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A Virtual Tool for Computer Aided Analysis of Spur Gears with Asymmetric Teeth

Fatih Karpas¹, Stephen Ekworo-Osire² and Esin Karpas¹

¹*Department of Mechanical Engineering, Uludag University, Bursa,*

²*Department of Mechanical Engineering, Texas Tech University, Lubbock,*

¹*Turkey*

²*USA*

1. Introduction

1.1 Background

There is an industrial demand in the increased performance of mechanical power transmission devices. This need in high performance is driven by high load capacity, high endurance, low cost, long life, and high speed. For gears, this has led to development of new designs, such as gears with asymmetric teeth. The geometry of these teeth is such that the drive side profile is not symmetric to the coast side profile. This type of geometry is beneficial for special applications where the loading of the gear is uni-directional. In such an instance, the loading on the gear tooth is not symmetric, thus calling for asymmetric teeth. Since one of the situations that demand high performance is the high rotational speeds, there is a need to understand the dynamic behavior of the gears with asymmetric teeth at such speeds. Such knowledge would shed light on detrimental characteristics like dynamic loads and vibrations. An efficient way in performing studies on the dynamic behavior of gears is using computer aided analysis on numerical models.

A number of studies on the design and stress analysis of asymmetric gears are available in literature. A large number of studies have been performed over the last two decades to assess whether asymmetric gears are an alternative to conventional gears in applications requiring high performance. In these studies, some standards (i.e., ISO 6336, DIN 3990), analytical methods (i.e., the Direct Gear Design method, the tooth contact analysis), and numerical methods (e.g., Finite element method) have been used to compare the performance of conventional and asymmetric gears under the same conditions (Cavdar et al., 2005; Kapelevich, 2000, Karpas, 2005; Karpas et al., 2005; Karpas & Ekworo-Osire, 2008; Karpas et al., 2008; Karpas & Ekworo-Osire, 2010). In the last ten years, the researches conducted in the area of gears with asymmetric teeth point to the potential impact of asymmetric gears on improving the reliability and performance requirements of gearboxes. The benefits of asymmetric gears which have been offered by researchers are: higher load capacity, reduced bending and contact stress, lower weight, lower dynamic loads, reduced wear depths on tooth flank, higher reliability, and higher efficiency. Each of the benefits can be obtained due to asymmetric teeth designed correctly by designers.

1.2 Dynamic analysis of involute spur gears with symmetric teeth

Gear dynamics has been a subject of intense interest to the gearing area during the last few decades. The dynamic response of a gear transmission system is becoming essential due to increased requirements for high speed, low vibration and heavy load in gear design. However, the numerous design parameters, manufacturing and assembly errors, tooth modifications, etc. make difficult to understand gear dynamic response. The dynamic load reducing in a gear pair may decrease noise, increase efficiency, improve pitting fatigue life, and prevent gear tooth failures. Thus far, many researchers have conducted theoretical and experimental studies on gear dynamics. Most of literature on mathematical models used to predict the gear dynamics have been reviewed by (Ozguven & Houser, 1988; Parey & Tandon, 2003). In these reviews, the theoretical studies use a numerical method which included the excitation terms due to errors and periodic variation of the mesh stiffness. This method was used by many researchers to calculate the dynamic contact load or the torsional response, depending on different gear parameters, i.e., tooth errors, addendum modification, mesh stiffness, lubrication, damping factor, gear contact factor, and friction coefficient.

In dynamic analysis of gears, the dynamic factor and static transmission are the two most important definitions. The dynamic factor is defined as the ratio of the maximum dynamic load to the maximum static load on the gear tooth. Dynamic loads of gears with low contact ratio (between 1 and 2) are affected by several parameters, namely: time-varying mesh stiffness, tooth profile error, contact ratio, friction, and sliding. Static transmission errors, which are defined as the difference between the position of an actual gear tooth and that of an idealized gear tooth, and dynamic loads, affect the gear vibrations, acoustic emissions, tooth fatigue, and surface failure. The static transmission errors change in a periodic manner, due to the variation of gear mesh stiffness during contact. This is the source of vibratory excitation in gear dynamics. The static transmission error has basic periodicities related to the shaft rotational frequencies and the gear mesh frequency. The mesh frequency and its first harmonics are the predominant contributors to the generation of noise. The Fast Fourier Transform (FFT) can be used to perform the frequency analysis of static transmission error.

1.3 Motivation and objectives

Involute spur gears with asymmetric teeth provide flexibility to designers for different application areas due to non-standard design. If they are correctly designed, they can make important contributions to the improvement of designs in aerospace industry, automobile industry, and wind turbine industry. This often relates to improving the performance, increasing the load capacity, reduction of acoustic emission, and reduction of vibration. In the past, most of the analysis of gears with asymmetric teeth has been limited to cases under static loading.

Dynamic loads and vibration are a major concern for gears running at high speeds. Therefore, dynamic behavior should be analyzed to determine the feasibility of asymmetric gears in different applications. In order to utilize asymmetric gear designs more effectively, it is imperative to perform analyses of these gears under dynamic loading. This study offers designers preliminary results for understanding the response of asymmetric gears under dynamic loading. The effect of some design parameters, such as pressure angle or tooth height on dynamic loads, is shown. The asymmetric gears considered will have a larger pressure angle on the drive side compared to the coast side. In this study, to investigate the response of asymmetric gears under dynamic loading, the dynamic loads and static transmission errors were used. The first objective of this chapter is to use dynamic analysis

to compare conventional spur gears with symmetric teeth and spur gears with asymmetric teeth. The second objective is to develop a MATLAB-based virtual tool to analyze dynamic behavior of spur gears with asymmetric teeth. For this purpose a MATLAB based virtual tool called DYNAMIC is developed.

The first part of the study is focused on assymmetric gear modelling. The second part focuses on the virtual tool parameters. In the third and the last part, the simulation results are given for different asymmetric gear parameters.

2. Dynamic model for involute spur gears with asymmetric teeth

There is an essential need to find the equations of motion for a gear tooth pair during a mesh to determine the variation of dynamic load with the contact position. A single-degree-of-freedom model of the gear system consists of a gear and a pinion shown in Fig. 1. The equations of motion can be expressed as follows:

$$J_g \ddot{\theta}_g = r_{bg}(F_I + F_{II}) \pm \rho_{gI} \mu_I F_I \pm \rho_{gII} \mu_{II} F_{II} - r_{bg} F_D \quad (1)$$

$$J_p \ddot{\theta}_p = r_{bp} F_D - r_{bp}(F_I + F_{II}) \pm \rho_{pI} \mu_I F_I \pm \rho_{pII} \mu_{II} F_{II} \quad (2)$$

where J_p and J_g represent the polar mass moments of inertia of the pinion and gear, respectively. The dynamic contact loads are F_I and F_{II} , while μ_I and μ_{II} are the instantaneous coefficients of friction at the contact points. θ_p and θ_g represent the angular displacements of pinion and gear. The radii of the base circles of the engaged gear pair are r_{bp} and r_{bg} , while the radii of curvature at the mating points are $\rho_{pI,II}$ and $\rho_{gI,II}$.

