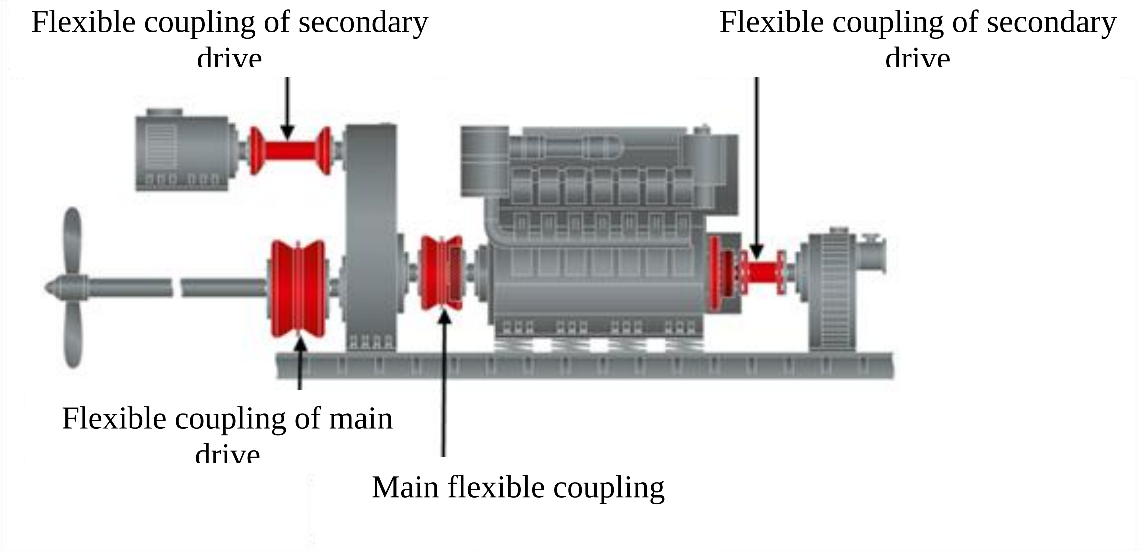
Flexible coupling for mechanical systems

Flexible shaft couplings are widely used in various torque-transmitting drives. Such drives transmitting mechanical energy are called mechanical systems. Flexible couplings can be used as part of both main and secondary drives. The Austrian company Geislinger, which deals with the production and application of flexible couplings, believes that each power branch driven by an internal combustion engine should be provided with a flexible coupling fig.

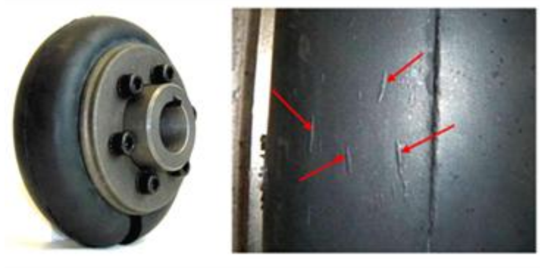


Thus, flexible shaft couplings must be able to transmit torque, but above all they must protect the mechanical system from torsional vibrations. This protection against torsional vibrations must be provided in a wide range of operating modes and operating speeds, not only in transient states during start-up and braking of the mechanical system. Another additional feature of flexible couplings, which results from their construction, is the elimination of expansion inaccuracies caused by the production and assembly of individual parts of the mechanical system.

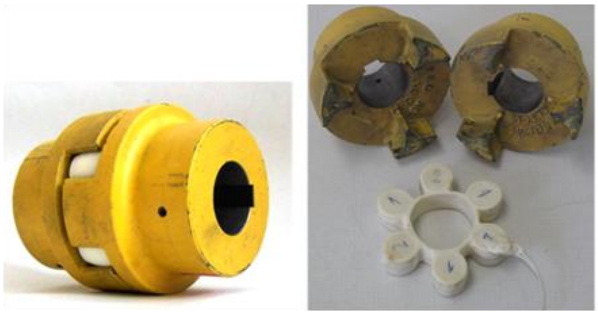
Improper design of the flexible coupling can cause serious failures of the coupling itself, but also of individual devices in the mechanical system. The initial failures of the couplings are usually manifested by the formation of cracks or undue deformations of the resilient members. Interestingly, such faults are often attributed to faults from elevated clutch temperatures. Research has shown that an inappropriate design of the coupling will cause its excessive dynamic load, which will cause a rise in temperature in the coupling, which is the cause of cracks and excessive deformation of the elastic members.

We also prove our claim with practical examples of failures of elastic connecting elements. The picture shows three different types of flexible couplings A) B-flex RB 116-4, B) Periflex PNA 10R, C) Gurimax GVW 100. Torsional vibrations of elastic couplings caused self-heating of elastic elements and consequently increased temperature caused failure of elastic elements. The experiments were performed at an ambient temperature of 20 ° C. The failures of the elastic elements are indicated by arrows.

