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Dynamic analysis of spur gear with backlash using ADAMS

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ABSTRACT

Gear tooth profile can deviate from its initial design shape and size as a result of increasing service time under time-varying load, introducing external agents like debris, overheating due to friction, wear, and in generally due to other nonlinear factors such as backlash. As a result, the dynamics of the gear will also vary depending on the resulting gear tooth profile. In this study, the worn-out gear tooth is modelled as a backlash by assuming a uniformly distributed worn out surface and the effect on the dynamic performance is investigated. By changing the amount of backlash of the gear tooth from 0 mm to 1 mm by 0.2 mm increment, the gear is modelled and analysed for three loading cases using MSC ADAMS software. This paper discusses the effect of backlash or uniformly worn out spur gear tooth faces on the dynamics specifically the contact and angular accelerations of the gear.

1. Introduction

Backlash is the gap between gear teeth at the pitch circle. This distance can be introduced in the gears either deliberately or without any intention such as due to the manufacturing error and assembly fault. The basic purpose of introducing backlash in gear mates is to avoid locking between gear teeth and to prevent simultaneous tooth contact on both sides of the gear. Introducing a small amount of backlash in gear mates is desirable to make the necessary gap for lubrication, free play and partial expansion of gears. On the other hand, if the amount of backlash is increased exceedingly, it will cause noise and vibration [1].

The dynamic behaviour of the gear tooth forces is quite different from the static or quasi-static forces in both magnitude and shape. These tooth forces show various nonlinear phenomena such as backlash-induced tooth separations and discontinuities. Therefore, surface wear is strongly related to the contact stresses and the dynamic performance. The dynamic response amplitudes of a backlashed gear should vary depending on the amount of the backlash and the dilation of the tooth profile from the perfect involute.

Gear tooth profile modifications such as tip and root relief are commonly used for the reduction of dynamic forces of the gear tooth and to minimize the stress riser profile by avoiding sharp edges of the gear. There are, however, unavoidable errors that arise from manufacturing and assembly errors that influence the dynamic response because they act as a transmission error excitation at the gear mesh interface. Surface wear is a material removal process from the contacting surfaces that results in a deviation from the initially designed tooth profiles. Therefore, the dynamics of the gear pair with backlash or worn out tooth face and the gear pair without backlash (a little backlash which in turn is considered as the gear pairs with no wear) have different behaviour.

Baumann and Bertsche [2] reached on a conclusion that by avoiding meshing impacts between the gear tooth, the level of gear rattle noise can be reduced. In addition, Fernandez Del-Rincon et al. [3] shown that the existence of lubricant plays tremendous advantages on the dynamic behaviour of the gear. In addition, Russo [4] conducted an experiment on helical gear pairs in the idle gear rattle condition by varying the lubrication mechanism to reveal the effect of lubrication on gear rattling. This shows indirectly that backlash has an effect on the dynamic behaviour of the gear.

Jian et al. [5] reported that backlash affects the vibro-impact properties significantly, where tooth separation and drive side mesh are observed with the decrease of load coefficient, leading to drive and back-side tooth impacts. Modelling of the dynamics is useful in cases such as conditional monitoring since it allows to simulate a vibration signal for different conditions of a gearbox

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which is difficult to conduct experiments. Gear mesh modelling is nowadays preferred due to the advantages to make virtual experiments by easily varying parameters. Several methods have been used to simulate the behaviour of gear pairs using mathematical modelling such as using the numerical methods available in MATLAB [6,7], finite element methods [8,9] in ABAQUS or ANSYS and multi-body dynamics methods [10 –12] using MSC ADAMS. MSC ADAMS is a computing software used for modelling, analysis and optimization of multibody mechanical systems.

Motivated by the strict noise level standards in certain applications involving spur gear meshes, Moradi and Salarieh [13] investigated backlash nonlinearity of spur gears using a classical single degree of freedom model by accounting for the dynamic transmission errors and the effect of operational dynamics and manufacturing parameters. Bahk et al. [14] investigated the effect of tooth profile modification on the dynamics of a planetary gear using analytical response solution, where the static and dynamic response can be varied based on tooth profile modification. To minimize the static and dynamic responses, the optimal amount of tooth profile modification should be done. Rocca et al. [15] provided an investigation into the influence of speed fluctuations and transmission error using analytical and experimental work. Theodossiades and Natsiavas [16] investigated the non-linear dynamics of the gear responses with periodic time varying mesh stiffness and backlash using steady-state and transient analysis.

Taking the above literature concepts into consideration, the aim of this article is to conduct analysis of the dynamics of the spur gear with different amount of backlash using MSC ADAMS software. The developed 3D model of the spur gear system is simulated by motions and forces and the observed dynamic responses are discussed.

2. Gear modelling parameters and boundary conditions

ADAMS provides machinery components that are used to model different types of gears by specifying the common parameters such as module, number of teeth for both gear pairs, face width and gear tooth thickness. The design parameters of spur gear considered in this study are given in Table 1. The gear mates are made from steel SAE-AISI 1045, whose physical properties are: density, $\rho = 786 Kg/m^3$, modulus of elasticity, E = 210 GPa and poison's ratio, $\nu = 0.3$.

