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Vibration analysis of multi-body dynamic model of combat vehicle gearbox

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Abstract. The combat vehicle gearbox, during the operation, generates vibration signals being related to the technical condition of gearbox. The analysis of the vibration signal could be used to determine accurately the behaviour of gearbox. Along with the development of the computer technology, the multi-body dynamic solution has been used widely to simulate, analyse, and determine the technical condition of gearbox. The purpose of this paper is to introduce the dynamic model of combat vehicle gearbox, and the simulation process based on the multi-body dynamic software, namely MSC.ADAMS. This proposed model allows the detection of failure conditions of individual gears and bearings in the gearbox. In this way, the fault conditions of the individual transmission components are identified. In the future, we would like to include a material wear module in the model, and we would like to model the life of the gearbox. We assume that we would also carry out accelerated tests of the gearbox to verify validity.

1 Introduction

It is obvious that the combat vehicle has required an advantage of mobility in operations, and to achieve the objective, the combat vehicle is required to work with high performance and reliability of transmission system. The gearbox consists of complex components, and it plays an important role in transmission system via transferring the torque from the engine to wheels at different ratios. In practice, the gearbox generates a gear noise and vibration signal that can be correlated with the technical condition of gearbox. The study about the behaviour of gearbox using vibration signal is called vibration diagnostics. Over past decades, there are a great number of researchers who have investigated the gearbox in experiment. For example, the spectral analysis technique was used to identify the gearbox failures (e.g. imbalance, misalignment, mechanical looseness, bearing defects, gear defects) [1]. Another good example is that the waveform can be detected not only the location of failures but also when it occurred [2]. The experiment could be used effectively to assess the technical condition of gearbox. However, it requires the knowledge of experts, and the long - term testing procedure with high investment costs.

To overcome those drawbacks, the simulation method has been used extensively to analyse the behaviour of gearbox in addition to the development of the computer technology. There is number of methods that are used to simulate, such as mathematical modelling (e.g. Matlab Simulink), finite element method (e.g. Ansys), and multi-body dynamic solution (e.g. MSC.ADAMS). Among these methods, the last one is the most proper technique for the vibration signal analysis. There are some computational multi-body dynamic models of gearbox that were developed to simulate the failure of gearbox, such as the simulation modern power train in the field of vibration [3], or the simulation of gear tooth breakage [4]. However, these models were simulated for separate failures, and the working condition of gearbox was not taken into account. Furthermore, the stiffness between different gear tooth surfaces will result in different vibration signal. This paper will introduce the dynamic model of the gearbox, and this model was solved by the multi-body dynamic simulation method. This model is simulated by using the computer software MSC.ADAMS in two cases, when the gear tooth is in good and breakage conditions. After that, the result of two cases is the vibration signal which is then analysed in both time domain and frequency domain.

2 The dynamic of gearbox

2.1 Fundamental of vibration analysis

In practice, the gearbox generates a gear noise and vibration during its operation. The gear noise is a vibrational phenomenon that is emitted from the meshing gears. Particularly, the vibration is transmitted through the mechanical component, such as gears, shafts, or bearings. After that, the gearbox housing is stimulated and vibration is converted into airborne. In fact, the gearbox is affected by working conditions, for example, excessive loads, insufficient lubrication, or manufacturing errors. Consequently, when the failure occurs, the transmission error is generated. The transmission error is the difference between the actual position of the output gear and the perfect position of the

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gear. It is assumed that the transmission error is associated with the geometry, deflection, dynamic of the gears and the variation of the applying loads. The transmission error is required to minimize to reduce the noise generation. The geometries of transmission error are shown in the Fig.1.

