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Transient free convective heat transfer from co-rotating concentric disks

BISWAJIT BANERJEE, K. V. CHALAPATHI RAO and V. M. K. SASTRI
Department of Mechanical Engineering, Indian Institute of Technology, Madras 600 036, India

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Abstract—Experimental investigations are carried out for the determination of transient heat transfer coefficients between parallel co-rotating and concentrically stationary disks with internal heat generation. The experimental model simulates a typical induction motor with unequal losses in the stator and rotor at the start-up condition. The data collected cover a range of Taylor numbers for various heat inputs. Quantitative assessment is made for the increase in heat transfer with speed of rotation under free convective ambient conditions. Temperatures along the axial, radial and tangential directions are measured. Unsteady heat transfer coefficients to the ambient air are evaluated in the radially diverging section. The results of the corresponding R-C network are obtained on a digital computer. The predicted values of temperatures at the corresponding nodal points are compared with the measured values and found to be in good agreement. The results are potentially very useful in the thermal design of electrical machines, more specifically, radially ventilated induction motors.

INTRODUCTION

PREDICTION of the temperature distribution in an electrical machine, both for steady and unsteady state conditions, is a subject matter of great interest to researchers and engineers. It is essential to know the magnitudes of the highest temperatures and their locations and deviations from the average value because of their bearing on the design of the machines. The problem assumes greater complexity if one has to predict these values under transient conditions. The temperature distribution also depends upon the variation in type and location of the heat source, the ventilation system and the transient nature of rotor thermal loading.

The electrical analogue approach is one of the most commonly used methods for the prediction of temperature distribution for both steady and unsteady state conditions.

The need for transient analysis arises due to several electrical design factors such as reactive overload capacity, negative phase sequence, etc.

Several authors [1–3] reported work on heat transfer from disks with or without enclosure and for the case of parallel disks [4] without rotation. Mochizuki and Yang [5] reported work with co-rotating parallel disks but with steam as the heating fluid. To the authors' knowledge, no single paper has so far been reported for the determination of the heat transfer coefficient for the case of co-rotating parallel disks together with stationary concentric parallel disks, a configuration that closely resembles the rotor–stator of an electrical machine, with forced or free convection taking place between the disks. The present work aims at determining the heat transfer coefficients under free convective conditions.

SCOPE OF THE PRESENT WORK

A model of a stator and rotor of a typical induction motor with radial duct cooling is simulated. Typical loss distribution in the stator and rotor will be considered and simulated by heat sources at suitable locations. Even though there is forced flow of air, the present analysis is limited to the no-flow condition, i.e. free convection is predominant, with rotation of the rotor superimposed. This may simulate the condition of a fan failure wherein heat loads are to be met by self-ventilation. The heat transfer coefficients, obtained experimentally for different Taylor numbers and different heat losses in the stator and rotor, are used to calculate the convective resistances for the analytical model. The analytical model is solved for the transient and steady state temperature distributions for comparison with the measured values of temperatures.

DESCRIPTION OF EXPERIMENTAL SET-UP

The experimental set-up simulated the rotor and stator of a typical induction motor. It consisted of essentially four pairs of disks forming three radial ducts. Attention was paid to the central duct whereas the remaining two ducts on either side took care of the end effects. A sectional view of the test section is shown in Fig. 1. For simplicity the laminations of the motor are idealized to be a simple disk of solid steel. The disks were held in position by means of tie rods and end flanges which were suitably insulated to minimize axial conduction. The rotor was fixed to two end flanges with a hollow shaft through which thermocouples and power leads were taken out. To simulate the heat generation due to copper losses in

NOMENCLATURE

- A heat transfer area, $(2\pi N/4)(D_2^2 - D_1^2) + \pi NL(D_2 + D_1)$ [m^2]
- B spacing between disks [m]
- D disk diameter: D_1, inner; D_2, outer; D_m, mean
- D_H hydraulic diameter, $2B$ [m]
- h average heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
- L axial dimension of disks [m]
- N number of disks
- n rotational speed of rotor [rps]
- Q heat input [W]
- T temperature [°C]
- Ta Taylor number, $B^2\Omega/\nu$.