A)



B)



C)

For flexible couplings, which we want to use for tuning or. to tune the mechanical system, it is appropriate to accurately identify their properties - especially torsional stiffness. Since the aim of a suitable tuning of a torsionally oscillating mechanical system is that the resonance from the main excitation frequency ω (resonance excited by the main harmonic component of the load torque) lies below the idle speed of the system. Thanks to this, during a fast start or they will not develop inadmissible torsional oscillations and will also be damped by the clutch. Then in the area of the operating mode of the system remain the so-called secondary frequencies i.ω, at which the yield of torsional oscillations is small in the case of trouble-free operation of the piston device.

Based on the equation, it is possible to state that the appropriate tuning of the system, as well as the magnitude of its torsional vibration depends on the degree of tuning η (it is recommended that the value 0.8> η> 1.4), whose magnitude is largely influenced by dynamic torsion stiffness k_{edyn} of the flexible shaft coupling.

$$\eta = \frac{i.\omega}{\Omega} = \frac{i.\omega}{\sqrt{k_{edyn} / I_{red}}}$$

Tuning of the mechanical system (drive) using a flexible coupling

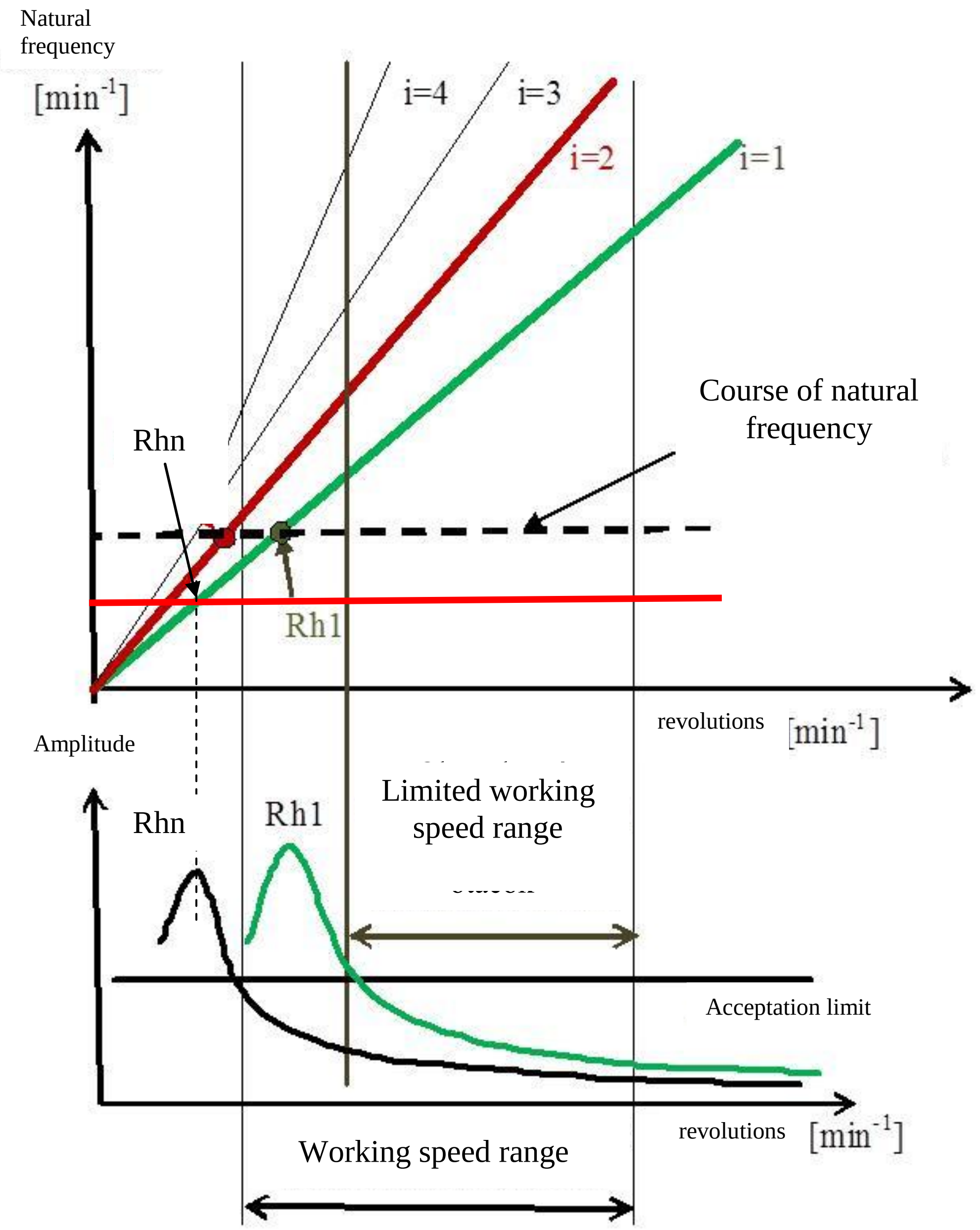
The basic principle of tuning based on the above equation can be clearly presented in the Campell diagram. The Campell diagram is shown in the following figure, the diagram shows on the vertical axis the natural frequency (natural speed of the system) on the horizontal axis the excitation frequency, resp. excitation (working) speed. The thin slashes show the waveforms of the excitation harmonics. For the illustrative case explaining the effect of the flexible coupling on the tuning of the mechanical system, the dominant harmonic component is indicated by a coarser green color. Below the Campell diagram is an associated amplitude-frequency diagram, which shows the magnitude of the oscillation amplitude at the excitation frequency. The required working area in which the mechanical system is to work - the working speed of the drive - is to be bounded by vertical solid lines. It is required that there are no resonances from the main excitation component in this work area - according to the formula above. The Campell diagram further indicates the waveforms of the natural frequencies of the drive (dashed thick line and thick red line). These waveforms are defined by the natural frequency of the drive Ω.

