Strategy for Reduction of the Negative Effects of Circumferential Flow Irregularity in Axial Compressor

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Abstract: Gas flow nonuniformity is one of the main sources of rotor blade vibrations in the gas turbine engines. Often, the source of the flow circumferential nonuniformity are the annular frames, located in the flow passage of the engine. This leads to the increased dynamic stresses in rotor blades and consequently to the blade destruction. The goal of the research was to find an acceptable method of reducing the level of gas flow nonuniformity. Thus, this study gives the ideas about methods of improving the flow structure in gas turbine engine. It allows the selection of the most suitable method for reducing gas flow nonuniformity.

1 INTRODUCTION

Gas flow circumferential non-uniformity is one of the main sources of oscillation of turbomachinery rotor blades (Ivanov, 1983; Vorob'ev., 1988; Cohen etc., 1996). The flow circumferential irregularity is caused by various factors: the axial asymmetry at the turbomachinery inlet; the presence of guide vanes (GV) and nozzle blades (NB) in the turbomachinery flow path; the influence of the structural elements (struts and bearings); bypass system operation and air bleeding; the operation of fuel nozzles of combustion chambers; the deformation of power casings; the buckling of the flame tubes of combustion chambers.

Several methods are used in practice to reduce high dynamic stresses: dynamic methods, gas dynamic methods and technological (Figure 1) (Ivanov, 1983; Birger etc., 1993; Logan, 2003).

Each method has its advantages and disadvantages, as well as the range of applicability. Methods that suggest design changes are impossible to apply to already designed engines.

However, it is possible to test the effectiveness of the proposed methods of dynamic stresses reduction without resorting to expensive and long experimental studies. For these purposes, it is possible to use CFD (Respondek, 2010) of the work processes occurring in the engine as a preliminary verification step.

The purpose of the research was to evaluate the of various methods and to indicate the most appropriate approaches for solving specific problems.

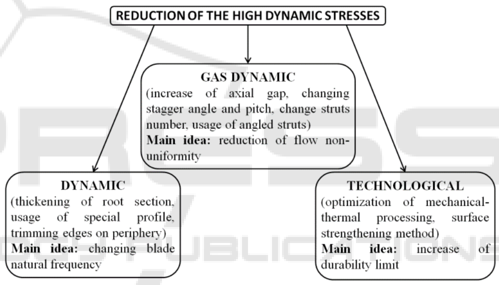

Figure 1: Classification of different methods to reduce high dynamic stresses in rotor blades.

2 MOTIVATION

The motivation for this research was the real-life destruction problem of the fifth stage rotor blades of the intermediate pressure compressor GTE NK-36ST (RW5 in Figure 2).

The reason of rotor wheel blades' destruction is the flow circumferential irregularity, which arises from the presence of middle annular frame struts in the GTE flow path. Struts have different profile thickness and distributed with different angular displacements relative to each other (Figure 3). Struts are the cause of the appearance of local zones of high pressure, which propagate upstream through the cascade of GV5 fixed blades. Passing through these zones, rotor wheel blades of the fifth stage experience dynamic effect, leading to their forced oscillations and high dynamic stresses (Figure 4).

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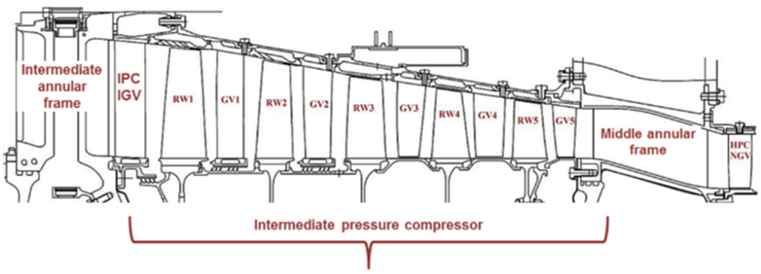


Figure 2: The diagram of investigated compressor flow path.

