Detection of centrifugal turbocompressor surge by drive motor current

M V Kipervasser¹ and A V Gerasimuk²

¹Siberian State Industrial University, 42 Kirova str., Novokuznetsk, 654007, Russia
²AO “Siberian Tyazhpromelektroproekt”, Novokuznetsk, 654000, Russia

E-mail: kipervasser2012@yandex.ru

Abstract. The article deals with the case of surging of a turbocompressor unit. Mathematical modeling of the situation under consideration is carried out and its influence on the stator current of the electric motor is estimated. The analysis of the simulation results with the corresponding conclusions is made. A surge detection device is proposed.

1. Introduction

When conducting drilling operations in iron ore mines, the energy of compressed air is used. The latter is mainly ensured through the use of centrifugal turbocompressors at compressor stations of mining enterprises. One of the emergency situations that can disrupt the operation of compressor units is surge. The emergency situation is accompanied by significant changes in pressure and volume of gas or air mixture supplied by consumers. The oscillation frequency ranges from fractions to units of hertz. The reasons for the surge are a large pressure difference in the suction and discharge pipelines and a significant decrease in gas consumption in the network [1, 2]. If the discharge pressure exceeds the pressure in the suction chamber by more than a certain critical value, a reverse flow of the working mixture appears (flow-in of the working mixture). Any centrifugal compressor has a so-called surge limit. Under certain conditions, fluctuations in the direction and magnitude of the gas mixture can be accompanied by resonance phenomena. The possibility of resonances is higher in systems that use several turbocompressors operating to supply common consumers or one common network [1, 2].

Surge causes strong vibration and heating of the turbocompressor components. The electric motor, multiplier and impeller parts are subjected to shock loads. In case of untimely obstruction of the developed surge phenomenon, the listed elements may fail for various reasons and with varying degrees of destruction. For individual components and parts and the unit as a whole, surge is an emergency mode, causing the accelerated wear and destruction of compressor elements, leading to failure of valves, pipelines and the turbocompressors themselves.

To protect against surge, turbochargers and associated networks are protected by special devices and fixtures. As such, by-pass valves, throttles, automatic control of compressor values during operation are used in order to prevent the development of surge.

2. The main approaches to the creation of anti-surge protection systems

There are two main types of surge protection systems [3]:

- anti-surge control systems that control the position of the compressor operating point on the gas-dynamic characteristic and, when it moves to the surge zone, initiating the start of anti-surge control (bypass, pressure relief into the atmosphere) [4-8];
• systems that respond to the characteristic signs of an already occurring surge. These include fluctuations in flow and pressure. The devices provide a signal to start anti-surge control or stop the compressor [3].

In [9], an approach is proposed to the creation of a device for detecting emergency situations of turbochargers using a drive motor as a diagnostic feature of the stator current. During surge, fluctuations in air flow occur through the compressor and fluctuations in pressure in the supply and suction pipelines. This leads to a change in the mechanical power consumed by the compressor. The electric motor reacts by changing the phase current, generating harmonic components of various orders in the network to the start of surge.

The mechanical power of the unit is [10, 11]:

\[ N_C = N_{CC} + N_{CD} \]  

where \( N_{CC} \) – the power consumed by the compressor to increase the gas pressure and to cover the friction losses in the units of the mechanism, kW; \( N_{CD} \) – the dynamic component of the unit’s power that appears during periods of change in the drive speed, kW.

The first term (1) is determined [10, 11]:

\[ N_C = \frac{P_{ATM} Q_C \ln \left( \frac{P_C}{P_{ATM}} \right)}{\eta_{IS} \eta_M} \left( \frac{n}{n_N} \right) \]  

where \( \eta_{IS} \) – isothermal efficiency; \( \eta_M \) – mechanical efficiency; \( P_{ATM} \) – atmospheric pressure, kPa; \( P_C = P_{Ch} + P_{ATM} \) – final air pressure at the outlet of the turbocompressor, kPa; \( Q_C \) – volumetric capacity of the turbocompressor, m³/s; \( n \) – the current compressor rotor speed, rpm; \( n_N \) is the nominal compressor rotor speed.