To conduct the analysis, the boundary conditions are introduced by fixing the pinion and gear to the ground in all directions except the rotational degree of freedom about Z-axis. Both gear mates rotate about the centre points C_1 and C_2 as shown in Fig. 1. The gear is subjected to three load cases and the pinion is subjected to a single constant angular speed of 2090 RPM. Based on the data given in Table 1, the values of the three load cases (torques on the gear) are as follows: load case 1 is 18729.92 Nmm, load case 2 is 37459.84 Nmm and load case 3 is 56189.76 Nmm. The contact settings are set as stiffness = $5.010^5 N/mm$and the static and dynamic friction coefficients are set to 0.15 and 0.11 respectively. By setting the above boundary conditions and contact settings, the simulation is run for 0.2 s with 10^{-4} step size.

Table 1
Geometric properties and the operating conditions of the meshing gear pair [3].

Parameter	Value
Module, m	4 mm
Pressure angle, α	20 degree
Number of teeth, Z_P and Z_G	14 and 22
Face width, B	6 mm
Normal tooth load per unit face width, w	75.5 N/mm
Angular velocity of pinion, ω_P	2090 RPM

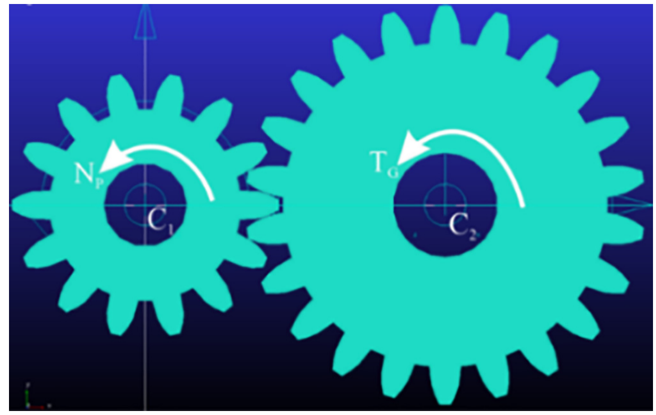

Fig. 1. Illustration of the definition of boundary conditions and loading for gear dynamics.

3. Discussion of results

The angular acceleration and contact forces of the gear with different backlash clearly show the effect of backlash on the dynamics of the gear pairs. The analysis contains six models of gear with 0 mm backlash to 1 mm backlash by 0.2 mm increments. The contact forces along the radial and tangential directions are determined and then the angular acceleration of the pinion and gear with different backlash are examined.

3.1. Contact force

During the process of designing the gear, the magnitude and direction of forces acting on the gear teeth is important. To make analysis of these forces, it is assumed that the force is acting upon the central part of the gear flank surfaces. The normal contact load can be resolved into radial and tangential load components, and each force component has its own function. The tangential load will produce the torque of the gears and the radial load is balanced by the reaction loads of the supports.

Fig. 2 shows the steady-state response history of the gear, under load case 1, for contact loads from mesh initiation to mesh termination. The dynamic radial force of the load (Fig. 2 (a)) varies from 105 N to 205 N. On the other hand, the designed tangential load for the same load case is 425.68 N, but Fig. 2(b) shows that the tangential load varies from 423 to 430.5 N, which can be due to the effect of the dynamics of backlash and discontinuities.

The maximum tangential load is found in a single tooth contact region which is equal to 430.5 N and it decreased to 423 N in a double tooth contact region. In addition, the tangential load shows discontinuity between the highest points of single tooth contacts of the mesh termination unlike a smooth increment along approaching mesh. The backlash affects the tangential load specifically around the recess region. The fluctuation of the radial load is greater than the fluctuation of tangential load, but the effect of backlash on the radial load is not significant like the tangential load. Load case 2 and 3 show similar trend (figures not included due to page limit). For load case 2, the maximum and minimum tangential loads are 862 N and 848.5 N respectively, while the radial loads vary between 220 N and 415 N. The radial and tangential loads for load case 3 vary between 1274 N and 1294 N.

In general, as the backlash increases, the contact loads along radial and tangential direction increase. In addition, the fluctuation

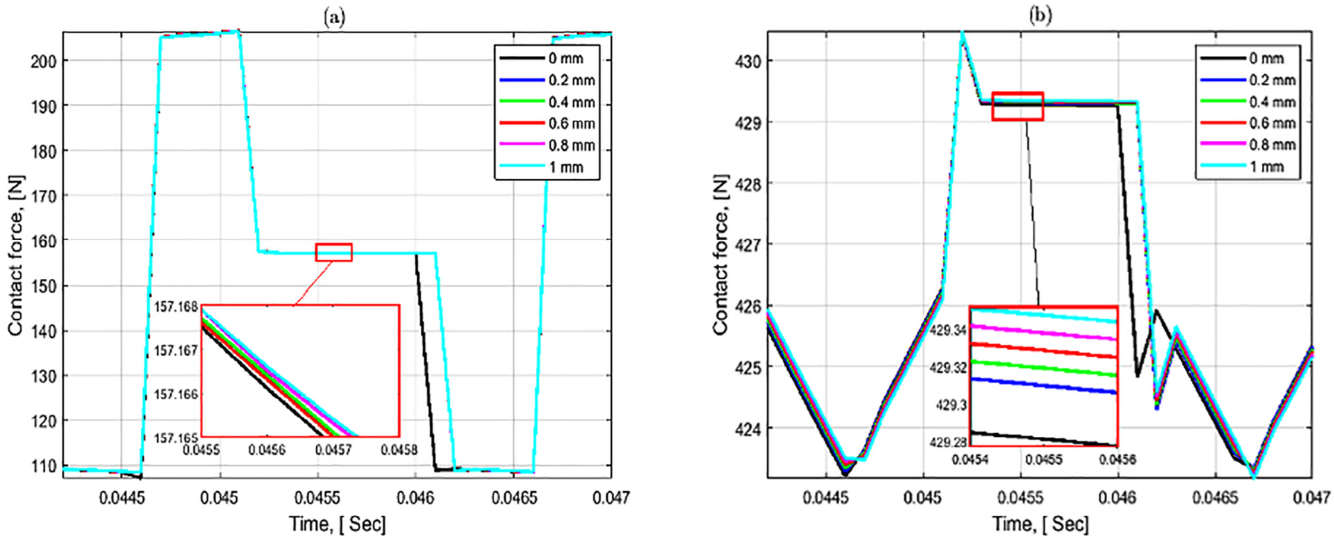

Fig. 2. Steady-state response history of the gear for load case 1 (a) Radial and (b) Tangential loads.

