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Uncertainty analysis of dynamic behavior of one stage gear system with eccentricity defect

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Abstract. This paper, we propose a method for considering uncertainties based on the projection on polynomial chaos. The proposed approach is used to study the nonlinear dynamic response of one stage gear transmission system with uncertainty associated to eccentricity defect and this uncertainty must be considered in the analysis of the dynamic analysis of this system. The simulation results are obtained by the polynomial chaos approach for dynamic analysis under uncertainty. The proposed method is an efficient probabilistic tool for uncertainty propagation. It was found to be an interesting alternative to the parametric studies. The polynomial chaos results are compared with Monte Carlo simulations.

Keywords: eccentricity defect; gear system; Monte Carlo simulation; polynomial chaos method; uncertainty

1. Introduction

The gearing is the best solution to transmit rotational motions and couple which has been offered numerous advantages (Guerine *et al.* 2015a, Guerine *et al.* 2024, Karmi *et al.* 2024): it ensures a mechanical reliability. Furthermore, its mechanical efficiency is of the order of 0.96 to 0.99. But today, several applications inquire for the gearing transmissions to be more and more reliable, light and having long useful life that requires the control of the vibratory behavior of these gearings (Guerine *et al.* 2015b, Guerine *et al.* 2016a, Guerine and El Hami 2022, Beyaoui *et al.* 2016).

Several parametric studies have shown the great sensitivity of the dynamic behavior of gear systems (Walha *et al.* 2009). However, these parameters admit strong dispersions. Therefore, it becomes necessary to consider these uncertainties to ensure the robustness of the analysis (Guerine *et al.* 2016b, Begg *et al.* 2000, Guerine *et al.* 2017, Dong *et al.* 2023, Georgoussis and Mamou 2020, Abegaz 2022, Ouazir 2022). Zhao *et al.* studied the effect of the geometric eccentricity on the dynamic behaviors of helical gear systems in both the quasi-static and dynamic analysis (Zhao *et al.* 2020). Walha *et al.* studied the nonlinear dynamic behavior of an automotive clutch coupled with a helical two stage gear system with eccentricity defect (Walha *et al.* 2011). They modeled the eccentricity defect located on the gear and the flywheel of the clutch. In addition, A general dynamic model for the cylindrical geared rotor system with local tooth profile errors and global mounting errors have been developed in (Yu *et al.* 2017). In this article, they studied the dynamic coupling behaviour of the transverse and rotational motions of gears subjected to gear eccentricities.

Many studies used an uncertainty method to analyzed the dynamic behavior of gear transmission system. For instance, Bel Mabrouk *et al.* studied the dynamic response analysis of Vertical Axis Wind Turbine geared transmission system with uncertainty (Bel Mabrouk *et al.* 2017). They used a new approach to determine the dynamic behavior of a bevel gear system with uncertainty associated to the performance coefficient of the input aerodynamic torque. The dynamic response analysis on torsional vibrations of wind turbine geared transmission system with uncertainty have been discussed in (Wei *et al.* 2015). In this article, they used the Chebyshev interval method to study the dynamic responses of a geared transmission system with uncertain parameters including the mesh stiffness, the transmission error, the mesh damping, the shaft damping, the moment of inertia of the input blades and the torsional stiffness of the driving coupling shaft.

Mélot *et al.* proposed a robust design of vibro-impacting geared systems with uncertain tooth profile modifications via bifurcation tracking (Mélot *et al.* 2023). They studied the influence of uncertain tooth profile modifications on the nonlinear dynamic response of a spur gear pair induced by a backlash nonlinearity. They proposed a methodology for a fast and reliable estimation of the tooth profile modification that minimizes the amplitude-jump instability by defining two criteria using the results of the bifurcation tracking algorithm. The stochastic dynamical response of a gear pair under filtered noise excitation have been discussed in (Hasnijeh *et al.* 2021). They studied the effect of the noise spectrum on the probabilistic response for different loading cases.

