CONTINUOUS TUNING OF SHIP PROPULSION SYSTEM BY MEANS OF PNEUMATIC TUNER OF TORSIONAL OSCILLATION

(DOI No: 10.3940/rina.ijme.2016.a3.378)

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SUMMARY

Mechanical system drives consist of driving machines and gearing mechanisms interconnected by shafts and couplings. In terms of dynamics it is possible to say that every driving mechanism is able to oscillate. Especially piston devices can create excessive torsional oscillation, vibrations, as well as noise. Important task of a designer is to reduce torsional oscillation in mechanical systems. Presently this problem is mainly solved by the flexible shaft couplings that are selected with regard to the dynamic properties of the given system. It means that every torsional oscillating mechanical system needs to be suitably tuned. The aim of this paper is to present the possibilities of controlling of dangerous torsional oscillations of the mechanical systems by the means of new method, i.e. its optimal tuning by means of the pneumatic coupling with self-regulation, which were developed by us.

NOMENCLATURE

I	Mass moment of inertia (kg.m^2)
I_{RED}	Reduced mass moment of inertia of a two mass torsional oscillating mechanical system (kg.m^2)
i	Gear ratio (–)
j	harmonic component order (–)
k	Dynamic torsional stiffness (N.m.rad^{-1})
M	Static load torque (N.m)
M_d	Dynamic load torque (N.m)
M_j	Amplitude of the j-th harmonic load torque component (N.m)
M_K	Load torque (N.m)
M_M	Engine torque (N.m)
M_Z	Static load torque computed from the propeller's fan characteristics (N.m)
m	Total number of masses (–)
N	Speed frequency (min^{-1})
n	Operating speed of engine (min^{-1})
PToTO	Pneumatic Tuner of Torsional Oscillations
p	Pressure of gaseous medium in coupling's compression space (kPa)
TOMS	Torsional Oscillating Mechanical System
γ_j	Phase angle of the j-th harmonic load torque component (N.m)
φ	Static twist angle (rad)
φ_k	Constant twist angle (°)
Ω	Natural angular frequency (rad.s^{-1})
ω	Angular speed (rad.s^{-1})

1. INTRODUCTION

Pumps, fans and especially piston combustion engines, as well as compressors, are consider to be the exciters or actuators of torsional oscillations in the mechanical systems [1…14]. Intensive torsional oscillations are the main cause of excessive dynamic load in these systems. As the result of this fact, some damages or failures of individual parts are arise in the system, for example damages of bearings, shafts, transmissions or flexible shaft couplings but also destructions of piston machinery.

The successful control of torsional oscillations can be achieved only by a detailed analysis of the system from the torsional dynamic point of view. The results of such analysis indicate that stated goal is possible to achieve only when the values of some elements from the system will be suitably adjusted for dynamic of the system. It means, that any "torsional oscillating mechanical system" (TOMS) has to be tuned. Practical experiences have shown that the most suitable medium for tuning of any TOMS is the properly modified flexible shaft coupling.

In our department, besides other activities, we are researching and developing a flexible pneumatic shaft coupling, particularly the differential pneumatic flexible shaft coupling with self-regulation, which works in given mechanical system as a "pneumatic tuner of torsional oscillations" (PToTO). Up to now, we focus our attention on the possibilities of application of mentioned coupling in the TOMS only in theoretical level.

The aim of this paper is to inform the technical community about the possibilities of control of dangerous oscillations of the mechanical systems by the means of new method, i.e., the application of pneumatic coupling with self-regulation, developed by us and also to present the calculation of equations for optimal tuning of the system.

