1992

# Heat Transfer Model of a Rotary Compressor

S. K. Padhy
*General Electric Company*

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**Sisir K. Padhy**
General Electric Company
Appliance Park 5-2North, Louisville, KY 40225

## ABSTRACT

Energy improvements for a rotary compressor can be achieved in several ways such as: reduction of various electrical and mechanical losses, reduction of gas leakage, better lubrication, better surface cooling, reduction of suction gas heating and by improving other parameters. To have a better understanding analytical/numerical analysis is needed. Although various mechanical models are presented to understand the mechanical losses, dynamics, thermodynamics etc.; little work has been done to understand the compressor from a heat transfer stand point. In this paper a lumped heat transfer model for the rotary compressor is described. Various heat sources and heat sinks are analyzed and the temperature profile of the compressor is generated. A good agreement is found between theoretical and experimental results.

## NOMENCLATURE

| | |
|---|---|
| $D_i$ | inner diameter |
| $D_o$ | outer diameter |
| $D_{mf}$ | upper bearing plate outer diameter |
| $\Delta x$ | conduction length |
| $\dot{Q}$ | rate of heat transfer |
| $S_i$ | heat generation within element $i$ |
| $\Delta \dot{E}$ | change in internal energy |
| $D_{casei}$ | case inside diameter |
| $D_{caseo}$ | case outside diameter |
| $A_{i,j}$ | coeffients of the simultaneous equations [coefficients of the matrix] |
| $h_r$ | effective radiation heat transfer coefficient |
| $\varepsilon$ | emissivity |
| $h_4$ | heat transfer coefficient between the case inside and the gas above the stator/rotor |
| $h_\infty$ | heat transfer coefficient of the case and outside |
| $z_1$ | height of the case above the stator |
| $z_2$ | height of the stator |
| $B_i$ | known parts of the system equations |
| $\dot{m}$ | mass flow rate of the gas |
| $\dot{W}$ | rate of work done |
| $\sigma$ | Stefan-Boltzmann constant |

| | |
|---|---|
| $\Delta T$ | temperature difference |
| $T_i$ | temperature of *i*-th element |
| $H_{i,j}$ | thermal conductances |
| $k_i$ | thermal conductivity of *i*-th element |
| $N_u = \frac{hL}{k}$ | average nusselt number |
| $Gr = \frac{g\beta\Delta T L^3}{\nu^2}$ | Grashof number, g acceleration due to gravity, β volumetric thermal expansion coefficient, ν kinematic viscosity |
| $h$ | heat transfer coefficient |
| $Pr = \frac{c_p\mu}{k}$ | Prandtl number, $c_p$ specific heat |
| $Ra = Gr.\ Pr$ | Rayleigh number |
| $Re = \frac{\rho v x}{\mu}$ | Reynolds number, ρ density, v velocity, x length, μ viscosity |
| $t$ | case thickness |
| $Ta$ | Taylor number |

## 1. INTRODUCTION

Rotary Compressors have been used in refrigerators and room airconditioners for several years. The enforcement of strict energy regulations for the power consumption as well as the goal of the CFC (the present refrigerant) removal by the year of 1995, are driving the design methodology of compressors to take new shapes. To achieve these goals the key is to understand the mechanics of the system and to develop analytical techniques to analyze the effect of various geometry, clearances, material properties, and temperature in the system. This understanding will lead to an optimum design of the compressor from energy, reliability, and economics stand point.

The rotary compressor is a high side compressor with all the components exposed to either lubricant oil or the hot compressed gas. The temperature profile as well as the heat transfer among the components of the compressor play a critical role on the reliability as well as energy efficiency. The suction gas heating has been of interest since it has an adverse effect on the energy rating of the compressor. Again an understanding of various heat transfers between the components, both qualitatively as well as quantitatively will help the designer to do design modifications that will result a compressor with improved reliability ands the energy rating. Another area of interest can be the effect of the temperature on the bearings dynamics of the compressor.