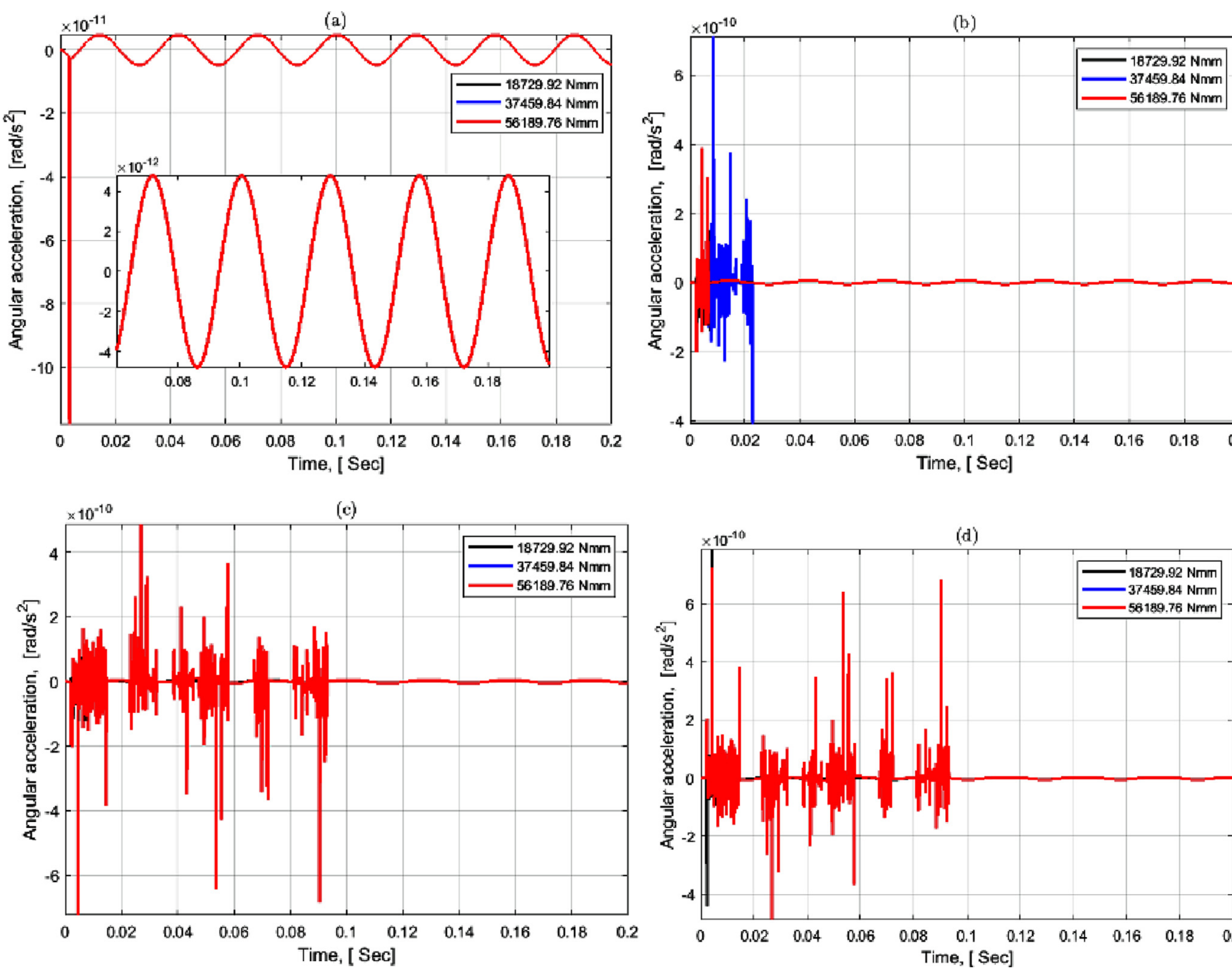

Fig. 3. Tangential angular acceleration of pinion with backlash of (a) 0 mm, (b) 0.2 mm, (c) 0.4 mm, (d) 0.6 mm,

load between the highest point of single tooth contact (HPSTC) and mesh termination becomes smooth as the load increases. It is also observed by applying higher loads that the tangential load of spur gear with different backlash shows the same curve history with the gear having no backlash. As the load of the gear increases, the lower and upper limit of the contact load variation is increased, the variation of the tangential load is 7.5, 13.5 and 20 N for load cases 1, 2 and 3 respectively.

3.2. Driving gear wheel (Pinion)

In the previous section, it was shown that the tangential and radial loads varied with time due to tooth separation and discontinuities, and these loads will produce acceleration in the tangential and radial directions respectively. In this section, these accelerations will be discussed in detail.