2. TORSIONAL OSCILLATION CONTROL

The theory of oscillations defines that the m-mass mechanical system has (m – 1) natural frequencies of torsional oscillations, as well as the same number of corresponding forms of oscillations. To each of these oscillation forms corresponds a certain range of critical speeds, which is the result of resonance of harmonic parts of actuating load torque of piston machinery with the natural frequency corresponding to the oscillations form of the mechanical system. In the case of failure-free operation of piston device during the lowest forms of oscillations (one-nodal, two-nodal), the most dangerous are the revolutions actuated by the main harmonic part of actuating in the piston device, which we call the main critical revolutions. In the case of performance of machinery, a desired position is in above-resonance level, the main critical revolutions were positioned under the non-load revolutions, or at least, in 40%-distance from the minimal revolutions of running regime of the system. Then, in the range of running regime of the system, the remaining revolutions are so-called secondary critical revolutions, in which the level of torsional actuating is small. The level of torsional actuating of secondary critical revolutions will increase, if damage occurs in piston machinery. As the consequence of unequal actuating of individual pistons of piston machinery in range of working regime of the system, a very intensive resonance arising from lower harmonic parts of actuating occurs. [1], [2], [3] Dangerous torsional oscillation can be controlled by means of detailed analysis of the system from the torsional dynamic point of view.

This problem is relevant for every manufacturer of the flexible shaft couplings. The worldwide well-known manufacturers are trying to solve this problem by utilizing of highly flexible shaft couplings that is the couplings with the high torsional flexibility [1], [2], [15], [16], [17].

Our department is also involved into solving of this problem. We are trying to control the dangerous torsional oscillations by application of the pneumatic flexible shaft couplings, in particular by utilization of the differential pneumatic flexible couplings as the PToTO, developed by us [18], [29], [20] By means of utilization of the pneumatic couplings we are able to control the dangerous torsional oscillations and in this way to ensure tuning of the TOMS during the running regime of the system [21].

The basic task of tuning of the TOMS is a suitable adjustment of dynamic properties of pneumatic coupling within running regime of the TOMS [22], [23] and [24].

Furthermore we focus our attention on the possibilities of controlling of dangerous torsional oscillations in a real ships driving system during the current operation, by application of the pneumatic differential coupling in function of the PToTO.

2.1 DIFFERENTIAL PNEUMATIC FLEXIBLE SHAFT COUPLINGS AND THEIR USE

Pneumatic differential couplings, according to the figures (Figure 1 and Figure 2) have to fulfill following important requirements:

- compensation of axial, radial and angular misalignments, caused due to manufacturing inaccuracies,
- to keep stable dynamic mechanical properties and constant flexible transfer of loading torque during the whole technical life of the mechanical system,
- ability to tune torsional oscillating mechanical systems continuously.

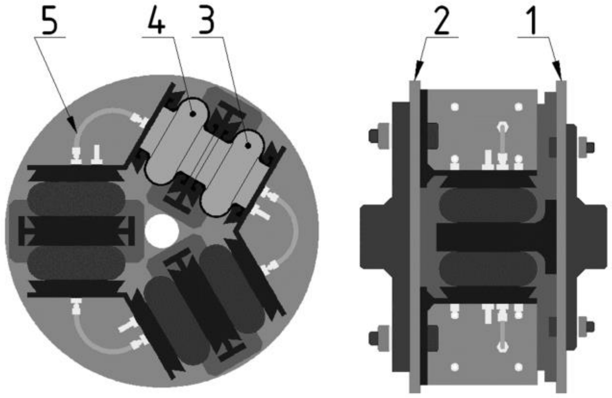

Figure 1: Schematics of a differential pneumatic flexible shaft coupling

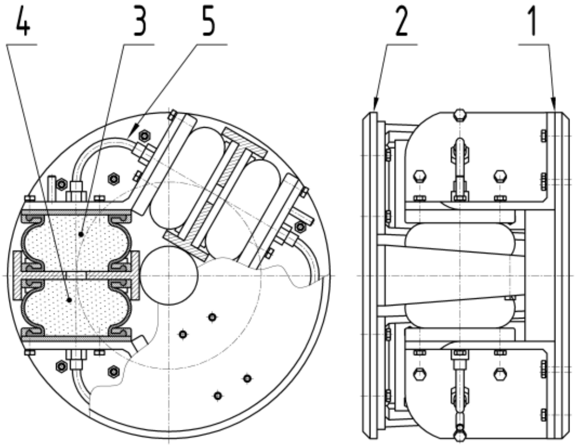

Figure 2: Real assembly of the pneumatic flexible differential shaft coupling

Pneumatic coupling, figures (Figure 1 and Figure 2), consists of the driving part (*1*) and the driven part (*2*), and between them is a compression space filled with a gaseous medium. The compression space consists of three circumferentially dislocated, differential elements, connected each other with the pipes (*5*). Every differential element has a compressed (*3*) and an expanded (*4*) pneumatic-flexible element.

