Determination of Pressure Fluctuations in Rotor Bundle of Centrifugal Compressor at Critical Conditions of Operation

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1. Introduction
One of the existing problems is unsteady flow phenomena in centrifugal compressors. These phenomena result in significant dynamic loads, which in turn can lead to equipment failures.

Due to fatigue failure of the impellers in the presence of variable aerodynamic loads, there is a necessity of research unsteady processes during the operation the centrifugal compressor. Among the known unsteady phenomena in centrifugal compressors an uneven distribution of flow parameters on the circumferential coordinate near impeller has greatest negative influence on the fatigue strength of the impeller. This is due to the complex flow structure at the outlet of the impeller, influence of the vaned diffuser (VD), inlet guide vanes (IGV) and the return guide vanes (RGV). At rotor rotation circumferential unevenness of flow parameters, stationary relative to the stator is transformed into the unsteady pressure field and velocity relative to the rotating impeller. Importantly, this kind of non-stationarity exists at all operating conditions of the centrifugal compressor, which are allowed during operation. Furthermore, there is a possibility of resonance when the frequency of the exciting aerodynamic load coincides with one of the natural frequencies of impeller, which can lead to a dangerous dynamic stress and fatigue fractures. The main source of forced vibrations in centrifugal compressors are so-called Tyler-Sofrin modes (TS) from the interaction of the rotor and stator elements of the compressor [1]. Also as a result of instability of the gas flow acoustic eigenmodes of the flowing part stage elements can be excited. Critical conditions of centrifugal compressors operation occur if frequencies of Tyler-Sofrin modes coincide with acoustic eigenfrequencies and natural frequencies of structural elements. In addition, Tyler-Sofrin-modes, structural and acoustic modes should be similar in shape.

There is a small number of works devoted to the study of non-stationary processes in the flowing part centrifugal compressors, aimed at identifying the causes of the impeller destruction. However, among the companies that design centrifugal compressors, there are those who have made progress in this direction. [2, 3, 4]. Special attention in these works is paid to acoustic processes taking place in the side cavities of impeller. It is shown that the resonant acoustic modes can cause intense excitation of forced vibrations in the impeller.

2. Determination of critical conditions of compressor operation
When analyzing the causes of the destruction of the impellers critical conditions of operation the centrifugal compressor were determined. We used the so-called resonance approach [5] based on a
comparison of the TS modes frequencies, the natural frequencies of the impeller and acoustic eigenfrequencies.

Research results of the impeller destruction in the third stage of centrifugal compressor are presented below.

Figure 1 shows the object of study - a five-stage centrifugal compressor; design scheme of the third stage, and also fragments of the impeller destruction.

Figure 1. The object of study: (a) - centrifugal compressor; (b) - stage design scheme; impeller; VD – vaned diffuser; IGV - inlet guide vanes; RGV - return guide vanes; SC - side cavity between shroud and housing; HC - side cavity between hub and RGV; (c) and (d) - fracture fragments of impeller.

The main stages of determining the critical conditions of operation the centrifugal compressor are as follows:

- Static stress analysis of the impeller;
- Modal analysis of the impeller;
- Determination of the order of circumferential modes resulting from the interaction between the rotor and the stator, predicting the resonance frequency of rotation;
- Calculation of acoustic eigenmodes in side cavities;
- Plot of Analysis diagrams of harmonic excitatory impacts.

All numerical studies have been performed using ANSYS software.

2.1 Static stress analysis of the impeller
 Calculation of stress-strain state of the impeller performed for rated speed n=5200 rpm with interference fit of impeller on the shaft. Figure 2 shows a 3D model of the impeller and calculated circumferential stress in the impeller.
Analysis of the results of calculations shows that the shroud on the outer diameter has static stress at which the strength of the impeller is provided in accordance with regulatory requirements. This means that intensive forced vibrations should provide the main influence on the fatigue failure of the impeller on this section.

2.2 Modal analysis of the impeller
Calculations were performed to determine the natural frequencies and mode shapes of the impeller.

Table 1 shows two natural frequencies for zero "umbrella" mode shape and mode shapes with 2, 4, 6 and 8 nodal diameters ($ND = m_{\text{struct}}$).

| Nodal diameters ($m_{\text{struct}}$) | Mode | Natural frequencies, Hz |
|--------------------------------------|------|-------------------------|
| 0                                    | 1    | 1103                    |
|                                      | 2    | 2945                    |
| 2                                    | 1    | 1010                    |
|                                      | 2    | 2489                    |
| 4                                    | 1    | 1635                    |
|                                      | 2    | 2369                    |
| 6                                    | 1    | 2002                    |
|                                      | 2    | 2215                    |
| 8                                    | 1    | 1973                    |
|                                      | 2    | 2284                    |