Figs. 3 and 4 show the tangential and radial acceleration of the pinion respectively. The pinion is subjected to a constant speed and it meshes with the gear which is subjected to a load. Though the pinion is subjected to a constant speed, it accelerates due to the dynamic effect of the gear. Fig. 3(a) shows that the gear with 0 mm backlash has a minimum angular tangential acceleration and smooth operation except for the first few mesh cycles. In addition, it shows that the radial acceleration of the driver gear or pinion is not affected by the load of the gear, which means it shows the same trend for all loading cases. As can be observed from the other figures (Fig. 3(b) - (d)), the tangential angular acceleration increases as the backlash increases. In addition, the tangential angular acceleration increases with the load. It is also observed that as the backlash increases, it takes a longer time to vibrate in steady-state conditions.

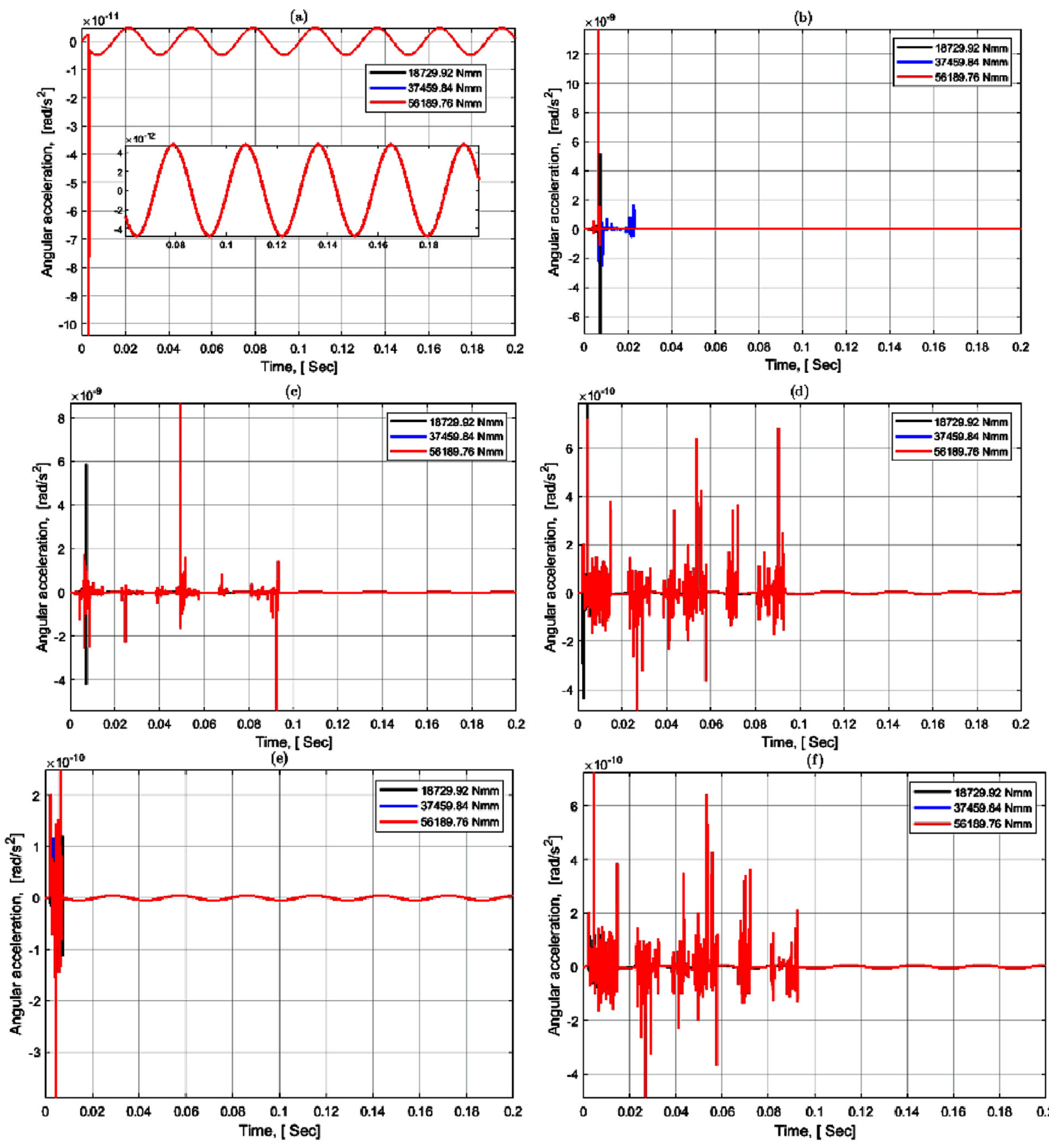

Fig. 4. Radial angular acceleration of pinion with a backlash of (a) 0 mm, (b) 0.2 mm, (c) 0.4 mm, (d) 0.6 mm, (e) 0.8 mm and (f) 1 mm.

Fig. 4(a) - (d) show the radial angular accelerations of the gear with different amount of backlash, which are results of a dynamic radial load. The gear without backlash has minimum radial angular acceleration. Like the tangential angular acceleration, the radial angular acceleration increases as the backlash increase except for 0.2 mm and 0.4 mm backlash gears as shown in Fig. 7(b) and (c). The radial acceleration is maximum for a backlash of 0.2 mm for the first few mesh cycles, and then shows a smooth response. Therefore, the pinion with 0.2 mm backlash shows a minimum tangential and radial angular acceleration.

