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Experimental Findings on Reduction of Vibrations in Transmission System by Phasing in Spur Gear Coupling

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ABSTRACT

Spur gear coupling is used to transfer torque and power from input shaft to output shaft. It is used in many applications such as automobile, aerospace, marine, industries etc. Transmission system consisting of gears and gear coupling are the major sources of vibrations. To reduce gear vibrations, method of phasing is used in this study. In this method input and output spur gear coupling sleeve are phased, in order to increase the number of teeth pairs in contact. Due to increased number of teeth pair in contact, mesh stiffness variation reduces. Mesh stiffness variation is a main contributor of production of internal excitation. It results in reduced amplitude of vibrations. The analytical study proved that maximum and minimum values of mesh stiffness for phased gear coupling are lower than non-phased gear coupling. This paper mainly focuses on experimentation of the proposed method. For this purpose experimental setup has been prepared and trials are taken to find the results with and without phasing of spur gear coupling. Experimental results show reduction in vibration parameter due to phasing arrangement of gear coupling sleeve. Maximum variation of vibration displacement and acceleration is observed between phased and non-phased arrangement at 1280 rpm speed and 3.5 Kg load.
Keywords: Spur Gear Coupling, Mesh Stiffness, Phasing, Vibration

I. INTRODUCTION

In many transmission systems shaft couplings are critical parts for providing the smooth transmission of power from drive to driven equipment. Rigid couplings if designed properly, selected and maintained rightly can have long life and give very satisfactory service on this drive and driven equipment. But however its life is limited due to various factors namely, Human Errors, Corrosion, Wear, Fatigue, Hardware failure, Shaft failure etc. The major reason of the above failures results because of vibrations in drive. To reduce the vibrations, lots of research work is carried out on gears. Many of them focus on gear tooth profile optimization, changing contact ratio. These techniques have certain limitations due to load dependency. From literature survey it is found that internal excitations produced due to variation of mesh stiffness is the responsible factor for generation of vibrations. This paper focuses on reduction of vibration in gear transmission system consisting of gears along with gear coupling. Hence to reduce vibration phasing is done of gear coupling instead of gears. Due to phasing of gear coupling sleeve, vibrations of transmission system reduces, due to reduction in mesh stiffness variation.

Cheon Gill-Jeong [1] studied vibration reduction in spur gear pair with phasing. He find that the phasing gear greatly reduced the dynamic response and nonlinear behaviour of the normal gears. S. H. Gawande and S. N. Shaikh [2] in their paper show experimental work to study the effect of planet phasing on noise and subsequent resulting vibrations of Nylon-6 planetary gear drive. They observed that by applying the meshing phase difference one can reduce planetary gear set noise and vibrations. D.K.Kokare , S.S.Patil [3] studied the effects of phasing gears on time-varying gear mesh stiffness. They find reduction in mesh stiffness variation and the possibility of reduction in vibration in simple spur gear pair. S.S. Ghosh , G. Chakraborty [4] studied vibration reduction by reducing material by profile modification, a graphical method using contour plots and semianalytical methods are presented for above purpose. Ankur Saxena, Manoj Chouksey [5] showed faults can be detected in gear tooth by observing changes in dynamic characteristics of the system. Model and frequency response characteristics of cracked gear tooth are compared with healthy rotor system. Yang Luo , Natalie Baddour, Ming Liang [6] did numerical simulation to indicate that gear center distance variation has major effects on time varying mesh stiffness(TVMS). They showed center distance variation along with other faults such as tooth crack, pitting also have influence on TVMS. Hui Man, XuPang [7] develope mesh stiffness model for profile modifications and verified it using FEM method.