By change of the gaseous medium pressure (p) in compression space of pneumatic coupling, it can change the characteristics of the coupling (Figure 3), as well as the characteristic mechanical properties of the coupling (torsional stiffness and dumping coefficient) can be changed. In this way there is change of, i.e. tuned, dynamic torsional stiffness (Figure 4), which is the most important factor of natural frequency in the given system ($\Omega = \sqrt{k/I_{red}}$. for a two mass system).

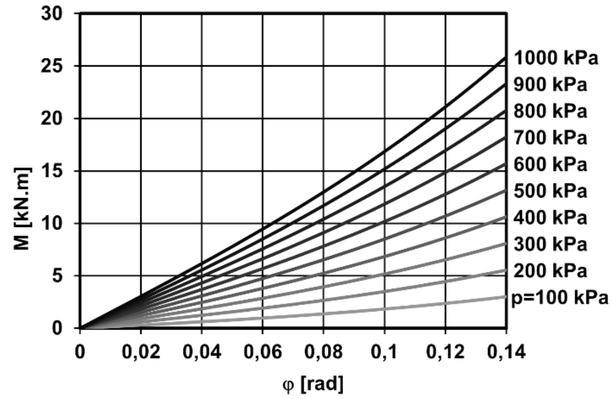

Figure 3: Static load characteristics of *3–1/210–D/A* type PToTO for gaseous medium pressures *p=100 ÷ 1000 kPa*

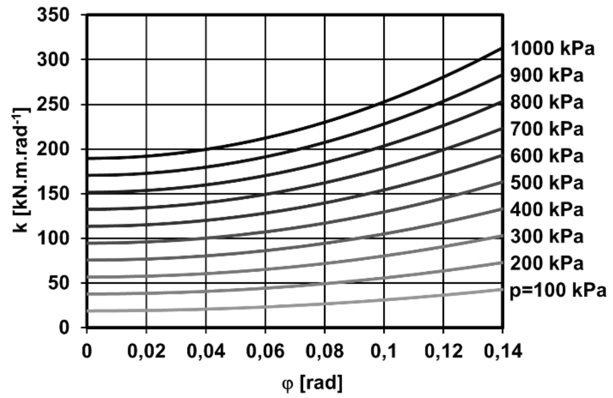

Figure 4: Dynamic torsional stiffness of *3–1/210–D/A* type PToTO for gaseous medium pressures *p = 100 ÷ 1000 kPa*

According to this fact the basic principle of continuous tuning of the TOMS by means of pneumatic couplings is evident, [26...33]. The natural angular frequency of the system (Ω) is modified with regard to the actuating angular frequency (ω), in order to avoid the resonance state ($\Omega = \omega$) or state very close to the resonance phenomenon.

Application of continuous tuning, according to the invention [26], requires also a new application of another kind of coupling. It is coupling, i.e. the PToTO, which is able to change its basic characteristics, like torsional stiffness and damping coefficient. This requirement is fulfilled in the case of pneumatic differential coupling, which is developed newly by us and which is able to operate in the function as torsional oscillations tuner (Figure 5).

Pneumatic tuner of torsional oscillations (Figure 5), protected by two patents for an invention [23], [24] is similar to the pneumatic differential coupling. The main difference consists in regulator (*6*), which enables to keep constant angle of twist in the coupling [24], [25], [26]. The basic principle of the PToTO is a self-regulation ability of the angular twisting, caused due to actual change of loading torque, into given constant angular value φ_k. This self-regulation of gaseous medium pressure in the compression space in tuner has an immediate influence on the characteristic of pneumatic tuner (Figure 3), and obviously, on the torsional stiffness *k* (Figure 6).

