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Matej URBANSKÝ[1], Peter KAŠŠAY[2]

Supervisor: Robert GREGA[3]

WPŁYW UNIERUCHOMIENIA TŁOKA W CYLINDRZE SILNIKA KOMPRESORA NA WIELKOŚĆ DRGAŃ SKRĘTNYCH UKŁADU MECHANICZNEGO

Streszczenie: Maszyny napędzane silnikami tłokowymi są istotnymi wzbudnikami drgań skrętnych w układach mechanicznych. W pracy rozważane są systemy mechaniczne, w których dynamiczne drgania skrętne mają być wytłumiane. Z tego względu, ważnym jest, aby znać specyficzny efekt zatrzymania pracy/ruch tłoka w cylindrze kompresora; w tym także: jaki jest wpływ tej czynności na amplitudę drań skrętnych układu mechanicznego.

Słowa kluczowe: silnik tłokowy, cylinder unieruchomiony, dynamika drgań skrętnych

THE EFFECT OF PISTON COMPRESSOR CYLINDERS DEACTIVATION ON THE TORSIONAL VIBRATION SIZE IN THE MECHANICAL SYSTEM

Summary: Piston machines are significant torsional vibration exciters in mechanical systems. We deal with mechanical systems tuning in terms of torsional dynamics and therefore it is very important for us to know the specific effect of putting piston compressor cylinder(s) out of operation on torsional vibration magnitude in a mechanical system.

Keywords: piston machine, cylinder out of operation, torsional dynamics.

1. Introduction

Piston machines are among the most important sources of excitation of torsional vibration in mechanical systems. During the operation of a mechanical system with a piston machine, the cylinder of the piston machine can be intentionally put out of

[1] Technical University of Košice; Faculty of Mechanical Engineering, Department of Machine Design and Transport Engineering; Matej.Urbansky@tuke.sk
[2] Technical University of Košice; Faculty of Mechanical Engineering, Department of Machine Design and Transport Engineering; Peter.Kassay@tuke.sk
[3] prof. Ing., PhD.; Technical University of Košice; Faculty of Mechanical Engineering; Department of Machine Design and Transport Engineering; Robert.Grega@tuke.sk

operation (temporary reduction of the power of a combustion engine, reduction of the supplied amount of compressed air by the piston compressor, etc.), or the cylinder of the piston machine may accidentally fall out of operation, in this case this is a malfunction (for example, a combustion engine ignition failure). Non-functionality, but also significant differences in the evenness of operation of one or more cylinders of a piston machine means that there will be a change in the torsional vibration excitation. Because at our workplace we deal with tuning as well as continuous tuning of torsionally vibrating mechanical systems during operation [i.a. 1-4], it is very important for us to know the exact influence of the aforementioned phenomena on dynamic conditions and on the magnitude of torsional vibration in mechanical systems. Therefore, the aim of this article is to describe the influence of cylinder-out-of-operation on the magnitude of torsional vibration in an experimental mechanical system of a piston compressor drive.

2. Experimental mechanical system

A torsionally vibrating mechanical system of piston compressor drive (Fig.1) was chosen as the experimental mechanical system.

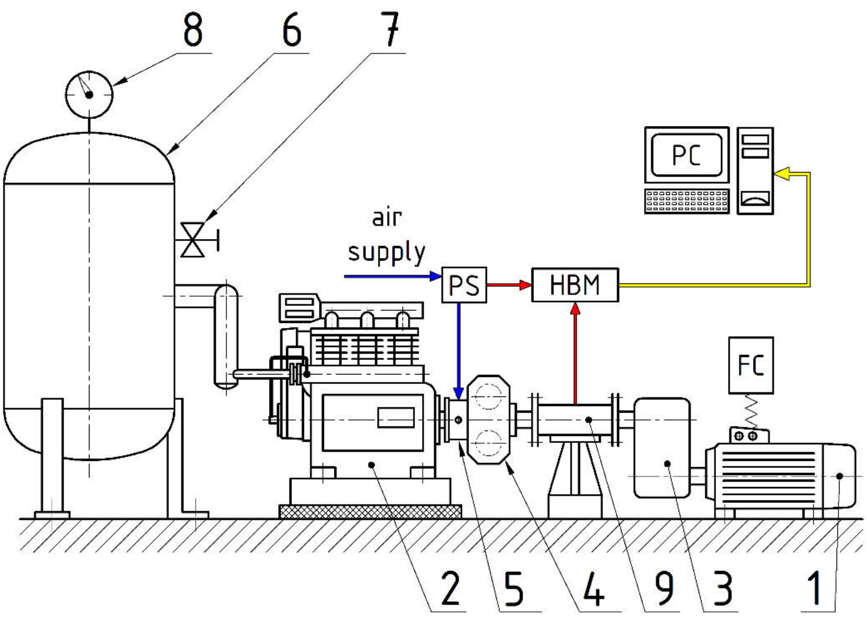

Figure 1. Experimental mechanical system

In Fig.1 we can see that the mechanical system is made up of 3-phase asynchronous electromotor Siemens 1LE10011DB234AF4-Z (11 kW, 1470 RPM) (1). The 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 of type 4-1/70-T-C (4). The compressor is mounted on a rubber layer