Greek symbols

- ν kinematic viscosity of air [$m^2\,s^{-1}$]
- Ω angular velocity of rotor, $2n$ rad s^{-1}.

Subscripts

- ∞ ambient
- av average
- R rotor
- S stator.

the conductor, rectangular heater elements each of 150 W capacity, were inserted into the slots made in the periphery of the rotor and stator disks. The stator assembly was similar and concentric with the rotor. After assembly the rotor–stator air gap was about 1 mm. Both rotor and stator surfaces were chrome-plated to prevent rust formation and maintain a clean surface. A power slip ring with brushes was used for power supply to the heating elements. Precautions were taken to prevent slipping and short circuiting of heating elements even at high rotational speeds. A variable speed drive was connected to the rotor shaft through a pulley mechanism for varying the speeds.

A number of copper–constantan thermocouples were fixed at different locations on the surface of the rotor and stator disks (complete details are given in ref. [6]). The experiments were carried out for different heat inputs and at various Taylor numbers. Temperatures in the axial, tangential and radial directions were measured to an accuracy of 0.1°C.

EXPERIMENTAL RESULTS AND DISCUSSION

The heat transfer performance for free convection through the radial ducts of the combined rotor–stator system with internal heat generation is expressed in terms of the average heat transfer coefficient h. Experiments were performed for rotor speeds of 0–700 rpm, which correspond to a Taylor number range of 0–102, where the Taylor number is defined as

$$Ta = B^2\Omega/\nu.$$

The air-side heat transfer coefficient is defined as

$$h = Q/A\Delta T$$

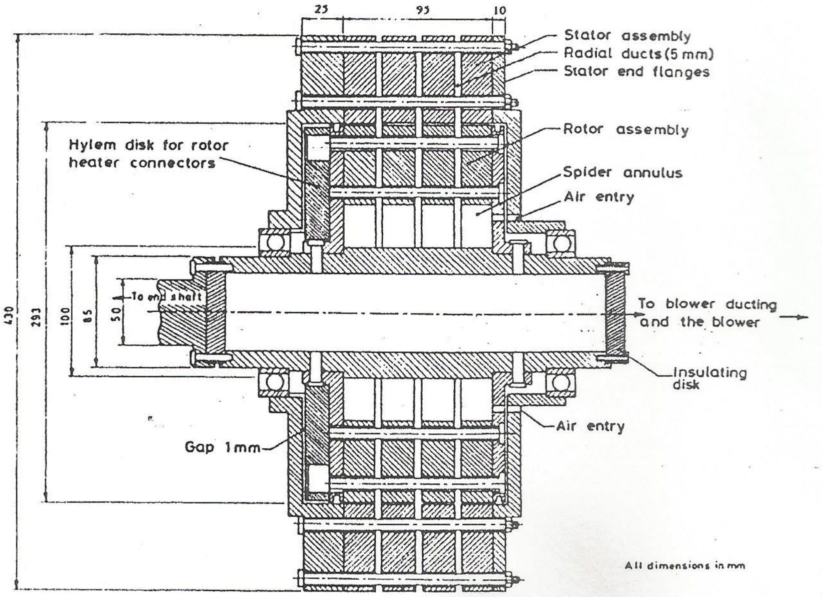

FIG. 1. Sectional view of the test section.

