CHAPTER 2
LITERATURE REVIEW

Stability analysis of hydrodynamic journal bearings is carried out by many researchers, R. Pai et al. (1991) considered a nonlinear transient method to analyze the stability characteristics of an unloaded rigid rotor supported in submerged oil journal bearings undergoing conical whirl. In their analysis the rotor had two degrees of freedom of motion in a self-excited conical whirl. The time-dependent form of the Reynolds equation was solved by them using finite-difference method with a successive over-relaxation scheme to obtain the moment components. Using these moment components, the equations of motion were solved by a fourth-order Runge Kutta method by them to predict the transient behavior of the rotor [3]. M. C. Majumdar et al. (1989) deal with a theoretical investigation of stability using a non-linear transient method for an externally pressurized porous gas journal bearing. Their analysis gave the journal center locus and from that the system stability was determined. With the help of graphics, several trajectories of the journal centre have been obtained by them for different operating conditions [4]. S.K.Kakoty et al. (2000) made an attempt to study the effect of fluid inertia on stability of journal bearing for a flow in laminar regime, a linear perturbation technique was used by him to find the dynamic characteristics [5]. An extensive survey of the experimental research on the static and dynamic characteristics of fixed geometry, hydrodynamic journal bearings was presented by E.E. Swanson et al. (1997). The type of bearing, size of bearing and range of parameters measured in each of their work were reported for slightly over 100 published experimental works. In addition, some general observations about the available experimental data sets were made [6]. A.H.Elkholy et al. (1995) presented a comprehensive technique, which could be applied to almost any rotating equipment to identify and diagnose journal bearing problems that relate to metal-to-metal bearing surface contact. Orbital measurements that describe bearing parameters in different modes of operation were experimentally obtained and analyzed by them. Such parameters included attitude angle, minimum oil film thickness, and the possibility of metal-to-metal rubbing occurrence. The general outline of the presented experimental technique was substantiated using the Raimondi-Boyd well documented design charts.
and good correlation between experimental and analytical results were obtained by them [7]. Anjani Kumar et al. (1996) considered the effects of geometric change due to wear on stability of hydrodynamic turbulent journal bearings following Constantinescu's turbulent lubrication theory in their analysis. Stability curves had been drawn for various values of wear depth parameter considering turbulence. Their analysis shown that wear causes the stability of the rotor to deteriorate in the case of lightly loaded bearings, and in case of worn bearings a lower L/D ratio gives better stability[8]. Madhumita Kalita et al. (2004) studied the dynamic behavior of Timoshenko beam supported on hydrodynamic bearings incorporating internal damping using FEM model. In their investigation critical speeds were estimated for synchronous whirl at different operating conditions using Campbell diagrams. It was observed in their analysis that in addition to the natural whirl frequencies, for every spin speed another whirling frequency appears in the solution, which was around half the spin speed. In case of dynamic coefficients evaluated using short bearing approximation, they observed that these additional frequencies were of the same order as that of the synchronous whirling frequencies[9]. Myunggyu Kim et al. (2010) investigated the stability of a disk-spindle system supported by coupled journal and thrust bearings considering the five degrees of freedom of a rotor-bearing system. The Reynolds equations and their perturbed equations of the coupled bearings were solved by FEM to calculate the stiffness and damping coefficients. Theoretical mass of the rotor-bearing system was determined by them by solving the linear equations of motion. As a result of their analysis the tilting motion was a major design consideration to determine the stability of the rotor-bearing system [10]. J.M.M.Guijosa et al. (2000) presented the bifurcation types that were associated with the loss of stability in order to gain insight into whether the linear stable regions were robust under finite perturbations. The effect of pad friction force on the rotor stability was also examined in their analysis[11].The effect of surface roughness on the stability characteristics of hydrodynamic journal bearings had been studied by R Turaga et al.(1998). Roughness was considered to be a stochastic variable, which was stationary, erotic with mean zero. Both one-dimensional roughness and two-dimensional roughness had been studied using the stochastic finite element method. Journal bearings with different L/D ratios were considered in their investigation. It was seen in their analysis that the transverse
roughness tends to increase the stability (non-dimensional mass parameter) whereas in the case of isotropic roughness there was a decrease when compared with the smooth bearing [12]. Chen-Chao Fan et al. (2010) described an experimental investigation of whirl elimination of a hydrodynamic bearing with an electromagnetic exciter (EE) and included a stability analysis through a root locus plot in their analysis. The test apparatus incorporated the EE, which provided additional stiffness through a PD algorithm in order to stiffen a rotating machine to resolve fluid-induced in stability. The stiffness of the hydrodynamic bearing gave a significant contribution to the occurrence of unstable vibrations as the speed or the load of a rotating machine was increased. With regard to the whirl-eliminated investigations the EE used to raise the stiffness of the rotating machine causes important changes in the stability of the rotating machine [13]. Sanxing Zhaoa et al.(2005) established motion equations for symmetrical single-disk flexible rotor-bearing system, and non-linear oil-film forces of finite journal bearings were calculated in their analysis. The rotor’s stiffness and damping were considered in their investigation. The motions of journal and disk were simulated with fourth-rank Runge-kutta method. The threshold speed of the system based on linear oil-film forces was derived. Non-linear transient simulation and unbalanced responses were investigated [14]. K. Raghunandana et al. (1999) investigated the effect of the non-Newtonian behavior of lubricants, resulting from the addition of polymers, on the performance of hydrodynamic journal bearings. The model of non-Newtonian lubricant developed by Dien and Elrod was taken into consideration in their analysis. An attempt was made by them to evaluate the mass parameter (a measure of stability) besides finding out the steady-state characteristics of finite journal bearings with non-Newtonian lubricants. A non-linear time transient analysis was carried out for the stability analysis[15]. Chris A. Papadopoulos et al. (2008) modeled the rotor using the finite element method with 4DOF’s per node, including the gyroscopic effect. The dynamic coefficients of the bearing were calculated by them by solving the Reynolds equation, The 4 x 4 stiffness and damping matrices, including the force moment and displacement-rotation relations with all non-diagonal coupling terms, were taken into account in their analysis. The stability of the system as a function of the rotational speed and the wear was examined by them [16]. Carine Boudesocque Duboisset al.(2003) presented a spectral numerical
method for solving one-dimensional systems of partial differential equations which arises from linearization of the Euler equations about an exact solution depending on space and time. Matching of quantities was performed by them in the space of characteristic variables. Time-dependent boundary conditions were handled following an approach proposed by Thompson. An exact numerical stability analysis valid for any explicit three-step third-order non-degenerate Runge–Kutta scheme was provided in their analysis [17].