Omar D. Mohammed, Matti Rantatalo [8] presented the effect of tooth mesh stiffness variation on the dynamic response of cracked gear tooth. They also investigated the effect of gear mesh stiffness o fault detection indicators, the RMS, kurtosis and crest factor.S.H. Gawande, S.N. Shaikh, R. N. Yerrawar, K.A. Mahajan [9] did experimental work on planetary gear set to find out the effect of phasing on noise level. They find approximately 6-7 db reduction in noise level due to phasing. A. Fernandez del Rincon, F. Viadero , M. Iglesias, P. García, A. de-Juan, R. Sancibrian [10] presented the effect of loaded transmission error in case of spur gear transmission. They developed a numerical quasi stastic model to find loaded transmission error under various conditions of load. D. Richards, D.J. Pines [11] used a passive method of reduction of vibration. They use the passive element as a periodic drive shaft. They find reduction in the level of vibration produce due to mesh frequency varion. The limitation of the work is that they do not take into consideration rotation effects of drive shaft. J Hedlund and A Lehtovaara [12] used parameterised numerical model for estimating mesh stiffness variation and transmission error for helical gear pair. They used finite element (FE) method to find out mesh stiffness variation. Jim Meagher, Xi Wu, Dewen Kong, and Chun Hung Lee [13] emphasised on gear health monitoring by reducing vibrations created due to gear mesh stiffness. Pair of high and low contact ratio spur gear are compared for their predicted stiffness cycle and dynamic response. Prof. S. B. Wadkar, Dr. S. R. Kajale ,Dr. K. Gupta , G. M. Desai [14] stated that tooth mesh stiffness depends on the number of teeth pairs in contact and point of contact of teeth.

II. Numerical Analysis

The study focuses on the reduction of vibration of spur gear coupling by minimising the mesh stiffness variations. Vibrations in case of gear coupling are produced due to internal excitations created by mesh stiffness variations. For numerical analysis system is considered to be a second degree of freedom system. A nonlinear model of spur gear given in the [1] is considered and schematically represented as below.

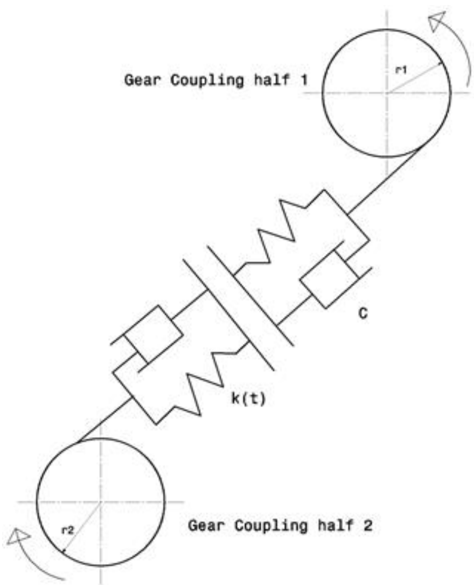

Fig. 1 Spur gear coupling model

According to [1] the equations of the motion of two gear coupling are as follows,

$$(I_1 + I_i)\ \ddot{\theta}_1 + r_1 C\left(r_1\, \dot{\theta}_1 - r_2\, \dot{\theta}_2\right) + r_1 K(t)\beta(t) = T_i \quad (1)$$

$$(I_2 + I_o)\ \ddot{\theta}_2 - r_2 C\left(r_1\, \dot{\theta}_1 - r_2\, \dot{\theta}_2\right) - r_2 K(t)\beta(t) = -T_o \quad (2)$$

The gear backlash nonlinearity was modelled as a piecewise linear function as below,

$$\beta(t) = \begin{Bmatrix} r_1\theta_1 - r_2\theta_2 - b \text{ when } r_1\theta_1 - r_2\theta_2 > b \\ r_1\theta_1 - r_2\theta_2 + b \text{ when } r_1\theta_1 - r_2\theta_2 < b \\ 0 \qquad\qquad \text{when } |r_1\theta_1 - r_2\theta_2| \le b \end{Bmatrix} \quad (3)$$

Where, I_1, I_2 are mass moment of inertia of gear couplings driver and driven gears, $\theta_i, \theta_o, \theta_1, \theta_2$ are vibrations of driver, load and gear 1 & 2.

Tooth mesh stiffness can be viewed as rectangular function [1, 3, 14]. When meshing of gear coupling is done, its frequency gets doubled and average mesh stiffness reduces. It can be represented as below,