## 2. PREVIOUS WORK

The understanding of the heat transfer of a rotary compressor involves compressor dynamics, mechanical and electrical losses, thermodynamic analysis, lubrication, & determination of heat transfer coefficients for the various interfaces of the compressor. Although works have been reported in the areas of dynamics, mechanical losses, lubrication (a complete theoretical study by the author [1] recently) and thermodynamics of the compressor, no work has been reported for the heat transfer of the rotary compressor except for an internal report of General Electric company.

Brok et al. [2] reported their research on the cylinder heat transfer for an reciprocating compressor. Applying simple overall thermodynamic analysis and mathematical modeling of the cylinder, suction chamber and discharge chamber they conclude that the heat transfer has a lesser effect on performance as opposed to the standard belief of higher effect. Kawai et al [3] discussed improvement in performance with the reduction of suction gas. The efficiency improvement of 6 to 10% is reported by modifying the suction muffler of an reciprocating compressor. Meyer and Thompson [4,5] carried out research both analytically and experimentally only for the suction gas of an reciprocating hermetic compressor. A steady state analysis was done. Heat transfer correlations were adapted from literature and some are experimentally determined. As a part of the same research Srikanth and Thompson [6] carried out a flow visualization study to evaluate the gas flow pattern inside a hermetic reciprocating compressor. Their conclusions are again to modify the muffler design for improved efficiency. Adair et al [7] studied the instantaneous heat transfer between the cylinder wall and the gas of an reciprocating compressor. They proposed a new correlation based on their experimental work. In an recent study Recktenwald et al [8] and Recktenwald [9] investigated the same heat transfer using numerical methods. The GE internal report [10] describes a lumped heat transfer model for a rotary compressor. However all the components of the compressor are not modeled and some experimental heat transfer coefficients are used. The present work models the compressor in total, to use better theoretical correlations and to produce a more accurate model.

## 3. PRESENT WORK

The present work deals with a lumped (averaged) parameter formulation of the heat transfer for a rotary compressor. The laws governing the system are said to be lumped if the their terms are independent of space and to be distributed if the terms depend on space[11]. The fundamental laws governing the system in the present discussion are conservation of mass, and first law of thermodynamics. The conservation of mass is given by

$$\dot{m} = 0$$

and the first law of thermodynamics is given by

$$\Delta\dot{E} = \dot{Q} - \dot{W}$$

In the present case there are interactions between more that one lumped mass and the law of conservation of mass for *i*-th element is

$$\sum_{j=1}^{n} \dot{m}_{j,i} = \sum_{k=1}^{m} \dot{m}_{i,k} + \frac{dM_i}{dt}$$

where *j*'s represent the elements from which the mass flows to *i-th* element and *k*'s represent the elements to which mass flows from element *i*. $M_i$ represents the instaneous mass of the element *i*. Similarly the first law of thermodynamics for the *i-th* element becomes

$$\dot{Q}_i + \sum_{j=1}^{n} \dot{m}_{j,i} h_{j,i} = \dot{W}_i + \sum_{k=1}^{m} \dot{m}_{i,k} h_{i,k} + \frac{d(M_i u_i)}{dt}$$

Although the inlet and exit enthalpy of the element are different an averaged value is used in the calculations.

The principal modes of heat transfer considered inside the rotary compressor are conduction and convection, and radiation is neglected. For the compressor case outside free and forced convection are considered in addition to the radiation. The conduction heat transfer is

$$\dot{Q} = \frac{k_i A_i \Delta T}{\Delta x}$$

The convective heat transfer is given by

$$\dot{Q} = h_i A_i \Delta T$$

and the radiative heat transfer equation is

$$\dot{Q} = \varepsilon\sigma A\left(T_i^4 - T_\infty^4\right)$$

Writing an effective heat transfer coefficient as

$$h_r = \varepsilon\sigma\left(T_i + T_\infty\right)\left(T_i^2 + T_\infty^2\right)$$

the radiative heat transfer equation becomes

$$\dot{Q} = h_r A\left(T_i - T_\infty\right)$$

All these equations can be conveniently represented by introducing the concept of thermal conductance and the general equation for heat transfer becomes

$$\dot{Q} = H_i \Delta T$$

$H$ is calculated from actual geometry, thermal properties of elements and heat transfer coefficients.