and has no flywheel; hence it has a higher dynamic impact. Compressed air from the compressor streams into air pressure tank (6) with volume of 300 l. Throttling valve (7) controls the air pressure in the tank and thereby also the output load of TOMS. Maximum air overpressure in the pressure tank is 800 kPa and its value we can see on the manometer (8). Through the rotation supply (5), the supply of the compressed air into the pneumatic coupling is possible. A torque sensor (9) (type 7934, producer MOM Kalibergyár with measuring range $0 \div 500$ N·m) was used for the measurement of torsional vibration magnitude. A pressure sensor (PS) (type MBS 3000, producer Danfoss with measuring range of overpressure $0 \div 1$ MPa) was used for the measurement of air pressure in compression space of the pneumatic coupling. Signals from both the sensors are amplified and processed by universal 8-channel measuring device MX840 from producer HBM 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 – nonlinearity, hysteresis and reproducibility).

The compressor cylinder(s) fall-out (cylinder without compression) can occur by the following ways in practice [5]:

- as the piston compressor inner failure because of various reasons, i.e. unwished state,
- regulation of piston compressor, i.e. targeted reduction of delivered amount of compressed air.

In our case we used an unloader (Fig.2 below) for the piston compressor cylinder(s) fall-out. Using this unloader, we opened permanently the intake plate valve of compressor cylinder so that the piston treads out the in-sucked air back into the intake piping (Fig.2 above).

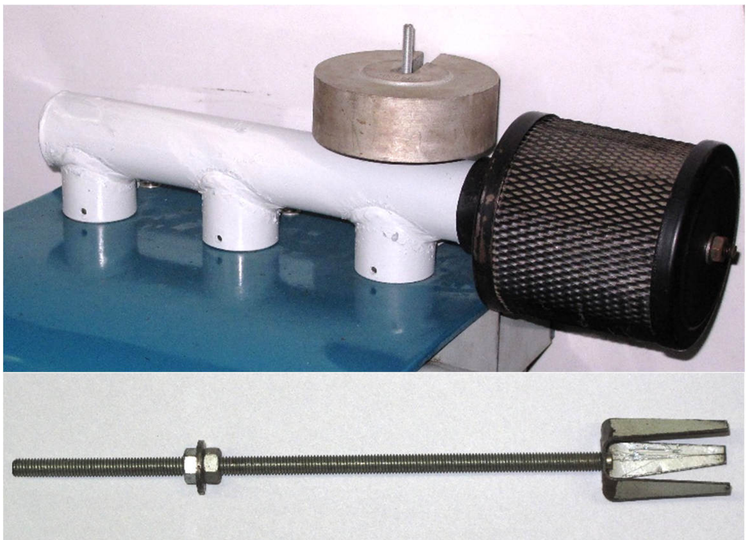

Figure 2. Cylinder unloader

3. Conditions of the measurements

As a quantity of torsional vibration magnitude, the effective value RMS of the dynamic component M_k of the load torque transmitted in the mechanical system was chosen. For the computation of RMS M_k, the running average method was used. The sample rate of 1200 Hz was used at measurements.
The constant air overpressure value in the pressure tank (6) in Fig.1 was chosen 500 kPa at each measurement.
The air overpressure in the pneumatic coupling p_{pS} was adjusted in the range from p_{pS} = 200 kPa to 800 kPa. The lower limit 200 kPa of this range was determined in consideration of the pneumatic coupling transmission capability. The upper limit 800 kPa of this range was determined in consideration of the maximal overpressure value in the pneumatic coupling defined by the air bellows manufacturer [6].

4. Results of the measurements

4.1. Failure-free operation

Resonance curves (dependences of the RMS M_k on the rotation speed at constant values of p_{pS}) of the mechanical system at failure-free operation (Fig.3) were measured at $p_{pS} = 200 \div 800$ kPa, with step 50 kPa and at rotation speed $300 \div 1200$ RPM with step 50 RPM and nearby the resonance peak with step 10 RPM in order to accurate determination of its position.