In addition, other studies used a Polynomial Chaos method to study the dynamic response of mechanical system. Polynomial chaos method is a popular surrogate modeling approach employed in uncertainty quantification for a variety of engineering problems (Guerine *et al.*

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2018a). The capabilities of polynomial chaos expansion have been tested in numerous applications, such as treating uncertainties in mechanical dynamic systems. But the method cannot solve some problems if the mechanical system is complex.

A Polynomial chaos expansion approximation for dimension-reduction model-based reliability analysis method and application to industrial robots have been proposed in (Wu *et al.* 2023). In this article, they proposed a new PCE-based surrogate-assisted meth to study the interacting variables from original high-dimensional input random variables by contribution-degree analysis. Guerine *et al.* studied the dynamic response of a Spur gear system with uncertain friction coefficient (Guerine *et al.* 2018b). They proposed a method for considering uncertainties based on the projection on polynomial chaos to determine the dynamic response of a spur gear system with uncertainty associated to friction coefficient on the teeth contact.

Many studies used a method taking into account the uncertainties to ensure the robustness of the analysis (Hu and Qui 2010, Wu *et al.* 2023) and to study the reliability for vibration structures taking into account the uncertainties (Boudhraa *et al.* 2021, Arian and Taghvaei 2021, Snoun *et al.* 2020, Wang *et al.* 2022, Liu *et al.* 2022, Lindsley and Beran 2005, Li and Ghanem 1998, Yu *et al.* 2024, Blanchard *et al.* 2009, Liu *et al.* 1986, Muscolino *et al.* 1999).

The main originality of the present paper is that the uncertainty of dynamic response of one stage gear transmission system with eccentricity defect is considered. The main objective is to investigate of the capabilities of the proposed method to determine the dynamic response of two stage gear transmission system subject to uncertain gear parameter. So, an eight degree of freedom system modelling the dynamic response of gear transmission system is considered. The modelling of gear transmission system is presented in Section 2. In the next section, the theoretical basis of the polynomial chaos is presented. In Section 4, the equations of motion for the eight degrees of freedom are presented. Numerical results are presented in Section 5. Finally, in Section 6, to conclude, some comments are made based on the study carried out in this paper.

2. Modelling of one stage gear transmission system

The global dynamic model of the one stage gear system in 3D is shown in Fig. 1. This model is composed of two blocks (j=1 to 2). Every block (j) is supported by flexible bearing which the bending stiffness is k_j^x and the traction-compression stiffness is k_j^y.

The wheels (11) and (22) characterize respectively the motor side and the receiving side. The shafts (j) admit some torsional stiffness k_j^θ.

Angular displacements of every wheel are noticed by $\theta_{(i,j)}$ with the indices j=1 to 2 designates the number of the block, and i=1 to 2 designate the two wheels of each block. Moreover, the linear displacements of the bearing noted by x_j and y_j are measured in the plan which is orthogonal to the wheels axis of rotation.

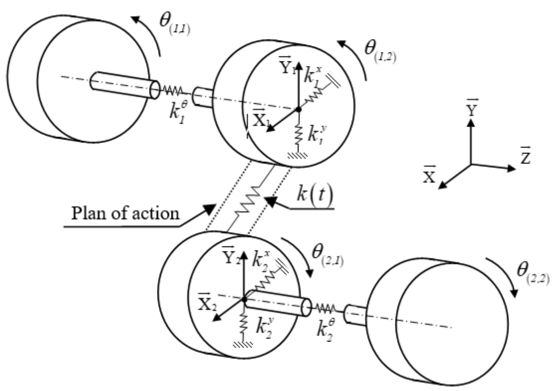

Fig. 1 Global dynamic model of the one stage gear system

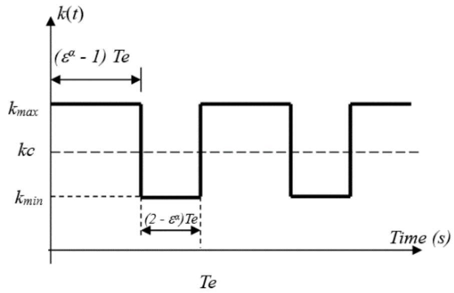

Fig. 2 Modelling of the mesh stiffness variation

In this study, we modelled the gear mesh stiffness variation $k(t)$ by a square wave form (Fig. 2). The gear mesh stiffness variation can be decomposed in two components: an average component noted by kc, and a time variant one noted by $kv(t)$ (Walha *et al.* 2009).

