HEAT TRANSFER IN A ROTATING DISK SYSTEM WITH RADIAL COOLANT PASSAGES

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Abstract

Experiments are carried out to investigate the heat transfer characteristics in a scale-model of a large rotating electrical machine. The model consists of four equi-spaced stator and rotor disks which simulate the stator and rotor packets of an induction motor. Axial heating elements are embedded in the circumferential slots located in the disks to simulate the heat generation in the machine. Disk temperature distributions in the radial, axial and circumferential directions, in addition to the temperature of air flowing through the ducts, are obtained. Heat transfer co-efficients are calculated for both stationary and rotating radial ducts for several values of heat generation in the system.

Introduction

The stator and rotor of a rotating electrical machine are made up of laminations (also called stampings) having circumferential slots and held together under pressure. Copper conductors, housed in these slots, extend axially from one end to the other end of the rotor and statot. Spacers are provided to separate one bunch of laminations from the other, thereby creating a diverging radial passage through which the cooling medium is forced to carry away the heat produced in the machine, while in operation. The bunch of laminations between two such radial passages is called a packet.

Due to the complex configuration of the machine and high rotational speed of the rotor, direct measurement of the temperature rise of the windings is often difficult. But the heat losses and the consequent temperature rise of the machine components are natural phenomena and simply can not be avoided.

A prior knowledge of the possible temperature rise is very essential to the designer to determine the class and type of insulations to be used for the windings of the machine. Direct measurements, as already mentioned, being difficult, indirect methods of assessing the temperature rise are resorted to. Oe of these is to build an electrical network with current sources, using lumped parameter approach and solving it by convenient methods. Evaluation of heat transfer coefficients in the radial cooling channels thus assumes great importance as they strongly define the convective resistances of the network.

Previous Work

Harada [1] made numerical investigations on the flow between two rotating disks using von Karman's similarity hypothesis. The analysis did not take heat transfer into account. Mochizuki et al [2] analysed the heat transfer mechanisms and performance in stationary multiple parallel disk assemblies and concluded that the disk device had performance characteristics comparable to high performance, plate-fin, compact surface. Sim and Yang [3] studied the heat transfer in laminar flow through co-rotating parallel disks numerically. A theoretical model was developed to determine the heat transfer performance through a pair of co-rotating parallel disks. Some other [4,5] studies heat transfer from a disk with or without enclosure.

As in seen, the literature reports studies about disks assemblies, which may be assumed to simulate either the stator or the rotor of an electrical machine, depending on whether the system is stationary or rotating. However, from heat transfer point of view, a realistic simulation of the packets and windings of an electrical machine is a system of co-rotating and stationary parallel concentric disks with embedded heating elements.

This paper aims at reporting the work done on such a set up and presenting results relevant to the rotor only.

Experimental Set up

A sectional view of the experimental set up is shown in Fig.1. The details of the rotor assembly are shown in Fig.2.

The disks are made of mild steel and measure 293 mm X 167.5 mm X 20 mm. To facilitate insertion of heaters, slots are made circumferentially in each disk (Fig.3 and 4). The stator disks measure 430 mm X 295 mm x 20 mm. Air gap between the stator and the rotor is 1 mm. Mild steel rings, 5 mm thick, are used as spacers between the disks and are inserted over the tie-rods holding the disks. A continously diverging radial duct of 5 mm width is thus formed between every two disks. Air enters the test section through the annulus formed between the rotor shaft and the rotor disks. The flow path of air is shown in Fig.5 of the three radial ducts formed, the central is treated as the test duct. Axial flow of air in the rotor-stator air gap is neglected. Rectangular box-shaped heaters, with nichrome wire as the heating element inside, are inserted into the slots of the rotor and stator disks. The heaters cover the full axial length of the disks and the ducts and thus, simulate the conductors of an actual electrical machine. The heat generation in the disks is varied from 300 W to 1000 W. Power supply to the rotor heaters are given through a slipring, mounted on the end shaft, with carbon brush contact.

Double insulated copper-constantan thermocouples are used to measure the disk and air temperatures. The locations of thermocouples are shown Figs. 6 and 7. For the measurement of duct air temperatures, thermocouples are brought out through the hollow rotor shaft and connected to a slipiring assembly via a junction box. The stator thermocouples are taken out radially. All the thermocouples are finally connected to the digital millivoltmeter through selector switch.

