International Journal of Mechanical and Industrial Technology ISSN 2348-7593 (Online)
Vol. 5, Issue 1, pp: (78-84), Month: April - September 2017, Available at: www.researchpublish.com

A Review on Investigating the Effects of Tightening Torque of Rigid Flange Coupling on Vibration Characteristics

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Abstract: **Condition based monitoring is a manufacturing active tool to predict and find out the causes of defect in any machine. Generally in industry almost all machines have rotating components and they rotate about their true axis, but due to difficulties in manufacturing, inspection and material handling it is very difficult to manufacture components with true length. So during manufacturing every process is carried out with given tolerance and some time it is very difficult to achieve that and due to this some defects are always present with components. In actual, machine working with some vibration may lead to breakage of mechanical linkages or component, so it is very important to find out that cause at early stage. In industry, flange coupling is used to transmit power and it has been found that rigid flange coupling is more prone to vibration. The objective of this paper is to review and summarize this literature to provide an extensive and good reference for researchers to utilize. The overview of the rigid flange coupling is introduced with existence problem and solution for them. The different parameter which affects vibration of rigid flange coupling and their results with FFT, Mathematical Modelling, ANSYS Workbench etc. are also reported.**

Keywords: **Misalignment, Coupling, FFT, Vibration.**

I. INTRODUCTION

Coupling is a device used to connect the shafts together for the purpose such as power and torque transmitting. Generally, couplings are used for connection of shafts that are manufactured separately. Some of the examples are motor, generator; electric motor and centrifugal pump etc. Inconvenience in transportation of shaft of greater length and limitations of manufacturing results in the reason of joining of two or more shafts by means of coupling. The shafts that are connected by coupling are simple enough to assemble and disassemble for the purpose of repair and alterations. The unadorned failure due to shearing of bolts head, key head, nuts and other projecting parts may cause accidents. So, it should be covered by fencing the flanges or by providing guards. The shaft connected by the coupling may have collinear axes, intercepting axes or a parallel axes with a small distance in between them. They are further classified into two types; Rigid and Flexible Coupling. Rigid flange coupling consists of two separate grey cast iron flanges. One keyed to the driving shaft and the other to the driven shaft by means of nuts and bolts arranged on a circle concentric with the axes of the shafts. There are two types of rigid flange couplings; Protected and Unprotected rigid flange coupling. A protected rigid flange coupling has a protective circumferential rim that covers the nut and bolt head. So in any case of failure of bolts during operation, broken piece of bolt will dash against this rim and eventually fall down, protecting the operator from any possible injuries. In unprotected rigid flange coupling there is no protective circumferential rim. So, in any case of failure of bolts, it may cause severe injuries to the operator.

Rigid flange is generally carried out by casting process as it consists of projection and recess. The commonly used material for flange coupling is grey cast iron which is characterized by graphitic microstructure causing fracture of the material to have a grey appearance. It is one of the most commonly and widely used form of cast iron and the cast material based on its casting characteristics. Most alloys of Iron contain 2.5-4% carbon, 1-3% silicon and the latter material is iron by weight proportion. It bears fewer tensile strength and shock resistance comparably to its compressive strength. The mechanical properties are controlled by the size and morphology of the graphite flakes. This deflect a passing crack and initiate counter less new cracks as the material breaks which procure good wear resistance and damping capacity. It also experiences less solidification shrinkage than other cast iron that does not form a graphitic microstructure during casting process. While casting, silicon promotes good corrosion resistance and tends to show increased fluidity. It also offers good weld ability

