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INFLUENCE OF PISTON COMPRESSOR INNER FAILURE ON MECHANICAL SYSTEM OBJECTIVE FUNCTION

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Summary

At our department we deal with continuous tuning of torsional oscillating mechanical systems during their operation using pneumatic tuners, mainly in terms of torsional vibrations magnitude, whereby we use the methods and means of technical diagnostics. One of the manners of continuous tuning realization is the application of extremal control – experimental optimization, which main advantage is that we don't need to know the mathematical model of mechanical system. We must know only that the objective function of mechanical system has an extreme. For that reason, it is very important to know this form of objective function not only at failure-free operation but at failure occurrence as well. The objective of this paper is therefore the investigation of the influence of piston compressor inner failure, namely cylinder(s) fall-out on mechanical system objective function form.

Keywords: torsional vibration, extremal control, objective function, inner compressor failure

1. INTRODUCTION

With continuous tuning of torsional oscillating mechanical systems (TOMS) during their operation deal i.a. publications [1], [2], [3], [4], [5], [6], [7], [8], [9], [10] and [11]. This continuous tuning we make mainly in terms of torsional vibrations magnitude (but also in terms of magnitude of rectilinear vibrations or noise arising from torsional vibrations). As means of this continuous tuning we use pneumatic flexible shaft couplings (pneumatic torsional vibrations tuners, thereinafter „pneumatic tuners") developed at our department i.a. publications [2], [3], [4], [5], [6], [7], [8], [9].

The torsional stiffness of given pneumatic tuners and so the natural frequencies of torsional systems can be changed by adjusting the gaseous media (the most commonly air) pressure in their pneumatic flexible elements. By suitable value of torsional stiffness k ($k_2 < k_1 < k_3$), resonances from individual harmonic components of excitation (Fig.1) can be moved from the operational speed (n) range (OSR) of mechanical system and herewith the value of dynamic component M_D of the transmitted load torque can be reduced, i.a. [12], [13], [14], [15], [16], [17], [18], [19] and [20].

One of the methods of continuous tuning is the application of the extremal control – experimental optimization. Extremal control gives us the possibility to minimize the magnitude of dangerous torsional vibrations in torsional oscillating mechanical systems during their operation directly by adapting the dynamic properties of the TOMS to actual operating parameters and failures. The main advantage of the extremal control is that we don't need to know the exact mathematical model of the system. We must know only that the objective function of the mechanical system has an extreme, i.a. [2], [3], [4], [5], [6], [7], and [8].

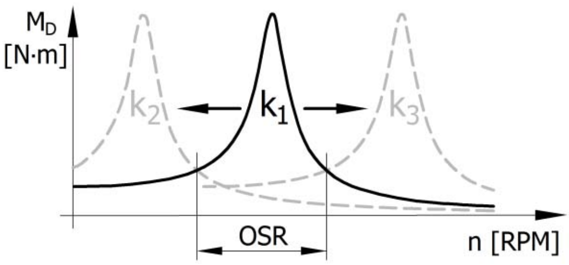

Fig. 1. Mechanical systems tuning principle

The objective function can be defined as the dependence of regulated parameter on the actuating variable at steady state of the mechanical system. In our case, the actuating variable is the air pressure in the pneumatic torsional vibration tuner and the regulated parameter is the magnitude of torsional vibrations (but also the magnitude of rectilinear vibrations or noise arising from torsional vibrations).

If some failures occur in the TOMS, these can be the cause of TOMS objective function form change. It is very important to know the objective function form not only at failure-free operation but at failure occurrence as well, because of the algorithm development, according to it the extremal control operates.

The piston compressor belongs to the group of piston machines, which are significant exciters of torsional vibrations in TOMS. The objective of this paper is therefore the investigation of the piston compressor inner failure influence, namely one or more cylinders fall-out (in meaning to cylinder without compression) on mechanical system objective function form.

2. DESCRIPTION OF THE EXAMINED MECHANICAL SYSTEM

The torsional oscillating mechanical system of the piston compressor drive (Fig.2) was chosen as examined mechanical system.

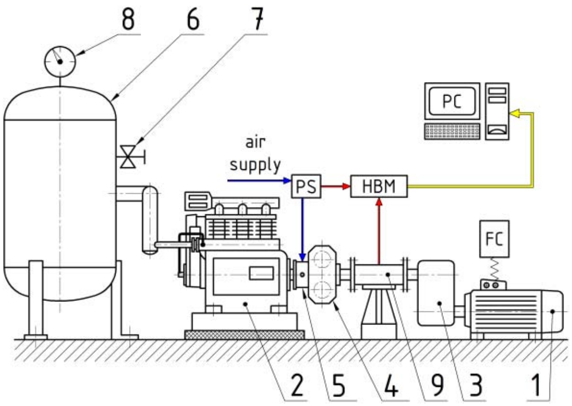

Fig. 2. Implemented torsional oscillating mechanical system

In Fig. 2 we can see that the TOMS consist 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 tuner of type 4-1/70-T-C (4) (Fig.3).

