodology of vanes modelling in 2D axisymmetric models

As mentioned in the introduction, the structural model of the core rotor is designed as a 2D axisymmetric for computational complexity reasons. Non-axisymmetric features in 2D finite element models need to be modelled to provide accurate component mass, stiffness and hoop carrying capability. Relatively accurate approximation of modelling blisk structures is based on mixed-dimension finite element model (MDFEM). The blade and the disk are modelled separately. The blade model is figured by 3D solid elements meanwhile, the disk is structured by 2D axisymmetric element [8]. However, the extensive use of 3D elements leads to a substantial increase in the computational complexity of the analyses. The compressor blades can in some cases be approximated directly by axisymmetric elements. For this approximation, a void ratio is determined which is defined as the ratio of actual volume occupied by discrete blades to the volume obtained by sweeping the axisymmetric mesh by 360 deg. This void ratio is used to scale the mass density and stiffness associated with the blades. This approach is used for example in articles [3, 5]. The most commonly used method is the approximation of blades using 2D elements under the plane stress with thickness conditions (for example [2, 6, 9]). In this approach the FE mesh is made by projecting the blades into the axisymmetric plane and the thickness of the individual elements is mapped in circumferential direction. In order to take into account the stiffness of all blades, the thickness of these elements is multiplied by the total number of blades. This approach is in many cases a sufficient approximation also for the purpose of evaluating the stress on the impeller disk in locations that are no longer affected by vanes (blades) stress peaking. However, in the case of significantly curved vanes the ability of the methodology to approximate reality is substantially reduced. Curved vanes modelled by

commonly used methodology have higher stiffness compared to reality. It produces differences in stress values even in locations that should no longer be affected by vanes stress peaking.

For the purposes of this paper, the completely new fictive geometry of the impeller was prepared in NX software [7] to demonstrate this issue. Fictive impeller which contains 22 identical curved vanes is shown in Fig. 2. In order to have reference results, a 3D FE model of the impeller was created and analysed in software Ansys [1]. Only one segment (1/22 of the full disk) is modelled under the cyclic symmetry conditions. The mesh consists of linear hexahedral SOLID45 elements (disk and vane) and quadratic tetrahedral SOLID95 elements (root fillet). The model contains 82 000 elements in total (103 000 nodes) and is shown in Fig. 3 (left). The initial version of the 2D axisymmetric impeller model is based on the original methodology mentioned above. Vane is projected into the axisymmetric plane and approximated by 2D linear PLANE42 elements under the plane stress with thickness condition. Identical elements are used for the disk but under the axisymmetric condition. The 2D disk mesh is based on a 3D model. Therefore, any influence of discretisation error is eliminated when the results are compared. The initial 2D axisymmetric impeller model is shown in Fig. 3 (right). The figure also shows the visualization of the vane elements thickness which correspond to the vane thickness in circumferential direction multiplied by the total number of vanes $n = 22$ to include all vanes weights and stiffness. The model contains 5 400 elements in total (5 500 nodes). For the purposes of this paper, the model is loaded only by the centrifugal force caused by the rotation of the impeller 45 000 RPM and an ideal elastic material is considered. Analysis includes large-deflection (NLGEOM is ON) and stress stiffening effects (SSTIF is ON). Analysis was performed in Ansys 17.2 [1].

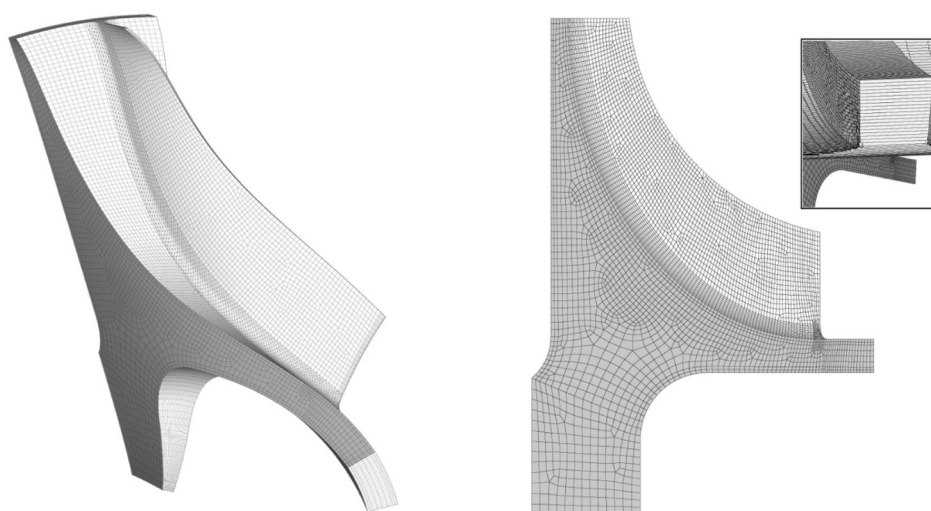


Fig. 3. 3D FE model of the fictive impeller (left), 2D axisymmetric impeller model based on the original methodology (right)