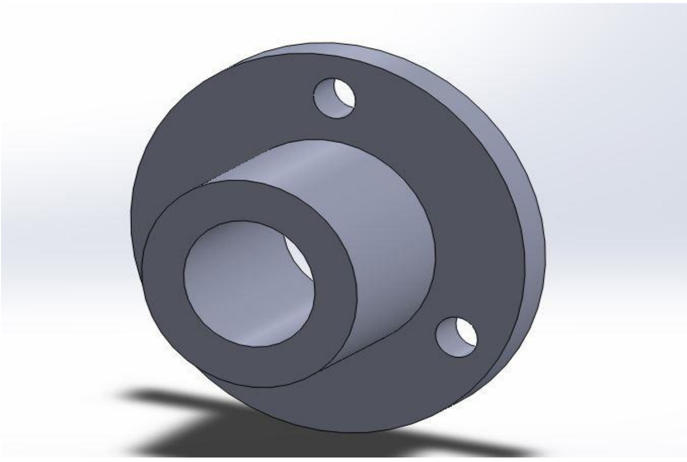

Fig.1 Flange Coupling

Brief overview of flange coupling

V. Hariharan and P.S.S. Srinivasan [1]

Machine vibration is caused because of Misalignment and unbalance in moving parts. Vibration may occur from unbalanced motor and also result in generation of excessive force in the bearing area and reduces the life of the machine. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration, reducing maintenance costs and increasing machine uptime. In this paper, experimental studies were performed on a rotor dynamic test apparatus to predict the vibration spectrum for rotor unbalance. They reported that due to presence of nut and bolt for assembly of coupling causes more vibration in coupling. So they designed and analysed a simplified 3 pin type flexible coupling for the experiments. The rotor shaft accelerations were measured at four different speed using accelerometer and dual channel vibration analyser (ADASH) under both balance (baseline) and unbalance conditions. The experimental and numerical (ANSYS) frequency spectra were also obtained for both conditions under different unbalanced forces. The experimental predictions support thoroughly with the ANSYS results. Both the spectra show that unbalance can be characterized primarily by one time (1X) shaft running speed.

Description of Pin Type Coupling [1]

The newly designed coupling is as shown in Figure 1. It has two flanges. One flange has a pin hole of required number at pitch circle diameter. The other flange will have a number of pins projected outside at pitch circle diameter to accommodate into the first flange holes with rubber bush. The driver and driven shafts are connected to their respective flanges i.e. output and input flanges by means of parallel square key. Mild-steel is considered for the input and output shafts, pins and keys. Over these pins, a circular natural rubber bush is provided and its length is equal to the length of the hole. The diameter of the flange holes is equal to the diameter of pin plus the thickness of rubber bush. The cast-iron material is chosen for both left and right flanges and the natural rubber is for bush. There is no nut and bolt to clamp the both input and output flanges. The following Figure 1 represent the two dimensional model of self-designed 3 pin type coupling. In between flanges a rubber material is introduced to give the flexibility.

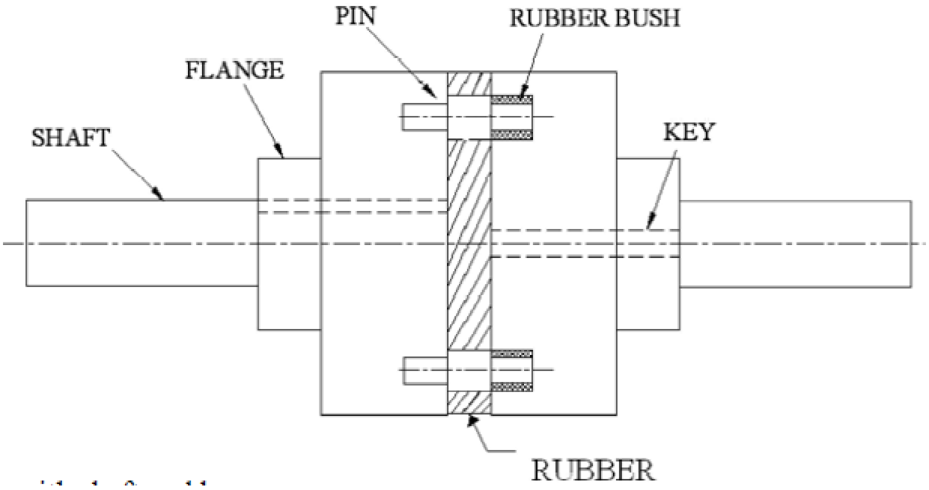

Fig. 2: Pin type coupling with shaft and key [1]

Frequency Spectrum of Base Line Condition [1]: The experimental and simulated frequency spectra were obtained to balanced condition. The perfect alignment and balance cannot be achieved in practice. Thus to show the residual unbalance and misalignment, a baseline (balanced) case is presented first. The measured acceleration of a well aligned and balanced system at drive end (DE) and non-drive end (NDE) with a self-designed 3 pin type.