Fig. 3. Pneumatic tuner of type 4-1/70-T-C

The compressor is mounted on a rubber layer and has no flywheel; hence it has a higher dynamic impact [2], [5]. 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) is realized the supply of the compressed air into the pneumatic tuner. We used a torque sensor (9) (type 7934, producer MOM Kalibergyár with measuring range 0 ÷ 500 N·m) for the measurement of torsional oscillation magnitude. We used a pressure sensor (PS) (type MBS 3000, producer Danfoss with measuring range of overpressure 0 ÷ 1 MPa) for the measurement of air pressure in compression space of the pneumatic tuner. 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 [21]:

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

In our case we used an unloader (Fig.4 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.4 above).

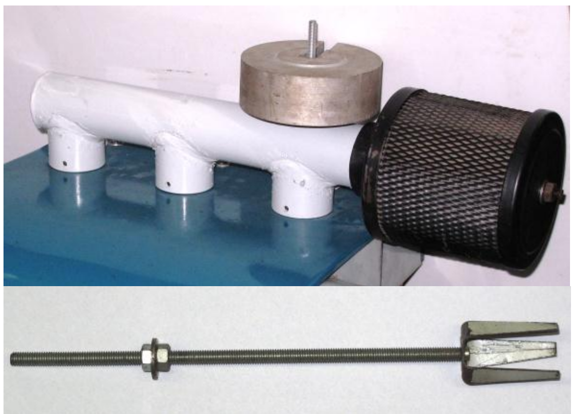

Fig. 4. Cylinder unloader

3. CONDITIONS OF THE MEASUREMENTS

From the description of the examined mechanical system it is obvious that torsional oscillations magnitude (specifically the effective value RMS of the dynamic component *Mk* of the load torque) as regulated parameter for the extremal control was chosen in this case. RMS *Mk* is transmitted by the pneumatic tuner and it was computed according to following equations:

$$RMS\,Mk = \sqrt{\frac{1}{N}.\sum_{i=1}^{N}(Mk_i)^2} \text{ , and} \tag{1}$$

$$Mk_i = M_i - \left(\frac{1}{N} . \sum_{i=1}^{N} M_i \right), \qquad (2)$$

where N is the number of samples and M_i is i^{th} sample of load torque time record. For the computation of RMS Mk according to equations (1) and (2) 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.2 was chosen 500 kPa at each realized measurement.

The air overpressure in the pneumatic tuner 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 tuner transmission capability. The upper limit 800 kPa of this range was determined in consideration of the maximal overpressure value in the pneumatic flexible elements (air bellows). This maximal overpressure p_{pS} value is defined by the air bellows manufacturer.

4. RESULTS OF THE MEASUREMENTS

4.1. Measured resonance curves of the mechanical system at failure-free operation

We measured the resonance curves (in our case the dependences of the RMS Mk on the rotation speed at constant values of p_{pS}) of the mechanical system at failure-free operation (Fig.5) at p_{pS} = 200 ÷ 800 kPa, with step 50 kPa and at rotation speed 300 ÷ 1200 RPM with step 50 RPM and nearby the resonance peak with step 10 RPM in order to accurate determination of its position.

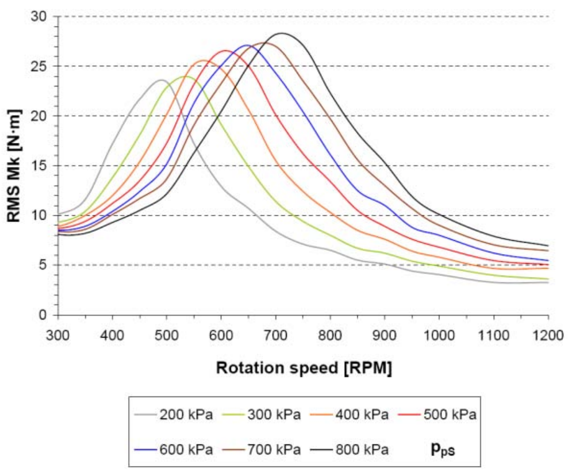

Fig. 5. Measured resonance curves of the mechanical system at failure-free operation

The resonance peaks in Fig.5 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.5 we can see that with pressure increase in the pneumatic tuner the resonance curve of mechanical system moves to the right. It occurs because the dynamic torsional stiffness of the tuner k_{dyn} increases with the increase of air pressure in the tuner, as we can see in Fig.6.