It is observed for the gear without backlash that the radial acceleration of the pinion is not affected by the load of the gear as shown in Fig. 4(a), the same phenomenon observed for tangential angular acceleration of the pinion in Fig. 3(a). The gear with a backlash takes longer time relative to the gear without backlash to

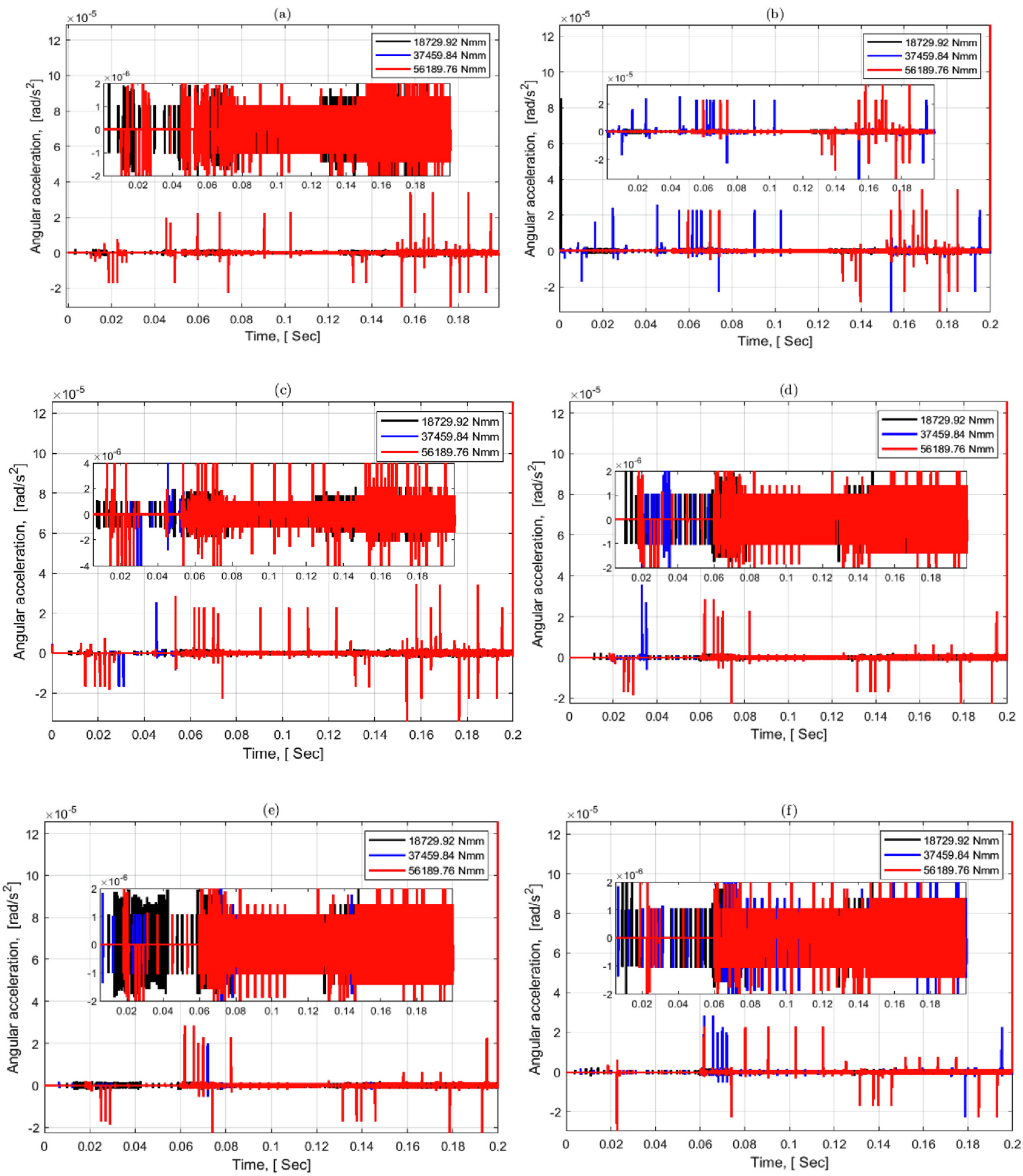

Fig. 5. Angular acceleration of pinion along Z-axis with: (a) 0 mm, (b) 0.2 mm, (c) 0.4 mm, (d) 0.6 mm, (e) 0.8 mm and (f) 1 mm backlash.

settle and give a steady-state response. The angular acceleration of the pinion is also shown in Fig. 8.

Angular acceleration along the rotational axis is produced mainly due to the geometrical effect. As depicted in Fig. 5 (a), the angular acceleration of the gear with 0 mm backlash$is2 \times 10^{-6} rad/s^2$. The pinion is subjected to the constant angular velocity, but there is an acceleration due to the gear. The pinion will accelerate and decelerate frequently due to the jamming effect, which means there is no free play. At 0.2 mm backlash (Fig. 5(b)) the angular acceleration has less frequency than the other responses of backlash values (Fig. 5(a), (c) - (f)). This may result due to the existence of free play, unlike the gear without