G. D. Jiang et al. (1997) made an attempt of investigating dynamic characteristics of oil film to a large full scale journal bearing test rig which was designed with only the function of static measurement. The considerations of impulse excitation, vibration measurement, data log and process, and dynamic characteristics identification were presented in their analysis [19]. M. Miyatake et al. (2006) investigated the whirling instability of a rotor supported by aerostatic porous journal bearings with a surface-restricted layer. Their analysis revealed that a surface-restricted layer can greatly improve the static stiffness and the whirling stability of a rotor at high speeds [20].

S. Singhal et al. (2005) investigated the stability of a journal bearing system, including the effects of inlet viscosity. Their investigation also revealed that it was possible to stabilize a journal bearing either by heating the oil or by cooling the oil depending on the operating region [21]. Xie Wenhui et al. (2008) studied the complicated dynamical behaviour of a flexible rotor-bearing system. Two new phenomena were found in their system first, the chaos with two attracting areas which cannot be distinguished from the stable period doubling motion on poincare section second, for the flexible rotor system with two unbalanced disks, the response varied in a large extent when the phase angle between the eccentricities of disks was different [22]. Anjani Kumar studied the analysis of conical whirl instability of an unloaded rigid rotor supported in a turbulent flow hybrid porous journal bearing. The effect of bearing feeding parameter (b), Reynolds number (Re), ratio of wall thickness to journal radius (H/R) and anisotropy of porous material on the stability of rotor-bearing system had been investigated by her. It was observed in her analysis that higher values of b gives better stability and higher stability was predicted if the porous bush was considered to be isotropic [23]. A. El-Shafei et al. (2010) used an approach to control the journal bearing instability. An active magnetic bearing (AMB) was used in their analysis to overcome the JB instability and to increase its range of
operation. Instead of using the AMB as a load carrying element, the AMB was used as a controller only, resulting in a much smaller and more efficient AMB. Different controllers for the AMB to control the JB instability were examined and compared theoretically and numerically in their analysis [24]. Byoung Hoo Rho et al. (2002) employed a control algorithm for an actively controlled hydrodynamic journal bearing in order to suppress whirl instability and to reduce the unbalance response of a rotor-bearing system. A cavitation algorithm, implementing the Jakobsson Floberg Olsson boundary condition, was adopted by them to predict cavitation regions in a fluid film more accurately than the conventional analysis. The unbalance responses and stability characteristics of the rotor-bearing system were investigated in their analysis for various control gains and phase differences between the bearing and journal motion [25]. K.M. Al-Hussain (2003) examined the effect of misalignment on the stability of two rotors connected by a flexible mechanical coupling subjected to angular misalignment. The dimensionless stability criteria of the non-linear system of differential equations of two misaligned rigid rotors were derived by him using Liapunov’s direct method [26]. B. C. Majumdar (1976) reported the whirl instability of externally pressurized gas-lubricated porous journal bearings. The theoretical analysis was obtained using a quasi-static assumption. In his analysis the stability characteristic of a particular journal bearing was given for various supply pressures, design dimensions and feeding parameters [27]. Guo Hong et al. (2009) presented a theoretical study and experimental method to recognize the dynamic performance (stiffness and damping coefficients) of an externally pressurized deep/shallow pockets hybrid conical bearing compensated by flat capillary restrictors [28]. L. Sun et al. (2000) considered a flexible sleeve as a new feature of active journal bearing. Their investigation also revealed that variation of the oil pressure in the chamber located under the sleeve caused deformation of the sleeve and consequently changed the geometry of the oil film as well as the pressure distribution of the oil film [29]. Cheng Hsien Chen et al. (2006) investigated the restriction effects of capillary and orifice on the stability of the rigid rotor–hybrid bearing system. They used finite difference method and numerical integration to solve the Reynolds lubrication equations and static and dynamic performances of lubrication film, respectively, and the Routh–Hurwitz method was used to determine the stability threshold in their analysis [30].