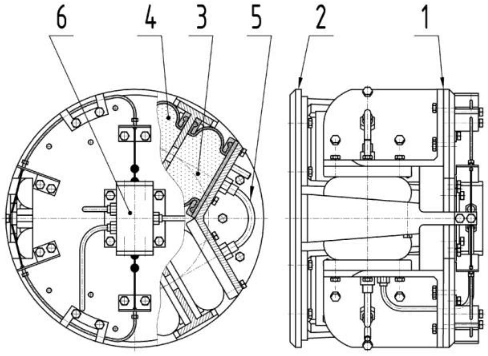

Figure 5: Pneumatic tuner of torsional oscillations

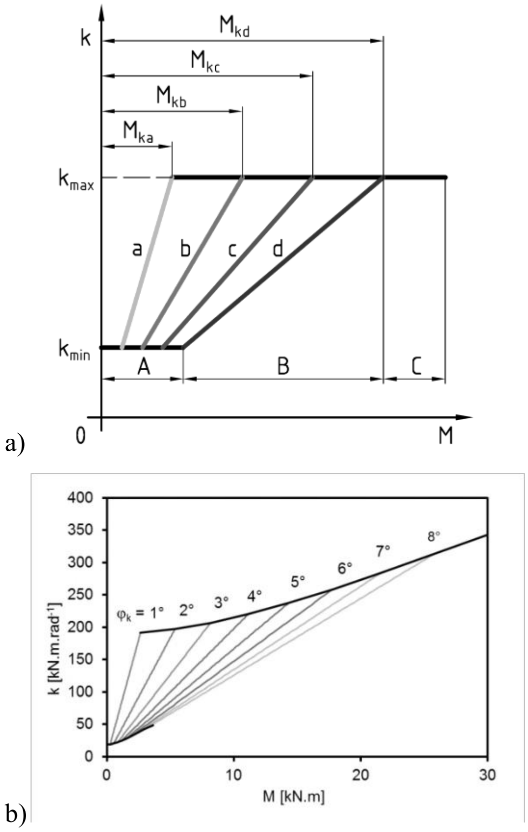

Figure 6: Behavior of torsional stiffness *k* in dependence on torque *M* and constant twist angle φ_k:
a) schematic representation for a linear PToTO,
b) for the *3–1/210–D/A* type PToTO

The above-mentioned behaviors are limited with minimum and maximum values of torsional stiffness k_{min} and k_{max} according to the pressures of gaseous medium from interval

p_{min} and p_{max} in compression space of the PToTO. There are also presented behaviors illustrated by a fractional line, which consists of three areas: pre-regulation – *A*, regulation – *B* and over-regulation – *C* area. From this illustration it is evident that change of φ_k influences the interval of pre-regulation and regulation area, but it influences mainly the value of pneumatic tuner torsional stiffness during an operational regime of the system. There are also influenced values of loading torque M_{ka}, M_{kb}, M_{kc} and M_{kd} with regard to the maximum value k_{max} of torsional stiffness. The pneumatic tuner with an increasing value of constant angle of twist, during a certain loading torque, has a declining torsional stiffness.

In the case of the PToTO with the maximum angle of twist value φ_{kmax}, from the relatively hard pneumatic coupling (behaviour *a*) becomes a high flexible pneumatic coupling (behaviour *d*), which is able to operate with considerably higher value of loading torque M_d at maximum value of dynamic torsional stiffness.

On the figure (Figure 6b.), dynamic torsional stiffness behaviour of the (nonlinear) *3–1/210–D/A* type PToTO is presented.