The total rate of heat transfer to an element $i$ can be written as the sum of heat transfer from other elements either to the element or from the element, and the rate of heat generation $S$ within the element.

$$\dot{Q} = \sum_{j=1}^{n} H_{j,i}\left(T_i - T_j\right) - \dot{S}_i$$

A steady state analysis is carried out and the equations became

$$\sum_{j=1}^{n} \dot{m}_{j,i} = \sum_{k=1}^{m} \dot{m}_{i,k} \qquad \text{as } \frac{d\mathrm{M}_i}{dt} = 0,$$

$$\dot{Q}_i + \sum_{j=1}^{n} \dot{m}_{j,i} h_{j,i} = \dot{W}_i + \sum_{k=1}^{m} \dot{m}_{i,k} h_{i,k} \qquad \text{as } \frac{d\left(\mathrm{M}_i u_i\right)}{dt} = 0$$

and

$$\dot{Q} = \sum_{j=1}^{n} H_{j,i}\left(T_i - T_j\right) - \dot{S}_i = 0$$

Heat addition to the element is considered positive and the heat transfer from the element is considered negative. Internal heat generation (including frictional heat generation) are considered positive.

The lumped and isothermal elements of the compressor are selected such that they conform with the natural geometric boundaries. These boundaries include solid parts, fluid masses and flow paths. Applying first law of thermodynamics and conservation of mass to the elements of the compressor a system of equations are formulated. These equations are solved by Gauss-Jordan method to calculate different element temperatures. The rest of this section deals with describing the system of equations, the solving procedure, the equations for the calculation of heat transfer coefficients, and the electrical and mechanical losses.

### *3.1 System of Equations*

The formulation of the system of equations is described in the following paragraphs. The compressor is divided into 22 elements [Figure 1] and these elements are modeled yielding 22 equations. For example element 1 is presented here.

*Element 1: Case portion above the stator*

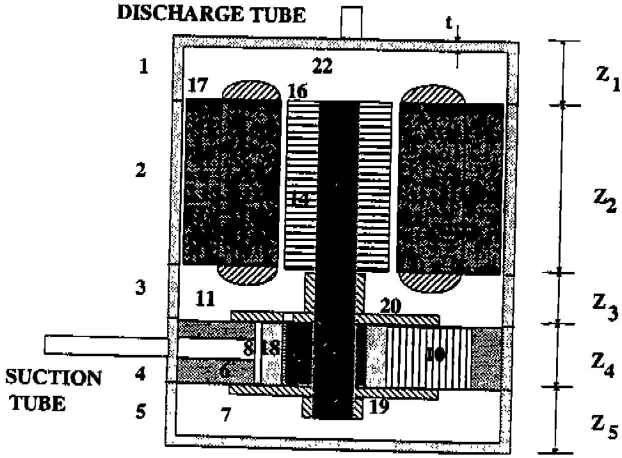
This element is connected to the outside of compressor, to the gas above the stator inside the compressor and to the case element that is attached to the stator. The energy balance gives

$$H_{1,2}\left(T_1 - T_2\right) + H_{1,\infty}\left(T_1 - T_\infty\right) + H_{1,22}\left(T_1 - T_{22}\right) = 0$$

or

$$T_1\left(H_{1,2} + H_{1,\infty} + H_{1,22}\right) - H_{1,2}T_2 - H_{1,\infty}T_\infty - H_{1,22}T_{22} = 0$$

This can be written in the following form, with the A's as the matrix elements to solve the




[Figure 1. Compressor Schematic with 22 elements are shown except the compressed gas (Element 9). The element names are described in Table 1.]

simultaneous equations.