INSTATIONÄRE FREIE KONVEKTION AN GLEICHSINNIG ROTIERENDEN KONZENTRISCHEN SCHEIBEN

Zusammenfassung—Die instationären Wärmeübergangskoeffizienten zwischen zwei parallelen, gleichsinnig rotierenden konzentrischen Scheiben mit innerer Wärmefreisetzung werden experimentell bestimmt. In dem Versuchsmodell wird ein typischer Induktionsmotor nachgebildet mit unterschiedlichen Verlusten im Stator und Rotor beim Anlaufen. Die Versuchsdaten decken bei unterschiedlicher Wärmezufuhr einen Bereich von Taylor-Zahlen ab. Das Anwachsen des Wärmeübergangs mit der Drehzahl unter den Bedingungen der Freien Konvektion wird quantitativ berücksichtigt. Die Temperaturverteilung wird in axialer, radialer und tangentialer Richtung gemessen. In dem sich in radialer Richtung erweiternden Abschnitt werden instationäre Wärmeübergangskoeffizienten zur umgebenden Luft hin berechnet. Das gesamte Problem wird mit Hilfe eines Widerstandskapazitätennetzwerks auf einem digitalen Rechner nachgebildet. Ein Vergleich von Messung und Berechnung zeigt gute Übereinstimmung. Die Ergebnisse sind möglicherweise bei der thermischen Auslegung elektrischer Maschinen sehr nützlich, insbesondere bei radial belüfteten Induktionsmotoren.

НЕСТАЦИОНАРНЫЙ СВОБОДНОКОНВЕКТИВНЫЙ ТЕПЛОПЕРЕНОС В СИСТЕМЕ ИЗ ВРАЩАЮЩИХСЯ В ОДНОМ НАПРАВЛЕНИИ КОНЦЕНТРИЧЕСКИХ ДИСКОВ

Аннотация—Проведены экспериментальные исследования по определению коэффициентов нестационарного теплопереноса в системе параллельных вращающихся в одном направлении и концентрически неподвижных дисков с внутренним тепловыделением. Такая экспериментальная ситуация моделирует типичный асинхронный двигатель с неравными потерями в статоре и роторе при запуске. Получены данные в диапазоне чисел Тэйлора, соответствующем различным тепловым нагрузкам. Определена качественная зависимость интенсивности теплопереноса от скорости вращения дисков при наличии свободной конвекции в окружающей среде. Получены температурные распределения в аксиальном, радиальном и тангенциальном направлениях. Рассчитаны коэффициенты нестационарного теплопереноса в окружающий воздух в радиально расходящемся сечении. С помощью цифровой вычислительной машины получены результаты для соответствующей РС-цепочки. Найдено, что расчетные значения температуры в узловых точках хорошо согласуются с экспериментальными данными. Полученные результаты могут быть использованы при тепловом расчете электрических машин, точнее, радиально вентилируемых асинхронных двигателей.

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., = $2\pi n$

ν - 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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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,

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

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

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HEAT TRANSFER IN A ROTATING DISK SYSTEM WITH RADIAL COOLANT PASSAGES

BISWAJIT BANERJEE, K.V. CHALAPATHI RAO and V.M.K. SASTRI

Department of mechanical engineering
Indian Institute of Technology
MADRA 600 036.

