# "THERMAL ANALYSIS OF HELICAL GEAR BOX MODEL PE-50"

**A Major Project Report**

*Submitted in Partial Fulfillment of the Requirements*
*for the degree of*

**MASTER OF TECHNOLOGY**
**IN**
**MECHANICAL ENGINEERING**

**(CAD/CAM)**

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**May, 2006**

# CERTIFICATE

This is to certify that the Major Project entitled "**Thermal Analysis of Helical Gear Box PE-50**" submitted by **Mr. Amit Dubey (04MME002)**, towards the partial fulfillment of the requirements for the degree of **Master of Technology in Mechanical Engineering** (**CAD/CAM**) of Nirma University of Science and Technology, Ahmedabad is the record of work carried out by him under my supervision and guidance. In my opinion, the submitted work has reached a level required for being accepted for examination. The results embodied in this major project, to the best of my knowledge, haven't been submitted to any other university or institution for award of any degree or diploma.

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## ABSTRACT

The model PE-50 is a product of ELECON Engineering Co. Ltd. PE- 50 is a four stage Gear Box with helical Gears. As the main problem of a gear industry is dissipation the heat generated inside the gear box, for this purpose Thermal analysis is required. Objective of present study is determination of thermal rating and power losses of the gear box and determination of temperature distribution of gear box casing. The tool used for this purpose are Pro/ENGINEER Wildfire 2.0 for solid modeling and ANSYS-10.0 for analysis.

Thermal Analysis of gearbox is carried out in Software ANSYS with transient and steady-state module and the temperature distribution along whole gear box is observed. During the operation of gear box, rise in temperature of lubricating oil due to total power losses inside the gear box is obvious. Heat from lubricating oil is transferred to the casing. Temperature at selected points on the casing is measured after eight hours running of the gear box. With this as input, cooling pattern is observed on ANSYS after two hours interval of time.

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## NOMENCLATURE

| | | |
|---|---|---|
| ø | = | Normal pressure angle |
| μ | = | Bearing coefficient of friction |
| φ | = | Helix angle |
| ν | = | Pitch line velocity (m/sec) |
| B | = | Face Width (m) |
| K | = | contact Load Factor |
| M | = | Mesh Mechanical Advantage |
| $\varphi_w$ | = | Working helix angle |
| $M_G$ | = | Stage ratio |
| $ø_t$ | = | Transverse pressure angle |
| m | = | Module (m) |
| $Z_w$ | = | No. Of Teeth In Wheel |
| $C_1$ | = | Lubrication factor |
| $H_s$ | = | Sliding ratio At Start of Approach |
| $H_t$ | = | Sliding Ratio At end of recess |
| $D_w$ | = | Reference diameter (m) |
| N | = | speed (RPM) |
| $C_p$ | = | Absolute Viscosity In Centipoises |
| $P_{in}$ | = | Input Power (watt) |
| $F_L$ | = | Radial Load (N) |
| $d_i$ | = | the inner dia of Bearing (m) |
| $d_o$ | = | the outer dia of Bearing. (m) |
| $Z_p$ | = | No. Of Teeth In pinion |
| $P_{cl}$ | = | Churning loss (watt) |
| $P_{wl}$ | = | Windage loss (watt) |
| $P_{bl}$ | = | Bearing Loss (watt) |
| $P_{ml}$ | = | Mesh loss (watt) |
| $T_p$ | = | Torque on pinion (N-m) |
| $N_p$ | = | Speed of pinion (RPM) |
| $R_o$ | = | Outer radius of gear (m) |
| $R_w$ | = | Reference radius of wheel (m) |
| $r_o$ | = | Outer radius of pinion (m) |

| | | |
|---|---|---|
| $r_w$ | = | Reference radius of pinion (m) |
| $P_n$ | = | Diameteral pitch (m) |
| A | = | Arrangement constant |
| $P_{wl+cl}$ | = | Sum of windage and churning loss (watt) |
| F | = | Surface Area ($m^2$ ) |
| $K_a$ | = | Thermal conductivity (kcal/ hr $m^2$ $^oC$) |
| $\eta$ | = | Efficiency of Gear box |
| $T_g$ | = | Gear box Temperature ($^oC$ ) |
| $T_a$ | = | Ambient temperature ($^oC$ ) |
| $P_g$ | = | Thermal heat Dissipation (watt) |

# CHAPTER 1

# INTRODUCTION

# INTRODUCTION

Helical gear boxes are having constantly sliding motion is going on between gear and pinion, this produces high frictional losses, which is converted in heat and results in temperature rise. This Temperature rise should be within permissible limits, otherwise causes thermal failures of gear box like scuffing of gear tooth reduction in the viscosity of lubrication oil, and reduction in the heat transfer rates which causes reduction in the efficiency of gear box. This is due to high heat generation that can raise the temperature in various parts of the helical gear box to higher level as well as the lubricant temperature to unacceptable levels when the box is operated continuously in practice.

The Gear Box PE-50 is a four stage Gear box having helical gears. 50 represents the centre distance and PE for four Stage as from ELECON Gear Pvt. Ltd.

ELECON was incorporated with 1951 initially at Bombay and was later shifted to V.V.Nagar 1961. initially it used to Manufacture material handling equipments only, but later it diversified into many other product such as reduction gear Boxes ,gear motors, mining equipments etc. The gear division of ELECON was established in 1963 to manufacture reduction gear boxes only. Today ELECON have a wide range of worm, parallel shaft and right angle helical, spiral bevel speed reduction units. ELECON gear division ISO9001 standards certified by RWTUV Germany since 1994.

As the biggest problem of a gear industry is to dissipate the heat which is generated within the gear box. This is depending on the maximum temperature of the lubricating oil. As to do the thermal analysis of any gear box, the main work is to calculate the various power losses. The main power losses in lubrication are the tooth losses, churning losses and bearing losses. The various governing equations are to taken in use by which these losses are calculated

Before going to the detailed thermal analysis very first step is to understand the need of thermal analysis, the influence of lubrication, when the gear box is loading condition various failure modes occur. These modes are wear, scuffing, micro pitting and pitting, these modes are affected by lubrication properties, oil temperature and film thickness.

In this present work heat losses are calculated and validated with the losses from AGMA. The heat generated is taken as a load and a methodology is established to predict the temperature distribution inside of gear box through steady and transient analysis and results are temperature are compared  in various locations inside and out side of gear box.

# CHAPTER 2

# LITERATURE REVIEW

# LITERATURE REVIEW

## 2.1 THERMAL ASPECTS OF GEAR LUBRICATION [2]

J. Bathgate[2] represents the theoretical and experimental procedures applicable to the thermal analysis of gear units. The life of the seals, bearings, gear materials and other components can be seriously reduced by excessively high operating temperatures. Many seals, for instance, only have acceptable lives at temperatures below $100^0$C; hardening and Cracking of the lips can occur above this temperature, resulting in lubricant leakage. Thermal degradation of the lubricant is related closely to its bulk temperature. A frequently used rule-of-thumb is that the life of a mineral oil is halved for each 18°F rise in temperature. Present industrial gear standards indicate that $200^o$F is a commonly accepted upper limit. A thermal network is proposed which enables the overall temperature distribution to be expressed in terms of local conditions of heat generation and conductance. Consideration is also given to the temperature limitations imposed on the gear unit with regard to component properties, lubrication and unit performance.

Any attempt to predict temperature levels at various locations within a gear Unit must recognize the interdependence of all parts of the system. The basis of any thermal analysis is, therefore, to represent the heat flow paths as a network of thermal resistances and to relate each one to the local conditions on which it depends. The complexity of the idealization will vary greatly according to the type of unit. A complex example might be a multi-reduction unit with several more-or-less distinct gear compartments with widely differing conditions of heat generation, heat transfer and temperature. The presence of an integral motor will require representation in the system both as a heat source and as a complication to the heat flow paths. If a cooling fan is fitted; this will necessitate distinction between the regions of the external surface over which the fan creates forced convection and the remainder under the influence of Natural convection

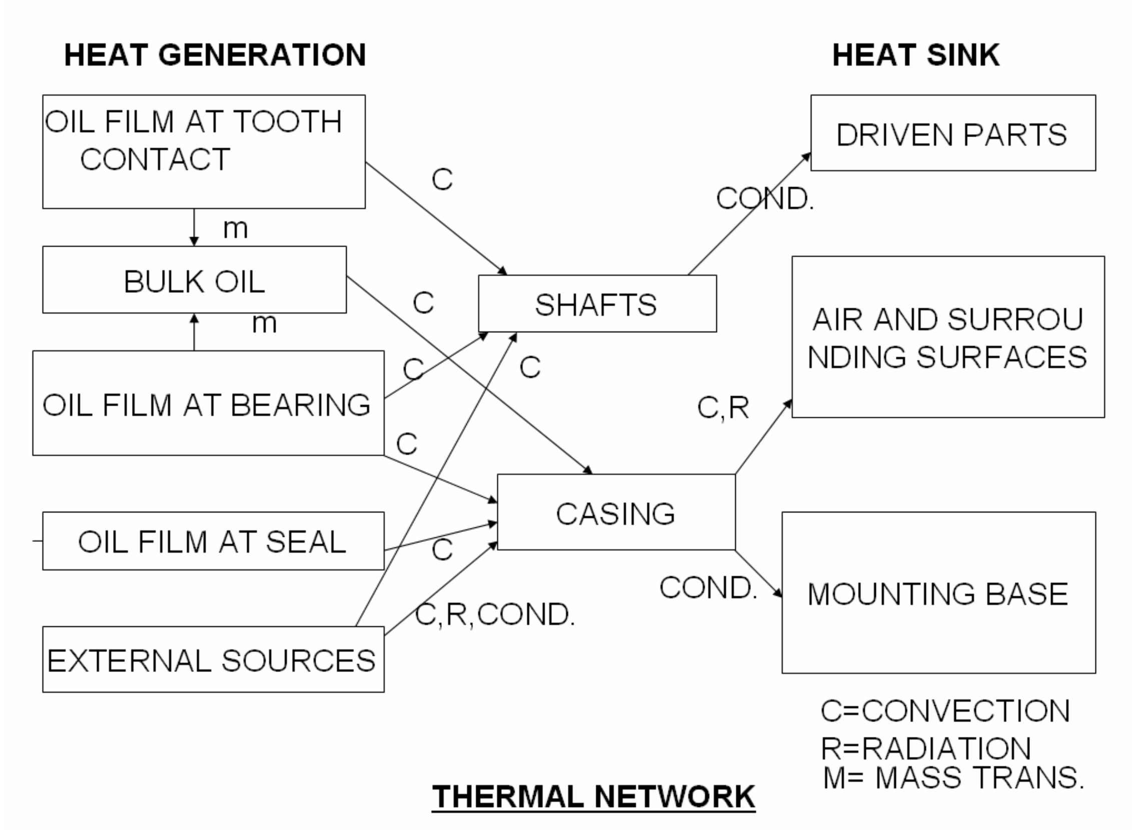



**FIG 2.1 BLOCK DIAGRAM OF GEAR THERMAL SYSTEM THERMAL NETWORK**

- **Gear tooth temperatures:**

The analysis of thermal conditions in a lubricant film between two rolling sliding surfaces has been attempted, with certain simplifying assumptions, by amongst others, Cheng and Sternlichent . There procedure was an iterative approach by means of computer program bases on the three following equations which are the Reynolds equations, the equations, and the film-thickness – surface elastic deformation equations. The complexity of these three equations rules out a direct analytical solution particularly as the viscosity is a complex function of both pressure and temperature and the density of the lubricant is temperature-dependent.

- **Tooth Friction:**

The coefficient of friction represents the instantaneous value for some point in the tooth contact. For the evaluation of power loss at the mesh, an effective value is required representative of the whole of the contact cycle. To obtain this, a series of contact positions must again be considered.

- **Bearing Resistance:**

Due to the complex conditions within rolling bearings, analytical solutions for power loss are still insufficiently developed. For a wide range of ball and roller bearings under the more usual operating conditions, the coefficient of friction, referred to the bearing bore radius, is found to be in the range 0.001—0.002.

- **Seal Resistance:**

The contribution to total heat dissipation made by the seals is derived from the power-loss information supplied by the manufacturer and, for lip seals, usually shows torque resistance to be nearly constant over a wide range of speeds.

- **Lubricant Churning:**

Power loss due to churning occurs as a result of viscous friction and kinetic effects wherever the rotating parts in a gear unit are in contact with the lubricant, All of this energy is converted into heat and is often a highly significant part of the overall gear unit power loss. In general, churning losses must be empirically evaluated for each design of unit for several operating conditions. Correlation of results over the whole range of required operating conditions and over a range of sizes of geometrically similar designs may then be represented as a relation between the usual dimensionless groups affecting fluid flow situations with free surface effects: Froude number, Reynolds number and a ratio of immersion depth to some characteristic dimension.

It is apparent that lubrication and thermal considerations are intimately allied and must be considered together if the full performance potential of a gear unit is to be realized. The local temperature rise in the film is superimposed on the bulk oil-gear temperature, the latter also being important with regard to lubricant life. Any attempt to predict temperature levels at the design stage will involve a heat transfer analysis of the complete gear unit. The guiding principles set out in the paper provide a basis for such analyses but their detailed application will vary widely according to the complexity of the unit and much of the data will be specific to particular designs. Data obtained from a limited amount of testing on prototypes or models can be extrapolated over a wide range of sizes of geometrically similar units and used in a computerized rating procedure. A further advantage of analyzing the heat transfer process in gear units is that it reveals the relative importance of various factors and indicates to the designers where changes will be most beneficial.

## 2.2 ON THE PREDICTION OF THE DESIGN CRITERIA FOR MODIFICATION OF CONTACT STRESSES DUE TO THERMAL STRESSES IN THE GEAR MESH [11]

It is seen in rolling element bearing and gears when the Hertzian pressure exceeds an allowable value. The crack usually propagates for a short distance in a direction roughly parallel to the tooth surface before turning up or branching to the surface.

They concluded that increasing the load, decreasing specific film thickness and maintaining negative relative sliding all increased the rate of micro-pitting wear. They also reported that micro pitting is almost completely eliminated at very low, but non-zero, slide to roll ratio. Decrease on the number of teeth also decreases the allowable mechanical stress. High carbon steel gears are significantly more sensitive to the thermal stress than low carbon steel as a result of lower ratio of allowable thermal strength to its ultimate strength.

In this paper the effect of material oil film thickness, surface roughness and geometric operating parameters are illustrated, on the destruction of lubricating film, the temperature rise is found, it helps to predict the thermal aspects in the thermal analysis.