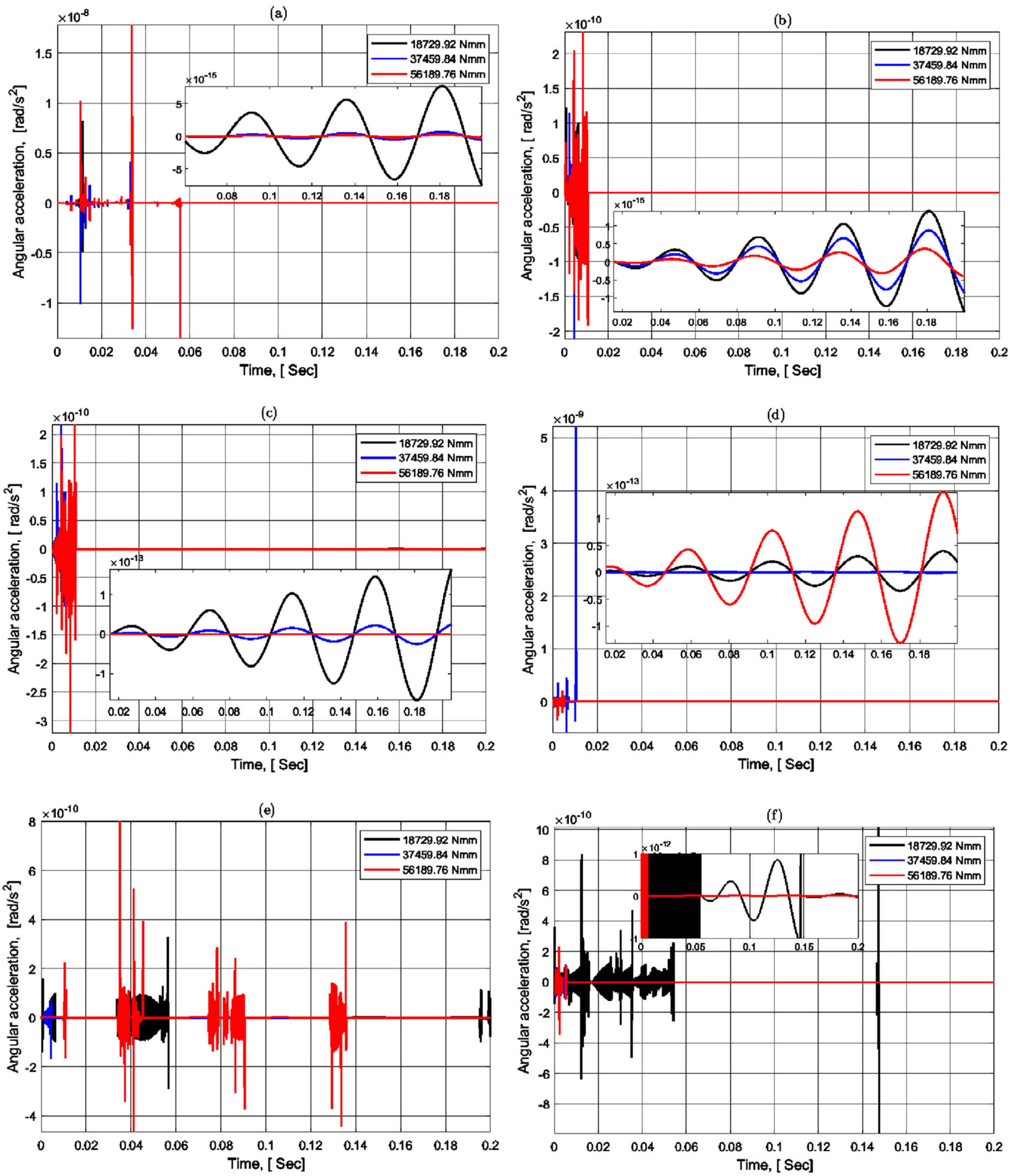

Fig. 6. Tangential angular acceleration of gear with: (a) 0 mm, (b) 0.2 mm, (c) 0.4 mm, (d) 0.6 mm, (e) 0.8 mm and (f) 1 mm backlash.

backlash and less gap that may cause an impact load as compared to the gear with a backlash between 0.4 and 1.0 mm backlash gears. In general, the angular acceleration of the pinion along the axis of rotation increases for 0 mm backlash as well as when the backlash increases. In addition, the radial angular acceleration is less than the tangential acceleration of the pinion.

