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COMPARISON OF PISTON AND TANGENTIAL PNEUMATIC FLEXIBLE SHAFT COUPLINGS IN TERMS OF HIGH FLEXIBILITY

Summary. The optimal tuning of mechanical systems in terms of torsional vibration magnitude is a very important function in flexible shaft couplings. Therefore, a flexible coupling with suitable dynamic properties, particularly dynamic torsional stiffness, has to be carefully chosen for each specific application. The current trend in the field of flexible shaft couplings, and the most noticeable in the automotive industry, is the development and utilization of highly flexible couplings, which means flexible couplings with a very low value of relative torsional stiffness. The aim of this article is to introduce a new type of flexible shaft coupling: a piston pneumatic flexible shaft coupling. This coupling was developed to improve the properties of pneumatic flexible couplings, especially the maximum angle of twist, in order to create a highly flexible pneumatic coupling. For illustration purposes, the piston pneumatic coupling is compared with the tangential pneumatic flexible shaft coupling of Type 3-1/110-T-C in terms of high flexibility characteristics, whereby the characteristic dimensions of both couplings are the same. Given that the piston pneumatic coupling has not been manufactured to date, only a computational model of this coupling was used. The results show that the design of the piston pneumatic flexible shaft coupling combines the advantages of a highly flexible and pneumatic shaft coupling.

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1. INTRODUCTION

Nowadays, reducing vibration and noise in machinery is a highly important task, mainly in terms of human health and the lifetime and safety of machines, e.g., [2-6,8,9,13]. Flexible shaft couplings are the most used machine parts for the flexible transmission of load torque and mechanical energy in mechanical systems. Another highly important function performed by them is the dynamic tuning of mechanical systems in terms of torsional vibration magnitude. Therefore, a flexible coupling with suitable dynamic properties, particularly dynamic torsional stiffness, has to be carefully chosen for each specific application to ensure that dangerous torsional vibration does not occur in a mechanical system, e.g., [8,9,12**Błąd! Nie można odnaleźć źródła odwołania.**]. From the point of view of the aforementioned dynamic tuning, the development and utilization of highly flexible couplings are particularly advantageous and most noticeable in the automotive industry nowadays (dual-mass flywheels). A highly flexible coupling possesses a very low relative torsional stiffness k_0. Relative torsional stiffness is expressed by the following formula:

$$k_0 = \frac{k_{DN}}{M_N}\left[rad^{-1}\right], \tag{1}$$

which is defined as the ratio of the nominal dynamic torsional stiffness of a coupling k_{DN} (at M_N) to the nominal torque M_N of a coupling. Common flexible couplings have a relative torsional stiffness value in the range of 10÷30 rad^{-1}. Shaft couplings marked as highly flexible have a relative torsional stiffness value lower than 10 rad^{-1}. With the application of a highly flexible coupling in a mechanical system (Coupling 2 in Fig. 1), the resonances from the individual harmonic components of a torsional vibration excitation can be moved from the operating speed (n) range (OSR) of the system to the low-speed area, which is far enough under the idle operating speed n_V, e.g., [6,9]. This low-speed area can be quickly run across at the start-up of a mechanical system, as shown in Campbell's diagram of a mechanical system (Fig. 1), where i represents the order of the harmonic component of a torsional vibration excitation.

Nowadays, the flexible elements of flexible shaft couplings are made of various materials. During the operation of mechanical systems, rubber and plastic flexible elements experience fatigue and ageing, while metal flexible elements of applied flexible couplings also experience ageing and wear and tear [1,7,15]. Consequently, an applied flexible coupling loses its original dynamic properties and thus its ability to carry out important functions in a torsional oscillating mechanical system. Flexible shaft couplings from the pneumatic flexible shaft couplings group (to which the following couplings belong, according to the granted patents: SK 288455 B6, SK 288344 B6, SK 288341 B6, SK 278750 B6, SK 278653 B6, SK 278152 B6) are able to facilitate the flexible transmission of mechanical energy without the loss of their characteristic properties, because the gaseous medium in the compression volume of couplings does not suffer from fatigue or ageing. The main advantage of pneumatic couplings is the possibility to change their torsional stiffness, which depends on the pressure value of the gaseous media. Based on the aforementioned grounds, the development of flexible couplings, with the benefits of both pneumatic and highly flexible couplings, is very advantageous.