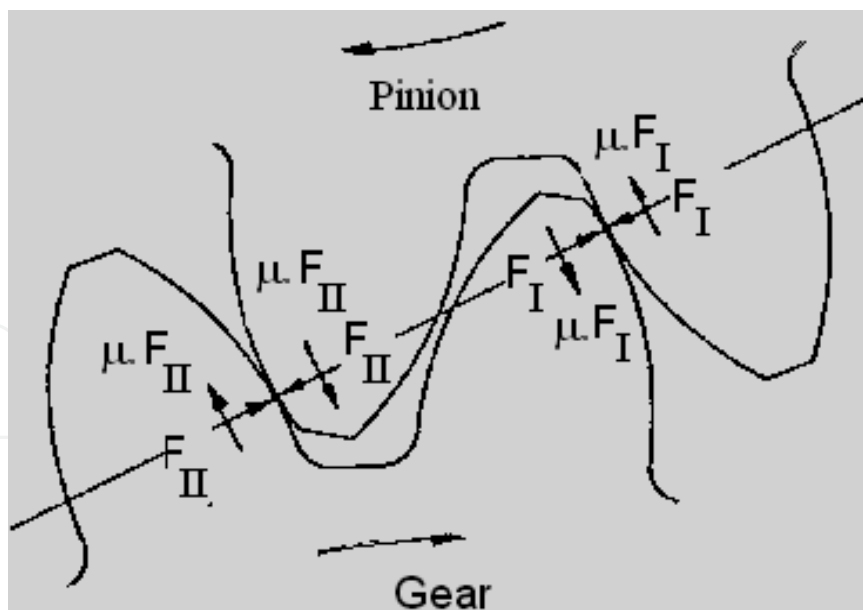


Fig. 1. The free body diagram of an engaging teeth pairs

The static tooth load is defined as:

$$F_D = \frac{T_p}{r_{bp}} = \frac{T_g}{r_{bg}} \quad (3)$$

The relative displacement, velocity, and acceleration can be written as follows:

$$x_r = y_p - y_g \quad (4)$$

$$\dot{x}_r = \dot{y}_p - \dot{y}_g \quad (5)$$

$$\ddot{x}_r = \ddot{y}_p - \ddot{y}_g \quad (6)$$

The effective gear masses are:

$$M_p = \frac{J_p}{r_{bp}^2} \quad (7)$$

$$M_g = \frac{J_g}{r_{bg}^2} \quad (8)$$

Including viscous damping, the equations of motion are reduced to:

$$\ddot{x}_r + 2\omega\xi\dot{x}_r + \omega^2x_r = \omega^2x_s \quad (9)$$

$$\omega^2 = \frac{K_I(S_{pI}M_g + S_{gI}M_p) + K_{II}(S_{pII}M_g + S_{gII}M_p)}{M_gM_p} \quad (10)$$

$$\omega^2x_s = \frac{(M_g + M_p)F_D + K_I\lambda_I(S_{pI}M_g + S_{gI}M_p) + K_{II}\lambda_{II}(S_{pII}M_g + S_{gII}M_p)}{M_gM_p} \quad (11)$$

The loaded static transmission errors can be obtained by dividing Eq. (11) by Eq. (10) to yield:

$$x_s = \frac{(M_g + M_p)F_D + K_I\lambda_I(S_{pI}M_g + S_{gI}M_p) + K_{II}\lambda_{II}(S_{pII}M_g + S_{gII}M_p)}{K_I(S_{pI}M_g + S_{gI}M_p) + K_{II}(S_{pII}M_g + S_{gII}M_p)} \quad (12)$$

The equivalent stiffness of meshing tooth pairs, in Eq. (10) through (12), can be written as:

$$K_I = \frac{k_{pI}k_{gI}}{k_{pI} + k_{gI}} \quad (13)$$

$$K_{II} = \frac{k_{pII}k_{gII}}{k_{pII} + k_{gII}} \quad (14)$$

The friction experienced by the pinion and the gear can be expressed as:

$$S_{pI} = 1 \pm \frac{\mu_I \rho_{pI}}{r_{bp}} \quad (15)$$

$$S_{gI} = 1 \pm \frac{\mu_I \rho_{gI}}{r_{bd}} \quad (16)$$

$$S_{pII} = 1 \pm \frac{\mu_{II} \rho_{pII}}{r_{bp}} \quad (17)$$

$$S_{gII} = 1 \pm \frac{\mu_{II} \rho_{gII}}{r_{bd}} \quad (18)$$

The signs in the above expressions are positive (+) for the approach and negative (-) for the recess.

The coefficient of friction is expressed by formula:

$$\mu_{I,II} = 18.1\nu^{-0.15} \left(\frac{v_{gI,II} + v_{pI,II}}{v_{gI,II} - v_{pI,II}} \right)^{-0.15} \left(|v_{gI,II} - v_{pI,II}| \right)^{-0.5} \left(\frac{\rho_{gI,II} \rho_{pI,II}}{\rho_{gI,II} + \rho_{pI,II}} \right)^{-0.5} \quad (19)$$

where ν is the viscosity of lubricant (cSt). And $v_{pI,II}$ and $v_{gI,II}$ are the surface velocities (mm/s), which can be formulated as follows:

$$v_{pI,II} = V \left(\frac{L_{pI,II} \cos \alpha_d}{r_{bp}} + \sin \alpha_d \right) \quad (20)$$

$$v_{gI,II} = V \left(-\frac{L_{gI,II} \cos \alpha_d}{r_{bg}} + \sin \alpha_d \right) \quad (23)$$

where $L_{pI,II}$ and $L_{gI,II}$ are the distances between the contact point and the pitch point along the line of action for pinion and gear, respectively, and V is the tangential velocity on the pitch circle.

The value of the damping ratio, ξ , in Eq. (9), is commonly recommended in literature as one between 0.1 and 0.2. In this study, a constant value of 0.17 proposed in literature for the damping ratio, ξ , was adapted in the solution of equations.

The dynamic contact loads, which include tooth profile error, can then be written as:

$$F_I = K_I(x_r - \lambda_I) \quad (21)$$

$$F_{II} = K_{II}(x_r - \lambda_{II}) \quad (22)$$

where λ_I and λ_{II} are the tooth profile errors. In this study, the effects of profile errors on the dynamic response of gears are not considered. Thus, the tooth profile errors are assumed to be zero. The developed computer program has a capability of using any approach for the determination of errors.

It should be noted that the above equations are valid only when there is contact between two gears. When separation occurs between two gears, because of the relative errors between the teeth of gears, the dynamic load will be zero and equation of motion will be given by:

$$T\ddot{x}_r = F_D \quad (23)$$

The meshing conditions are described as follows:

If $x_r > \lambda_I; x_r > \lambda_{II}$	$F_I, F_{II} > 0$	Double tooth contact
If $x_r \leq \lambda_I; x_r \leq \lambda_{II}$	$F_I = F_{II} = 0$	Tooth separation
If $\lambda_I < x_r \leq \lambda_{II}$	$F_I > 0$ and $F_{II} = 0$	Single tooth contact
If $\lambda_{II} < x_r \leq \lambda_I$	$F_I = 0$ and $F_{II} > 0$	Single tooth contact

3. Tooth stiffness

According to Equations (13) and (14), in order to calculate the equivalent stiffness of a meshing tooth pair, the tooth stiffness has to be known beforehand. In this study, a 2-D finite element model was developed to calculate the deflections of both the asymmetric and the symmetric gear teeth. By using this model, nodal deflections are calculated for pre-determined contact points. The load applied for each contact point is taken as a constant in order to determine tooth deflection under unit load. By putting the calculated nodal deflection values into Equations (24-27), the tooth stiffness are calculated and then the approximate curves for the single tooth stiffness along the contact line are obtained with respect to the radius of the gears. This process was repeated for each gear previously designed for different gear parameters.

$$k_{pI} = \frac{F}{\delta_{pI}} \tag{24}$$

$$k_{gI} = \frac{F}{\delta_{gI}} \tag{25}$$

$$k_{pII} = \frac{F}{\delta_{pII}} \tag{26}$$

$$k_{gII} = \frac{F}{\delta_{gII}} \tag{27}$$

where F is the load applied, and δ_{pI} , δ_{pII} , δ_{gI} , and δ_{gII} are the deflections of the teeth in the direction of this load.