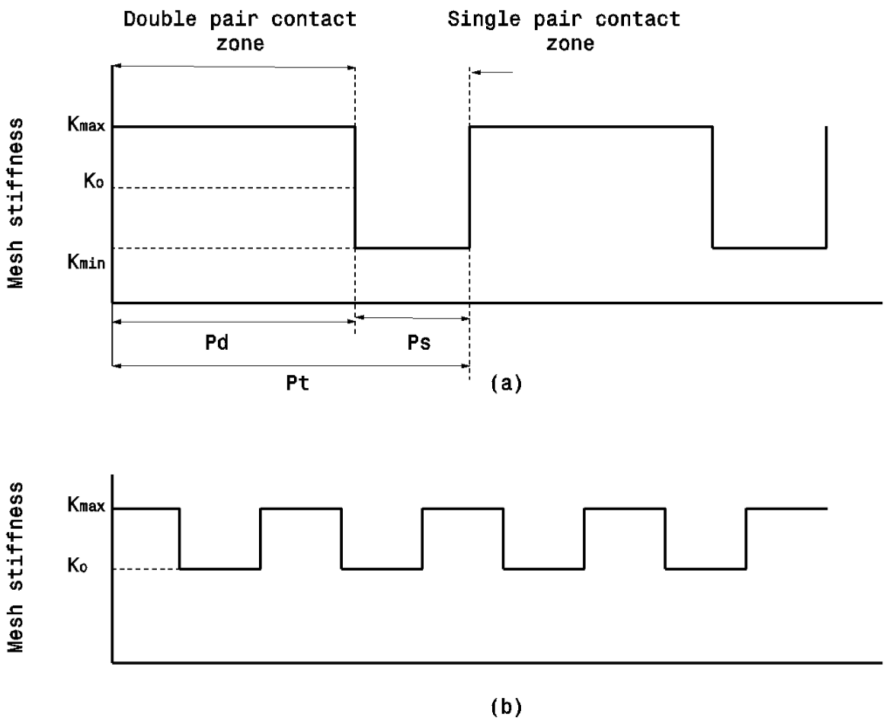

Fig. 2 Mesh stiffness (a) non-phased gear coupling (b) phased gear coupling

Mesh periods Pt, Pd and Ps are determined as follows,

$$Pt = \frac{60}{Nt} \quad (4)$$

$$Pd = (CR - 1)Pt \quad (5)$$

$$Ps = (2 - CR)Pt \quad (6)$$

Where, N is rotational speed in rpm, t is number of teeth, CR is contact ratio, Pd is mesh stiffness for double tooth contact, Ps is mesh period for single tooth contact, Pt is total mesh period.

III. Mesh stiffness calculation

ABS polymer material is used for gear coupling sleeve with and without phasing. The selection is based on mechanical properties, wear résistance, material availability, torque and other parameters. Table 1 shows material selection and gear coupling specifications.

Table 1 Gear coupling specifications

Material	ABS Polymer
Module	1.428 mm
Number of teeth	28
Pitch circle diameter	40 mm
Pressure angle	20 degree

The average mesh stiffness values K_o = 1.2115 x 10^8 taken from ISO 6336 standard as specified in [10] for non-phased condition, Contact ratio is 1.54 for above gear coupling, Speed is 700 rpm, R = 5. The timely varying mesh stiffness K(t) is calculated as given in [3] as below,

$$K(t) = K_o + \sum_{r=1}^{R} K_r \cos(2\pi r f_m t - \emptyset_r) \quad (7)$$

The value of K_r and $\emptyset_r$ can be find out from following relations,

$$\frac{K_o}{K_s} = CR \quad (8)$$

$$\frac{K_r}{K_s} = \frac{\sqrt{2-2\cos[2\pi r\,(CR-1)]}}{\pi r} \quad (9)$$

$$\emptyset_r = \frac{1-\cos[2\pi r(CR-1)]}{\sin[2\pi r(CR-1)]} \quad (10)$$

In above notations K_o is average mesh stiffness, CR is contact ratio, K_s is gear mesh stiffness during single tooth contact, K_r and $\emptyset_r$ are rth Fourier coefficient and phase angle of K(t). By plotting mesh stiffness vs time curves for Non-phased and phased gear coupling is given in figure 3 & 4.