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-

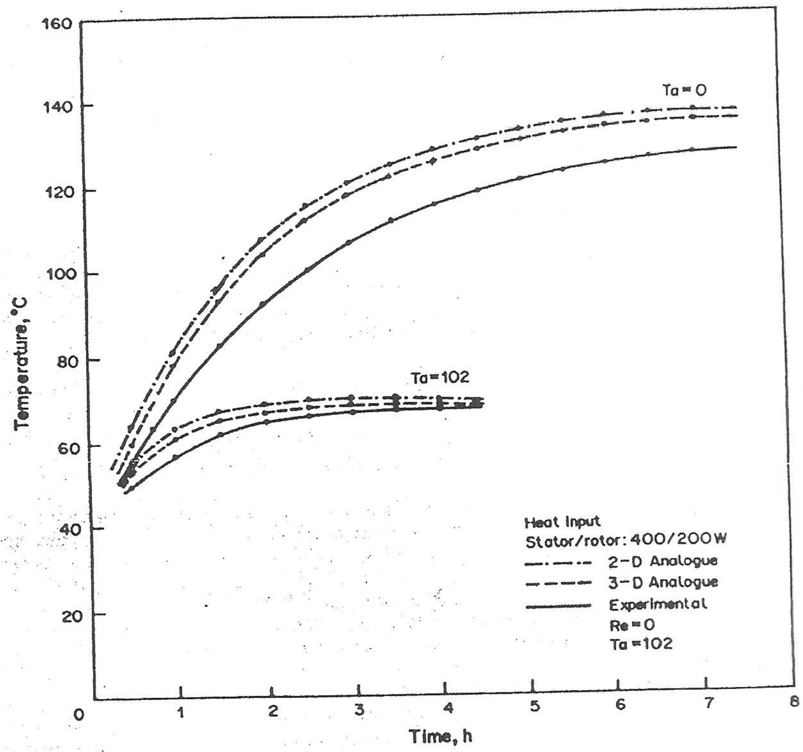

FIG. 7. Comparison of temperature variation of a typical stator node for different *Ta*.

CONCLUSIONS

(1) The unsteady state heat transfer coefficients are significantly different from the steady state values. This fact can be used for better thermal design of the rotating electrical machines.

(2) The simple network analogue method can be effectively used to estimate the transient temperature distribution in rotating systems.

(3) Neither Taylor number nor heat flux appears to have any significant effect on the axial temperature distribution of the disks.

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TRANSFERT THERMIQUE VARIABLE PAR CONVECTION NATURELLE POUR DES DISQUES CONCENTRIQUES CO-ROTATIFS

Résumé—On conduit des études expérimentales pour la détermination des coefficients variables de transfert thermique entre des disques parallèles co-rotatifs avec génération thermique interne. Le modèle expérimental simule un moteur à induction typique avec des pertes différentes dans le stator et le rotor au départ. Les résultats collectés couvrent un domaine de nombres de Taylor pour différentes conditions thermiques. On mesure les températures dans les directions axiale, radiale et tangentielle. Des coefficients variables de transfert thermique vers l'air ambiant on été évalués dans la section radiale. Les résultats du réseau R-C correspondant sont obtenus sur un ordinateur. Les valeurs prédites de température aux points nodaux correspondants sont comparées aux valeurs mesurées et on trouve un bon accord. Les résultats sont utilisables dans la conception des machines électriques, plus spécifiquement dans les moteurs à induction radialement ventilés.

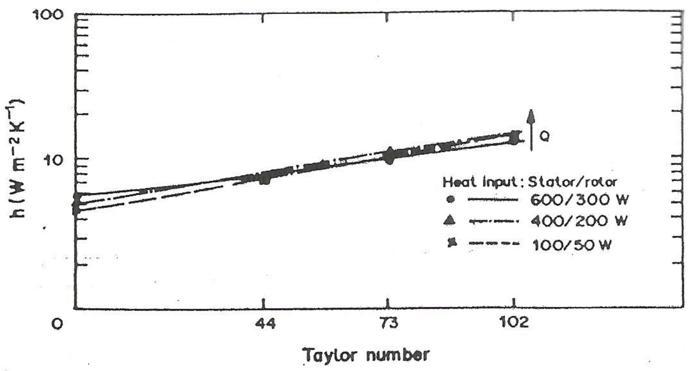

FIG. 4. Variation of HTC in stator radial duct.

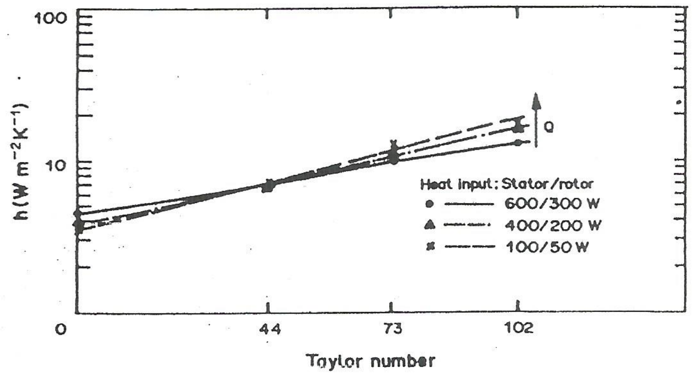

FIG. 5. Variation of HTC in rotor radial duct.