Fig. 4 shows the von Mises stress field and deformed shape (magnification $10\times$) of reference 3D model. Due to the centrifugal force, the impeller back wall is deflected (like umbrella shape). Also a significant deflection of the vane in circumferential direction is observed. Except the local area, which is affected by vane (vane stress peaking), there is almost axisymmetric stress fields in the rest of the disk. Fig. 5 (right) shows 2D versus 3D deformed impeller disk model overlay (magnification $20\times$). The results show that the vane of 2D model prepared by common methodology has higher stiffness compared to 3D model. The axial deformation of the disk based on 2D is 29.7% lower at the top of the impeller (location E). The von Mises stress percent

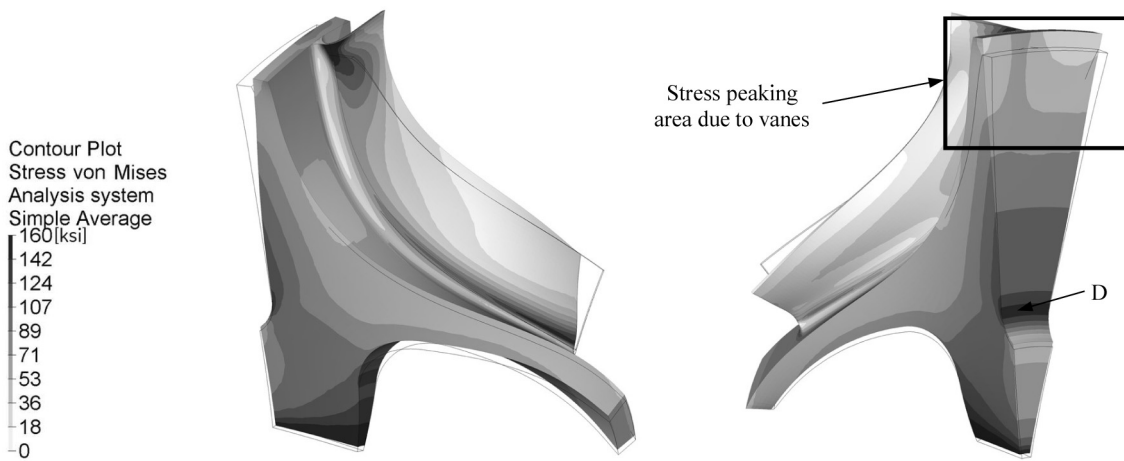


Fig. 4. Von Mises stress field and deformed shape (magnification 10×) of the reference 3D model

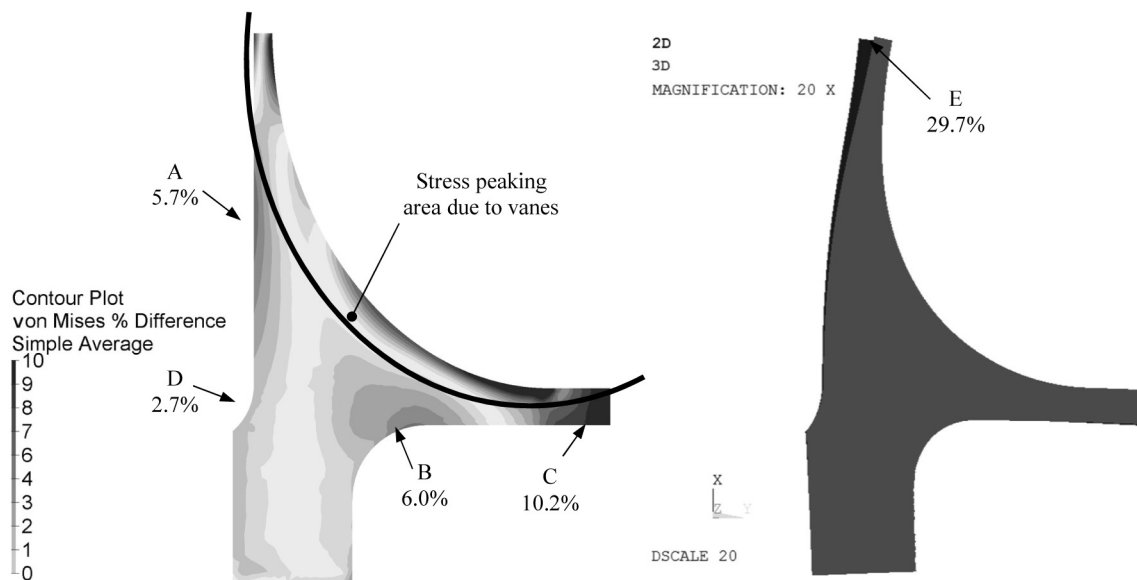


Fig. 5. Original commonly used methodology; von Mises stress percent difference in absolute value between 3D and 2D results (left) and 2D versus 3D deformed impeller disk overlay with magnification 20× (right)

difference in absolute value between 3D and 2D results is shown in Fig. 5 (left). In the case of the 3D model, the average stress values (averaged in circumferential direction) were evaluated. Major differences are mainly in the area that is directly affected by stress peaking due to vanes.

Stress cannot be calculated correctly in this area using 2D. However, significant differences are evident also in the other locations (A, . . . , D) that are no longer affected by vanes stress peaking and therefore more accurate results can be expected. Especially on the back wall (locations A and D), there is high stress and temperature combination under the real operating conditions. In terms of calculating fatigue, knowledge of the exact stress values is crucial at these areas. Stress differences depend mainly on the impeller geometry (especially on the vanes configuration). Similar stress differences were observed for the real Catalyst impeller geometry (up to 10%). It demonstrates that the commonly used methodology for building 2D axisymmetric models is not sufficient for the curved vanes impellers.