From FIG. it is obvious that the dashed line showing one of the courses of the natural frequency intersects with the main harmonic component (green line) at the point Rh1- at this point a resonance occurs. The course of the amplitude of the oscillation of the drive at the natural frequency of the value (dashed line) is shown in the amlitude frequency characteristic by a green line. As we can see, there is a significant increase in oscillation in the required working speed range - above the value of the acceptance criterion.

Such a drive is therefore unusable for the required operating range!

The solution is offered in two forms:
1. Limit the working range so that the resonance is below the working speed. This will have a direct impact on the unusability of the drive at lower speeds.
2. Tuning the mechanical drive - We will design a suitable type of flexible coupling, which will have torsional stiffness suitable for use in the required working range of the drive. Appropriate torsional stiffness means a torsional stiffness that changes the natural frequency of the system so that a change in resonance is achieved.

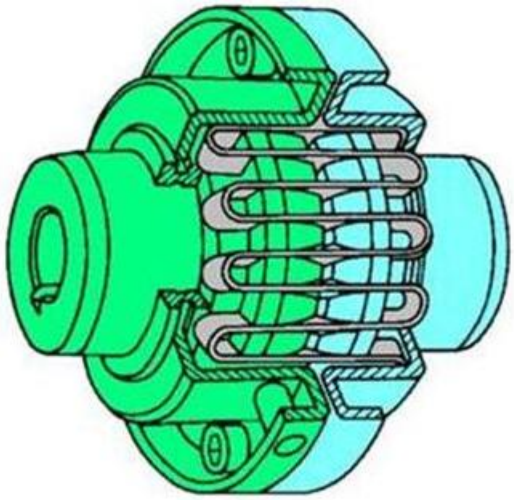
This case of designing a suitable torsional stiffness of a flexible coupling is presented by a red course of the natural frequency. This course has a common intersection with the excitation harmonic component at the point Rhn. This point, representing the resonance, is outside the range of the required operating speed. When shown in the amplitude-frequency graph, we see that the maximum amplitude generated in the resonance is outside the required operating range of the drive. As can be seen from the course, the vibrations that will be in the drive when a suitable flexible coupling is applied have a lower value than the required acceptance criteria. Such a design will significantly improve the working area of the drive and protect it from unwanted vibrations.



Design of flexible couplings

Despite the fact that the construction of flexible couplings has undergone extensive development, it is possible to determine the basic design features for all flexible couplings. Each flexible coupling consists of three basic parts. The first part is the drive part, or the primary part. This part is usually a disc or flange which allows the flexible coupling to be attached to the drive machine. The second part is the driven or even the secondary part. This part is mounted on the driven equipment or on the drive shaft of the driven machine. Usually this part is made as a rotating disc or flange. Between these two parts there is still a third part of the flexible coupling, which is a flexible member. The resilient members can be of different constructions, but can also be made of different materials. The most common materials used to make flexible members include: ferrous and non-ferrous metals, plastics, rubber, leather, various composite materials, and various gases and liquids. According to the type of material used to make the flexible member, we divide flexible couplings into three main types:

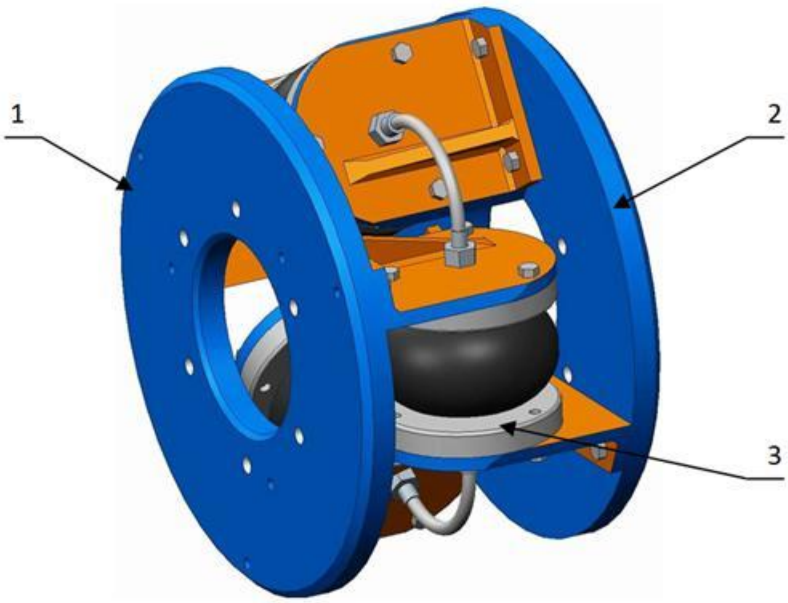
- flexible couplings with metal flexible member,