Summary [1]: The simulation of the pin type coupling-ball bearing system with unbalance was performed. The validity of the model was successfully verified throughout the torsional experimental and simulation works and the investigation of rotor dynamic system characteristics related to unbalance was also done. The experimental and simulated frequency spectra were obtained. The experimental predictions are in good conditions and torsional vibration characteristics of a speed and Ansys results are backing each other. Both the experiment and increasing geared rotor-bearing system simulation results spectra shows that unbalance can be

Shamim Pathan and Pallavi Khaire [2]

Rotor systems have been widely used in mechanical engineering. The dynamics of rotor systems are study from decades ago. It became more important than ever with the increasing demand of the machineries. A rotating machine has one or more machine elements such as rolling-element bearings, impellers, and other rotors that rotate with a shaft, in a perfectly balanced machine, all forces are equal and all rotors rotate about their central axis. In industrial machinery, it is a common feature that an imbalance of forces to occur. In addition to imbalance generated by a rotating element, vibration may be caused by instability, therefore it can be concluded that the main causes of mechanical vibration are unbalance, misalignment, looseness, distortion, defective bearings, gearing defect, coupling inaccuracies, rotor shaft misalignment, bent rotor shaft and so on. The vibration caused by unbalance may destroy critical parts such as bearings, seals, gears and couplings, misalignment, looseness, distortion, defective bearings, gearing defect, coupling inaccuracies, rotor shaft misalignment, bent rotor shaft. In real system, faults are inevitable due to errors in manufacturing provision of tolerances on mating parts, errors while assembling different parts of the system, faults may develop in the system due to operating conditions such as heat generation, looseness and wear etc. Thus to avoid catastrophic failure it is very important to identify the following faults in machine in early stages itself. The defect in a rotor system will affect vibration behavior. Also the nature of this effect may result differently for different types of faults .Therefore it is said that machine vibration leads to the indication of machine failures. The identification of faults at early stage is a significant aspect of industrial strategy to avoid breakdown of machineries. In this sense the machine failure could be judged by thoroughly identification of faults and with proper maintenance.

Summery [2]:

A. Jaw coupling

Fig.2 and fig.3 shows a typical vibration spectrum on middle bearing (900 rpm) for balanced and unbalanced case at a frequency of 15Hz. Fig.3 shows peaks at 1×, 2× and 3 × .The magnitudes is 0.0041mm for the highest amplitude observed at 1×. From fig.4 it is observed that the unbalance effect shows dominant peak at 1× and there also show increase in vibration level when compared with balanced condition. The amplitude of vibration for unbalanced condition is 0.023mm.The vibration spectrum is also acquired for other two bearings.