3. New methodology based on direct modelling of vanes spatial curvature using shell elements

As shown in the previous chapter, during impeller loading with centrifugal force there is significant bending of the vanes. The vanes bending effect is significant only in the case of curved vanes and is a major source of inaccuracy in the approximation of the impeller by the 2D axisymmetric model using the above-mentioned methodology. Therefore a new methodology was tested that models the real curved shape of the vane through shell elements. The elements represent the midsurface of the vanes as shown in Fig. 6 (left). The fundamental problem of this construction is that the vane nodes generally do not lie in the axisymmetric plane (it is no more a vanes projection). Therefore, CP (coupled degrees of freedom) was used to connect the vane elements to the disk. The vane and the disk are meshed so that the coupled nodes have the same radial and axial positions. The position differs only circumferentially. The vane is modelled using SHELL181 elements. It is a four-node element with six degrees of freedom at each node: translations in the x , y , and z directions, and rotations about the x , y , and z axes. The impeller disk is again modelled by axisymmetric formulation of the PLANE42 elements. This element is defined by four nodes having two degrees of freedom at each node: translations in the nodal x (radial) and y (axial) directions [1]. To have vane and disk displacement consistent, the vane nodes were rotated into local cylindrical coordinate system where x corresponds to the radial and y to axial axes, see Fig. 6 (right). Between the nodes of PLANE42 and SHELL181 elements a CP was created in radial ($ux_2 = ux_1$) and axial direction ($uy_2 = uy_1$). Assuming a small disk torsion, displacement in circumferential direction ($uz_2 = 0$) and rotations ($rotx_2 = 0$) and ($roty_2 = 0$) were fixed in the case of nodes where CP was defined. The rotation $rotz_2$, which corresponds to the rotation in the axisymmetric plane, was not prescribed.

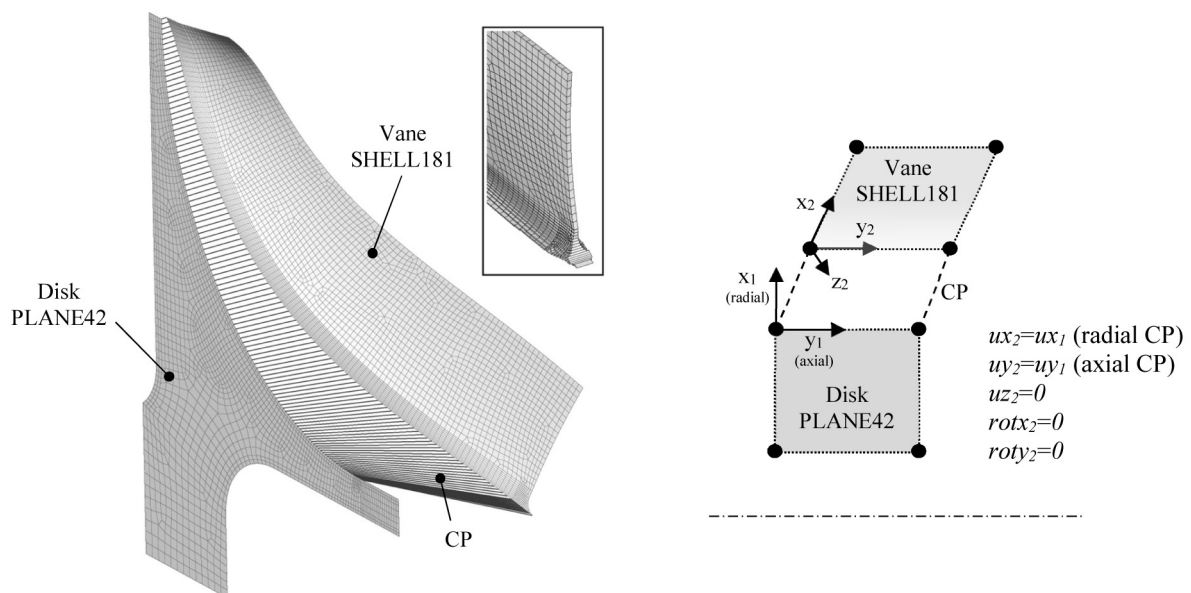


Fig. 6. 2D Model based on direct modelling of vanes spatial curvature using shell elements

The thickness of the shell elements was mapped based on the vane geometry in HyperMesh v14.0 using the “Map Mid-Mesh Thickness” tool [4]. Unlike the previous approach, the thickness in this case is mapped in the normal direction of the individual element, Fig. 6 (left). Again, the key is to include the stiffness and weight of all blades around the impeller. Equivalent vane mass and stiffness increases in direct proportion to the number of vanes n .

