CHAPTER 2
LITERATURE REVIEW

2.1 INTRODUCTION

The constant pressure to build stronger, less expensive, lighter weight, efficient and quieter running gears has resulted in a rich knowledge base for gear design. A survey of literature published on gears shows that a large number of aspects of gear design, manufacture and working have been addressed in the past. Out of these, a review of the following areas of research on gears is considered essential in order to assess the status of research pertaining to the design of gear drives.

- Gear tooth stress analysis
- Lubrication analysis
- Thermal analysis
- Design optimization

2.2 GEAR TOOTH STRESS ANALYSIS

A review of the published literature on gears shows that in-depth studies on the mechanical behavior of power transmission gears have been carried out in the past and the results have been well documented and utilized for improving the design of gears. Most of the early research works on gears pertained to the stress analysis of the different types of gears. These were classical approach to the problem of stress determination in gear teeth based on the cantilever beam theory, with additional semi-empirical stress concentration factors to estimate the root stress at the tooth fillet.
Tooth bending failure at the root of the gear tooth is a major concern in gear design. If the bending stress exceeds the fatigue strength of the gear material, the gear tooth has a high probability of failure. Another important concern in gear design is the surface fatigue failure. These two modes of failure are taken care of in all the gear design procedures by imposing the following constraints on the design:

- The induced bending stress in the gear should be lesser than the allowable bending strength of the gear material
- The contact stress induced in the gear tooth should be lesser than the permissible contact compressive strength of the material of the gear

All the standard procedures of gear design rate the strength of the gear teeth based on these stresses and include some additional safety factors that account for dynamic effects, quality of gears, tooth geometry and size of gears, notch sensitivity effects and temperature effects [1].

In the early research works, analytical approaches [2] and experimental techniques were adopted to investigate static and dynamic stresses and deflection in gears. The analytical techniques were approximate because of the numerous assumptions made to simplify the analysis. Experimental methods had the drawback of limited range of experimentation. The advent of computers and advances in numerical techniques saw large-scale application of numerical methods for solving design problems. Among them, the finite element method has been adopted to a very large extent for solving design and analysis problems on gears.
The analytical methods used for gear design are based on many assumptions and generally give the stress values at a particular point on the tooth. They do not give the distribution of stresses or deflection over the entire tooth. For instance, the Lewis equation considers the average tangential load applied at the midpoint along the face width of the gear tooth for the calculation of bending stress at the root of the tooth. Thus it gives only an approximate value of the bending stress induced at the root section of the tooth [2]. Also, the influence of the axial and radial loads on the bending stress is not considered. In the face of such limitations, the finite element method (FEM) offers itself as a powerful numerical tool for the stress analysis of gear teeth.

Chabert et al [2] have evaluated the stresses and deflection of spur gear teeth under strain using finite element method. They have first carried out an investigation that aimed at contour delimitation of the gear structure and concluded that it is enough to model and carry out analysis on a single gear tooth since the results do not deviate too much from the results of multi-tooth analysis. The delimited contour is shown in Figure.2.1. PQ and RS form the boundaries of the tooth that requires to be modeled. Using this concept of modeling, they have drawn formulas for the determination of deflection and stresses in spur gears of different ratios and of 20-degree pressure angle, from a two-dimensional finite element analysis

Finite element method has been adopted by Winter and Hirt [3] to determine the root stresses on gear tooth and the influence of fillet radius on stresses and tooth strength. The measurement of actual strains at the gear tooth root has been carried out using the photo elastic technique and the results have been used to validate the finite element model developed.
Wallace and Seireg [4] have carried out computer simulation of dynamic stress and deformation in spur gears using two-dimensional finite element method. They have investigated the effect of impulsive loads applied at different points on the tooth surface and moving loads normal to the tooth profile, on the pattern of stress and deflection of the tooth. The accuracy of the simulation technique has been verified using a simple experiment on a cantilever plate.

Ramamurti and Gupta [5] studied the application of the finite element method to spur gear tooth stress analysis. The tooth was treated as a plane stress model. The study was confined to a single tooth with fixed boundaries.