## 2.3 INFLUENCE OF OIL TEMPERATURE ON GEAR FAILURES [1]

Typical gear failures like wear, scuffing, micro pitting and pitting are influenced by the oil temperature in the lubrication system. As the temperature varies, viscosity plays an important role in gear failure. Protecting elastohydrodynamic lubricant film thickness is dominated by lubricant viscosity at operating temperature. Reactivity of gear oil additives forming tribological layers is decisively influenced by temperature and time. When discussing temperature, its location has also to be defined: bulk oil temperature in the oil sump, or gear bulk temperature of the gear blank, or total contact temperature as instantaneous value in the gear mesh, including flash temperature, as the temperature rise due to contact friction.

Mr. B. R. Hohn & K. Michaelis [1] relates oil temperature with viscosity, took a lots of comparative study and then arrived on some important conclusion about how does the gear failure occur. As High temperatures lead to low viscosities and thus thin lubricant films in the gear mesh with generally detrimental influence on failure performance. On the other hand, for gear oils with additives higher temperatures correspond with higher chemical activity and, at least in some cases, with better failure performance of the lubricant. Last, but not least, at very high temperatures even metallurgical changes have been found with a reduction in material endurance limits.
It helps to understand the various failure modes in the lubrication system and how they influence by the oil temperature.

## 2.4 THERMAL ASPECTS OF LUBRICATION OF CONCENTRATED CONTACTS [13]

Deterioration of gears occurs by abrasive wear, pitting (surface fatigue) or severe adhesive wear (scuffing). The effects of the latter mode may be mitigated by the use of extreme pressure (EP) additives but these sometimes accentuate the risk of pitting-type failure. Accordingly, EP lubricants are not recommended for industrial gearing. The introduction of elastic-hydrodynamic lubrication theory has demonstrated that it is possible to operate gears without physical contact between the interacting tooth faces.

Thus Blok [13] assumed that given oil can sustain a 'temperature flash' of 500°C, the resulting criterion governing applied conditions taking the form

$$\text{Load x Speed}^{2/3} = \text{Constant} \qquad (1.1)$$

When a considerable amount of sliding occurs between discs in edgewise contact the measured film thickness has been found to be less than the value predicted by Eq (2). Thus Burwell measured film thickness values which were only about five-eighths of the calculated values, and Wilmer and Cameron measured film thickness both electrically and optically, which was found to be less than the predicted value by about 15%.

The effect of pressure on the viscosity of lubricant, combined with the local elastic deformation of surfaces, bringing them into close conformity over the contact area, enables substantial viscous films to be generated between interacting surfaces. Theory presupposes that an adequate supply of lubricant is present between the gear flanks to enable the theoretical film thickness to be generated. However it is possible for a system to operate even if the thickness is reduced below the theoretical value by oil starvation. It may therefore be desirable to evaluate those characteristics of a lubricant which determine the quantity which is deposited on a gear flank before contact occurs. Tacky lubricants are common for large slow-running gears, and modern formulations of automobile lubricants intended for both engines and gears contain viscosity index improvers which have non-Newtonian properties.

With the help of this paper [13] it is to understand that the damage in gears by abrasive wear, pitting, or scuffing is to be reduced by the use of extreme pressure (EP) additives like Zinc Dialkyldithiophosphate (ZDDP)

## 2.5 ASSESSMENT OF THE THERMAL PERFORMANCE OF BONDED GEARBOXES [5]

The effect of convection between the inner surface of the gearbox and the circulated oil has been found to be negligibly small. A critical assessment of the thermal performance of bonded steel and bonded cast iron gearboxes, with reference to a commercially available cast iron one, has been carried out.

**TABLE 2.1 TEMPERATURE DISTRIBUTION AT VARIOUS MACHINE SETTINGS**

| Machine configuration | Idling 2500 rpm | | Idling 1200 rpm | | Cutting 370 rpm | |
|---|---|---|---|---|---|---|
| Heat flux $q$ (W m$^{-2}$) | 1900 | | 1351 | | 644 | |
| Base temperature (°C) | 78.1 | | 60.4 | | 46.3 | |
| | | | localized temperature (°C) | | | |
| $r/r_o$ | $T$* | $T$** | $T$* | $T$** | $T$* | $T$** |
| 1.30 | 77.5 | 77.2 | 60.2 | 60 | 46.1 | 45.9 |
| 1.59 | 77.1 | 76.7 | 59.9 | 59.7 | 46.0 | 45.7 |
| 1.88 | 76.7 | 76.5 | 59.7 | 59.5 | 45.8 | 45.5 |

* Calculated temperature
** Measured temperature

Theoretical and finite element model has been developed for determining the temperature distribution for bonded and solid gearboxes. In the theoretical analysis the convection between the inner surface of the front panel of the gearbox and the oil circulating inside is neglected. The effect of this has been shown to be negligibly small.

This paper gives the idea about the temperature distribution for the gear box casing. By experimentally shown that the effect of convection between the inner surface of gear box and circulated oil has been formed to be negligibly small.

## 2.6 FINITE ELEMENT THERMAL ANALYSIS OF WORM GEAR BOX

Worm speed reducers, which are used for high-speed reductions, consist primarily of sliding motion. This causes high fractional losses, which is converted into heat and results in temperature rise. This temperature should be within permissible limit. In present work, P. Thyla [3] attempts have been made to establish a methodology to predict the temperature distribution in a worm gearbox using Finite Element Analysis which can be used as a basis for the thermal rating. Experiments have been conducted for various loads and temperature was measured at different locations of the gearbox. The results obtained experimentally have been used to validate the software model.

### 2.6.1 Convective Boundary Conditions

The heat generated from worm and worm gear teeth portion elements and bearing elements is conducted through the shafts to the casing from where it is carried away to the surrounding air by convection; also heat is convicted from worm to lubricating oil. Along with the material properties of the lubricating oil elements, convective boundary condition is specified on the faces of the element, which are in contact with the worm and on the casing elements to air. The heat transfer coefficients for the sides and top surface of the worm gear teeth listed in bellow table.

**TABLE 2.2 HEAT TRANSFER COEFFICIENTS USED IN THE FEA MODEL**

| ELEMENTS INVOLVED | MODE OF HEAT TRANSFER | HEAT TRANSFER COEFFICIENT (h)W/m$^2$ K |
|---|---|---|
| Worm to Oil | Forced convection | 200 |
| Oil to casing | Forced convection | 200 |
| Outer casing exposed to atmospheric air | Natural convection | 7 |

| | | |
|---|---|---|
| Wheel gear tooth surface to air | Forced convection | 20 |
| Wheel sides to air | Forced convection | 10 |

The finite element thermal analysis was carried out to the gear box described earlier, this may be attributed to the fact that there is only a localized movement of oil inside the gear box and there is no complete mixing of oil. Hence the entire cooling area is not being properly utilized for heat transfer. This suggests that more effective cooling can be brought about by providing a means of mixing the entire volume of oil for better heat transfer, thus eliminating the need for extern al cooling. This may also help in maintaining the bulk temperature of the worm and wheel at a lower value.

Worm gear boxes are presently not being utilized to their full capacity because of lack of knowledge of their thermal rating. This necessitates the development of a procedure for predicting the maximum temperature in gearboxes, this can be achieved by experimental and analytical or numerical means. Experimentation is a time consuming and costly process and moreover, measurements at all desired locations may not be possible. Hence in this work an FEM model is developed to predict the temperature at various locations. The predicted values of the model are verified at all locations where measurement is possible. The experimental data correlates well with the FEM model.

It helps to predict the modes of heat transfer involved in the various elements in the gear box, and a comparative study made between analytical approach and thermal analysis through ANSYS.

## 2.7 A METHOD FOR THERMAL ANALYSIS OF SPIRAL BEVEL GEARS

A modeling method for analyzing the three-dimensional thermal behavior of spiral bevel gears has been developed. The model surfaces are generated through application of differential geometry to the manufacturing process for face-milled spiral bevel gears. Contact on the gear surface is found by combining tooth contact analysis with three-dimensional

Hertzian theory. The tooth contact analysis provides the principle curvatures and orientations of the two surfaces. This information is then used directly in the Hertzian analysis to find the contact size and maximum pressure. Heat generation during meshing is determined as a function of the applied load, sliding velocity, and coefficient of friction. Each of these factors change as the point of contact changes during meshing. A nonlinear finite element program was used to conduct the heat transfer analysis. This program permitted the time- and position-varying boundary conditions, found in operation, to be applied to a one-tooth model. An example model and analytical results are presented

# CHAPTER 3

# DESCRIPTION OF GEAR BOX

# DESCRIPTION OFGEAR BOX

## 3.1 Description of Stage Efficiency

PE-50 is a four stage helical gear box which is widely used in cement industries. It is a low speed gear box. Designed and manufactured by ELECON.

From [21]

- Input Power = 61 KW,
- Stages = 4,
- Gear Ratio = 250,
- Efficiency = 96%,
- Input RPM = 1500,
- Output RPM = 5.76,

## 3.2 Details of Gear Box Internals

➢ **Quantity of internals**

**TABLE 3.1 QUANTITY OF BEARING INTERNALS**

| Name of Internal | Quantity |
|---|---|
| Shafts | 5 |
| Wheel | 4 |
| Bearing | 10 |
| Bearing cover | 10 |

➢ **Parameters Of Wheel**

**TABLE 3.2 DIMENSIONS FOR GEAR WHEEL**

| Wheel No. | No. of Teeth | Inside Dia (m) | Out side Dia (m) | Module (m) |
|---|---|---|---|---|
| 1 | 56 | 0.090 | 0.2679 | .0045 |
| 2 | 55 | 0.130 | 0.4145 | .007 |

| | | | | |
|---|---|---|---|---|
| 3 | 59 | 0.200 | 0.626 | .010 |
| 4 | 50 | 0.300 | 0.7524 | .014 |

Helix Angle = $12^o$

➢ **Parameters of Pinion**

**TABLE 3.3 DIMENSIONS FOR PINION**

| Pinion on | No. Of Teeth | Module (m) |
|---|---|---|
| Input shaft | 14 | 0.0045 |
| Intermediate 1 shaft | 12 | 0.007 |
| Intermediate 2 shaft | 13 | 0.010 |
| Intermediate 3 shaft | 16 | 0.014 |

➢ **Bearing Used:**

**TABLE 3.4 VARIOUS BEARING USED IN GEAR BOX**

| Bearing | Quantity |
|---|---|
| Taper Roller Bearing 33215 | 2 |
| Taper Roller Bearing 32318 | 2 |
| Cylindrical Bearing Roller NJ2332 | 2 |
| Cylindrical Bearing RollerNCF3052 | 2 |
| Roller Bering Spherical 22326 | 2 |

## ➢ Schematic Diagram to Represent The Stages in Gear Box

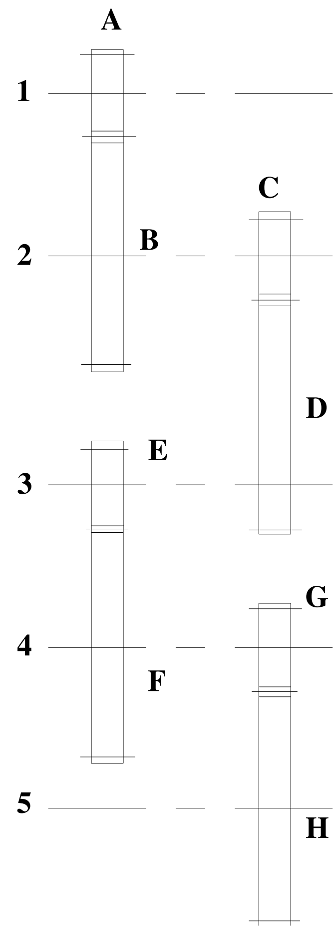



**FIG (3.1) SCHEMATIC DIAGRAM OF GEAR BOX**

1,2,3,4 and 5 represent the shafts.

A = input pinion.

C, E and G are the pinions on first second and third intermediate shafts.

B, D and F are the wheels on the first second and third intermediate shafts.

H is the wheel on output shaft.

distance between shaft 1 and 2 = 0.055m

shaft 2 and 3 = 0.193 m

shaft 3 and 4 = 0.375 m

shaft 4 and 5 = 0.50 m

# CHAPTER 4

# THERMAL HEAT DISSIPATION

# THERMAL HEAT DISSIPATION

## 4.1 Testing of Gear Box PE- 50

Testing of Gear box is done at ELECON manufacturing cell.

These are the measure After Running At 8 hours at no load.

Oil Temperature = 36 $^{o}$C

Input Rpm = 1500

Output Rpm = 5.75

Gear Ratio = 250

**Location of Temperature Measurements:**

Temperature at various locations from the front view

1. Temperature at input bearing.
2. Temperature at bearing of first intermediate shaft.
3. Temperature at bearing of second intermediate shaft.
4. Temperature at bearing of third intermediate shaft.
5. Temperature at bearing of output shaft.

A. Temperature at side casing**.**

are shown.

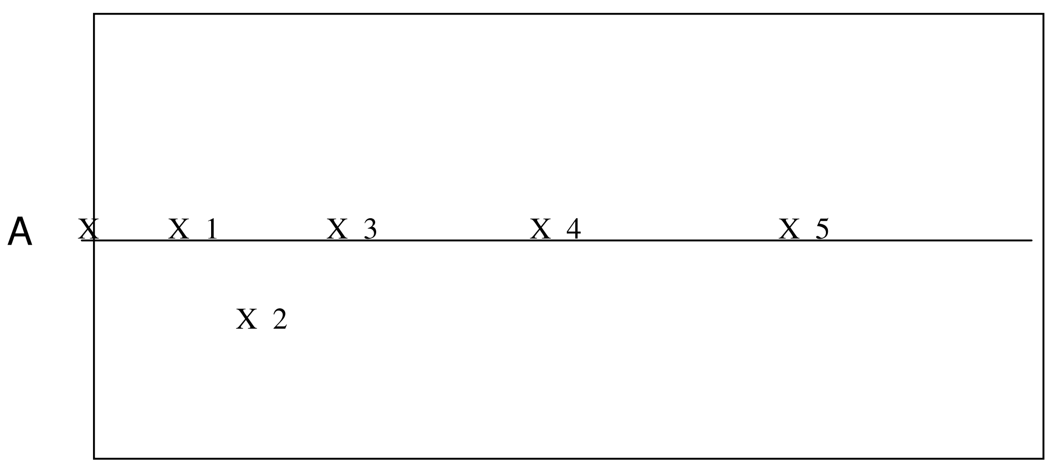



**FIG. 4.1 VARIOUS LOCATIONS FOR TEMPERATURE MEASUREMENT**

**Temperature Measurements At Various Locations:**

After 8 Hours Running at No Load Condition.