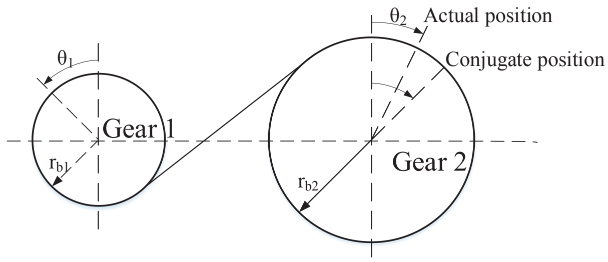

Fig. 1. Geometries of transmission error [5]

In the Fig.1, the transmission error can be formulated as the difference between the theoretical and actual angular displacement of two gears, and it can be expressed as follows [6]

$$T\mathrm{E}(\varpi) = \theta_1(\varpi) - \frac{N_2}{N_1}\theta_2(\varpi). \tag{1}$$

Or, it also can be written:

$$T\mathrm{E}(\varpi) = \theta_1(\varpi) - \frac{r_{b2}}{r_{\mathrm{b1}}}\theta_2(\varpi) \tag{2}$$

where: $TE(\omega)$ - transmission error; $\theta_1(\omega)$, $\theta_2(\omega)$ - the angular positions; N_1, N_2 - angular velocities; r_{b1}, r_{b2} - base radii of gear 1 and gear 2.

2.2 Gearbox dynamic model

In general, it is likely that the vibration part could be considered as the spring - mass - damper system. This system consists of three fundamental parameters, including a mass (m), stiffness (k), and damping (C). To describe the vibration motion of meshing gear, the dynamic system is assumed as the coordinate. That is called multi-DOF system (degree of freedom), or and N-DOF system, where N is the number of coordinates required. The simple coordinated system of the meshing gear is shown in the Fig.2. The system is the 6 DOF model, in which each gear has 3 DOF (one rotational and two translational) [6].

The equations of motion for this model can be expressed as follows. The equation in the 'x' direction for the pinon and gear [6]:

$$m_p\ddot{x}_p = -K_{xp}x_p - C_{xp}\ddot{x}_p + F_p \tag{3}$$

$$m_g\ddot{x}_g = -K_{xg}x_g - C_{xg}\ddot{x}_g + F_g. \tag{4}$$

The equation of motion in the 'y' direction for the pinion and gear [6]:

$$m_p\ddot{y}_p = -K_{yp}y_p - C_{yp}\ddot{y}_p - \mathrm{N} \tag{5}$$

$$m_g\ddot{y}_g = -K_{yg}y_g - C_{yg}\ddot{y}_g + \mathrm{N}. \tag{6}$$

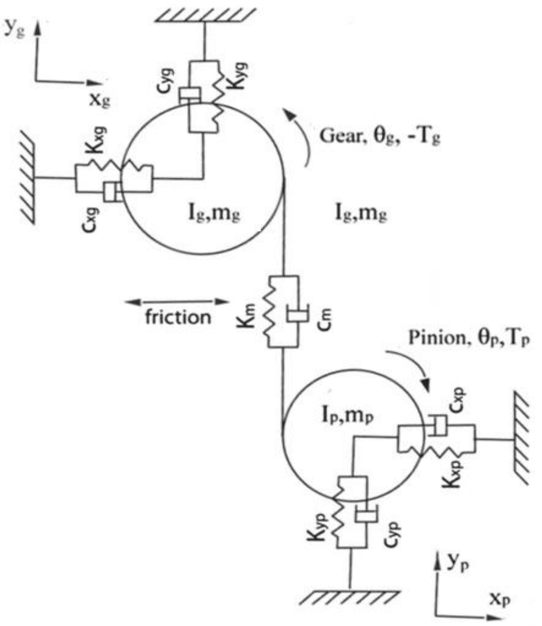

Fig. 2. Dynamic model of meshing gear with six DOF [6].

The equation of motion in the θ direction for the pinion and gear [6]:

$$I_p\ddot{\theta}_p = r_pN + T_p + M_p \tag{7}$$

$$I_g\ddot{\theta}_g = -r_gN - T_g + M_g \tag{8}$$

where: m_p/m_g - mass of the pinion/gear; I_p/I_g - mass moment of inertia of the pinion/gear; K_{xp}/K_{yp} - radial stiffness in the x/y directions of the pinion; K_{xg}/K_{yg} - radial stiffness in the x/y directions of the gear; C_{xp}/C_{yp} - radial damping in the x/y directions of the pinion; C_{xg}/C_{yg} - radial damping in the x/y directions of the gear; K_m - equivalent mesh stiffness; C_m - mesh damping coefficient; T_p/T_g - torque applied on the pinion/gear; r_p/r_g – radius of pinion/gear; F_p/F_g - friction forces for pinion/gear; N - equivalent tangential force.

