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OPTYMALNA REGULACJA DRGAŃ SKRĘTNYCH SYSTEMU MECHANICZNEGO

Streszczenie: Jednym ze sposobów zmiany ilości sprężonego powietrza dostarczanego przez kompresor tłokowy jest zmiana jego prędkości roboczej. Jednak, powoduje to niewłaściwą regulację systemu mechanicznego napędu kompresora tłokowego w odniesieniu do jego dynamiki ruchów skrętnych. W celu uniknięcia nadmiernych drgań skrętnych, można w rozważanym systemie zastosować odpowiednie pneumatyczne podatne sprzęgło. Optymalne ciśnienie powietrza w sprzęgle pneumatycznym zostało wyznaczone eksperymentalnie dla trzech różnych trybów działania eksperymentalnego układu mechanicznego tj. napędu kompresora tłokowego.

Słowa kluczowe: pneumatyczne sprzęgło podatne, drgania skrętne, optymalna regulacja

OPTIMUM TUNING OF TORSIONALLY VIBRATING MECHANICAL SYSTEM

Summary: One of possible ways to change the amount of compressed air delivered by a piston compressor is to change its operating speed. However, it can cause an improper tuning of the mechanical system of piston compressor drive in terms of torsional dynamics. In order to avoid excessive torsional vibration, a suitable pneumatic flexible shaft coupling can be used in the system. An optimum air pressure value in a pneumatic coupling was determined experimentally for three various operating modes of an experimental mechanical system of piston compressor drive.

Keywords: pneumatic flexible shaft coupling, torsional vibration, optimum tuning

1. Introduction

Excessive torsional vibration in mechanical systems causes noise and various serious failures, such as breakages of shafts, gear teeth and machines feet, pressure damages

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of shaft keys and keyways, failures of flexible couplings and others, so we try to reduce it. The value of torsional stiffness of a flexible shaft coupling applied in a mechanical system directly influences the natural frequencies of the mechanical system. By suitable value of torsional stiffness k ($k_2 < k_1 < k_3$) (Fig.1), resonances from individual harmonic components of excitation can be moved from the operating speed range (OSR) of a mechanical system and herewith the value of dynamic component M_D of the transmitted load torque can be reduced, i.a. [1–4], [6]. Thus, we say that we can tune the mechanical system in terms of the magnitude of torsional vibration. The dependence of the M_D on the speed n of a mechanical system (Fig.1) is so-called resonance curve of the mechanical system. The speed of a mechanical system at which the natural and the excitation frequencies match (peak of the resonance curve) is called critical speed.
The torsional stiffness of a pneumatic flexible coupling, and so the natural frequencies of torsional system can be changed by adjusting the air pressure in its pneumatic flexible elements.

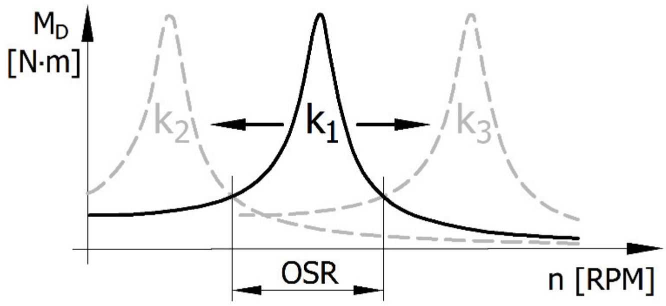

Figure 1. Mechanical system tuning principle

Pneumatic couplings allow us to tune the mechanical system even during its operation after changing its operating mode, then we talk about the continuous tuning of the mechanical system during its operation (if we want to change the torsional stiffness of a flexible shaft coupling with flexible elements made of rubber, plastic or steel, we have to replace the elements when the mechanical system is out of operation).

The change of operating speed of a piston compressor, for example using a frequency converter, is one of possible ways to change the amount of compressed air delivered by it. However, it can cause an improper tuning of the mechanical system in terms of torsional dynamics. In order to avoid excessive torsional vibration, a pneumatic flexible shaft coupling was used in an experimental mechanical system of piston compressor drive. The goal of the paper is to determine experimentally an optimum air pressure value in the pneumatic coupling for three various operating modes, given by various operating speeds n of the system: $n = 400\ \text{min}^{-1}$; $n = 700\ \text{min}^{-1}$; $n = 1000\ \text{min}^{-1}$.