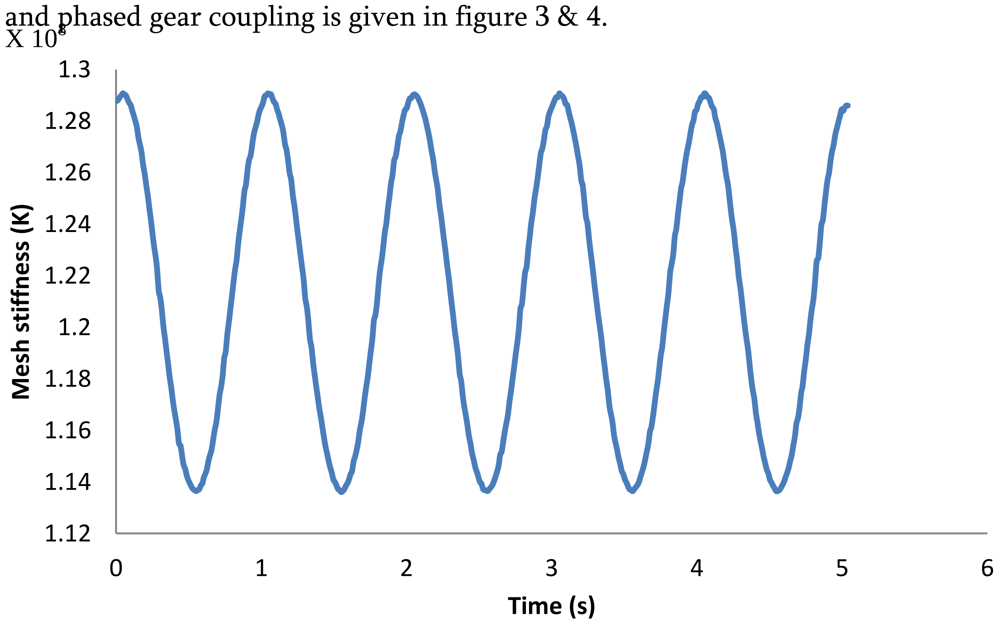

Fig. 3 Mesh stiffness variation for Non-phased gear coupling

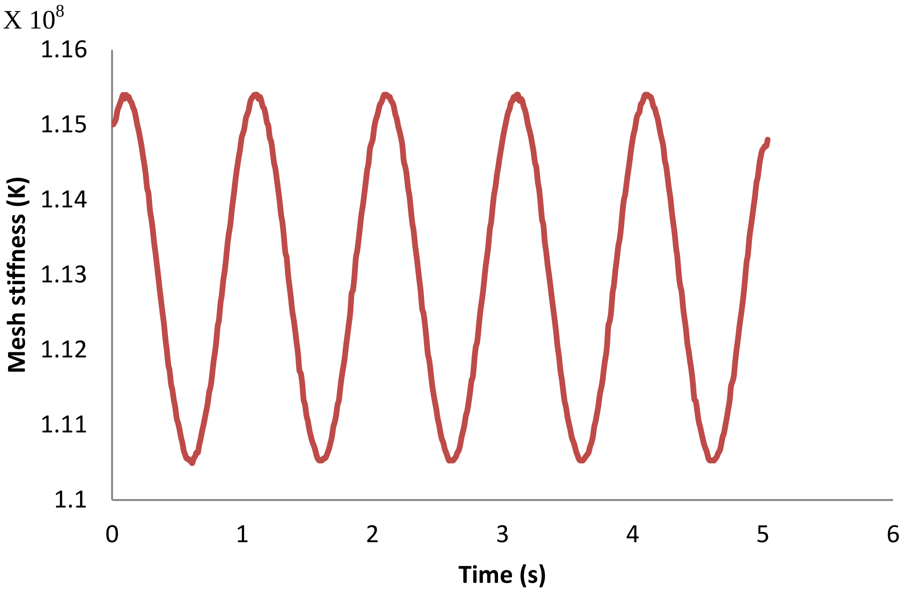

Fig. 4 Mesh stiffness variation for phased gear coupling

From above figure 3 & 4 it is observed that there is reduction in the mesh stiffness variation due to phasing of gear coupling sleeve. Maximum and minimum values of mesh stiffness variation are plotted in table 2.

Table 2 Comparison of maximum and minimum mesh stiffness for phased and non-phased condition

Mesh stiffness (K)	Non-phased gear coupling	Phased gear coupling
Kmax (N/m)	1.2907×10^8	1.1539×10^8
Kmin (N/m)	1.1360×10^8	1.1049×10^8

Above table shows that mesh stiffness variation is reduced due to phasing of gear coupling. Mess stiffness variation is a main contributor to production of vibrations. Hence this numerical study shows possibility of vibration reduction by the method of phasing to validate it experimentation is carried out.