- flexible couplings with rubber or plastic flexible member,

- pneumatic flexible couplings

Current trends in the field of mechanical system dynamics also require innovation in the design of flexible couplings. Various special types of flexible couplings are created, which are used mainly in the automotive industry. Dual-mass flywheels are such an example of successful innovation of flexible couplings. We can also consider as a successful innovation of flexible couplings those types of flexible couplings in which it is possible to adapt their dynamic properties to the operational requirements of the mechanical system during their service life. Such flexible couplings include, in particular, pneumatic flexible couplings. Pneumatic flexible couplings, especially in combination with control systems, make it possible to effectively change the dynamic conditions in the mechanical system. Such targeted changes in dynamic properties during the operation of the device are called tuning of the mechanical system. Pneumatic flexible couplings, which will be equipped with a system for controlling their properties during the operation of the mechanical system, can be called pneumatic torsion oscillators fig.



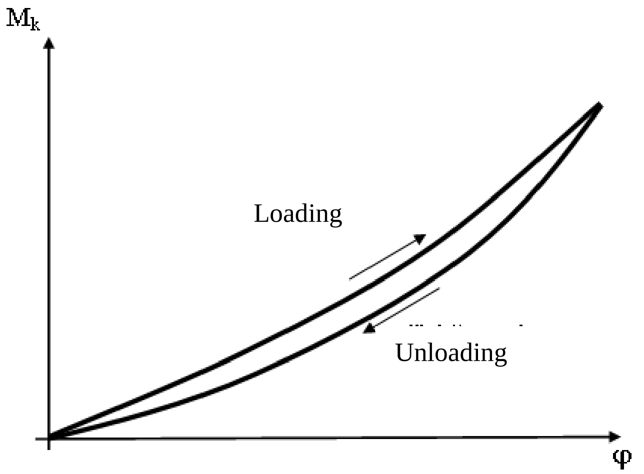
Pneumatic torsional tuner
1- primary part, 2- secondary part, 3- flexible members

The pneumatic torsion tuner consists of a primary (1) and a secondary part (2), between which a pneumatically flexible member (3) is placed. The pneumatically resilient member is filled with the pressure of air or another gaseous medium, and this pressure can be varied during operation by means of a control system as required. By changing the pressure of the gaseous medium in the compression space of pneumatic tuners, it is possible to change their torsional stiffness, and thus the natural frequency of the entire mechanical system.

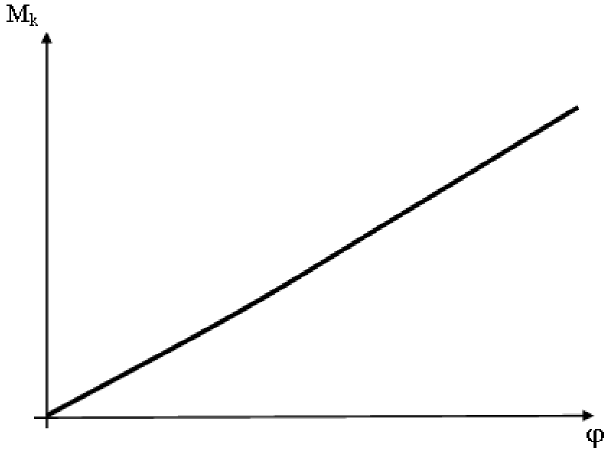
Properties of flexible couplings

Due to ignorance and ignorance of the main importance of flexible couplings even today, many manufacturers of flexible couplings still provide data that are sufficient only for the design of a flexible coupling using the operating factor according to STN 026408. For a more thorough dynamic calculation and for the correct design of the coupling into the mechanical

