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VERIFICATION OF TORSIONAL OSCILLATING MECHANICAL SYSTEM DYNAMIC CALCULATION RESULTS

Summary. On our department we deal with optimization and tuning of torsional oscillating mechanical systems. When solving these problems we often use results of dynamic calculation. The goal of this article is to compare values obtained by computation and experimentally. For this purpose, a mechanical system built in our laboratory was used. At first, classical HARDY type flexible coupling has been applied into the system, then we used a pneumatic flexible shaft coupling developed by us. The main difference of these couplings over conventional flexible couplings is that they can change their dynamic properties during operation, by changing the pressure of the gaseous medium in their flexible elements.
Keywords: torsional vibration, Measurement, Dynamic Calculation.

WERYFIKACJA WYNIKÓW OBLICZEŃ DYNAMICZNYCH W WYKONANYM UKŁADZIE MECHANICZNYM DRGAJĄCYM SKRĘTNIE

Streszczenie. W naszej pracowni zajmujemy się optymalizacją i strojeniem układów mechanicznych drgających skrętnie. W trakcie rozwiązywania tych problemów często korzystamy z wyników obliczeń dynamicznych. Celem niniejszego artykułu jest porównanie wartości uzyskanych na podstawie obliczeń z uzyskanymi w trakcie doświadczeń. Wykorzystaliśmy do tego układ mechaniczny skonstruowany w naszym laboratorium, przy czym w jednym przypadku zostało w układzie zastosowane klasyczne elastyczne sprzęgło typu HARDY, a w drugim wynalezione przez nas elastyczne sprzęgło pneumatyczne łączące wały. Najważniejszą cechą różniącą te sprzęgła od klasycznych sprzęgieł elastycznych jest to, że możliwe jest zmienianie ich właściwości dynamicznych bezpośrednio podczas pracy za pomocą zmiany ciśnienia medium gazowego w elementach elastycznych.
Słowa kluczowe: drganie skrętne, pomiar, obliczenie dynamiczne.

1. INTRODUCTION

When designing torsional oscillating mechanical systems, it is usually necessary to rely on the results of dynamic calculation. There we try to find out especially the size of dynamic component of load torque during operation. This calculation should reflect the situation in the

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already built mechanical system as closely as possible. This requires to have the most accurate input values. In order to check the accuracy and reliability of dynamic calculation, we decided to use a mechanical system implemented in our laboratory. By identifying the input values of mechanical system individual elements, we relied on catalogue data of producers and on parameter values measured by us. We also investigated the effect of the torque sensor location on the resulting measured torque time course.

2. EXAMINED MECHANICAL SYSTEM

The dynamic calculation results verification we performed on a mechanical compressor drive system (Fig. 1). The system consists of three-phase asynchronous electric motor which drives three-cylinder air compressor. The time course of load torque is sensed by a torque sensor. Pneumatic supply of flexible shaft flexible coupling is handled by use of radial rotary air supply. Measurements were performed on the system with a HARDY type flexible shaft coupling and with tangential pneumatic flexible shaft couplings with fully interconnected pneumatic flexible elements type 4-2/70-T-C.

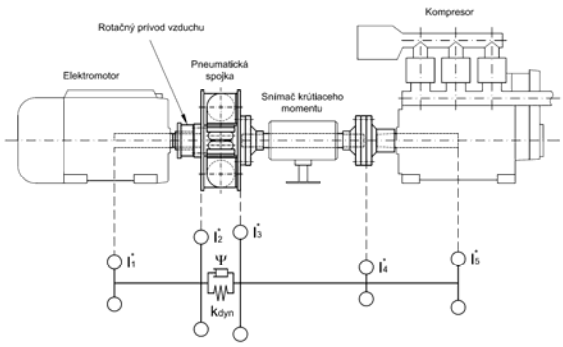

Fig. 1. Mechanical system of compressor drive
Rys. 1. Układ napędu mechanicznego kompresora

Individual elements of examined mechanical system have the following parameters:

Parameters motor:

- Three-phase asynchronous squirrel-cage motor SIEMENS 1LE1002-1CB,
- rated power of the electric motor P_M = 7,5 kW,
- rated speed electric motor n_M = 1450 ot/min,
- mass moment of inertia of the electric motor $I_E = I_1^*$ = 0,024 kg.m^2.