IV. Experimental Setup and Measurements

Many researchers did lots of research in the field of vibration reduction. Some of them use macro and micro technics for vibration reduction. Macro technic include changing tooth numbers, changing centre to centre distance, changing materials whereas micro-technics include optimisation of tooth profile, changing contact ratio, but these technics have limitations due to load dependency. These are all passive methods of vibration reduction. Active methods of vibration reduction include use of actuators, signal processing, external energy source to reduce vibrations. These methods are costly and requires additional energy source. Hence in order to reduce vibrations, a new method is proposed which is based on phasing of gears by phasing gear coupling sleeve. In this method phasing is done within gear coupling sleeve so as to increase the total number of teeth pairs in contact. In phasing slight angular twist is given between input and output side of gear coupling sleeve. Due to increased number of teeth pairs in contact, variation of mesh stiffness reduces which is responsible for vibrations. For this purpose experimental setup has been prepared and many trials are taken to compare dynamic response of system with and without phasing of gear coupling sleeve. Vibrometer is used for finding various vibration parameters and tachometer is used to measure the speed fluctuations. Figure 5 shows the schematic diagram of the experimental setup. Actual experimental arrangement is shown in figure 6. A power transmission system consisting of spur gears and gear coupling is shown. Due to power transmission vibrations are created due to internal excitation caused by gear tooth mesh stiffness variation. These vibrations are transferred to the bearing housing and measured by placing probe of vibrometer in the bush welded to the bearing housing. Dyno pully is used for varying the load. Voltage regulator is used for varring voltage to the electric motor. It results into varring speed.

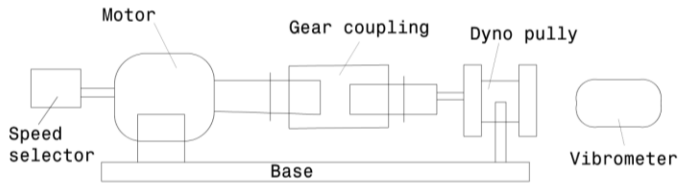

Fig. 5 Schematic layout of experimental setup for Gear couling vibration reduction.

Fig. 6 Photograph of experimental setup

V. Results and Discussions

To study the effects of method of phasing on the vibration parameters, experimentation is carried out with and without phasing of gear coupling sleeve. The results obtained are plotted in table 3 and table 4. To find the relation between power, torque, efficiency, vibration displacements, vibration acceleration for various values of load and speed are plotted below.

Table 3 Experimental observations without phasing arrangements

Sr. No.	load (KG)	Speed (rpm)	Vibration displacements (mm)	Vibration acceleration m/sec^2
1	1.5	1386	1.15445	4.529902
2	2	1378	1.163871	4.502582
3	2.5	1369	1.174639	4.448025
4	3	1358	1.18805	4.393579
5	3.5	1354	1.192997	4.366399
6	4	1280	1.291799	4.033575
7	4.5	1095	1.617852	3.244962
8	5	961	1.961949	2.656389

Table 4 Experimental observations with phasing arrangements

Sr. No.	Load (Kg)	Speed (rpm)	Vibration displacements (mm)	Vibration acceleration (mm/sec^2)
1	1.5	1394	1.125168	4.48748
2	2	1388	1.142117	4.45711
3	2.5	1376	1.156248	4.41325
4	3	1364	1.1707	4.35640
5	3.5	1358	1.16805	4.31429
6	4	1284	1.26408	3.97572
7	4.5	1104	1.588632	3.20641
8	5	964	1.942839	2.62405

The experimental results are carried out with the help of vibrometer and tachometer for measuring vibration parameter and speed respectively. Figure 7 shows variation of vibration displacements with speed for with and without phasing arrangement of gear coupling sleeve. It indicates reduction in vibration displacement by using phasing arrangements. Maximum reduction in vibration displacement is observed at 1280 rpm speed with a difference of 0.027 mm with and without phasing arrangement. It is also observed that as speed increases vibration displacement decreases due to increase in load. Vibration acceleration verses speed is plotted in figure 8. It is observed that as speed increases, vibration acceleration increases. Vibration acceleration with phasing arrangement is less as compared to without phasing arrangement. Its maximum difference of 0.0578 mm/sec^2 is observed at 1280 rpm.

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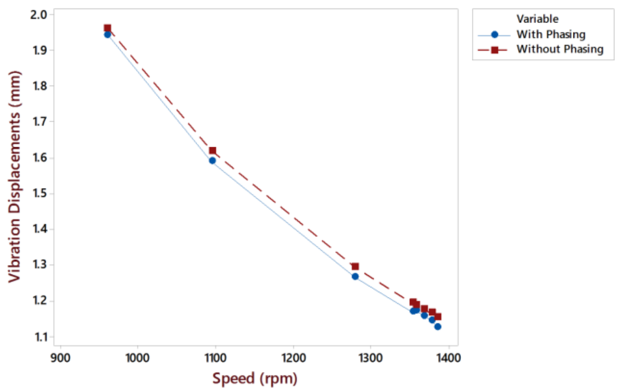

Fig. 7 Comparison of vibration displacements vs speed of Gear Coupling sleeve.