P. Vleugels et al. (2006) presented the simulation techniques to predict the steady behaviour of a foil bearing. A method to calculate the dynamic properties was proposed in their analysis. Using stiffness and damping coefficients, a stability analysis was carried out. Their investigation also revealed that, even without additional damping, a foil bearing was more stable than a rigid surface aerodynamic journal bearing [31]. Guang Qiao et al. (2007) described a mathematical model to study the linear stability of a tilting-pad journal bearing system. By employing the Newton-Raphson method and the pad assembly technique, the full dynamic coefficients involving the shaft degrees of freedom as well as the pad degrees of freedom were determined by them [32]. Jerry C.T et al. (2003) analysed numerically the high-speed gas-lubricated porous journal bearings up to 200000 rpm. The effects of rotation speed, bearing eccentricity, permeability and thickness of the porous wall on bearing load capacity and attitude angle were investigated by them [33]. T. V. V. L. N. Rao et al. (2005) modified the governing equation incorporating the effects of roughness and cavitation in a journal bearing. The available theories of Reynolds roughness and cavitation algorithm proposed by Elrod were utilized in their work to develop a numerical procedure for stability analysis of a liquid lubricated rough journal bearing [34]. S Das et al. (2004) reported the theoretical study using both the linear analysis based on the perturbation method and non-linear analysis using the Runge Kutta method. Parametric studies had been conducted and stability characteristics had been obtained and a comparison of the results of linear and non-linear analyses for different parameters had been done in their analysis [35]. S K Guha et al. (2007) investigated theoretically the dynamic performance characteristics of finite-hydrodynamic porous journal bearings lubricated with coupled stress fluids using finite difference techniques. Using the first-order perturbation of the modified Reynolds equation, the stability characteristics in terms of threshold stability parameter and whirl ratios were obtained for various parameters in their analysis [36]. Helio Fiori de Castro et al. (2008) validated a complete nonlinear solution to simulate the fluid-induced instability during run-up and rundown in rotor bearings. A nonlinear hydrodynamic model was considered for short bearing and laminar flow in their analysis [37].
Above [2-17 and 19-37] researchers carried stability analysis of hydrodynamic journal bearings using different techniques like nonlinear transient method, effect of fluid inertia, spectral numerical method, determination of mass parameter etc.

[3] Considered a nonlinear transient method to analyze the stability characteristics of an unloaded rigid rotor supported in submerged oil journal bearings.[5] Considered the effect of fluid inertia on stability of journal bearing for a flow in laminar regime using a linear perturbation technique.[7] Developed a comprehensive technique, which could be applied to almost any rotating equipment to identify and diagnose journal bearing problems.[8] Considered the effects of geometric change due to wear on stability of hydrodynamic turbulent journal bearings.[9] Applied FEA to determine dynamic coefficients of journal bearings.[10] Investigated the stability of a disk-spindle system supported by coupled journal and thrust bearings considering the five degrees of freedom of a rotor-bearing system.[12] Studied the effect of surface roughness on the stability characteristics of hydrodynamic journal bearings. [13] Described an experimental investigation of whirl elimination of a hydrodynamic bearing with an electromagnetic exciter and stability analysis through a root locus plot.[15] Investigated the effect of the non-Newtonian behavior of lubricants, resulting from the addition of polymers, on the performance of hydrodynamic journal bearings.[16] Modeled the rotor using the finite element method to determine the dynamic coefficients of the bearing by solving Reynolds equation.[17] Used a spectral numerical method for solving one-dimensional systems of partial differential equations to analyze stability.[19] Considered the impulse excitation, vibration measurement, to find dynamic characteristics.[20] Revealed that a surface-restricted layer can greatly improve the static stiffness and the whirling stability of a rotor at high speeds.[21] Concluded that it was possible to stabilize a journal bearing either by heating the oil or by cooling the oil depending on the operating region.[22] Explained that the response varied in a large extent when the phase angle between the eccentricities of disks was different. [23] Observed in her analysis that higher value of $b$ gives better stability and higher stability was predicted if the porous bush was considered to be isotropic.
In present work stability analysis of hydrodynamic journal bearing with L/D ratio 1 is carried out at different operating conditions of speed and load using stiffness coefficients. Stability speed is further verified experimentally on journal bearing test rig.

Xiaobin Lu et al. (2006) reported the results of a series of experiments performed on a journal bearing together with a theoretical prediction of the Stribeck-type behavior. Various loads and oil temperatures were considered in their analysis. The comparison between the experimental results with a mixed elastohydrodynamic lubrication model for line contacts were reported [39]. Jean Frenea et al. (2006) presented different flow regimes, which occur in bearings and seals. The theories to obtain the characteristics of bearings operating in turbulent flow regime were presented in their analysis. The effects of inertia forces in laminar and in turbulent flows were shown. Results obtained using the complete Navier Stokes equations were presented and was shown how they were included in the classic lubrication theory [40]. Z M Jin et al. (1994) measured the lubricating film thickness in a model of compliant layered bearings for total joint replacements by means of optical interferometer under entraining motion. The essential features of the interferometer technique were OH-normal incidence light, a combination of polyurethane elastomer and a crown glass plate as bearing surfaces and the use of silicone fluid or water as lubricants. The film thickness in the lubricated contact was measured in their analysis for both water and silicone fluid under a range of entraining velocities [41]. L Gustafsson et al. (1994) proposed an image processing method for the analysis of film thickness. Their method makes it possible to extract considerably more information about film thickness fluctuations which were achievable by the naked eye. Their method primarily matches description from digitized color interferometric images of the unknown film shapes with calibration values obtained with known geometric shapes. Their method shown to work well in the range from 95 \( \mu \text{m} \) to 700 \( \mu \text{m} \) with white light and makes the results impartial by the observer [42]. I. Sherrington (2011) focused on development and use of a tool to study the hydrodynamic lubrication of piston rings in internal combustion engines, which included the development of experimental sensors to measure lubricant film thickness [43]. P. Harper et al. (2003) considered that a wave of ultrasound will reflect from a lubricant film between two bearing components. They reported that for thin lubricant films the layer behaves as a spring and the proportion of
