MINIMIZATION OF DYNAMIC FORCES IN GEAR MESHING BY SELECTION OF THE FLEXIBLE COUPLINGS PARAMETERS

Grzegorz Wojnar

The Silesian University of Technology, Faculty of Transport
Krasińskiego Street 8, 40-019 Katowice, Poland
tel.: +48 32 6034193, fax: +48 32 6034118
e-mail: grzegorz.wojnar@polsl.pl

Abstract

Torsional vibrations occur during drive systems operation. They may be especially hazardous if the frequency of input functions is close to resonance frequency or if system resonance frequency is equal to subharmonic of input function frequency. What's more, there is always a possibility of more than one resonance zone since the system is usually non-linear and multi-body.

In design process of power transmission systems very important is correctly selection of the couplings parameters (especially flexibility). Couplings stiffness and damping have principle influence on shaft torsional vibrations. In case of power transmission with tooth gear the vibrations which are connected with dynamic forces in meshing. The paper presents influence of flexible couplings stiffness characteristics on dynamic forces in meshing. The investigations based on the measured couplings stiffness as well as damping and computer simulation of power transmission system with tooth gear operates at varying load.

On the basis of conducted tests and analysis it may be stated that toothed wheels rotational speed increase does not always result in meshing dynamic forces increase. Dynamic model of drive system with toothed gear is therefore a useful tool for analysing couplings impact on meshing dynamic forces. This is illustrated by examples presented in current paper.

Keywords: *couplings stiffness, couplings damping, power transmission systems, computer simulation, dynamic forces in meshing*

1. Introduction

Clutch and couplings are among the elements universally used in drive construction. Their principal task is to join (couple) drive shafts, but they also help to ease changes in rotating torque, to compensate tolerances in relative shaft neck positioning, and in case of clutches they are also used to switch the drives on and off.

Torsional vibrations occur during drive systems operation. They may be especially hazardous if the frequency of input functions is close to resonance frequency or if system resonance frequency is equal to subharmonic of input function frequency. What's more, there is always a possibility of more than one resonance zone since the system is usually non-linear and contains different masses. That is why it is so important to select proper parameters of the drive, e.g. couplings adapted to minimisation of possible torsional vibrations. One of possible solutions used in „fine tuning" of drive systems is application of elastic pneumatic couplings [4-6]. Another answer to the problem is application of appropriate dampers or eliminators of torsional vibrations [3, 7]. If the coupling parameters are not selected properly, then torsional vibrations generated in the drive system influence the work of all system elements and hence they affect reliability and durability of whole kinematic chain. In case of systems using toothed gears these vibrations may cause significant increase of meshing forces. During toothed gear design process the influence of both external and internal loads acting upon the wheel teeth is taken into account. The internal loads present during toothed gear operation are affected by: gear wheels rotations speed, transverse and pitch contact ratios, dimensional tolerances of toothed wheels, meshing vibrations damping (which depend on, inter alia, lubrication method), contour modification, inter-tooth space (clearance) and other factors, which are presented in detail in references [2, 8, 9, 13, 14, 16, 18]. The influence of external loads acting upon gear under construction is usually considered by using a suitable coefficient determined on the basis of general appropriation of motor and load operation. The impact of non-linear stiffness and coupling damping on the forces acting in the meshing, in case of specific input functions, is so far a poorly recognised issue. That is why this particular problem is undertaken and discussed in current paper.

2. Test object

Drive system consisting of motor, input coupling, toothed gear, output coupling and load machine has been subjected to simulation tests. Geometric parameters of the toothed wheels are:

- number of pinion teeth $z_1 = 16$,
- number of gear teeth $z_2 = 24$,
- gear ratio 1.5,
- helix angle $\beta = 0$ deg,
- face width $b = 17$ mm,
- module pitch $m_n = 4.5$ mm,
- coefficient of pinion addendum modification $x_1 = 0.864$,
- coefficient of gear addendum modification $x_2 = -0.5$,
- distance between the centres of the two gears $a_w = 91.5$ mm.