$$A_{1,1}T_1 + A_{1,2}T_2 + A_{1,22}T_{22} = B_1$$

Where

$$A_{1,1} = H_{1,2} + H_{1,\infty} + H_{1,22}\ ,\quad A_{1,2} = -H_{1,2}\ ,\quad A_{1,22} = -H_{1,22}\ ,\quad B_1 = H_{1,\infty}T_\infty$$

$$H_{1,2} = k_1\pi\left(D_{caseo}^2 - D_{casei}^2\right)/4z_2\ ,$$

$$H_{1,\infty} = \pi\left(h_r + h_\infty\right)\left(D_{caseo}z_1 + \left(D_{caseo}^2 - D_{casei}^2\right)/4\right)$$

$$H_{1,22} = h_4\pi\left(D_{caseo}(z_1 - t) + \left(D_{casei}^2 - D_{dis}^2\right)/4 + D_{dis}t\right)$$

Similarly the other elements are modeled. In general the equations can be represented as

$$\sum_{j=1}^{n} A_{i,j}T_i = B_i$$

where

$$\underset{j\neq 2}{A_{i,j}} = -H_{i,j}\ ,\quad A_{i,i} = \sum_{j=1(j\neq i)}^{n} H_{i,j}$$

and $B$ is the heat quantity that may be the heat transferred to the sorrounding or the mechanical losses or may be the electrical losses.

### 3.2. Heat transfer coefficients calculation

#### *3.2.1. Case outside*

The compressor is placed in the cabinet of the refrigerator or an air conditioner unit. Depending on the air flow on the compressor the heat transfer can be natural convection or forced convection in addition to the radiation. Modeling the compressor as a cylinder in cross flow to the air stream the forced convection correlation used is [13]

$$N_u = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1 + \left(\frac{0.4}{\mathrm{Pr}}\right)^{2/3}\right]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}}{282000}\right)^{5/8}\right]^{4/5}$$

for $10^2 < \mathrm{Re} < 10^7$; $\mathrm{Pe} > 0.2$

The following equation is used for the free convection [13]

$$N_u = \left[ 0.6 + \frac{0.378 Ra^{1/6}}{\left\{1+(0.559/\Pr)^{9/16}\right\}^{8/27}} \right]^2$$

for $10^{-5} < Ra < 10^{12}$

and the heat transfer coefficient is calculated as

$$h = \frac{N_u k}{x}$$

In calculating the radiation heat transfer, the temperature of the case is unknown. Hence a temperature is assumed to be close to the gas temperature coming out of the compressor.

*3.2.2. Oil sump heat*

The heat transfer between the case and oil is very complicated. The heat transfer between the oil and the vertical case portion is modeled as free convection as the oil circulating through the compressor mechanism comes down to the sump and there is a continuous motion. However as the shaft rotates, the bottom portion of the case is subjected to both free and forced convection. The oil velocity in the sump is calculated knowing the volume rate of oil flow [previous paper] through the shaft. The equation for the vertical portion is [14]

$$N_u = 0.68 + \frac{0.67 Ra^{1/4}}{\left[1+(0.492/\Pr)^{9/16}\right]^{4/9}}$$

for $Ra < 10^9$

Otherwise the following formula is used

$$N_u = 0.825 + \frac{0.387 Ra^{1/6}}{\left[1+(0.492/\Pr)^{9/16}\right]^{8/27}}$$

For the bottom circular disk for free convection the equation used is[15]

$$N_u = 0.818 \Pr^{0.034} Ra^{0.2}$$

and for the forced convection

$$N_u = 0.664 \mathrm{Re}^{0.5} \Pr^{1/3}$$

The overall mixed convection Nusselt number is calculated as follows;

$$(N_u)^3_{total} = (N_u)^3_{forced} + (N_u)^3_{free}$$

*3.2.3. Suction tube heat transfer calculation*

In one cycle the rotary compressor compresses the gas as well as draws the gas into the suction chamber. Knowing the volume of gas, and the diameter of the suction tube portion inside the cylinder as well as outside the compressor, the velocity of the gas is determined and Reynolds number is calculated. This is a case of flow through pipes and the Nusselt number is calculated using the following correlation[15]. For laminar flow