3. CHARACTERISTICS OF SELECTED MECHANICAL SYSTEM

Described torsional oscillating mechanical system (Figure 7) consists of driving part (*1*), driven part (*2*), first PToTO (*4*) with gearbox (*3*), second PToTO (*5*) and shaft (*6*). The driving part is the six-cylinder in-line Diesel engine – type *6–27,5 A 2 L S*, with power output $P = 515\ kW$ and speed range $n = 200 \div 600\ min^{-1}$. The driven part is a three-vane propeller with diameter *1700 mm*. The transmission of torque between the driving and driven parts consists of the PToTO - type *3-1/210-D/A*, reverse gearbox - type *VSR 10 B (3)* with gear ratio $i = 1{,}766$, PToTO - type *3 – 1/245 – D/A(5)* and shafts.

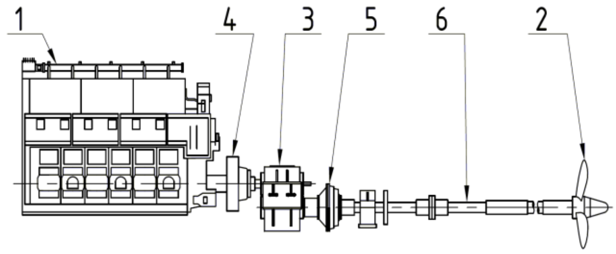

Figure 7: Examined torsional oscillating mechanical system

The properties of *3–1/210–D/A* type PToTO are determined by static torque *M* load characteristics (Figure 3) and dynamic torsional stiffness *k* (Figure 4) depending on static twist angle *φ*.

This driving unit is installed in a real ships system, namely in a river-push boat (Figure 8).

Figure 8: River-push boat

3.1 TORSIONAL ANALYSIS OF SELECTED MECHANICAL SYSTEM

Analysis of loading of two PToTO during a steady state operation of the mechanical system will be performed by means of a schematic model of torsional oscillating mechanical system (Figure 9).

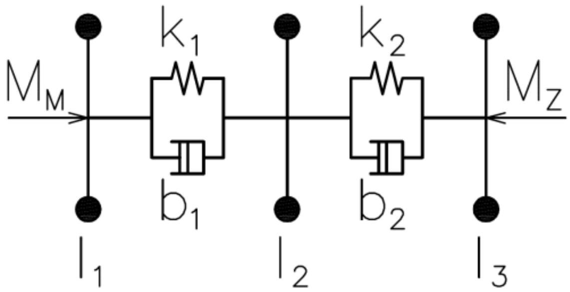

Figure 9: Schematic model of given torsional oscillating mechanical system

During a calculation process of the mechanical system loading during the steady state operation in the range of its working regime, it can be supposed that this mechanical system is rotating with angular speed *ω*, which is changing in framework of working regime. The mass (*1*) with moment of inertia I_1 is loaded by loading torque in the form: $M_Z + \sum M_j.\sin(j.\omega.t + \gamma_j)$. From this relation it is evident that the PToTO, as well as the whole torsional oscillating mechanical system, is loaded by loading torque with fan characteristics, i.e.: $M_Z = 0{,}043 \cdot n^2$ and there is also added a harmonic component of oscillation M_j. According to this, the PToTO has to also transmit additional component of dynamic torque M_d. So, in our case, the pneumatic tuner is loaded with loading torque M_K and maximum angle of twist φ_K:

$$M_K = M_Z + M_d, \tag{1}$$

$$\phi_K = \phi_Z + \phi_d. \tag{2}$$

The additional dynamic torque M_d and a dynamic component of maximum angle of twist φ_d can be calculated from equations of motion (*3*).