The tests were conducted at different wheel rotational speeds. During tests the gear operated as reduction gear and was loaded with sinusoidally-varying torque $M_h = 71{,}8$ Nm ±25%. One of the measures of gear load is the rated load intensity indicator [13, 14] calculated from the following relationship:

$$Q = \frac{P_o}{b \cdot d_{w1}} [\mathrm{MPa}], \tag{1}$$

where:

P_o - circumferential force,
b - face width,
d_{w1} - pinion's pitch diameter.

In the analysed case $Q = 1{,}05$ MPa ±25%.

On the basis of method shown in [15], tyre couplings 060 ASO-165 [19] have been selected for the drive system. The rated torque of these couplings $M_{zn} = 127$ Nm.

The drive motor moment of inertia was equal to 0.08 kgm^2, while load machine moment of inertia was twice as high.

Dimensional tolerances of the toothed gear were neglected in the tests.

3. Dynamic model

References presenting coupling models [1, 15] describe coupling operation in great detail, for instance they take into account coaxiality errors [1]. In order to determine coupling characteristics impact on meshing forces, it was decided to apply dynamic model of the drive system with toothed gear [8]. The meshing phenomena are described in accordance with L. Müller model [2, 13, 14]. The mathematic description of the extended model, which takes into consideration non-linear couplings characteristics (Fig. 1) has been presented in reference [18].

4. Couplings stiffness and damping

Couplings 060 ASO-165 used in the model are characterised by high torsional flexibility, good vibration damping, they should nicely ease (smooth) abrupt changes in rotational torque and they are also able to compensate significant tolerances in shaft necks positioning.

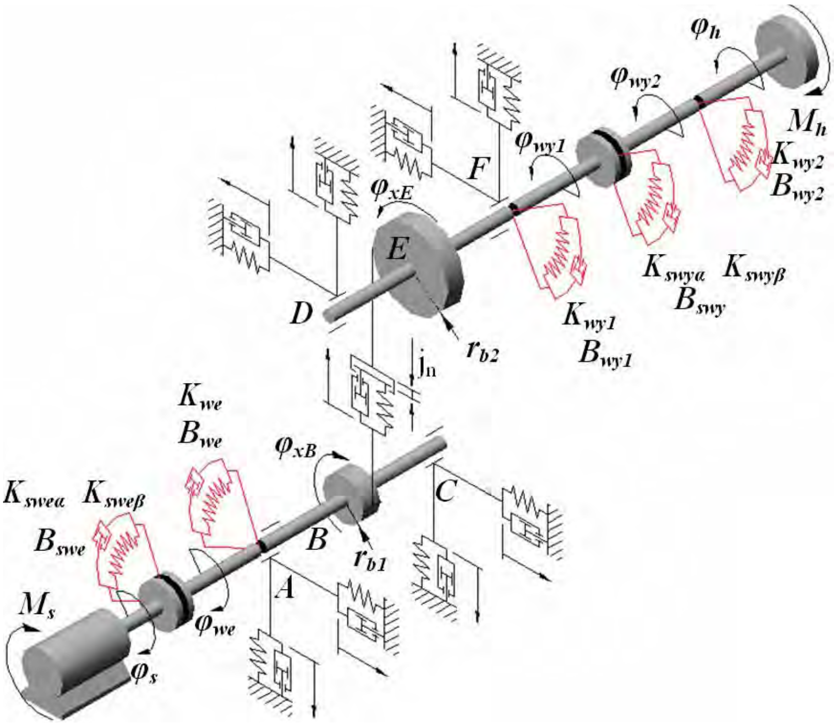

Fig. 1. Dynamic model of power transmission system with gearbox

Simulation tests take into account non-linear stiffness characteristics of the discussed coupling. This characteristic has been obtained as a result of experimental tests presented in reference [15] (Fig. 2). In order to determine coupling stiffness impact on meshing forces, another coupling (characterised by higher stiffness) has also been tested (Fig. 2). In case of this particular coupling, achieving torsional angle identical as in case of 060 ASO-165 coupling, requires application of rotational torque 6 times as great as for 060 ASO-165 coupling.