the wave reflected was related to the stiffness of the oil layer. This provided a method for determining the film thickness. An ultrasonic transducer was mounted on the outside of a hydrodynamic journal bearing bush. The film thickness was calculated from the reflection spectra[44]. Zhiqiang Liu et al.(2002) investigated experimentally and theoretically the effect of a deposited soft thin metallic film on friction properties of a hardened steel substrate. The dependency of the film thickness and contact load on the static friction coefficient was presented in their analysis. Their experimental observations shown that deformation of the film in contact was plastic, thereby confirming the assumption of the theoretical calculation [45]. B.R.Hohn et al.(2001) did a survey of the research work that has been carried out in this field in Germany in the past 20 years and explained that highly loaded lubricated rolling and sliding contacts, such as gear contacts or roller bearing contacts, are extremely important for the durability of most gear applications [46]. To assess the reliability of new simulations, S J Sochting et al. (2009) compared the output of computer models with experimental measurements of parameters on operating engines. An experimental study of an investigation into the effect of load on the minimum oil film thickness between piston rings and cylinder liner in a fired compression ignition engine was carried out by them. Oil film thickness data were collected using capacitance-based transducers located near top dead centre and mid-stroke. Experiments were performed by them at 2000 r/min using two mono-grade oils (SAE 50 and SAE 20) and one multigrade oil (SAE 5W50) under a range of fixed engine loads[47]. H Moreau et al.(2002) carried experimental measurements and the theoretical calculations of oil-film thickness in a dynamically loaded connecting-rod big-end bearing. Four eddy current gap sensors for each bearing were used by them to measure the oil-film thickness and to deduce the bearing trajectories. Their solution used a mechanical linkage fixed under the piston axis. Their calculation process used finite element and Newton Rapson methods for the numerical analysis [48]. A system for monitoring oil film thickness based on a capacitive measurement technique was adapted by Kenneth Iraniet al. (1997) to measure hydrodynamic lubricating oil film thicknesses in the middle main bearing of a heavy-duty diesel engine. Transducers were developed to suit the extreme conditions in the bearing, with very high and varying pressures created by the lubricating film. Measurements of the film thickness as a function of the crank
angle was performed with a variation of the engine speed, load and oil temperature in their analysis [49]. Q W Qu et al. (2001) derived an equivalent viscosity model based on adsorption theory under the condition of different film thicknesses. Their investigation also revealed that thickness of adsorption layers hardly changes, but the load capacity of the oil film was increased rapidly as the oil-film thickness decreased [50]. Sas Bukovnika et al. (2006) developed a numerical method based on the hydrodynamic, elastohydrodynamic and thermo-elasto-hydrodynamic lubrication theory. Several crankshaft main bearings and connecting rod big end bearings were investigated in their analysis. The comparison of peak oil film pressure, minimum oil film thickness and oil flow was reported in their analysis [51]. C. H. Venner (2005) reported that EHL models based on the Reynolds equation in a steady state circular contact predicts a positive film thickness as long as the grid used in the calculations is sufficiently dense [53]. C. H. Venner et al. (2008) investigated, by means of numerical simulations using a starved lubrication model, the film thickness modulations in the centre of the contact induced by a harmonically varying inlet supply. A simple formula was presented by them for use in engineering predicting the ratio of the amplitude of the film modulations in the centre of the contact to the amplitude of the layer variations in the inlet [54]. O Marklund et al. (2001) discussed a computer-based analysis of interferogram recorded using an elastohydrodynamic lubrication Fitzeu interferometer (a so-called ball and disc apparatus) was discussed, the main objective of their analysis was to extract the absolute oil-film thickness. Intensity based methods most importantly, calibration look-up procedures where colour parameters from recorded dynamic interferogram were compared with table values corresponding to known film thicknesses in their analysis [56]. Z M Jin et al. (1994) used an optical interferometry technique to study the lubricant film thickness in a compliant layered bearing model for total joint replacements under squeeze-film motion. In their analysis experiments had been carried out for both thin and thick layers of compliant bearing material. It was demonstrated that the film thickness patterns depend significantly upon the layer thickness if other parameters were kept constant [57]. O Smith et al. (2011) described the development of an ultrasonic system to measure film thickness in main engine bearings under firing conditions. This unobtrusive method of measuring film thickness in operating contacts had been applied to the main
bearings of operating diesel engines. In their analysis preliminary measurements were taken during dynamic engine ramping to identify the thinnest operating film condition [58]. R S Dwyer-Joyce et al. (2006) evaluated an ultrasonic method of oil film measurement in PTFE pad bearings in hydro generator for plant condition monitoring. In their analysis an ultrasonic transducer was mounted on the back face of the thrust pad. Pulses were generated and transmitted through the pad material, bonding interlayer, and PTFE surface layer. The proportion of the wave that reflects back from the oil film layer was determined. This was then related to the oil film thickness using a series of calibration experiments and a spring stiffness model [59]. D M C McCarthy et al. (2005) investigated the influence of pad facing material on hydrodynamic lubrication in tilting-pad thrust bearings in terms of pad and oil-film temperatures and thicknesses. In their analysis two tilting-pad thrust bearings were examined one with babbitt pad facing, the other with a layer of PTFE-based composite material. Frictional torque, pad, collar, and oil-film temperatures and thicknesses were all monitored by means of a comprehensive array of sensors mounted in the bearing and shaft [60]. A simple iterative method was used by J O Ostensen (1995) for the computation of the central film thickness in a fully flooded elastohydrodynamic point contact lubricated with polymeric viscosity index improved base oil. The shear stress dependence of the VI improved base oil was included in the iterative method. In his analysis experimental data was presented on temporary viscosity loss and central film thickness at different temperatures from -8 to 40°C [61]. T Reddyhoff et al. (2005) studied how phase shift was explored as the film changes and was evaluated as an alternative means to measure oil film thickness. A quasi-static theoretical model of the reflection response from an oil film was developed in their analysis. The developed model relates the phase shift to the wave frequency and the film properties. Measurements of reflection coefficient from a static model oil film and also from a rotating journal bearing were recorded. These were used to determine the oil film thickness using both amplitude and phase shift methods [62]. H van Leeuwen (2009) described a method to deduct film thickness by adapting the value of the pressure viscosity coefficient until the differences between accurate film thickness approximation values and accurate film thickness measurements over a wide range of values were at a minimum. Eleven film thickness approximation formulas were compared in describing
the film thickness of a test fluid with known value of the pressure–viscosity coefficient [63]. R. S. Dwyer-Joyce et al. (2011) studied the reflection of ultrasonic waves from the lubricated contact between a sliding steel ball and a flat steel disc when substantial solid contact occurs. In their analysis the liquid film stiffness was calculated by using a predicted film thickness and a bulk modulus estimated from published rheological models of lubricants under high pressure [64]. Above researchers 39-64 carried out experimentation to determine fluid film thickness in different bearings using different displacement measuring techniques. [41] Measured the lubricating film thickness in a model of compliant layered bearings for total joint replacements by means of optical interferometer. [42] Used an image processing method for the analysis of film thickness. [43] used experimental sensors to measure lubricant film thickness of piston rings in internal combustion engines.[44] used a wave of ultrasound that will reflect from a lubricant film between two bearing components to measure oil film thickness.[45] investigated experimentally and theoretically the effect of a deposited soft thin metallic film on friction properties of a hardened steel substrate.[47] Collected Oil film thickness data in journal bearing using capacitance-based transducers located near top dead center and mid-stroke.[48] Carried experimental measurements and the theoretical calculations of oil-film thickness in a dynamically loaded connecting-rod big-end bearing.[49] Developed a system for monitoring oil film thickness based on a capacitive measurement technique.