$$N_u = 4.36$$

and for turbulent flow

$$N_u = 5 + 0.015 \mathrm{Re}^a \Pr^b \; ; \quad \text{for } 0.1 < \Pr < 10^5 \text{ and } 10^4 < \mathrm{Re} < 10^6$$

where $a=0.88-\dfrac{0.24}{4+\Pr}$ , $b=0.3333+0.5e^{(-0.6\Pr)}$

*3.2.4. Suction Chamber heat transfer coefficient*

The suction chamber is modeled as a circular duct and the hydraulic diameter is evaluated. The velocity in the chamber is calculated from the mass flow rate which is known from the calorimeter tests. No swirl effects are assumed in the suction chamber. The Nusselt number are calculated using correlations described in 3.2.3.

*3.2.5. Compression chamber heat transfer coefficient*

There are no correlations reported in literature for the heat transfer coefficient calculation for the rotary compressor. The gases involved in heat transfer at high temperatures and undergo a rapid physical change. Further difficulty is introduced by the uncertainties of gas flow patterns, the state of boundary-layer developments as well as the complexities induced by the change of compression volume boundaries. In absence of more exact data the following equation is used for the calculation of Nusselt number [7] although it was originally formulated for reciprocating compressors.

for laminar: $N_u=4.36$

for turbulent: $N_u=0.053\,\mathrm{Re}^{0.8}\,\Pr^{0.6}$

where the Reynolds number is calculated using the swirl velocity. The swirl angular velocity is taken to be twice that of shaft angular velocity[7]. The compression chamber is modeled as a circular duct and the hydraulic diameter is used in calculation of the Reynolds number.

*3.2.6. Stator and case channel heat transfer coefficient*

The volume of gas flow through the channel is determined with known volume rate of flow from the compression chamber (In the present the compression volume is calculated using ideal gas law and polytropic coefficient of 1.16[16]). As the gas flows through the channel and rotor gap, volume flow rate is calculated for each of them knowing the cross sectional areas of the channel and the gap. The effect of heat interaction in gap as well as in the channel, on the gas is neglected. The velocity of flow and the Reynolds number is calculated using the hydraulic diameter concept. Depending on the Reynolds number the Nusselt number is calculated from the formulations described in section 3.2.3.

*3.2.7. Heat transfer coefficients in rotor and stator gap*

The calculation of heat transfer coefficient in the rotor gap has been of great interest for many years [17-26]. Although many experimental works are undertaken, no attempts have made to simulate theoretically the heat transfer rates for the combined effects of rotation and axial flow[26]. The flow field in the gap is complicated by the onset of vortices and axial flow velocity. In the compressor, the stator is fixed while the rotor rotates. The radial velocity is determined from the angular speed while the axial velocity in the present case is calculated from the volume rate of flow through the gap. The calculation of Nusselt number involves mean radius, axial Reynolds number, Taylor's number, and critical Taylor's number. The following correlations are used in the present work.

$N_u=4.294\sigma^{0.4845}$; for $1<\sigma<4.817$

and $N_u=5.08\sigma^{0.3507}$; for $\sigma>4.817$

where $$\sigma = Ta/Ta_{cr}\ ,\ Ta = \frac{\omega^2 r_m \delta^3}{\nu^2}\ ,\ Ta_{cr} = \frac{\omega_{cr}^2 r_m \delta^3}{\nu^2}\ ,\ \omega_{cr}^2 = \frac{\pi^4 \nu^2 r_m}{S\delta^3 r_i^2}$$

$$\delta = r_o - r_i\ ,\ r_m = (r_o + r_i)/2$$

and $$S = 0.0571\left(1-\frac{0.652\delta}{r_i}\right)+0.00056\left(1-\frac{0.652\delta}{r_i}\right)^{-1}$$

These correlations cover a very good range of axial Reynolds number of about 250 to 1000. In the present work the axial Reynolds number is approximately 720.