The dynamic mechanical power of a turbocharger is determined by:

\[ \Delta N_{CD} = J_{rk} \omega_r \frac{d\omega_r}{dt} = \omega_r \frac{d\omega_t}{dt} \left[ k \left( J_{CSi} + \sum_{i=1}^{i=m} J_{CCI} \right) \right] \]  

where \( J_{rk} \) – total moment of inertia of the compressor rotor, kg·m²; \( J_{CSi} \) – moment of inertia of the compressor shaft, kg·m²; \( J_{CCI} \) – moment of inertia of the i-th compressor shaft, kg·m²; \( k \) – coefficient of bringing the moment of inertia to the motor shaft. Most often, for turbine units, the total rotor moment of inertia is given, given to the engine shaft.

Dynamic power will also occur during surge, since load fluctuations will cause fluctuations in the rotor speed around the synchronous one, which will lead to fluctuations in the stator phase currents.

When considering the surge phenomenon, it should be borne in mind that the pressure and volumetric capacity of the compressor in this mode are variable parameters. The pressure and volumetric capacity of a turbocharger depend on the following factors: rotor speed, tangential gas velocity at the outlet of the turbocharger impeller, parameters of the distribution network connected to the compressor. The latter include the current air flow, the number and volume of receivers, the volume of pipelines.

To describe the dynamics of transient processes in a centrifugal turbocompressor, it is convenient to use the well-known model based on the Moore – Greitzer equations [12]:

\[
\begin{align*}
\frac{d\Phi_x}{d\tau} &= \frac{1}{l_p} \left( \Psi_{\nu} (\Phi) - \Psi_C \right); \\
\frac{d\Psi_C}{d\tau} &= \frac{1}{4B^2 l_p} \left( \Phi_C - \Phi_{\nu} (\Psi_{\nu}) \right).
\end{align*}
\]
where \( l_p \) – the total length of all pipelines at the suction and discharge of the turbocompressor unit, m; \( B \) is the Greitzer parameter; \( \Psi_C \) – compression ratio of the working medium (air) by the compressor; \( \Psi_N \) – is the compression ratio of the working medium (air) in the network; \( \Phi_C \) – volumetric capacity of the compressor, p.u.; \( \Phi_N \) – network consumption, p.u.; \( \tau \) is the relative time; \( \Psi_C(\Phi) \) – gas-dynamic characteristic of the compressor, determined by the function:

\[
\Psi_C(\Phi) = \Psi_{C0}(\omega) + H(\omega) \left[ 1 + \frac{3}{2} \left( \frac{\Phi_C}{W(\omega)} - 1 \right) \left( \frac{\Phi_C}{W(\omega)} - 1 \right)^{\frac{1}{3}} \right];
\]  

(5)

where \( \psi_{C0}(\omega) \) – compression ratio in the absence of flow rate, m\(^3\)/s; \( W \) is the half-width of the gas-dynamic characteristic; \( H \) is the half-height of the gas-dynamic characteristic; \( \omega \) – rotational speed of the turbocharger wheels; rad/s.

The parameters included in equations (4) – (5) are determined by the expressions:

\[
\psi_{C0}(\omega) = K_{\psi} \omega_r^2,
\]

(6)

\[
H = K_H \omega_r^2,
\]

(7)

\[
W = K_W \omega_r,
\]

(8)

\[
B = \frac{U}{a_s} \sqrt{\frac{V_R}{\pi D_p^2 l_p}} = \frac{\omega_r R_r}{2a_s} \sqrt{\frac{V_R}{V_p}}
\]

(9)

\[
\tau = \frac{U t}{R_r}
\]

(10)

where \( K_{\psi}, K_H, K_W \) – are coefficients depending on the design features of a particular turbocompressor; \( U = \omega_r R_r \) – average tangential air flow velocity at the outlet of the turbocompressor impeller, m/s; \( R_r \) is the average radius of the turbocompressor impellers, m; \( a_s \) – speed of sound, m/s; \( V_R \) – receiver volume; \( t \) – time, s; \( \omega_r \) – rotational speed of the turbocharger wheels, rad/s; \( D_p \) – diameter of the pipeline network, m; \( V_p \) – volume of pipelines in the network, m\(^3\).