Equations of motion for the three-mass mechanical system:

$$\begin{aligned} &I_1 \cdot \ddot{\phi}_1 + b_1 \cdot (\dot{\phi}_1 - \dot{\phi}_2) + k_1 \cdot (\phi_1 - \phi_2) = M_M(t) \\ &I_2 \cdot \ddot{\phi}_2 - b_1 \cdot (\dot{\phi}_1 - \dot{\phi}_2) - k_1 \cdot (\phi_1 - \phi_2) + b_2 \cdot (\dot{\phi}_2 - \dot{\phi}_3) + k_{12} \cdot (\phi_2 - \phi_3) = 0 \\ &I_3 \cdot \ddot{\phi}_3 - b_2 \cdot (\dot{\phi}_2 - \dot{\phi}_3) - k_2 \cdot (\phi_2 - \phi_3) = -M_Z(t) \end{aligned} \tag{3}$$

Whereas the loading torque of engine is described in a complex form (4):

$$M_M = M_Z + M_d = M_Z + \sum_{j=1}^{n} M_j \cdot e^{i \cdot (\omega \cdot j \cdot t - \Delta\phi_{Mj})} \tag{4}$$

where:

M_Z – is loading torque, this torque is stationary in steady state of the system,

M_d – is component of additional engine's dynamic torque,

M_j – is amplitude of *j*-th harmonic component of additional engine's dynamic torque,

$\Delta\varphi_{Mj}$ – is the phase of *j*-th harmonic component of additional engine's dynamic torque.

Natural speed frequencies of this system are:

$$N_{1,2} = \frac{\Omega_{1,2} \cdot 60}{2 \cdot \pi} , \tag{5}$$

where $\Omega_{1,2}$ are natural angular frequencies of the system, according to the relation:

$$\Omega_{1,2} = \sqrt{\frac{q_2 \mp \sqrt{q_2^2 - 4 \cdot q_1 \cdot q_3}}{2 \cdot q_1}} , \tag{6}$$

where coefficients q_1, q_2 and q_3 are calculated from the next equations:

$$q_1 = \frac{I_1 \cdot I_2 \cdot I_3}{k_1 \cdot k_2} , \tag{7}$$

$$q_2 = \frac{(I_2 + I_3) \cdot I_1}{k_1} + \frac{(I_1 + I_2) \cdot I_3}{k_2} , \tag{8}$$

$$q_3 = I_1 + I_2 + I_3 \tag{9}$$

4. THEORETICAL RESULTS OF THE OPTIMAL TUNING OF THE SHIPS SYSTEM

We are using two PToTO [33] for tuning the ship propulsion system [34, 35]. The first PToTO located after engine has a constant value twist angle $\varphi_{K1} = 3,2°$, and the second PToTO located after gearbox has a constant value of twist angle $\varphi_{K2} = 2,5°$. Constant twist angles were optimized so that the natural frequencies lie as close as possible to *0,75*-th and *1,25*-th harmonic components, because in reality, an engine torque doesn't contain these harmonic components, and they lie in the middle between existing minor harmonic components of the engine torque.

On Campbell's diagram (Figure 9), the two natural frequencies of examined three-mass mechanical system are displayed.

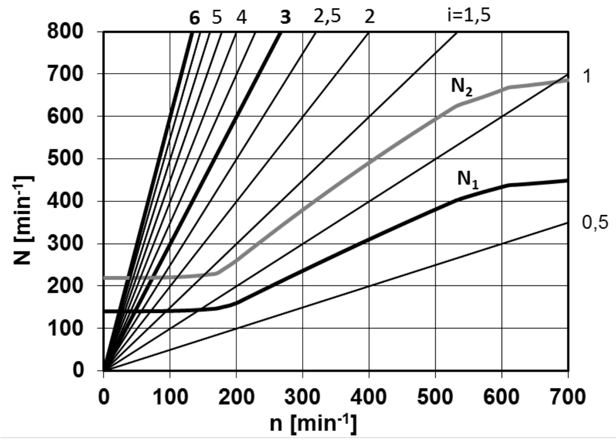

Figure 9: Campbell's diagram of given ships system

Based on the above-mentioned facts it is possible to say that there is no risk of any dangerous torsional oscillation in the range of the operational speed. This fact is supported by the figure (Figure 10), where the behaviors of amplitudes of dynamic torque for both couplings $M_{d1,2}$ depending on the speed *n* for the given ships system are displayed.