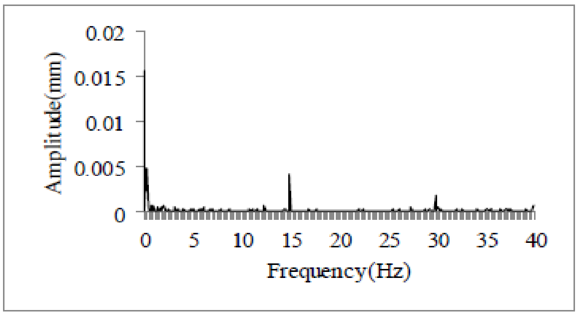

Fig.3 Frequency Spectrum at 900 rpm (15 hz) Balance Condition

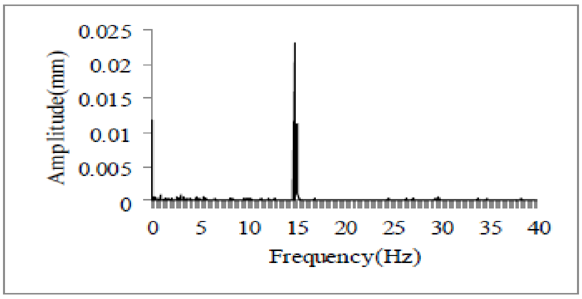

Fig.4 Frequency Spectrum at 900 rpm (15 hz) Unbalance Condition

B. Flexible flange coupling: The frequency spectrum for four pin type flexible flange coupling is acquired on all three bearing at 15Hz (900 rpm) for both conditions. Frequency spectrum for balanced condition is shown in fig.5.shows peak at 1× and the magnitude of amplitude is 0.012 mm. Fig.6 indicates the severity and unique characteristics of unbalance showing dominant peak at 1×. The amplitude of vibration for unbalanced condition for flange coupling is 0.303 mm.

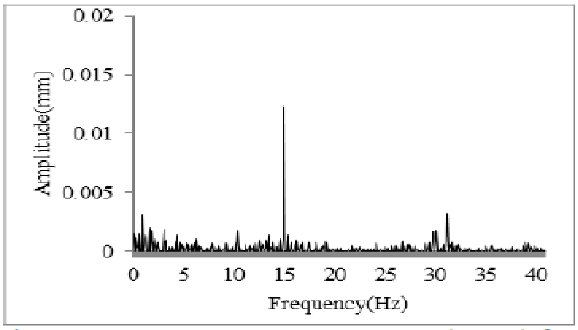

Fig 5.Frequency spectrum at 900 rpm(15 Hz) Unbalanced

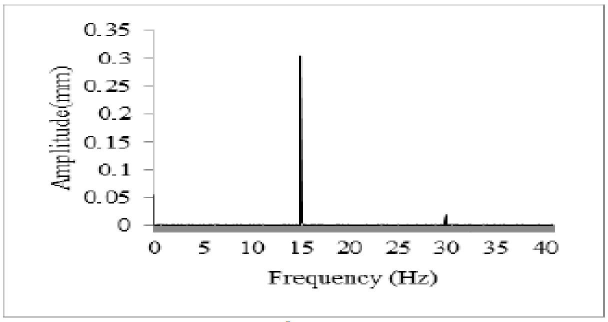

Fig.6. Frequency spectrum at 900 rpm(15 Hz) for for balanced

C. Rigid flange coupling:

The maximum amplitude can be seen at 1× for balanced and unbalanced condition on all bearings.Fig.7 and fig.8 shows frequency spectrum for balanced and unbalanced condition respectively for rigid coupling. From the above results, the rigid flange coupling shows higher vibrations at 15 Hz with maximum amplitude of 1X. All above couplings show the dominant peak on 1× for unbalanced condition and peaks at 1×, 2× and 3× for balanced condition. There is a reduction in vibration level as compared to other two bearings compared to jaw coupling.

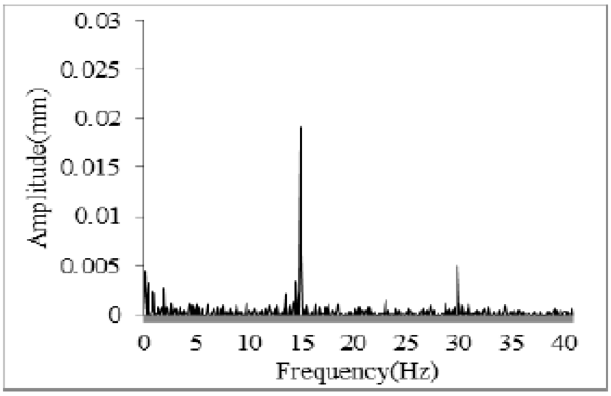

Fig.7 Frequency spectrum at 900 rpm for balanced flange coupling

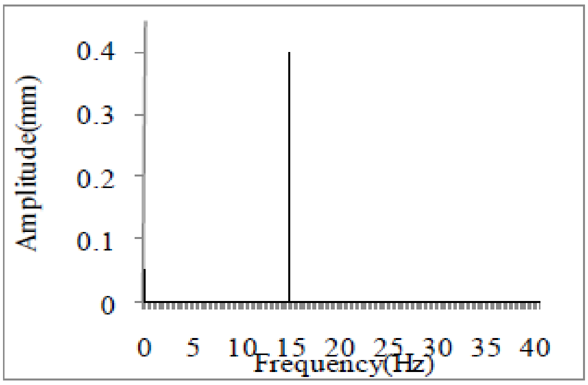

Fig.8 Frequency spectrum at 900 rpm for using condition rigid unbalanced condition using rigid flange coupling