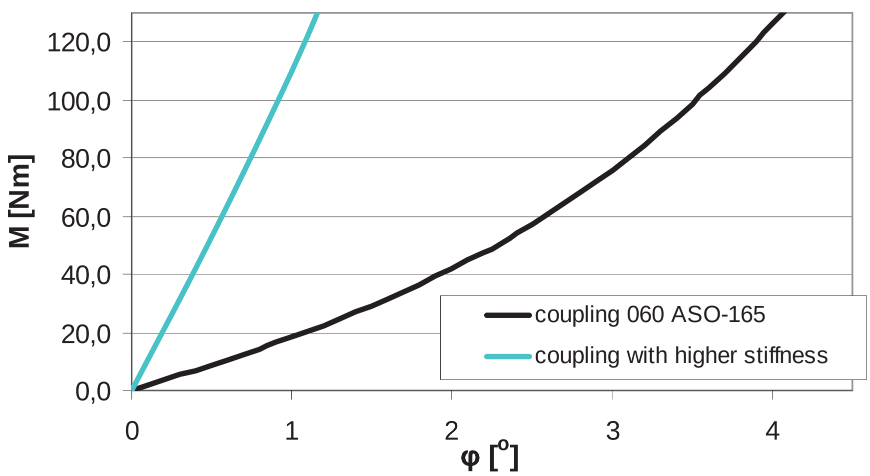

Fig. 2. Stiffness characteristics of two couplings with different stiffness's (060 ASO-165 and coupling with higher stiffness)

Shaft stiffness is high in relation to coupling stiffness; therefore it has been assumed to be infinitely great for sake of simulation tests.

The damping coefficient ψ (non-dimensional), which is defined as the ratio of heat-dissipated energy in each duty cycle A to elastic strain energy A_s in accordance with equations presented in reference [15], has been re-calculated to obtain dimensional damping coefficient used in the present model (Tab. 1).

Tab. 1. Relative damping coefficient of 060 ASO-165 tyre coupling [15]

Torque amplitude	Hysteresis loop area	Elastic strain area	Relative damping coefficient
M_A [Nm]	A_r [mm^2]	A_s [mm^2]	ψ
10% M_{zn} = 12.70	48.14	39.82	1.24
25% M_{zn} = 31.75	241.68	187.35	1.29
50% M_{zn} = 63.50	953.76	676.43	1.41
75% M_{zn} = 92.25	2379.32	1652.31	1.44
100% M_{zn} = 127	4582.63	2794.29	1.64

5. Meshing dynamic forces

K coefficient has been adopted as a measure of dynamic forces. It is defined as the ratio of maximum inter-teeth force to static force; it corresponds to the product of usage coefficient (K_A) and dynamic coefficient K_υ, these are recommended in ISO standards computational procedures.

Figure 3 shows dynamic forces coefficient in the meshing, in case of 060 ASO-165 coupling and pinion operating at low rotational speed oscillating around 500 rpm. Oscillation were caused by changes in the load torque M_h = 71.8 Nm ±25%. Time course of meshing dynamic forces coefficient displays variations due to sinusoidally varying gear load torque (torque varying frequency was equal to 2 Hz).