4. Computational procedure

The reduced equation of motion is solved numerically using a method that employs a linear iterative procedure. This involves dividing the mesh period into many equal intervals. In this study, the flowchart of this computational procedure developed in MATLAB, used for calculating the dynamic responses of spur gears, is shown in Fig. 2. The time interval, between the initial contact point and the highest point of single contact, is considered as a mesh period. In the numerical solution, each mesh period is divided into 200 intervals for good accuracy. Within each of the sub-intervals thus obtained, various parameters of equations of motion are taken as constants, and an analytical solution is obtained. The

calculated values of the relative displacement and the relative velocity after one mesh period are compared with the initial values x_r and v_r . Unless the differences between them are smaller than a preset tolerance (0.000001), the iteration procedure is repeated by taking the previously calculated values of x_r and v_r at the end point of single pair of teeth contact as the new initial conditions. Then the dynamic loads are calculated by using the calculated relative displacement values. After the gear dynamic load has been calculated, the dynamic load factor can be determined by dividing the maximum dynamic load along the contact line to the static load.

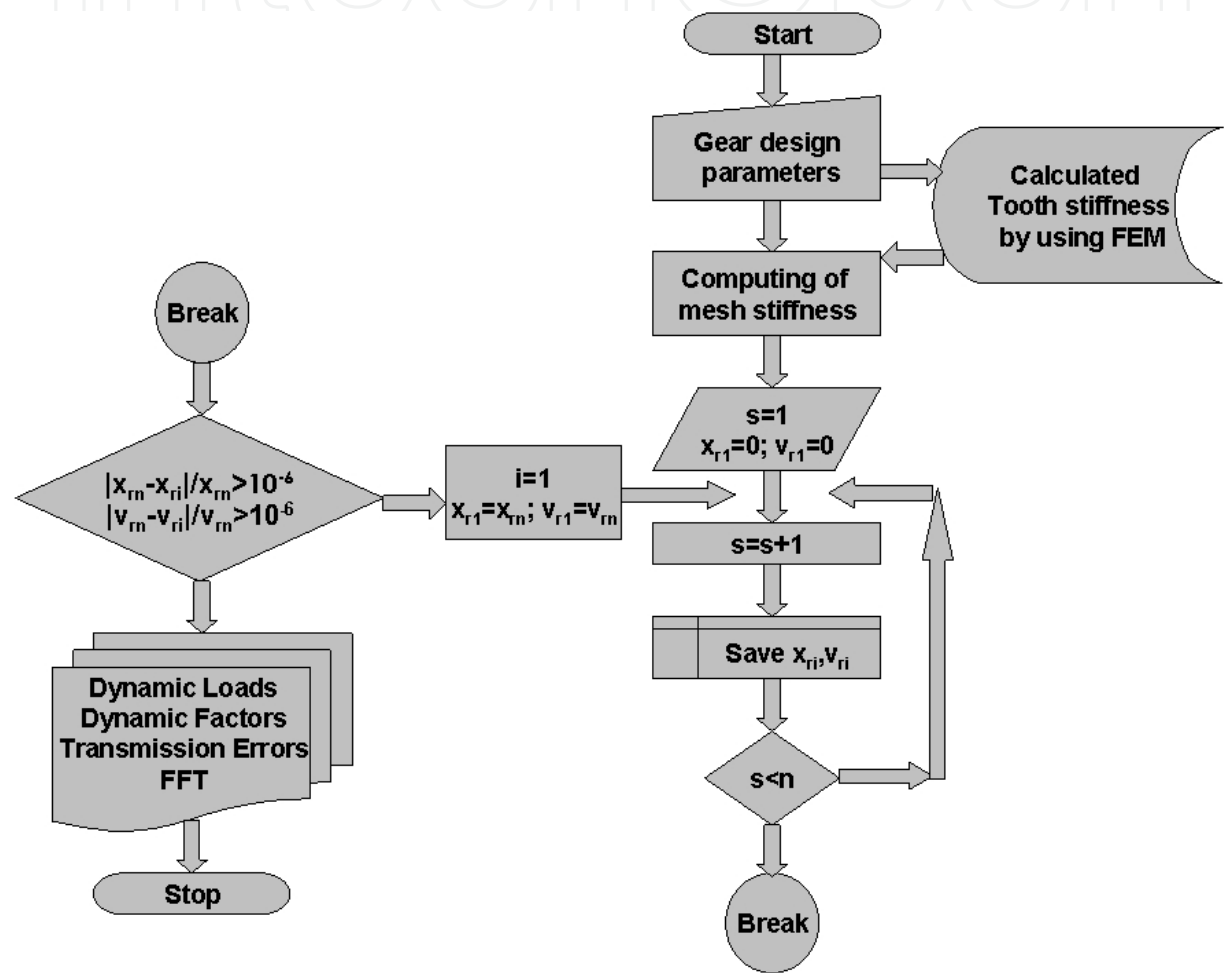


Fig. 2. Flowchart of the developed computer program in MATLAB

5. DYNAMIC virtual tool

Physics-based modeling and simulation is important in all engineering problems. The current mature stage of computer software and hardware makes it possible complex mechanical problems, such as gear design, to be solved numerically. In-house prepared codes to handle individual research projects, graduate, and/or PhD studies; commercial packages for engineers in industry are widely used to solve almost every engineering problem. Tailored with graphical user interfaces (GUIs) and easy-to-use design steps, anyone-even a beginner- can design a gear pair and obtain results, e.g Dynamic Load,

Transmitted Torque, Static Transmission Error as a function of time, and Static Transmission Error Harmonics etc., just by pressing a command button. Lecturers have been increasingly using these packages to increase their teaching performance and student understanding. Based on and triggered by these thoughts, a virtual tool DYNAMIC is prepared that can be used for educational and research purposes. The DYNAMIC is a general purpose gear analyzing tool (Fig. 3).



Fig. 3. The Front panel of the DYNAMIC tool

There are six blocks and a figure block on the front panel of the tool. Three blocks on the right side of the front panel, belong to the parameters which will be defined by the users (Fig. 2 a, b). *Pinion* and *Gear* blocks are reserved for the tooth parameters and *Mechanism* block is for the parameters related to the mechanical variables. Material is set to “Steel” by default and can not be changed by the user.