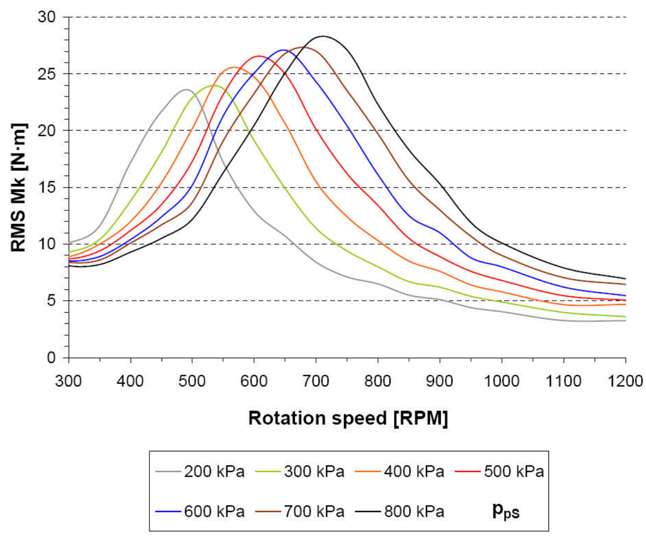

Figure 3. Measured resonance curves of the system at failure-free operation

The resonance peaks in Fig.3 arise from the coincidence of the main – 3rd harmonic component of the excitation (This fact was also verified by the frequency analysis) with the 1st natural frequencies of the mechanical system. In Fig.3 we can see that with pressure increase in the pneumatic coupling, the resonance curve of mechanical system moves to the right. It occurs because the dynamic torsional stiffness of the coupling increases with the increase of air pressure in the coupling.

From the known positions of resonance peaks we created the Campbell's diagram of the mechanical system (Fig.4). The main harmonic component and its multiples are shown as black bold lines and the other harmonic components are shown as black thin lines. Considering various other realized measurements, we can say that at current conditions the dynamic torsional stiffness k_{dyn} of the pneumatic coupling depends very lightly on the static component, amplitude and frequency of the transmitted load torque. For that reason, the natural frequencies of the mechanical system are shown as horizontal lines in the Campbell's diagram.

If the mechanical system operates at failure-free state, only the main harmonic component and its multiples are in torsional vibration excitation spectrum. If the compressor cylinder(s) fall-out occurs, the vector balance of harmonic components of torsional vibration excitation is disturbed. In this spectrum, there is decrease of main harmonic component amplitudes and the increase of the other harmonic components amplitudes at one or two cylinders fall-out.