The design coefficients of the turbocompressor are determined from expressions (7) – (8) according to the passport gas-dynamic characteristics.

\[
\begin{align*}
\frac{dQ_C}{dt} &= \frac{U}{R l_p} \left( \omega_r^2 \left[ K_K + K_N \left[ 1 + \frac{3}{2} \left( \frac{Q_C}{K_N \omega_r} - 1 \right) \right] \right] - P_C \right); \\
\frac{dP_C}{dt} &= -\frac{U}{4 RB^2 l_p} (Q_C - Q_S);
\end{align*}
\]

(11)

where \( P_C \) – the pressure of the working medium (air) of the compressor, Pa; \( Q_C \) – volumetric capacity of the compressor, m\(^3\)/s; \( Q_N \) – network consumption, m\(^3\)/s.
In addition to the turbocompressor itself, the compressor unit contains a synchronous motor and a multiplier. The multiplier can be modeled with the following equations:

\[
\begin{align*}
\frac{dQ_K}{dt} &= \frac{1}{l_p} \left( \omega_0^3 \left[ K_p + K_H \left[ 1 + \frac{3}{2} \left( \frac{Q_c}{K_p \omega_r} - 1 \right) - \frac{1}{2} \left( \frac{Q_c}{K_p \omega_r} - 1 \right)^3 \right] \right) - \omega_0 P_c; \\
\frac{dP_c}{dt} &= \frac{\omega_0}{4B^2l_p} \left( Q_c - Q_N \right); \\
N_C &= \frac{P_{ATM}Q_K}{\eta_B \eta_M} \ln \left( \frac{P_c}{P_{ATM}} \right) + J_{ra} \omega_r \frac{d\omega_r}{dt}.
\end{align*}
\]  

(12)

In addition to the turbocompressor itself, the compressor unit contains a synchronous motor and a multiplier. The multiplier can be modeled with the following equations:

\[
\begin{align*}
\Delta N_M &= \Delta N_{MS} + \Delta N_{MD}; \\
\Delta N_{MS} &= N_C \left( 1 - \eta_g^n \eta_B^n \right); \\
\Delta N_{MD} &= J_M \omega \frac{d\omega}{dt} = \omega_0 \frac{d\omega}{dt} \left( \sum_{i=1}^{k} k_i J_{Gi} + k \sum_{j=1}^{l} J_{Si} \right); \\
\omega_r &= k \omega_c,
\end{align*}
\]  

(13)

where \( \Delta N_{MS} \) – the static power loss (friction loss in bearings and gear), kW; \( N_{DP} \) – dynamic power loss (manifested in dynamic modes, due to the presence of rotating masses of gears; \( \eta_g \) – gear efficiency; \( \eta_B \) – rolling bearing efficiency; \( n \) – number of rolling bearings in the multiplier, \( m \) – number of multiplier stages; \( J_M \) – total moment of inertia of the multiplier reduced to the motor shaft, kg\cdot m\(^2\); \( J_{Gi} \) – moments of inertia of each individual gear of the multiplier, kg\cdot m\(^2\); \( J_{Si} \) – moments of inertia of the multiplier shafts, kg\cdot m\(^2\); \( k \) is the number of gear wheels of the multiplier; \( i \) is the number of shafts of the multiplier; \( k_i \) – coefficient of inertia reduction of the \( i \)-th wheel to the motor shaft; \( k_j \) – coefficient of inertia reduction of the \( j \)-th shaft in the motor shaft; \( k \) – gear ratio; \( \omega_c \) – frequency of rotation of the electric motor.