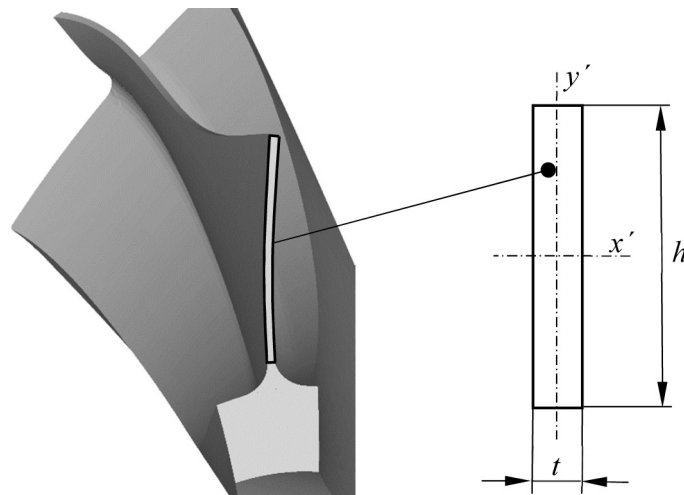


Fig. 7. Vane cross-section approximation by a rectangle

Let us approximate the vane cross-section by a rectangle characterized by the height h and the thickness t (Fig. 7) then the vane mass m is directly proportional to the thickness t

$$m \sim \rho \cdot h \cdot t \cdot l, \quad (1)$$

where ρ is the material density and l is the equivalent vane length. Similarly, the bending stiffness of the vane around the x' axis is also directly proportional to the thickness t

$$K_{x'} \sim E \cdot \frac{1}{12} \cdot h^3 \cdot t, \quad (2)$$

where E is the Young's modulus. These facts are used in the methodology where the vanes are approximated by 2D elements in plane stress with thickness and the equivalent (modelled) element thickness corresponds to the vane thickness multiplied by the total number of vanes n . The equivalent weight m and the bending stiffness $K_{x'}$ are directly proportional to the thickness and therefore to the number of vanes. However, in the case of modelling the vanes using shell elements, the vane may additionally deform in circumferential direction. In contrast to the previous case, it is also necessary to consider the bending stiffness of the vane around the y' axis

$$K_{y'} \sim E \cdot \frac{1}{12} \cdot h \cdot t^3. \quad (3)$$

It is obvious that the bending stiffness $K_{y'}$ does not increase linearly with the vane thickness t . Therefore, the approach based on the equivalent vane thickness cannot be used because the modelled vane would be too stiff. One way to solve this problem is not to modify the thickness but the material properties. It can be seen from (1) that the vane weight m is directly proportional to the density of the material ρ . Similarly, it is clear from relations (2) and (3) that both bending stiffnesses are directly proportional to the Young's modulus of material E . The material properties of the vane ρ and E must be multiplied by the total number of impeller vanes n . In this way, the equivalent material properties of the modelled vane can be obtained. Disadvantage of this approach is that the vane stress will be n times higher due to the use of an equivalent E . In the case that vane stress needs to be evaluated, it is therefore necessary to divide the stress values by the total number of vanes n . An alternative solution is to model the actual number of vanes n directly. In this case the vane elements are copied n times to the same position and the newly created nodes are merged. Then, the total number of degrees of freedom of the model

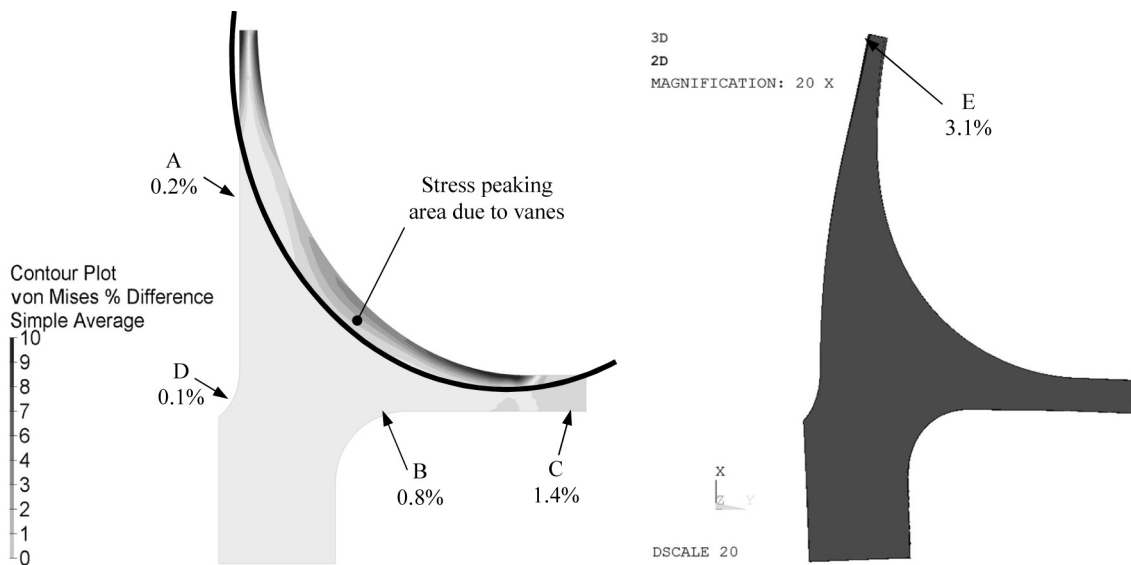


Fig. 8. Model using shell elements; von Mises stress percent difference in absolute value between 3D and 2D results (left) and 2D versus 3D deformed impeller disk overlay with magnification 20× (right)

does not increase. Nevertheless, especially if the large deflections are taken into account and the global stiffness matrix is then repeatedly assembled, the computational time increases due to the higher number of elements in the model. The advantage is the correct stress values on the vane which no more need to be scaled. Percentage difference of von Mises stress in absolute value on the disk between 3D and 2D is shown in Fig. 8 (left). Approach with equivalent material properties is used (ρ and E are multiplied by the number of blades n). Obviously, outside the area that is directly affected by vane stress peaking, there is a significant reduction of stress difference. In the case of impeller back wall (locations A and D), the maximum von Mises stress difference is about 0.2% (the original methodology: 5.7%). Fig. 8 (right) shows 2D versus 3D deformed impeller disk model overlay (magnification 20×). It is obvious that even in case of deflections there is a significant reduction of differences. The difference in axial displacement at point E is 3.1% (the original methodology: 29.7%).