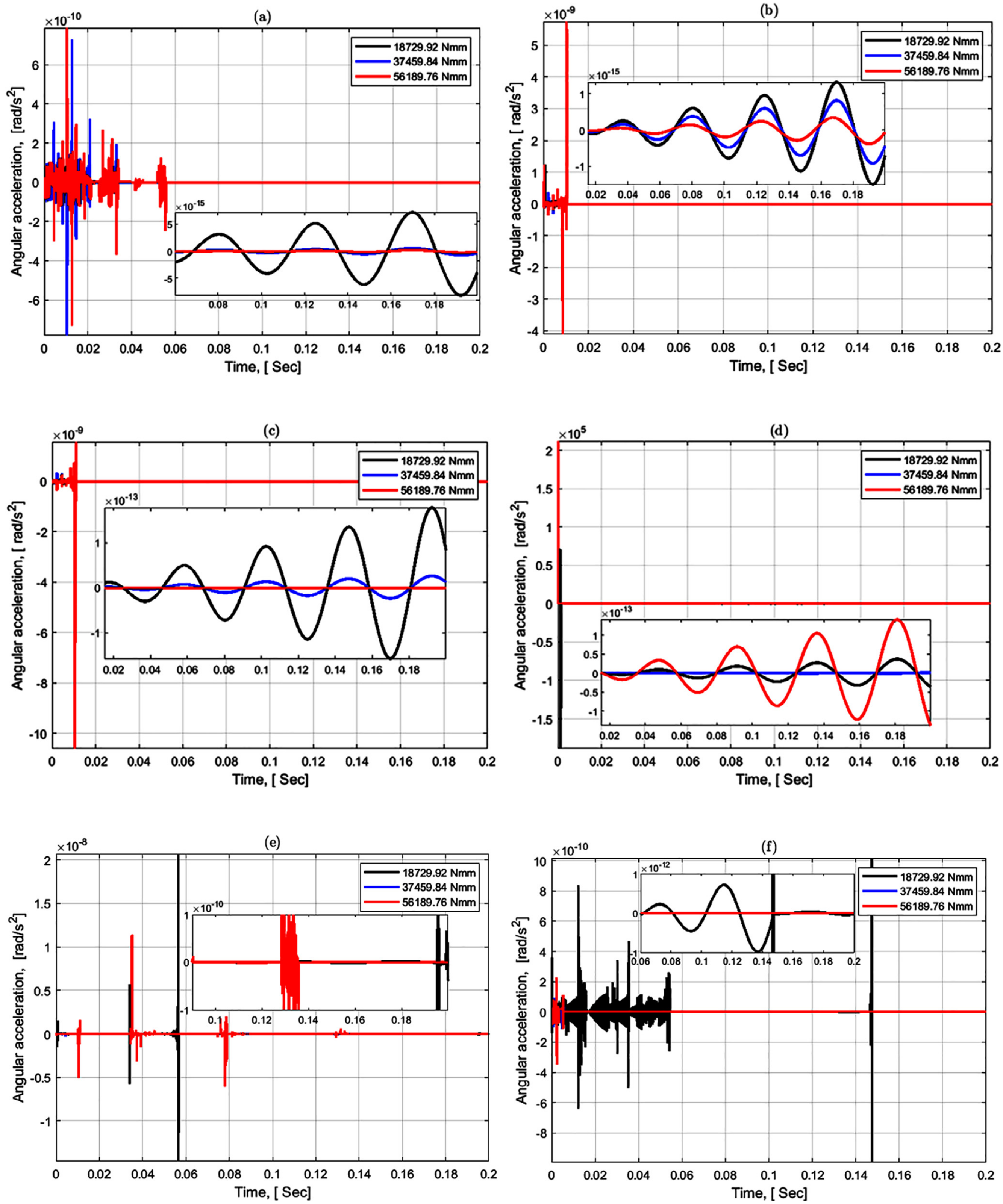

Fig. 7. Radial acceleration of gear along Y-axis with: (a) 0 mm, (b) 0.2 mm, (c) 0.4 mm, (d) 0.6 mm, (e) 0.8 mm and (f) 1 mm backlash.

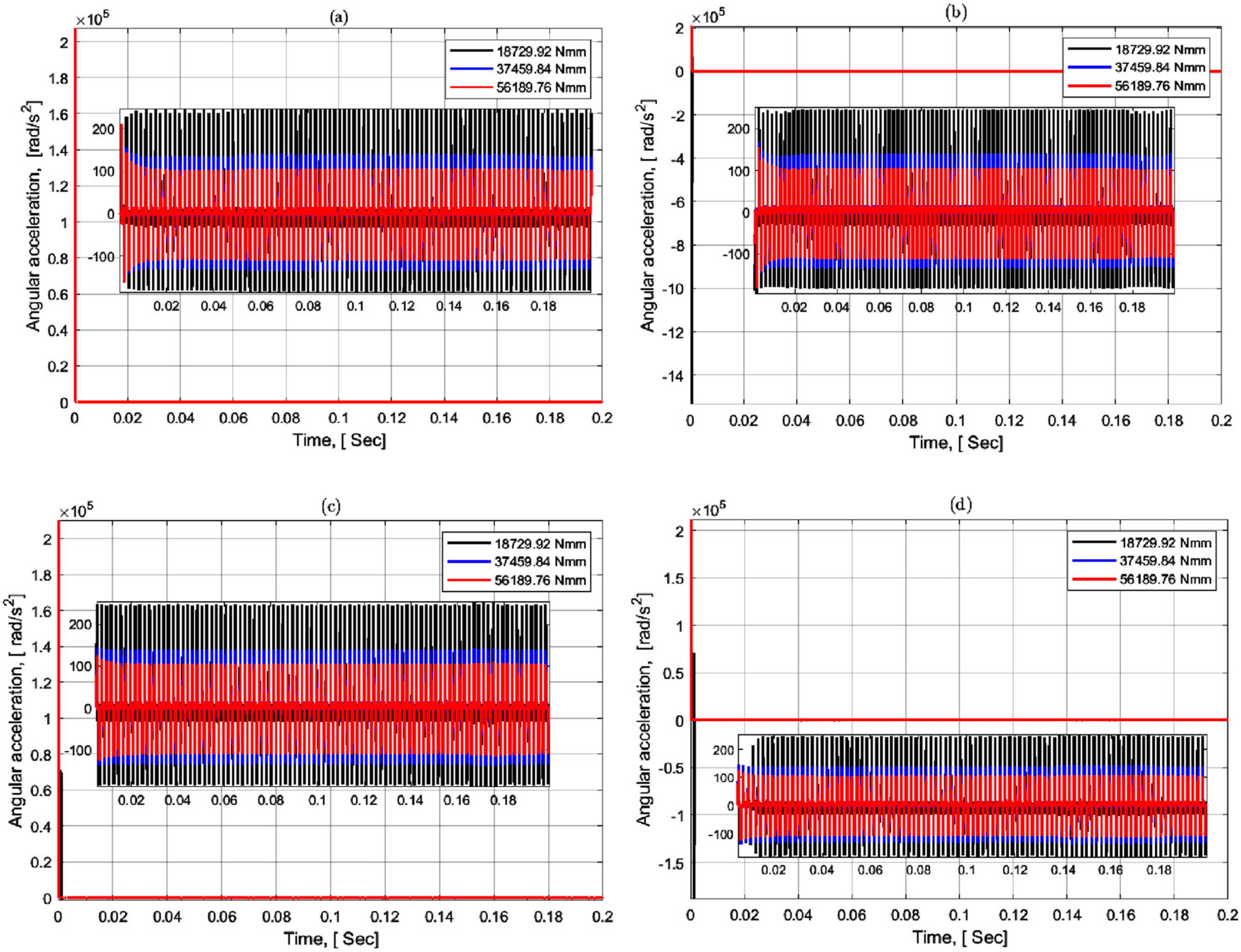

Fig. 8. Angular acceleration of gear with: (a) 0 mm, (b) 0.2 mm, (c) 0.4 mm, (d) 0.6 mm backlash.