The air required for the test is obtained from a centrifugal blower. Its flow rate is monitored by a by-pass and a damper provided in the blower ducting and measured with an inclined-tube manometer. The air is uniformly admitted all over the annulus between the rotor disks and the rotor shaft through the circular opoenings in the stator and rotor end flanges. It then branches out in the radial ducts of the set up and is let off to the atmosphere.

The rotor is driven by a variable speed motor through a pulley and belt drive. Stepped pulleys are used to get different speeds. A rectifier unit controls the speed of the D.C. motor.

Results and Discussions

Experimental are carried out for both flow and no flow conditions. The latter corresponds to a situation where Re=0, or, the rotor is self-ventilated. The flow Reynolds number is defined as:

$$Re = GD_H / \mu \qquad (1)$$

Rotor speed is varied from 0 to 700 rpm. This corresponds to a Taylor number range of 0-1-2, where Ta is defined as:

$$Ta = B^2 \Omega / \nu \qquad (2)$$

The heat transfer coefficients are defined as follows:

(i) when Re=0,

$$h = Q / (A \Delta T) \qquad (3)$$

where,

$$\Delta T = T_w - T_\infty \qquad (4)$$

(ii) when Re>0,

$$h = \frac{Q \ln \frac{(T_{w,2} - T_{f,2})}{(T_{w,1} - T_{f,1})}}{A[(T_{w,2} - T_{f,2}) - (T_{w,1} - T_{f,1})]} \qquad (5)$$

The Nusselt number isdefined as:

$$Nu = h D_H / k \qquad (6)$$

Fig.8 shows a set of curves drawn for Nu Vs. Ta for different heat inputs under self-ventilating conditions. The increase in Nusselt number is well marked with increase in Taylor number. For the range of Ta investigated, the increase is about three times as compared with stationary condition. Further, it is seen that with increase in heat loss, the Nusselt number has a decreasing tendency at higher Ta, thus indicating that the buoyanty effects are not predominant. However, irrespective of the heating conditions, Ta has a marked effect on Nu.

The variation of Nu with Ta for pure rotation (Re=0) and with radial flow superimposed is shown in Fig.9. Graphs are plotted for Re=0, 1825 and 3850, covering conditions below and just above transition flow. The cure at each higher Re is steeper than the previous one, establishing combined effect of rotation and flow on heat transfer augmentation.

The variation of temperature of a typical node for flow and no-flow situations and with increasing rotation is given in Fig.10. Temperatures have been measured at sixteen other locations and in each case, a similar trend has been observed. It is seen that the graph for temperature drop with rotation for no-flow condition is much steeper than the corresponding graphs with different flow situations. This, of course, is to be expected. With the introduction of flow, the wall temperature itself is low and the temperature potential is much smaller than that for the no-flow codition.

The axial temperature distribution at a particular radius is shown in Fig.11. All the graphs show a more or less uniform axial temperature distribution. The variation of tangential temperature is not considerable due to symmetry. The radial temperature distribution is shown in Fig. 12. The gradient is in the flow direction and temperatures near the heat generation points are higher. This is because these points are in contact with relatively hot air and with reduced velocity of air due to the diverging radial duct.

Fig. 13 shows a typical three-dimensional temperature distribution for the rotor disks.

Nomenclature

A - heat transfer area, $m^2, = \frac{\pi N}{4}(D_2^2 - D_1^2) + \pi NL(D_2 + D_1)$

A_c - area of flow of air, $m^2, = \pi \times D_m \times L$

B - spacing between disks, m

D - disk diameter, m; D_1, inner; D_2, outer; D_m, mean

D_H - hydraulic diameter, m, = 2B

G - mass velocity, kg/(m^2.s)=, m/A_c

h - heat transfer coeffocient, W/m^2 ° K

k - thermal conductivity of air, W/m°K

L - disk thickness, m

m - mass flow rate of air, kg/s..

N - numer of disks

Nu - Nusselt number

n - rotational speed of rotor, rps

Q - heat input, W

Re - Reynolds number

T - temperature, °C

Ta - Taylor number

Ω - angular velocity of rotor, rad/s., = $2\pi n$

v - kinematic viscosity of air, m^2/s

μ - absolute viscosity of air, kg/(m.s)