Summery [2] - It is found from the experimental study that the unbalance shows dominant peak at 1× rpm. The type of couplings and vibrations are inter-related. Rigid coupling show more reaction towards the vibrations as compared to other two couplings in unbalanced condition. It is also observed the vibration level experienced is less when jaw coupling is brought into action. Hence it is concluded that the resistance of rigid flange coupling to vibrations due to unbalance is more as compared to others. Their observations are in close agreement as reported in literature.

Simone Delvecchio et al [3]

Diesel engines run very roughly at low speed range between 500-1000 rpm which is a major problem. And so in marine applications they introduce vibrations into the body of the boat. Thus performance in quietness is what expected by the overall marine machineries customers. Usually, the leaving from the ports requires the respect of the speed limits.

Summery [3]

The analysis of vibration signal picked up on suitable measurement points monitors the internal combustion. Regarding engines for marine applications, overall customer satisfaction is based on performance in quietness. In this paper the evaluation of the influence of two different types of flexible coupling over the vibrational behavior of the engine is executed. These couplings are mounted between the flywheel of the marine diesel engine and the propeller shaft. It is inadequate to carry out a meaningful comparison among couplings spectrum analysis, led in different operating conditions. On the other hand, the use of time-frequency analysis method is well suited. In particular, the application of the wavelet transform is considered in the present work. The high sensitivity of the wavelet transform technique to transient phenomena makes it suitable to provide monitoring features of the couplings. Since, to prove the efficiency of this method, tests were performed by changing the gear repeatedly. Different impulsive phenomena were obtained and the calculation of the mean value of the peaks in time domain is evaluated with the help of such conditions. In order to precisely localize in time-frequency domain these phenomena for monitoring purposes, the wavelet transform amplitude for a frequency range is taken into considerations. In addition the discrete wavelet transform is used in order to point out the impulsive phenomena in signals where the signal to noise ratio is low. According to the presented results, the wavelet transform appears to be a successful monitoring tool for the analysis of the vibrational effects of

Flexible marine couplings

K. M. Al-Hussain and I. Redmond [4]:

Nowadays, the problem which concerns designers and maintenance engineers much is of misalignment encountered in rotating machinery. For the purpose of trouble shooting the need for understanding the phenomena is very imperative to practical engineers. Most rotating machinery consists of a driver and a driven machine, which are coupled through some type of coupling. There are many types of industrial couplings. Some of them are of rigid, gear, and flexible type. To transmit torque from the driver to the driven machine is the main function of coupling. There are two types of coupling misalignments: parallel and angular. A combination of parallel and angular misalignment is common in the industry. This causes high vibrations with different symptoms which seldom has some explanations. The engineers and designers are into the discussions in the industry regarding the interpretation of the vibration signal, caused by misalignment, coming from a machine. But no accurate answer has been still found out. It was found out that the level of vibration is strongly dependent on the location of the coupling with respect to the bending mode shape. The flexible coupling behaves exactly as a universal joint to take the misalignment erect into account was some of the assumptions made in their studies.

Summery [4] - Some of the results were found through their studies, they were- a model for coupled lateral and torsional vibrations of two rotors subjected to pure parallel misalignment. The system degrees of freedom are the model's four orthogonal lateral defections, the rigid-body rotation, and the model's torsional deformation. The system equations of motion are obtained using the Lagrange equations. Successive partial differentiations of the kinetic and potential energies were also related with the above conclusions. The equations of motion are coupled in the stiffness matrix and the force vector as a result of the presence of misalignment. The equations of motion are put in dimensionless form for the general situation.