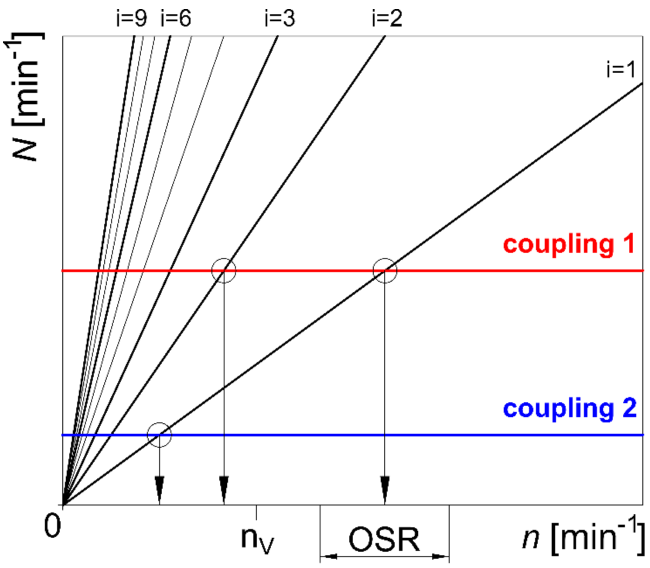


Fig. 1. Campbell's diagram of a mechanical system

From the point of view of physics, a flexible coupling with a low torsional stiffness must have a large twist angle in order to transmit a high load torque. This is the next prerequisite for creating a highly flexible coupling. Therefore, the aim of this article is to introduce a piston pneumatic flexible shaft coupling, which was developed to improve the properties of pneumatic flexible couplings, especially the maximum angle of twist, in order to create a highly flexible pneumatic coupling. For illustration purposes, this coupling is compared with the tangential pneumatic flexible shaft coupling of Type 3-1/110-T-C in terms of high-flexibility characteristics, in which the characteristic dimensions of both couplings are the same.

2. EXAMINED PNEUMATIC FLEXIBLE SHAFT COUPLINGS

In the following, the piston pneumatic flexible shaft coupling of Type 2-1/110-P-C[2] (Fig. 2) is compared to the tangential pneumatic flexible shaft coupling of Type 3-1/110-T-C[3] (Fig. 3), manufactured by the FENA company in cooperation with our department. Both couplings are designed to transfer load torque in one sense (their pneumatic flexible elements must be pressed), have the same pitch diameter, D_R=180 mm, and the same outer diameter of pneumatic elastic elements, D_E=110 mm. Given that the piston pneumatic coupling has not yet been manufactured, only a computational model of this coupling was used.

2.1. Piston pneumatic flexible shaft coupling of Type 2-1/110-P-C

The piston pneumatic flexible shaft coupling (Fig. 2) consists of a driving flange (1), a driven flange (2), pneumatic flexible elements (4), curved hollow cases (5), curved piston bodies (3), fastening flanges (6) and valves (7). The compression volume of the coupling is created from two pneumatic flexible elements (4), which are motionlessly placed in the

[2] Urbanský, M., Homišin, J. Pneumatic Flexible Piston Coupling. Patent No. SK288390 (B6).
[3] Homišin, J. Pneumatic Flexible Shaft Coupling. Patent No. SK222411 (B6).

hollow cases (5), which are attached to the driven flange (2). The piston bodies (3) are attached to the driving flange (1). The pneumatic flexible elements (4) are inflated to the required overpressure of gaseous media through the valves (7), and the basic position of the piston bodies and the driving flange (1) in relation to the driven flange (2) is herewith defined (Fig. 2a). The transmission of load torque causes a twist in the driving flange (1) in relation to the driven flange (2), and the piston bodies (3) are therefore pushed into the pneumatic flexible elements (4) so that the piston bodies (3) are coated with the pneumatic flexible elements (4) (Fig. 2b). The design of the coupling allows for a maximum angle of twist of 76°.