3.3. Driven gear wheel (Gear)

The tangential and radial acceleration of the gear are shown in Figs. 6, 7 and 8 respectively while the figures for angular acceleration are partly omitted due to space limit. As given in Fig. 6(a), the gear with 0 mm backlash has maximum tangential angular acceleration ($1.8 \times 10^{-8} rad/s^2$) under load case 3. As the load increases, the time to reach a steady-state response will also be longer. The figures show that the settling time for the three load cases is10, 35 and 58 ms respectively.

The tangential angular acceleration is maximum for the first few mesh cycles when the gear drives without backlash, but the steady-state response of the gear with 0 mm backlash is minimum as compared to the gears with a backlash except for the gear with 0.2 mm backlash. The gear with a maximum load has minimum tangential acceleration. Fig. 6(b) shows the gear with 0.2 mm backlash, which indicates that there is acceleration at the instant of meshing, but it's for a very short time compared to the gear without backlash, which is attributed to a free play. The steady-state response of 0.2 mm backlash gear is less than that for gear with 0 mm backlash. The minimum load has a maximum tangential angular acceleration like the gear with 0 mm backlash.

In general, it is observed, for high backlash values (Fig. 6(c) – 6 (f)) that the tangential angular acceleration increases with increasing backlash. However, the maximum tangential acceleration is found on the gear without backlash for the first few mesh cycles and increases as the backlash increases. The optimum (minimum) tangential angular acceleration is found on the gear with 0.2 mm backlash. In some cases, such as those given in Fig. 6(d) and (e), the steady-state response shows maximum tangential angular acceleration at a low load value, which may be due to the fact that the resisting torque is easily affected by the driver acceleration. The more the load imposed on the gear, the less acceleration response is found. Therefore, higher loads have the effect of reducing vibration in cases where backlash exists in gear mates.

As depicted in Fig. 7(a), the radial angular acceleration with a minimum design load (load case 1) exceeds the gear with a larger designed load (load case 2 and 3) and the gear with 0 mm backlash has maximum tangential acceleration than the others. Like the tangential angular acceleration, the radial angular acceleration shows the same phenomena, the optimum (minimum) radial angular acceleration is found on the gear with 0.2 mm backlash. Fig. 7(d) shows the response of the gear with 0.6 mm backlash, where load case 2 exhibits minimum radial angular acceleration than load case 2 and 3. This is an unexpected result similar to those shown previously on the tangential angular acceleration of the gear with 0.6 mm backlash. The gear with 1 mm backlash accelerates frequently up to 55 ms with a load case 1, but for load case 2 and 3 the radial acceleration settles in few seconds (Fig. 7(f)).

In general, the acceleration of the gear is the measure of vibration of the gear. A further investigation of the angular acceleration for 3 different load cases along the axis of rotations (Fig. 8) indicated that the angular acceleration of the gear decreases when the load increases. The angular accelerations oscillated depending on the load cases. The oscillations have been between -250 and $250 rad/s$ for the first load case, between -150 and $150 rad/s$ for the second load case and between -100 and $100 rad/s$ for the last load case. The angular acceleration of the gear takes very few seconds to run within a steady state fashion. The gear accelerates around $2.1 \times 10^5 rad/s^2$ for load case 3

4. Conclusions

In this paper, the effect of backlash on the dynamics of the spur gear has been discussed with respect to contact forces and angular acceleration. The vibration of the gear is affected by the non-linear parameters of the gear, such as geometrical (backlash) and material non-linearity. The simulated results using ADAMS show that the tangential and radial angular acceleration of the gear increase when the value of the backlash increases, and the optimum (minimum) acceleration is found on the gear with 0.2 mm backlash. In addition, the angular acceleration (tangential and radial) of the gear decrease as the load increases except for the gear with 0.6 mm and 0.8 mm backlash. Thus, this paper demonstrated the dynamics of the selected spur gear with different amounts of backlash, based on the simulation results, and the following conclusions are drawn.

- o The contact force components of the gear along the tangential and radial direction increased slightly as the backlash increased.
- o The acceleration of the mating gear pairs are affected by the backlash, for 0 mm backlash the acceleration of the gear is greater than the gears with a backlash, and when the gear backlash varies from 0.2 mm to 1 mm backlash the angular acceleration of the gear is increased with some unexpected simulation results. The tangential vibration has not shown a significant variation for the gear with different amounts backlash, while the lateral and transversal angular acceleration shows the variation with backlash.
- o The angular acceleration of the gear decreases as the load imposed on the gear increases. This shows that, though two gears have the size of backlash, their vibration level is a function of the size of the imposed load.

The continuing work of the study reported in this paper will be devising a mechanism of validating the simulation results using experimental approach by accounting for the complexities in the number of gears pairs to be availed and design and manufacturing of proper tooth profiles.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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