A Frequency Model of Vibrational Processes in Gas-Turbine Drives of Compressor Stations of Main Gas Pipelines

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Abstract. At compressor stations, systematic measurements of noise and vibration of power equipment - gas compressor units - are carried out. The article presents basic equations for calculating natural and forced frequencies at which the main defects appear. According to the studied dependences, results of calculations are obtained on the following types of drives for gas-compressor units GTK-10-4, Avon-1534, DG-90.

1. Introduction

If vibration levels exceed the allowable, breakage of parts and components may occur, leading to failures of power equipment. The presence of excessive vibration and noise indicates the presence of various defects, which are one of the reasons for the accelerated wear and fatigue failure of parts. Measuring vibration and noise during operation can detect many defects at their early stages, prevent possible failures of equipment, as well as put it into operation and repair according to its actual state, instead of conducting planned preventive maintenance.

Methods of vibroacoustic diagnostics provide ample opportunities. They are based on the use of information contained in the oscillatory processes that accompany operation of machines and mechanisms. Vibroacoustic diagnostics is one of technical diagnostics sections. As a source of information, it uses not a series of static parameters characterizing the state of a mechanism, but a number of dynamic parameters causing the occurrence of vibration and acoustic waves. Vibration signal providing capacious enough information on the work of the unit and its components can be a good indicator of its status.

2. Research

Vibration and noise of the axial compressor (AC) of gas compressor units (GCU). The vortex noise of the compressor and the high-pressure turbine (HPT) arises from vortices occasionally shedding from the blades, the turbulent flow, and the turbulent jet. The spectrum of the vortex noise caused by periodic vortex shedding is determined from the equation [1, 2, 3]

\[ f = 0.212 \left(1 - \frac{21.2}{Re} \right) \frac{D}{d} \]  

(1)

where \( Re = \frac{Vd}{\nu} \) - the Reynolds number; \( d \) – the determining linear dimension; \( V \) – the linear flow velocity, \( \nu \) - the viscosity of the medium.

The level of wideband noise can be determined according to the equation:
\[ \alpha = 10 \log \left[ k'(B\Delta T)^{1.5} \left( \frac{B\Delta T}{z^{0.27} \text{Re}^2} \right) 10^{1.5} \right], \]  
(2)

where \( \alpha \) - the sound pressure level, \( B \) – the flow of steam through the HPT or air through the AC, \( \Delta T \) – the temperature differential in the HPT or AC, \( z \)– the number of blades, \( k' \) - the coefficient determined from the experimental data on the measured noise level and the geometric characteristics of the AC or HPT.

The level of noise of the discrete components [1, 2, 3]:

\[ \alpha = 10 \log \left[ k' B^2 \eta \Delta T / \left[ D_p^2 (1 - d_h^2) \text{Re}^2 \right] 10^{1.5} \right], \]  
(3)

where \( D_p \) – the impeller diameter, \( d_h \) – the hub-tip ratio, \( \eta \) - the efficiency of the first stage, \( k'' \) - the coefficient determined experimentally.

For multistage compressors, the acoustic power is determined by summing the power of each stage.

During operation, the AC rotates in a pulsating flow, whereby each blade transfers momentum to the working fluid that is a multiple of the number of blades. When summing these impulses and subsequently transmitting to the body a vibration is excited with frequencies that are multiples of the number of blades [1, 2, 3]:

\[ f_{bi} = i \cdot z \cdot f_p \]  
(4)

where \( i \) – the harmonic number, \( i=1,2\ldots n \), \( f_p \) – the rotor rotational frequency. The amplitude of the blading vibration depends on the steam or air density (the ratio of expansion or compression pressures) and some other factors; the smaller the density, the smaller the aerodynamic imbalance of the working fluid flow. The vibration frequency during a developed shedding is determined by the equation [1, 2]:

\[ f_{sp} = (0.3 \div 0.6) f_p \]  
(5)

**Vibration of the rotor.** The vibration state of the AC rotor can best be determined by the speed characteristics when accelerating or running down the rotor.

The oscillatory system rotor-body may be in a certain critical condition in the presence of a number of critical rotational frequencies of the rotor. These frequencies can be higher, lower or within the range of the operating frequency, which is highly undesirable. Reducing the vibration amplitude in this case is achieved by using different damping devices or by balancing.