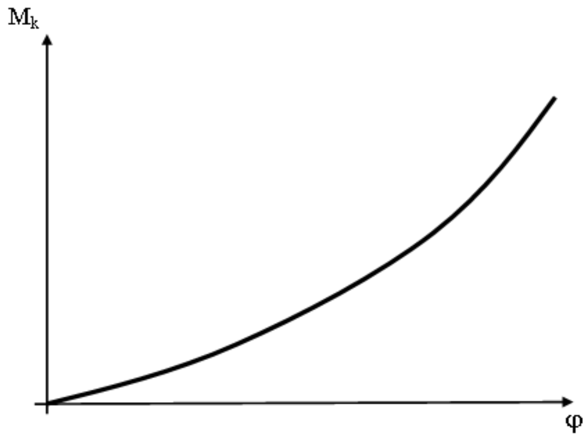
system, it is necessary to know the properties of flexible couplings in more detail. In general, it is appropriate to identify the following properties for each flexible coupling:
1.) Strength properties, i. the ability to transmit different types of torque.
2.) Operating characteristics, i. the ability to tune and tune the mechanical system and dampen torsional vibrations.
3.) Expansion properties, i. ability to compensate for misalignment, axial misalignments and angular misalignments.
4.) Ability to withstand the effects of heat
5.) In the case of pneumatic flexible couplings also the ability to react to different types of gases
The properties of flexible couplings are determined by experimental measuring methods, which are divided into static and dynamic methods. The coupling characteristic is defined as the dependence of the torque on the torsion angle Mk = f (φ). When measuring the characteristic, the coupling is loaded with gradually increasing and decreasing torque. However, the load torque must not exceed the value of M_{km}. Repeat the whole load cycle 4 times, the resulting loop is drawn from the last two loads. The loading procedure is important and can be summarized as follows:
¬ it is not possible to prescribe the size of the jump and the duration of this jump. We must strive for the narrowest possible loop, this is a prerequisite for more credible results,
¬ the load time does not need to be extended so that the result is not distorted by the coupling flow,
¬ for non-linear couplings it is necessary to divide the whole measurement as densely as possible,
¬ torque jumps and load time must be the same,
¬ for couplings with a hardening characteristic, it is recommended to take measurements in both directions.



The static measurement methods can be used to determine the static load characteristics of each flexible coupling. The flexible coupling must be loaded with a step load up to the value of the maximum torque during non-variable operation M_{st}. From the course of the measurement, we then make a load characteristic. Depending on the type of flexible coupling, it is possible for the load characteristic of the coupling to have a linear FIG. or a non-linear characteristic FIG



Linear characteristic of flexible coupling



Non-linear characteristic of flexible coupling

The static characteristics of a flexible coupling are expressed by the type equation:

$$Mk = a_1.\varphi + a_3.\varphi^3$$

We determine the coefficients a_1 and a_3 as follows:

$$a_1 = \frac{s_2.s_3 - s_1.s_4}{s_3^2 - n.s_4}; \qquad a_3 = \frac{s_1.s_3 - n.s_2}{s_3^2 - n.s_4};$$

if: $$s_1 = \sum_{i=1}^{n} \frac{T_i}{\varphi_i}; \qquad s_2 = \sum_{i=1}^{n} T_i.\varphi_i; \qquad s_3 = \sum_{i=1}^{n} \varphi_i^2; \qquad s_4 = \sum_{i=1}^{n} \varphi_i^4$$

The calculated values of the coefficients a_1 and a_3 are an expression of their mutual ratio. It is possible to classify a flexible coupling among couplings with a linear or non-linear characteristic.

Static torsional stiffness is a property of the clutch, which is defined as the ratio of the torque dM_k to the twist of the coupling dφ, which this torque induces if the measured clutch has no damping.

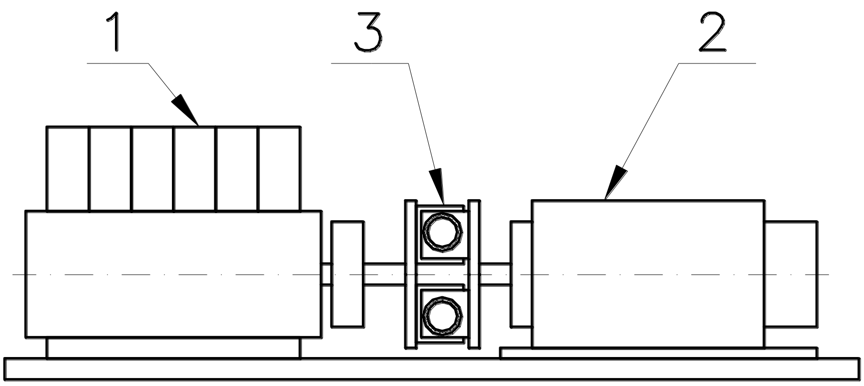
$$k = \frac{dM_k}{d\varphi}$$

The determination of static torsional stiffness defined in this way is suitable for flexible couplings with both linear and non-linear characteristics. However, static torsional stiffness is only stiffness that is found at rest. This torsional stiffness is measured only for a very slow change in torque frequency. At higher frequencies, this characteristic is different and is called dynamic torsional stiffness. The difference between static and dynamic torsional stiffness is typical mainly for high molecular weight materials. The applied clutch is in motion and for this reason it is appropriate to identify the properties that correspond to the actual state. Dynamic operating characteristics include:

- dynamic torsional stiffness,
- damping
- dynamic clutch characteristics.