Fig. 9 (left) shows 2D versus 3D deformed impeller vane overlay (magnification 10×). The deformed shape of a 2D vane based on shell elements is in good agreement with the

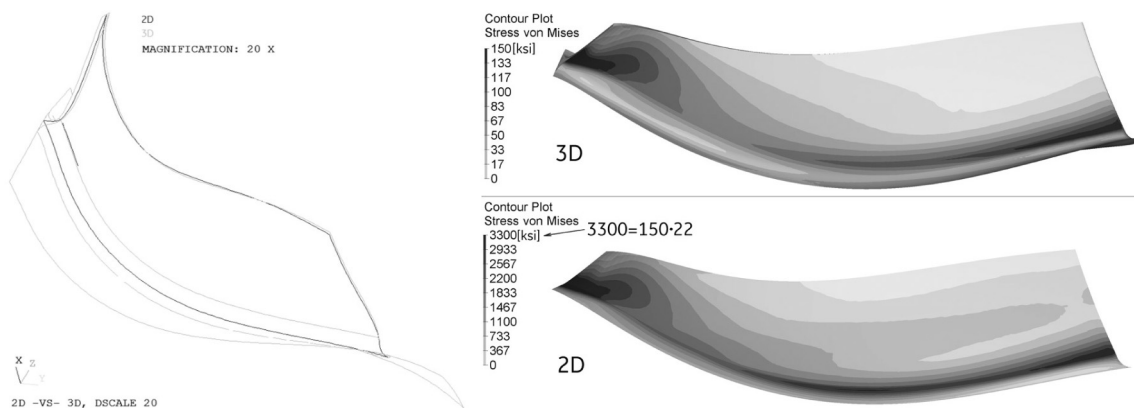


Fig. 9. Model using shell elements; 2D versus 3D deformed impeller vane overlay with magnification 10× (left) and comparison of the von Mises stress field between 3D and 2D vane results (right)

reference 3D model. Therefore, the results can be used to estimate clearance (only approximately) between vanes and stator under the operating conditions. These clearances are key to compressor efficiency. Especially, if the vanes deformation is planned to be evaluated, it is important to model the root fillet with a sufficiently fine FE mesh. The mapped thickness of individual shell elements changes a lot in this area but is constant within the element. It is therefore necessary to create a sufficiently fine mesh in the direction of the root fillet thickness gradient. A sensitivity study on the effect of root fillet mesh density was performed. The use of 13 elements in the direction of the root fillet thickness gradient has proven to be sufficient for a given geometry. Fig. 9 (right) shows the comparison of the von Mises stress field between 3D and 2D vane results. As mentioned above, the vane stress values are n times higher due to the use of an equivalent E . To obtain the actual stress, it is therefore necessary to divide the stress values by the total number of vanes (in this case by $n = 22$). It should be highlighted that the purpose of this methodology is not to calculate exact values of the vane stress. The goal is to model the vane so that the correct stress values on the disk are calculated. However, the results can be used to estimate approximately vane stress field.

Two different shell elements have been tested for vanes modelling. At first the SHELL63 element was tested. This element uses Kirchhoff-Love shell theory which does not calculate transverse shear stresses. But analysis was more difficult to converge in the case that large-deflection (NLGEOM, ON) was included. There was also a problem with elements warping limit which is very restrictive. Convergence was improved by using KEYOPT(2) = 2 which is used to turn off the consistent tangent stiffness matrix when SOLCONTROL is ON. It is the same as KEYOPT(2) = 0, but independent of SOLCONTROL. In the case of using SHELL181 element that uses the Reissner-Mindlin shell theory, the model converged substantially better. For this reason, SHELL181 element was chosen for final analysis.

The described methodology has been tested on a GE Catalyst core rotor model for real ADP (aero design point) under the real operating conditions (temperatures, pressures, temperature dependent material data, etc.). Similar results as presented above were achieved. Outside the area that is directly affected by vane stress peaking, there was a significant reduction of stress difference between 3D and 2D results. Better results have also been achieved when comparing deflection of the disk and vanes. When using the methodology based on equivalent material properties, the computational time has increased by 20% over the original model. When using the methodology based on the direct modelling of the number of vanes, the computational time was more than double. As a result, an approach that uses equivalent material properties can be recommended.