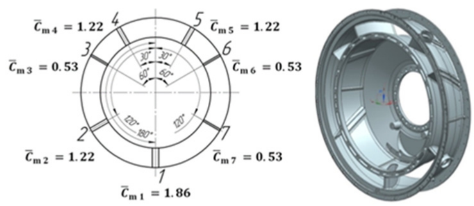
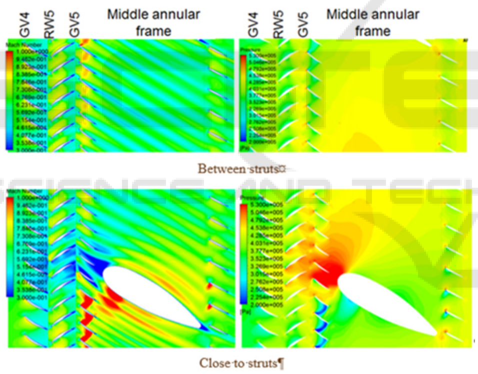


Figure 3: The appearance of middle annular frame.

Because of this, the fifth rotor wheel (RW) blades in the original design have anti-vibration segments. However, such a solution has several shortcomings (Cumpsty, 2004).



a) b)

Figure 4: Mach number (a) and pressure (b) fields.

Thus, the aim of the work was the search of an alternative approach for reduction of dynamic stresses level in the fifth rotor wheel blades of intermediate pressure compressor (IPC). For further research, the fifth-stage rotor blades were modeled without anti-vibration segments.

3 METHOD OF DYNAMIC STRESSES CALCULATION

At the initial stage of the work, the method of dynamic stresses calculation in the fifth rotor wheel blades of IPC was developed (Figure 5).

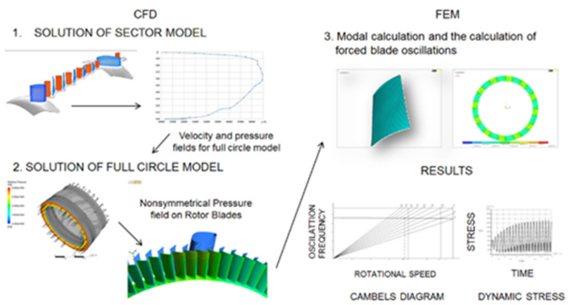


Figure 5: Scheme of method for calculation compressor rotor blades forced oscillations.

Stage 1. CFD-simulation of IPC sector model on the main modes of GTE operation. The sector is one blade channel of blade row, the lateral boundaries of which are imposed by boundary conditions of periodicity. The IPC sector model consists of the intermediate frame domain, the domains of all IPC blade rows, the middle frame domain and the inlet guide vane of high pressure compressor (HPC IGV). When transmitting the flow parameters between domains, the Mixing Plane interface is used (NUMECA, 2008), averaging the flow parameters in the circumferential direction. The purpose of this calculation is to determine the flow parameters distribution (total pressure, total temperature, flow angle) along the flow path height in the cross section behind the RW of the IPC fourth stage, as boundary conditions for subsequent calculations.

Stage 2. The CFD modelling of full circle model will allow to detect the presence of gas flow circumferential irregularity, which arises from the nonuniformly located struts of different thickness along the circumference.

This model consists of GV4, RW5, GV5 of IPC, middle annular frame and HPC IGV domains. Gas dynamic load acting on all blades of the fifth stage rotor wheel is determined in this calculation.

Creation the grid of finite volumes of the full circle model was carried out in AutoGrid5 (NUMECA, 2008). Parameters of finite volume grids and computational models tuning were chosen according to the recommendations given in papers (Matveev etc., 2014; Popov etc., 2014). The computational models were verified in studies (Kolmakova etc., 2014; Ermakov etc., 2014) in assessing the vibration level of rotor blades. This indicates the reliability of the calculations performed in the study described in this article.

Total pressure and temperature, and inlet flow direction at the inlet, which were obtained from the sector model calculation at the previous step (Ermakov etc., 2014); static pressure at the outlet of the computational model were set as the boundary conditions for the full circle model.

To transfer parameters between the rotating and fixed domains, the standard Frozen Rotor interface was used without their averaging in the circumferential direction (ANSYS, 2017).

Stage 3. Determination the values of exciting harmonics amplitudes of the load, as well as dynamic stresses arising in the RW5 blade. For this purpose, in the APDL programming language, the program that imports the load distribution from the finite volume blade row model in CFX into the finite element blade row model in Ansys Mechanical APDL, were written. The gas load, which has a complex character of the distribution along the circumference, was represented as a sum of the harmonics components. Each harmonics component represents a load waves chain that locate along the flow path circumference. The load rotates at angular velocity. Thus, the circumferential stationary inhomogeneity of the gas flow for a rotating rotor wheel is equivalent to the action of an infinite set of exciting harmonics, each of which is a load chain of backward running waves performing harmonic oscillations in time.