**TABLE 4.1 TEMPERATURE AT VARIOUS LOCATION AFTER 8 HOURS**

| LOCATION | FRONT TEMP($^0$C) | BACK TEMP ($^0$C) |
|---|---|---|
| 1 | 40 | 39.8 |
| 2 | 38 | 38.5 |
| 3 | 38.2 | 37.2 |
| 4 | 35.6 | 35 |
| 5 | 37.8 | 36.9 |
| A | 38 | 37.2 |

From Testing Max Temp. of Gear Box at input bearing is 40 $^0$C ,

## 4.2 CALCULATION FOR MAXIMUM HEAT DISSIPATION

## From [9]

$$p_g = \frac{F\,(T_g - T_a)\,K_a}{860 \times (1-\eta)} \qquad \textbf{4.1}$$

Where,

F = 4.3 m$^2$ ,

$K_a$ = 30 Kcal /hr m$^2$ $^0$C,

η = 0.96,

$\mathbf{T_g}$ = 40 $^0$C ,

$T_a$ = 23 $^0$C ,

So from Formula

Heat Dissipated is came out 63.75 KW.

But At Field It Varies Under Loading maximum temperature Reaches up to 80 $^0$C.

So Again Max Heat Dissipation

$T_g = 80\ ^0C$

$T_a = 35\ ^0C$

So maximum heat dissipation calculated as 168.7 KW

The Value Of Heat Dissipated Should Be Greater Then the Input Power

So,

$$P_g \geq P_{in} \qquad 4.2$$

For Our Case $P_{in}$ = 61 KW.

And **Max. Heat dissipated = 168.7 KW**

**<u>It shows Gear Box PE- 50 is safe.</u>**

# CHAPTER 5

# POWER LOSSES IN GEAR BOX

# POWER LOSSES IN GEAR BOX

## 5.1 LOSSES FROM FORMULAE OF ELECON:-

From [21]

**1. CHURNING LOSSES:-**

Churning Losses,

When value of m < 5.5

$$p_{cl} = \frac{\left[V^{2.637661} \times B\right]}{\left[10^{3.5812501} \times 324 \times (m/4.67)^{1.2}\right]} \quad 5.1$$

When 5.5 ≤ m ≤ 8

$$p_{cl} = \frac{\left[V^{2.2653876} \times B\right]}{\left[10^{2.6607005} \times 432 \times (m/6.7)^{1.2}\right]} \quad 5.2$$

When m > 8

$$p_{cl} = \frac{\left[V^{2.14834} \times B\right]}{\left[10^{2.1969847} \times 500 \times (m/9.125)^{1.2}\right]} \quad 5.3$$

Here V is calculated from = $\dfrac{\pi \times d \times N}{60}$

**TABLE 5.1 CALCULATION FOR PITCH LINE VELOCITY**

| Gear Wheel | D (m) | N (rpm) | V (m/sec) |
|---|---|---|---|
| 1 | 0.2576 | 375 | 5.057 |
| 2 | 0.393 | 81.8 | 1.6832 |
| 3 | 0.603 | 18.02 | 0.5692 |
| 4 | 0.7156 | 5.764 | 0.21606 |

**:-**

**TABLE 5.2 CALCULATION OF CHURNING LOSSES**

| Velocity V (m\s) | Face Width B (m) | Module m (m) | $P_{cl}$ (Watt) |
|---|---|---|---|
| 5.0579 | 0.070 | 0.0045 | 16.951 |
| 1.6832 | 0.1105 | 0.007 | 6.8 |
| 0.5692 | 0.1558 | 0.010 | 2.09 |
| 0.21606 | 0.2108 | 0.014 | .23607 |

Total Churning Losses = 26.077 Watts

**2. WINDAGE LOSSES:-**

Windage losses= 1.25 x $P_{cl}$ 5.4

**TABLE 5.3 CALCULATION FOR WINDAGE LOSSES**

| No. | $P_{cl}$ | (1.25 X $P_{cl}$) |
|---|---|---|
| 1 | 16.95 | 21.18 |
| 2 | 6.8 | 8.5 |
| 3 | 2.09 | 2.61 |
| 4 | 0.23607 | 0.295 |

Total Windage Losses = 32.59 Watts

**3. BEARING LOSSES:-**

For calculating Bearing loss first load on Bearing is calculated.
So firstly Torque from formula

$$T = \frac{60 \times P_{in} \times 10^3}{2\pi N} \tag{5.5}$$

**TABLE 5.4 CALCULATIONS FOR TORQUE**

| Shaft No. | $P_{in}$ (Watt) | N (N) | T (N-m) |
|---|---|---|---|
| 1 | 61000 | 1500 | 388 |
| 2 | 61000 | 375 | 1553 |
| 3 | 61000 | 81.8 | 7121 |
| 4 | 61000 | 18.02 | 32325 |
| 5 | 61000 | 5.76 | 101129 |

After finding T

Tangential Force

$$F_{tp} = F_{tw} = \frac{2T_p}{d_p} = \frac{2T_w}{d_w} \tag{5.6}$$

Radial Force

$$F_{rp} = F_{rw} = \frac{F_{tp} \times \tan\alpha}{\cos\beta} \tag{5.7}$$

And Axial Force

$$F_{ap} = F_{aw} = F_{tp} \times \tan\beta \tag{5.8}$$

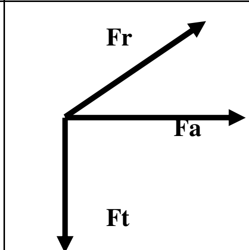
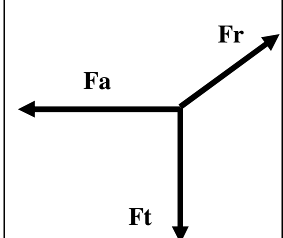
**TABLE 5.5 VALUE OF ALL THE AXIAL RADIAL AND TANGENTIAL FORCES**

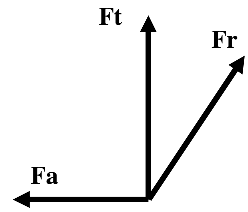
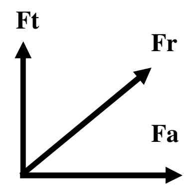
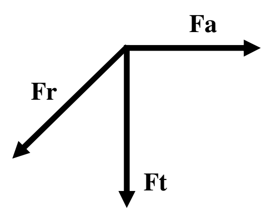
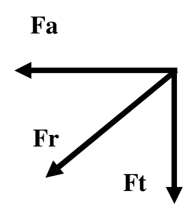
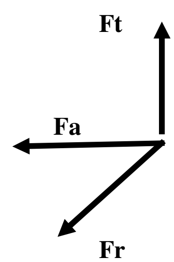
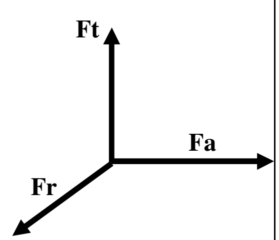
| No. | $F_{tw}$ | $F_{rw}$ | $f_{aw}$ |
|---|---|---|---|
| 1 | $12.04x10^3$ | $4.48\ x10^3$ | $2.25x10^3$ |
| 2 | $36.23x10^3$ | $13.48x10^3$ | $7.7x10^3$ |
| 3 | $107.18x10^3$ | $39.88x10^3$ | $22.78x10^3$ |
| 4 | $282.6x10^3$ | $105.15x10^3$ | $60.06x10^3$ |

After calculated the Axial, Tangential and Redial force for calculating the load on the Bearing by applying all the forces on line diagram and calculate the rection forces. Direction of force is taken with direction from table

Initially the condition for input shaft is RH and CW.

**TABLE 5.6 DIRECTION OF FORCES**

| Driver or Driven | Hand of Helix | Direction of Rotation | Direction of Force |
|---|---|---|---|
| Driver | LH | CCW |  |
| Driver | RH | CCW |  |

| | | | |
|---|---|---|---|
| Driver | LH | CW |  |
| Driver | RH | CW |  |
| Driven | RH | CCW |  |
| Driven | LH | CCW |  |
| Driven | RH | CW |  |
| Driven | LH | CW |  |

$$P_{bl} = \frac{\mu \times F_l \times d \times N}{19100} KW \qquad 5.9$$

Value of μ For

Taper Roller = .001

Spherical Roller Bearing = .0018

Cylindrical Roller Bearing = .001

**TABLE 5.7 CALCULATION FOR BEARING LOSSES**

| Brg No. | $F_L$ (N) | μ | d (m) | N(rpm) | $P_{bl}$ (watt) |
|---|---|---|---|---|---|
| 33215 | 8896 | .001 | 0.075 | 1500 | 52 |
| 33215 | 3962.1 | .001 | 0.075 | 1500 | 23 |
| 32318 | 15921 | .001 | 0.090 | 375 | 28 |
| 32318 | 33697 | .001 | 0.090 | 375 | 59 |
| 2332 | 87734.7 | .0018 | 0.130 | 81.8 | 80 |
| 22326 | 60276.6 | .0018 | 0.130 | 81.8 | 60 |
| 2332 | 171540 | .001 | 0.160 | 18.02 | 25 |
| 2332 | 230323 | .001 | 0.160 | 18.02 | 34 |
| 3052 | 120983 | .001 | 0.260 | 5.76 | 9 |
| 3052 | 189587 | .001 | 0.260 | 5.76 | 14 |

Total Bearing Losses = 384 watts

**4 MESH LOSS IN PAIR OF GEAR TEETH:-**

Mesh loss is calculated from

$$P_{ml} = \frac{.09 \times P_{in} \times (Z_p + Z_w)}{Z_p \times Z_w} \qquad 5.10$$

**TABLE 5.8 CALCULATIONS FOR MESH LOSS**

| Shaft | $Z_p$ | $Z_W$ | $P_{ml}$(Watt) |
|---|---|---|---|
| 1 | 14 | 56 | 490 |
| 2 | 12 | 55 | 557 |
| 3 | 13 | 59 | 515 |
| 4 | 16 | 50 | 452 |

Total Losses Due to Mesh = 2000 Watt

## 5.2 LOSSES FROM AGMA STANDARD:-

From [10]

**1. BEARING LOSSES:-**

For Bearing Losses

$$P_{bl} = \frac{\mu \times F_l \times (d_o + d_i) \times N}{252100} \qquad 5.11$$

**TABLE 5.9 CALCULATION FOR BEARING LOSSES FROM AGMA**

| Brg No. | $F_L$ lb | μ | $d_o$ inch | $d_i$ inch | N | $P_{bl}$ HP |
|---|---|---|---|---|---|---|
| 33215 | 1998.8 | .001 | 5.07 | 2.92 | 1500 | .095 |

| | | | | | | |
|---|---|---|---|---|---|---|
| 3321 | 890.6 | .001 | 5.07 | 2.92 | 1500 | .042 |
| 32318 | 3579.1 | .001 | 7.41 | 3.51 | 375 | .058 |
| 32318 | 7575.3 | .001 | 7.41 | 3.51 | 375 | .123 |
| 22326 | 19723.5 | .0018 | 8.97 | 5.07 | 81.8 | .161 |
| 22326 | .0018 | 13550.7 | 8.97 | 5.07 | 81.8 | .111 |
| 2332 | .001 | 38563.7 | 13.26 | 6.24 | 18.02 | .0527 |
| 2332 | .001 | 51778.6 | 13.26 | 6.24 | 18.02 | .0721 |
| 3052 | .001 | 27198.2 | 15.6 | 10.14 | 5.76 | .0159 |
| 3052 | .001 | 42620.8 | 15.6 | 10.14 | 5.76 | .0250 |

Total Loss in Bearing = 0.7567 HP

= 563.7 Watt

## 2 TOOTH FRICTION LOSSES:-

Tooth friction loss is calculated from

$$P_{ml} = \frac{\mu_1 \times T_p \times N_p \times \cos^2 \varphi_w}{63025\ M} \qquad 5.12$$

For calculate the value of M first calculate the value of $H_s$ and $H_t$.

$$H_s = (M_G + 1)\left[\left\{\left(\frac{R_0}{R_w}\right)^2 - \cos^2\phi_t\right\}^{0.5} - \sin\phi_t\right] \qquad 5.13$$

**TABLE 5.10 CALCULATION FOR $H_S$**

| No | $R_0$ inch | $R_w$ inch | $Cos^2 ø_t$ | Sin $ø_t$ | $M_G$ $(N_1/N_2)$ | $H_s$ |
|---|---|---|---|---|---|---|
| 1 | 5.19 | 5.02 | .878 | .348 | 4.66 | .548 |
| 2 | 7.93 | 7.66 | .878 | .348 | 4.58 | .554 |
| 3 | 12.15 | 11.76 | .878 | .348 | 4.53 | .513 |
| 4 | 14.5 | 13.95 | .878 | .348 | 3.12 | .442 |

Now For $H_t$,

$$H_t = \frac{(M_G + 1)}{M_G}\left[\left\{\left( r_0 / r_w \right)^2 - \cos^2 \phi_t \right\}^{0.5} - \sin\phi_t \right] \quad 5.14$$

**TABLE 5.11 CALCULATION FOR $H_T$**

| No | $M_G$ | $r_o$ | $r_w$ | $Cos^2 ø_t$ | Sin $ø_t$ | $H_t$ |
|---|---|---|---|---|---|---|
| 1 | 4.66 | 1.42 | 1.24 | .878 | .348 | .367 |
| 2 | 4.58 | 1.94 | 1.67 | .878 | .348 | .418 |
| 3 | 4.53 | 2.98 | 2.59 | .878 | .348 | .390 |
| 4 | 3.12 | 5.01 | 4.46 | .878 | .348 | .355 |

Now using the value of $H_s$ and $H_t$ to calculate M,

$$M = \frac{2\cos\phi_t (H_s + H_t)}{(H_s^2 + H_t^2)} \quad 5.15$$

**TABLE 5.12 CALCULATION FOR M**

| No. | Cos $\Phi_t$ | $H_s$ | $H_t$ | M |
|---|---|---|---|---|
| 1 | 0.937 | 0.548 | 0.367 | 3.94 |
| 2 | 0.937 | 0.554 | 0.418 | 3.78 |
| 3 | 0.937 | 0.513 | 0.390 | 4.07 |
| 4 | 0.937 | 0.442 | 0.355 | 4.64 |

To Calculate the value of $\mu_1$ first contact load factor (K) is calculated

$$K = \frac{T_p \left(Z_p + Z_w\right)}{2 \times F \times \left(R_w\right)^2 \times Z_w} \qquad 5.16$$

**TABLE 5.13 CALCULATION FOR K**

| No. | Tp(lb-inch) | $Z_p$ | $Z_w$ | F(inch) | $R_w$(inch) | K |
|---|---|---|---|---|---|---|
| 1 | 3434.09 | 14 | 56 | 2.73 | 5.02 | 31.1 |
| 2 | 13747.8 | 12 | 55 | 4.3 | 7.66 | 33.18 |
| 3 | 63027.1 | 13 | 59 | 6.07 | 11.76 | 45.8 |
| 4 | 286100 | 16 | 50 | 8.22 | 13.95 | 118.04 |

$$\mu_1 = \frac{K^{0.35}}{C_1 \times V^{0.23}} \qquad 5.17$$

**TABLE 5.14 CALCULATION FOR $M_1$**

| No. | K | $C_1$ | V (Fpm) | $\mu_1$ |
|---|---|---|---|---|
| 1 | 31.1 | 76.3 | 995.4 | 0.008 |
| 2 | 33.48 | 76.3 | 331.2 | 0.0118 |
| 3 | 45.8 | 76.3 | 112.0 | 0.016 |
| 4 | 118.04 | 76.3 | 42.53 | 0.029 |