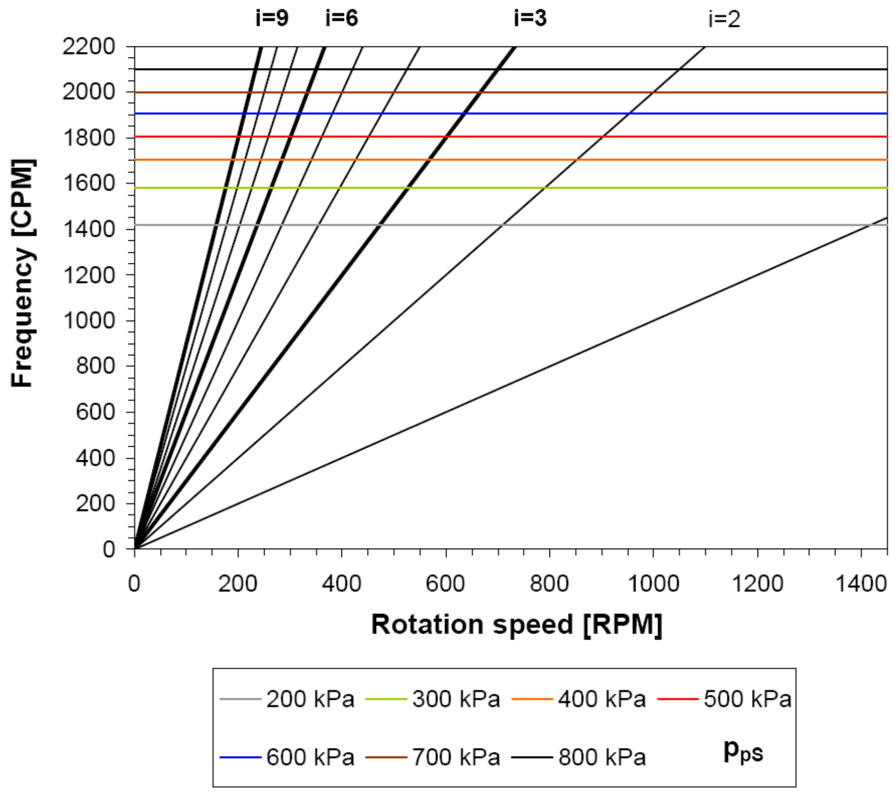

Figure 4. The Campbell's diagram of the mechanical system

4.2. One cylinder fall-out

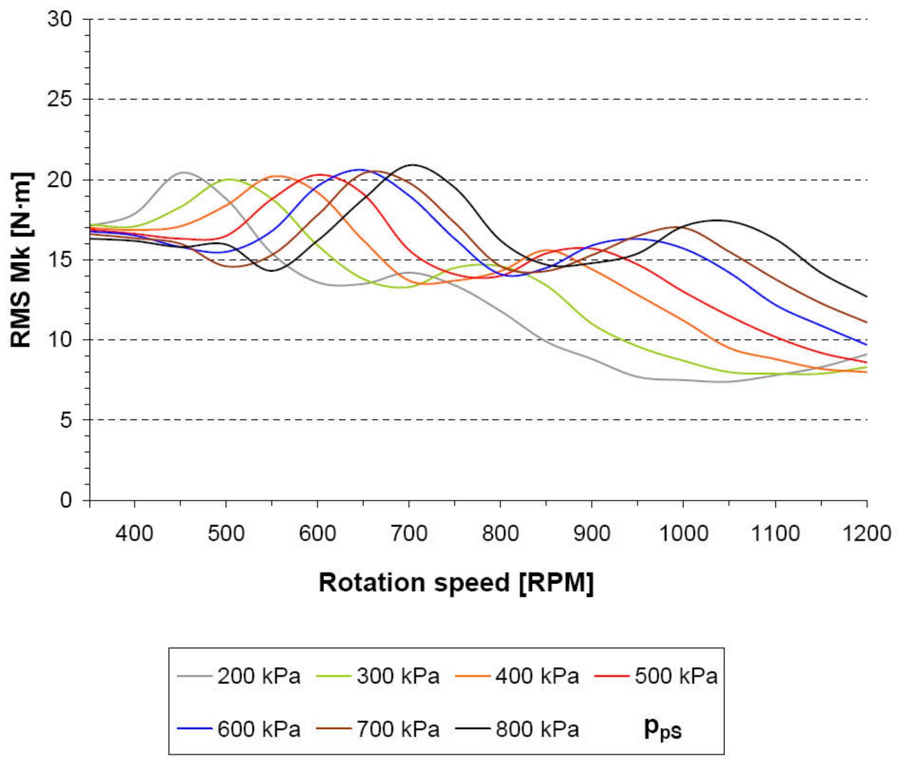

Figure 5. Measured resonance curves of the system at one cylinder fall-out operation

Resonance curves of the mechanical system at one cylinder fall-out operation (Fig.5) were measured at $p_{pS} = 200 \div 800$ kPa, with step 50 kPa and at rotation speed $300 \div 1200$ RPM with step 50 RPM and nearby the resonance peak with step 10 RPM in order to accurate determination of its position.

In Fig.5 we can see the resonance peaks arisen from the coincidence of the 3rd and 2nd harmonic component of the excitation (this fact is also verified by the frequency analysis) with the 1st natural frequencies of the mechanical system (Fig.4). The resonance with 2nd harmonic component causes increased dynamic load (in comparison with Fig.3).

Conclusion

If all 3 cylinders of the piston compressor of the given mechanical system are working and the deviations of the work of the individual compressor cylinders are minimal, then, in the operation speed range, resonances will arise only from the main (in this case the third) harmonic component of the torsional vibration excitation (Fig.3). In Fig.3, it is also possible to see that with a suitable adjustment of the torsional stiffness of the pneumatic coupling (given by the overpressure of a gaseous medium in the compression volume of the coupling) we can always move in the region outside of resonance (i.e. in the supra- or sub-resonance region = tuning of the mechanical system from the point of view of dynamics) in the entire operation speed range of given mechanical system. It is very convenient in the case if we change the amount of compressed air supplied by the compressor just by changing the speed of the given mechanical system.

In the case that the deviations of the work of the individual compressor cylinders are significant, or the compressor cylinder(s) are out of operation, then resonances from the minor harmonic components of the torsional vibration excitation will also appear in the operation speed range of the mechanical system. In this case (Fig.5), resonances from the 3rd and 2nd harmonic components are manifested in the operation speed range. This results in an increase in the value of the dynamic component of the load torque M_k in the mechanical system in the area of higher operating speeds, even with appropriate tuning of the mechanical system with respect to the main harmonic component. In Figure 5, it is also possible to see that by choosing the appropriate torsional stiffness of the pneumatic coupling, we can still move in the region outside of resonance in the entire operation speed range of given mechanical system. It is advantageous in the case that we change the amount of compressed air supplied by the compressor by applying decommissioning of the cylinder(s) of the piston compressor.

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