In present analysis oil film thickness of hydrodynamic journal bearing with L/D ratio 1 is investigated theoretically and experimentally at different operating conditions by attachment of inductive transducer to journal bearing test rig. A.O. Kurban (2004) investigated a classical problem using an experimental apparatus and applying an artificial neural network (NN) optimization approach for determining the pressure distribution in a journal bearing at elasto-hydrodynamic state of lubrication. In his investigation the experimental system was employed at different working speeds with different surface roughness of shafts for getting pressure distribution. A NN was employed by him to predict the real data of the system as an optimization technique [66]. Ping Qin et al. (2005) used a feed-forward neural network to model the nonlinear oil-film force database of a finite length hydrodynamic journal bearing, which was constructed by
continuous transformation of Reynolds equation. Neural network models trained were utilized to investigate motion characteristics of a rigid unbalanced rotor supported on elliptical bearings in 300 MW steam turbine generator set. Their investigation also revealed that there exist similar motion behaviors between neural networks and numerical method [67]. A theoretical analysis on the general behavior of a thrust bearing was presented by Ali Osman Kurban et al. (2003) The model programme using a method adaptation of finite differences was developed by them to solve the Reynolds equation for lubrication. Their model in the theoretical analysis used a single one-dimensional grid. The altering of total lubrication load obtained in the result of under-cutting in the thrust bearing had been determined by them together with the parameters such as oil film thickness and pressure [68]. Abhinav Saxena et al. (2007) presented the results of investigation into the use of genetic algorithms (GAs) for identifying near optimal design parameters of diagnostic systems that are based on artificial neural networks (ANNs) for condition monitoring of mechanical systems. They proposed that GA can be used to select a smaller subset of features that together form a genetically fit family for successful fault identification and classification tasks. At the same time, their investigation also revealed that an appropriate structure of the ANN, in terms of the number of nodes in the hidden layer, can be determined, resulting in improved performance [69]. Cem Sinanoglu et al. (2005) presented an analysis of pressure development of journal bearing in a various shaft surface texture and velocity variations using a neural network. In their analysis the effects of the parameters, which act on performances of journal bearing, on the pressure development and load-carriage capacity were examined which were number of revolution and shaft surface texture. In their analysis the data from the experiments was used as learning information for the neural network to establish a reliable prediction model that can be applied to journal bearings [70]. J. Ghorbanian et al. (2011) developed a rapid and globally convergent predictive tool for dynamically loaded journal bearing design. A neural network model of crankshaft and connecting rod bearings in an internal combustion engine was developed by them as an alternative for the complicated and time-consuming models. In their analysis six most important parameters were selected as inputs of neural network. These parameters were, oil viscosity, engine speed, bearing radial clearance, bearing diameter, slenderness ratio and maximum force applied on
bearings. Also, some significant parameters were calculated by them as neural network outputs. These parameters were all components of friction loss, all components of oil consumption, minimum oil film thickness, eccentricity, oil temperature rise and displacement relative to shell [72]. According to the non-stationary characteristics of roller bearing fault vibration signals, a roller bearing fault diagnosis method based on empirical mode decomposition energy entropy was developed by Yang Yu et al. (2006). In their analysis original acceleration vibration signals were decomposed into a finite number of stationary intrinsic mode functions, and then the concept of empirical mode decomposition energy entropy was proposed. Their analysis results from roller bearing signals with inner-race and out-race faults shown that the diagnosis approach based on neural network by using empirical mode decomposition to extract the energy of different frequency bands as features can identify roller bearing fault patterns accurately and effectively [73]. C. Sinanoglu et al. (2004) investigated the pressure variations on the steel shafts on the journal bearing system with low temperature and variable speed. They carried analysis in two parts, experimental and simulation. In the experimental work, journal bearing system was tested with different shafts speed and temperature conditions. In their investigation collected experimental data of pressure variations was employed as training and testing data for an artificial neural network. The neural network used in their analysis was a feed forward three layered network. Quick propagation algorithm was used to update the weight of the network during the training [74]. Fazil Canbulut et al. (2004) investigated the effects of surface roughness on lubrication in slippers with varying hydrostatic bearing areas and surface roughness. Their network was capable of predicting the leakage oil quantity of the experimental system. In their analysis the network had parallel structure and fast learning capacity. It was seen from their experimental results that the leakage oil quantity was caused by surface roughness, orifice diameter and the size of hydrostatic bearing area, loading pressure and the number of rotations [75]. Experimental as well as theoretical work was done by Mansour Karkoub et al. (1997) to calculate the pressure distribution inside the bearing. The pressure distribution and the load-carrying capacity were predicted by them using feed forward architecture of neurons. The inputs to the networks in their analysis were a collection of experimental data. Collected data was used to train the network using the
Levenberg-Marquardt optimization technique. The results of the neural network model were compared to a theoretical model [76]. Khalid F. et al. (2008) developed a new technique for an automated detection and diagnosis of rolling bearing. The time-domain vibration signals of rolling bearings with different fault conditions were pre-processed in their study, using Laplace-wavelet transform for features extraction. The extracted features for wavelet transform coefficients in time and frequency domains were applied as input vectors to artificial neural networks (ANNs) for rolling bearing fault classification [77]. D. M. Yang et al. (2002) developed a novel condition monitoring procedure for rolling element bearings which involved a combination of signal processing, signal analysis and artificial intelligence methods. Seven approaches based on power spectrum, bispectral and bi coherence vibration analyses were investigated by them as signal pre-processing techniques for application in the diagnosis of a number of induction motor rolling element bearing conditions [78]. Nikos G. Pantelelis et al. (2000) developed a simple finite element models of a turbocharger (rotor, foundation and hydrodynamic bearings) combined with neural networks and identification methods and vibration data obtained from real machines towards the automatic fault diagnosis of naval turbochargers [79].