Figure 3 shows the mode shapes of the impeller with 0, 2, 4, 6 and 8 nodal diameters.
2.3 Determination of the order of circumferential modes resulting from the interaction between rotor and stator, predicting the resonance frequency of rotation

Number of nodal diameters forced TS modes is characterized by the order of circumferential modes \( m_{\text{TS}} \), which can be determined from the following equation [2]:

\[
m_{\text{TS}} = h_B Z_B + h_{\text{VD}} Z_{\text{VD}} + h_{\text{IGV}} Z_{\text{IGV}} + h_{\text{RGV}} Z_{\text{RGV}}
\]  

where \( z_B \) – number of impeller blades; \( z_{\text{VD}} \) – number of VD vanes; \( z_{\text{IGV}} \) – number of IGV vanes; \( h_B \) – index, which covers the positive integers (values are taken for calculation - 1, 2, 3), \( h_{\text{VD}}, h_{\text{IGV}}, h_{\text{RGV}} \) – indices covering the positive and negative integers (values are taken for calculation - -3, -2, -1, 0, 1, 2, 3).

TS modes are characterized by a frequency in the rotating reference frame, which are determined by the formula:

\[
\omega^{\text{TS}} = \omega^{\text{shaft}} | h_{\text{VD}} Z_{\text{VD}} + h_{\text{IGV}} Z_{\text{IGV}} + h_{\text{RGV}} Z_{\text{RGV}} |
\]

and order of harmonic impact frequency component, which is defined by the following formula:

\[
m^{\text{TS}} = h_{\text{VD}} Z_{\text{VD}} + h_{\text{IGV}} Z_{\text{IGV}} + h_{\text{RGV}} Z_{\text{RGV}},
\]

where the parameter \( \omega^{\text{shaft}} \) refers to the angular velocity of the rotor shaft in a stationary reference frame. Basic data for the calculation of the order and frequency TS modes are as follows: the number of impeller blades - 18; the number of VD vanes - 36; the number of RGV vanes - 16; number of IGV vanes - 16. The calculation was performed for the compressor condition of operation on the rotation frequency is 4900 rpm. Table 2 shows some of the results of the calculation mode order \( m_{\text{TS}} \), order harmonic frequency impact \( m^{\text{TS}} \) and exciting frequency \( \omega^{\text{TS}} \) at rotation frequency of the rotor 4900 rpm, as well as the predicted resonance rotation frequencies \( \omega^{\text{shaft,rez}} \).

**Table 2.** Main mode orders, harmonic impact frequency orders and harmonic impact rotation frequency at \( n=4900 \text{ rpm} \) and predicted resonance rotation frequencies

| \( m_{\text{TS}} \) | \( h_B \) | \( h_{\text{VD}} \) | \( h_{\text{IGV}} \) | \( h_{\text{RGV}} \) | \( m^{\text{TS}} \) | \( \omega^{\text{TS}} \), Hz | \( f_{\text{st,rez}} \), Hz | \( n_{\text{shaft,rez}} \), rpm |
|---|---|---|---|---|---|---|---|---|
| -6 | 1 | -2 | 0 | 3 | 24 | 1960,00 | 2002 | 5005 |
| -6 | 1 | -2 | 1 | 2 | 24 | 1960,00 | 2002 | 5005 |
| 2 | 1 | 0 | -1 | 0 | 16 | 1306,67 | 1010 | 3787,5 |
| -2 | 1 | -1 | 0 | 1 | 20 | 1633,33 | 1010 | 3030 |
| 8 | 2 | -3 | 3 | 2 | 28 | 2286,67 | 1973 | 4227,86 |
| 8 | 2 | -3 | 3 | 2 | 28 | 2286,67 | 2284 | 4894 |
| 4 | 2 | 0 | -2 | 0 | 32 | 2613,33 | 1652,3 | 3908,06 |
| 0 | 2 | -1 | 0 | 0 | 36 | 2940,00 | 2944,7 | 4907,83 |

Predicted resonant frequencies are determined by the following formula:

\[
n_{\text{shaft,rez}} = 60 \cdot \frac{f_{\text{st}}}{m^{\text{TS}}}
\]

where \( f_{\text{st}} \) - natural frequency of the impeller with the number of nodal diameters corresponding to the order of the TS modes \( m_{\text{TS}} \) in modulus.