The speed of the electric motor can be continuously changed with frequency converter.

Parameters compressor:

- Three-cylinder piston air compressor ORLIK 3JSK-75,
- mass moment of inertia of the compressor I_K=I_5^* =0,02967 kg.m^2
- mass moment of inertia of the flanges I_4^* =0,0151 kg.m^2
- atmospheric pressure p_a = 100 kPa,
- overpressure at the inlet p_{ps} = 0 kPa,
- overpressure at the discharge p_{pv} = 400 kPa

By calculating the torque dependency on the crankshaf rotation angle, in addition to pressure and inertial forces, we counted with constant friction force acting on the piston. Magnitude of this force was chosen so, that calculated compressor static torque responded to real (measured) values as recommended in [5]. Calculated course of compressor load torque at selected speed n = 1450 rpm is shown on Fig. 2.

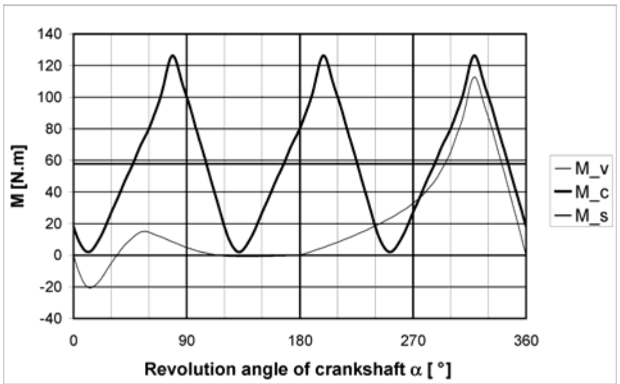

Fig. 2. Load torque of the compressor at speed n = 1450 rpm
Rys. 2. Obciążający moment skrętny kompresora przy obrotach n = 1450 obr/min^{-1}

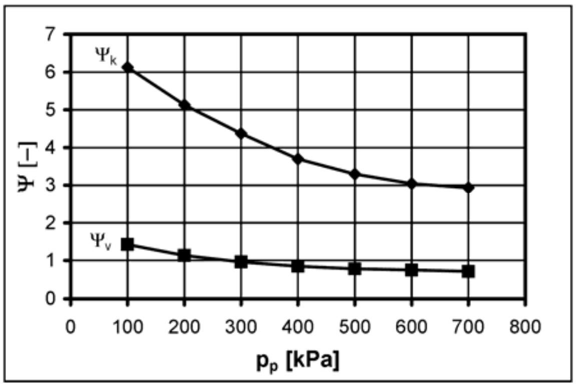

Fig. 3. Relative damping of pneumatic flexible shaft coupling
Rys. 3. Stopniowe tłumienie elastycznego sprzęgła pneumatycznego łączącego wały

where: M_v torque of first cylinder, M_c torque of whole compressor, M_s static (mean) torque of compressor.

Parameters of HARDY flexible coupling:

- Dynamic torsional stiffness $k_{dyn} = 3{,}117 \cdot M_{stat} + 1280$ N.m.rad^{-1},
- coefficient of relative damping $\Psi = 1$.
- moment of inertia on motor side $I_2* = 0{,}00178$ kg.m^2,
- moment of inertia on compressor side $I_3* = 0{,}009319$ kg.m^2.

Parameters of of pneumatic flexible shaft coupling type 4-2/70-T-C:

- Dynamic torsional stiffness $k_{dyn} = 1{,}925 \cdot p_p + 740{,}22$ N.m.rad^{-1},
- moment of inertia on motor side $I_2* = 0{,}0672$ kg.m^2,
- moment of inertia on compressor side $I_3* = 0{,}0757$ kg.m^2.

Mechanical properties measurements were performed with both of flexible shaft couplings. Dynamic properties were determined by the method of free oscillations [4]. This method gives good results for dynamic torsional stiffness. The values of relative damping coefficient however, are more influenced by the frequency and amplitude of twist angle. Therefore we used data from the manufacturer for HARDY coupling. For the pneumatic shaft couplings data from the work [8] are used for comparison. Where the author established damping parameters by kinematic excitation with frequency 22 rad.s^{-1}. Courses of relative damping coefficient depending on the overpressure in the coupling p_P by free oscillation Ψ_v and by kinematic excitation Ψ_k we can see on Fig. 3.