Oda et al [6] studied the root stress distribution in a thin rim spur gear using a two-dimensional finite element model. Triangular elements were used in their analysis. The
effect of boundary constraints of adjacent teeth on the root stress distribution was studied.

Chong et al [7] investigated the influences of radius of curvature of root fillet, the width of the bottom of the tooth space, the pressure angle and the loading position on the root stress distribution.

The effects of the shape of the tooth on the root stresses and tooth deflection of a thin rimmed gear with ribs and webs were studied by Syama et al [8] using three dimensional finite element method.

Ramamurti and Rao [9] proposed a method for the stress analysis of spur gear tooth by finite element method using cyclic symmetry concept. Each tooth was treated as a two-dimensional plane stress problem. The stress distribution in a loaded tooth and its influence on adjacent teeth were studied.

Simon [10] has applied the finite element method for the analysis of stress in double enveloping worm gears. The influence of the design parameters on the deflections and stresses has been investigated. On the basis of the results obtained, equations have been derived for the calculation of deflections and stresses in the worm thread and in the tooth of the worm wheel using regression analysis and interpolation functions.

Vijayarangan and Ganesan have studied the behaviour of composite spur gear [11] and composite bevel gear [12] from a static load point of view using three-dimensional finite element method. The performance of composite gears is compared with that of carbon steel gears. They have also made an attempt to study the contact stress of a pair of
mating gear teeth under static conditions using two-dimensional finite element method [13]. The variation of contact stress along the contact surface in a direction normal to the mating surfaces is presented.

Bibel et al [14] have carried out three-dimensional stress analysis of spiral bevel gears using multi-tooth finite element model. Four gear teeth and three pinion teeth have been modeled and the surface contact stress has been evaluated using gap elements. The results have been found to be in good agreement with calculated Hertzian contact stress.

Ramamurti et al [15] have demonstrated the use of cyclic symmetry concept in the finite element analysis of rotationally periodic structures like gears. They have applied this concept for the static and dynamic analysis of spur and bevel gears. Static stresses and deflection, and the first ten natural frequencies and mode shapes have been obtained based on their analysis.

Litvin et al [16] have proposed a methodology for the determination of load share, contact ratio, precision of motion and stress analysis of loaded gear teeth. Loaded tooth contact analysis is carried out by applying the finite element method and the load sharing between the teeth, the real contact ratio and the stresses developed in the teeth are determined. The method developed by them has been demonstrated on a hypoid gear.

Umeyama et al [17] have proposed a method of calculation of the loaded transmission error of helical gears using a finite element model of an equivalent spur gear tooth. Loaded transmission error of a tooth pair is determined as the sum of the bending
deflection, the approach deflection and the amount of tooth surface modification. The bending deflection in the tooth is predicted using finite element method and the approach deflection using Hertzian formulas. The investigations have revealed that the transmission error is the maximum under no load condition.

Static and dynamic tooth loading in spur and helical geared systems has been investigated by Baud and Velex [18]. A numerical model has been developed based on a finite element procedure for the prediction of dynamic tooth loads in geared rotor systems. The gear pair is modeled by two rigid cylinders linked by a series of stiffnesses distributed along the potential lines of contact on the base plate, as shown in Figure 2.2. Quasi-static and dynamic stresses at the root of spur and helical gears have been obtained and experiments have been conducted for validating the numerical model.

Shuting [19] has adopted three-dimensional finite element method for the tooth contact analysis of three-dimensional thin-rimmed gears. The aim was to obtain three-
dimensional tooth load distributions, tooth root stresses and the tooth contact pattern. The calculated results are verified by conducting experiments.

A method for the three-dimensional finite element analysis of long-fiber reinforced composite spur gears has been developed by Cagdas Alagoz et al [20]. The gear is modeled with two distinct regions, the long-fibre reinforced region and the chopped-fibre reinforced region. The gear tooth stresses are determined and are compared with the results given for composite gears by Shiratori et al [21].