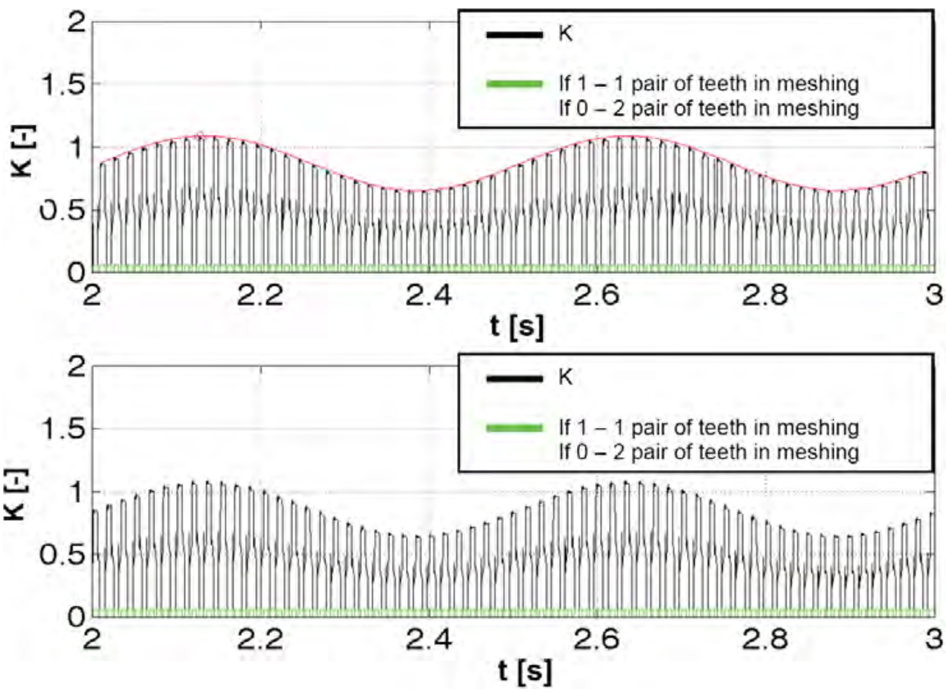

Fig. 3. Coefficient of dynamic forces in meshing – pinion rotational speed ~500 rpm: for even number of teeth (a), for odd number of teeth (b)

Along the length of line of action we can observe the impact of increase in pinion rotational speed and torque varying frequency on the dynamic changes in meshing forces coefficient (Fig. 4)

On the basis of results shown in Fig. 5a we can say that when 060 ASO-165 couplings are used, the increase in dynamic meshing forces does not directly correspond to increase in rotational speed, since it is also dependent on load torque frequency. Fig. 5b in turn demonstrates that if the gear operates at pinion's rotational speed oscillating around 3000 rpm and load torque frequency equal to 6 Hz, then decreasing this frequency down to 2 Hz or increasing it up to 10 Hz results in 10%