Practical design of flexible coupling

Design a flexible coupling for a torsionally oscillating mechanical system according to fig. The drive part consists of a six-cylinder four-stroke diesel engine type 6S160PN, the driven part consists of a DC generator type DN 1144-4. The mechanical system will operate at speed $n = 600 min^{-1}$ at 100% fuel supply of the diesel engine, ie 100% power P = 98kW with idle speed $n_v = 380 min^{-1}$.



1 – combustion engine, 2 – DC generator, 3 – flexible coupling

A. A common but unsuitable design of a flexible coupling

Calculation of load torque:

$$P = M_k.\omega$$

$$M_k = 1560 Nm$$

Choice of flexible coupling based on maximum load torque:

$$M_{km} = K.M_k$$

$$M_{km} = 4368Nm$$

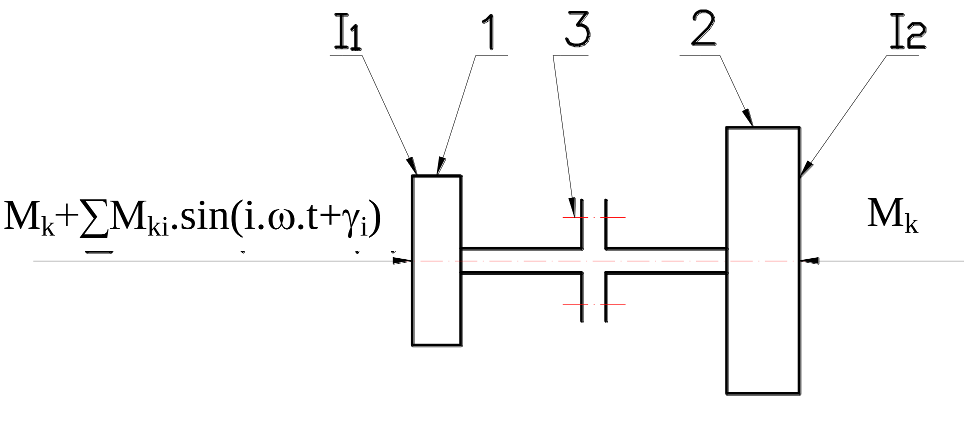
if: **K = 2,8 – operating factor**

From the Vulkan catalog, we choose a flexible coupling type EZR1001 for the next higher value M_{kN} = 5000Nm.

Catalog value:

M_{kN} = 5000Nm; - maximal torque,
M_{km} = 15000Nm; - maximum short - term torque,
M_{kw} = 2000Nm; - dynamic component under fluctuating symmetrical load,
k_{d100} = 83kNm.rad^{-1}; - dynamic torsional stiffness at maximum load,
ψ = 1.13; - relative damping,
I_{sp} = 0,08kg.m^2 - mass moment of inertia of the flexible coupling.

B. Determination of flexible coupling load based on dynamic calculation.



1 - engine, 2 - generator, 3 – flexible coupling

In FIG. is a dynamic model of the system

Determination of the load for the designed flexible coupling. The values for the system are: flexible coupling EZR1001 with catalog parameters

I_1=3,69kgm^2
I_2=40,37kgm^2

We describe the system by equations of motion according to the theory of block 11.

Natural frequency of mechanical system: **Ω_0=156,68rad.s^{-1}**

Tuning factor for the third harmonic component - **i = 3** calculation:

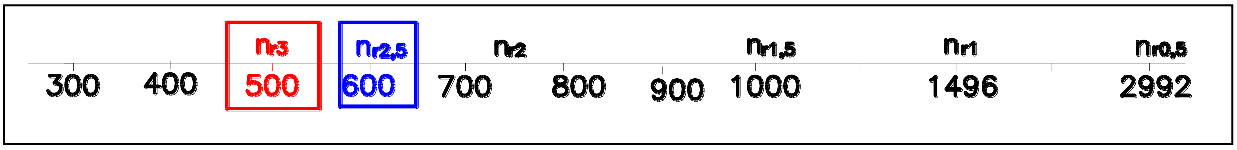
- **For idle speed:** $\mathbf{n_v = 380min^{-1}}$ **je** $\mathbf{\eta_v = 0,762}$
- **for operating speed:** $\mathbf{n_p = 600min^{-1}}$ **je** $\mathbf{\eta_p = 1,2}$

Determination of resonances from the main harmonic component and lower harmonic components of the load torque

i = 3 ; 2,5 ; 2 ; 1,5 ; 1 ; 0,5.

$$n_{r3} = 499min^{-1} \approx 500min^{-1}$$

$$n_{r2,5} = 600min^{-1};\ n_{r2} = 748min^{-1};\ n_{r1,5} = 998min^{-1};n_{r1} = 1496min^{-1};\ n_{r0,5}=2992min^{-1}$$



Calculation of the dynamic component M_{kdr} and the total load torque M_{kcr} of the mechanical system from i = 3 for:

- **resonance** - $\mathbf{n_r = 500min^{-1}}$,
- **operating speed** - $\mathbf{n_p = 600min^{-1}}$,
- **idle speed** - $\mathbf{n_v = 380min^{-1}}$

For calculation are apply forms to the theory of block 11.