**TABLE 5.15 CALCULATION FOR TOOTH FRICTION LOSSES AGMA**

| No. | $T_p$(lb-inch) | Np | $Cos^2\phi w$ | $\mu_1$ | M | Pml(HP) |
|---|---|---|---|---|---|---|
| 1 | 3434.09 | 1500 | 0.956 | .008 | 3.94 | .158 |
| 2 | 13747.8 | 375 | 0.956 | .0118 | 3.78 | .244 |
| 3 | 63027.7 | 81.8 | 0.956 | .016 | 4.07 | .307 |
| 4 | 286100 | 18.02 | 0.956 | .029 | 4.64 | .488 |

Total Losses in Watts = 812

## 3. WINDAGE AND CHURNING LOSSES:-

**Calculation For Windage And Churning Losses:-**

Formula for calculating Windage and churning losses

$$P_{(wl+cl)} = \frac{{d_w}^2 \times N^2 \times B \times \cos^3 \varphi}{12600 \times p_n \times A} \tag{5.18}$$

For ISO 320 $C_p = 50$

**TABLE 5.16 CALCULATION FOR TOTAL CHURNING AND WINDAGE LOSS FROM AGMA**

| No. | $d_w$ | N | B | $\cos^3\Phi$ | m | $P_n$ | $P_{(wl+cl)}$ HP |
|---|---|---|---|---|---|---|---|
| 1 | 2.49 | 1500 | 2.73 | 0.935 | 0.1771 | 5.772 | 0.1097 |
| 2 | 10.04 | 375 | 2.73 | 0.935 | 0.1771 | 5.772 | 0.1108 |
| 3 | 3.34 | 375 | 4.3 | 0.935 | 0.2755 | 3.71 | 0.030 |
| 4 | 15.32 | 81.8 | 4.3 | 0.935 | 0.2755 | 3.71 | 0.031 |
| 5 | 5.18 | 81.8 | 6.07 | 0.935 | 0.393 | 2.60 | 0.006 |
| 6 | 23.52 | 18.02 | 6.07 | 0.935 | 0.393 | 2.62 | 0.0068 |
| 7 | 8.93 | 18.02 | 8.2 | 0.935 | 0.551 | 1.872 | 0.0018 |
| 8 | 27.9 | 5.764 | 8.2 | 0.935 | 0.551 | 1.872 | 0.0018 |

So Total Loss In H.P. = 0.2979

**= 222 watt**

## 5.3 COMPARISON OF LOSSES

**TABLE 5.17 COMPARISON OF LOSSES CALCULATED FROM AGMA FORMULAE AND ELECON FORMULAE**

<table>
<tr><td>Losses<br>Watts</td><td>AGMA Standard</td><td>ELECON<br>Gears</td></tr>
<tr><td>Bearing</td><td>563.7</td><td>384</td></tr>
<tr><td>Tooth<br>Friction</td><td>812</td><td>2000</td></tr>
<tr><td>Windage</td><td rowspan="2">222</td><td>32.58</td></tr>
<tr><td>Churning</td><td>26.07</td></tr>
<tr><td>Stage efficiency</td><td>97.38%</td><td>96 %</td></tr>
</table>

## 5.4 C PROGRAM TO CALCULATE POWER LOSSES

### 5.4.1 INTRODUCTORY REMARKS

C- Program is prepared for calculating heat losses in Gear box. C- Program gives Power losses in gear box from formulae of AGMA and ELICON. It gives facility to select from which methods losses to be calculated. With variation in stages of gear box. It will give churning, windage, mesh and bearing losses individually and total sum also. The flow chart and algorithm have been prepared for understanding logic behind C- program.

### 5.4.2 ALGORITHM

**Algorithm:**

STEP: 1

- Enter which method to use: a
- Enter the number of stages :s
- If method selected is ELECON METHOD then
- Enter the value of velocity of wheel or pinion : v
- Enter the value of face width : B
- Enter the value of module: m
- Depending on value of module, formulae to calculate churning loss will vary.
- Calculate churning losses from respective formulae
- Calculate windage loss which is 1.25 * churning losses.

STEP: 2

- Enter the value of teeth on pinion and wheel.
- Enter the value of input bearing power.
- Calculate meshing loss using respective formulae.

STEP: 3

- Enter the value of bearing . Coeff. Of friction.
- Enter the value of load on bearing.
- Enter the value of diameter of shaft.
- Enter the value of speed in rpm.
- Calculate bearing losses using formula.
- Calculate total power losses in gear box using all the above losses.

STEP: 4

- If method selected is AGMA then.
- Enter the value of helix angle.
- Enter the value of absolute viscosity.
- Enter the reference diameter.
- Enter the value of module.
- Enter the value of speed in rpm.
- Enter the value of face width.
- Calculate churning and windage losses using AGMA formulae.

STEP: 5

- Enter the value of lubrication factor.
- Enter the value of teeth on pinion and wheel.
- Enter the value of reference radius and transverse pressure angle.
- Enter the value of pinion speed in rpm and torque on pinion.
- Enter the value of helix angle and normal pressure angle.
- Enter the value of stage ratio.
- Enter the value of reference radius of wheel and pinion.
- Enter the value of velocity of wheel.
- Calculate meshing losses using formulae.

STEP: 6

- Enter the value of bearing. Coefficient Of friction
- Enter the value of load on bearing.
- Enter the value of inner diameter of shaft
- Enter the value of outer diameter of shaft
- Enter the value of speed in rpm
- Calculate bearing losses using formula for that.
- Calculate total power losses in gear box using all the above losses.

### 5.4.3 FLOW CHART

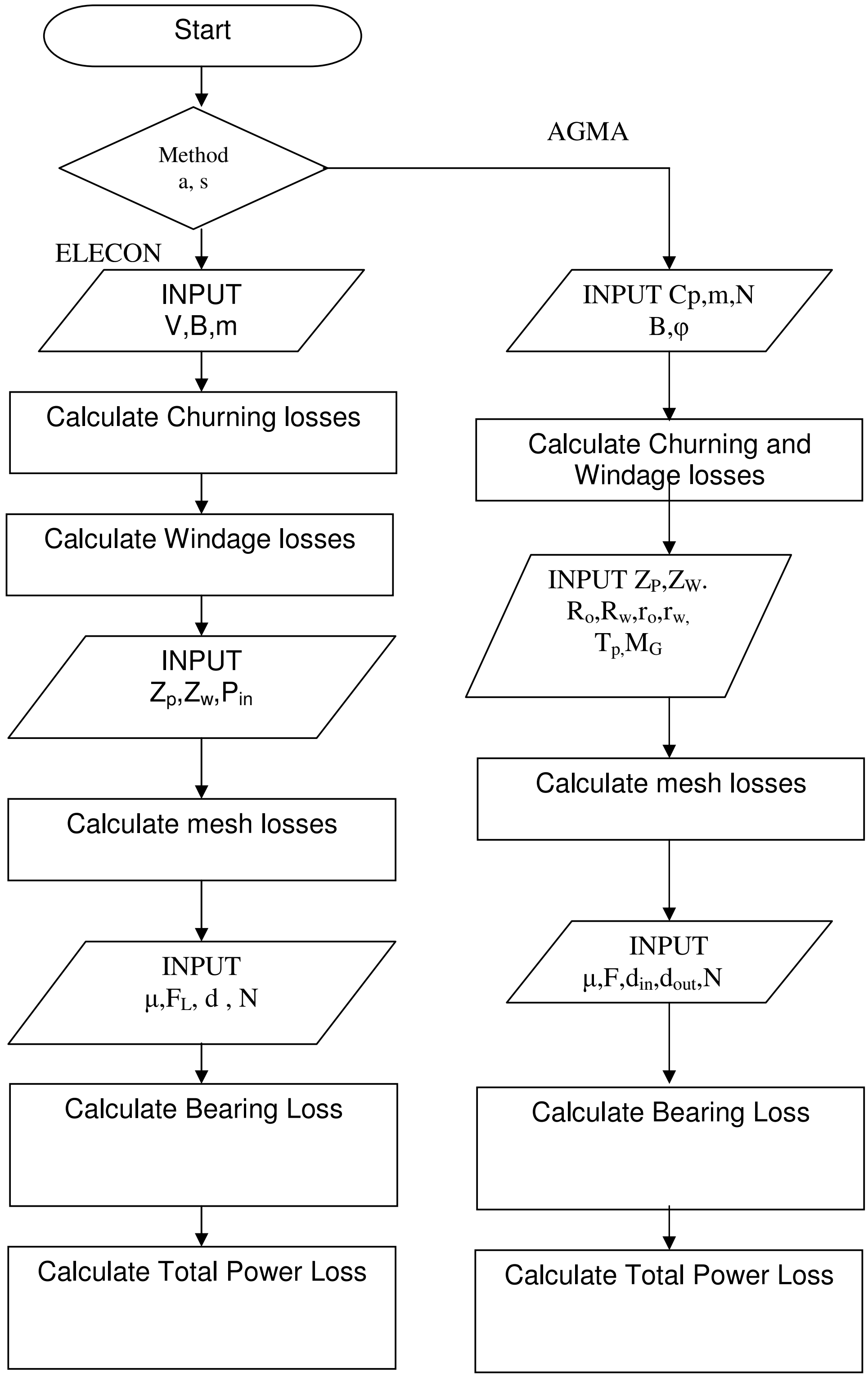



**FIG. 6.1 FLOW CHART FOR PROGRAM IN C**

### 5.4.4 OUTPUT

ENTER 1:- LOSSES FROM ELECON

ENTER 2:- LOSSES FROM AGMA

WHICH METHOD YOU WANT TO USE:-1

ENTER THE NUMBER OF STAGES:->2

ENTER THE VALUE OF VELOCITY OF WHEEL OR PINION:->5.0579

ENTER THE VALUE OF FACE WIDTH:->.070

ENTER THE VALUE OF MODULE:->.0045

ENTER THE VALUE OF VELOCITY OF WHEEL OR PINION:->1.6832

ENTER THE VALUE OF FACE WIDTH:->.110

ENTER THE VALUE OF MODULE:->.007

TOTAL CHURNING LOSS IS :-> 23.795303 WATT

TOTAL WINDAGE LOSS IS :-> 29.744130 WATT

ENTER THE VALUE OF TEETH ON PINION:->14

ENTER THE VALUE OF TEETH ON WHEEL:->56

ENTER INPUT BEARING POWER:->61

ENTER THE VALUE OF TEETH ON PINION:->12

ENTER THE VALUE OF TEETH ON WHEEL:->55

ENTER INPUT BEARING POWER:->61

TOTAL MESHING LOSS IS :-> 1047.496704 WATT

ENTER THE VALUE OF SPEED IN RPM:-> 375

ENTER THE VALUE OF BRG. COEFF. OF FRICTION:->.001

ENTER THE VALUE OF LOAD ON BRG.:->33697

ENTER THE VALUE OF DIAMETER OF SHAFT:->.090

ENTER THE VALUE OF SPEED IN RPM:-> 375

ENTER THE VALUE OF BRG. COEFF. OF FRICTION:->.0018

ENTER THE VALUE OF LOAD ON BRG.:->87734.7

ENTER THE VALUE OF DIAMETER OF SHAFT:->.130

ENTER THE VALUE OF SPEED IN RPM:-> 81.8

ENTER THE VALUE OF BRG. COEFF. OF FRICTION:->.0018

ENTER THE VALUE OF LOAD ON BRG.:->60276.6

ENTER THE VALUE OF DIAMETER OF SHAFT:->.130

ENTER THE VALUE OF SPEED IN RPM:-> 81.8

TOTAL VALUE OF BRG. LOSSES IS :->311.741272 WATT

TOTAL POWER LOSSES IN GEAR BOX:->1412.59 WATT

# CHAPTER 6

# SOLID MODELING

# AND

# THERMAL ANALYSIS

# SOLID MODELING OF GEAR BOX PE-50

## 6.1 SOLID MODELING

Solid Modeling of Gear Box Is done on the Pro/ENGINEER Wildfire 2.0.
This modeling is done mainly by extrude command in Pro/E.Solid modeling is done according the manufacturing drawing which are issued by ELECON Gear Division.
First the solid modeling of lower casing is done then after mirroring upper casing is modeled.
The Assembly of both casing is prepared and then hole for second shaft is prepared in lower casing as per manufacturing drawing.