The electric motor can be modeled by various approaches, for example, using the equations of a generalized synchronous machine [13-15].

3. Results and discussion

The turbocompressor unit was simulated in the Matlab Simulink environment using a set of standard blocks. Figure 1 shows the results of modeling the operation of a K-1500-62-2 type turbocompressor with a R-8000-1.49 type multiplier and an STD-10000-2 drive motor for a network with a variable flow rate.

As follows from the graphs in figure 1, the turbocompressor model based on system (12) describes its operation quite accurately, and the results are correlated with the passport data.
Figure 1. Results of modeling the operation of a K1500-62-2 type turbocompressor with a varying network flow (a), pressure (b), volumetric capacity (c) and consumed mechanical power (d).

Figure 2 shows a graph of fluctuations in mechanical power consumed by the K1500-62-2 compressor during pumping.

Figure 3 shows a graph of the stator current of a drive motor during turbocompressor surge. As it follows from figures 1-3, a drop in the air flow in the network below the critical value is accompanied by the compressor reaching the unstable area of the gas-dynamic characteristics and the occurrence of surge, accompanied by fluctuations in the pressure and air flow through the compressor with a frequency of about 4.5 Hz. Moreover, as follows from figure 1, the air flow can be reduced down to negative values, which indicates a reverse air flow. Fluctuations of the consumed mechanical power in frequency and magnitude of deviations are close to fluctuations in the flow rate (figures 2). In the graph of the stator current, there are low frequency fluctuations in amplitude relative to the nominal value. The magnitude of fluctuations in the amplitude of the stator current reaches 60% of the nominal value.

Figure 3. Graph of the stator current of the drive electric motor during turbocompressor surge.
Power fluctuations in the turbocharger causing stator current modulation are sources of the harmonic spectrum in the overall stator current signal of the motor. To confirm this, an additional analysis of the harmonic composition of the stator current was carried out using a Kaiser filter and a sampling frequency range of 0 – 500 Hz. The analysis results are shown in figure 4.

Figure 4. Spectrogram of the stator phase current during turbocompressor surge.

4. Conclusion
As follows from the graph in figure 4, the surge of the turbocompressor is accompanied by the generation of a wide range of harmonic components into the supply network. It is advisable to perform detection of surging of the turbocompressor by current in the low-frequency zone. The values of the harmonics indicate the possibility of filtering them and separating them from the total stator current without any special technical difficulties by systems of both standard analog and digital filters.

Figure 5 shows a functional diagram of a turbocompressor surge detection unit (TSDU). This circuit is relevant when implementing its filtering part based on analog passive filters.

Figure 5. TSDU functional diagram based on passive analog filters (CT – current transformer, TD1,2 – time delay unit).

Alarm recognition is based on filtering the low-frequency component of the current in the range of 0 – 35 Hz by suppressing signals with a frequency above this range. The block has a two-stage structure. Each stage operates at its own setting and can have its own time delay. The first stage is triggered at the initial stage of the surge, when the fluctuations in the motor current are small and amount to 1.1-1.2 of the maximum operating current (1,1 \( I_{\text{max,work}} < I_{\text{mean}} < 1,2I_{\text{max,work}} \)). The stage performs the function of warning about the start of surge. The second stage acts to disconnect the
compressor from the network and is triggered when the surge is fully developed, when the amplitude of the current fluctuation is more than 1.2 times ($I_{\text{mean}} > 1.2 I_{\text{max.work}}$).

The use of the described scheme for detecting surging of a turbocompressor makes it possible to diagnose the phenomenon at an early stage of development, which in turn reduces the degree of damage to unit components and economic losses of the enterprise as a whole.

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