Dynamic coefficient:	$\xi_{r3} = 5,65$
Dynamic torque:	$M_{kdr} = 10700Nm$
Total torque:	$M_{kcr} = 12260Nm$

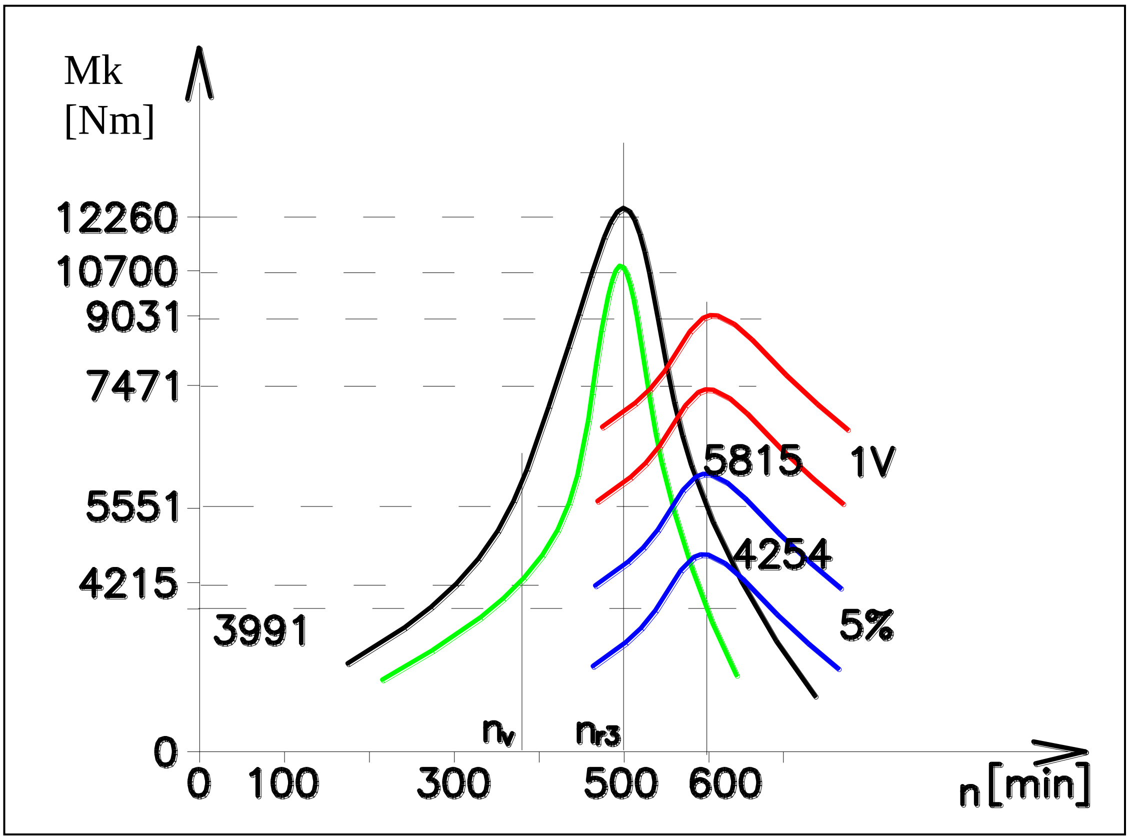
Dynamic coefficient:	$\xi_{p3} = 2,11$
Dynamic torque:	$M_{kdp} = 3991Nm$
Total torque:	$M_{kcp} = 5551Nm$

Dynamic coefficient:	$\xi_{v3} = 2,23$
Dynamic torque:	$M_{kdv} = 4215Nm$
Total torque:	$M_{kcv} = 5775Nm$

Checking for anomaly operating range

Uneven running ±5% .... $\mathbf{M_{kdp} = 4254Nm}$, $\mathbf{M_{kcp} = 5815Nm}$ (blue colour)

One cylinder deactivation...................... $\mathbf{M_{kdp} = 7471Nm}$, $\mathbf{M_{kcp} = 9031Nm}$ (red colour)



Determination of the load for a flexible coupling designed on the basis of the load torque M_k

Based on the calculated load torque M_k = 1560Nm, we propose from the Vulkan catalog a flexible coupling type EZR 0702/1 with the following catalog values:

M_{kN} = 2200Nm; - maximal torque
M_{km} = 5000Nm; - maximum short - term torque
M_{kw} = 640Nm; - dynamic component under fluctuating symmetrical load
k_{d100} = 24,6kNm.rad^{-1}; - dynamic torsional stiffness at maximum load
ψ = 1,13; - relative damping
I_{sp} = 0,041kg.m^2 - mass moment of inertia of the flexible coupling

Then the natural frequency of the system will be: **Ω_0 = 86rad.s^{-1}** and the resonance from the main harmonic component i = 3 will occur at **n_{r3} = 274min^{-1}**. In case of resonance:

Tuning coefficient: **η_r = 1,00**
Dynamic coefficient: **ξ_{r3} = 5,65**
Dynamic torque: **M_{kdr} = 10700Nm**
Total torque: **M_{kcr} = 12260Nm**

In the case of trouble-free operation of the piston machine, the secondary harmonic components have no effect on the magnitude of the dynamic component of the load torque.