### 6.1.1 Solid modeling of Gear Casing

- Bottom casing
- Top casing
- Gear Case Assembly

➢ **Bottom casing:**

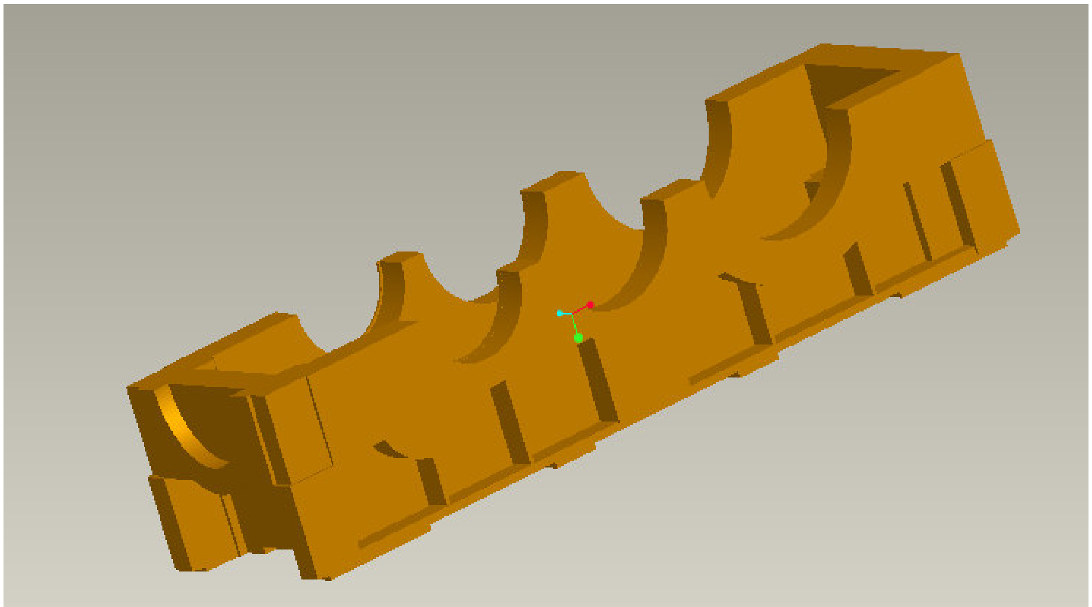


**FIG.6.1 BOTTOM CASING OF PE- 50**

➢ **Top Casing:**

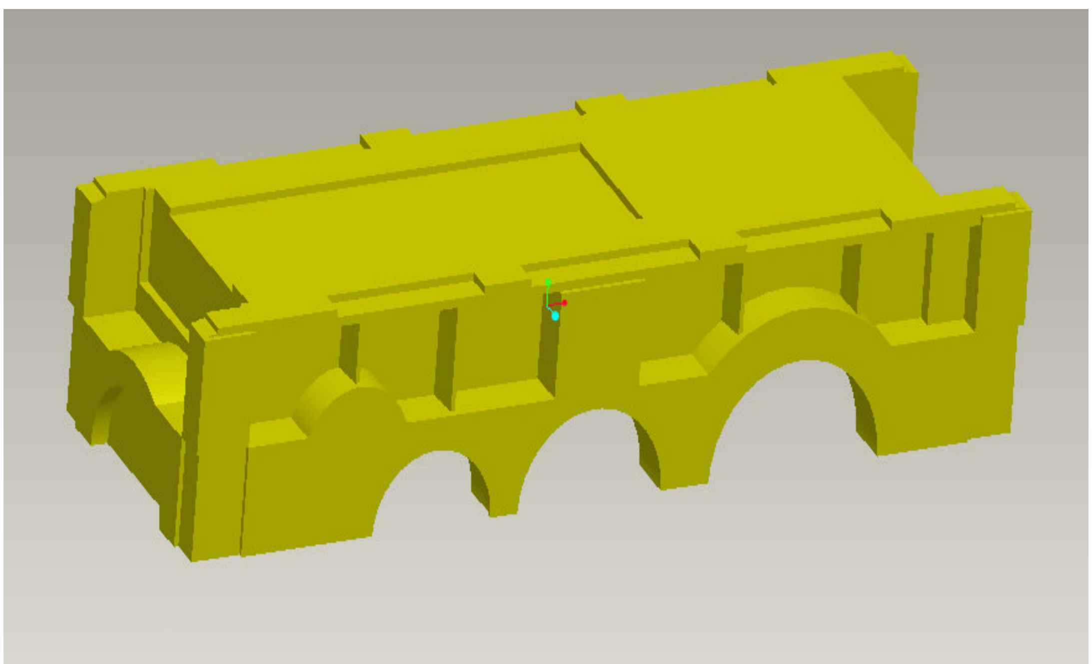


**FIG.6.2 TOP CASING OF PE 50**

- **Gear Case Assembly:**

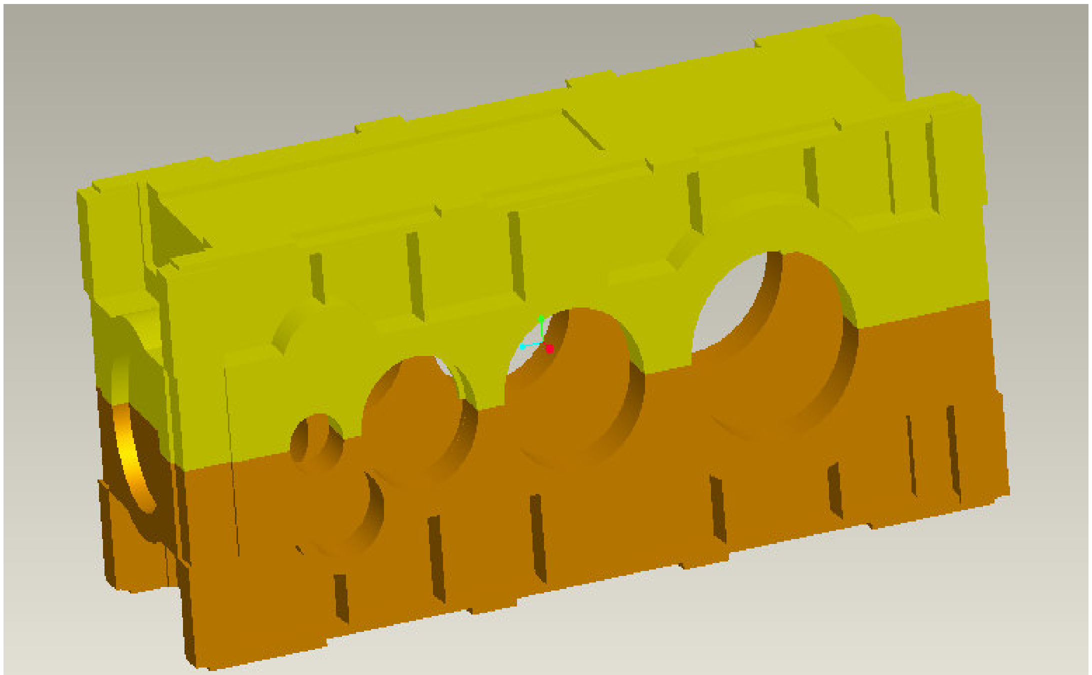


**FIG.6.3 GEAR CASE ASSEMBLY**

### 6.1.2 Solid modeling Of gear case Assembly with Internals

- View of Half Section With all Internals
- Front View Of complete Assembly
- Back View of Complete Assembly

- **View of Half Section With all Internals**

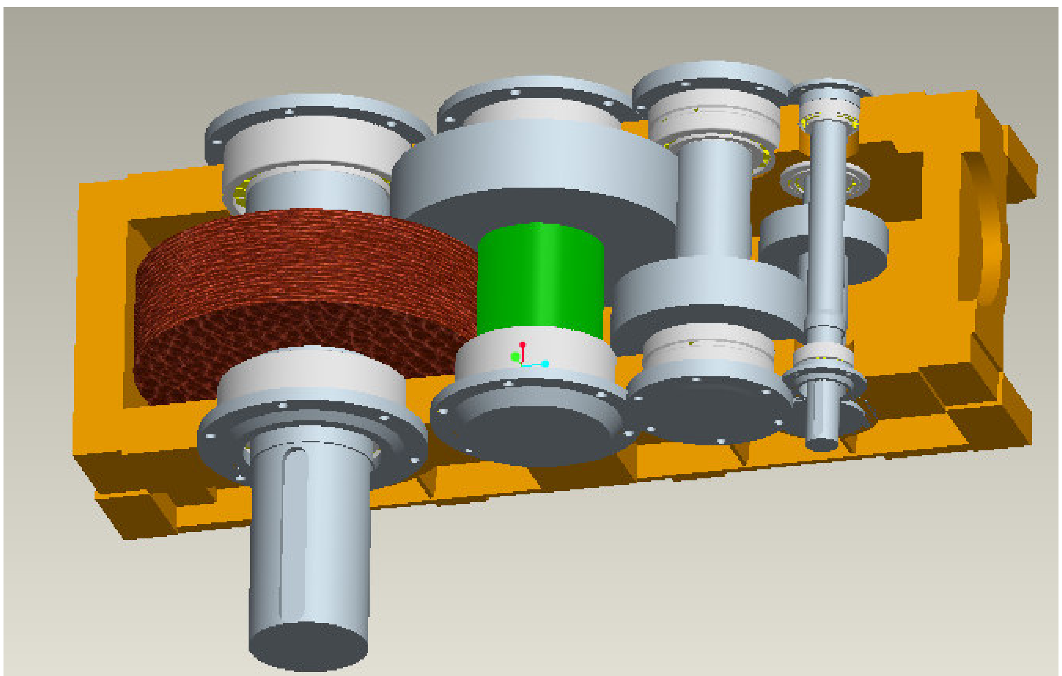


**FIG.6.4 VIEW OF ASSEMBLY WITH INTERNALS (TOP CASING REMOVED)**

- **Front View Of Complete Assembly**

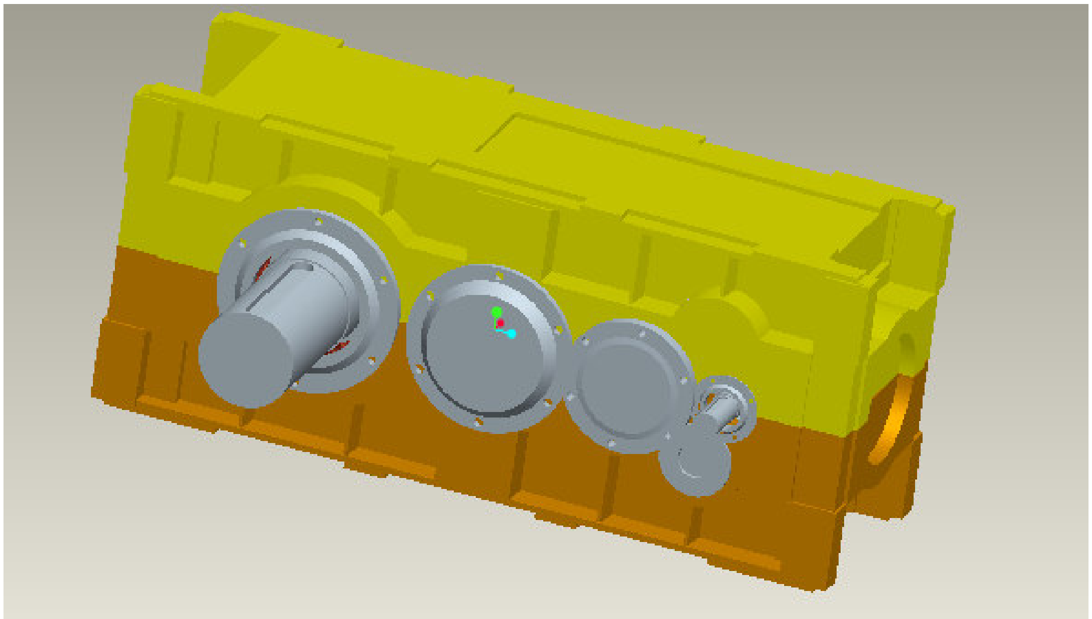


**FIG.6.5 FRONT VIEW OF ASSEMBLY WITH INTERNALS**

- **Back View Of Complete Assembly**

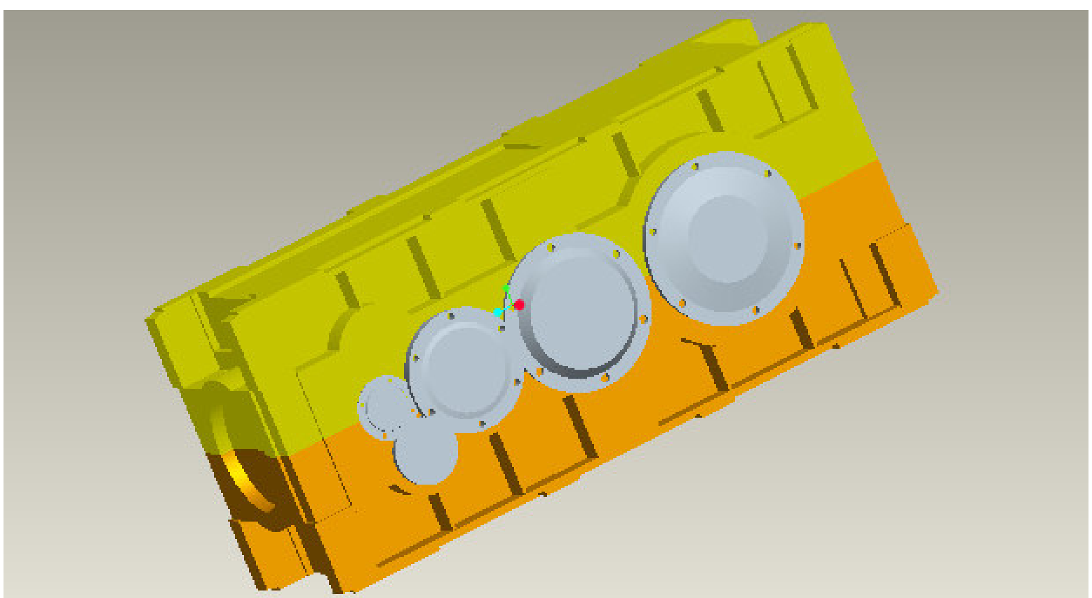


**FIG.6.6 BACK VIEW OF ASSEMBLY WITH INTERNALS**

## THERMAL ANALYSIS

Thermal analysis is done with the help of ANSYS- 10.0.

## Material Properties :

Conductivity = 54 W /m K

Specific heat = 460 J /Kg K

Density = 7250 Kg /$m^3$

Element = Solid - 20 Bricknode 90

**Thermal Analysis of Gear Box PE-50**

Thermal analysis of Gear Box is done By Two ways in ANSYS-10.0

- Steady Analysis
- Transient Analysis

The conditions for load in thermal analysis are given below :

- Taking Heat Generation as a Load
- Convection of Casing Surface With oil Temperature as a Load
- Convection of Air With the Temperature of external Surface

## 6.2. Steady State Analysis :

In Steady State, Analysis is carried out through the temperature difference. In this process the time can not be varied.

Meshing is done in ANSYS-10.0

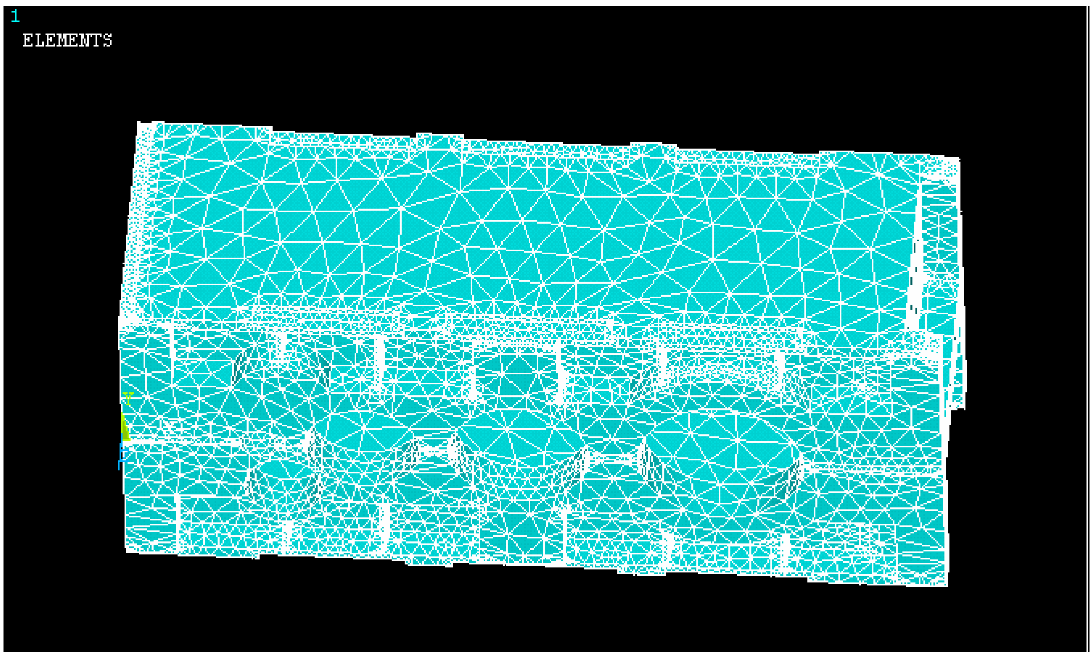



**FIG.6.7 MESHED MODEL**

The maximum Temperature of Gear Box is taken as 80°C.