Subscripts

f - fluid

w - wall

∞ - ambient

1 - inlet

2 - outlet

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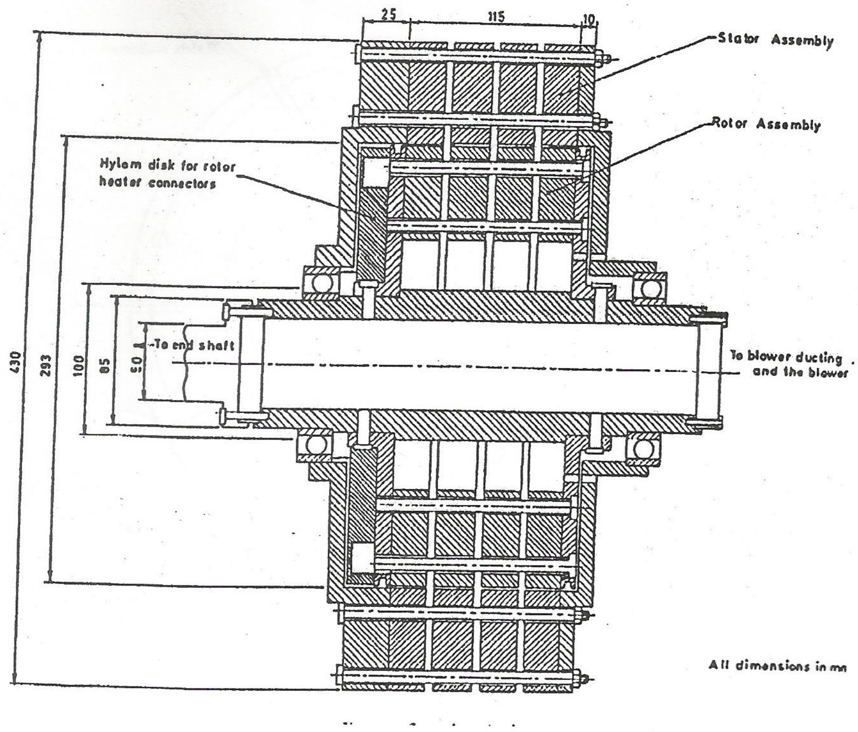