The two blocks above the figure are *Simulation* and *Figure Selection* panels (Fig 3a). Once the user inputs the needed parameters, he/she clicks the CALCULATE pushbutton to obtain the solution for the specified parameters. In the *Figure Selection* block, from the pop-up menu, user can select which solution to be plotted: Dynamic Load, Transmitted Torque, Static Transmission Error or Static Transmission Error Harmonics (Fig 3b). Then the required figure can be plotted with the PLOT button. Once the solutions are calculated, it is not needed to run the program again and again for each figure option. CLEAR is to clean the figure axes before each plot.

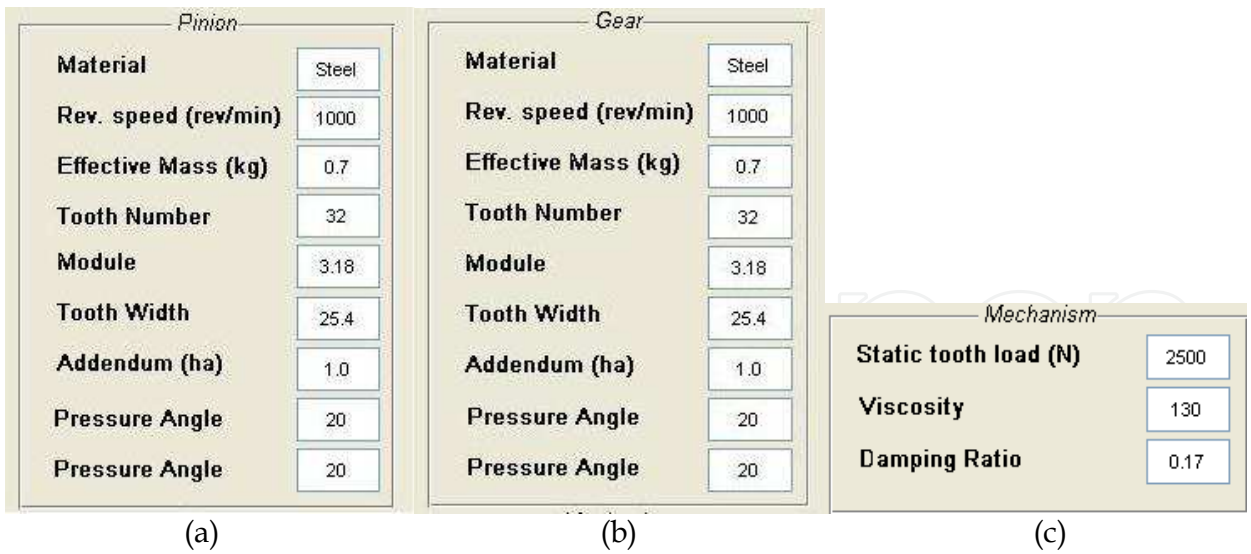


Fig. 4. Variable input blocks: a) pinion, b) gear, c) mechanism

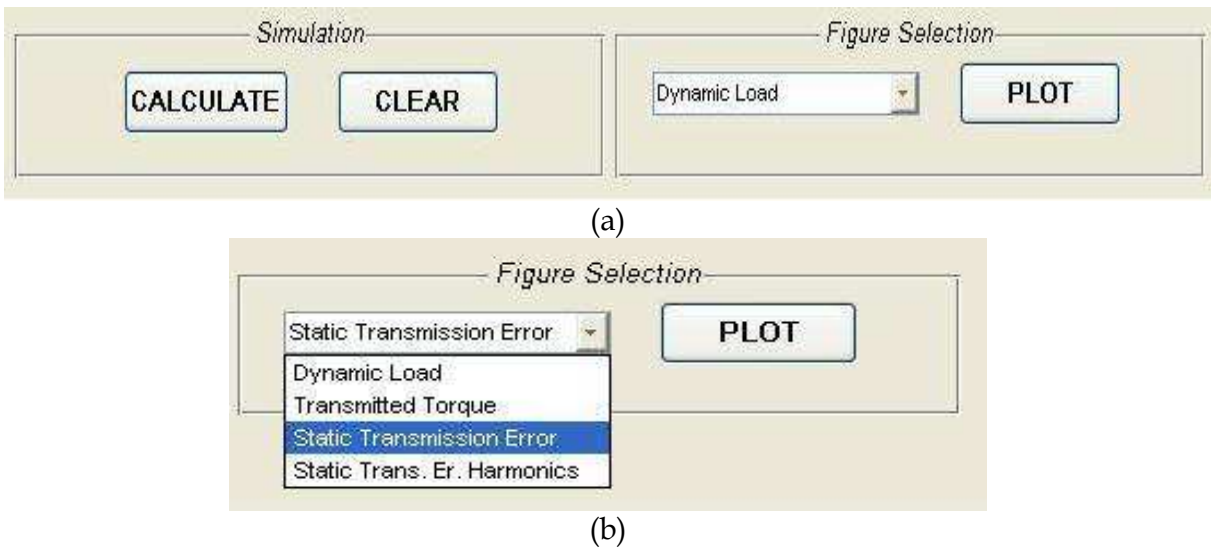


Fig. 5. Simulation command blocks

The variation of dynamic load with respect to time can be seen in Fig. 6. The solutions for different variables can be plotted in one figure, for comparison. In Fig. 6 two different solutions for dynamic load are plotted for different revolution speed. Fig. 7 is an example for Transmitted Torque solution.

6. Results and discussions

The computer program developed has been used for the dynamic analysis of spur gears with symmetric and asymmetric teeth. In this study, seven different gear pairs are considered for the dynamic analysis of spur gears with asymmetric teeth. In order to simplify the analysis, all gear parameters are kept constant, apart from the pressure angle on the drive side and the tooth height. Since the effects of the tooth profile errors are not considered in this study, the analyzed gears are assumed to be “perfect gears” without tooth errors. The properties of these gear pairs are provided in Table.