3 The simulation of the mechanical gearbox using multi-body dynamics software MSC.ADAMS

3.1 Multi-body dynamics simulation

The multi-body dynamics simulation is used to research the dynamics of moving parts, and the change of loads and forces being distributed throughout mechanical systems. This is a method of numerical simulation in which the multi-body system consists of various parts (rigid or elastic bodies). The parts' connections can be modelled with kinematic constraints (e.g. joints) or force elements (e.g. spring, dampers). Unilateral constraints and Coulomb – friction can also be used to model frictional contacts between bodies. In practice, multi-body simulation is a useful software for conducting motion analysis. Specially, the core of multi-body simulation software is Solver, in which the equations of motion is solved by computation algorithms.

The multi-body simulation process that performed vibration analysis consists of five main activities. First, the physical model with geometries (e.g. length, width, depth, dimension, distance) is created. Generally, the model is created by the professional 3D-CAD designed software. Next, the physical characteristics, for example, inertia properties, mass, and dynamic friction coefficient are set. Then, the kinematic definitions, e.g. translational and rotational, are fulfilled. The next activity is that the model is simulated with initial conditions. Finally, the results are analysed and evaluated.

There are a number of multi-body simulation software, such as MBDyn, COMSOL Multiphysics, or MSC.ADAMS. Among these software, the last one is the most famous computational software that has been broadly used to perform multi-body dynamics simulation. MSC.ADAMS (Automatic Dynamic Analysis of Mechanical System) is a virtual prototype simulation software which has been developed by Mechanical Dynamics, Inc. company. This software is significantly used for modelling, analysing, and optimizing the multi-body mechanical system. ADAMS has several modules, for instant, ADAMS/View, ADAMS/Post-processing, ADAMS/Solver, ADAMS/Driveline, ADAMS/Car…, and each module has different function.

3.2 Simulation combat vehicle gearbox

3.2.1 Modelling 3D gearbox

The combat vehicle gearbox that was chosen for simulation is equipped with the vehicle UAZ 469. This is a mechanical gearbox that has four forward speeds and a reverse. Shafts and gears were made from steel, and bearings were considered as SKF gear in the mechanical dynamic handbook.

The 3D gearbox model is generated in the Inventor software. This software provides the possibility to set up parameters of model. Parameters of gearbox, such as a structure, a shape, a dimension and feature part are created by command tools. Furthermore, the effective library tool provides standard parts such as shafts, gears, bearings… with different specifications. The basic geometrical parameters of gearbox that used for modelling are shown in the Table 1.

Table 1. Gearbox technical specification [4].

Description	Item	Modul (mm)	Face width (mm)	Type	Helical Angle (°)	Tooth number
Constant gear mesh	Pinion	3	20	Helical	28,85	$Z_{0R} = 15$
	Gear	3	20	Helical	28,85	$Z_{0N} = 32$
1st speed	Pinion	3,5	20	Spur		$Z_{1R} = 15$
	Gear	3,5	20	Spur		$Z_{1N} = 29$
2nd speed	Pinion	3	20	Helical	28,85	$Z_{2R} = 21$
	Gear	3	20	Helical	28,85	$Z_{2N} = 26$
3rd speed	Pinion	3	20	Helical	28,85	$Z_{3R} = 27$
	Gear	3	20	Helical	28,85	$Z_{3N} = 20$
Reverse	Pinion	3,5	15	Spur		$Z_{RR} = 15$
	Gear	3,5	15	Spur		$Z_{RN} = 19$

3.2.2 Simulation in MSC.ADAMS

The 3D gearbox model is transferred to multi-body dynamic model by MSC.ADAMS software. However, the model after being transferred does not have properties, like mass, constraints, or kinematics…, and this leads to the fact that the new virtual model is required to fulfil. To perform this model, the material of parts is chosen, and then the bodies' properties such as, centre mass, mass, stiffness and rotational inertia are obtained.

