Research into the impact of shafts misalignment of turbocompressor installation on power characteristics of a drive engine

M V Kipervasser and A V Gerasimuk

Siberian State Industrial University, 42 Kirova Street, Novokuznetsk, 654007, Russia

E-mail: kipervasser2012@yandex.ru

Abstract. In the present paper the emergency situation in operation of a turbocompressor installation is considered, which leads to shafts misalignment, connected by rigid couplings, of the drive synchronous electric motor and multiplicator is considered. Misalignment of shafts causes vibrations in the course of work of a turbine unit that, in its turn, can result in untimely wear of bearings, loosening of different fasteners and breakdown of the whole installation. The analysis of shafts misalignment of electric motor and planetary gearbox (multiplicator) regarding the emergence of mechanical power surge and derivation of corresponding equations is performed. The mathematical modelling of a turbine unit operation taking into account misalignment in MatLAB Simulink is carried out. The graphs of mechanical power surges on the shaft of the electric motor and current surges in the phase of its stator are received. The conclusion about the possibility of recording stator current surges by standard measurement methods and use of this fact for development of a current protection against misalignment of shafts is made.

1. Introduction

Turbocompressor installations are widely used at the mining and metallurgical enterprises. The compressor machines of centrifugal type, intended for pumping of large volumes of air or gas with a rather small pressure (0.15 – 1.0 MPa), are usually used. Turbocompressor installation usually is a high-speed and energy-intensive unit. Capacities of synchronous motors reach 12.5 MW, and rotation rate of turbocompressors wheels in some cases make 30000 rpm. Similar capacities and rotation rates condition high requirements to quality of installation and pre-commissioning activities.

One of the moments requiring especial attention is alignment procedure of turbocompressor and planetary gearbox shafts, and also - planetary gearbox and the electric motor. At high rotation rates the shafts misalignment leads to vibrations which negatively influence the whole installation. Vibrations result in the accelerated wear of bearings, loosening of bolted connections, failure of the connection couplings. Besides, during start-up of turbocompressor installations the emergence of the resonance mode of natural oscillation frequency and oscillations caused by shafts misalignment can lead to a severe accident.

Control of the installation vibrations is performed by means of vibration diagnostics. However, as practice shows, vibration control is not always applied. Sensors are a weak link of the monitoring system. Their failure paralyzes the system; it either becomes not operational, or loses its sensitivity. It is reasonable to use such monitoring system where the vibration sensors would be absent per se.

Content from this work may be used under the terms of the Creative Commons Attribution 3.0 licence. Any further distribution of this work must maintain attribution to the author(s) and the title of the work, journal citation and DOI.
Published under licence by IOP Publishing Ltd

2. Calculation of additional load torque

For detection of shafts misalignment it is possible to use the indirect method of emergencies recognition based on current oscillations of the drive engine stator [1]. Misalignment of shafts creates some additional load torque on a shaft leading to the current oscillations of the drive engine stator which can be recorded [2].

In Figure 1 the emergence of shafts misalignment in the horizontal plane during assembly of the turbocompressor unit is shown. Shift in horizontal direction makes some value "s". In the initial point of time, before the beginning of the rotation, the shafts in the place of connection keep their the in-line state.

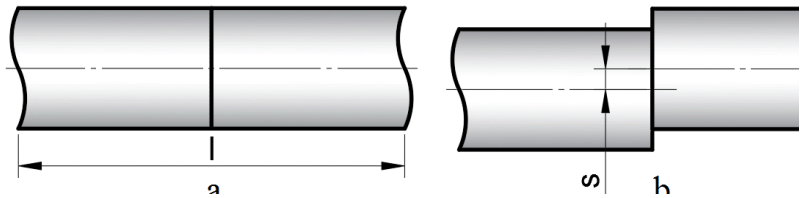


Figure 1. Position of the shafts The provision of shaft prior to rotation: a – side view, b – top view.

After energising the engine and the beginnings of rotation, due to misalignment of axial lines and under the influence of elastic forces, the rigidly connected shafts begin periodically bent at some value *f* (Figure 2). At the same time, the maximum value of the bend will be observed during rotation by 180 degrees.