### *3.2.8. Heat transfer coefficient between the gas above the cylinder and rotor bottom*

The compressed gas emerges from the clearances of the volume cup and comes in contact with the rotor bottom. The rotation of the rotor induces velocity in the gas. The effect of the axial velocity impinging the rotor bottom is neglected. The Reynolds number is calculated using an average radial velocity of the rotor which is calculated utilizing mean diameter concept. The formula used for the calculation of Nusselt number is adapted from Kreith[27].

$$N_u = \frac{\mathrm{Re}\,\mathrm{Pr}\left(\sqrt{C_{dr}/2}\right)}{5\,\mathrm{Pr}+5\ln(5\,\mathrm{Pr}+1)+\sqrt{2/C_{dr}}-14}$$

where $$C_{dr} = \frac{1}{-2.05+4.07\log_{10}\mathrm{Re}\sqrt{C_{dr}}}$$

and $C_{dr}$ is solved numerically using Newton-Raphson [28] method.

### *3.2.9. Copper winding of Stator and the gas above the cylinder*

The location of copper winding is very close to the rotor and hence the gas velocity at copper windings will be fairly same that of the gap. Hence the heat transfer coefficient is taken as same as that of the rotor gap.

### *3.2.10. Heat transfer coefficient between the stator bottom and the gas above cylinder*

The velocity of the gas near the rotor is equal to the radial velocity of the rotor . Assuming a swirl motion in this chamber, the velocity of the gas is assumed to be equal to the rotor velocity. This value is used to calculate the Reynolds number. The heat transfer is modeled as a flat plate and the following correlations [14] are used.

$$N_u = 0.664\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3};\qquad \text{for Re} < 10^5$$

$$N_u = \mathrm{Pr}^{1/3}\left(0.037\,\mathrm{Re}^{0.8}-850\right);\ \text{for } 10^5 < \mathrm{Re} < 10^7$$

and the heat transfer coefficient is determined as

$$h = \frac{N_u k}{(D_o - D_i)}$$

### *3.2.11. Heat transfer coefficient between the case and the gas above cylinder*

As it is not known how the gas behaves in the space below the stator and above the cylinder, the gas and case interaction is modeled as gas flowing to the case perpendicularly. The velocity of the gas is taken as the rotor velocity value as described in section 3.2.10. Following Schlichting [29] the equation becomes

$$N_u = \frac{\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/2}}{\sqrt{0.5\pi}}$$

*3.2.12. Heat transfer coefficients between the gas above cylinder and cylinder*

As the cylinder has three kidney shaped ports, it is more complicated to calculate the heat transfer coefficient accurately. In the present work the cylinder is modeled as a circular disk and the heat transfer is modeled as a flat plate heat transfer for this particular calculation. The velocity of the gas above the cylinder is taken to be equal to the rotor velocity assuming swirl is there. The equations governing the heat transfer coefficient [14] are

$$N_u = 0.664\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}; \qquad \text{for Re} < 10^5$$

$$N_u = \mathrm{Pr}^{1/3}\left(0.037\,\mathrm{Re}^{0.8} - 850\right); \text{for } 10^5 < \text{Re} < 10^7$$

and the heat transfer coefficient is determined as

$$h = \frac{N_u k}{\left(D_{case1} - D_{mf}\right)}$$

*3.2.13. Heat transfer coefficient between the upper bearing plate and the gas above the cylinder*

The analysis is same as that described in section 3.2.12 except that the cylinder dimensions are replaced by the bearing plate dimensions. The velocity of gas is again taken to be the rotor radial velocity.

*3.2.14. Heat transfer coefficient between the stator top, rotor top, copper winding, shaft top and the gas above stator/rotor*

The analysis is same as that of the gas above cylinder and the stator/rotor bottom and copper winding. The velocity of gas is calculated from continuity of mass. However for the heat transfer calculation for the rotor and the shaft top, velocity is taken as the rotor velocity and Kreith's formula is used.

*3.2.15. Heat transfer coefficient between the case above the stator and the gas above the stator/rotor*

The gases coming from the rotor gap, stator channel mix in this chamber. The gas flows to the condenser through the discharge tube which is connected to this chamber. Using continuity of mass the velocity in the chamber is evaluated. The heat transfer is modeled as a flat plate heat transfer and the equations usedare described in section 3.2.12.