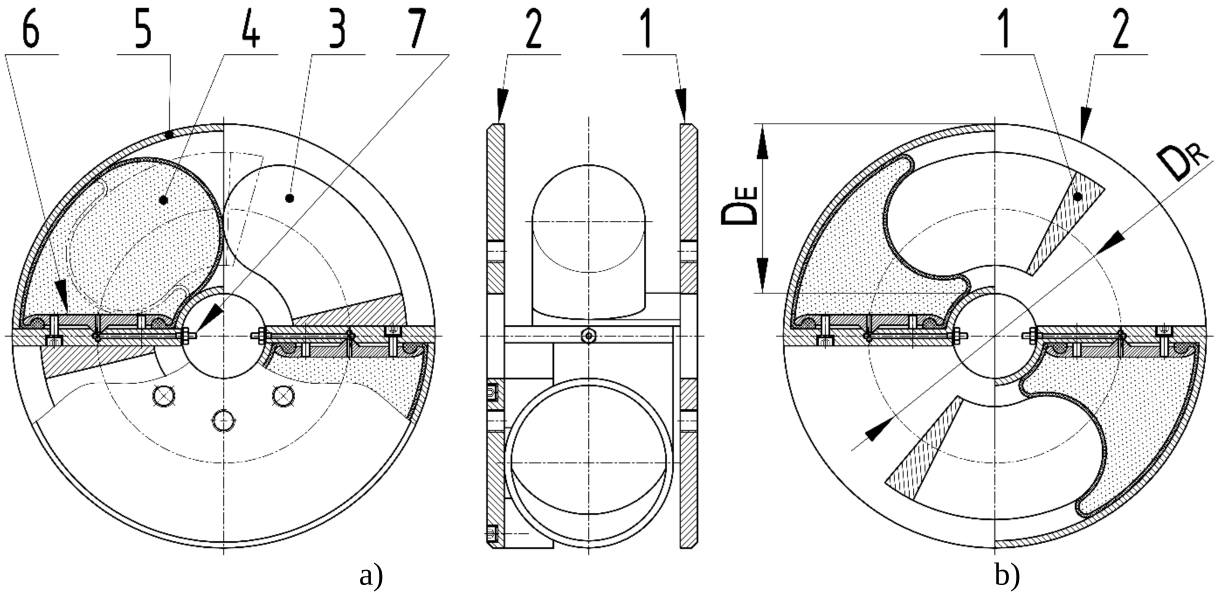


Fig. 2. The piston pneumatic flexible shaft coupling of Type 2-1/110-P-C: a) unloaded state; b) partially loaded state

The compression of gaseous media in the pneumatic flexible elements (4) is proportional to the load, resulting in the flexible transmission of load torque in mechanical systems.

2.2. Tangential pneumatic flexible shaft coupling of Type 3-1/110-T-C

By this coupling (Fig. 3), load torque is transmitted flexibly from the driving flange (1) to the driven flange (2) by the compression volume of the coupling. The compression volume is created from three pneumatic flexible elements (3), which are tangentially positioned around the perimeter of the coupling and completely interconnected with the hoses (4). The gaseous medium streams into the compression volume of the coupling through the quick-acting pneumatic fitting (5). The design of the coupling allows for a maximum angle of twist of 15°.