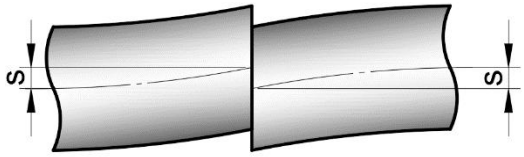


Figure 2. Position of the shafts during rotation by 180 degrees relative to the initial position. Top view.

The value of shafts deflection, with their lengths and diameters being equal, will be:

$$f = \frac{2s}{2}, \tag{1}$$

where *f* – the deflection value, m; *s* – misalignment value of axial lines of shafts, m.

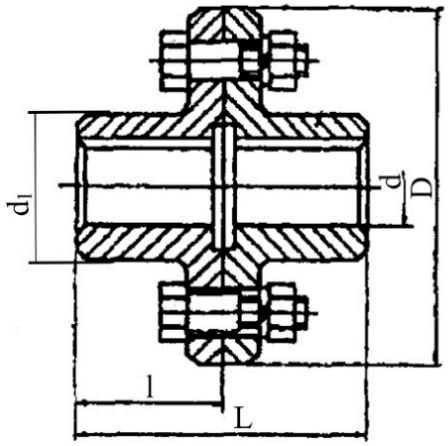


Figure 3. Drawing of a flange coupling.

It is necessary to consider that couplings are fitted over the shafts of the engine and planetary gearbox (it is accepted that the rigid flange couplings are used since when using other types of couplings, e.g., gear-type coupling, some misalignment of the shafts is compensated by the coupling design features) and the shaft as well as coupling bend (stretching of the connecting bolts is neglected). The drawing of the coupling is presented in Figure 3 [2].

The shaft deflection arises under the influence of bending force F defined as follows [3]:

$$F = \frac{48EIf}{L_f^3}, \tag{2}$$

where E – elasticity module equals $2 \cdot 10^6$kg/cm^2; I – equatorial moment of inertia of a solid shaft, m^4; f – shaft deflection, m; L_f – full length of couplings and a shaft (from the engine bearing to the reducer bearing), m.

$$L_f = L + 2\left(l_{48} - l_1 - 168\right) \tag{3}$$

where L = 250 – full length of the coupling, mm; l_{48} = 435, l_1 = 210 – the sizes according to the drawing of the engine, (Figure 4), mm.

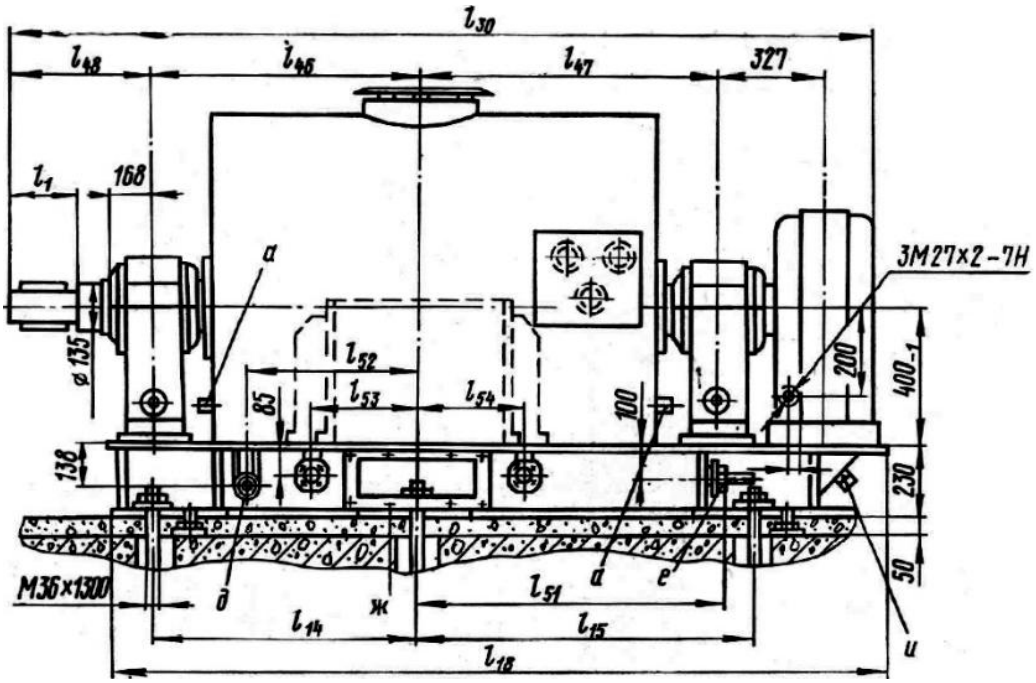


Figure 4. Drawing of the STD-1600 engine.

$$I = \frac{d_1^4}{64}, \tag{4}$$

where d_1 – diameter of the coupling, m (see Figure 3).