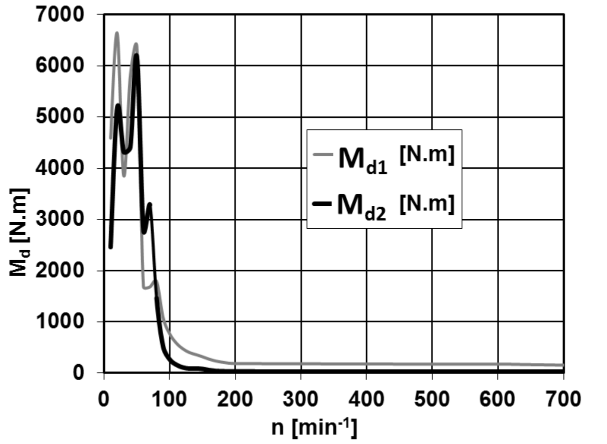

Figure 10: Behaviors of amplitudes of dynamic torque M_d depending on the speed *n* for the given ships system

For the speed values from the interval $n = 0 \div 200\ min^{-1}$ there is increased amplitude of the dynamic moment M_d. The given figure shows peaks of torque M_d amplitudes in the interval $n = 20 \div 80\ min^{-1}$, what are the resonances with the major harmonic components ($j = 3$ and $j = 6$) from the first and the second natural speed frequencies. If the operational speed is more far away from the resonance area, the loading of individual flexible couplings descends, what is favorable situation for the whole ships system. Loading of the individual flexible couplings in the range of operational speed $n = 200 \div 700\ min^{-1}$ is following:

- in the first PToTK: $M_{d1} = 186 \div 156\ N.m$,
- in the second PToTO: $M_{d2} = 34 \div 30\ N.m$.

According to the all above-mentioned results it is possible do declare that by application of two PToTO with constant values of angles of twist $\varphi_{K1} = 3{,}2°$, and $\varphi_{K2} = 2{,}5°$, we are able to achieve a smooth operation of the ships system with regard to its torsional oscillation.

5. CONCLUSIONS

Results from this work confirmed the important fact that the dangerous torsional oscillations in the mechanical system can be reduced into an acceptable level by means of a suitable modification of dynamic properties of flexible couplings in this system. Torsional oscillating mechanical systems have to be tuned in advance. For the proper tuning of torsional oscillating mechanical systems it is necessary to perform detailed dynamic calculation with regard to the torsional oscillation. Taking into consideration the above-mentioned facts, our suggestion is: application of the pneumatic flexible shaft couplings developed by us for tuning of the ships system in order to reduce dangerous torsional oscillations. These flexible shaft couplings are in fact the pneumatic tuners of torsional oscillations. They have not only one, but the whole range of characteristics and characteristic features in the framework of gaseous medium pressure in the compression area with regard to chosen constant value of angle of twist.

During application of pneumatic tuners of torsional oscillations with the constant angles of twist $\varphi_{K1} = 3{,}2°$, and $\varphi_{K2} = 2{,}5°$, no resonance occurred in the whole range of operational regime because the pneumatic tuner fulfils also a function of high-flexible shaft coupling after the change of constant angle of twist.

The use of two PToTO in a three mass torsional oscillating mechanical system gives us the possibility to tune both natural frequencies. We can theoretically tune the natural frequencies so that their value is nearly a constant multiple of operating speed. But this is limited by the operating range of gaseous medium pressure in PToTO.

Finally, it is possible to say, that this tuning method during current operation (continuous tuning of systems) can be applied in every situation where dangerous torsional oscillation of mechanical systems should be eliminated. The results presented in this paper confirm our presumption that these new tuning methods are able to improve technical standard and operational reliability in all torsional oscillating mechanical systems.

6. ACKNOWLEDGEMENTS

This paper was written in the framework of Grant Projects: „*VEGA 1/0197/14 Research of the new methods and innovative design solutions to increase efficiency and to reduce emissions of a vehicle drive unit with an assessment of the potential operational risks*."

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