When analyzing the predicted resonance rotation frequencies it can be noted that the resonance frequencies for TS modes orders \( m_{\text{TS}} = -6, m_{\text{TS}} = 8 \) and \( m_{\text{TS}} = 0 \) are close to the operating rotation speed \( n = 4900 \text{ rpm} \), therefore they are the main source of excitation of the impeller. Table 2 shows that the resonance rotation frequencies for mode shapes \( m_{\text{TS}} = -6 \) and \( m_{\text{TS}} = 8 \) occur in the complex interaction.
of the impeller (interaction of the impeller with multiple blade elements running of the compressor) with the diffuser vanes, reverse guide vanes and inlet guide vanes. At the same time for zero mode shape rotation frequency occurs in interaction of the impeller with the VD only (simple interaction).

Acoustic eigenmodes in the side cavities at the impeller can also be an additional excitation source of impeller natural frequencies.

2.4 Calculation of acoustic eigenmodes in side cavities

In ANSYS Workbench software calculation of acoustic eigenmodes in the side cavities at the impeller was performed. Example of acoustic eigenmodes is shown in Figure 4.

![Acoustic Eigenmodes](image)

**Figure 4.** The mode shape with 8 nodal diameters $m_{ac} = 8$: a - axonometry, b - a plane section, and zero mode shape $m_{ac} = 0$: c - axonometry, d - flat section.

The rotation of the gas in the side cavities has a significant influence on the value of the acoustic eigen frequencies. Usually it is sufficient to assume that the core gas flow rotates at a circumferential speed being equal to half the circumferential speed of the rotor $\omega_{shaft}$, that is $\omega_{swirl} = 0.5 \omega_{shaft}$ [2].

The resonance condition must be satisfied in the same reference frame, namely in a rotating reference frame. In this case, the frequency of the acoustic eigenmodes must be determined in a rotating reference frame [2]:

$$\omega_{ac} = \omega_{ac}^A + m_{ac} \cdot (\omega_{swirl} - \omega_{shaft})$$  \hspace{1cm} (5)

where $\omega_{ac}^A$ - frequency of acoustic eigenmodes in the reference frame associated with the rotating flow core;

$m_{ac}$ – order (number of node diameters) of acoustic eigenmodes.
The values of acoustic eigenfrequencies in side cavity for different rotational speeds are shown in Table 3.

| Table 3. Acoustic eigenmodes. |
|-----------------------------|
| Acoustic eigenfrequencies, Hz |
| n=3640 rpm | n=4900 rpm | n=5200 rpm | n=5460 rpm |
|-----------------------------|
| \( m_{\omega} = -6 \)    |   |   |   |
| 1 | 1863 | 1926 | 1941 | 1954 |
| 2 | 1996 | 2059 | 2074 | 2087 |
| 3 | 2298 | 2361 | 2376 | 2389 |
|-----------------------------|
| \( m_{\omega} = 8 \)  |   |   |   |
| 1 | 1484 | 1400 | 1380 | 1363 |
| 2 | 2305 | 2221 | 2201 | 2184 |
| 3 | 2377 | 2293 | 2273 | 2256 |
|-----------------------------|
| \( m_{\omega} = 0 \)  |   |   |
| 1 | 2565 | 2983 |
| 2 | 2983 |
| 3 | 3117 |

2.5 Plot Analysis diagrams of harmonic excitatory impacts
Analysis of the interaction between the mechanical natural frequencies, TS modes and acoustic modes is convenient to do using analysis diagrams of harmonic excitatory impacts.

Analysis diagrams of harmonic excitatory impacts are construct for mode orders \( m_{TS} = -6 \) and \( m_{TS} = 8 \) and shown in Figure 5.

![Figure 5. Analysis diagrams of harmonic excitatory impacts for mode orders \( m_{TS} = -6 \) and \( m_{TS} = 8 \).](attachment:image.png)

The frequency in the rotation reference frame is displayed on the horizontal axis of abscissas. The rotor speed is shown on the vertical ordinate axis. The red vertical lines indicate the natural frequencies of the impeller, which are assumed to be constant for different rotor speeds. Blue lines indicate the frequency of exciting TS modes for the first, second and third harmonic component. Pink lines indicate the frequency of acoustic eigenmodes in the rotation reference frame in the side cavity SC. Green lines limit the range...
of compressor operation \((n_{\text{min}}=3640\ \text{rpm} \text{ and } n_{\text{max}}=5460\ \text{rpm})\) and two conditions of operation \((n=5200\ \text{rpm} \text{ and } n=4900\ \text{rpm})\).

Analysis diagram of harmonic excitatory impacts for zero mode shapes is shown in Figure 6.