During shaft rotation the process of bending will have a periodic character and, obviously, will be described by function of a sine (provided that in initial position the shaft was in the non-bent state).

Then, taking into account formulas (2) – (4) it is possible to write down the bending force function on the shaft rotation angle:

$$F\left(\theta\right) = \frac{3Ef\pi d_1^4}{4L_f^{\,3}}\sin\left(\theta\right), \tag{5}$$

where θ is a shaft rotation angle.

Dependence of a bending force on time is important for research. The variable dependent on time is the rotation angle:

$$\theta(t) = \omega t = 2\pi n t, \tag{6}$$

where n – shaft roation rate, rps; t – time, s.

Let us find the work, which the force F performs during the shaft rotation. To make one rotation the following work should be done:

$$A = \frac{d_1}{2}\int_o^{\theta} F\left[\theta(t)\right] d\theta(t). \tag{7}$$

The mechanical power, energizing the shaft and allowing it to perform one rotation, can be calculated by the formula:

$$P_M(t) = \frac{d_1 n}{2}\int_o^{\theta} F\left[\theta(t)\right] d\theta(t). \tag{8}$$

Taking into account formula (7) formula (8) will take the form:

$$P_M(t) = \frac{3Ef\pi d_1^5 n}{8L_f{}^3}\int_o^{\theta} \sin\left[\theta(t)\right] d\theta(t). \tag{9}$$

After solution of the integral we will receive the following expression:

$$P_M(t) = \frac{3Ef\pi d_1^5 n}{8L_f{}^3}\left\{1-\cos F\left[\theta(t)\right]\right\} = \frac{3Ef\pi d_1^5 n}{8L_f{}^3}\left[1-\cos\left(2\pi n t\right)\right]. \tag{10}$$

Upon transition to dimension of rotation rate – rpm, formula (10) will take the form:

$$P_M(t) = \frac{Ef\pi d_1^5 N}{160L_f{}^3}\left[1-\cos\left(\frac{\pi N t}{30}\right)\right]. \tag{11}$$

Besides the power consumption for a periodic shaft bend, during the bend the losses of power due to bearing friction increase.

Dependence of friction force in the bearings on time has the form:

$$F_B(t) = z\mu F(t), \tag{12}$$

where z – number of bearings; μ – friction coefficient of bearings.

The power supplied for bearings friction is described by the expression:

$$P_B(t) = z\mu\frac{Ef\pi d_1^5 N}{160L_f{}^3}\left[1-\cos\left(\frac{\pi N t}{30}\right)\right]. \tag{13}$$

Then, total additional losses of power due to misalignment of two shafts will be:

$$P(t) = \frac{Ef\pi d_1^5 N}{160 L_f^{\ 3}}\left[1-\cos\left(\frac{\pi N t}{30}\right)\right](1+z\mu) \quad (14)$$

To obtain data on behavior of electric parameters of the drive electric motor in the situation of shafts misalignment the research of mathematical model was conducted. Mathematical modelling of operation of the turbocompressor installation being a part of compressor K-250-61-1 and electric motor STD-1600 with the planetary gearbox P1500/3,69 was carried out in the environment of MatLAB Simulink. Implementation of formula (14) by the standard blocks from Simulink library is presented in Figure 5.