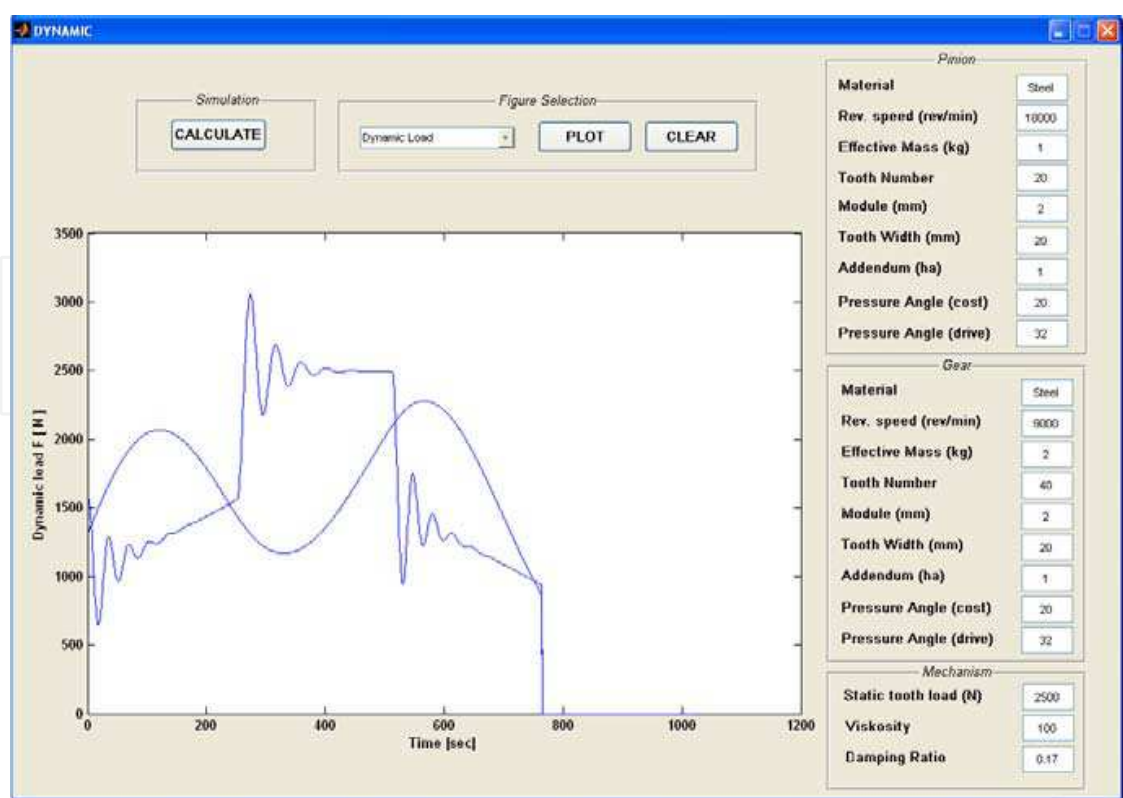


Fig. 6. The comparison of variation of dynamic load for different rotational speeds

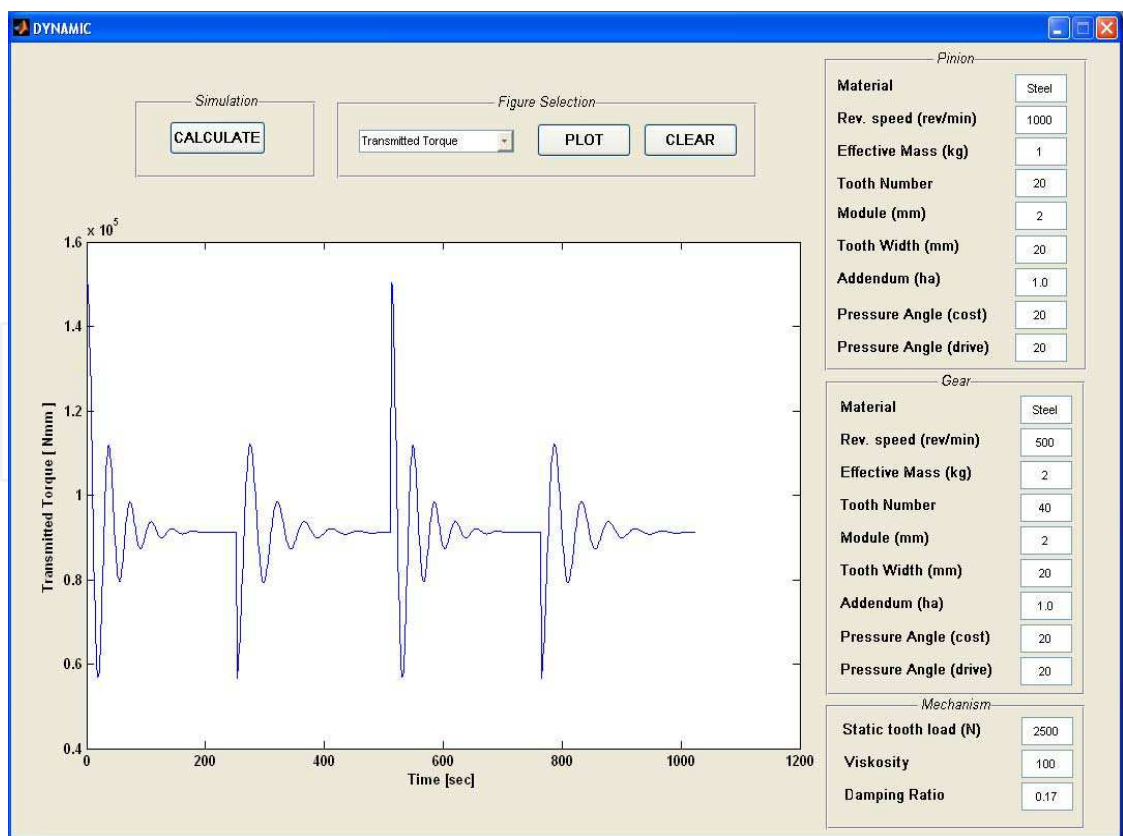


Fig. 7. An example of transmitted torque solution

In a previous work (Karpát, 2005), different approaches for minimizing the dynamic factors and the static transmission errors, in low-contact ratio gears, were reviewed in details. In one of the approaches discussed, the usage of high gear contact ratio was included. It was observed that increasing the gear contact ratio reduced the dynamic load. In literature, minimum dynamic loads were obtained for contact ratios between 1.8 and 2.0. A way of increasing the contact ratio is by using higher addendum values. It should be noted that increasing the value of the addendum leads to a reduction in the bending stress at the tooth root. This occurs through the lowering of the location of the highest point of single tooth contact (HPSTC). The other gear characteristics impacted by high addendum are the thickness of tooth tip and undercut. In this study, for asymmetric gears, high addenda are analyzed, as a means of minimizing the dynamic factors and the static transmission errors (Gear Pair 4 and 5).

	Gear Pair				
	1	2	3	4	5
Module m_n	2 mm	2 mm	2 mm	2 mm	2 mm
Teeth number of pinion z_{n1}	20	20	32	32	32
Pressure angle on coast side α_c	20°	20°	20°	20°	20°
Pressure angle on drive side α_d	20°	24°	32°	24°	32°
Gear ratio	2	2	2	2	2
Mass of pinion M_p	1 kg	1 kg	1 kg	1 kg	1 kg
Mass of gear M_g	2 kg	2 kg	2 kg	2 kg	2 kg
Material	Steel	Steel	Steel	Steel	Steel
Kinematic viscosity	100 cSt	100 cSt	100 cSt	100 cSt	100 cSt
Damping ratio	0,17	0,17	0,17	0,17	0,17
Tooth width	20 mm	20 mm	20 mm	20 mm	20 mm
Addendum h_a	1 m_n	1 m_n	1 m_n	1.32 m_n	1.17 m_n
Contact ratio	1.64	1.49	1.31	1.90	1.52

Table 1. The data of the gear pairs

For the sample gear pair whose dimensions and properties are given in Table 1, variations of dynamic loads are determined for various pinion speeds between 1000 rpm and 20 000 rpm. As an example, the dynamic load variation of gear pair 1 for 1000 rpm, 3000 rpm, 10 000 rpm and 18 000 rpm is shown in Figure 8.