The critical rotational frequencies of the rotor are calculated by the equation [4]:

\[ f_{crit} = K \sqrt{\frac{EI}{L^2}} \]  
(6)

where \( K \) – the coefficient obtained as a result of calculating the system of equations of small oscillations of the rotor, \( E \) – the modulus of elasticity, \( I \) – the reduced moment of inertia, \( L \) – the rotor length.
For the axial compressor blades, the range of static natural frequencies of the first three bending forms of vibrations in the plane of its least stiffness is determined by the equation of I.M. Meerovich:

\[
f_{\text{bend}} = 0.0398\alpha_{\text{bend}}\beta_{\text{bend}} \frac{\delta_k}{\rho} \sqrt{\frac{0.8\psi^2\Delta_i}{\gamma_i^2(1 + 4\Delta_i\phi_0^2)}} \tag{7}
\]

where \(\alpha_{\text{bend}}, \beta_{\text{bend}}, \psi, \gamma, \Delta_i\) – for each waveform there is a wedge, curvature and radial function, \(a_i\), \(\delta_k\) – the maximum curvature and thickness of the blade profile at the root section, \(\phi_0\) – the feather twist (the angle between the chords of the root and peripheral sections), \(l\) – the length of the blade feather.

The spectrum of the natural frequencies of torsional oscillations of the axial compressor rotor blades is determined by the equation [1, 2]:

\[
f_{\text{cr}} = 0.343\alpha_{\text{cr}}\beta_{\text{cr}} \frac{\delta_k}{b_{cp}l} \sqrt{\frac{G}{\rho}} \tag{8}
\]

where \(\alpha_{\text{cr}}, \beta_{\text{cr}}\) – the coefficients determined by [1, 2], \(b_{cp}\) – the width of the blade chord in the middle part, \(G\) – the shear modulus.

The rotor unbalance, bending by the first form, the thermal instability of the rotor occur at a frequency [1, 2, 3]:

\[
f_{o.r} = f_p \tag{9}
\]

The uneven stiffness of the rotor cross-section, the resonant oscillations of the rotor blades due to loose connections, the misalignment of couplings, the rotor bending by the second form occur at a frequency [1, 2]:

\[
f_{p.k} = 2f_p \tag{10}
\]

Rotor elements touching the stator, separation of supports from the foundation [1, 2]:

\[
f_{z.r} = (2.5 \div 2.8) f_p \tag{11}
\]

**Vibration and noise of the bearings.** This noise is an indication of a malfunction, since it is not possible to determine it if the bearings are serviceable. The main defects of the bearings are the waviness on the bearing races, the increase in the radial clearance, oval balls and rings, the unroundness of rolling elements, increasing clearances in the sockets of separators, which we denote by (b). Then the forced frequency of faults is determined by the general equation:

\[
f_n = b f_p \tag{12}
\]
The value (b) is determined by the geometric dimensions of bearings. Specific equations are given in [1, 2, 3].

The frequency content of the vibration spectrum of the defective bearing depends on the type of defect. Working with defective bearings we connect the five main bearing frequencies:

1 – \( F_r \) - the rotor rotational frequency;
2 – \( F_{sep} \) - the separator rotational frequency;
3 – \( F_{o.r} \) - the ball (roller) frequency of passing the outer ring;
4 – \( F_{i.r} \) - the ball (roller) frequency of passing the inner ring;
5 – \( F_{r.b} \) - the frequency of natural rotation of the ball, roller.

The frequency of natural ball rotation is excited when the defect on the ball or roller meets the race track. The frequency may equal 2FTK, since its defect meets both rings. Excited frequencies seldom reach such high values as:

1. the ball is not always in the loading zone when the defect meeting the rings, and
2. the energy is lost when the signal passes through additional structural joints, when the defect meets the inner ring.

The frequency of natural rotation is also excited when the balls are pressed against the separator or the separator is broken. The separator rotational frequency is equal to the rotational speed of a set of balls or rollers and the separator around the axis of the shaft.

**Noise of the reduction box gear wheels.** This noise occurs when the gears collide in the short-term contact during operation. The short-term contact leads to the appearance of stress that is cyclical in nature and in turn causes stress wave propagation along the gear wheel and the emergence of noise. The level of noise of gear wheels depends on their residual imbalances, compression of air and oil, torsional oscillations of shafts, irregular shock loads, and uneven teeth surface.

The frequency of contact meshing noise is determined by the equation [1,2]

\[
f_z = i \cdot z \cdot f_p
\]  

where \( z \) – the number of teeth.