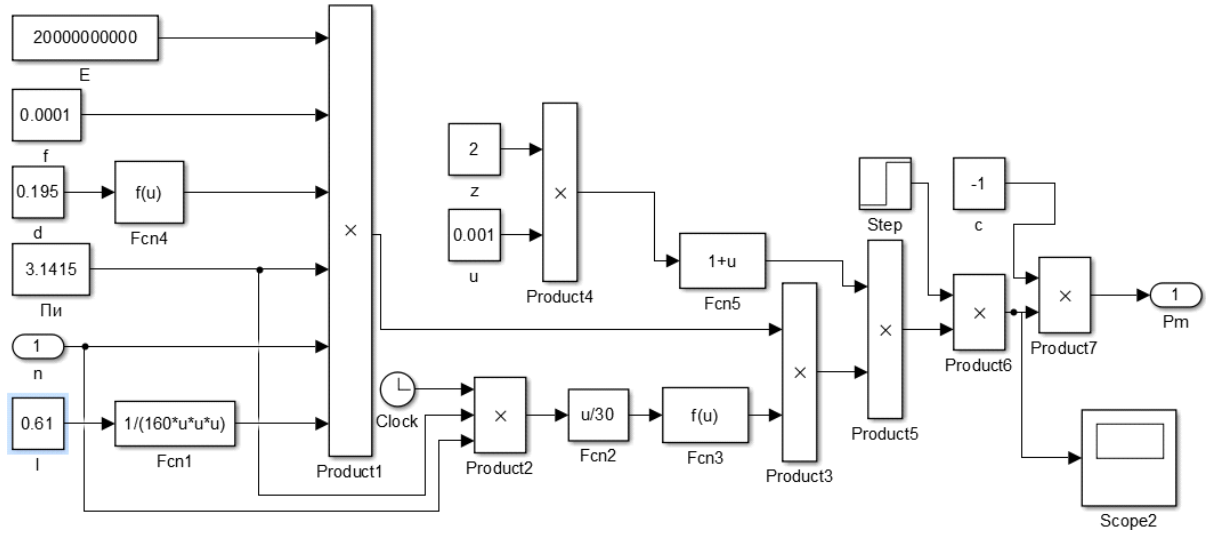


Figure 5. Functional scheme of imitation block of shafts misalignments.

The functional diagram of a model of the compressor unit, misalignment of shafts and the supply mains is given in Figure 6.

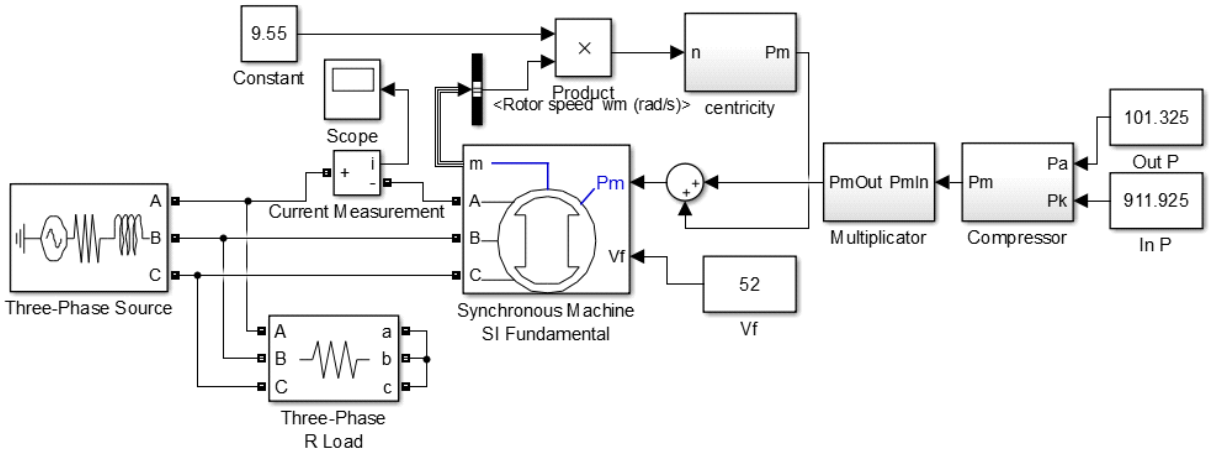


Figure 6. Functional structure of the mathematical model of the compressor unit and supply mains.

When modeling the shaft misalignment value of the engine and planetary gearbox was set 0.1 mm. The similar misalignments can be observed in real situation of turbocompressor installation

functioning. The length of the free shaft end of the engine, according to the data of the producer, is 0.21 m, and diameter – 0.13 m. The diameter of the connecting couplings – 195 mm, and the full length of the coupling is 0.5 m. The parameters of the shaft of the planetary gearbox are accepted similar to the engine shaft. Modelling was carried out for the mode of nominal performance and air pressure [4]. The mode of misalignment arises on the third second of modelling.

3. Results of modelling

Figure 7 provides the current diagram of one of the electric motor phases during shafts misalignment, and Figure 8 shows the diagram of total additional losses of power due to misalignment.