FIG.1 EXPERIMENTAL SET-UP

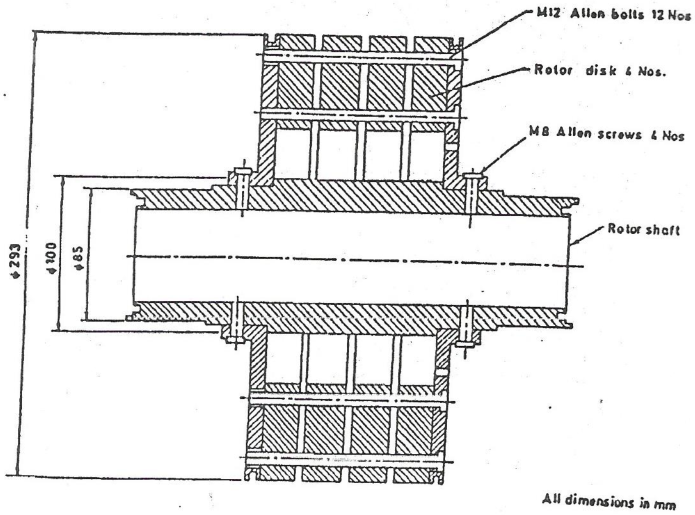

FIG.2 ROTOR ASSEMBLY

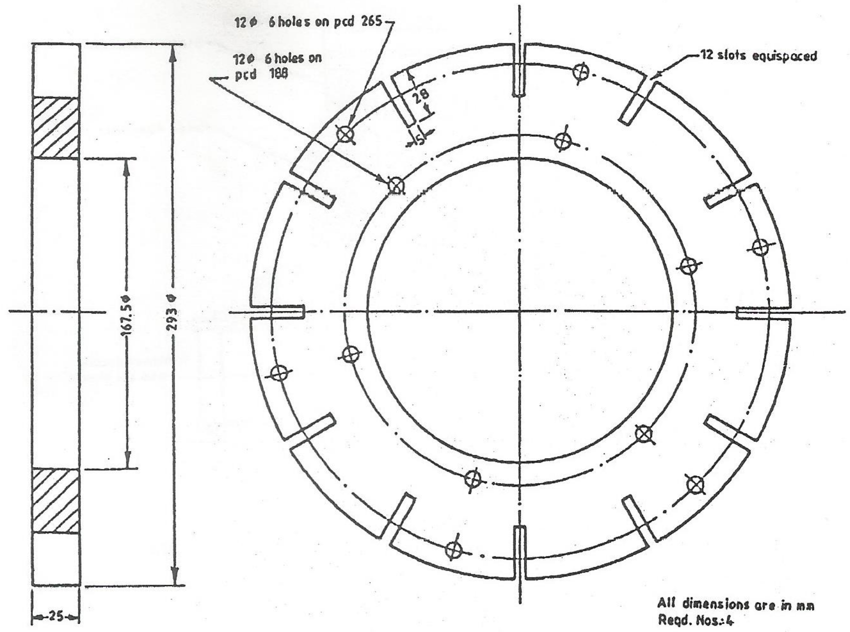

FIG.3 ROTOR DISKS

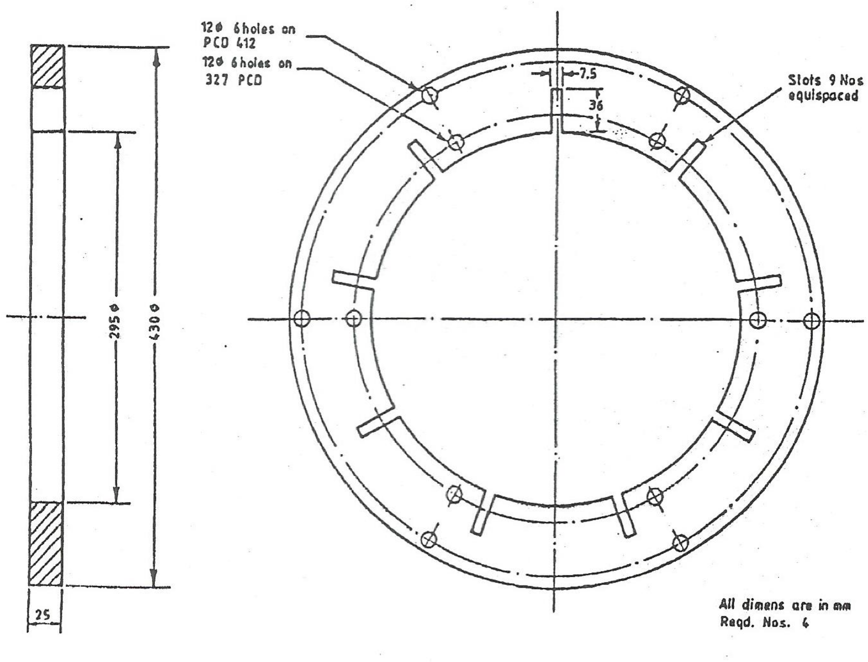

FIG.4 STATOR DISK

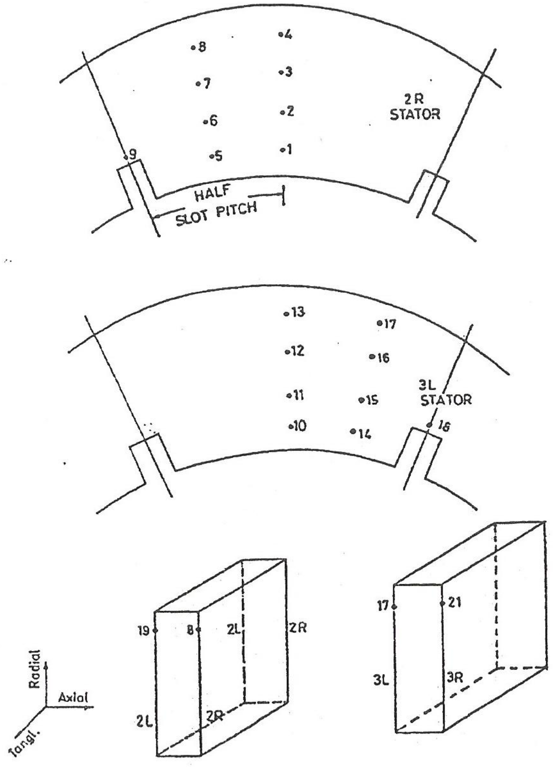

Fig.6. Location of thermocouples in stator disks