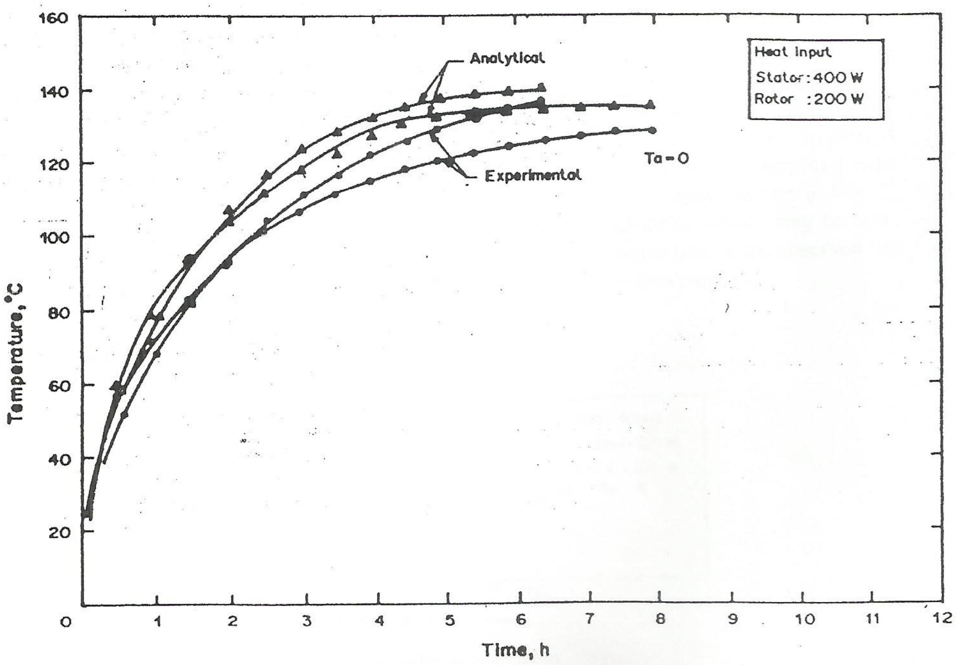

FIG. 6. Variation of temperature of a typical stator and rotor node with time.

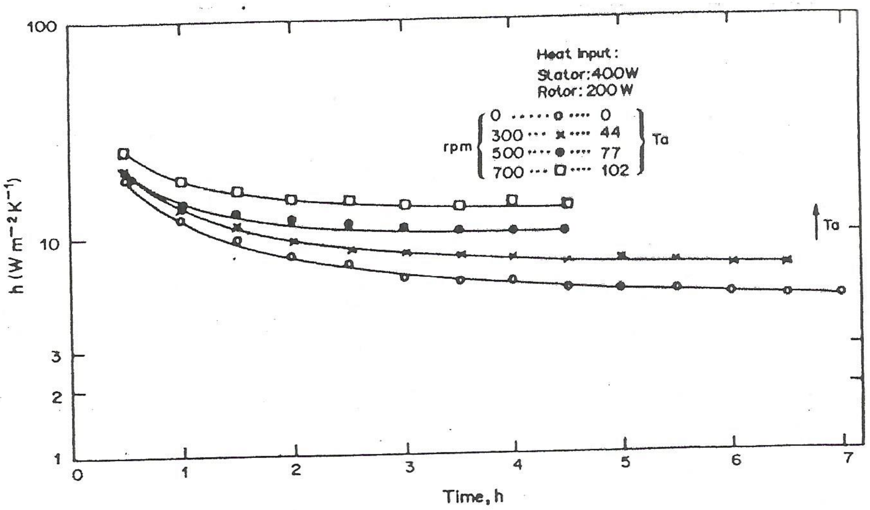

FIG. 2. Variation of HTC for stator radial duct with time.

where ΔT is the difference of average wall temperature and ambient temperature.

The variation of average transient heat transfer coefficient in the stator and rotor radial ducts is shown in Fig. 2 for different Taylor numbers. A similar trend was observed at various heat inputs. The graphs show a definite trend of increase in 'h' with increasing Ta. However, at each Ta, the heat transfer coefficient approaches a steady state value after a time period.