Surface Convection Coefficient is taken 200W/$m^2$ °C.

Atmospheric Temperature is 35 °C.

In ANSYS-10.0 The load is applied as Convection and

Temperature Distribution in the Gear Box is given below.

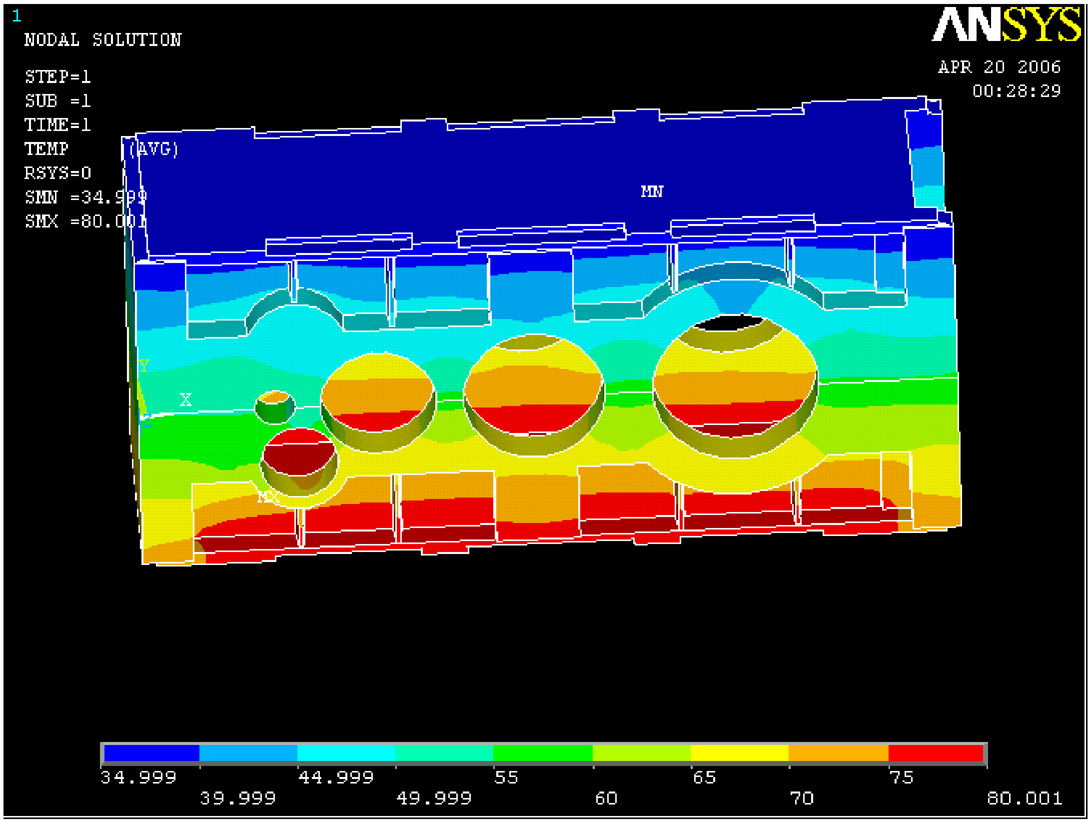



**FIG.6.8 STEADY STATE TEMPERATURE DISTRIBUTION**

## 6.3 Transient Analysis:

By the Transient Analysis the change in temperature is analyzed with variation in time. In different time Slot the change in temperature can be observed.

For model PE-50 the loads are applied by three ways.

- Taking Heat Generation as A Load.
- Convection of Casing Surface With oil Temperature as a Load.
- Convection of Air With the Temperature of external Surface.

### ➢ Transient Analysis Through Heat Generation

As total loss in Gear box is already calculated. This total loss is taken as the heat generation in the gear box .

As the surface area = 4.3 $m^2$

Total heat loss = 2500 watt

So

Total Heat Generation For half Casing is --- 1162.7w/$m^3$

This Heat generation is taken as the Load and temperature distribution is found.

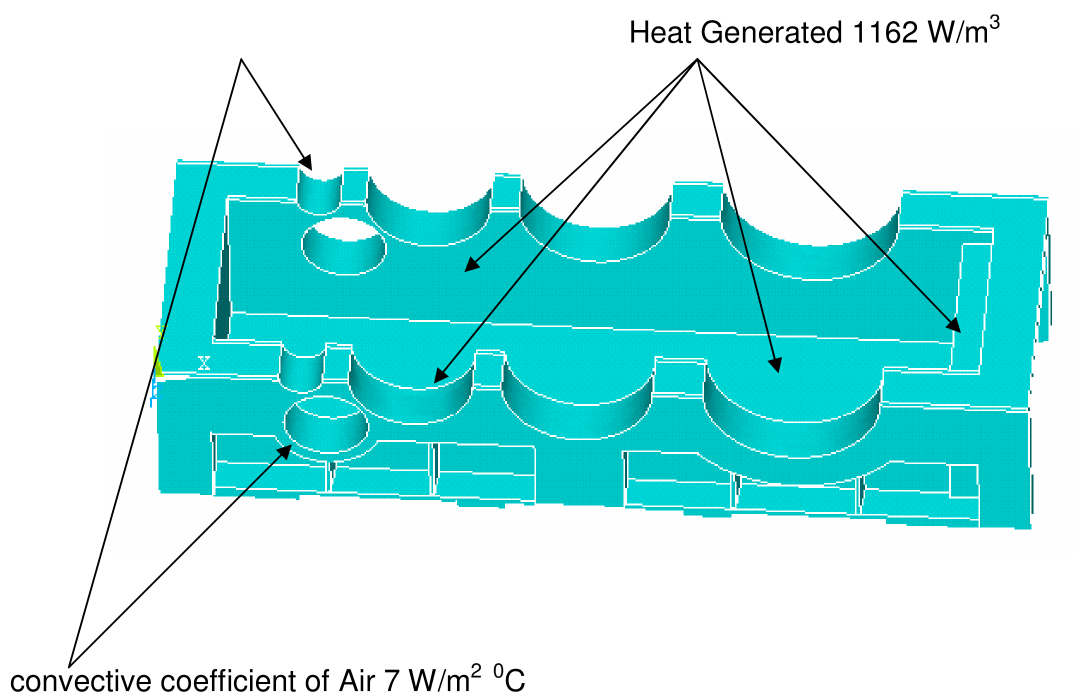



**FIG.6.9 LOADING CONDITION FOR HEAT GENERATION**

**Temperature Distribution** –

For two hour (120minutes),four hours (240min) and eight hours (480 min)

Transient Analysis is Carried out.

Eight Sub steps are given .

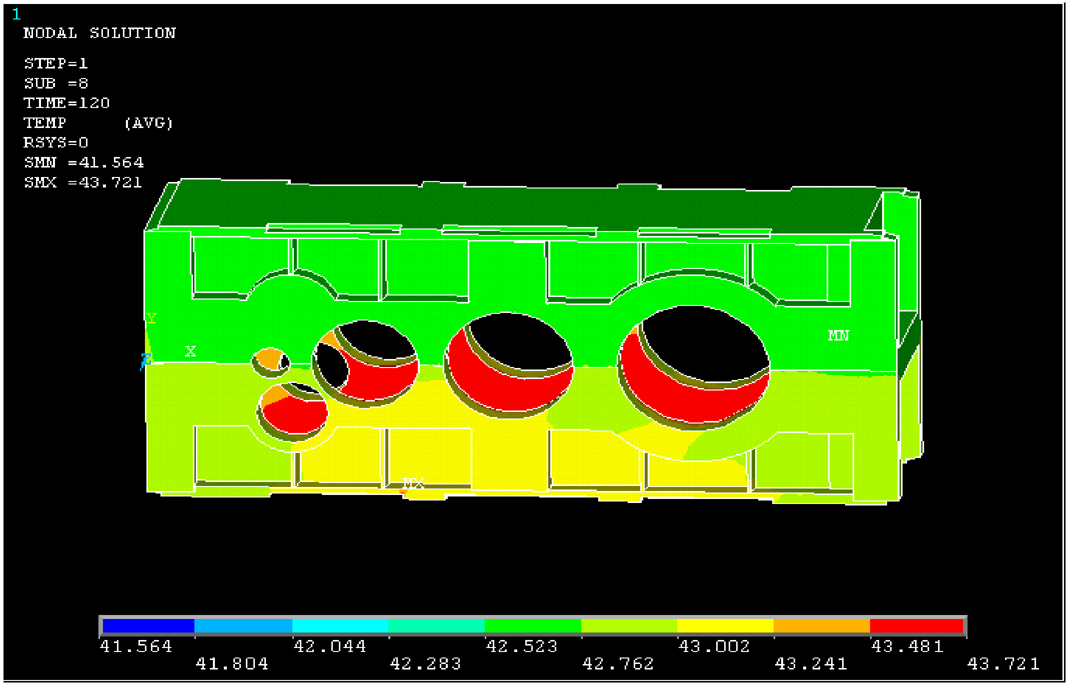



**FIG.6.10 TEMPERATURE DISTRIBUTION AFTER 2 HOURS IN HEAT GEN**

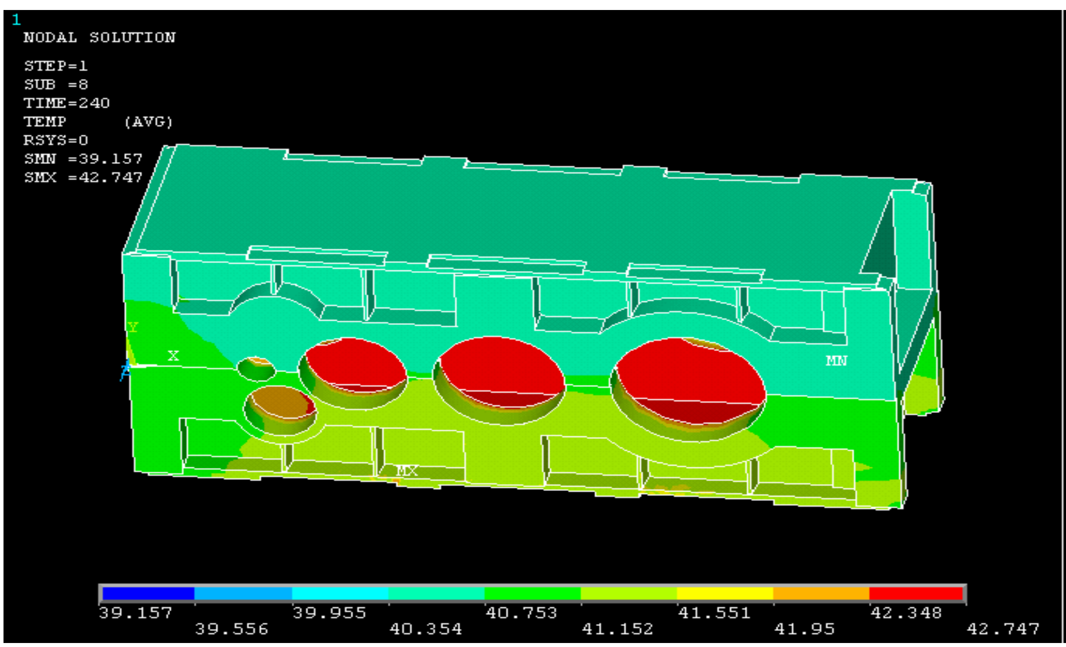



**FIG.6.11 TEMPERATURE DISTRIBUTION AFTER 4 HOURS IN HEAT GEN**

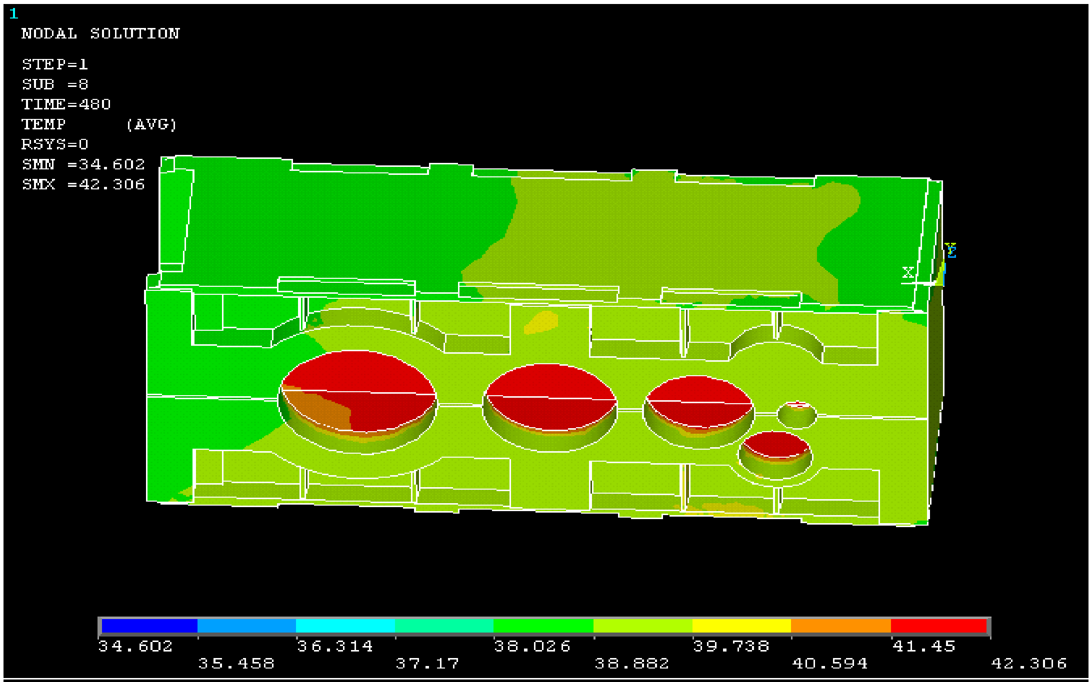



**FIG 6.12 TEMPERATURE DISTRIBUTION AFTER 8 HOURS IN HEAT GEN**

- **Transient Analysis Through Convection of Casing Surface With oil Temperature as a Load**

Since From Testing Of Gear Box maximum temperature of gear box measured is 80 $^0$C.