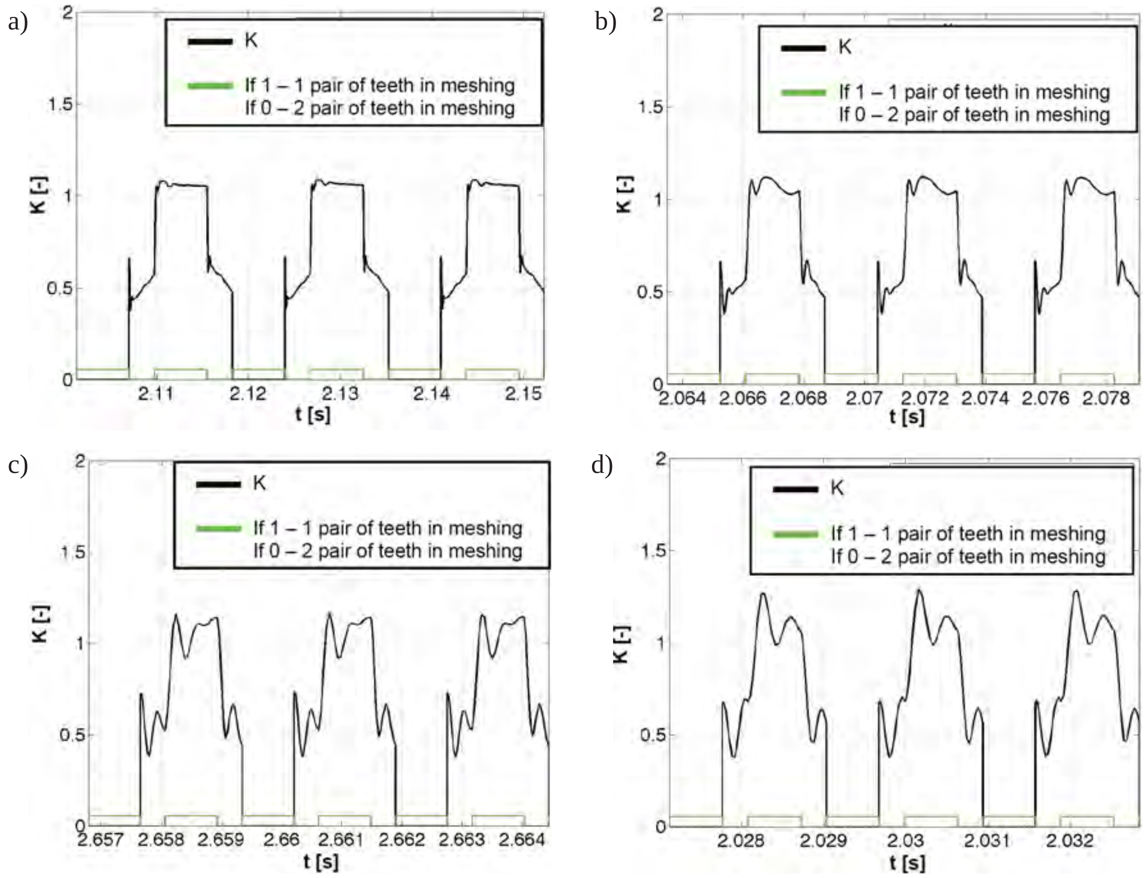

Fig. 4. Coefficient of dynamic force s in meshing for even number of teeth, pi nion's rotational speed and frequency for sinusoidally varying load torque is equal to: (a) ~500 rpm, 2 Hz, (b) ~1500 rpm, 8Hz, (c) ~3000 rpm, 8Hz, (b) ~4000 rpm, 10Hz

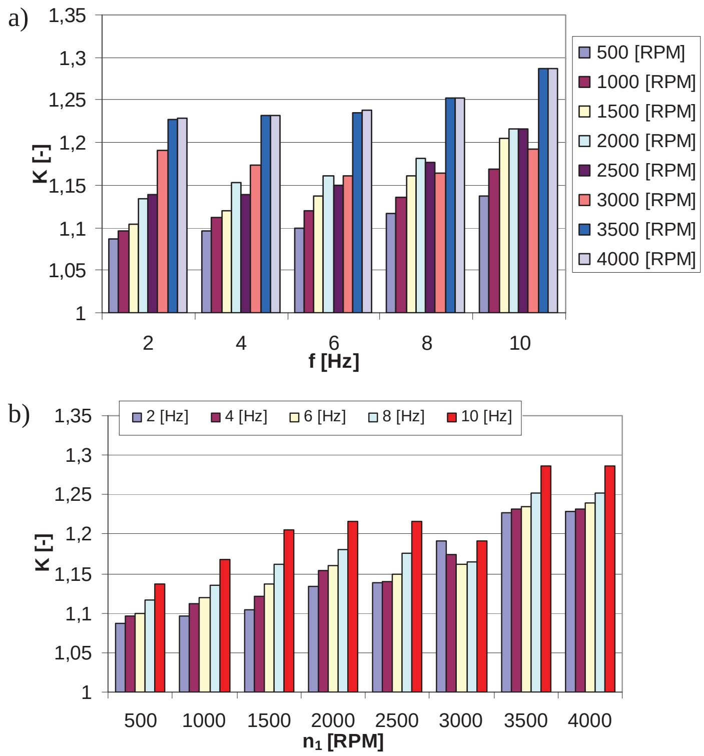

Fig. 5. Coefficient of dynamic forces in meshing K as function of freque ncy of load torque variation (a) and pinion rotational speed (b) - coupling 060 ASO-165

increase of meshing dynamic forces. Using a coupling characterised by higher stiffness (see Fig. 2) with damping identical as in case of 060 ASO-165 coupling, helps to decrease dynamic meshing forces down to 6% in most cases, with load torque frequency equal to 10Hz (Fig. 6). For rotational speeds *c.* 1000 and *c.* 2000 rpm the meshing dynamic forces coefficient increases a little, i.e. by 1%.