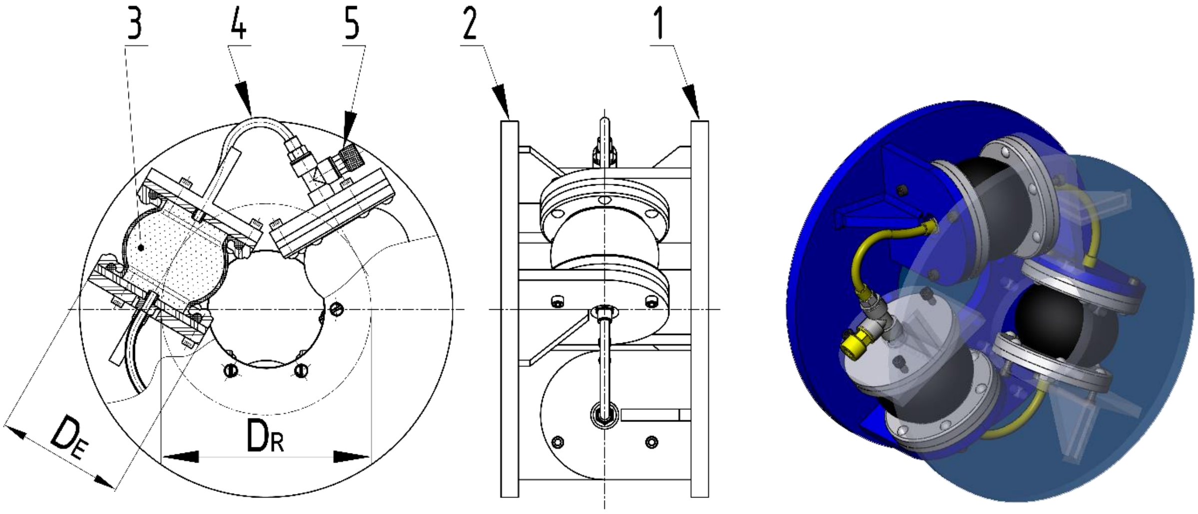


Fig. 3. The tangential pneumatic flexible shaft coupling of Type 3-1/110-T-C

3. DETERMINING THE BASIC CHARACTERISTIC PROPERTIES OF THE COUPLINGS

3.1. Dependence of pressure in the pneumatic couplings on their twist angle

As for pneumatic flexible shaft couplings, load torque is transmitted flexibly from the driving flange to the driven flange by pneumatic flexible elements. As the transmission of load torque causes a twist in the driving flange in relation to the driven flange, the compression volume of a pneumatic coupling is compressed. The compression of gaseous media in the pneumatic flexible elements is proportional to the load.

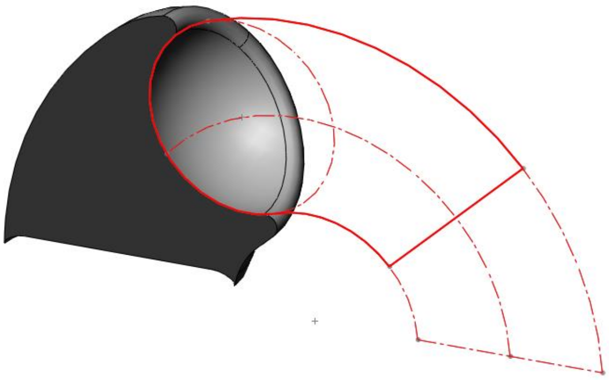


Fig. 4. The principle of modelling the air volume deformation caused by the piston body, as a section view

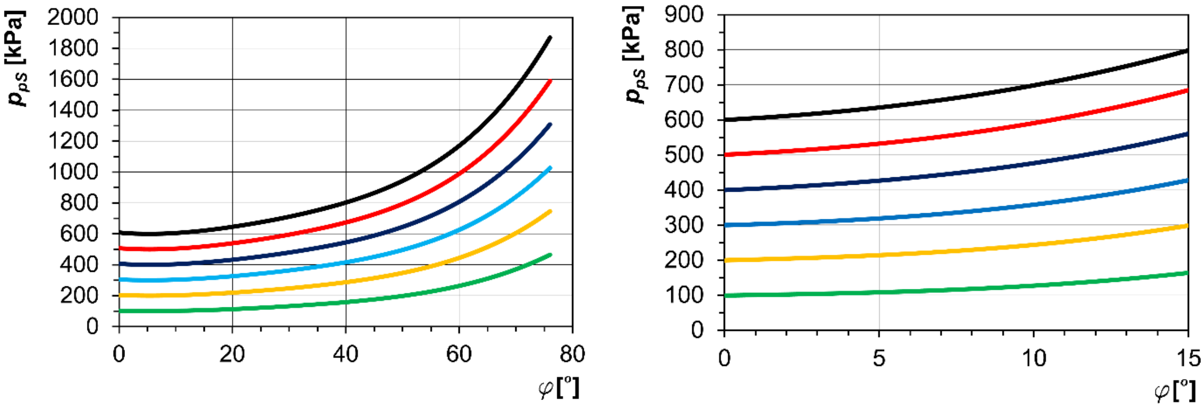
In Fig. 5, we can see the dependencies of air overpressure p_{pS} in the compression volume of the pneumatic couplings on their angle of twist φ. Initial air overpressure values (at φ=0°) in the couplings were p_{pS0}=100÷600 kPa. For the tangential coupling, the dependencies in Fig. 5b were measured and the maximally allowed p_{pS} value was 800 kPa [10]. For the piston coupling, the dependencies in Fig. 5a were computed as follows:

1. The dependence of the air volume V on the twist angle φ was determined using 3D CAD software (Fig. 4)

2. The dependence of the air overpressure p_{pS} in the pneumatic coupling on the twist angle φ was determined by considering isothermal compression using the following formula:

$$p_{pS(\varphi=x)} = \left(p_a + p_{pS0}\right)\cdot\left(\frac{V_{(\varphi=0)}}{V_{(\varphi=x)}}\right) - p_a;\ \ x \in (0, \varphi_{\max}\rangle\,, \qquad (2)$$

where p_a is the atmospheric pressure (p_a=101,325 Pa).



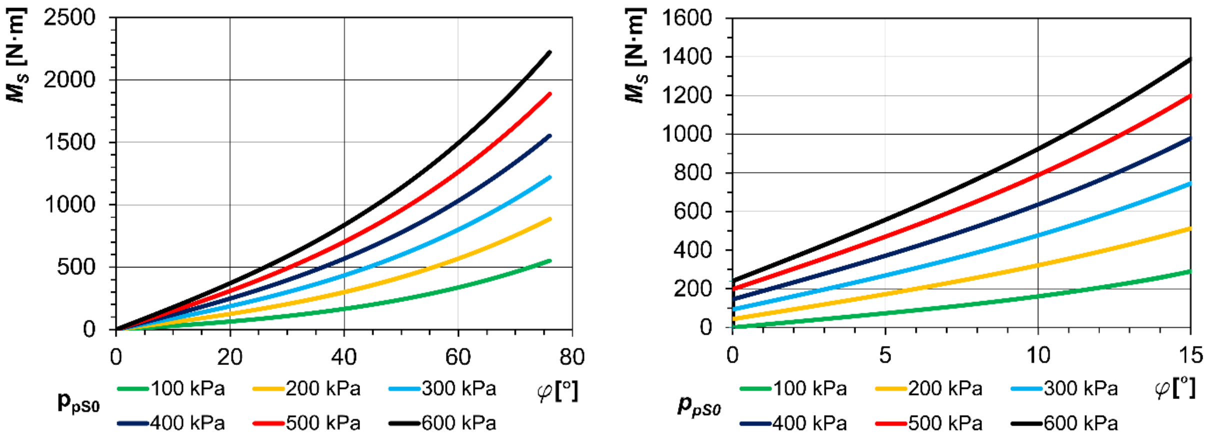
a) b)

Fig. 5. The air overpressure p_{pS} in the couplings, which is dependent on the twist angle φ at various values of p_{pS0}: a) piston coupling of Type 2-1/110-P-C; b) tangential coupling of Type 3-1/110-T-C

3.2. Static load characteristics

According to the standard [16], the static load characteristic of a flexible coupling is the dependence of the coupling twist on load torque during slow change in the load torque. The loading and unloading of a coupling should be performed smoothly at a constant speed <0.002 rad/s^{-1}. Five cycles of loading and unloading should be performed and the static characteristic should be obtained as the loading part of the fifth cycle. The coupling should be loaded up to the maximum static load torque M_{Smax}, which can be defined, for example, according to the maximum twist angle of a flexible coupling. The condition is not to damage the coupling. In Fig. 6, we can see the static characteristics of both couplings at initial air overpressure values in their compression volume p_{pS0}=100 ÷ 600 kPa. The static characteristics of the tangential pneumatic coupling (Fig. 6a) were determined in the laboratory of our department and the static characteristics of the piston pneumatic coupling (Fig. 6b) were computed from the equality of the works of the gaseous medium and torque:

$$p_{pS} \cdot dV = M_S \cdot d\varphi \Rightarrow M_S = p_{pS} \cdot \frac{dV}{d\varphi}. \tag{3}$$



a) b)