After that, the force or torque and kinematic constraints between bodies are added. Based on the driveline of torque from the input shaft to output shaft, the suitable joints (e.g. fixed, revolute, translational…) are added to the relative contact surface. The motion of model (e.g. force or torque, speed) is added at the input shaft. Finally, the model is simulated and the results is showed in Postprocessor. The multi-body dynamic model of gearbox is shown in the Fig. 3.

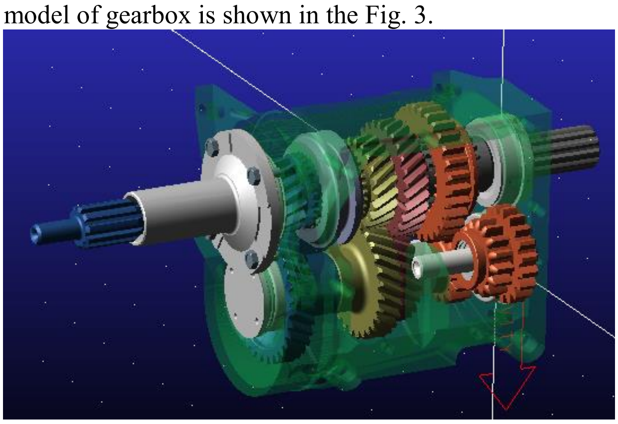

Fig. 3. The multi-body dynamic model of gearbox.

3.2.3 The mathematical formulas of contact force

Regarding the simulation about the vibration signal of meshing gears, the contact force between gears bodies were modelled. The contact force depends on the geometries of contact surface, the load, and the material properties. In ADAMS, the contact force model is shown in Fig. 4, and the equation can be expressed [7]:

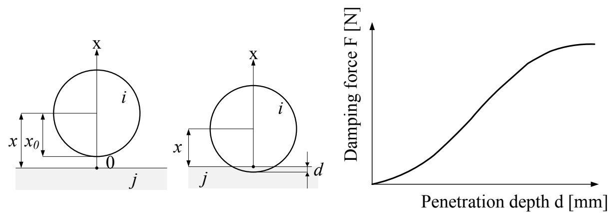

Fig. 4. Contact force (left) and the relation between damping force and penetration (right) in MSC.ADAMS [8].

$$F = \begin{cases} K(x_0 - x)^e + CS\dot{x}, & x < x_0 \\ 0 & x \geq x_0 \end{cases} \tag{9}$$

where: F - contact force; K - stiffness; C - damping coefficient; x_0-x - deformation; e - contact force exponent; S - step function.

$$F = \begin{cases} 0 & x > x_0 \\ (3 - 2\Delta d)\Delta d^2 & x_0 - d < x < x_0 \\ 1 & x \le x_0 - d \end{cases} \quad (10)$$

where: $\Delta d = x_0 - x$ - the deformation of the body.

The equation (10) shows that the contact force includes two parts. The sooner is an elastic component $K(x_0-x)^e$ that is functioned like a nonlinear spring. The latter is the damping force *CS (dx/dt)* that acted like the contact - collision velocity. The equation (11) displays the step function, in which the damping force is proportional to the penetration depth. To avoid the discontinues caused by the dramatic variation of damping force while contact - collision occurs, the damping force is defined as zero when the penetration depth is zero. The contact force between bodies are divided into two types. The first type is the discontinuous contact, such as the falling ball bouncing on the floor. While, another type is the continuous contact, such as a nonlinear spring. In ADAMS, there are two algorithms methods for computing a contact force, namely the Restitution Method and the Impact Method. In this study, the Impact Method was chosen to simulate the gearbox model. The necessary parameters for computing are shown in the following stiffness K. According to the Hertzian elastic contact theory, the stiffness of the two contact parts could be described by a pair of ideal contacted cylindrical components. This can be expressed [11]:

$$\begin{cases} K = \frac{4}{3}R^2E^* = \frac{4}{3}\left[\frac{id_1\cos\alpha_t\tan\alpha_t}{2(1+i)\cos\beta_i}\right]^{\frac{1}{2}}E^* \\ \frac{1}{E^*} = \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2} \\ \beta_b = \alpha\tan(\tan\beta\cos\alpha_t) \\ \frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \end{cases} \quad (11)$$

where: R_1, R_2 - radius of two gear contact points, E^* - equivalent Young's modulus, i - gear ratio, d_1 - diameter of the standard pitch circle, α_t', α_t – transverse pressure angle at engaged and standard pitch circle; β, β_b - helical angle of the pitch base circle, v_1, v_2 - Poisson ratio of the pinion and gear; E_1, E_2 – Young's modulus of the material of two gears respectively.