*3.2.16. Contact resistances*

The stator is shrink fit to the case and similarly is the rotor to the shaft. The contact stresses are evaluated knowing the interfearance of the contacting surfaces. The contact pressure is calculated [30] and the contact resistance is found from heat transfer hand book [31]. The contact pressure is given by

$$p = \frac{\eta}{b\chi}$$

where
$$\chi = \frac{1}{E_i}\left(\frac{b^2 + a^2}{b^2 - a^2} - v_i\right) + \frac{1}{E_o}\left(\frac{c^2 + b^2}{c^2 - b^2} - v_o\right)$$

and $a$ is the radius of the hole of the inner member, $b$ is the radius of contact surface, $c$ is the radius of outer surface, $E$'s are the youngs modulus for innet and outer members, v's are the poisson's ratio, and $\eta$ is the radial interference.

*3.2.17. Thermal conductivities*

The thermal conductivities are adapted from various sources [31,32].

**3.3. Mechanical and Electrical Losses**

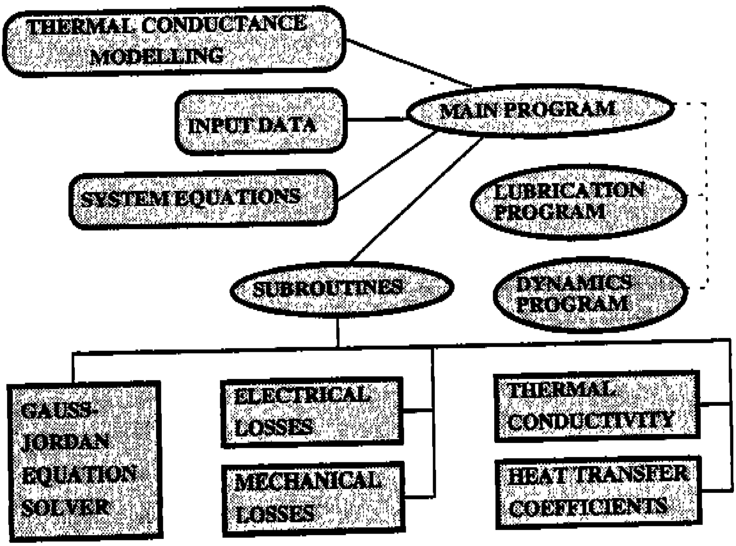
Mechanical losses in the rotary compressor have been described in the literature extensively [33,34] and also they can be found from fundamental principles [35]. Electrical losses on the other hand depends on the configuration and mass of the components and is usually provided in forms of charts from the manufacturer. For the present work the electrical losses, i.e. iron and copper losses are determined from the data of the particular motor used in the compressor. However the bearing losses should be modified by a correcting factor for the turbulent case. If $P_l$ is the loss at laminar condition then the loss $P_t$ at turbulent condition becomes[35]

$$P_t = 0.039\,\mathrm{Re}^{0.57}\,P_l\,, \qquad \text{for Re} > 1600$$

It is assumed that all the work transforms into heat. The present work accounts for the mechanical losses occur at the upper bearing plate journal, lower bearing plate journal, the journal bearing between the shaft eccentric and the roller, upper and lower thrust bearing losses, vane side, vane top and bottom losses, vane and roller loss at the contact surface, roller and bearing plate losses, and the loss at stator and rotor gap. In addition to these mechanical losses copper and iron losses at the motor is also included in the analysis.

**3.4. Implementation**

A computer code is developed for the heat transfer analysis of the compressor. The architecture of the computer program is shown in Figure 2. The program code is developed in FORTRAN and it consists of more than 2000 lines of code. In the main program the data is declared, and the thermal conductances and the system equations are coded. There are five subroutines in this program executing




[Figure 2. Program Architecture]

the tasks of simultaneous equation solution, electrical loss calculation, mechanical loss calculation, providing values of thermal conductivity of the components used in compressor, and heat transfer coefficients calculations. Some inputs for this program are generated using the Lubrication and Dynamics computer programs. The simultaneous linear equations are solved using Gauss-Jordan method.