Convective Coefficient of surface is 200 W/m$^2$ $^o$C.

Convective Coefficient of air is 7 W/m$^2$ $^o$C.

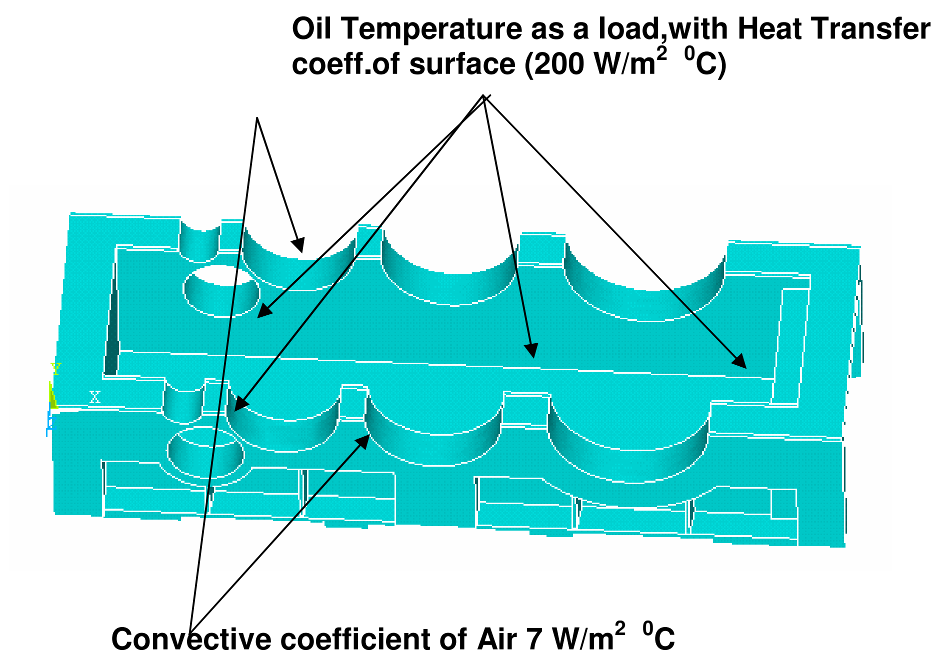



**FIG.6.13 LOADING CONDITIONS FOR OIL TEMPERATURE**

Temperature distribution is obtained by above loading condition from transient analysis in ANSYS 10.0.

Analysis is done for eight hours (480 min) and other material properties and element are kept same.

**Temperature Distribution :**

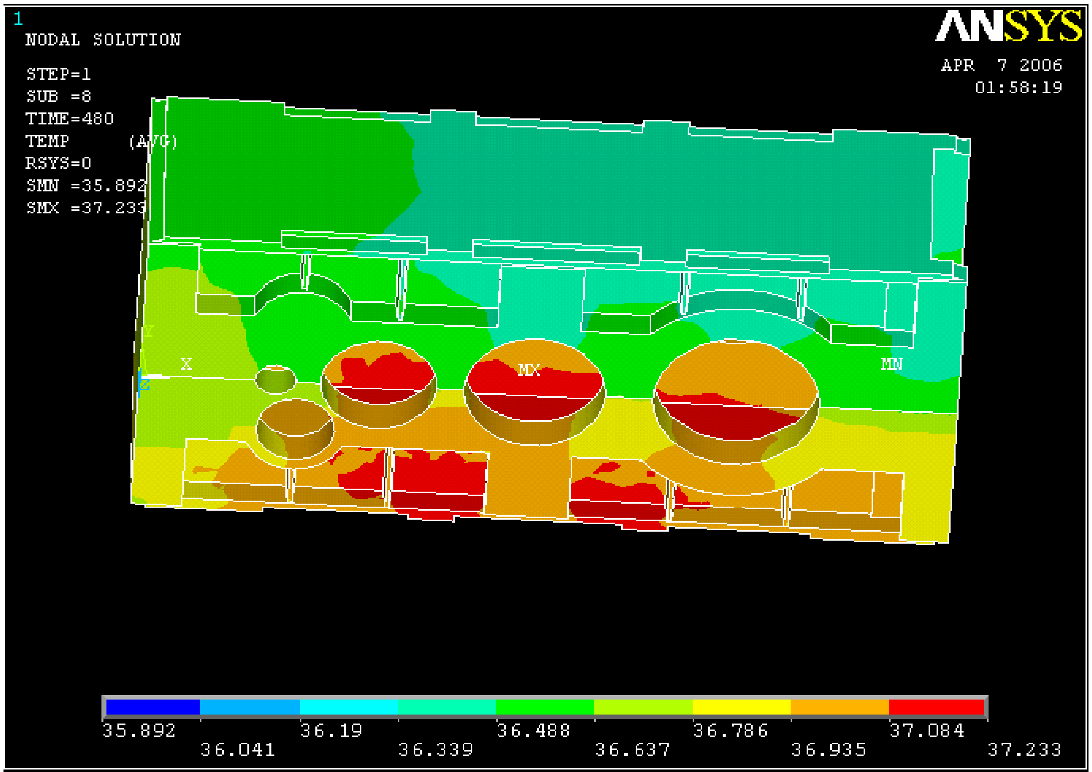
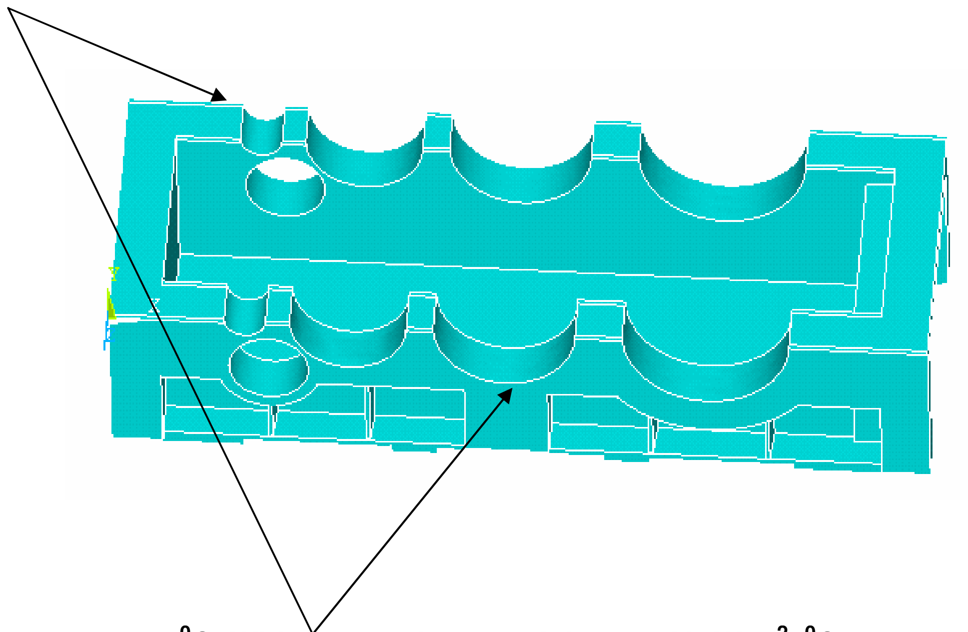



**FIG.6.14 TEMPERATURE VARIATION DUE TO LOAD AS OIL TEMPERATURE**

- **Convection of Air With the Temperature of external Surface as a load**



**Temperature 80$^{0}$C With convective coefficient of Air W/m$^{2}$ $^{0}$C**

**FIG.6.15 LOADING CONDITIONS IN SURFACE CONVECTION**

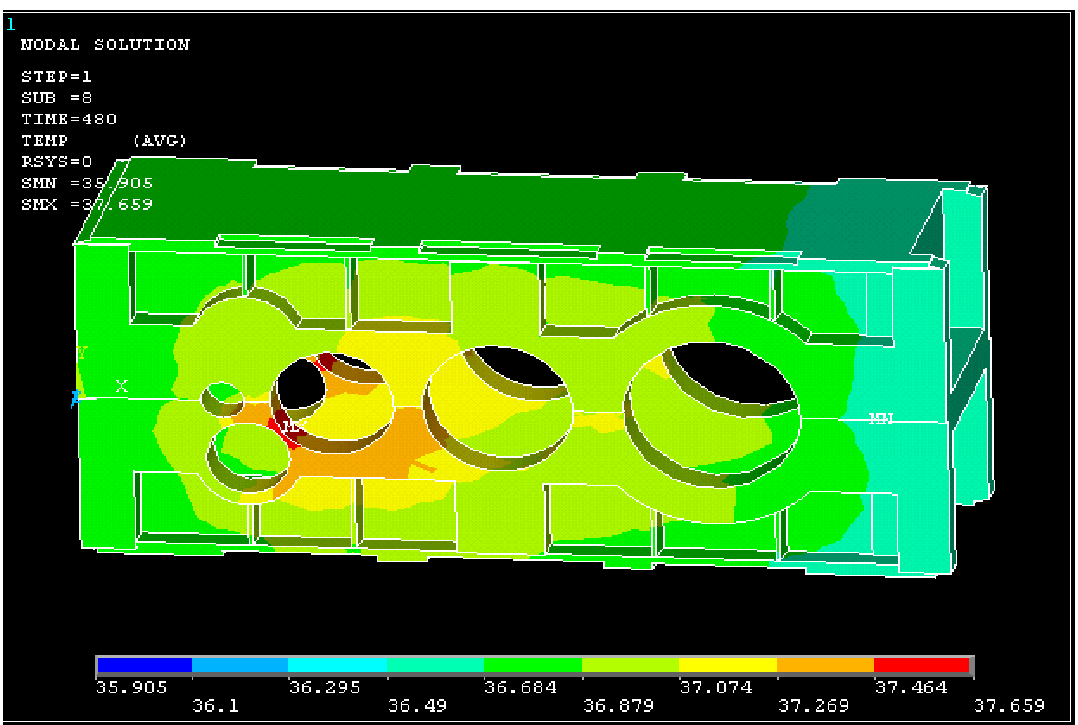



**FIG.6.16 TEMPERATURE VARIATION AFTER 8 HOURS IN CONVECTION**

# CHAPTER 7

# RESULTS AND DISCUSSION

# RESULT AND DISCUSSION

Transient analysis and steady state analysis is done for gearbox casing with the heat generation and oil temperature taken as a thermal load in ANSYS 10.0.In the temperature distribution is obtained for two, four and eight hours. Comparison of results are done for various location in gear box.

## 7.1.1 COMPARISON OF RESULTS

### The Temperature of Various Locations in Gearbox with Respect to Time :

For comparing the cooling in various parts of gear box these six locations are taken :

1. Temperature at outer bottom casing nearer to input bearing
2. Temperature at middle of outer bottom casing nearer to third intermediate shaft
3. Temperature at right side of outer bottom casing when viewed from front.
4. Temperature at back side of outer bottom casing nearer to second intermediate shaft.
5. Temperature at bottom most portion inside of bottom casing.
6. Temperature at inside of bottom casing nearer to third intermediate shaft.

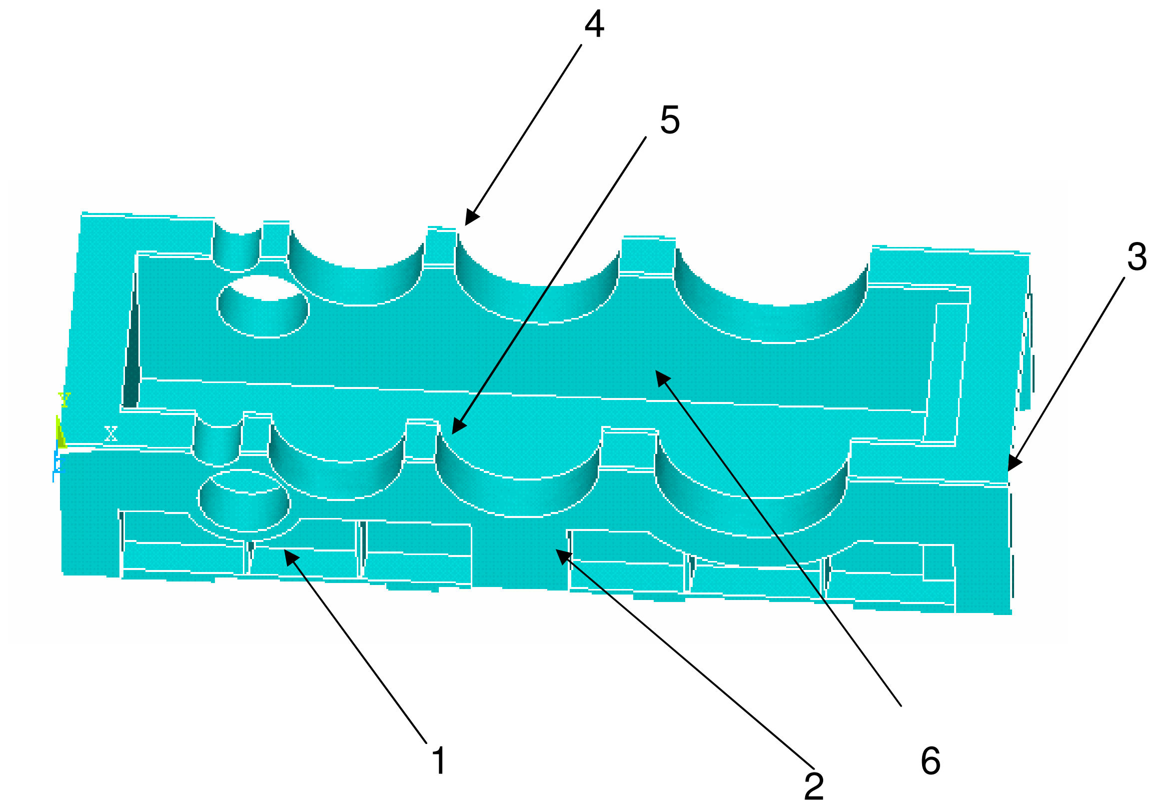



**FIG. 7.1 LOCATION FOR COMPARING THE RESULT**

## Temperature At Various Locations

**TABLE 7.1 TEMPERATURE VARIATION WITH RESPECT TO TIME**

| Temp at Location ($^oC$) | 120 min | 240 min | 480 min |
|---|---|---|---|
| 1 | 43.3 | 41.4 | 39.1 |
| 2 | 43.0 | 41.2 | 38.89 |
| 3 | 42.8 | 41.0 | 38.8 |
| 4 | 43.1 | 41.2 | 39.14 |
| 5 | 43.1 | 42.4 | 41.4 |
| 6 | 43.6 | 42.6 | 42.2 |

### 7.1.2 GRAPHICAL REPRESENTATION

As the temperature at various location are known. A graph is plotted to observe the variation.