The gear mesh contact stiffnesses were calculated and shown in Table 2 [4]:

Table 2. Contact stiffness.

Pair of gears	Stiffness (N/mm$^{3/2}$)
The constant - mesh gears	$6{,}48.10^5$
1st speed gears	$6{,}84.10^5$
2nd speed gears	$5{,}59.10^5$
3rd speed gears	$5{,}12.10^5$
Reverse gears	$6{,}98.10^5$

The materials of couple gear engaged were made from alloy steel, so the Young's modulus $E = 2{,}1.10^5$ and Poisson ratio $v = 0{,}29$. The force exponent was determined after several trial simulations and the value is $e = 2{,}2$.

Penetration depth was considered as the numerical convergence in ADAMS $d = 0{,}1$.

Damping coefficient is from 0,1 % to 1 % of the stiffness K, and in this paper, the damping coefficient used for simulation is C = 1000 Ns/mm. The dynamic and static friction coefficient, and viscous velocity are chosen from the mechanical handbooks and are shown in the Table 3.

Table 3. Friction value [8].

Frictions	Values
Static friction coefficient (μ_s)	0,1
Static transonic speed (ν_s)	1 (mm/s)
Dynamic friction coefficient (μ_d)	0,08
Dynamic transonic speed (ν_d)	10 (mm/s)

3.2.4 The initial simulated condition

After the multi-body dynamic model was established (including 3D modelling, adding material, parameters, constraints and motion), the model was simulated. The investigation was performed in two cases, the one was in the health-gear condition while another was in the breakage-gear condition. The breakage was created as similar as the failure of gearbox in practice. In addition, this was located in the addendum of the pinion tooth of the 4th mating gear, which is helical with helical angle being 28,85 the width being 20 mm, and the module being 3. The drive gear has 15 teeth and the driven gear has 32 teeth. Both simulated cases were investigated in the operation condition that is similar to the realistic state when the engine speed was set at 2000 rpm in accordance with maximum torque 166,7 Nm. The modes were simulated with the end time being 1 and the step size being 2048. The results were shown in time domain and frequency domain.

3.3 Results and discussions

The multi-body dynamic model of gearbox is simulated in two cases. Regardless of the first case, the health gear is simulated; meanwhile, the second case is the breakage gear. Both cases were investigated at the same operation condition: loads and the rotational speed. The acceleration was measured and analysed in the time domain and frequency domain. Fig. 5 and Fig. 6 show the acceleration of 4th mating gear when the gear is in the health condition and the gear is in the breakage condition.

In the time domain, both cases witness the dramatic fluctuation of the acceleration signal that is correlated with the mating gear. In the first case, the fluctuation followed by the period of stability. Meanwhile, the second case experiences the significant increase at the breakage positions that remains for a short time prior to quick recovery to the vibrated trend.

In the frequency domain, the overall signal of the breakage gear is higher than that of the heath gear. It can be explained that the overall vibration signal is resonated by those faults. While at the breakage position, the acceleration signal roars up remarkably.