Transient temperatures were recorded every 30 min for each heat input and rotational speed until steady state was attained. However, it was noticed that the system took a considerable time to attain steady state due to large thermal inertia.

The heat transfer coefficient values are high initially and subsequently attain asymptotic values. The influence of rotation on heat transfer augmentation is very much evident and this is shown in terms of h_r/h_0 in Fig. 3.

Figures 4 and 5 show the absolute values of the heat transfer coefficients in both the stator and rotor radial ducts for different heating conditions. The values do not seem to depend much on the heat flux but are strong functions of Taylor number only.

A three-dimensional resistance–capacitance network was formulated for the two pairs of stator and rotor disks forming the central duct. The calculation of conductive resistances and capacitances poses no problem as they are dependent mainly on the thermophysical properties of the disk material. However, it is the convective resistance which relies heavily on the values of the heat transfer coefficients. The values of heat transfer coefficients obtained experimentally are made use of to calculate the convective resistances from the disks to the radial ducts and around all other convective zones. Assuming uniform heat flux, current injections to the nodes close to the heating elements were incorporated consistent with the heating of each lump representing a node. The network was solved on an IBM 370 computer using the software package SPICE with all the convective nodes grounded [7].

The values of temperature rise obtained analytically for a typical stator and a rotor node, when plotted against time, are shown in Fig. 6 and compared with the experimentally measured values shown in Fig. 7. The variation is about 10–20%, which may be considered as satisfactory. Similar trends are observed for other Taylor numbers and heat inputs.

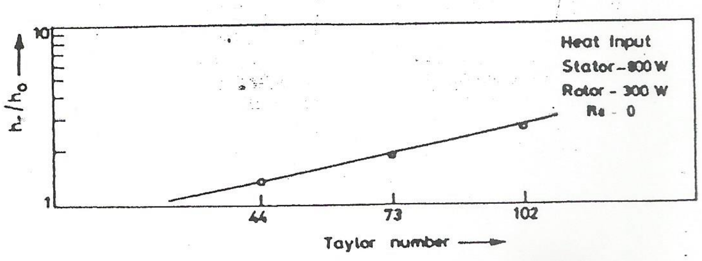

FIG. 3. Increase in heat transfer with rotation.