Fig. 9 shows the relationship between the dynamic factors and the rotational speed. When comparing the maximum dynamic factors in the corresponding gear pairs in Fig. 9. (e.g., Gear Pair 1 versus Gear Pair 3), it is generally stated that the dynamic factor for spur gears with asymmetric teeth increases with increasing pressure angles on the drive side. Furthermore, it is obvious that the sample Gear Pair 4, which is the gear pair with the

highest gear contact ratio 1.90, has a lower dynamic load, at all speeds; this indicates that the impact of gear contact ratio on dynamic loads. The highest dynamic factor is observed at the resonant rotational speed (about 12 000). Beyond this speed, the asymmetric teeth have consistently higher dynamic factors than symmetric teeth. One of reasons for that may be the effect of contact ratio on dynamic loads. As the pressure angle on drive side increases, the contact ratio decreases. However, the dynamic factor in gear systems decreases with increasing the contact ratio. This result may be due to the narrow single contact zone. Because of the narrow single contact zone, this zone is passed speedily as gear rotate and system can not respond. Other reason may be seen by analyzing the variation of mesh stiffness with respect to time. As can be seen from this figure, in the single contact zone, the asymmetric gear (Gear Pair 4) has higher mesh stiffness than the symmetric gear (Gear Pair 1). The high mesh stiffness is one of the reasons for the high dynamic factor observed in Fig.9.

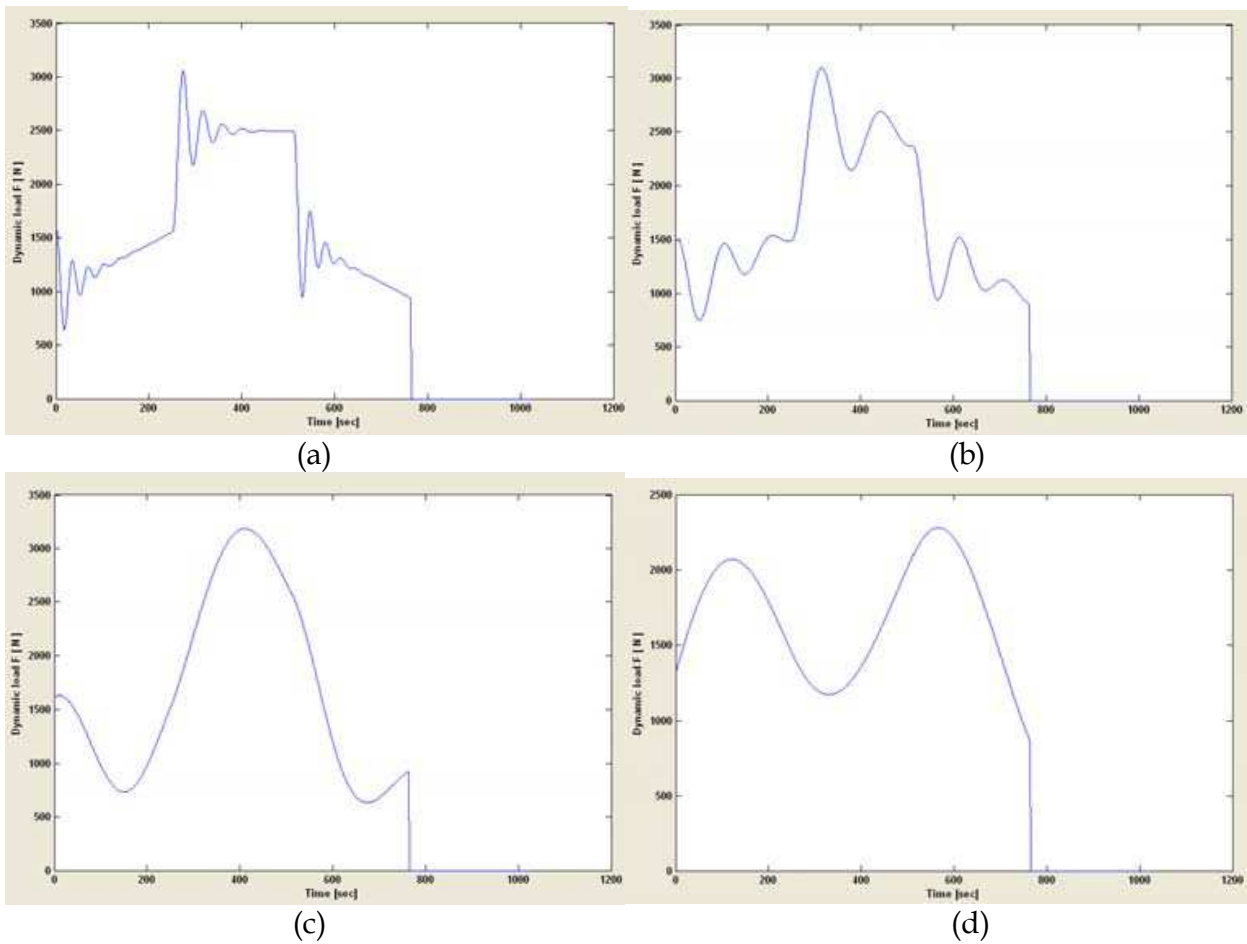


Fig. 8. Variation of dynamic load with rotational speed of pinion: a) 1000 rpm b) 3000 rpm c) 10 000 d) 18 000 rpm

Fig. 10 shows the impact of increasing the pressure angle, on the drive side, on the static transmission error. Generally, changing the pressure angle will impact the tooth mesh characteristics, such as the tooth contact zone and contact ratio. Fig. 11 indicates that the single tooth contact zone increases with increased pressure angle. Thus, compared to gears with symmetric teeth, gears with asymmetric teeth have a larger single tooth contact zone.