The input to the program are the ambient temperature, the power consumption of the compressor, the gas temperature at the inlet to the compressor, pressure ratio, the exit gas temperature, and the amount of air flow over the compressor. The outputs of the program include the temperatures at various location and heat transfer between components.

### 3.5. Verification and Discussion

To verify the theoretical prediction a compressor is modified to attach 24 thermocouples at various locations. The experiment was carried out on a calorimeter and various temperatures are measured [Figure 3]. The experimental data is compared against theoretical predictions and a good agreement is found. Table 1 describes the theoretical as well as experimental temperatures (in degree Fahrenheit) at various locations.

[Figure 3. Experimental setup with airflow over compressor using a fan]

It is found that the case temperature above the stator/rotor differs that of experimental measurement by approximately 8 degrees. This is attributed to the fact that the heat transfer due to radiation is calculated assuming a temperature of the case which is ten degrees higher than that of the exit temperature. In calculating the oil sump temperature, a more accurate determination of velocity profile of the lubricant oil can be done using computational fluid dynamics and at present this is not done. Otherwise a good agreement between the thoery and experiment is found.

| Element Number | Element Name | Theoretical temperature | Experimental temperature |
|---|---|---|---|
| 1 | Case above the stator top | 176.21 | 184.71 |
| 2 | Case attached to the stator | 197.56 | 192.82 |
| 3 | Case between stator and cylinder | 196.03 | 192.03 |
| 4 | Case attached to the cylinder | 191.98 | 191.41 |
| 5 | Case portion attched to the oil sump | 187.08 | 188.84 |
| 6 | Cylinder | 200.45 | 198.25 |
| 7 | Oil sump | 192.00 | 197.64 |
| 8 | Suction gas in the cylinder | 125.38 | 123.11 |
| 9 | Compressed gas in cylinder | 237.51 | 231.89 |
| 10 | Vane | 202.31 | Not measured |
| 11 | Gas above cylinder | 200.44 | 203.44 |
| 12 | Stator copper windings | 200.03 | 198.33 |
| 13 | Stator iron | 198.07 | 195.28 |
| 14 | Rotor | 200.88 | Not measured |
| 15 | Shaft above upper bearing | 200.89 | Not measured |
| 16 | Gas at rotor gap outlet | 200.46 | 195.28 |
| 17 | Gas at channel outlet | 199.9 | 198.55 |
| 18 | Roller | 203.04 | Not measured |
| 19 | Lower bearing plate | 201.72 | 202.8 |
| 20 | Upper bearing plate | 201.66 | 205.21 |
| 21 | Shaft in the bearings | 201.74 | Not measured |
| 22 | Gas above the stator/rotor | 199.68 | 196.48 |

[Table 1. Comparision of theoretical and experimental results]

## 4. CONCLUSION

A lumped mass model for the rotary compressor is developed. System equations are developed using the thermal conductance concept. Heat transfer coefficients are calculated from the empirical/theoretical equations adapted from various sources. No empirical correction factors are introduced into the system equations. Mechanical as well as electrical losses are considered as heat addition to the system. A computer code is generated to predict the temperature of the compressor at various locations and a good agreement is found with the experimental data.


## ACKNOWLEDGMENT

The author is indebted to Mr. J. J. O'Neill, Manager-Compressor & Refrigerant Systems Design, for his many valuable discussions, help and encouragement; without which the publication of this paper might not have been possible. The author wish to thank his academic advisor Prof. S. N. Dwivedi at West Virginia University for his technical discussions. The author appreciates the help of Dr. Robert Bergholz at GE-Air Craft Engines in explaining his prior work. Thanks to our technicians Mr. Robert Smith and Mr. James O'Dell for their excellent work on the experiments. Special thanks to Mr. R. Mark Schreck, General Manager-Refrigeration and Dr. William J. Sheeran, Vice President-Technology for their encouragement and for allowing the paper to be published.

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