2. Description and scheme of the experimental mechanical system

In following Fig.2, the scheme of the experimental mechanical system of piston compressor drive is displayed. The mechanical system is made up of 3-phase

asynchronous electromotor Siemens 1LE10011DB234AF4-Z (11 kW / 1470 RPM) (1). Rotation speed of this electromotor is continuously vector-controlled by the frequency converter Sinamics (FC). Electromotor drives a 3-cylinder piston compressor ORLIK 3JSK-75 (2) through a gearbox with gear ratio 1:1 (3) and through a pneumatic flexible shaft coupling type 4-1/70-T-C (4). The piston compressor excites torsional vibration in the mechanical system. Compressed air from the compressor streams into air pressure tank (6) with volume of 300 l. Using throttling valve (7), the air pressure in the tank and thereby also the output load of the mechanical system can be controlled. Maximum air overpressure in the pressure tank is 800 kPa and we can see its value on the manometer (8). Through the rotation supply (5), the supply of the compressed air into the pneumatic coupling is implemented.
For the measurement of torsional vibration magnitude, a torque sensor (9) (type 7934, producer MOM Kalibergyár with measuring range 0 ÷ 500 N·m) is used. For the measurement of air pressure in compression space of the pneumatic coupling, a pressure sensor (PS) (type MBS 3000, producer Danfoss with measuring range of overpressure 0 ÷ 1 MPa) is used.

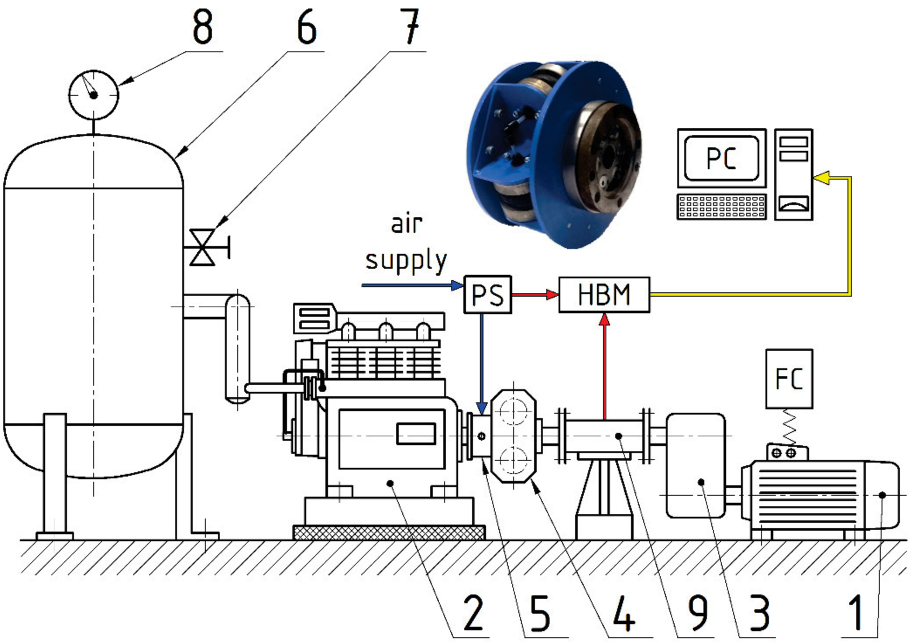

Figure 2. Experimental mechanical system of piston compressor drive

Signals from both the sensors are amplified and processed by universal 8-channel measuring device HBM MX840 and the data is subsequently sending to PC. The accuracy of the MBS 3000 sensor with metal membrane is 0,5% of its measuring range, i.e. 5 kPa (combined fault – nonlinearity, hysteresis and reproducibility) and the accuracy of the torque sensor is 0,1% of its measuring range i.e. 0,5 N·m (combined fault).