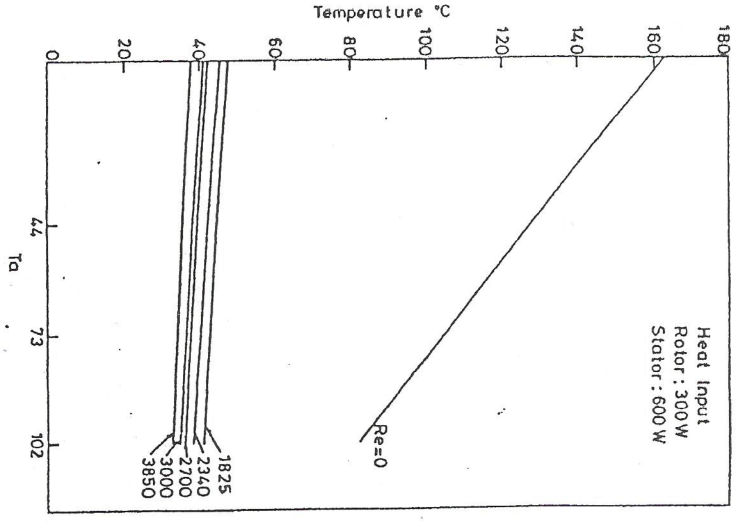

FIG.10. Temperature variation of rotor node with Ta for different Re

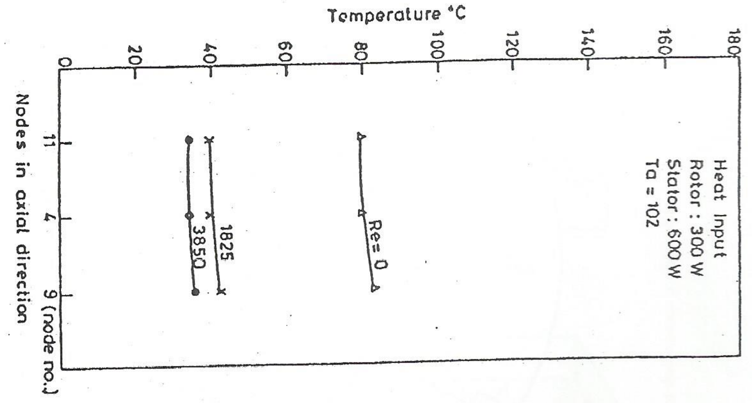

FIG.11. Axial temperature distribution in rotor for different Re

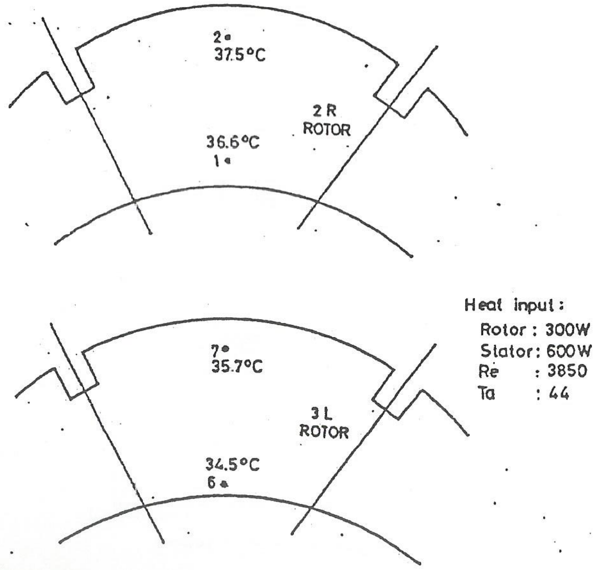

FIG. 12. Radial temperature distribution in rotor

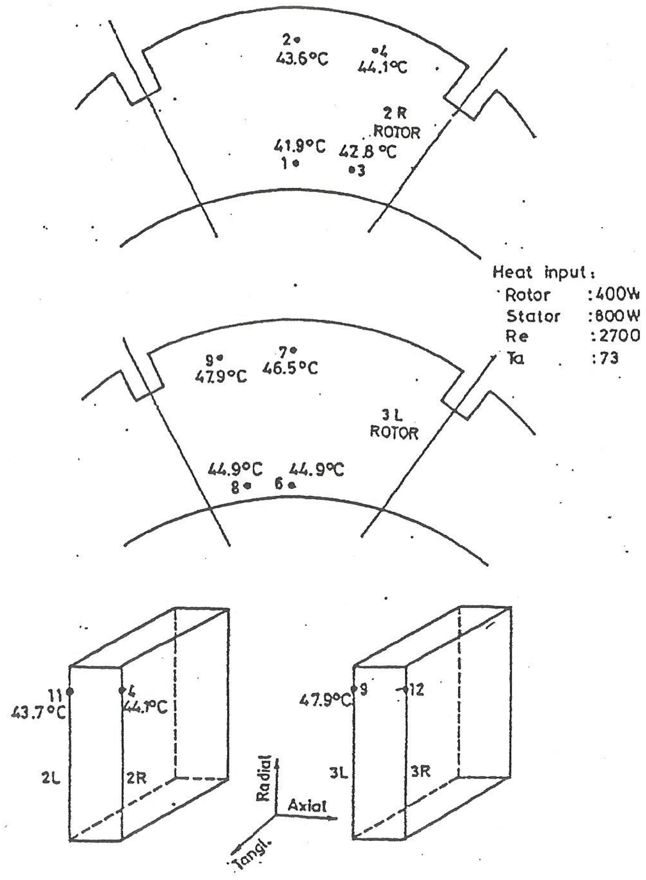

FIG. 13. 3-D Temperature distribution in rotor disks