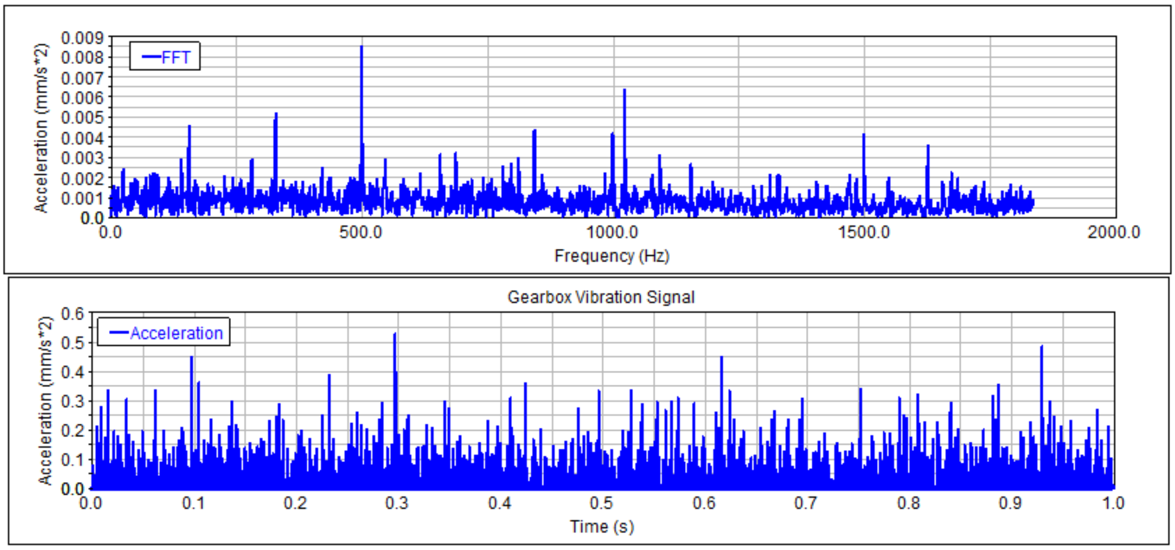

Fig. 5. Vibration signal of health gear.

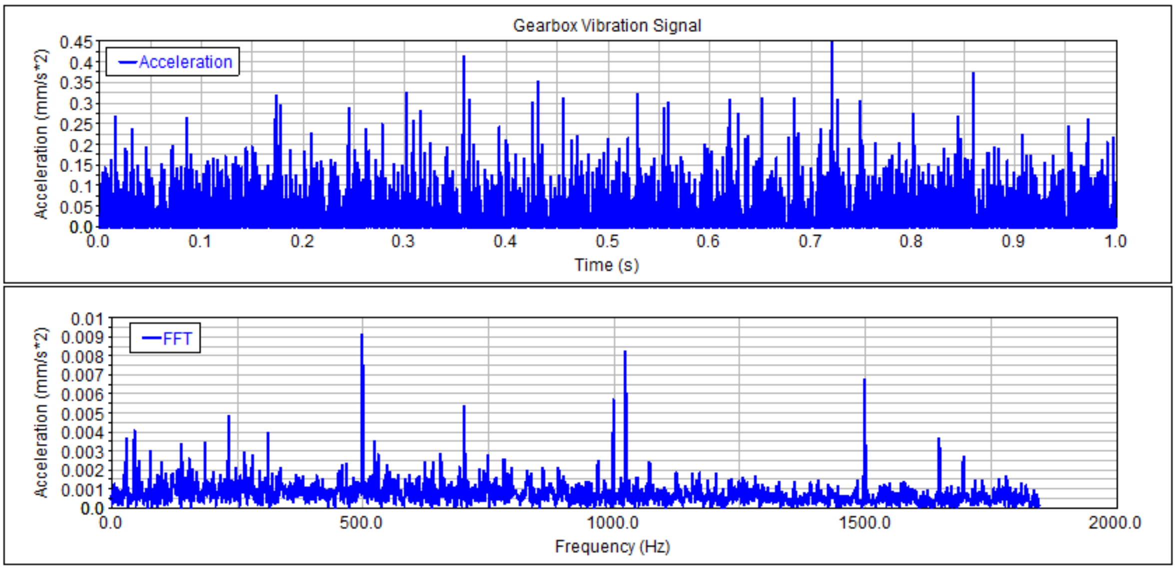

Fig. 6. Vibration signal of breakage gear.

4 Conclusion

To conclude, in this paper, the dynamic model of gear contact with six degree of freedom was developed. Also, the dynamic model was solved by the multi-body dynamic simulation method. The simulation process was conducted in MSC.ADAMS software. The gearbox was investigated in two cases when the gear tooth is in good and breakage conditions. The results show the acceleration signal of the third mating gear, and the signal was analysed in the time domain and frequency domain. The result of the health gear model was showed clearly difference from the one of the breakage gear model. The gearbox failures could be identified obviously after the vibration signal analysis.

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