The extreme values of the mesh stiffness variation are defined by

$$k\frac{kc}{2\varepsilon^{\alpha}}_{min} \text{ and } k\frac{2-\varepsilon^{\alpha}}{\varepsilon^{\alpha}-1}min_{max} \quad (1)$$

ε^α and Te represent respectively the contact ratio and mesh period corresponding to the two gear meshes contacts.

3. Polynomial Chaos method

In this section, we propose a new methodological method based on the projection on polynomial chaos. This method consists in projecting the stochastic desired solutions on a basis of orthogonal polynomials in which the variables are Gaussian orthonormal. The properties of the base polynomial are used to generate a linear system of equations by means of projection. The resolution of this system led to an expansion of the solution on the polynomial basis, which can be used to calculate the moments of the random solution. With this method, we can easily calculate the dynamic response of a mechanical system.

Let us consider a multi-degrees of freedom linear system with mass and stiffness matrices $[M_T]$ and $[K_T]$ respectively. The equations of motion describing the forced vibration of a linear system are

$$[M_T]\{\ddot{u}_T\} + [K_T]\{u_T\} = \{f_T\} \quad (2)$$

Where $\{u_T\}$ is the nodal displacement vector and $\{f_T\}$ is the external excitation.

The chaotic polynomials ψ_m corresponding to the multidimensional Hermite polynomials obtained by the Eq. (3)

$$\psi_{\mathrm{m}}(\alpha_1,\ldots,\alpha_{\mathrm{P}}) = (-1)^{\mathrm{P}} e^{\left(\frac{1}{2}{}^{T}\{\alpha\}\{\alpha\}\right)} \frac{\partial^{\mathrm{P}} e^{\left(-\frac{1}{2}{}^{T}\{\alpha\}\{\alpha\}\right)}}{\partial\alpha_1 \ldots \partial\alpha_{\mathrm{P}}} \tag{3}$$

Where $\{\alpha\}$ is the vector grouping the random variables and P is the number of random variables.

$$T\{\alpha\} = \langle \alpha_1 \ldots \alpha_P \rangle \tag{4}$$

The random matrices mass and stiffness $[M_T]$ and $[K_T]$ of the mechanical system can be written as

$$[M_T] = [M_T]_0 + [\tilde{M}_T] \tag{5}$$

$$[K_T] = [K_T]_0 + [\tilde{K}_T] \tag{6}$$

The matrices $[M_T]_0$ and $[K_T]_0$ are deterministic matrices, the matrices $[\tilde{M}_T]$ and $[\tilde{K}_T]$ correspond to the random part of the mass and stiffness matrices.

$[\tilde{M}_T]$ and $[\tilde{K}_T]$ are rewritten from an expression of type Karhunen-Loeve in the following form

$$[\tilde{M}_T] = \sum_{p=1}^{P} [M_T] \quad \alpha_p \tag{7}$$

$$[\tilde{K}_T] = \sum_{p=1}^{P} [K_T] \quad \alpha_p \tag{8}$$

Where α_p are independent Gaussian centered reduced which may correspond to the first polynomial ψ_p, while the matrices $[M_T]$ and $[K_T]$ are deterministic.

We set $\alpha_0 = 1$, we can write then

$$[M_T] = [M_T]_0 \,.\, \sum_{p=0}^{P} \alpha_p \tag{9}$$

$$[K_T] = [K_T]_0 . \sum_{p=0}^{P} \alpha_p \tag{10}$$

In the same way, we can write for $\{f_T\}$

$$\{f_T\} = \{f_T\}_0 \sum_{p=0}^{P} \alpha_p \tag{11}$$

The dynamic response is obtained by solving the following equation knowing that the initial conditions are predefined

$$[K_{eq}]\{u \quad (t+\Delta t)\} = \{F_{eq}\} \tag{12}$$

Where

$$[K_{eq}] = [K_T] + a_0[M_T] \tag{13}$$

$$\{F_{eq}\} = \{\mathrm{f_T}(t+\Delta t)\} + [\mathrm{M_T}](a_0\{u \quad (t)\} + a_1\{\dot{u} \quad (t)\} + a_2\{\ddot{u} \quad (t)\}) \tag{14}$$

Where

$$a_0 = \frac{1}{A\Delta t^2},\ a_1 = \frac{B}{A\Delta t} \text{ and } a_2 = \frac{1}{A\Delta t} \tag{15}$$

A and B are the parameters of Newmark.