Kim et al [22] have adopted the finite prism method, a particular case of the finite element method, to design external cylindrical spur gears with a web. Based on their investigations on the load sharing, pressure distribution, meshing stiffness and three-dimensional tooth fillet stresses, they have derived formulas to estimate the maximum principal stress in the root of the tooth.

Katsuzo et al [23] have developed a method for the determination of load distribution, Hertzian contact pressure and oil film thickness in worm gear drives. Empirical equations are formulated for tooth deflection based on measurement of tooth deflection using laser holographic interferometry. The matrix of influence coefficient is formulated in order to calculate root stress of the worm and worm wheel.

More recent works have concentrated on the prediction of crack propagation in gear teeth. Analytical methods have been developed to estimate the stress intensity factors in gears, which are the key parameters for the estimation of the characteristics of cracks [24, 25]. These estimations are based on linear elastic fracture mechanics, Numerical techniques such as finite element and boundary element methods have been employed.
to study crack propagations in gears. Based on the stress intensity factors estimated using the linear elastic fracture mechanics concept, fatigue crack growth and gear life predictions have been investigated by Glodez [26]. A study to determine the effect of moving tooth load on crack propagation in gears has been carried out by David Lewicki et al [27] in a recent work. They have adopted the finite element method for two-dimensional analysis on spur gears and the boundary element method for three-dimensional analysis on spiral bevel gears and have compared their results with experimental results.

2.3 GEAR LUBRICATION ANALYSIS

Lubricating the meshing zone of gear teeth is the primary function of gear oil. In addition, it has to provide cooling also, so that excessive surface temperatures are not attained. This can be achieved by the use of appropriate oil and appropriate lubricant application procedures. Proper understanding of the phenomenon of lubrication is essential so that lubrication breakdown can be avoided.

When lubricating oil is supplied to the meshing zone, it is flung off and the teeth faces enter the meshing zone starved of oil. This fling-off of oil is an adverse effect as far as hydrodynamic lubrication of the teeth is considered. On the other hand, it is a desirable effect since a part of the heat generated at the mesh is absorbed and carried away by the flung oil. Considerable amount of research has been done on the fling-off of oil in gear systems.

Fundamentals of fling-off cooling of gear teeth have been investigated by DeWinter and Blok [28]. In their work, they have formulated and evaluated the hydrodynamic and thermal aspects of intermittent cooling in spur gears using analytical method and solved
using numerical method. They have shown that the heat flung-off through the terminal tip section is lesser than the heat flung off from the face. For spur gears, an upper bound to the amount of fling-off heat from the tooth face concerned per unit of its width has been established as

\[ Q_{\text{ft}} = 5.6mb\theta_s\omega^{-0.5} \]  

...(2.1)

where, \( m \) is the tooth module, \( b \) the thermal contact coefficient of oil, \( \theta_s \) is the difference between the gear tooth temperature and the oil supply temperature, and \( \omega \) is the angular velocity of the gear under investigation. Any consistent set of units can be adopted for the above equation.

van Heijningen and Blok [29] have investigated continuous fling-off cooling of gear teeth by supply of oil to the root of the tooth and compared with the results of intermittent fling-off cooling. The investigations reveal that the cooling capacity of the intermittent fling-off cooling is adversely affected by the choice of oil having a lighter viscosity grade and/or steeper fall of viscosity with an increase in temperature. The reverse proved true of the continuous fling-off cooling by root supply. In this type of cooling, a higher viscosity of oil results in a slower flow of coolant, and a correspondingly longer thermal exposure of the oil, which will more than offset the usual adverse effect of an increase of viscosity on the forced convective heat transfer coefficients.

Lubricant jet flow phenomenon has been studied by Akin et al [30] to predict the lubricant jet flow impingement and penetration depth into a gear tooth space. This investigation is aimed at the determination of how much of the impinging oil is involved in the lubrication and cooling processes and how much of it is flung off. An analytical model
has been developed for the vectorial impingement depth and compared with experimental results.