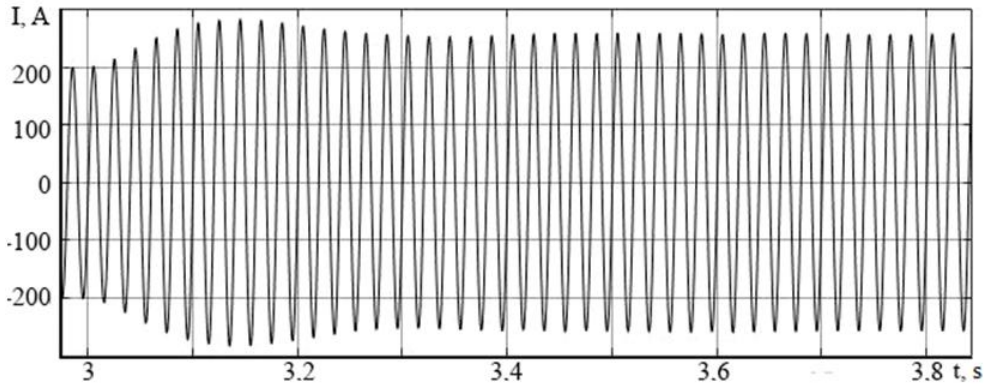


Figure 7. Current diagram in the phase of the drive engine during shafts misalignment.

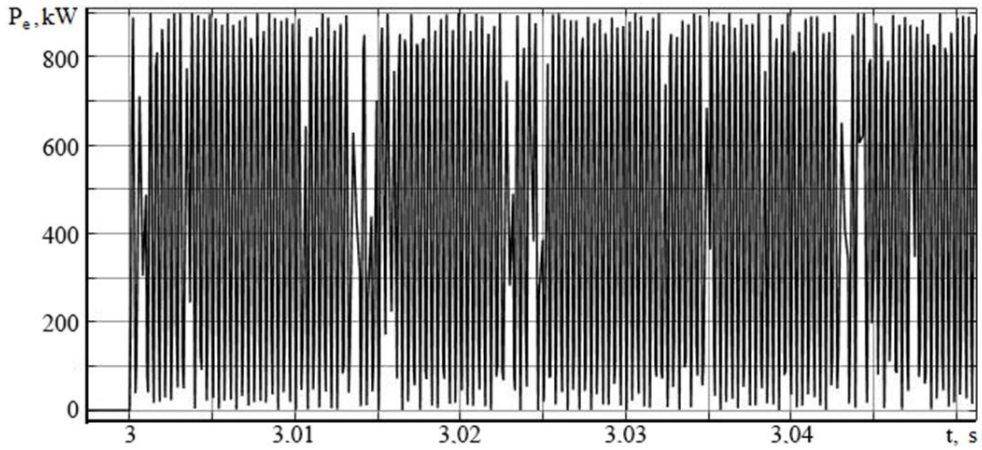


Figure 8. Diagram of mechanical power surge due to misalignments of shafts (the timescale is increased).

According to the diagrams the misalignment of shafts creates periodic (confirmed by the analysis of formula (13)) mechanical power surge with an amplitude up to 800 kW at the set parameters, that makes 50% of the engine nominal performance. Amplitude of the phase current surge of the electric motor equals to 290 A, that is 45% of the rated current and in the course of attenuation of transition process decreases to 255 A (28% of the rated current). Duration of transition process is 0.3 s.

4. Conclusions

On the basis of the obtained data it is possible to conclude that the value of current surge is sufficient enough to be registered by standard methods of measurement, and, therefore, it is possible to apply the method of indirect diagnosing of emergency situation [1] and to create the protection system against shafts misalignment as one of the components in the protection complex of turbine units.

References

[1] Pugachov E V, Kipervasser M V and Inzhelevskaya O V 2011 *Bulletin of KuzSTU* **4** 51–55.
[2] Pugachyov E V, Kipervasser M V and Gerasimuk A V 2016 *Bulletin of KuzSTU* **2** 84–90
[3] *Russian National Standard: Flange Couplings 20761-96* pp 2–11
[4] Postnikov I M 1960 *Design of Electrical Machines* (Kiev: GITL) p 274
[5] Sidorov M D 1962 Reference Book on Air and gGs-blowing (Moscow-Leningrad: Mashgiz) p 220