The magnitude of the dynamic load is affected only by the third harmonic component of the torque and its magnitude at working speed will be:

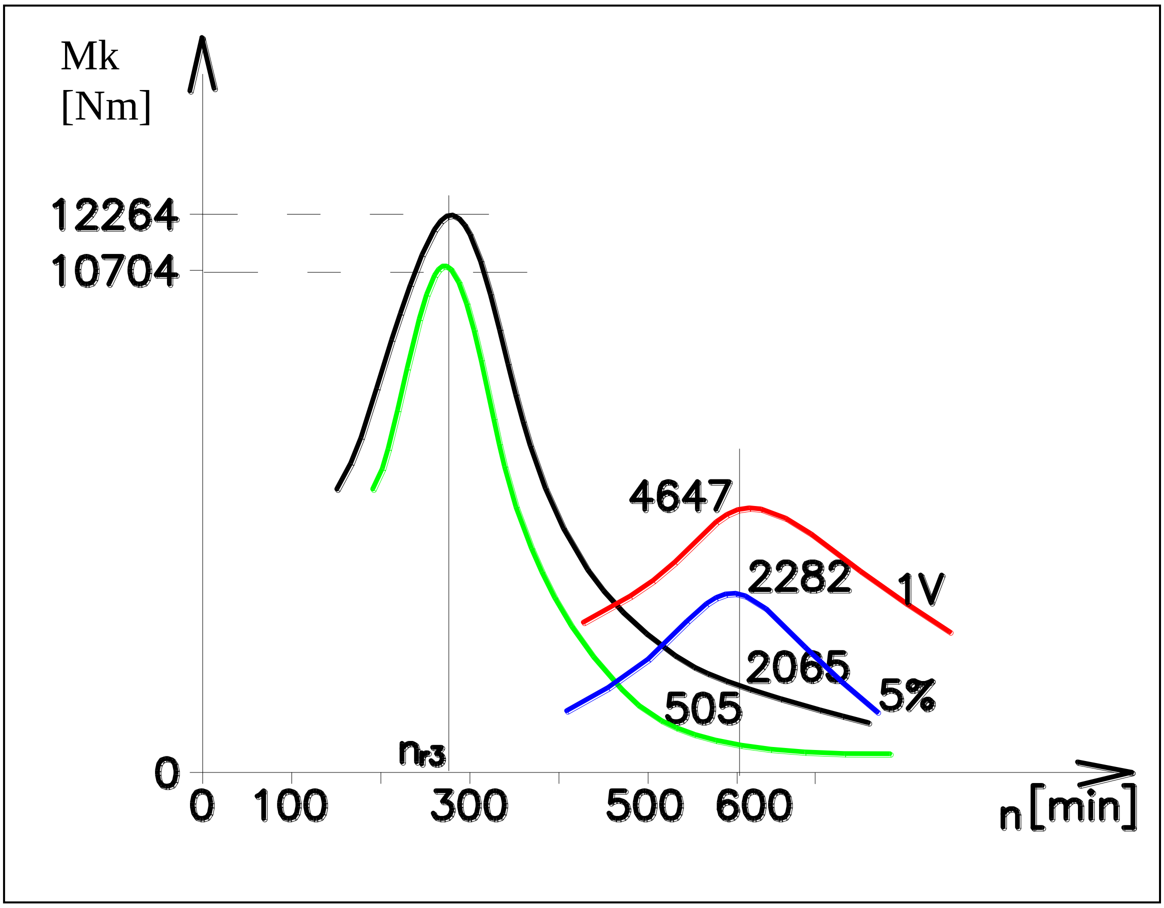
Tuning coefficient:	$\eta_v =$ **1,39**
Dynamic coefficient:	$\xi_{p3} =$ **0,267**
Dynamic torque:	$M_{kdp} =$ **505,5Nm**
Total torque:	$M_{kcp} =$ **2065,5Nm**

$$M_{kcp} < M_{kN} \quad (2200Nm)$$

<u>It is sufficient to dimension each rotating part of the mechanical system for the load torque M_k=2065,5Nm!</u>

Uneven running ±5%.................................... M_{kdp} **= 722Nm,** M_{kcp} **= 2282Nm**
One cylinder deactivation.. M_{kdp} **= 3087Nm,** M_{kcp} **= 4647Nm**

The aim of the design of each flexible coupling in the dynamic calculation is that the resonance from the main harmonic component of the load torque lies below the idle speed, ie that the mechanical system operates in the supra-resonant region. This condition is provided by the tuning coefficient (factor).



Thus, the aim is to keep the value of the tuning factor to a minimum η = 1,3.

The stated flexible coupling of the type meets this requirement EZR 0702/1.