A hydrodynamic analysis of fluid flow between meshing spur gear teeth has been carried out by Wittbrodt and Pechersky [31]. A numerical model based on one-dimensional flow has been developed and the computed results include the velocity of the fluid and the pressure and temperature at the mesh region.

2.4 ELASTO-HYDRODYNAMIC ANALYSIS OF GEARS

Elastohydrodynamic (EHD) lubrication is a form of hydrodynamic lubrication where elastic deformation of the lubricated surfaces becomes significant. A high load carried by a small area, as between the ball and race of a rolling contact bearing and that between meshing gears, causes elastic deformation of the bodies in contact. The elastic deformation of the contacting bodies under load as well as the pressure-viscosity characteristics of the lubricant is to be incorporated in solving the lubrication problem.

Bathgate and Yates [32] describe the application of elementary line-contact elastohydrodynamic lubrication (EHL) theory of worm gear together with calculations of flash and total contact temperature. Discharge voltage measurements of film thickness in a worm, which were calibrated in a disk machine, suggested values in the range of 0.03-0.3 μm with the particular oil used.

Fuan et al [33] also applied line contact EHL theory to a worm gear and predicted film thickness values of 0-2.5 μm, and concluded that lubrication in the middle part of the contact area is weak because of poor entrainment conditions in this region.
Wellauer and Holloway [34] have shown how EHD film thickness theory can be applied to the assessment of industrial gearing. Oil film thickness and surface texture of the gear tooth are important factors that affect gear tooth surface performance. The correlation of gear tooth oil film thickness and surface texture with surface distress is made using a specific film thickness $\lambda$, which is defined as the ratio of film thickness to the composite surface texture. A nomograph, utilizing readily available gear geometry and operating and lubricant parameters is developed which allows rapid determination of gear tooth oil film thickness for a wide range of gear drive conditions.

Jackson [35] adopted the thermal EHL analysis of Murch and Wilson [36] to produce simple charts for the determination of thermal correction factors for the isothermal film thickness calculations in gears and rolling element bearings.

A full thermal EHL model of worm contacts has been published by Simon [37, 38]. Finite difference method has been employed for the simultaneous solution of Reynolds's, elasticity, energy and Laplace equations. Results are given in terms of performance curves using non-dimensional flash temperature, EHL load carrying capacity, and friction factor ratios.

Investigations on the prediction of elastic contact and EHD film thickness in worm gears have been carried out by Kong et al [39]. Three-dimensional elastic simulation technique has been adopted for the calculation of true area of elastic contact under load and the contact stress. EHD lubrication effects have been investigated using a special non-Newtonian thermal solver for predicting the lubricant film thickness.
2.5 THERMAL ANALYSIS OF GEARS

Since gears are now being used for higher power transmission and at high speeds, temperature rise of the gear system is a major concern for the designers. High gear tooth surface temperatures may lead to scoring or scuffing failures. Also, thermal expansion of the components of the gear system may result in large values of stresses. In order to avoid failure of the gears due to scoring or high thermal stresses, knowledge of the thermal characteristics of the gear drive units and the influence of various operating variables on the temperature of the gears is essential. In this context, thermal analysis of gears becomes an important step in the gear design process. The thermal aspects of gears can be accounted for, by the flash temperature and the bulk temperature of the gear teeth, and, the lubricant oil temperature.

2.5.1 Flash Temperature Analysis

The most critical factor in hot scoring of gear tooth is the flash temperature. This is a momentary peak temperature at the contact surfaces between gear teeth. When it occurs, the mating teeth destroy the oil film between them, and instantaneously weld together. Then, as the gear teeth rotate out of contact, the welded bond breaks and the tooth surface fractures.