3. Experimental measurement conditions

- An operating mode of the mechanical system was characterized by a constant value of overpressure in the pressure tank and by a constant operating speed of the mechanical system.
- The constant value of overpressure in the pressure tank during measurements was 500 kPa.
- Constant operating speed of the system was n_{O1} = 400 min^{-1}; n_{O2} = 700 min^{-1} and n_{O3} = 1000 min^{-1}.
- The overpressure value in the compression space of the pneumatic coupling could be in the range from p_{pS} = 200 kPa (minimum overpressure to ensure the required transfer capacity of the coupling in given mechanical system) to 800 kPa (maximum overpressure prescribed by the manufacturer [5] of the pneumatic flexible elements).
- 3 resonance curves of the mechanical system were measured at the p_{pS} = 200 kPa, p_{pS} = 500 kPa, and p_{pS} = 800 kPa.
- The operating speed of the mechanical system varied in the range n_O = 300 ÷ 1100 min^{-1}, with step 50 min^{-1}.
- The compressor operated without failure; thus its cylinders operated evenly. For a 3-cylinder compressor, the main harmonic component is the 3rd harmonic component. The resonance peaks in our operating speed range arose from the coincidence of the main – 3rd harmonic component of the excitation with the 1st natural frequencies of the mechanical system (This fact was verified by the frequency analysis).
- To quantify the torsional vibration magnitude, the effective value RMS of the dynamic component M_D of the load torque was chosen. RMS M_D was computed according to following equations:

  RMS $M_D = \sqrt{\frac{1}{N} . \sum_{i=1}^{N} (M_{Di})^2}$ and $M_{Di} = M_i - \left(\frac{1}{N} . \sum_{i=1}^{N} M_i \right)$,

  where N is the number of samples and M_i is i-th sample of load torque time record. For the computation of RMS M_D according to the equations, the running average method was used.
- Torque signal measurement sampling frequency was 1200 Hz.

4. Results

In following Fig.3, resonance curves of the experimental mechanical system of piston compressor drive are displayed. They were measured according to the above stated conditions.

We can see that the most appropriate value of the air overpressure in the pneumatic coupling is p_{pS} = 800 kPa at n_{O1} = 400 min^{-1}; p_{pS} = 200 kPa at n_{O2} = 700 min^{-1} and p_{pS} = 200 kPa at n_{O3} = 1000 min^{-1}.

The mechanical system is operating in the sub-resonance area at p_{pS} = 800 kPa / n_{O1} = 400 min^{-1} and in the over-resonance area at p_{pS} = 200 kPa / n_{O2} = 700 min^{-1} and p_{pS} = 200 kPa / n_{O3} = 1000 min^{-1}.

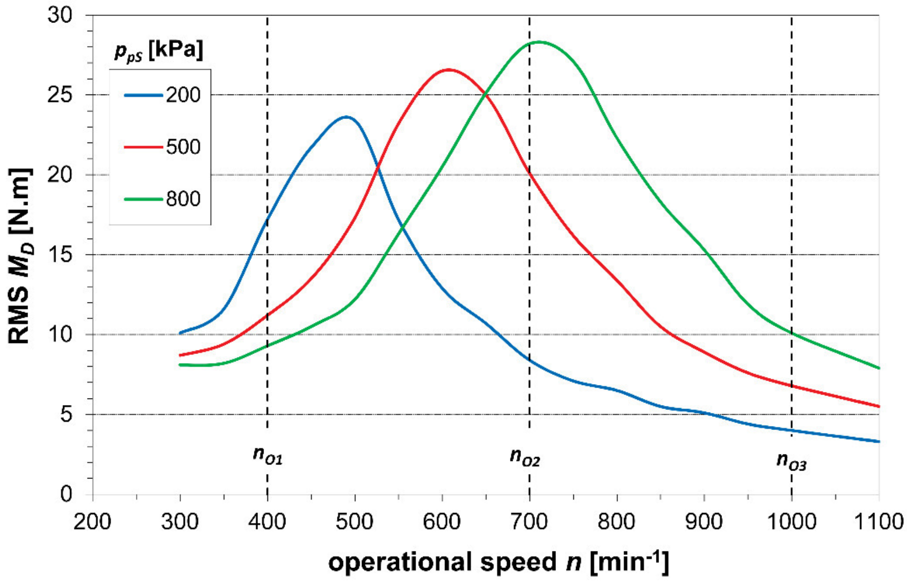

Figure 3. Measured resonance curves of the mechanical system

Conclusion

Using a suitable pneumatic flexible shaft coupling in a mechanical system of piston compressor drive, the change of operating speed of the piston compressor is possible without causing an excessive torsional vibration in the mechanical system due to improper tuning of the mechanical system in terms of torsional dynamics.
For a specific operating mode of the mechanical system, the most appropriate value of the air overpressure in the pneumatic coupling p_{pS} has to be selected and set.

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