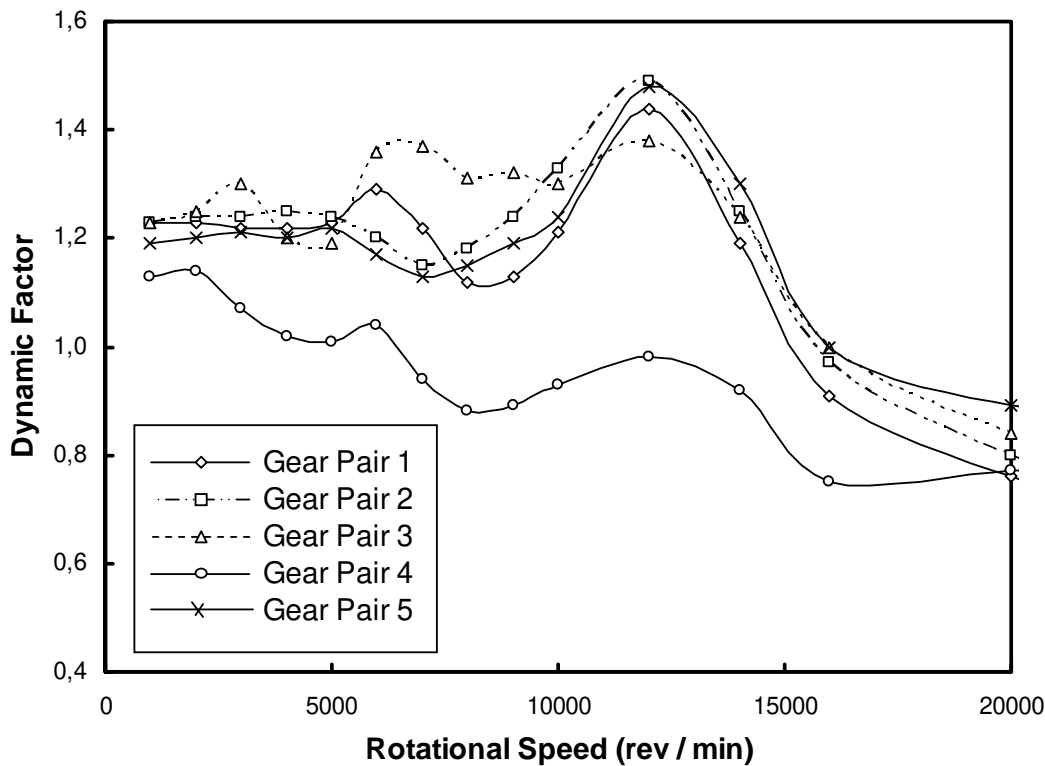


Fig. 9. The maximum dynamic factors with respect to rotational speeds

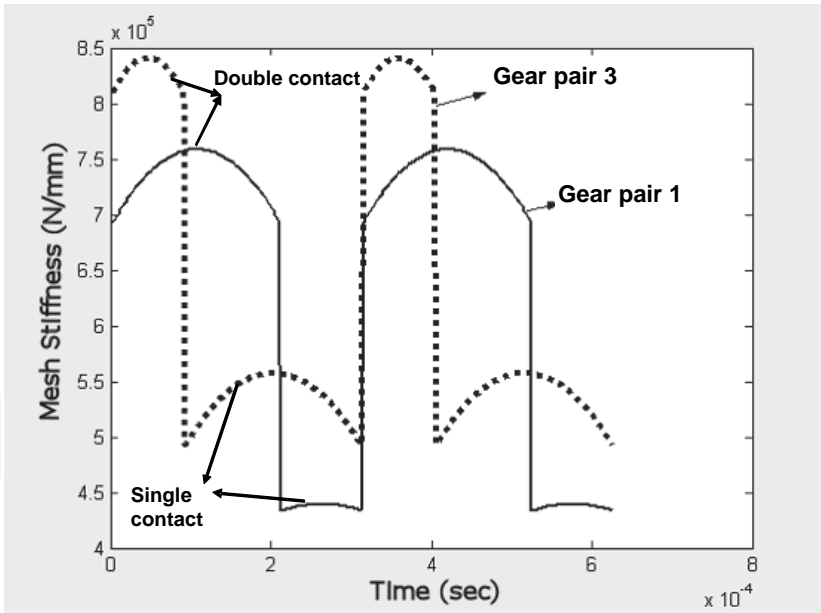


Fig. 10. The variation of mesh stiffness with respect to time for Gear Pair 1 (symmetric teeth) and Gear Pair 3 (asymmetric teeth)

Furthermore, the static transmission error, at the center of the single tooth contact zone, decreases with increasing of pressure angle. The frequency spectra of the static transmission errors are depicted in Fig. 11. In these figures, the sum of first five harmonics slightly increases with increasing pressure angle.

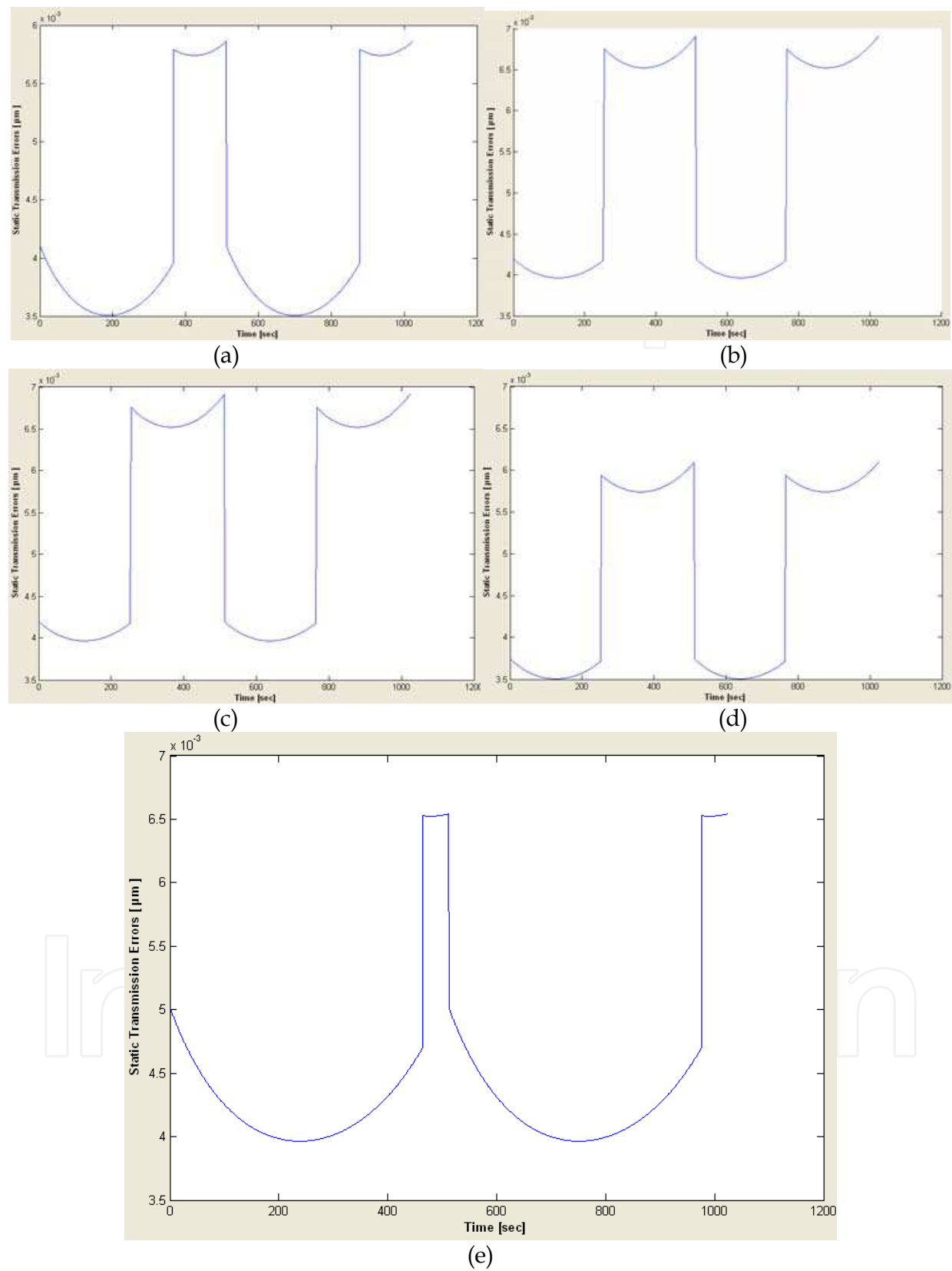


Fig. 11. Static transmission errors (a) Gear Pair 1 ($\alpha_c = 20^\circ, \alpha_d = 20^\circ$), (b) Gear Pair 2 ($\alpha_c = 20^\circ, \alpha_d = 24^\circ$), (c) Gear Pair 3 ($\alpha_c = 20^\circ, \alpha_d = 32^\circ$), (d) Gear Pair 4 ($\alpha_c = 20^\circ, \alpha_d = 24^\circ$), (d) Gear Pair 5 ($\alpha_c = 20^\circ, \alpha_d = 32^\circ$)