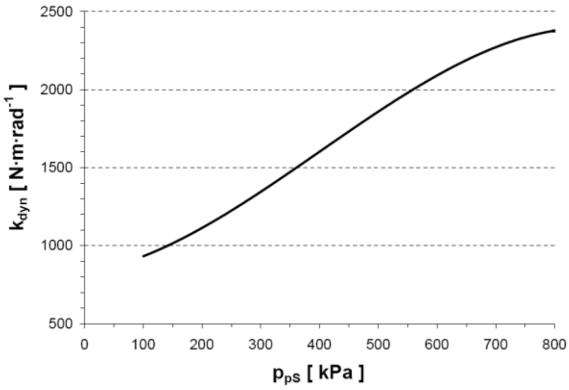

Fig. 6. The dependence of the dynamic torsional stiffness of the tuner k_{dyn} on the air overpressure p_{pS} in the tuner

From the known positions of resonance peaks we created the Campbell's diagram of the mechanical system (Fig. 7). 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 tuner 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.

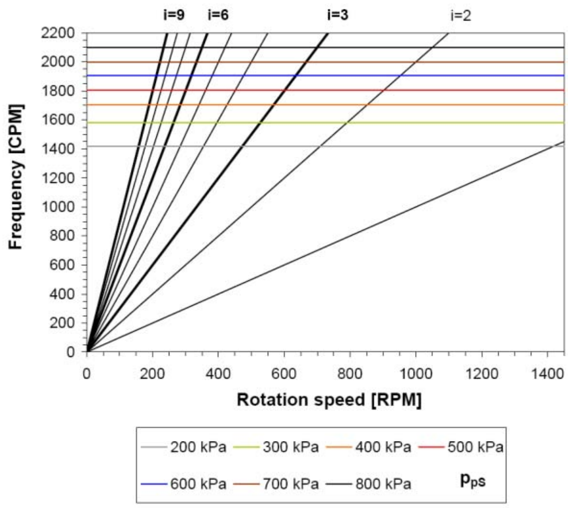

Fig. 7. The Campbell's diagram of the mechanical system

In terms of torsional dynamics the given mechanical system can be considered as a dual mass TOMS, because each 2nd natural frequency of the TOMS is coincident only with higher harmonic components of compressor excitation in the examined operation speed range (their magnitude is negligible).

4.2. Measured resonance curves of the mechanical system at one cylinder fall-out

If the mechanical system operates at failure-free state, only the main harmonic component and its multiples are in torsional oscillation excitation spectrum (column A in the Tab.1). If the compressor cylinder(s) fall-out occurs, the vector balance of harmonic components of torsional oscillation excitation is disturbed. In this spectrum at one or two cylinders fall-out we can see the decrease of main harmonic component amplitudes and the increase of the other harmonic components amplitudes (columns B and C in the Tab.1).

Table 1. Theoretical computation of harmonic component (H.C.) amplitudes at: A – failure-free operation, B – one cylinder fall-out, C – two cylinders fall-out

H.C.	A [N·m]	B [N·m]	C [N·m]
$i = 1$	0	22,8	22,8
$i = 2$	0	19,54	19,54
$i = 3$	40,02	26,32	12,66
$i = 4$	0	7,84	7,84
$i = 5$	0	3,16	3,16
$i = 6$	1,39	0,91	0,44
$i = 7$	0	1,51	1,51
$i = 8$	0	1,68	1,68
$i = 9$	3,66	2,44	1,22

We measured the resonance curves of the mechanical system at one cylinder fall-out operation (Fig.8) at $p_{pS} = 200 \div 800$ kPa with step 50 kPa and at rotation speed 300 ÷ 1200 RPM with step 50 RPM and nearby the resonance peak with step 10 RPM in order to accurate determination of its position.

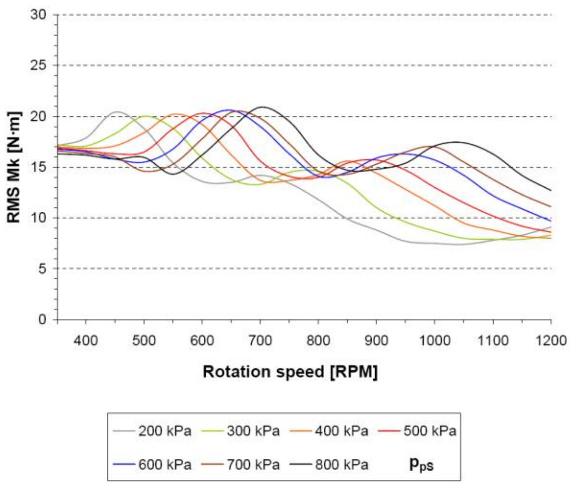

Fig. 8. Measured resonance curves of the mechanical system at one cylinder fall-out operation

In Fig.8 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.7). The resonance with 2nd harmonic component causes increased dynamic load (in comparison with Fig.5).