$\{u \quad (t+\Delta t)\}$ is decomposed on polynomials to P Gaussian random variables orthnormales

$$\{u \quad (t+\Delta t)\} = \sum_{n=0}^{N} (\{u \quad (t+\Delta t)\})_n \psi_n(\{\alpha_i\}_{i=1}^{P}) \tag{16}$$

Where N is the polynomial chaos order.

$[K_{eq}]$ and $\{F_{eq}\}$ are written in the following form

$$[K_{eq}] = [K_T]_0 . \sum_{p=0}^{P} \alpha_p + \alpha_0 . [M_T]_0 \sum_{p=0}^{P} \alpha_p \tag{17}$$

$$\{F_{eq}\} = \sum_{p=0}^{P} (\{f_T(t+\Delta t)\})\, \alpha_p + \sum_{p=0}^{P} [\mathrm{M_T}] \quad \alpha_p (a_0(\{u_T(t)\})_0 + a_1(\{\dot{u}_T(t)\})_0 + a_2(\{\ddot{u}_T(t)\})_0) \tag{18}$$

Substituting Eqs. (16), (17) and (18) into Eq. (12) and forcing the residual to be orthogonal to the space spanned by the polynomial chaos ψ_{m} yield the following system of linear equation

$$\sum_{p=0}^{P} \sum_{n=0}^{N} [K_{eq2}] \quad \{u \quad \}_n \langle \alpha_p \quad \psi_n \quad \psi_m \rangle = \sum_{p=0}^{P} \{F_{eq2}\}_p \langle \alpha_p \quad \psi_m \rangle \qquad m = 0,\ 1,\ .\ .\ .\ , N \tag{19}$$

Where N is the order of Polynomial Chaos.

Where $\langle .. \rangle$ denotes the inner product defined by the mathematical expectation operator

This algebraic equation can be rewritten in a more compact matrix form as

$$\begin{bmatrix} [D]^{(00)} & \cdots & [D]^{(0N)} \\ \vdots & [D]^{(ij)} & \vdots \\ [D]^{(N0)} & \cdots & [D]^{(NN)} \end{bmatrix} \begin{Bmatrix} (\{u \quad (t+\Delta t)\})_0 \\ \vdots \\ (\{u \quad (t+\Delta t)\})_j \\ \vdots \\ (\{u \quad (t+\Delta t)\})_N \end{Bmatrix} = \begin{Bmatrix} \{f\}^{(0)} \\ \vdots \\ \{f\}^{(j)} \\ \vdots \\ \{f\}^{(N)} \end{Bmatrix} \tag{20}$$

Where

$$[D]^{(ij)} = \sum_{p=0}^{P} [K_{eq2}] \quad \langle \alpha_p \quad \psi_i \quad \psi_j \rangle \tag{21}$$

$$\{f\}^{(j)} = \sum_{p=0}^{P} \{F_{eq2}\} \quad \langle \alpha_p \quad \psi_j \rangle \tag{22}$$

After resolution of the algebraic system (20), the mean values and the variances of the dynamic response are given by the following relationships

$$E[\{u \quad \}] = \frac{1}{N} \sum_{n=0}^{N} (\{u \quad (t+\Delta t)\})_n \tag{23}$$

$$Var[\{u \quad \}] = \sum_{n=0}^{N} ((\{u \quad (t+\Delta t)\})_n)^2 (\psi_j)^2 \tag{24}$$