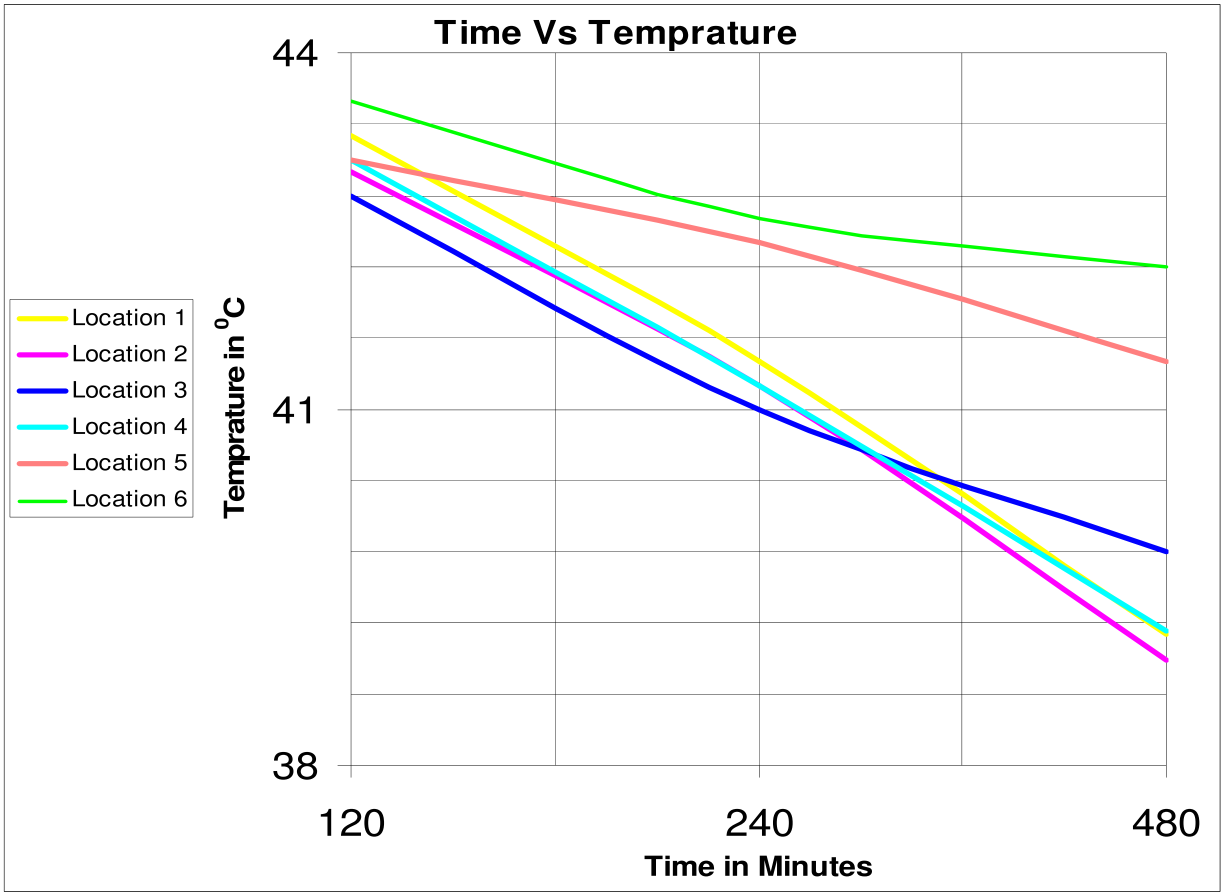



**FIG. 7.2 GRAPH FOR TEMPERATURE AT VARIOUS LOCATIONS WITH RESPECT TO TIME**

## 7.2 DISCUSSION

- .The Heat losses which came from AGMA are less than the loss from ELECON Engg. Co. Ltd, The reason for this difference is because ELECON Engg Co. Ltd is using formulae those are fourteen years old and some parameter are taken in those formulae are experiment based where as formulae from AGMA are recently revised.

- Thermal analysis is done with the basis of power losses which are calculated from ELECON because they are more than losses from AGMA so for Factor of safety purpose analysis is done on the basis of ELECON formulae.

- Graph shows that These Locations

 5 -- Temperature at bottom most portion inside of bottom casing bearing.

 6 – Temperature at inside of bottom casing nearer to third intermediate shaft.

are having the cooling rate are very slow.

# CHAPTER 8

# CONCLUSION
# AND
# SCOPE FOR FUTURE WORK

# CONCLUSION AND SCOPE FOR FUTURE WORK

## 8.1 Conclusion

1. The heat losses are calculated from ELECON formulae are more than those from AGMA Formulae. The probable reason as known from ELECON company personnel is, using very high factor of safety at the time of determination of the relationship.

2. In the Steady State analysis, it is observed that the portion which is in contact with oil having higher temperature than other part of gear box. As moving from bottom assembly to top, the temperature is getting low.

3. In Transient analysis when the heat generation is taken as a load, the drop in the temperature with respect to time is approximately linear in every time slot two hours, four hours and eight hours.

4. It is observed that the temperatures at bottom most portion inside of bottom casing bearing and inside of bottom casing nearer to third intermediate shaft are greater than the atmosphere temperature even after eight hours cooling.

5. As analyzed in Transient analysis for eight hours using three modes of load viz. heat generation, inside oil temperature with surface convection coefficient and lower outer casing temperature, the temperatures which came from the heat generation is taken as a load are slightly higher than the temperatures from the other two conditions.

## 8.2 Scope For Future Work

As the thermal analysis of helical gearbox casing PE-50 is done taking inside oil temperature and heat generation to be consideration ,further precise analysis can be done with taking the gearbox internals like shafts wheel bearings in account. and results can be compared Effect of heat fumes can also be taken as consideration The temperature distribution and variation in the internals of gear box can be obtained.

Effect of air velocity inside the gear box can also be taken in account and computational fluid dynamics of lubricating oil can be done with the help of analysis software.

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# REFERENCES

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# APPENDIX

# APPENDIX

## PROGRAM TO CALCULATE POWER LOSSES IN GEAR BOX

```
#include<stdio.h>
#include<conio.h>
#include<math.h>

void main()
{
  int a,Zp,Zw,s,i,Z;
  float V,B,m,Pkw,u,Fl,d,N,Pcl[20],Pwl[20],Pml[20],Pbl[20];
  float TPcl=0,TPwl=0,TPml=0,TPbl=0,din,dout,TPwlcl=0;
  float sy,A,cp,dw,Pn,Pwlcl,c1,rw,Tp,np,fin,Mg,Ro,ro,Rw1,rw1,f;
  float Hs,Ht,K,M,fyt;

  clrscr();
  printf("\n\nENTER 1:- LOSSES FROM ELECON");
  printf("\n\nENTER 2:- LOSSES FROM AGMA");
  printf("\n\nWHICH METHOD YOU WANT TO USE:-");
  scanf("%d",&a);
  printf("\nENTER THE NUMBER OF STAGES:->");
  scanf("%d",&s);

  if(a==1)
   {
        {
          for(i=1;i<=s;i++)
          {
                printf("\nENTER THE VALUE OF VELOCITY OF WHEEL OR PINION:->");
                scanf("%f",&V);
                printf("\nENTER THE VALUE OF FACE WIDTH:->");
                scanf("%f",&B);
                printf("\nENTER THE VALUE OF MODULE:->");
                scanf("%f",&m);

          if(m>=8e-3)
          {
            Pcl[i]=(pow(V,2.14834)*B)/(pow(10,2.1969847)*500*pow((m/9.125),1.2));

            TPcl=TPcl+Pcl[i];
           }

          if(m<=5.5e-3)
```

```
{
  Pcl[i]=(pow(V,2.637661)*B)/(pow(10,3.5812501)*324*pow((m/4.67),1.2));

  TPcl=TPcl+Pcl[i];
}

if(m>5.5e-3 && m<8e-3)
{
  Pcl[i]=(pow(V,2.2653876)*B)/(pow(10,2.6607005)*432*pow((m/6.7),1.2));

  TPcl=TPcl+Pcl[i];
}
}
        printf("\n\n\tTOTAL CHURNING LOSS IS :-> %f WATT",TPcl*1000);


for(i=1;i<=s;i++)
{
 Pwl[i]=(1.25*Pcl[i]);
 TPwl=TPwl+Pwl[i];


}
printf("\n\n\tTOTAL WINDAGE LOSS IS :-> %f WATT",TPwl*1000);

getch();
clrscr();

for(i=1;i<=s;i++)
{
  printf("\nENTER THE VALUE OF TEETH ON PINION:->");
  scanf("%d",&Zp);
  printf("\nENTER THE VALUE OF TEETH ON WHEEL:->");
  scanf("%d",&Zw);
  printf("\nENTER INPUT BEARING POWER:->");
  scanf("%f",&Pkw);

      Pml[i]=((0.09*Pkw*(Zp+Zw))/(Zp*Zw))*1000;

      TPml=TPml+Pml[i];

}
 printf("\nTOTAL MESHING LOSS IS :-> %f WATT",TPml);


}
```

```
getch();
clrscr();
for(i=1;i<=(2*s)+2;i++)
{

printf("\nENTER THE VALUE OF BRG. COEFF. OF FRICTION:->");
scanf("%f",&u);
printf("\nENTER THE VALUE OF LOAD ON BRG.:->");
scanf("%f",&Fl);
printf("\nENTER THE VALUE OF DIAMETER OF SHAFT:->");
scanf("%f",&d);
printf("\nENTER THE VALUE OF SPEED IN RPM:-> ");
scanf("%f",&N);

Pbl[i]=((u*Fl*d*N)/(2*9550))*1000;

TPbl=TPbl+Pbl[i];



}
        clrscr();
        printf("\nTOTAL VALUE OF BRG. LOSSES IS :->%f WATT",TPbl);

        printf("\n\n\nTOTAL POWER LOSSES IN GEAR BOX:->%f WATT",TPcl+TPbl+TPml+TPwl);


}
else

{

 clrscr();
 printf("\nENTER THE VALUE OF HELIX ANGLE:->");
 scanf("%f",&sy);
 sy=sy*(M_PI/180);
 printf("\nENTER THE VALUE OF ABSOLUTE VISCOSITY:->");
 scanf("%f",&cp);



for(i=1;i<=(2*s);i++)
{
  printf("\nENTER THE REFERANCE DIAMETER:->");
  scanf("%f",&dw);
```

```
  printf("\nENTER THE VALUE OF MODULE:->");
  scanf("%f",&m);
  printf("\nENTER THE VALUE OF SPEED IN RPM:->");
  scanf("%f",&N);
  printf("\nENTER THE VALUE OF FACE WIDTH:->");
  scanf("%f",&B);

A=(22440/cp);

Pn=1/(m*cos(sy));


 Pwlcl=((dw*dw)*(N*N)*B*(cos(sy)*cos(sy)))/(126000*Pn*A);

 TPwlcl=TPwlcl+Pwlcl;

  }

 printf("\nCHURNING AND WINDAGE LOSSES ARE:->%f",TPwlcl);
 getch();

 clrscr();

 for(i=1;i<=s;i++)
 {
        printf("\nENTER THE VALUE OF LUBRICATION FACTOR:->");
        scanf("%f",&c1);
        printf("\nENTER THE VALUE OF TEETH ON PINION:->");
        scanf("%d",&Zp);
        printf("\nENTER THE VALUE OF TEETH ON WHEEL:->");
        scanf("%d",&Zw);
        printf("\nENTER THE VALUE OF REFERANCE RADIUS:->");
        scanf("%f",&rw);
        printf("\nENTER THE VALUE OF TORQUE ON PINION:->");
        scanf("%f",&Tp);
        printf("\nENTER THE VALUE OF PINION SPEED IN RPM:->");
        scanf("%f",&np);
        printf("\nENTER THE VALUE OF HELIX ANGLE:->");
        scanf("%f",&sy);
        printf("\nENTER THE VALUE OF NORMAL PRESSURE ANGLE:->");
        scanf("%f",&fin);
        printf("\nENTER THE VALUE OF STAGE RATIO:->");
        scanf("%f",&Mg);
        printf("\nENTER THE VALUE OF ADDENDUM RADIUS OF WHEEL:->");
        scanf("%f",&Ro);
        printf("\nENTER THE VALUE OF ADDENDUM RADIUS OF PINION:->");
```

```
        scanf("%f",&ro);
        printf("\nENTER THE VALUE OF REFERANCE RADIUS OF WHEEL:->");
        scanf("%f",&Rw1);
        printf("\nENTER THE VALUE OF REFERANCE RADIUS OF PINION:->");
        scanf("%f",&rw1);
        printf("\nENTER THE VALUE OF FACE WIDTH IN INCH:->");
        scanf("%f",&f);
        printf("\nENTER THE VALUE OF VELOCITY OF WHELL:->");
        scanf("%f",&V);
        printf("\nENTER THE VALUE OF TRANSVERSE PRESSURE ANGLE:->");
        scanf("%f",&fyt);

         fyt=fyt*(M_PI/180);


        Hs=(Mg+1)*(pow((pow((Ro/Rw1),2)-pow(cos(fyt),2)),0.5)-sin(fyt));

        Ht=((Mg+1)/Mg)*(pow((pow((ro/rw1),2)-pow(cos(fyt),2)),0.5)-sin(fyt));

        M=(2*cos(fyt)*(Hs+Ht))/(pow(Hs,2)+pow(Ht,2));

        K=(Tp*(Zp+Zw))/(2*f*pow(Rw1,2)*Zw);

        u=(pow(K,0.35))/(c1*pow(V,0.23));

        Pml[i]=(u*Tp*np*pow(cos(sy),2))/(63025*M);

        TPml=TPml+Pml[i];

 }

 printf(" \nMESHING LOSSES ARE %f",TPml);

getch();
clrscr();
for(i=1;i<=(2*s)+2;i++)
{

printf("\nENTER THE VALUE OF BRG. COEFF. OF FRICTION:->");
scanf("%f",&u);
printf("\nENTER THE VALUE OF LOAD ON BRG.:->");
scanf("%f",&Fl);
printf("\nENTER THE VALUE OF INNER DIAMETER OF SHAFT:->");
scanf("%f",&din);
printf("\nENTER THE VALUE OF OUTER DIAMETER OF SHAFT:->");
scanf("%f",&dout);
```

```
    printf("\nENTER THE VALUE OF SPEED IN RPM:-> ");
    scanf("%f",&N);

    Pbl[i]=((u*Fl*(dout+din)*N)/(252100));

    TPbl=TPbl+Pbl[i];

    }
    printf("\nTOTAL VALUE OF BRG. LOSSES IS :->%f WATT",TPbl*745.7);


  }



getch();
}
```