4. Methodology based on modification of vane material model

4.1. Isotropic material

The second approach that is tested in this paper modifies the original methodology which uses 2D linear PLANE42 elements in plane stress with thickness conditions. The effect of spatial curvature of the vanes is empirically considered by reducing Young's modulus of vanes material. The advantage of this approach is its easy implementation into already existing models and no increase in the computational complexity. It is purely empirical not much robust approach based on comparing the results between 2D and 3D models. Model is based on the original 2D model shown in Fig. 3 (right). The Young's modulus of the vane material (and root fillet) is multiplied by the weighting parameter α which expresses the percentage of the original E . The weight

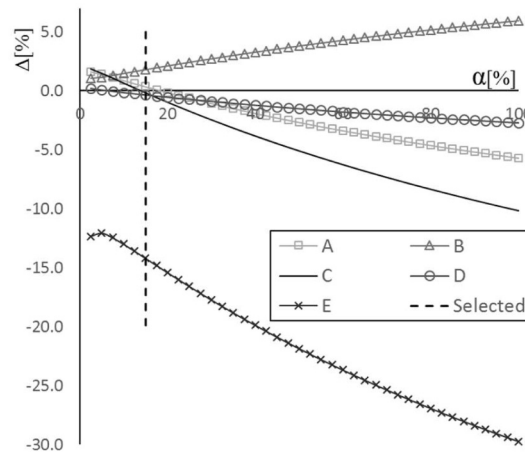


Fig. 10. Isotropic material model; dependence of von Mises stress difference between 2D and 3D in areas A, . . . , D (respectively, the difference of axial displacement at E) on the parameter α

parameter is optimised iteratively by the trial-and-error method. Fig. 10 shows the dependence of von Mises stress difference between 2D and 3D in areas A, . . . , D (respectively, the difference of axial displacement at E) on the parameter α . Based on this study, the optimal value $\alpha = 15\%$ was chosen. For this value, the 2D model shows the minimum averaged stress difference in the evaluated locations. The graph also shows that it is not possible to achieve axial displacement difference below 12% at E for the given geometry.

Percentage difference of von Mises stress in absolute value is for $\alpha = 15\%$ shown in Fig. 11 (left). Obviously, outside the area that is directly affected by vane stress peaking, there is a significant reduction of stress difference. The biggest localised stress difference is 3.7%. Fig. 11 (right) shows 2D versus 3D deformed impeller disk model overlay (magnification $20\times$). Although the difference in axial displacement at the point E has been reduced, it is still quite significant: 14.2% (the original methodology 29.7%).

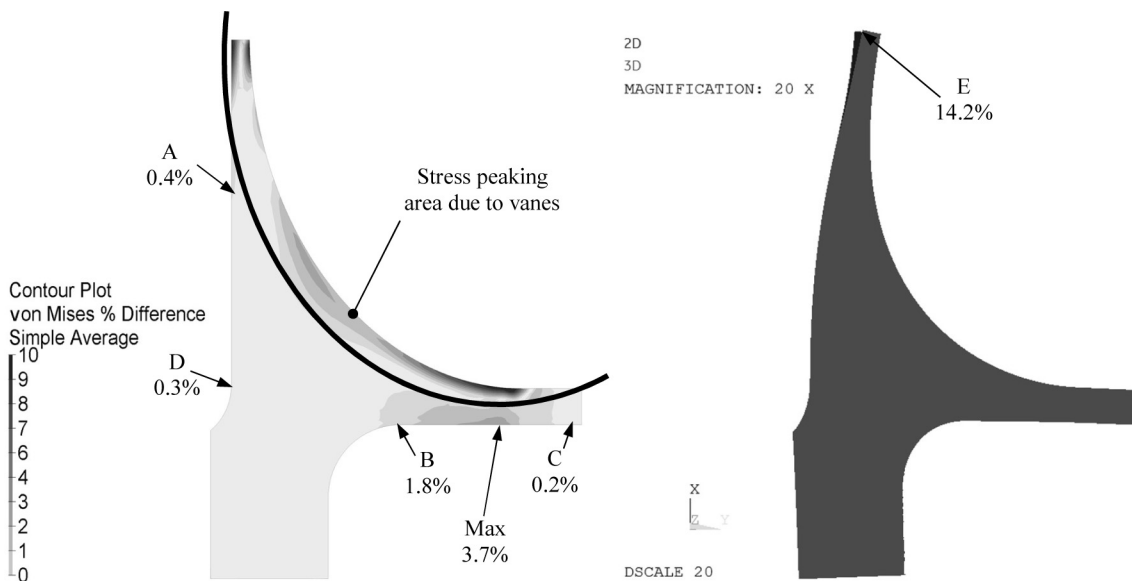


Fig. 11. Model with scaled isotropic material; von Mises stress percent difference in absolute value between 3D and 2D results (left) and 2D versus 3D deformed impeller disk overlay with magnification $20\times$ (right)

4.2. Orthotropic material

The reason why the previous approach based on reducing the vane modulus of elasticity provides better results than the original methodology is the vane stiffness reduction, especially in the axial direction. However, by reducing the modulus of elasticity E the stiffness of the vane also decreases in the radial direction in which the centrifugal forces act. It leads to large radial deformations of the vane. It ultimately leads to unrealistic deformation of the disk at the point E. A possible solution to this problem is to model the vane again with PLANE42 elements but with an orthotropic material that allows to reduce elastic modulus only in the axial direction. The vane is therefore deformed radially less significantly as shown in Fig. 12 (right). Mechanical or thermal properties of orthotropic material are unique and independent in three mutually perpendicular directions. Under plane stress conditions for the in-plane components of the stress and strain, the constitutive relations are of the form