4. Equation of motion

The equation of motion describing the dynamic behavior of our system (Fig. 1) is obtained by applying Lagrange formulation and is given by

$$m\,\ddot{x}_1 + k_1^x\, x_1 + \sin(\alpha) \ k(t) \ \langle L \quad \rangle \{u(t)\} = 0 \tag{25}$$

$$m\,\ddot{y}_1 + k_1^y\, y_1 + \cos(\alpha) \ k(t) \ \langle L \quad \rangle \{u(t)\} = 0 \tag{26}$$

Table 1 System parameters

Material: 42CrMo4	ρ=7860 Kg/m^3
Motor torque	Cm=200 N.m
Bearing stiffness	k_j^x=10^7 N/m k_i^y=10^7 N/m
Torsional stiffness of the shaft	k_j^θ=10^5 Nm/rad
Number of teeth	Z(12)=40, Z(21)=50
Module of teeth	Module=4 mm
Contact ratio	ε^α=1.7341
The pressure angle	α=20°

$$m\,\ddot{x}_2 + k_2^x\,x_2 - \sin(\alpha)\;k(t)\;\langle L\ \ \rangle\{u(t)\} = 0 \tag{27}$$

$$m\,\ddot{y}_2 + k_2^y\,y_2 - \cos(\alpha)\;k(t)\;\langle L\ \ \rangle\{u(t)\} = 0 \tag{28}$$

$$I\ddot{\theta}_{(1,1)} + k_1^\theta\big(\theta_{(1,1)} - \theta_{(1,2)}\big) = Cm \tag{29}$$

$$I\;\;\ddot{\theta}_{(1,2)} - k_1^\theta\;\big(\theta_{(1,1)} - \theta_{(1,2)}\big)\;+\; r_{(1,2)}^b\;k(t)\;\langle L\ \ \rangle\;\{u(t)\} = 0 \tag{30}$$

$$I\;\;\ddot{\theta}_{(2,1)} + k_2^\theta\;\big(\theta_{(2,1)} - \theta_{(2,2)}\big)\;-\; r_{(2,1)}^b\;k(t)\;\langle L\ \ \rangle\;\{u(t)\} = 0 \tag{31}$$

$$I\ddot{\theta}_{(2,2)} + k_2^\theta\big(\theta_{(2,1)} - \theta_{(2,2)}\big) = 0 \tag{32}$$

Where I is the moment of inertia of the wheels.

Where $\langle L\ \ \rangle$ is defined by

$$\langle L\ \ \rangle = \big[sin(\alpha)\quad -sin(\alpha)\quad \cos(\alpha)\quad -\cos(\alpha)\quad 0\quad r_{(1,2)}^b\quad -r_{(2,1)}^b\quad 0\big] \tag{33}$$

$r_{(1,2)}^b$, $r_{(2,1)}^b$ represent the base gears radius. α is the pressure angle.

$\{u(t)\}$ is the vector of the model generalized coordinates, it is in the form

$$\{u(t)\} = \big[x_1\quad y_1\quad x_2\quad y_2\quad \theta_{(1,1)}\quad \theta_{(1,2)}\quad \theta_{(2,1)}\quad \theta_{(2,2)}\big]^T \tag{34}$$

5. Numerical simulation

The technological and dimensional features of the one-stage gear transmission system are summarized in the Table 1.

5.1 Dynamic response with eccentricity defect

An eccentricity is theoretically the distance between the geometric and rotating axis of the gear. In this paper, Fig. 4 represents the case of an eccentricity on the gear (12) belonging to the first train of the two-stage gear system. O_{12} and G_{12} represent respectively the rotational and geometric centers of the gear (12). The eccentricity defect is defined by the parameter e_{12}, which represents the distance between the axis, and by a phase λ_{12} to specify the initial position.