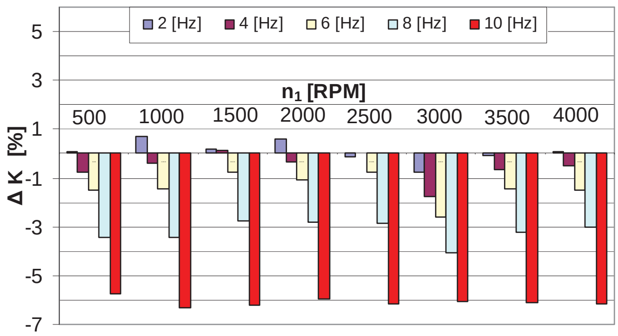

Fig. 6. Changes of meshing dynamic forces coefficient caus ed by using both input and output couplings with higher stiffness (Fig. 2) than that of 060 ASO-165 coupling

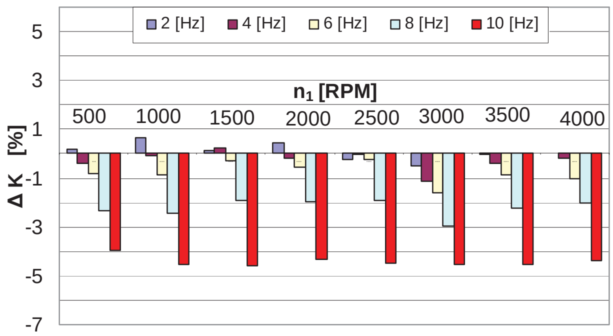

Fig. 7. Changes of meshing dynamic forces coefficient due to using 060 ASO-165 input coupling together with output coupling characterised by higher stiffness (Fig. 2) relative to 060 ASO-165 coupling stiffness

If stiffness of a single coupling is increased, the decrease of meshing forces is less than in case of using two identical couplings characterised by higher stiffness (Fig. 6 and 7). If one coupling with higher stiffness was used, the reduction in gear inter-teeth forces was greater when this coupling was applied as an output device (Fig. 8).

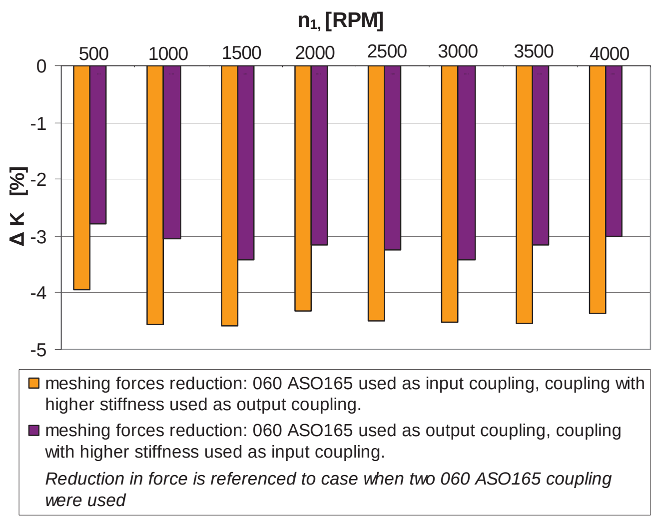

Fig. 8. Comparison of changes in meshing dynami c forces coefficient due to using a single coupling with higher stiffness (Fig. 3), load torque variation frequency was equal to 10 Hz

6. Conclusions

On the basis of conducted tests and analysis it may be stated that:

- toothed wheels rotational speed increase does not always result in meshing dynamic forces increase,
- increase in load torque variation frequency does not always lead to increase in meshing dynamic forces,
- in most cases, using a coupling with higher stiffness and damping identical to 060 ASO-165 coupling did lead to meshing dynamic forces decrease; this decrease was as high as 6% at load torque variation frequency equal to 10 Hz,
- dynamic model of drive system with toothed gear is a useful tool for analysing couplings impact on meshing dynamic forces, this is fully illustrated by examples presented in current paper.

References

[1] Batko, W., Dąbrowski, Z., Pakowski, R. et al., *Nowoczesne metody badania procesów wibro-akustycznych – cz. II*, Wydawnictwo i Zakład Poligrafii ITE w Radomiu, Katowice-Radom 2006.