G.N.D.S. Sudhakar, A.S. Sekhar [5]-

Due to their ability to identify location and severity of the fault, model based methods for fault identification in rotating systems is gaining importance for the last three to four decades. Model based methods are of different types. Among them, equivalent loads minimization method is one. In this method, by minimizing difference between equivalent loads estimated in the system due to the fault and theoretical fault model loads fault is identified in a rotor bearing system. The error in identified fault parameters increases with decrease in number of measured vibrations is the biggest limitations of this process. Thus a comprehensive methodology for fault identification with minimum error even in case of fewer measured vibrations is attempted in the present work. Two different approaches: equivalent loads minimization and vibration minimization method are applied for the identification of unbalance fault in a rotor system. Some proposed methods by measuring transverse vibrations at only one location are used to obtain results. He studied that theoretical fault models used in the least squares algorithm of equivalent loads minimization method are modified to reduce error in identified fault parameters. Also, another method based on vibration minimization is applied for fault identification where difference between measured vibrations and calculated vibrations (obtained by adding theoretical fault model loads at the fault location on undamaged system) are minimized to identify fault parameters. Both methods are simulated for unbalance fault identification in a rotor system. Also, unbalance fault is experimentally identified using these methods in a SPECTRAQUEST MFS rotor system successfully by measuring transverse vibrations at one location (i.e., 2 dof) only.

Saurav Rajgadia1, Debayan Das2, Pawan Jaiswal3, Ankit Basnet4, Anupam Raj Jha5, Rakesh Jaiswal6,

Anush Karki7, Rabindra Nath Barman8 [6]:

The purpose of this report is to reduce the shear-stress on the nuts and bolts section of a rigid-flange coupling by carrying out the analysis of stress using FEM. Solid works 12.0 was used to design the 3-D CAD model and ANSYS 14.5 was used to do the analysis the stress. Coupling selection involves a number of design criteria including application, torque (Power), misalignment, stiffness, inertia, rotational speed, environment factors, space limitation, service factor, cost and others. All criteria were considered and addressed in the selection process to ensure that the coupling will work properly without premature failure. The result obtained from the analysis of the bolts and keys of a rigid-flange coupling using ANSYS workbench is better than that of calculation using analytical method. From this comparison we can conclude that the design of coupling done using Solid works is more suitable and safe.

K.M.AI-Hussian, I. Redmond [7]

The parallel misalignment on the lateral and torsional responses of two rotating shafts is observed with theoretical and numerical study. The general equations of motion are derived and given in dimensionless form to represent the general case. The equations of motion revealed that parallel misalignment couples the translation and angular defections through the stillness matrix and the force vector. The non-linear Equations are solved numerically using a combination of new mark and Newton Rap son methods to determine the dimensionless frequency and transient responses in terms of misalignment magnitude. The results shows that the system natural frequencies are excited at transient condition because the presence of pure parallel misalignment. At steady state condition, the rotational speed excitation is present in the translation and angular. A direction, which shows that parallel misalignment, can be a source of two torsional and lateral excitations.

M.Xu, R.D. Marangoni Part I [8]

Shaft misalignment and rotor unbalance are the main problem in all machines that are in rotating motion. In order to understand the characteristics of these machines theoretical model developed by them was motor flexible coupling and they studied universal joint effect. Theoretical analysis was developed using the component mode synthesis technique. A computer program was developed on the basis of theoretical model. The natural frequencies of the components and system were calculated using the FEA and component mode synthesis.

M.Xu, R.D. Marangoni Part II [9]

Rotor dynamics study was performed to verify the theoretical model of rotor misalignment and shaft misalignment. In this study self-designed flexible coupling and a helical coupling were used. They measured and observed frequency spectrum and their conclusion was some defects remained hidden and does not show in the frequency spectrum. After increasing the speed of shaft up to its natural frequency, a resonance type condition occur. As speed increasing further frequency density also increases.

II. CONCLUSION

From the literature study it can be concluded that

1. Vibration affects the behavior of any rotating machines & for that lot of study has been carried out and need to study more.
2. For alignment between two shafts coupling is most used and vibration level varies with its type

3. Generally it is observed that flange coupling is more prone to vibration, but it is need to find out the root cause.
4. Vibration effect can be analyzed with the help of tightening and loosening of flange coupling parameter.

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