Fig. 6. The static load characteristics at various values of p_{pS0}: a) piston coupling of Type 2-1/110-P-C; b) tangential coupling of Type 3-1/110-T-C

The static load characteristics of both couplings are slightly non-linear and can be described by cubic equations in the form, $M_S=a_0\cdot\varphi + a_3\cdot\varphi^3$. The coefficients a_0 and a_3 (Tab. 1) were determined by the method of least squares.

Tab. 1.

The coefficients a_0 and a_3 of static load characteristic equations

Piston coupling **2-1/110-P-C**			Tangential coupling **3-1/110-T-C**		
p_{pS0}	a_0	a_3	p_{pS0}	a_0	a_3
100 kPa	820.0	3,858.9	100 kPa	168.4	140.5
200 kPa	1,434.8	5,178.2	200 kPa	338.4	187.0
300 kPa	1,959.6	7,718.2	300 kPa	508.4	233.6
400 kPa	2,506.6	9,851.8	400 kPa	678.3	280.1
500 kPa	3,030.4	11,563.2	500 kPa	848.3	326.6
600 kPa	3,532.3	12,414.5	600 kPa	1,018.3	373.1

According to the standard [16], the nominal torque M_N of a flexible coupling can be determined as the third of its maximum static load torque M_{Smax}. In Fig. 7, we can see the M_{Smax} and M_N dependencies on initial air overpressure p_{pS0} for the piston pneumatic coupling (Fig. 7a) and tangential pneumatic coupling (Fig. 7b).

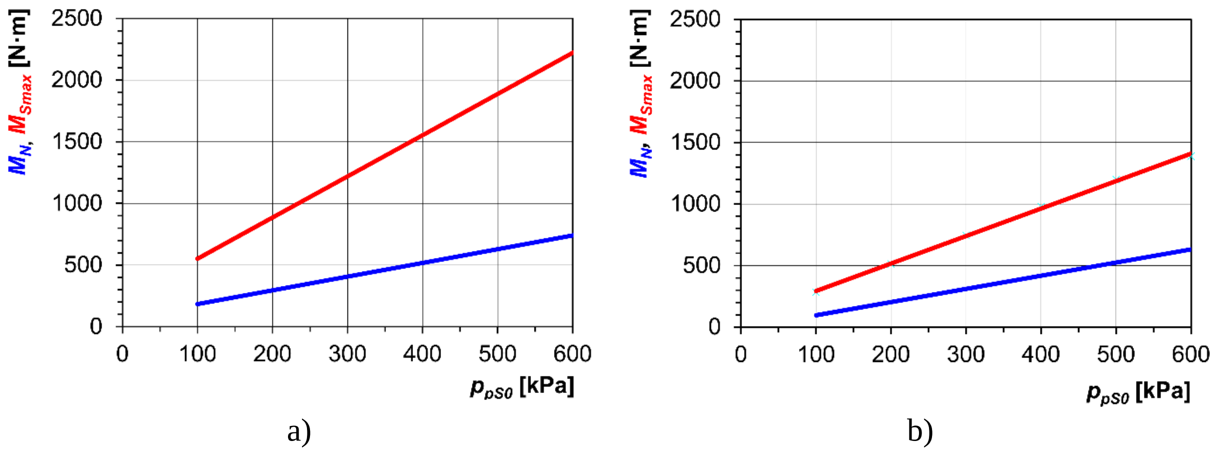
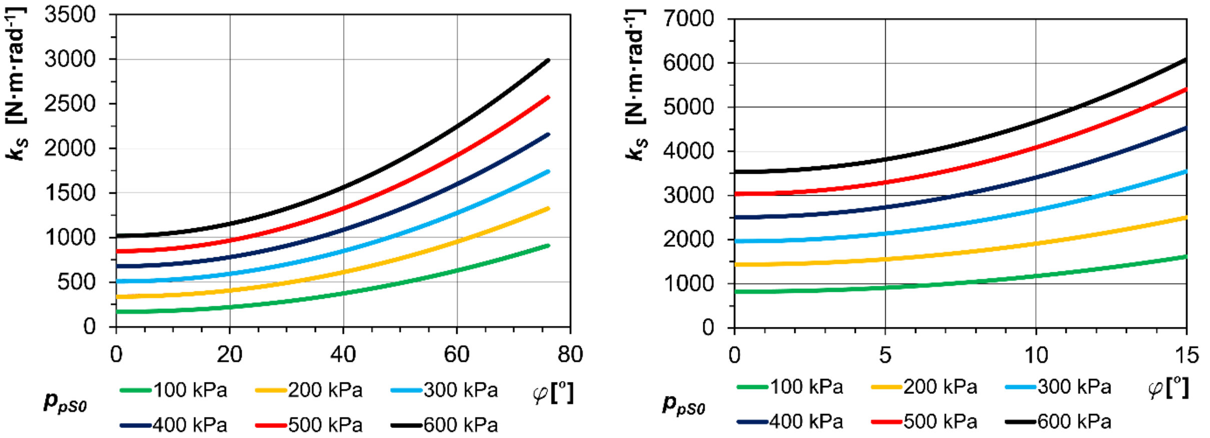