The load is expanded in Fourier series along the similar blades' nodes and is represented in the form of a backward running wave. Campbell diagram is used to determine the most dangerous operating modes of the engine, and the most dangerous harmonics. The dynamic stresses are calculated only with the most dangerous harmonic.

4 METHODS OF REDUCTION THE DYNAMIC STRESSES

4.1 Detuning the Rotor Blade from the Dangerous Harmonics

IPC testing without shroud segment of RW blades (the blade without shroud segment is a standard blade airfoil without anti-vibration segment) was carried out by JSC "Kuznetsov".

The experimental investigations were carried out using a cold test rig for the IPC with standard pressure at the inlet and middle annular frame at the outlet. Two types of the tests were conducted. Strain gauges were installed at five rotor blades in the first case and at 12 rotor blades in the second test case (total number of rotor blades was 84). The maximum rotor speed, during the tests was the speed at which resonance occurs with the most dangerous harmonics.

These tests revealed the destruction of the IPC fifth stage rotor blades at resonance with the 7th and 12th harmonics. Figure 6 shows the histogram of the exciting load amplitude distribution, calculated using the method described in clause 3. The average value on the blade surface that is calculated by adding the amplitudes of the harmonics in each node and dividing by the number of nodes is referred to the mean amplitude. Similar calculations are performed for all blades in the fifth blade row.

As can be seen from Figure 6, the 12th harmonic has the highest amplitude (harmonics from 7 to 14 fall into the region of GTE operating modes).

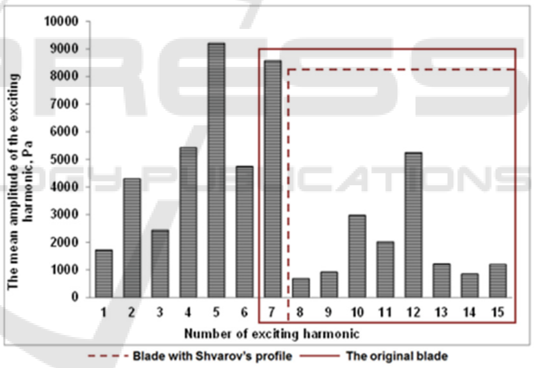


Figure 6: Level of the amplitudes of the exciting harmonics.

An attempt to turn out from the most dangerous 7th harmonic, and to increase the flexural strength of the airfoil blade to reduce the dynamic stresses at resonance with the 12th harmonic, was made at JSC "Kuznetsov". For this purpose, a special Shvarov's profile was used. The shape of the suction side surface coincides with the initial profile; however, the pressure side surface is convex. This profile shape allows to significantly reduce the flexural stresses in the airfoil. Finite element harmonic analysis showed a double decrease in the dynamic stresses of blade airfoil at the resonance with the 12th harmonic (Figure 7). Increase of IPC efficiency was about 0.5% for the compressor variant with Shvarov's profile on the rotor blades compare to the compressor with shrouded blades.

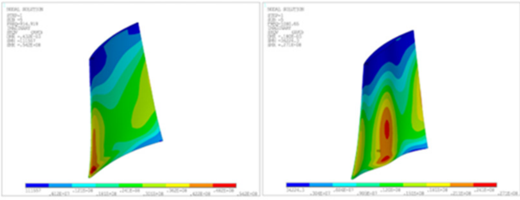


Figure 7: Equivalent stresses at the resonance in RW5 blade (original is on the left, Shvarov's profile is on the right).

Figure 8 shows the Campbell diagram for the 5th rotor wheel with the original unshrouded rotor blade and with blade with Shvarov's profile. Thus, the use of blade with the special Shvarov's profile allows to significantly reduce the dynamic stresses in the airfoil, as well as to turn out from dangerous operating modes. For the blades with Shvarov's profile the strongest seventh harmonic is out of region of the engine operating modes (Figure 6).