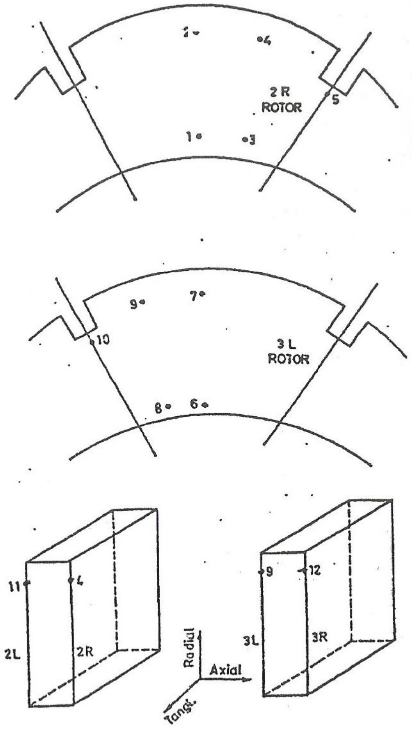

Fig.7. Location of thermocouples in rotor disks

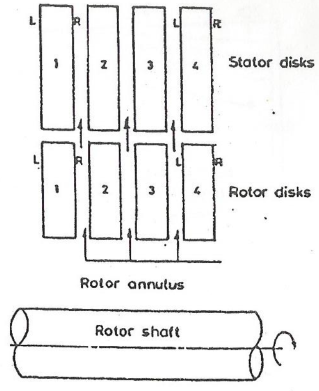

Fig.5. Numbering of stator & rotor disks for thermocouple locations
Arrows indicate the air flow path

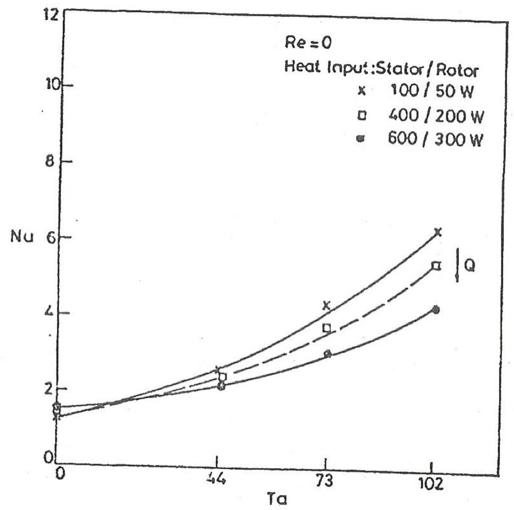

FIG.8. Nu Vs Ta for rotor radial duct for various heat inputs.