H. Blok [40] was the first to determine the contact temperature of the gear teeth. He used the method of the heat source in order to solve the equation of heat conductivity for the case of short operation of gear transmissions. He adopted the hypothesis of the “thermal shock” as the basis of his method, i.e., the instantaneous transmission of heat at each point of contact, which the moving heat source touches at a given instant. The outcome of this research work was a simple and approximate equation for flash temperature based on one-dimensional heat flow.
A numerical solution to the dynamic load, film thickness and surface and flash temperatures in spur gears has been developed by Wang and Cheng [41, 42]. Part I [41] of their study is concerned with the study of dynamic loads and the effect of various geometric and operating parameters on the tooth bulk and flash temperatures and Part II [42] investigates the lubricant performance. From their investigations, it is found that the dynamic load distribution deviates drastically from the static load distribution. The results show that an increase in the face width of the gears results in an increase in both the maximum equilibrium surface temperature and the flash temperature and hence a decrease in the oil film thickness. Increased pitch line velocity and increase in viscosity of the lubricant result in a marked improvement in film thickness.

Toshimi Tobe and Masana Kato [43] have carried out investigations on the flash temperature of spur gear teeth. They have used a moving heat source for the analytical determination of flash temperature and have studied the effect of tooth load, speed of revolution, module and profile modification on the flash temperature. They have found that high flash temperatures occur at the start and end of the gear mesh, where the radius of curvature and sliding velocity are relatively large. The flash temperature is found to increase linearly with the applied load and the increase is non-linear with increase in speed and module. Profile modification also results in a decrease in flash temperature and hence can be an effective means to reduce temperature.

Bathgate J & Yates F [32] carried out analytical and experimental investigations on the application of film thickness, flash temperature and surface fatigue criteria to worm gears. They have shown how accurate prediction of scuffing and pitting may be made at the design stage itself using their criteria. They have modified the formula derived by Blok [40] for the prediction of flash temperature of spur gears so that it can be adopted
for worm gears. Experiments have been conducted for the measurement of film thickness using voltage discharge technique.

Temperature rise in plastic gears has been investigated by Gauvin et al [44]. A mathematical model has been developed from basic heat transfer theory which estimates the convective heat transfer coefficients based on measured gear tooth surface temperatures. A running gear was preheated to a known temperature and the drop in the surface temperature of the tooth measured as it loses heat to the surroundings by convection. These measured values are used in the mathematical model developed to evaluate the heat transfer coefficients.

2.5.2 Bulk Temperature Analysis

The contribution of the gear tooth bulk temperature to the scoring phenomenon and its contribution in bringing about the onset of this failure mode have been studied by Akin [45] and De Winter [28].

Nadir Patir and Cheng [46] have adopted finite element method for the prediction of steady state bulk temperature of spur gear teeth. Different heat transfer coefficients have been used for different regions of the gear tooth. To study the effect of different geometrical and operating parameters on the bulk temperature of the gear tooth, three dimensionless parameters have been derived. The dimensionless bulk temperature \( \theta \) is studied with respect to the dimensionless heat transfer coefficients \( H_s \) and \( H_t \) and, \( N \), the number of teeth on the gear. Here, \( H_s \) and \( H_t \) are the convective heat transfer coefficients on the gear hub surface and the tooth face respectively. It has been established that, the temperature gradients in the tooth are highly dependent on the cooling rates \( H_s \) and \( H_t \). These convective coefficients control the mode of cooling, i.e., whether the gear is
cooled more through the hub surfaces or through the tooth faces. It is also found that high temperature is confined to a thin layer below the meshing tooth face.

Figure 2.3 Isotherms on gear tooth obtained by Townsend and Akin [47]

Townsend and Akin [47] have carried out gear tooth temperature analysis using finite element method. The work was based on a calculated heat input and estimated heat transfer coefficients for the different parts of the spur gear tooth that are oil cooled and air-cooled. The isotherms on the tooth surface show that the maximum temperature occurs near the pitch point. A typical pattern of isotherms obtained by them is shown in Figure 2.3. It is found that the tooth surface temperature increases linearly with increase in load. Increase in oil jet pressure and jet size help in reducing the surface temperature. There is an increase in temperature due to an increase in speed. The results of the numerical model are compared with experimental results and it is found that there are large differences at lower loads and speeds.
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Fumio Obata, et al, [48] have examined the effects of various factors on the bulk temperature rise of the working tooth flank when a moving heat source with varying heat quantity is supplied repeatedly at regular intervals to the surface of an isotropic and homogeneous rectangular body used as the model of a spur gear tooth. The results indicate that the rise in bulk temperature of the tooth is inversely proportional to the heat supply interval. It is also reported that the size of the heat supply region and the moving direction and velocity of the source have no effect on the temperature rise.