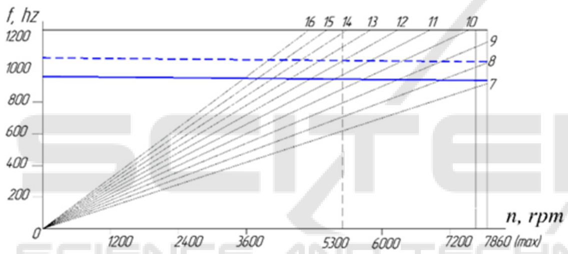


Figure 8: RW 5 resonance diagram (--- Blade with Shvarov's profile, - original blade).

Using the blades with special Shvarov's profile allow to reduce the level of dynamic stresses by 1.8 times. This result was confirmed experimentally during strain-gauging at JSC "Kuznetsov". The stresses were measured using heat-resistant strain gauge, located on the surface of the blade pressure side. The single amplitudes were estimated. However, this reduction was not enough to eliminate the problems of blade destruction. It is necessary to influence not only the blades shape, but also the gas flow circumferential irregularity. In further researches, we used the blade with Shvarov's profile.

4.2 Design Change

One of the methods of reduction the gas flow circumferential irregularity in front of the struts is the use of guide vane with a different pitch along the circumference and different stagger angle of the GV blades (Sladojević etc., 2007; Yang etc., 2012). This method allows to redistribute the flow between various blade channels and to regulate the position of high pressure zones (Saren, 1984).

Parametric model of IPC fifth stage GV was created to implement the method. All GV blades were divided into 7 groups, according to the location of the 7 struts. Angles and pitch changes occurred within each group. The key factor was the minimum number of variable blades.

For the GV5 blades located in the symmetry plane of the strut, the first and the last blades in the group, the stagger angle does not change. The change of blades stagger angles within the groups was carried out according to a linear law. The blades located on opposite sides of the strut symmetry plane rotate in opposite directions, relative to the initial position (Figure 9). The blades located closer to the strut rotate a larger angle. If the stagger angle increases, the sign "+" stands in front of the angle, the sign "-" means decreasing stagger angle. All angles are measured from the leading edge plane.

In the case of pitch change, the number of changeable blades was not limited. Different pitch was set in the range -0.35 ... + 0.35 of the pitch. The sign "-" means that in the strut area the pitch between the blades decreased, and the sign "+" - that the pitch increased. The number indicates the maximum increase (decrease) of the pitch between the blades in the group in relative values from the nominal pitch with uniformly arranged blades. When different blade pitches are introduced in groups, the position of the extreme blades groups did not change. The law of pitch change is also linear.

In total, 10 variants of the GV5 configuration with different angles and pitches were tested. Table 1 shows same of them. Variant 1 corresponds to the original version of the IPC without an anti-vibration segment on the RW5 blades with Shvarov's profile.

Table 1: The results of the parametric IPC model calculation.

Variant number	Parameter of different stagger angles, maximum stagger angles (number of blades) for the groups:			Parameter of alternating blade pitch for the groups:					Number of variable blades	Dynamic stresses MPa
	2, 5, 6	1, 3, 7	4 (3)	1, 7	2, 6	3	4	5		
1	0	0	0	0	0	0	0	0	0	86.7
4	3 (6)	3 (6)	3 (8)	0.3	0.3	0.3	0.3	0.3	42	31.877
9	6 (2)	6 (2)	6 (2)	0	0	0	0	0	14	46.737

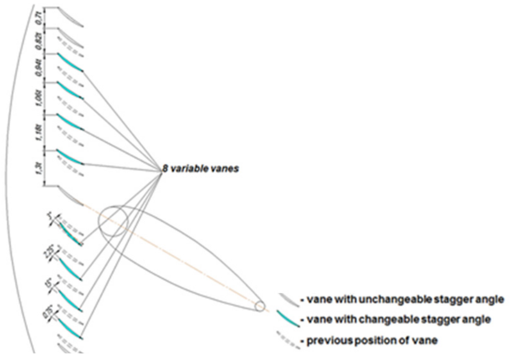


Figure 9: The diagram of GV blades rotation.

The analysis of the obtained results has shown that it is possible to achieve a significant reduction in the level of the exciting harmonics acting on the RW5 rotor blades, and, consequently, the level of dynamic stresses with the help of different angles and pitches.