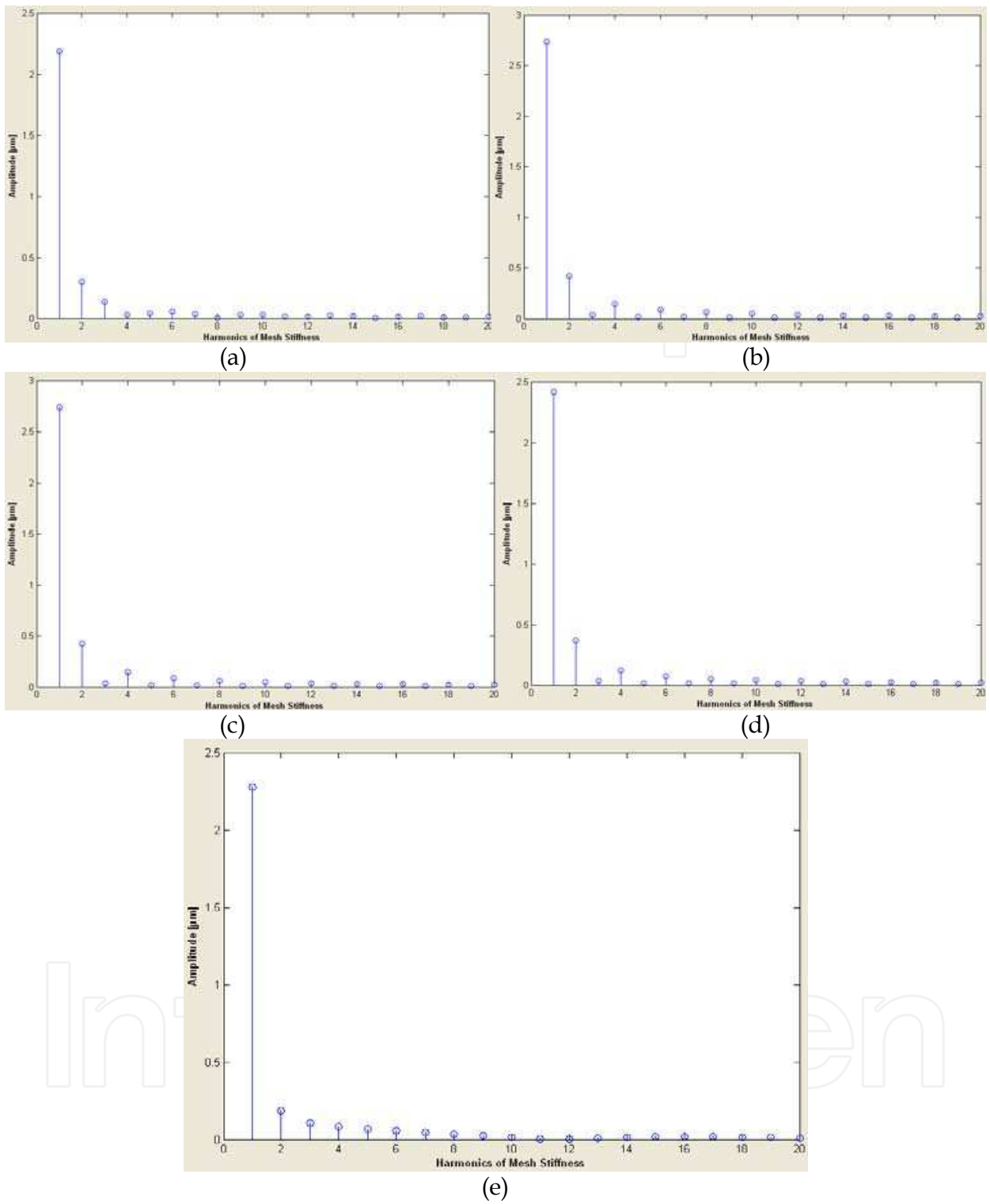


Fig. 12. Frequency spectra of the static transmission errors (a) Gear Pair 1 ($\alpha_c = 20^\circ, \alpha_d = 20^\circ$), (b) Gear Pair 2 ($\alpha_c = 20^\circ, \alpha_d = 24^\circ$), (c) Gear Pair 3 ($\alpha_c = 20^\circ, \alpha_d = 32^\circ$), (d) Gear Pair 4 ($\alpha_c = 20^\circ, \alpha_d = 24^\circ$), (e) Gear Pair 5 ($\alpha_c = 20^\circ, \alpha_d = 32^\circ$)

Fig. 12 (d) and (e) shows the static transmission error for increased values of addendum for asymmetric teeth. Increasing the addendum, the amplitude of the static transmission errors is decreased for a comparable pressure angle. Additionally, the single tooth contact zone

decreased for a comparable pressure angle. In Fig. 12 (d), it is noted that the asymmetric tooth with $h_{ap} = 1.32m_n$, $\alpha_c = 20^\circ$, and $\alpha_d = 24^\circ$, has the lowest static transmission error. Furthermore, the difference in the magnitude of the error, in the single tooth contact and double teeth contact zones, is also smallest for this tooth configuration. In Fig. 12 (d), the amplitudes of harmonics of static transmission errors are significantly reduced when asymmetric teeth with long addendum, providing high gear contact ratio close to 2.0 are used. By referring to Fig. 12, it can be inferred that when designing asymmetric gears, for achieving reduced dynamic response, one may consider using a high addendum. In summary, for asymmetric teeth, increasing the addendum leads to a significant decrease in the dynamic factor. The maximum reduction of the dynamic factor is achieved for a gear contact ratio of about 2.0. The result implies that the usage of long addendum for involute spur gears with asymmetric teeth may be an alternative way to reduce the dynamic response as well decreasing tooth stress at root.

7. Conclusions

Virtual tools have become very effective in teaching engineering problems. The user need not to know graphical user interface details, programming tips, etc. Instead, such tools have the capability of handling a variety of different gear problems.

A MATLAB-based virtual tool, DYNAMIC, is introduced to analyze dynamic behavior of spur gears with asymmetric tooth design. The DYNAMIC is used to compare conventional spur gears with symmetric teeth and spur gears with asymmetric teeth in this study. The results for dynamic load, dynamic factor, transmitted torque, static transmission error and static transmission error harmonics are obtained for various tooth parameters to show the powerful aspects of asymmetric teeth.

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The book consists of 24 chapters illustrating a wide range of areas where MATLAB tools are applied. These areas include mathematics, physics, chemistry and chemical engineering, mechanical engineering, biological (molecular biology) and medical sciences, communication and control systems, digital signal, image and video processing, system modeling and simulation. Many interesting problems have been included throughout the book, and its contents will be beneficial for students and professionals in wide areas of interest.

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InTech Europe

University Campus STeP Ri
Slavka Krautzeka 83/A
51000 Rijeka, Croatia
Phone: +385 (51) 770 447
Fax: +385 (51) 686 166
www.intechopen.com

InTech China

Unit 405, Office Block, Hotel Equatorial Shanghai
No.65, Yan An Road (West), Shanghai, 200040, China
中国上海市延安西路65号上海国际贵都大饭店办公楼405单元
Phone: +86-21-62489820
Fax: +86-21-62489821

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