Above researchers 66-79 applied Artificial Neural Network analysis to analyse and predict different bearing parameters. [66] applied artificial neural network (NN) optimization approach for determining the pressure distribution in a journal bearing at elasto-hydrodynamic state of lubrication.[67] used a feed-forward neural network to model the nonlinear oil-film force database of a finite length Hydrodynamic journal bearing.[68] The hydrodynamic behavior of thrust bearing was analyzed by considering of different dimensionless system pressure, speed and geometry of the bearing.[69] used genetic algorithms (GAs) for identifying near optimal design parameters of diagnostic systems based on artificial neural networks (ANNs) for condition monitoring of mechanical systems.[70] presented an analysis of pressure development of journal bearing in various shaft surface texture and velocity variations using neural network.[72] Developed a neural network model of crankshaft and connecting rod bearings in an internal combustion engine for optimization of important parameters.[73] Developed roller bearing fault diagnosis method based on empirical mode decomposition.
(EMD) using neural network model.[74] Investigated the pressure variations on the steel shafts on the journal bearing system with low temperature and variable speed using neural network model.[75] The effects of surface roughness on lubrication in hydrostatic bearing areas and surface roughness in axial piston pumps bearings using artificial neural networks and predicting the leakage oil quantity of the experimental system.[76] Predicted the pressure distribution and the load-carrying capacity in bearings using feed forward architecture of neurons.

In present work Artificial Neural Network technique is applied for prediction of oil film pressure and oil film thickness in hydrodynamic journal bearing. Experimental values of oil film pressure and oil film thickness at different operating conditions of speed, load and viscosity are considered to train the network.

M. Gevers et al. (2006) examined the identification of multi-input systems motivated by an experiment design problem. The effect of an additional input on the variance of the estimated coefficients of parameterized rational transfer function models, with special emphasis on the commonly used FIR, ARX, ARMAX, OE and BJ model structures. In their analysis it was shown that, for model structures that have common parameters in the input–output and noise models (e.g. ARMAX), any additional input contributes to a reduction of the covariance of all parameter estimates. Their investigation also revealed that the accuracy improvement extends beyond the case of common parameters in all transfer functions, and shown exactly which parameter estimates are improved when a new input is added [80]. Chiou fong chung et al. (2002) applied a system identification method to obtain a nonlinear equation for an oil film rotating system. In their analysis the centrifugal force induced by an unbalanced mass was used as the input signal for identification, and the phase between the input signal and the measured output vibration amplitude was calculated to perform the identification. In their analysis stability analysis and system performance were evaluated by using the root locus [81]. Jerzy T. Swaicki et al.(1996) described system identification and data reduction methods used for extracting rotor dynamic coefficients of fluid-film journal bearings. In their analysis data was used from a test apparatus incorporating a double-spool shaft spindle which permits independent control over the journal spin speed and the frequency of an adjustable-magnitude circular orbit, for both forward and backward whirling. To assess the quality
of the measured signals, in their analysis coherence functions were calculated to relate the time-averaged input motion signals and the time-averaged output force signals [82]. Yeong-shu Chen et al. (2010) proposed an identification technique in the dynamic analyses of rotor bearing foundation systems called the pseudo mode shape method. Two procedures, namely, phase modification and numerical optimization, were proposed in their algorithm of pseudo mode shape method to effectively improve its accuracy. Their investigation also revealed that pseudo mode shape method uses the frequency response function (FRF) data of joint positions between the rotor and the foundation to establish the equivalent mass, damping, and stiffness matrices of the foundation without having to build the physical model [83]. An identification algorithm for bearing dynamic characterization by using unbalance response measurements was developed by R. Tiwari et al. (2002) for multi-degree-of-freedom flexible rotor-bearing systems. Their algorithm identifies the bearing dynamic parameters, consisting of four elective stiffness and four damping coefficients for each bearing, utilizing frequency domain synchronous unbalance response measurements from the accelerometers attached to the bearing housings in the horizontal and vertical directions, for a minimum two different unbalance configurations [84]. Helio Fioride et al. (2010) discussed the identification of parameters in rotary systems namely, the unbalance magnitude, phase and position in the rotor system. These parameters were identified using the measured orbits in the hydrodynamic bearings. In their analysis oil film forces were evaluated in the different positions of the orbit of the journal and were applied to the model of the shaft. Their model, integrated in time domain, allows with an assumed unbalance, to simulate the orbits [85]. Frank Peeters et al. (2001) discussed the non-parametric identification of multiple input multiple output rotor-bearing systems in the frequency domain for real as well as for complex modal testing. Their FRFs were estimated on the basis of the maximum likelihood estimator, considering noise on both inputs and outputs. In their analysis experiments were performed on a rotor test rig and the evaluation was made for both real and complex modal analysis [86]. Sharad Shekhar Palariya et al. (2011) presented the dynamic analysis of a dual-disk rotor supported over a double-row ball bearing system. Finite element model of the rotor was employed in their analysis to know the approximate range of first few bending frequencies and the corresponding resonance