$$\begin{pmatrix} \varepsilon_x \\ \varepsilon_y \\ \gamma_{xy} \end{pmatrix} = \begin{pmatrix} 1/E_x & -\nu_{yx}/E_y & 0 \\ -\nu_{xy}/E_x & 1/E_y & 0 \\ 0 & 0 & 1/G_{xy} \end{pmatrix} \cdot \begin{pmatrix} \sigma_x \\ \sigma_y \\ \tau_{xy} \end{pmatrix}, \quad (4)$$

where only four of the five material constants are independent because of

$$\frac{\nu_{yx}}{E_y} = \frac{\nu_{xy}}{E_x}. \quad (5)$$

In Ansys, the Poisson’s ratio ν_{xy} is referred as the “major Poisson’s ratio” (denoted by PRXY), while ν_{yx} is referred as the “minor Poisson’s ratio” (denoted by NUXY). The vane material parameters with orthotropic material are based on the original isotropic material values E and ν

$$E_x = E, \quad E_y = \alpha_y \cdot E, \quad \nu_{xy} = \nu, \quad G_{xy} = \frac{E}{2(1 + \nu)}, \quad (6)$$

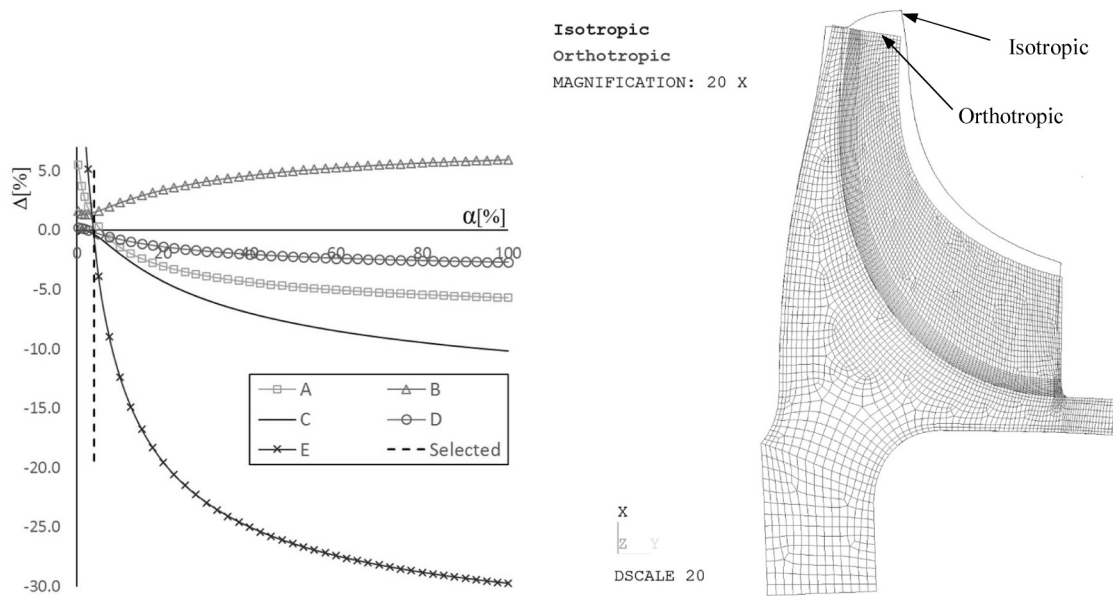


Fig. 12. Orthotropic material model; dependence of von Mises stress difference between 2D and 3D in areas A, . . . , D (respectively, the difference of axial displacement at E) on the parameter α (left), isotropic versus orthotropic 2D deformed impeller disk overlay with magnification 20× (right)

where E_x and E_y are Young's modulus in the radial and axial direction, respectively, ν_{xy} is the major Poisson's ratio (PRXY), G_{xy} is the shear modulus and α_y is the weighting parameter which again represents the percentage value of the original E . This parameter is optimised iteratively by the trial-and-error method. Fig. 12 (left) shows the dependence of von Mises stress difference between 2D and 3D in areas A, . . . , D (respectively, the difference of axial displacement at E) on the parameter α_y . Based on this study the optimal value $\alpha_y = 4\%$ was chosen. Compared to the isotropic vane material it is evident that there is a significant improvement in the axial displacement difference at the point E.

Percentage difference of the von Mises stress in absolute value is for $\alpha_y = 4\%$ shown in Fig. 13 (left). Obviously, outside the area that is directly affected by vane stress peaking, there is again significant reduction of stress difference. The biggest localised stress difference is 4.8%. Fig. 13 (right) shows 2D versus 3D deformed impeller disk model overlay (magnification 20×). Difference in axial displacement at the point E is 1.0% (the original methodology 29.7%). In terms of stress difference similar results were obtained for orthotropic and isotropic material. However, the deformed shape of the disk corresponds to the reference 3D model significantly better. Other modifications of orthotropic material parameters including some of their combinations were also tested. Nevertheless, none of the tested modifications led to better results. The methodology based on isotropic and orthotropic material was tested on a real Catalyst core rotor model under real operating conditions. Also, in this case the previous conclusions can be considered as valid.