![Figure 6. Analysis diagram of harmonic excitatory impacts for zero mode shapes.](image)

Critical condition of compressor operation corresponds to point of intersection of the blue and red lines. The diagram indicated in figure 5a shows that for mode shape with order \(m_{\text{struct}}=6\) condition of compressor operation is operation on the rotor speed of 5005 rpm, which is very close to the operating speed of 4900 rpm. In this case, the first natural frequency of the impeller for sixth mode shape is excited. For mode shape with 8\(^{th}\) order \(m_{\text{struct}}=8\) (see figure 5b) the critical condition of operation is operation on the rotor speed of 4894 rpm, but second natural frequency is excited. Analysis diagrams in figure 5a and 5b show that the additional excitatory impact on the natural frequencies of the impeller have acoustic eigenfrequency (pink line) in the side cavity, which lie near the critical conditions of operation compressor.

For zero mode shape (see figure 6) a critical condition of compressor operation is also operation on the rotor speed of 4900 rpm and second natural frequency is excited. It also shows that acoustic eigenmodes create additional impact in the side cavity SC. It is also necessary to note that critical conditions of compressor operation appeared on one of the rotor speed - 4900 rpm, and this significantly reduced the number of fluid dynamics calculations in transient formulation.

3. **Transient CFD Analysis of compressor stage**

Existing critical conditions of compressor operation suggest a possible cause of impeller destruction, but not assert that the cause of the destruction is found. Therefore, transient problem of viscous flow in the compressor stage has been solved that evaluate risk of critical operation conditions. The goal solving the problem is the definition of pressure fluctuations and finding the amplitude and frequency of the fluctuation.

Transient CFD problem was solved using ANSYS-CFX software. To solve the problem the third stage of the compressor was taken (see, figure 1b), in which radial vanes of RGB from second stage were used as an IGV. 3D model of the third stage jointly with second stage RGV vanes is shown in figure 7.
The total pressure of 1,114 MPa and the total temperature of 288 K are set as the boundary conditions at the inlet to the stage. The mass flow rate of 62.23 kg/s was set in the outlet stage. Methane gas similar in properties to a real gas was used for the working fluid.

Finite element characteristic of stage is shown in figure 8.

Transient CFD calculation was performed for 15 of the impeller revolutions. Time interval discreteness corresponds to one degree of rotation angle of the rotor to provide sufficient resolution when analyzing the pressure distribution within a single revolution of the rotor.

Several weeks of computer calculations were required to produce stable solution for the selected finite element mesh breakdown.

Figure 9 shows the instantaneous pressure distribution for the meridian section of the test stage of the compressor.

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**Figure 7.** 3D model of the third stage: a - view from the inlet of the gas flow; b - view from the outlet of the gas flow.

**Figure 8.** Finite element characteristic.

**Figure 9.** The instantaneous pressure distribution for the meridian section of the test stage of the compressor.
Figure 10 shows the instantaneous pressure distribution for outer surface of the impeller shroud and instantaneous pressure distribution for the middle section of the impeller and the VD.

![Figure 10](image)

**Figure 10.** Instantaneous pressure distribution: a - for the outer surface of impeller shroud; b - for the middle section of the impeller and VD.

Pressure fluctuations for two points in the stage cavity was analyzed as a result of the transition solution, as shown in figure 11. The first point is located at a distance 0.97D2 in the side cavity at the shroud, and the second point is located above the impeller at a distance 1.01D2.

![Figure 11](image)

**Figure 11.** Location points in the stage cavity for the analysis of pressure fluctuations.

Fast Fourier Transform procedure for the instantaneous pressure distribution at the same time interval discreteness for these points was performed for numerical analysis of pressure fluctuations. The resulting amplitude for each circumferential mode order from 1 to 64 is shown in figure 12a and 12b for the two points, respectively.

![Figure 12](image)

**Figure 12.** Harmonic analysis results for the two points obtained by the calculation data in accordance with transient CFD.
As the results of harmonic analysis (see figure 13), there is practically no pressure fluctuation amplitude for the mode orders $m_{TS}=-6$ and $m_{TS}=8$. Low levels of amplitudes can be explained by the fact that the forced TS modes of these orders occur from a complex combination of several elements of the stator and rotor components of the stage, as shown in the Table 2.

Analysis of the results showed that most dangerous critical conditions of compressor operation is critical conditions of operation for zero mode order $m_{TS}=0$. The frequency of forced vibrations is equal to a multiple of the first blade blade frequency of 2940 Hz (see figure 13) and practically coincides with the acoustic eigenfrequency of 2983 Hz and natural frequency of the impeller of 2944.7 Hz. It is also worth noting that the zero mode shape of forced TS modes occurs from the interaction of impeller ($h_b = 2$) and the VD ($h_{VD} = -1$) only.

4. Conclusions
Using the Fast Fourier Transform, the processing of the results of numerical calculation in accordance with the procedure CFD, determination of the amplitude of the pressure fluctuations was performed. This amplitude affects the impeller for the finding a critical condition of compressor operation, the value of which will be used to evaluate the fatigue strength of the impeller.

5. References
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