3. DETERMINING THE COURSE OF LOAD TORQUE ON TORQUE SENSOR

Time course of load torque transmitted by flexible shaft couplings, we can determine by dynamic calculation of two-mass torsional oscillating mechanical system [3], [7]. Where for I_1 we substitute $I_1 = I_1^* + I_2^*$, and for I_2 we substitute $I_2 = I_3^* + I_4^* + I_5^*$.

Since the torque sensor is placed between the compressor (torsional vibration exciter) and flexible coupling, so the measured time course will differ from the dynamic calculation results of the torque transmitted by flexible shaft couplings. Values that we should obtain by experimental measurements can be determined using the following procedure.

Let us have two masses (0) with mass moments of onertia I_A and I_B, which are rigidly connected by an intangible shaft representing the torque sensor. Torques M_A and M_B. are acting on them

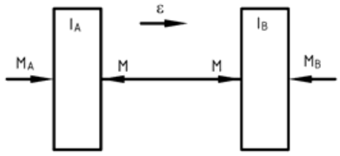

Fig. 4. Action of torques on two rigidly connected masses
Rys. 4. Działanie momentów skrętnych na dwie sztywno ze sobą połączone masy

Torque M acting on intangible shaft will have a value:

$$M = M_A \cdot \frac{I_B}{I_A + I_B} + M_B \cdot \frac{I_A}{I_A + I_B} \tag{1}$$

Torque M will be affected by torque courses of both masses, as well as by the mass ratio I_A/I_B. When applied to our case the first mass will be represented by the halves of flexible coupling and the rotating parts of torque sensor, together with the connecting flanges ($I_A = I_3^*$). The second mass consists of other half of the torque sensor rotating parts, compressor crank mechanism and connecting flanges ($I_B = I_4^* + I_5^*$).

4. COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS FOR MECHANICAL SYSTEM WITH HARDY FLEXIBLE COUPLING

In the mechanical system with HARDY flexible coupling the mass ratio is $I_A/I_B = 0{,}21$. So mass I_A, on which acts dynamic load torque of flexible coupling M_{dS} is significantly smaller than mass I_B on which acts dynamic load torque of compressor M_{dK}.

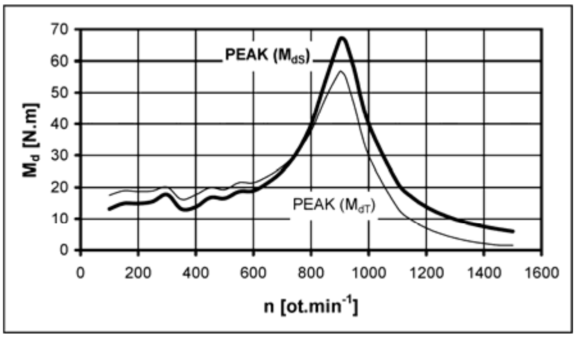

Fig. 5. Course of calculated dynamic torques
Rys. 5. Przebieg obliczonych dynamicznych momentów skrętnych

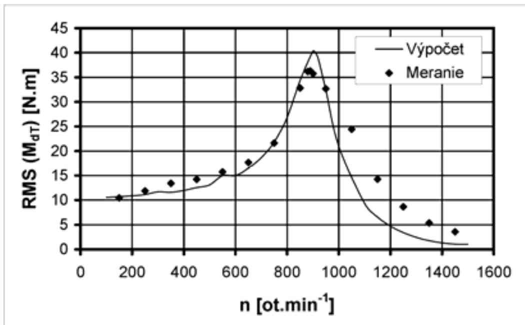

Fig. 6. Effective torque on sensor for HARDY flexible coupling
Rys. 6. Wartość efektywna momentu skrętnego na czujniku sprzęgła podatnego HARDY

According to equation (1) the resulting time course on torque sensor M_{dT} should be more close to time course of flexible coupling torque. This is shown on 0, where the course of peak torques on the coupling *PEAK(*M_{dS}*)* and on the sensor *PEAK(*M_{dT}*)* are similar. On the mechanical system measurements of effective torque *RMS(MdT)* were taken. Comparison of the values obtained by calculation and measurement we can see on 0.