For example, variant 9 allows to reduce the dangerous harmonic amplitude by almost 2 times. In this variant of the GV5 configuration the number of changeable blades is 14 (a total of 76 blades). Variant 4 allows to obtain the greatest dynamic stresses decrease, however the number of changeable blades will amount to 42, which is more than half of the total number.

Figure 10 shows the mean of average exciting harmonics amplitudes for the original compressor version, GV variant 4 and GV variant 9.

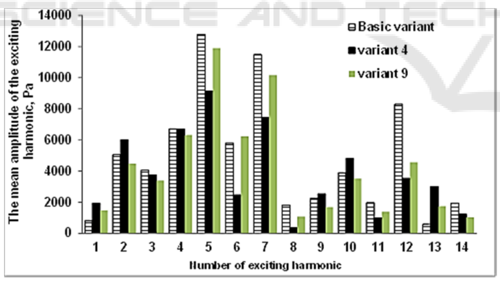


Figure 10: The mean of exciting harmonics amplitudes acting on RW5.

From a gas dynamic point of view, the positive effect of different angles and pitches on the dynamic stresses value can be explained by the following. Due to redistribution of the flow, its structure becomes more homogeneous, with smaller high pressures areas. In addition, the pressure peaks values behind the 5th rotor wheel is also markedly reduced (Figure 11).

Thus, the introduction of GV of different angles and pitches can reduce the gas flow circumferential irregularity and dangerous harmonics amplitude. However, the use of this method leads to a complication of production technology. GV blades in this case will have to be manufactured by nonuniversal technologies. In addition, due to the change in the blades pitch, it will be necessary to develop a new technology for grooving.

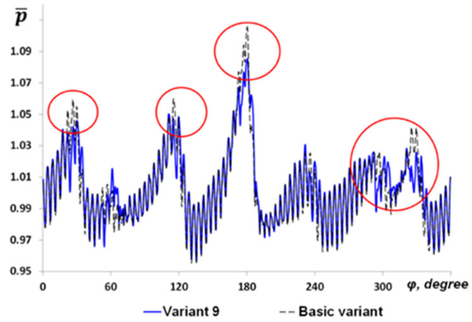


Figure 11: Graphs of relative static pressure variation for basic compressor variant and variant 9.

4.3 Modification of Frame Design

Experimental 13-struts annular frame was made by JSC "Kuznetsov" to reduce the resonant stresses level in the IPC fifth stage. Variable stresses level at resonances with the strongest 12th harmonic in the test with 13-struts annular frame decreased approximately 2 times. Experimental 13-struts annular frame calculation using the method described above confirmed the decrease in the average amplitude of the 12th harmonic (Figure 12) by 1.8 times. This fact is another confirmation of the reliability of the developed methodology.

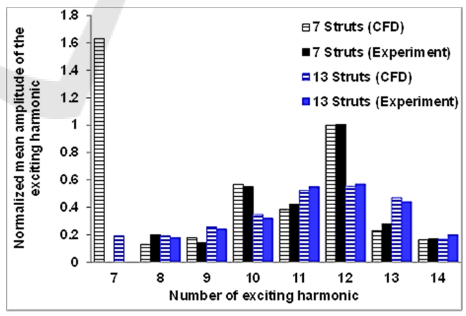


Figure 12: Comparison of excitation amplitudes of harmonics of the basic 7-struts and 13-struts annular frame obtained in experiment and CFD calculation.

Comparison of normalized mean harmonic amplitudes of the basic 7-struts annular frame and experimental 13-struts annular frame obtained in experiments and by CFD is shown in Figure 13. The calculation results were converted into relative values by considering the mean amplitude of 12th harmonic of base 7-struts frame as 1 to compare calculation data with experimental results.

Experimental 13-struts annular frame proposed by JSC "Kuznetsov" specialists allowed to decrease the resonant stresses in the fifth stage rotor blades and to exclude the anti-vibration segment. However, in this annular frame in addition to increasing the struts number, their thickness was reduced. This is critical towards to the struts through which the various engine systems of communications pass. Therefore, the use of this frame on the engine is impossible.

For this reason, it was decided to develop a new design of 13-struts annular frame with saving the position and required cross-sections in the struts No. 1, 2, 4, 5 of 7-struts with engine systems.

When designing the new annular frame, all the previous modifications of the compressor were not considered (different stagger angles and pitches) except the implementation of Shvarov's profile on the rotor blades of the fifth stage.