The natural oscillation frequency is determined by the formula [1,2]:

\[
f_c = \frac{\lambda}{(2 \pi)} \sqrt{\frac{c}{m}},
\]  

where \( \lambda \) - the factor considering the system flexibility; \( c \) - the teeth stiffness, \( m \) - the reduced gear wheel mass.

**Vibration and noise of the combustion chamber.** Due to the fact that the combustion chamber is located behind the air intake in the GTU and AC, its noise is the resultant noise caused not only by the combustion process, but also by the previous process. The combustion chamber noise, as well as the AC, is continuous in nature and has a wideband range. The most dangerous is the vibrating combustion that occurs as a result of strong vortices in the flow of gas mixture.

A stable combustion mode gives the maximum value of amplitudes in the low frequency zone of 40 ÷ 60 s\(^{-1}\), and during the vibrating one the amplitude shifts to the zone of 200 ÷ 300 s\(^{-1}\) and above. Flame tubes of the combustion chamber of the investigated GCU are a cylindrical shell simply supported at the ends, so the approximate value of frequencies of natural oscillations is described by the equation [1, 2]
where $\bar{\omega}$ - a dimensionless frequency of the normal mode of vibration supported at the ends of the cylindrical shell; $R$ – the radius of shell curvature; $E$, $\mu$, $\rho$ - the modulus of elasticity, the Poisson's ratio and the density of the shell material.

**Vibration and noise of the GCU turbine rotor.** The first form of static natural bending vibrations for turbine blades is determined by the equation of A.E. Shneydman

$$f_{ct} = \frac{\bar{\omega}}{R} \sqrt{\frac{E}{\rho(1-\mu^2)}} \quad (15)$$

where $\bar{\omega}$ - a function depending on the area and moments of inertia of the cross sections of the blades.

The relationship between the static and dynamic ($f_d$) natural oscillation frequency of the blades [1, 2]:

$$f_p = \sqrt{f_{ct}^2 + \beta f_d^2} \quad (16)$$

where $\beta$ - the dynamic frequency coefficient determined from the literature data [1, 2].

Table 1 shows the results of the calculation of natural and forced oscillation frequencies of main units of the gas generator Avon-1534 at $f_p=125$ Hz.

**Combination frequencies.** In addition to the abovementioned fundamental frequencies excited on the body of a product by rotors, blades, bearings, a gearbox and drive units, the occurrence of combination frequencies is possible, the general formula of which is given by:

$$f_k = K_i \cdot n_i + K_j \cdot n_j \quad (18)$$

where $n_i$ and $n_j$ - fundamental frequencies, $K_i$ and $K_j$ - integer-valued coefficients.

A combination frequency corresponds to a certain physical model of excitation, different for each individual case.

**Table 1.** An example of calculation of the frequency components of vibration processes in the main units of the gas generator Avon-1534.

| Axial compressor | Stages | $f_{bb}$, Hz | $f_{cp}$, Hz | $f_{crit}$, Hz | $f_{bend}$, Hz | $f_{cr}$, Hz | $f_{br}$, Hz | $f_{pk}$, Hz | $F_{m}$, Hz | $f_{zr}$, Hz |
|------------------|--------|--------------|---------------|----------------|----------------|-------------|-------------|-------------|-----------|-------------|
| 00               | 2750   |              |               | 196            | 865            | 125         | 250         |             | >250-55000 | >312-360    |
| 0                | 3625   | 375-75       |               | 198            | 984            |            |             |             |           |             |
| 1                | 4625   |              |               | 201            | 1187           |            |             |             |           |             |
| 2                | 6625   |              |               | 216            | 1267           |            |             |             |           |             |
| 3                | 7875   |              |               | 219            | 1290           |            |             |             |           |             |
| 4                | 9125   |              |               | 270            | 1536           |            |             |             |           |             |
| 5                | 11125  |              |               | 298            | 1864           |            |             |             |           |             |
| 6                | 12125  |              |               | 323            | 1898           |            |             |             |           |             |
3. Conclusion

With the help of the vibrating diagnosing method, a direct control of the dynamic force action is carried out, making it possible to detect and prevent the development of a defect at an early stage. So, damage to individual elements of the gas turbine unit causes an instantaneous change in the level and frequency characteristics of the vibration spectrum. The system of diagnostics by the frequency model will increase the service life of individual components, reduce the number of unscheduled repairs and downtime, and improve the reliability and efficiency of gas compressor units in general.

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