Handschuh and Kicher [49] have developed a model for analyzing three-dimensional thermal behaviour of spiral bevel gears. Steady state and transient state thermal behaviour have been analysed. Three-dimensional contact analysis has been performed to obtain the contact ellipse and time averaged heat flux is used in their finite element analysis to obtain the temperature distribution in the tooth of the bevel pinion.

Terauchi et al [50] have investigated the heat balance of gear equipment for predicting the bulk temperature rise of gears and the temperature rise of oil with dip cooling.

2.6 GLOBAL THERMAL ANALYSIS OF GEARBOX

Most of the research work carried out on the thermal characteristics of gear drives focus on one mechanical element isolated from its environment. A gearbox is composed of many elements that act as sources of heat as well as take part in the heat dissipation process. Hence, in order to obtain a more accurate prediction of the thermal behaviour of gearboxes, it is necessary to include all the major elements of the system in the model developed. This will aid the determination of the global thermal behaviour of the gearbox unit.
Only a few theoretical and numerical studies deal with the global thermal behaviour prediction of gear transmissions. Numerical techniques such as the finite element method and the thermal network method have been attempted for the prediction of global thermal behaviour of gearboxes. However, these approaches require assumptions about oil flows along the inner walls of the gearbox housing. Oil flow observations have been made by Joule et al [51, 52] and others, and in all cases, reasonable hypotheses for oil flow definition are made for the thermal model definition.

Joule et al have studied global thermal behaviour of spur gear box by modeling all the elements of the gearbox. In the first part of their study [51], they have carried out a steady state thermal analysis of a spur gear box using the finite element method. The gears have been represented as plain discs. The geometrical arrangement of their gearbox is shown in Figure.2.4. The physical measurement of the total power loss in the test gearbox was made and the bearing power losses are estimated using Palmgren's equation. The difference between these two losses is calculated as the power loss due to tooth friction and churning of oil. This forms the frictional heat input to the gearbox modeled using FEM. This model of the gearbox is analysed to predict the steady state temperature distribution in the gearbox.

The second part of the study carried out by Joule et al [52] pertains to the transient thermal behaviour of a spur gearbox. Two different transient conditions have been investigated. The first is continuous running from start-up until steady-state conditions are attained, and the second is a start-stop-restart cycle. The thermal boundary conditions are arrived at based on experimental measurements coupled with appropriate theory.
Thermal network method of analysis has been adopted by Manin and Play [53] for investigating the global thermal behaviour of power gearing transmission. In the thermal network method, each component of the gearbox is modeled as a thermal finite element and a network is established to model the entire gearbox. Thus, this method permits the prediction of the global thermal behaviour of the gearbox. The results obtained from this numerical model have been compared with those obtained by FEM and experiments. The comparison shows that the mean surface temperature obtained by the thermal network method differs slightly from those obtained by FEM, but the bulk temperatures are almost the same. It is found that the temperature of the gear wheel increases linearly with pitch line velocity and the applied torque. The temperature of the lubricant is found to increase linearly with the power transmitted.
2.7 OPTIMUM DESIGN OF GEAR DRIVES

The current trend in gear design is to design smaller, lighter weight gears, having higher capacity for tooth loads, speeds and wear resistance. These criteria are often conflicting with one another and the designer has to make a compromise to obtain the best design to suit a set of practical design constraints.

The research work carried out in the past few decades have resulted in the evolution of many approaches for improved design of gears. The use of optimization techniques is one among them, which has received considerable attention.