amplitudes as a function of bearing stiffness coefficient. In their analysis a non-parametric model based on back-propagation neural network was developed in order to assess the bearing stiffness for known values of frequency shifts and resonance amplitudes with respect to rigid bearing support conditions [87]. Gaetan Kerschen (2006) carried a survey containing a review of the past and recent developments in system identification of nonlinear dynamical structures. The objective was to present some of the popular approaches that have been proposed in the technical literature, to illustrate them using numerical and experimental applications, to highlight their assets and limitations and to identify future directions in this research area [88]. S.X. Zhaoa et al. (2005) proposed an experimental method to identify coefficients and present their characteristics under various operational conditions. In their analysis a delicate test rig was constructed and experimental data were acquired under various testing conditions. From their experimental data, the relative velocity of the Journal and the oil-film forces was obtained by using a differentiator. The coefficients were identified using least-mean-square method in time domain [89]. V. Meruaneet al. (2008) proposed a framework to the numerical identification of nonlinear fluid film bearing parameters from large journal orbital motion (20–60% of the bearing clearance). Nonlinear coefficients were defined by a third order Taylor expansion of bearing reaction forces and were evaluated through a least mean square in time domain technique. In their analysis the journal response was obtained from a computational fluid dynamical model of a plain journal bearing on high dynamic loading conditions [90]. A.W.Lees et al. (2009) explained that vibration-based condition monitoring had become well accepted and widely used to identify faults in rotating machines. In the recent past, it has been observed that model-based identification has played a significant role in the rapid resolution and quantification of faults [91]. Two separate identification algorithms, for the simultaneous estimation of the residual unbalance and bearing dynamic parameters in a rigid rotor-bearing system, had been reported by R. Tiwari et al. (2009). Their first method uses the impulse response measurements of the journal from bearing housings in the horizontal and vertical directions, for two independent impulses on the rotor in these directions. Time-domain signals of impulse forces and displacement responses were transformed to the frequency domain, and were used for the estimation of the residual unbalance and bearing dynamic
parameters. Their second method employs the unbalance responses from three different unbalance configurations for the estimation of these parameters. Unbalance response measurements were taken for both the clockwise and counter-clockwise rotations of the rotor [92]. K.P. Gertzos et al.(2011) reported that to prevent catastrophic failure of a rotating system, it is necessary both to detect wear precisely, without shutting down and dismantling the machinery and to predict future replacement needs. In their analysis Computational Fluid Dynamics analysis was used to solve the Navier-Stokes equations. Diagrams of bearing characteristics such as relative eccentricity, attitude angle, lubricant side flow and friction coefficient versus Sommerfeld number were presented by them for various wear depths and used for online wear identification. A graphical detection method was analytically presented by them to identify the wear depth associated with the measured dynamic bearing characteristics [93]. S.X. Zhao et al. (2005) proposed that hydrodynamic journal bearings under large perturbations should be treated as nonlinear systems. Three kinds of nonlinear oil-film force models were proposed to denote the oil-film forces by retaining certain terms of Taylor series expansion of the oil-film force. In their analysis Least-mean squares method in time domain was proposed to identify the oil-film coefficients [94]. Hua Zhou et al. (2004) presented an experimental method to recognize set of linear stiffness and damping coefficients and establish their characteristics under varieties of operating conditions. The fundamental test model was obtained from a Taylor series expansion of bearing reaction force. In their analysis the coefficients were evaluated by means of least mean square in time domain [95]. M. Chouksey et al. (2012) studied the influences of internal rotor material damping and the fluid film forces (generated as a result of hydrodynamic action in journal bearings) on the modal behaviour of a flexible rotor-shaft system. Their investigation revealed that correct estimation of internal friction and the journal bearing coefficients at the rotor spin-speed were essential to accurately predict the rotor dynamic behaviour [96]. Keir Harvey Groves et al. (2010) used the Chebyshev polynomial fits to identify finite difference solution of the incompressible Reynolds equation. Their method manipulates the Reynolds equation to allow efficient and accurate identification in the presence of cavitation, the feed-groove, feed-ports, end-plate seals and supply pressure [97]. Two efficient calculation methods, based on the free boundary theory and variational
principles for the unsteady nonlinear Reynolds equation in the condition of Reynolds boundary, were presented by Zhipeng Xia et al. (2009). By employing the two mentioned methods in their analysis, the nonlinear dynamic forces as well as their Jacobians of the journal bearing can be calculated saving time but with the same accuracy [98]. An identification algorithm for simultaneous estimation of residual unbalances and bearing dynamic parameters by using impulse response measurements was presented by R. Tiwari et al. (2006) for multi-degree of freedom flexible rotor bearing systems. In their analysis the algorithm identifies speed dependent bearing dynamic parameters for each bearing and residual unbalances at predefined balancing planes [99]. Above researchers 80-99 used different identification algorithms for numerical identification of rotor bearing systems.[81] Applied a system identification method to obtain a nonlinear equation for an oil film rotating system.[82] Described system identification and data reduction methods used for extracting rotor dynamic coefficients of fluid-film journal bearings.[83] proposed an identification technique in the dynamic analyses of rotor–bearing–foundation systems called the pseudo mode shape method (PMSM).[84] Developed an identification algorithm for bearing dynamic characterization by using unbalance response measurements.[85] Studied the identification of parameters in rotary systems, namely, the unbalance magnitude, phase and position.[86] Discussed the non-parametric identification if MIMO rotor-bearing systems in the frequency domain for real as well as for complex modal testing.[87] Presented the dynamic analysis of a dual-disk rotor supported over a double-row ball bearing system.[89] Proposed an experimental method to identify coefficients and present their characteristics under various operational conditions.[90] Developed a numerical identification of nonlinear fluid film bearing parameters from large journal orbital motion. [91] Explained the vibration-based condition monitoring used to identify faults in rotating machines.[92] Developed two separate identification algorithms, for the simultaneous estimation of the residual unbalance and bearing dynamic parameters in a rigid rotor-bearing system.[93] used Computational Fluid Dynamics to detect and identify the wear precisely to prevent catastrophic failure of a rotating system.