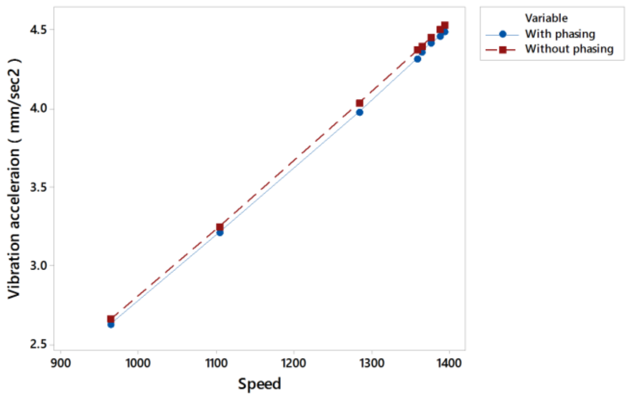

Fig. 8 Comparison of vibration acceleration verses speed of Gear Coupling sleeve.

In figure 9 variation of vibration displacement with load is shown with and without phasing. It shows reduction in vibration displacement with phasing arrangement. Maximum reduction in vibration displacement is observed at 3.5 Kg load with difference of 0.0249 mm between phased and non-phased condition. It also observed that with increase in load, vibration increases. Figure 10 indicates variation of vibration acceleration with load with and without phasing arrangement of gear coupling sleeve. As load increases, vibration acceleration decreases. Vibration acceleration is less with phasing arrangement than non-phased condition. Maximum difference of 0.0578 mm/sec^2 is observed at 4 Kg load.

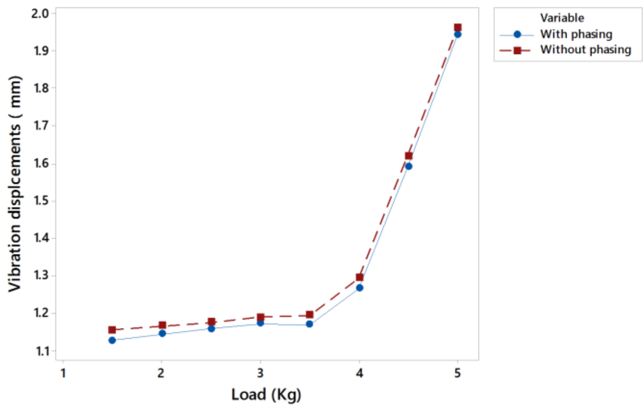

Fig. 9 Comparison of vibration displacements vs load of Gear Coupling sleeve.

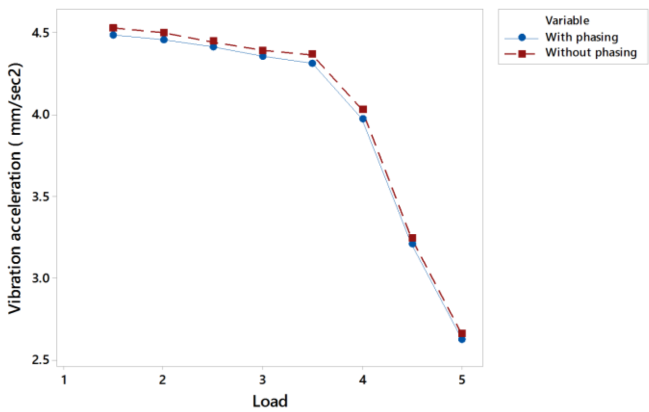

Fig. 10 Comparison of vibration acceleration vs load of Gear Coupling sleeve.

VI. Conclusion

The main purpose of this work was to reduce vibrations in power transmission system. Analytically maximum and minimum values of mesh stiffness for non-phased gear coupling are 1.29 x 10^8 N/m and 1.13 x10^8 N/m, for phased gear coupling are 1.15 x10^8 N/m and 1.10 x10^8 N/m respectively. Analytical results showed reduction in mesh stiffness variation and possibility of vibration reduction. Experimental setup has been prepared and trials are taken with the help of vibrometer and tachometer for measurement of vibration parameter (displacement, acceleration) & speed respectively. Various plots have been plotted with relevance to speed and load. It is observed that reduction in vibration displacements and acceleration is obtained with phasing arrangement. With maximum variation in vibration parameter of phased and non-phased condition is found out at 1280 rpm and 3.5 Kg load.

VII. Acknowledgment

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