An eccentricity defect causes additively teeth deflection on their own line of action. The deflection $\delta_1(t)$ is then added with a transmission error $e_{12}(t)$ definite by

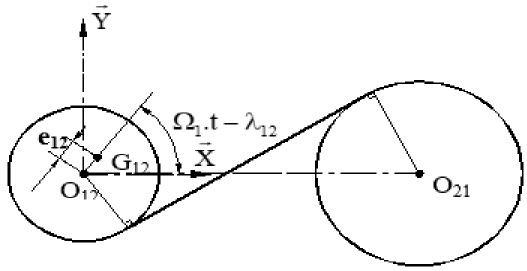

Fig. 4 Eccentricity defect

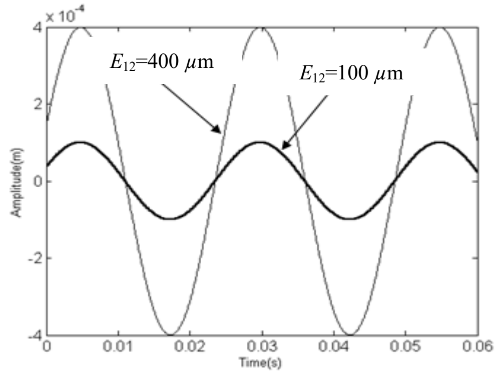

Fig. 5 Transmission error signal of the eccentricity defect

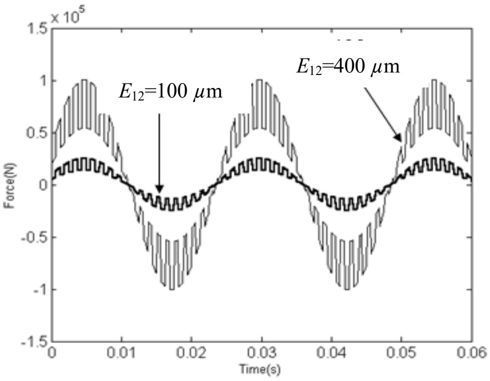

Fig. 6 Time response of the eccentricity defect external force $\langle Fecc(t)\rangle$

$$e_{12}(t)=E_{12}.\ \sin(2.\pi.f_e.t\text{-}\ \lambda_{12}) \tag{35}$$

f_e represents the frequency of rotation of the gear (12). Fig. 5 shows the transmission error signal for two eccentricity amplitude defects. To analyze the really consequences of the defect on the dynamic behavior of the model, we study two values of eccentricity: E_{12}=100 μm and E_{12}=400 μm.

We can assimilate the effect of the eccentricity defect by an external force

$$\langle Fecc(t)\rangle = k(t).\ e_{12}(t)\ \langle L\ \ \rangle \tag{36}$$

Fig. 6 shows the time response of the eccentricity defect external force.

To analyze the really consequences of the defect on the dynamic response of the model, we study two values of eccentricity: E_{12}=100 μm and E_{12}=300 μm.

Fig. 7 shows the time dynamic displacement of the first and second bearings following x direction. It appears a new

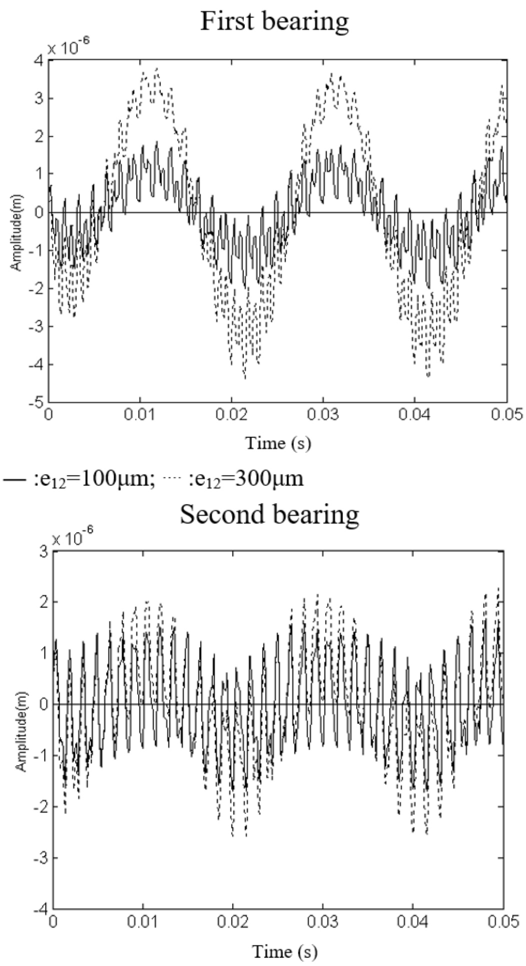

Fig. 7 Time dynamic displacement of the first and second bearings following *x* direction

period in the signal. This periodicity will be detected on all dynamic responses of all model degrees of freedom.