In present work System Identification of hydrodynamic journal bearing system is carried out using oil film pressure measurement and oil film thickness measurement to determine
transfer function of hydrodynamic journal bearing system. This transfer function is further used for simulation of feedback control system and determines step response of same.

S.B. Glavatskih (2004) presented a method to improve temperature monitoring of fluid film bearings. His method was tested in an industrial tilting pad thrust bearing. Temperatures monitored by a thermocouple through the utilization of his proposed method were compared to those measured by thermocouples in the pad backing and in the collar. His method was also tested on a PTFE-faced bearing. His investigation revealed that the proposed method improves sensitivity to thermal transients in conventional babbit bearings and provides adequate means of temperature monitoring in the PTFE-faced bearings [100]. A method for on-line monitoring and control of hydrodynamic journal bearings using film thickness measurement was proposed by S. K. Roy Chowdhury (2000). An adequate film thickness was maintained at all times. The method uses the journal speed as the controlling parameter purely for demonstration purposes [101]. Sergei B. Glavatskih (2001) developed the application of an eddy current sensor with an active compensation for changes in sensor temperature to simultaneous monitoring oil film thickness and temperature in a tilting pad thrust bearing. In his study sensor design, calibration procedure, sensitivity and accuracy were described. Test equipment along with sensor mounting was also presented. In his analysis tests were run at different rotational speeds and bearing loads as well as different supplied oil flow rates to evaluate sensor performance in various operating conditions. During the tests film thickness and temperature were simultaneously measured. Temperatures were compared with data from thermocouples installed in the pads and thermistors mounted in the collar. Tests have shown that the sensor can successfully be used to reliably monitor the conditions within the bearing [102]. A method for detection of wear in thrust ball bearings coated with molybdenum disulphide MoS2 was presented by Ali Kahirdeh et al. (2010). They employed an energy feature obtained from time-frequency representation of the vibration signal. Extensive experimental studies were conducted by them to verify the efficiency of the proposed method for fault diagnosis of MoS2 coating [103].

T Akagaki et al. (2006) studied friction and wear behaviours of rolling bearing in contaminated oil containing white fused alumina particles. In their investigation the
friction and wear processes were monitored using wear debris analysis, such as ferrography and spectrometric oil analysis program, and vibration analysis [104]. V Hariharan et al. (2010) investigated the effect of contamination of lubricant by solid particles on the dynamic behaviour of rolling bearings. Silica powder at three concentration levels and different particle sizes was used to contaminate the lubricant. In their analysis experimental tests had been performed on the ball bearings lubricated with grease, and the trends in the amount of vibration affected by the contamination of the grease were determined [105]. I. F. Santos et al. (2003) investigated the theoretical and experimental behaviour of rigid rotors controlled by tilting pad journal bearings with active oil injection. In their analysis the global model of the system was obtained by coupling the equation of motion of the rigid rotor with the stiffness and damping of the active oil film [106]. M. S. Patil (2010) used RSM to study the influence of defect size, load, and speed on the bearing vibrations. Kurtosis was used as response factor by him. In his analysis experiments were performed using 6305 ball bearings [107]. Z.Q. Chen et al. (2005) developed an ultrasonic technique to monitor the dynamic behaviour of condensing and non-condensing fluid films. Both stable (upwards-facing) and unstable (downwards-facing) liquid films were examined by them. Their ultrasonic system utilized 5 MHz planar piston transducers operated in pulse-echo mode [108].

Above researchers 100-108 developed different techniques for monitoring and controlling different bearing parameters. [100] Developed a method to improve temperature monitoring of fluid film bearings used in an industrial tilting pad thrust bearing. [101] Developed a method for on-line monitoring and control of hydrodynamic journal bearings using film thickness measurement. [102] Developed the application of an eddy current sensor for changes in sensor temperature to simultaneous monitoring oil film thickness and temperature in a tilting pad thrust bearing. [103] Developed a method for detection of wear in thrust ball bearings coated with molybdenum disulphide.

In present work simulation of feedback control system is done in Matlab. Further feedback control system is developed for stability of oil film thickness in hydrodynamic journal bearings. Journal speed is considered as controlling parameter and oil film thickness is considered as controlled parameter.