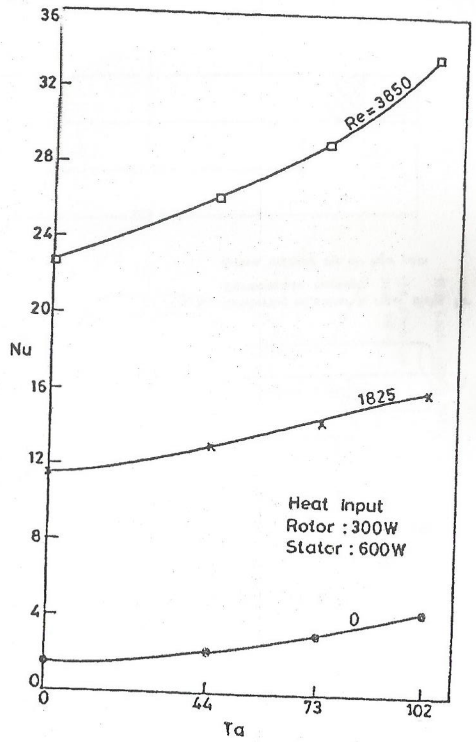

FIG.9. Nu Vs Ta for rotor radial duct for different Re

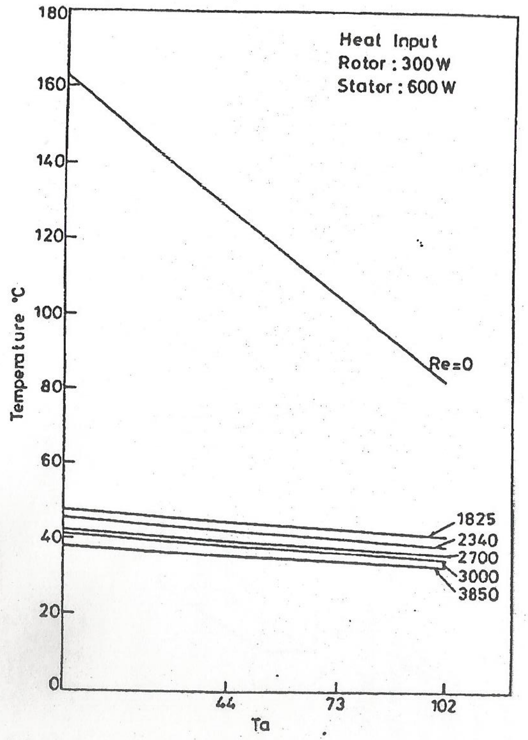

FIG.10. Temperature variation of rotor node with Ta for different Re

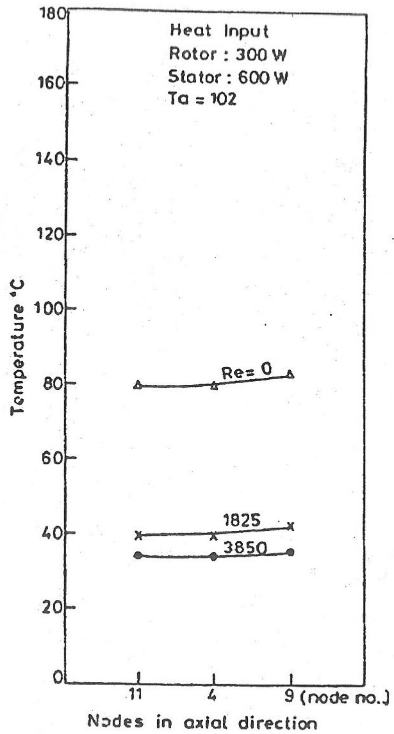

FIG.11. Axial temperature distribution in rotor for different Re

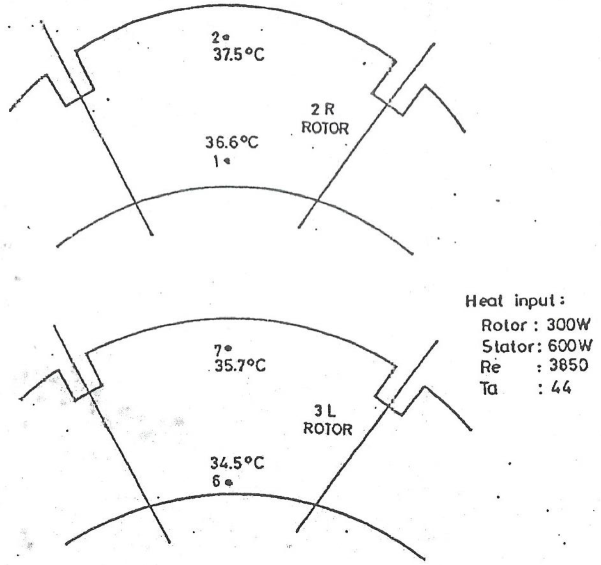

FIG.12 Radial temperature distribution in rotor

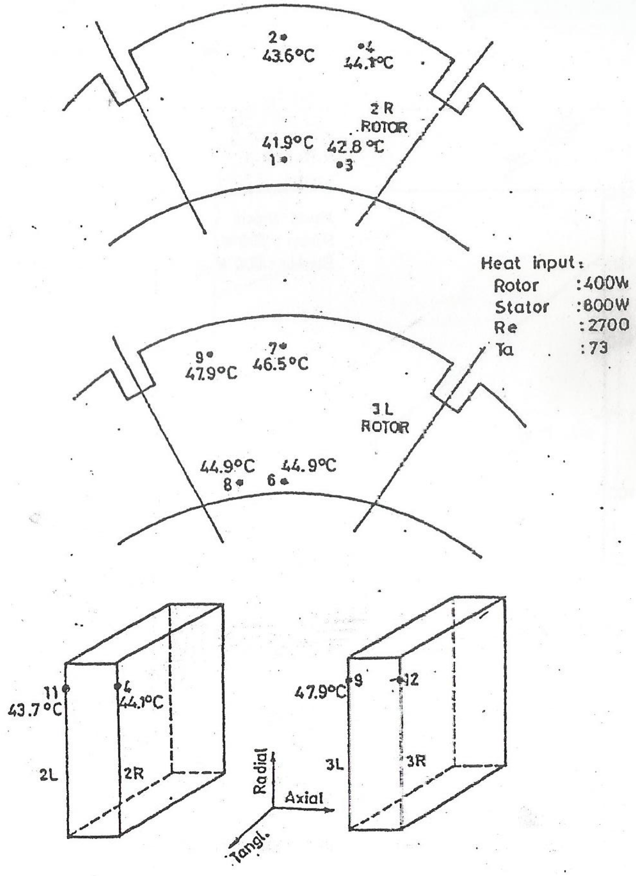

FIG. 13. 3-D Temperature distribution in rotor disks