5.2 Dynamic response with uncertain eccentricity defect

In this section numerical results are presented for the new method formulations derived in the Section 3. The polynomial chaos (PC) results are compared with Monte Carlo (MC) simulations with 100000 simulations.

The eccentricity defect is defined by

$$e_{12} = e_{12_0} + \sigma_{e_{12}}\xi \qquad (37)$$

Where ξ is a zero mean value Gaussian random variable, e_{12_0} is the mean value and $\sigma_{e_{12}}$ is the standard deviation of this parameter.

The mean value of the dynamic component of the linear displacement of the first bearing in two directions *x* and *y* have been calculated by the polynomial chaos method. The obtained results are compared with those given by the Monte Carlo simulations with 100000 simulations. The results are plotted in Figs. 8 and 9.

These figures show that the obtained solutions oscillate around the Monte Carlo simulations reference solution. It can be seen that for small standard deviation $\sigma_{e_{12}}$=1%, the

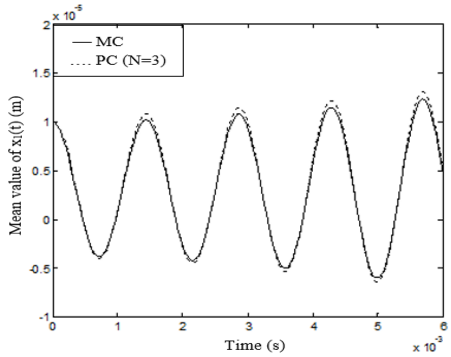

Fig. 8.1 Mean value of $x_1(t)$ for $\sigma_{e_{12}}$=1%

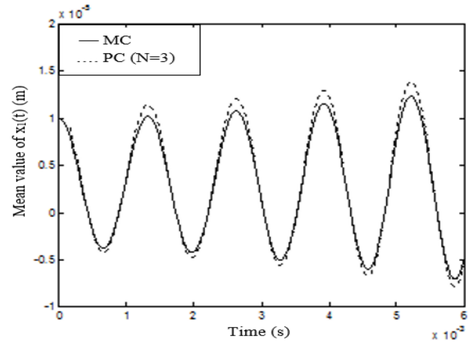

Fig. 8.2 Mean value of $x_1(t)$for $\sigma_{e_{12}}$=3%

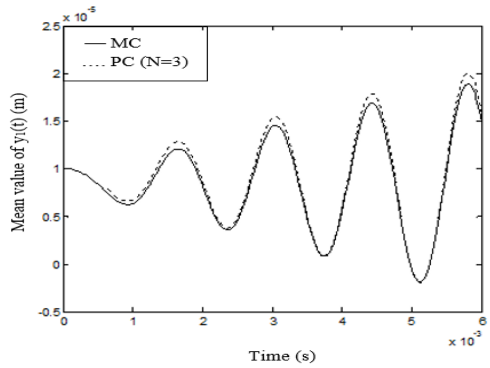

Fig. 9.1 Mean value of $y_1(t)$ for $\sigma_{e_{12}}$=1%

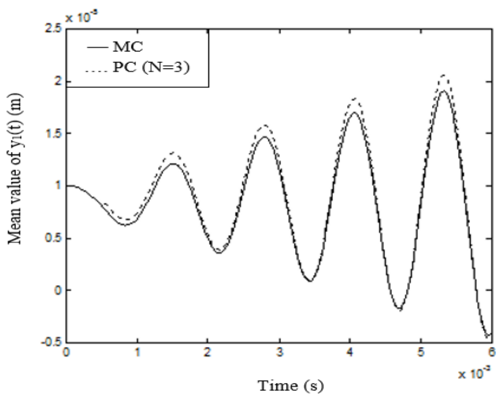

Fig. 9.2 Mean value of $y_1(t)$ for $\sigma_{e_{12}}$=3%

polynomial chaos solutions in third order polynomial (*N*=3) provides a very good accuracy as compared with the Monte Carlo simulations. When the standard deviation of the