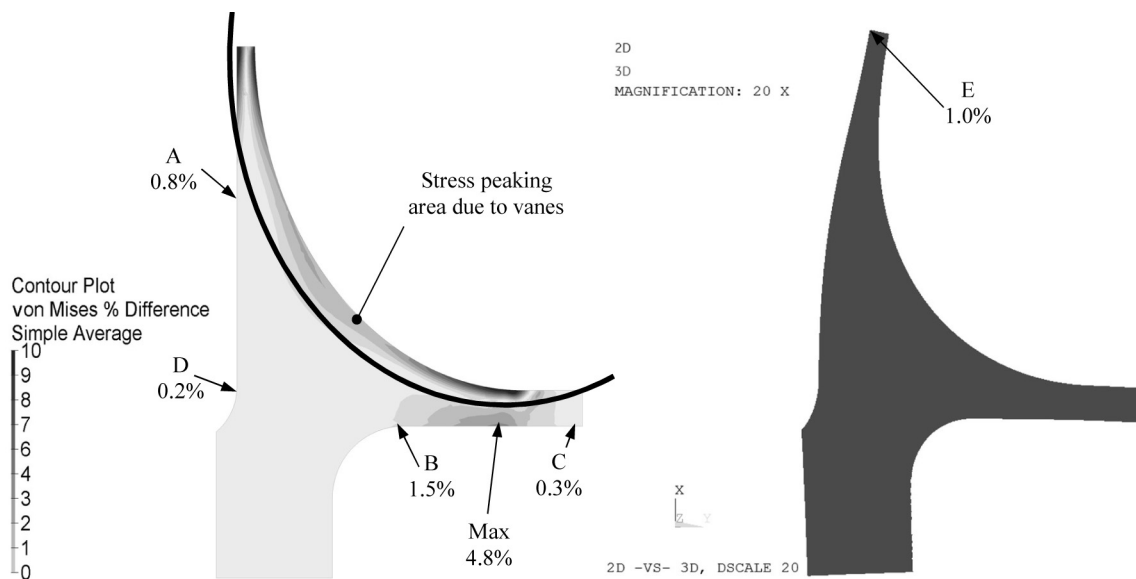


Fig. 13. Model with scaled orthotropic material; von Mises stress percent difference in absolute value between 3D and 2D results (left) and 2D versus 3D deformed impeller disk overlay with magnification 20× (right)

5. Conclusion

Nonlinear structural analysis of large systems such as a turboprop core rotor does not need to be advantageous to be simulated as full 3D models. Axisymmetric 2D models are often used for rotors. The advantage is a considerable reduction of computational time and stored result data size. However, this approach involves the problem of modelling non-axisymmetric features

as for example impeller vanes. Commonly used methodology of modelling impeller vanes in the axisymmetric model gives acceptable results in the case of a straight vanes impeller, but in the case of curved vanes, the ability of the methodology to approximate reality is substantially reduced. Curved vanes modelled by commonly used methodology have higher stiffness compared to reality. It produces differences in stress values even in locations that should no longer be affected by vanes stress peaking. Two different approaches have been tested that led to an improvement in 2D axisymmetric model results. The first approach directly models the spatial curvature of the vanes using shell elements. In this case, the real behaviour of the vanes is described more precisely at the cost of a slight increase in the computational complexity of the analysis. The second approach modifies the commonly used methodology which simulates the vanes using plane elements under the plane stress conditions. The influence of spatial curvature of the vanes is then empirically considered by reducing the Young’s modulus of elasticity of vanes material. In addition, this approach has been extended using orthotropic material that allows material properties to be changed in certain directions.

In terms of the disk stress difference between 2D and 3D results, the best match was achieved using a shell element-based methodology that exactly models the spatial curvature of the vanes. Since the method is based on more accurate geometrical description of the vane, its advantage is great robustness. In the case of a good mesh quality (especially in the root fillet area), the model could be used for the clearance approximation or for a vane stress analysis. The disadvantage consists in the higher computational complexity of the analysis. In the case of a real Catalyst core rotor model, computational time increases by 20%. In contrast, computational complexity does not increase in the case of modified original methodology which empirically reduces the Young’s modulus of vanes material. Another advantage is simple implementation into already existing models. However, it is a purely empirical approach in which the vane material parameters are calibrated to achieve optimal match with the reference 3D model. Depending on the specific case, this methodology can achieve results of different quality. It is therefore not very robust approach. Nevertheless, the given model was optimised to achieve better results compared to the original methodology. Significantly worse results were achieved by comparing the shape of the deformed impeller disk. This inaccuracy has been eliminated by using orthotropic vane material and reducing the modulus of elasticity in the axial direction only.

The methodologies described in this paper have been tested on both the presented fictive impeller geometry and the real core rotor of turboprop Catalyst which is currently being developed by GE aviation. All the conclusions of this paper can be applied to both models. The comparison of the real Catalyst core rotor model computational times and the results obtained from the fictive impeller geometry studied for all tested variants are summarized in Table 1.

Table 1. The comparison of the real Catalyst core rotor model computational time and the fictive impeller geometry study results summary

Vanes Modelling Approach		CP Time [hour]	Max Stress Difference [%]	Max Displacement Difference [%]
Plane Elements	Original Methodology	4.9	10.2	29.7
	Reduced Isotropic Material	5.0	3.7	14.2
	Reduced Orthotropic Material	5.0	4.8	1.0
Shell Elements	Equivalent Material	5.9	1.4	3.1
	Real Number of Vanes	13.2		

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