Different approaches have been adopted by various researchers for the optimal design of gears. Errichello [54] adopted a hand calculation procedure for the determination of minimum weight gears. Savage et al [55] considered involute interference, contact stresses and bending fatigue. They concluded that the optimal design usually occurs at the intersection point of curves relating the tooth numbers and diametral pitch required to avoid pitting and scoring. Carroll and Johnson [56] expanded the model to include the AGMA geometry factor and AGMA dynamic factor in the tooth strength formulas. Their analysis found that the theoretical optimal gear set occurred at the intersection of the bending stress and contact stress constraints at the initial point of contact. Design of gears to minimize wear has been investigated by Gay [57].

Later works have expanded to include a wider range of considerations. Andrews and Argent [58] approached the optimal strength design for non-standard gears by calculating the hob offsets to equalize the maximum bending stress and contact stress between the pinion and gear. Savage et al [59] in their work have treated the entire transmission as a complete system. In addition to the gear mesh parameters, the
selection of bearing and shaft proportions were included in the design configuration. Wang HL and Wang HP [60] have introduced a mathematical formulation and an algorithm to solve the multi-objective gear design problem, where feasible solutions can be found in a three-dimensional solution space.

Most of the literature on optimum design of gears deals primarily with static tooth strength. These studies are based on the Lewis formula assuming that the static load is applied at the tip of the tooth. Some considered stress concentration and the AGMA geometry and dynamic factors. Ping Hsun Lin et al [61] have included the effect of operating speed to design compact spur gear sets giving the minimum dynamic response. Since root fillet stress is important in determining tooth-bending failure in gear transmission, they adopted a modified formula that includes dynamic considerations. Constraint criteria employed for their investigation include involute interference limits combined with the tooth bending strength and contact stress limits. Their study was limited to spur gears with standard involute tooth profile.

Saravanan et al [62] have adopted genetic algorithm and simulated annealing for the optimal design of gears with the objective of maximizing transmission efficiency and minimizing composite errors and weight.

Choy et al [63] have carried out the analysis of multistage gear systems coupled with gearbox vibrations. Takatsu et al [64] have adopted a building block approach for the analysis of vibration in a single stage gearbox. Inoue et al [65] have attempted the optimum design of gearboxes for low vibration levels.
2.8 SUMMARY OF THE OBSERVATIONS

The observations made in the survey of literature on gears can be summarised as follows.

2.8.1 Investigations on Gear Tooth Stresses

A large number of researchers have carried out research on the different types of gears in order to determine the stresses and deflections in the gear teeth to a better accuracy. Different techniques like analytical methods, numerical methods and experimental methods have been tried out. Among these, the finite element method, a very powerful numerical method, has been adopted to a very large extent. Experimental methods like strain measurements and photo elasticity have been used to validate the numerical methods.

Many of the published work have attempted to determine the static and dynamic stresses and deflection in gear teeth. Their major aim was to determine the bending stress and deflection, root stress and the surface contact stress in the gear tooth. The influence of tooth shape, tooth modification, speed ratios, pressure angle, radius of curvature of root fillet, etc, on stresses and deflection, load share, root stress, contact ratio, precision of motion and load transmission error have been investigated.

Investigations on crack propagation in gears and prediction of gear life have also been carried out. Studies on these lines have been done on all the type of gears, like spur, bevel, helical and worm and worm wheel drives. Attempts have also been made to study stresses and deflection in composite gears.
It can be seen that, all the above-mentioned work primarily aim at the determination of stresses and deflections of gear teeth due to the mechanical load transmitted. The stresses set up in the gears due to high temperatures and thermal gradients are not addressed. In high-speed and heavy-duty gears and in worm gears, significant amount of heat is generated due to friction, and the gears operate at high temperatures. The uneven temperature distribution in the gear tooth in the longitudinal direction causes the tooth load to be uneven, which frequently results in failure. Moreover, since the gearbox consists of a number of elements made of different materials, there will be differential expansion of the parts, and thus there is likelihood of inducement of large values of thermal stresses. Failures of gears due to these reasons can be minimized by carrying out a temperature analysis in the gearbox followed by thermal stress analysis.