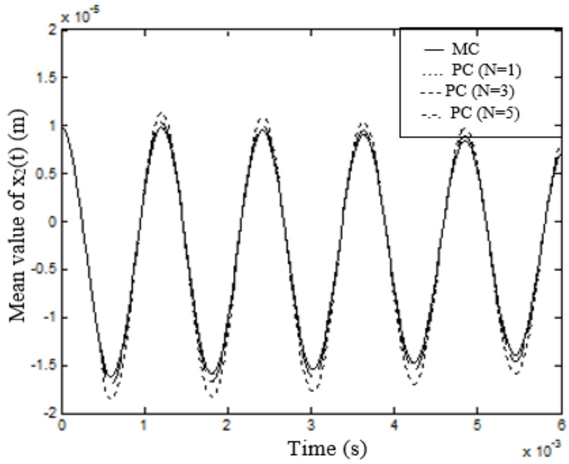

Fig. 10.1 Mean value of $x_2(t)$ for $\sigma_{e_{12}}$=7%

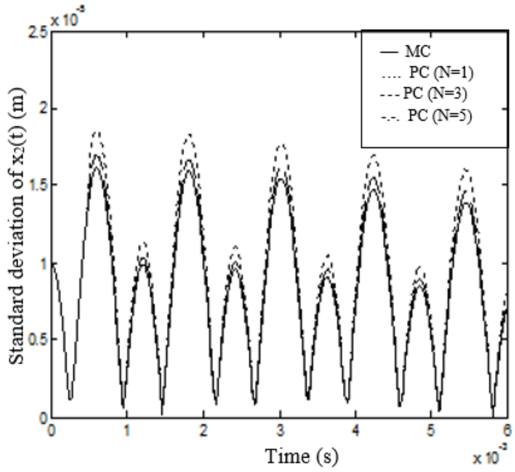

Fig. 10.2 Standard deviation of $x_2(t)$ for $\sigma_{e_{12}}$=7%

uncertain parameter increases, the error increases. Because the amplitude of the displacement of the bearings increases. So, the difference between the Monte Carlo method results and the Polynomial Chaos method results increase.

The mean value and standard deviation of the dynamic component of the linear displacement of the second bearing in direction x obtained with different orders of polynomial chaos N=1, N=3 and N=5 are presented in Fig. 10 for $\sigma_{e_{12}}$ =7% in order to check the capabilities of the polynomial chaos approach in the analysis of the dynamic behavior of spur gear system.

The polynomial chaos results are compared with Monte Carlo simulation with 100000 simulations. It is evident from these figures that N=1 case clearly does not have enough chaos terms to represent the output. As N increases, the results seem to become better, and with N=5, the dynamic response of the linear displacement of the second bearing with polynomial chaos values almost exactly match with the Monte Carlo simulation results. The uncertainty of the radius parameter affects the amplitude of the system responses. It can be noted that the amplitudes of the mean values and the standard deviation are approximated more accurately with N=5 than the first and the third order polynomial.

6. Conclusions

An approach based on the polynomial chaos method has been proposed to study the dynamic response of one stage gear transmission system with eccentricity defect. A complete study of the dynamic analysis has been carried out for an eight degree of freedom model describing one stage gear transmission system characterised by an uncertain eccentricity defect. The polynomial chaos method has been used to determine the dynamic response of gear transmission system. The advantage of the polynomial chaos method is that only a small number of simulations are required to extract the dynamic response of gear transmission system. The polynomial chaos method has demonstrated excellent computation efficiency compared to the Monte Carlo method. This efficiency is more clearly seen in the case for small standard deviation $\sigma_{e_{12}}$=1% that the polynomial chaos results provides a very good accuracy as compared with the Monte Carlo method. The main results of the present study show that the polynomial chaos may be an efficient tool to consider the dispersions of eccentricity defect in the dynamic behavior study of gear transmission system.

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