5. COMPARISON OF THEORETICAL AND EXPERIMENTAL RESULTS FOR MECHANICAL SYSTEM WITH PNEUMATIC FLEXIBLE COUPLING

In the mechanical system with pneumatic flexible coupling the mass ratio is I_A/I_B = 1,69. So mass I_A, on which acts dynamic load torque of flexible coupling M_{dS} is larger than mass I_B on which acts dynamic load torque of compressor M_{dK}. Acoording to equation (1) the final time course of torque M_{dT} on the sensor will be significantly influenced by the dynamic component of compressor torque. We can see it also on the following course, where in speed area over the resonance the peak value of torque on sensor PEAK(M_{dT}) increases (0). Pak values curses of dynamic torque on the coupling *PEAK(*M_{dS}*)* and on the sensor *PEAK(*M_{dT}*)* depending on operating speed *n* are shown on 0 for relative damping determined by free oscillation. Listed courses were calculated for p_p = 100 kPa value of overpressure in the coupling. Comparison of experimental results with calculated values for coefficient of relative dampig determined by free oscillation Ψ_v and by kinematic excitation Ψ_k is on 0.

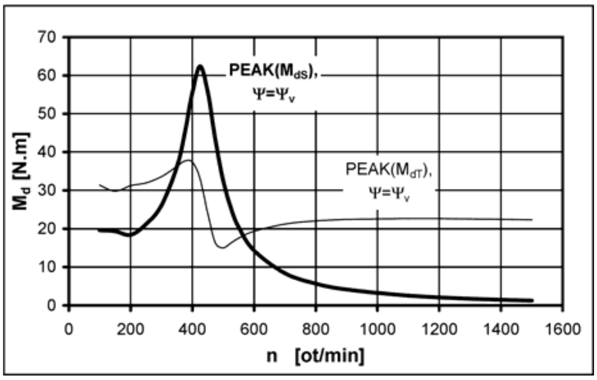

Fig. 7. Course of calculated dynamic torques
Rys. 7. Przebieg obliczonych dynamicznych momentów skrętnych

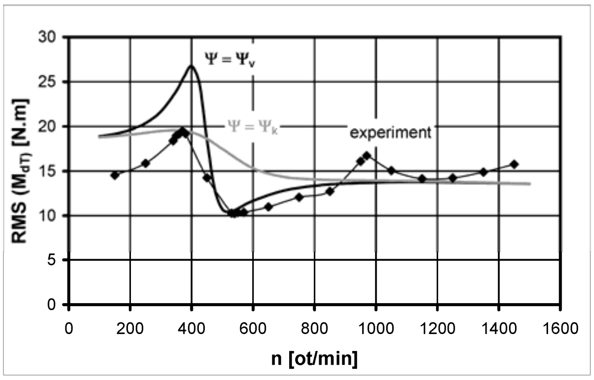

Fig. 8. Effective value of torque on the sensor, at a overpressure p_p = 100 kPa
Rys. 8. Wartość efektywna momentu skrętnego na czujniku przy nadciśnieniu p_p = 100 kPa

6. CONCLUSION

We can say that the measured and calculated values for HARDY coupling reported a quite good agreement. Position of the speed at resonance against measurement varies approx. 2%, and the maximum value at the resonance approx. by 11%. Higher tolerances at higher speeds are probably caused by the fact that flexible coupling changes its damping properties by influence of amplitude and frequency of vibration.

For pneumatic flexible shaft coupling is, for damping by free oscillation Ψ_v, the difference in speed at resonance approx. 8%, but the maximum of effective value varies by as much as 37%. Good agreement shows that course in speed area over the resonance.

In contrast, for the damping by kinematic excitation Ψ_k, the difference in speed at resonance is approx. 5%, but the maximum of effective value differs by only 0.5%. Thus, the size and position of resonance (local maximum) is very close to the measured values. In the future, it would be necessary to pay more attention to damping of pneumatic flexible shaft couplings. Significant impact on its value has the airflow in their interconnection tubes, which is influenced by the frequency and amplitude of the coupling twist angle.

If we want to measure torque transmitted by flexible shaft coupling with torque sensor, it is best to place the torque sensor on the opposite side as torsional vibration exciter is located. Then only dynamic load torque transmitted by flexible shaft coupling will be reflected in measured data, In case that sensor is positioned between shaft coupling and exciter of torsional vibration, the mass moment of inertia on the side of flexible coupling should be as small as possible.

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