Fig. 7. Maximum static torque M_{Smax} and nominal torque M_N, which are dependent on initial air overpressure p_{pS0}: a) piston coupling of Type 2-1/110-P-C; b) tangential coupling of Type 3-1/110-T-C

3.3. Static torsional stiffness

The dependence of the static torsional stiffness k_S of a flexible coupling on the twist angle φ can be computed by deriving the equation of the static characteristic of a flexible coupling.

In Fig. 8, we can see the static torsional stiffness k_S, which is dependent on the twist angle φ of the couplings at initial air overpressure values in the pneumatic couplings p_{pS0}=100÷600 kPa for the piston pneumatic coupling (Fig. 8a) and tangential pneumatic coupling (Fig. 8b).



a) b)

Fig. 8. The dependence of the static torsional stiffness k_S on the twist angle φ at various values of p_{pS0}: a) piston coupling of Type 2-1/110-P-C; b) tangential coupling of Type 3-1/110-T-C

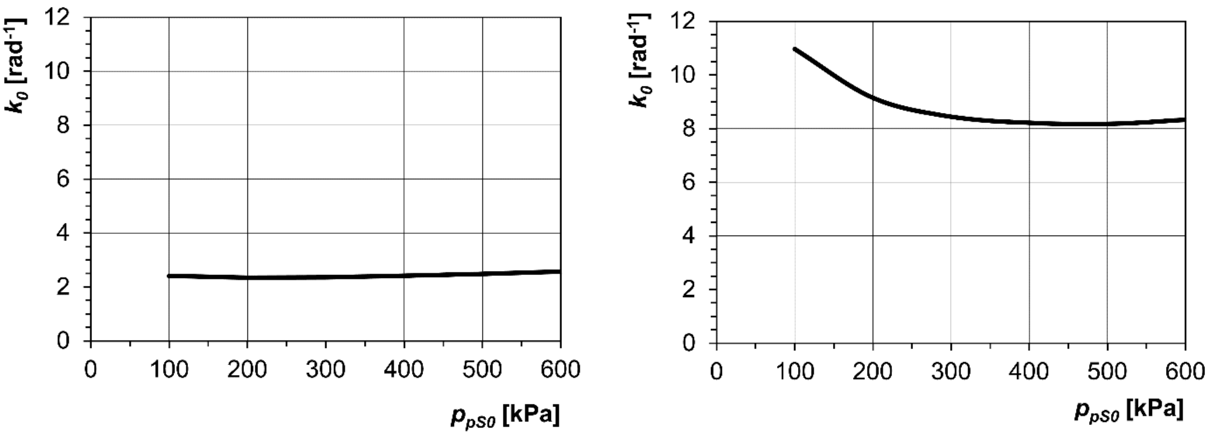
4. HIGH-FLEXIBILITY CHARACTERISTICS OF THE COUPLINGS