2.8.2 Investigations on Gear Lubrication

Lubricant is a main component of a gear system and selection of lubricating oil is an essential design element. Lubrication function is the primary function of the gear oil to protect the tooth faces from excessive heat generation by tooth friction, and from various kinds of wear such as scoring and pitting. In addition to this main function, the oil also has to remove the heat from the meshing zone and thus act as a cooling agent.

From the foregoing literature review, it is evident that many research works have concentrated on the selection of lubricating oil and the method of application of the oil in order to avoid lubrication breakdown in service. Studies have also been carried out to determine the amount of heat that is carried away by the oil. In this context, numerical and experimental analyses have been carried out in a number of research works on the intermittent and continuous fling-off cooling. Elastohydronamic theory also has been developed to predict the film thickness and pressure and velocity of the fluid at the
meshing zone, and the effect of the gear geometry on these parameters. Nomographs have been developed, which can be used to estimate oil film thickness under a wide range of gear drive conditions.

In all the above analysis, knowledge of the temperature of the gear tooth is essential. Determination of the cooling capacity of the lubricating oil requires knowledge of the amount of heat removed by the fling-off process. The determination of oil film thickness using the EHD lubricant analysis requires the effective viscosity within the gap between the meshing gears to be known. This in turn, requires the temperature at the conjunction to be known. It is observed from the survey of the literature presented above, that the investigations have been carried out based on assumptions of the bulk temperature of gear teeth. Thus, it is seen that temperature analysis of the gear teeth becomes essential to know the performance of the lubricant selected, in respect of its lubricating and cooling capacities.

2.8.3 Investigations on Thermal Aspects of Gears
The study of the literature shows that considerable amount of work has been done on the flash and bulk temperatures of gears. But most of these pertain to spur gears and not much work has been reported on worm gears. Also, there is only a limited literature on global thermal analysis of gearboxes that takes into analysis all the major components of the gearbox. In many of the thermal analysis of gears that aim to predict the temperature distribution, the inputs are based on experimental measurements of power loss. Hence, in order to make an analysis using the above methods, physical tests have to be conducted, which means that the prototype should be available. But in order to make a performance evaluation at the design stage, experiments are to be
conducted on virtual computer models, with calculated estimates of heat losses and convective heat transfer coefficients.

2.8.4 Investigations on Optimisation of Gear Design
The survey of published literature on the optimal design of gears shows that in many of the attempts on design optimisation of gears, the main objective is to minimise the size and weight of the gears and maximize the transmission efficiency. These designs imposed constraints on involute interference, contact stresses and bending fatigue. Design of gearboxes for low vibration levels also has been attempted. The survey shows that these attempts on design optimisation of gears have addressed most of the mechanical aspects of concern but consideration of thermal aspects is lacking. Constraints on temperature rise of gear teeth or of the lubricating oil have not been imposed in most of these works. Since there are many instances of failures of gears due to temperatures exceeding safe limits, it is imperative that objectives and constraints be framed based on temperature rise of the elements of the gearbox too.

2.9 CONCLUDING REMARKS
The review of literature presented above shows that a large portion of the work related to gear drives has addressed the mechanical requirements. The design optimizations proposed have mechanical aspects such as maximum efficiency, minimum size and weight, minimum vibration levels, minimum interference, etc as major objectives. Factors like induced bending stress, contact stress, etc are considered as constraints. But, it is found that thermal aspects like temperature rise of gears are not given much significance. But heat generation and the resulting temperature rise are important factors that affect the performance of gears, especially in high power transmission drives and
worm gear drives. Hence, it is very important that these factors are addressed in the design of gear drives.

Prediction of temperature rise in a gear box under operating conditions in any application before the completion of the design will help to improve and arrive at a design that will meet the loading conditions without exceeding the temperature limits. This will avoid the underrating of gearboxes to ensure lower operating temperatures. Hence, the development of a comprehensive design procedure that includes thermal aspects along with the mechanical strength requirements is an important requirement.