4.3. Objective functions of the mechanical system

The objective function in our case is the dependence of the RMS *Mk* on the air overpressure p_{pS} in the pneumatic tuner at steady state of mechanical system, i.e. at constant rotation speed and at constant air overpressure value in the pressure tank. In the following figures, we can see the objective functions forms at failure-free operation (Fig.9) and at one cylinder fall-out operation (Fig.10). The variable rotation speed of the mechanical system of piston compressor drive can be used in practice, in order to targeted reduction of delivered compressed air amount. Therefore we selected several constant rotation speed values of the TOMS within its operation speed range. In Fig.9 we can see that the objective functions at failure-free operation are in the whole p_{pS} range either monotonic, or concave. The objective functions at one cylinder fall-out operation can be more complex with more local minima and maxima, as it is shown in Fig.10 using bold lines.

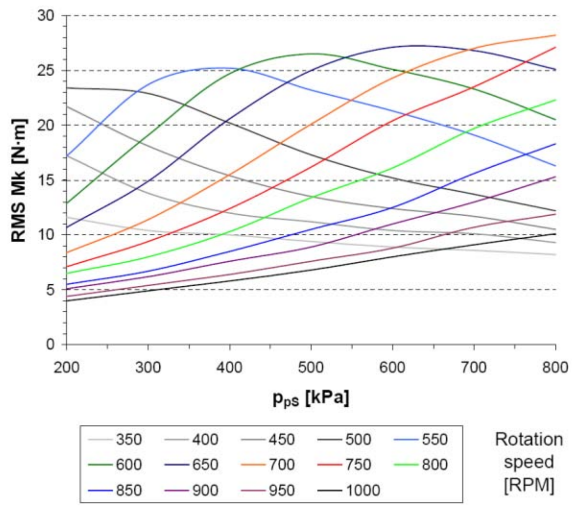

Fig. 9. Objective functions of the mechanical system at failure-free operation

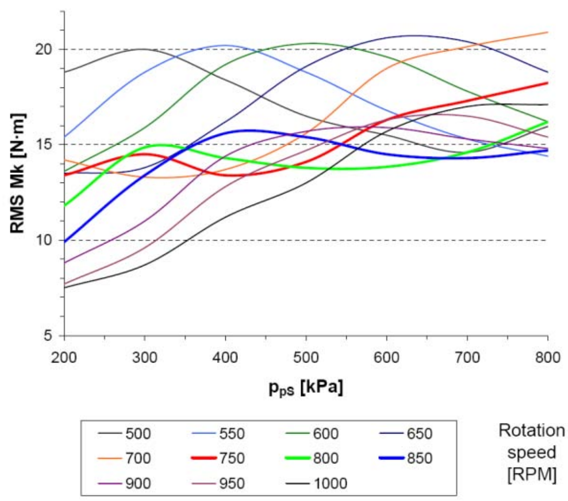

Fig. 10. Objective functions of the mechanical system at one cylinder fall-out operation

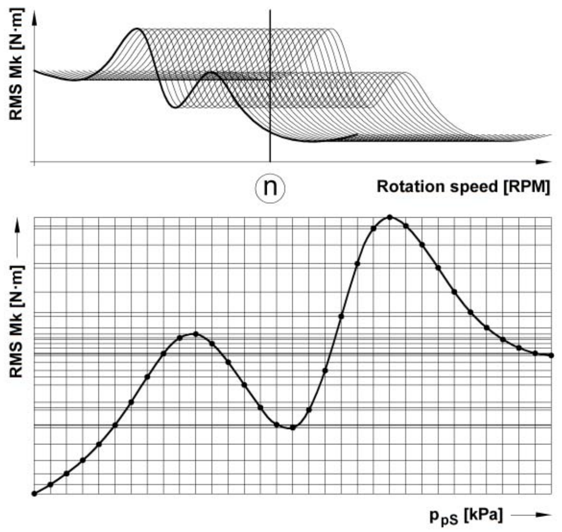

Fig. 11. Possible objective function form at rotation speed *n* at the increase of actuating variable p_{pS} range

As we can see in Fig.11 (the Fig.11 is dimensionless, simplified and the objective function is drawn using of graphic method), we can expect even more complex objective functions with increasing of the range of air overpressure in the pneumatic tuner p_{pS}.

5. CONCLUSION

We can now use the information obtained from the realized measurements (Fig.7, Fig.8, Fig.9 and Fig.10) for the algorithm development, according to the extremal control operates. It is necessary that the extremal control can react on the failure occurrence in the TOMS (e.g. piston machine cylinder fall-out, irregular excitation of cylinders, etc.) and it is very advisable that the extremal control can find not only the objective function local minimum, but also the objective function global minimum.

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• Diagnostics of torsionally oscillating mechanical systems performed with regard to the characteristics of torsional oscillation and excited mechanical oscillation.

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