In order to compute the relative torsional stiffness values k_0 of the pneumatic couplings, the values of the dynamic torsional stiffness k_{DN} of the pneumatic couplings at the nominal torque M_N need to be determined. According to the research, the k_{DN} values of the pneumatic couplings can be determined by the following formula:

$$\frac{k_{DN}}{k_{SN}} = 1{,}05 + 4{,}14.10^{-4}.p_{pS0}\,, \tag{4}$$

where k_{SN} is the value of the static torsional stiffness of the pneumatic coupling at the nominal torque M_N at a certain value of initial air overpressure in the pneumatic coupling in the range of p_{pS0}=100÷600 kPa.

Finally, the relative torsional stiffness k_0 of the couplings can be determined according to Eq. (1). In Fig. 9, we can see the relative torsional stiffness k_0, which is dependent on initial air overpressure in the pneumatic couplings p_{pS0}=100÷600 kPa for the piston pneumatic coupling (Fig. 10a) and the tangential pneumatic coupling (Fig. 10b).



a) b)

Fig. 9. The relative torsional stiffness k_0, which is dependent on initial air overpressure p_{pS0}: a) piston coupling of Type 2-1/110-P-C; b) tangential coupling of Type 3-1/110-T-C

As we can see from Fig. 9, the piston pneumatic flexible coupling of Type 2-1/110-P-C meets the requirements for high flexibility in the whole range of the initial air overpressure p_{pS0} within its compression volume, while the tangential pneumatic flexible coupling of Type 3-1/110-T-C can be marked as highly flexible at approximately p_{pS0}>150 kPa.

5. CONCLUSION

With the transfer of load torque by the compression volume of pneumatic flexible shaft couplings filled with a gaseous medium, we achieve the compression of the medium proportional to load. This is how the continuous flexible load torque transmission in the driving and driven machine system is characterized. Gaseous media throughout their lifetime are not subject to ageing, meaning that pneumatic couplings do not lose their initial positive dynamic properties.

From the above-presented graphs and tables, we can see that the Type 2-1/110-P-C piston pneumatic coupling, in comparison with the Type 3-1/110-T-C tangential pneumatic flexible coupling, offers the following advantages:

- It has lower values of torsional stiffness (Fig. 8), while its transmission ability is higher (Figs. 6-7), thanks to its large maximum twist angle. The piston pneumatic coupling is able to transmit about 160% larger torque by the same pitch diameter D_R. The outer diameter of both couplings is approximately the same too.
- According to Fig. 9, both compared pneumatic couplings are highly flexible. The piston coupling has much smaller values of relative torsional stiffness (Fig. 9), which means that it has a higher flexibility than the tangential coupling.
- The design of the piston coupling allows for M_S=0 at φ=0 (Fig. 6); in other words, the piston coupling do not have initial torsional rigidity unlike the tangential coupling.

Meanwhile, the piston coupling has the following disadvantages:

- An unconventional pneumatic flexible element needs to be developed and manufactured for the piston pneumatic coupling.
- To achieve the presented transmission ability (Figs. 6-8), the piston coupling needs to work with relatively high levels of overpressure in its compression space (Fig. 5).

Both compared couplings are designed to transfer load torque in one sense, while their pneumatic flexible elements are pressed. Therefore, neglecting the influence of the rubber-cord coating of pneumatic flexible elements [14], the difference between static and dynamic torsional stiffness in the pneumatic couplings is mainly caused by a polytrophic process in their compression volume under dynamic load [11]. Under dynamic load, pneumatic flexible elements do not generate too much heat, unlike the flexible elements made of rubber and plastic [7]; therefore, the pneumatic couplings can work with large amplitudes of twist at high frequencies.

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