It was agreed that the development of new annular frame design would be carried out using optimization methods implemented in the IOSO software package (Egorov etc., 2002; Sigma Technology, 2017). Since the use of large-scale 3-D model during optimization requires significant computational resources and time, it was decided to use surrogate struts models.

With this approach, each strut is represented as a pressure peak, which the strut causes in the axial gap between RW5 and GV5. The shape of the peaks is determined by the geometric parameters of the struts, and their location in the circumferential direction is the arrangement of the corresponding struts.

Additional research was carried out (Kolmakova etc., 2015) using struts parametric models (Figure 13) to determine the relationship between the struts geometric dimensions and the peaks' shape. These models allowed to change the struts design taking into account the existing limitations.

As a result, the obtained data array allowed to derive equations for each strut type, reflecting the dependence of pressure peaks' height on the position of the strut leading edges relative to the outlet edges of GV blades located upstream.

The found equations after verification were used to search for a new annular frame design using IOSO.

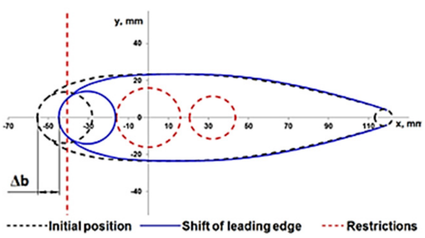


Figure 13: Strut parametric model.

The optimization problem is the following. Struts position and the values of the shift of the strut leading edges are used as variable parameters. Optimization criteria were to reduce the amplitude values of 12th and 10th harmonics (the second largest after 12th). Pareto set was obtained which is the compromise between the decrease in 12th and 10th harmonics. The point of the Pareto set, which gives the maximum decrease in the 12th harmonic amplitude, was chosen as a point for further research.

The annular frame configuration corresponding to the selected point of Pareto set is shown in Figure 14. For this configuration, a CFD simulation was performed using the method described in Section 3 of this article. As a result, the amplitude of the 12th harmonic was reduced (Figure 15).

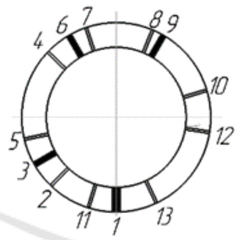


Figure 14: Optimized 13-struts annular frame.

Thus, the change in the annular frame structure makes it possible to significantly reduce the exciting harmonics amplitudes.

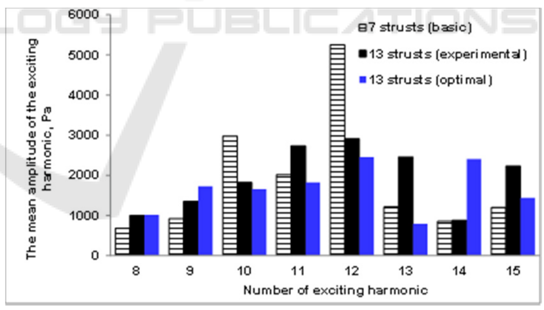


Figure 15: Comparison of exciting harmonics amplitudes of the original 7-struts, 13-struts frame by JSC "Kuznetsov" and optimized 13-struts annular frame.

5 CONCLUSIONS

Several approaches to increase the dynamic strength of GTE compressor rotor wheels were developed. Each of the proposed approaches, possessing a set of advantages and disadvantages, allowed to achieve the set goal which was to reduce the dynamic stresses level in rotor blades. Also, the following aspects can be noted as the obtained research results:

1) The calculation method of dynamic stresses caused by the flow circumferential irregularity is developed;
2) The use of special Shvarov's profile on the RW5 blades makes it possible to reduce the dynamic stresses by almost 2 times;
3) It is revealed that the change in the angular position of the GV5 blades makes it possible to reduce the amplitude of the dangerous 12th harmonic by 2.3 times with the change of 42 blades and by 1.8 times with the change of 14 blades;
4) A technique has been developed to optimize the struts angular position without using complex CFD calculations using surrogate struts models;
5) Different design variants of the engine middle annular frame different number of struts are obtained. With an increase in the number of struts to 13, the amplitude of the most dangerous 12th harmonic is reduced by 2.4 times;
6) For all annular frame variants, the shape and angular position of the struts through which the engine systems pass is saved.

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