[2] Dąbrowski, Z., Radkowski, S., Wilk, A., *Dynamika przekładni zębatych – Badania i symulacja w projektowaniu eksploatacyjnie zorientowanym*, ITE, Radom 2000.

[3] Homik, W., Projektowanie *wiskotycznych tłumików drgań skrętnych*, Przegląd Mechaniczny, No. 10, pp. 27, Warsaw 2007 .

[4] Homišyn, J., Dostrajanie *układów mechanicznych drgaj ących skrętnie za pomoc ą sprzęgieł pneumatycznych*, Wyd. ATH, Bielsko-Biała 2008.

[5] Homišyn, J., Grega, R., Kopas, M., *New kinds of pneumatic flexible shaft couplings with low torsional rigidity*. Transactions of the Universities of Košice Vol. 1, pp. 17-20, Košice 2007.

[6] Homišyn, J., *Elaboration of issue of torsional oscilations of mechanical systems via new types of flexible shaft clutches – pneumatic clutches*, Acta of Mechanica Slovaca, Vol. 1, ročnik 8, pp. 57-68, 2004.

[7] Lacko, P., Seman, J., Lacko, V., *Analýza dynamického modelu torzne pružného prenosového zariadenia*, Inženýrská mechanika roč. 4, , č.3, pp. 183-190, 1997.

[8] Łazarz, B., *Zidentyfikowany model dynamiczny przek ładni zębatej w układzie napędowym jako podstawa projektowania*, ITE, Radom 2001.

[9] Łazarz, B., Wojnar, G., *Model dynamiczny uk ładu napędowego z przekładnią zębatą*, XVII Ogólnopolska Konferencja Przekładnie Zębate, pp. 101-108, Węgierska Górka 2000.

[10] Łazarz, B., Wojnar, G., *Identyfikacja strat mocy w przek ładni zębatej walcowej*, Zeszyty Naukowe Politechniki Śląskiej, s. Transport, Z. 43, pp. 143-152, Gliwice 2001.

[11] Markusik, S., *Sprzęgła mechaniczne*, WNT, Warszawa 1979.

[12] Markusik, S. Opasiak, T., *Badanie charakterystyk statycznych i dynamicznych nierozłącznych sprzęgieł podatnych oponowych*, Zeszyty Naukowe Pol. Śl., s. Transport, Z. 56, pp. 215-222, Gliwice 2004.

[13] Müller, L., *Przekładnie zębate – badania*, WNT, Warszawa 1984.

[14] Müller, L., *Przekładnie zębate – dynamika*, WNT, Warszawa 1986.

[15] Opasiak, T., *Wpływ zmian momentu obrotowego na charakterystyk ę statyczną i dynamiczną sprzęgieł podatnych oponowych*, Rozprawa doktorska, Politechnika Śląska, Gliwice 2005.

[16] Skoć, A., Grzesica, P., *Wpływ obciążenia zewnętrznego na dynamik ę przekładni zębatych w aspekcie badań modelowych*, Materiały XXII Sympozjonu Podstaw Konstrukcji Maszyn, Wydawnictwo Fundacji Rozwoju Akademii Morskiej, Gdynia-Jurata 2005.

[17] Wojnar, G., Łazarz, B., *Wpływ uszkodzeń elementów przek ładni zębatej na si ły dynamiczne w zazębieniu*, Materiały XXII Sympozjonu Podstaw Konstrukcji Maszyn, Wydawnictwo Fundacji Rozwoju Akademii Morskiej, pp. 407-414, Gdynia-Jurata 2005.

[18] Wojnar, G., Łazarz, B., Madej, H., *Diagnozowanie przekładni zębatych w przypadku pracy ze zmiennymi w czasie pr ędkościami obrotowymi i obciążeniami*, Materiały XXXIII Ogólnopolskiego Sympozjum Diagnostyka Maszyn, Węgierska Górka 2006.
[19] *Katalog firmy Fena sp. z o.o, Polska*, Katowice1999.