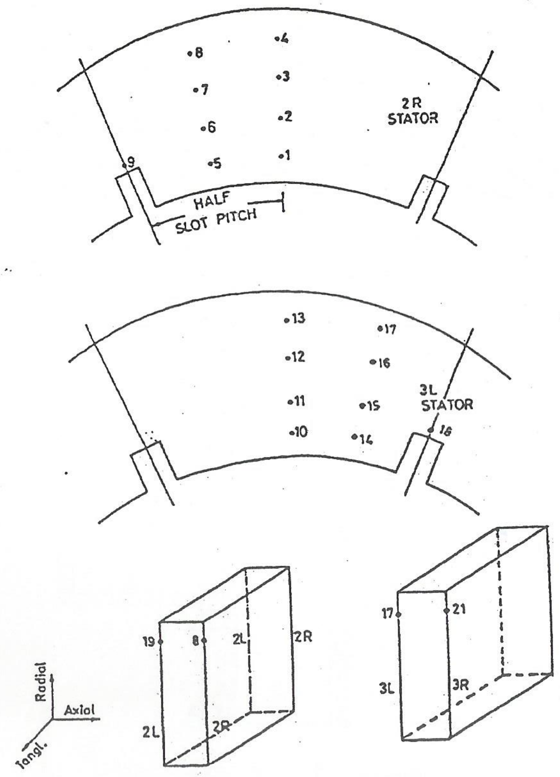

Fig.6. Location of thermocouples in stator disks

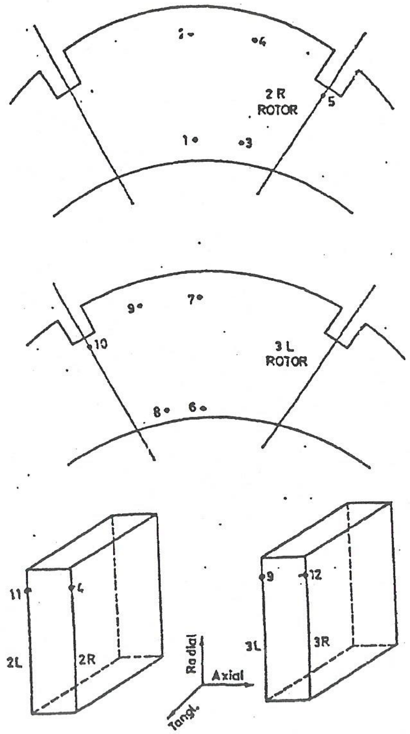

Fig.7. Location of thermocouples in rotor disks

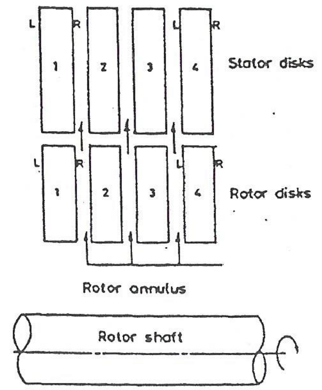

Fig.5. Numbering of stator & rotor disks for thermocouple locations
Arrows indicate the air flow path

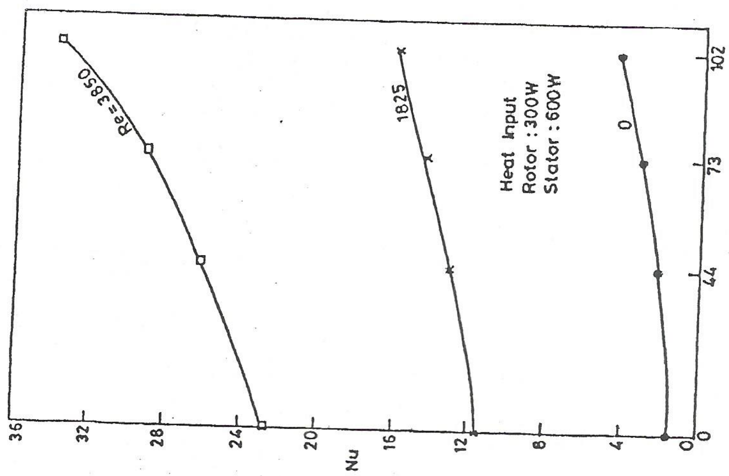

FIG.9. Nu Vs Ta for rotor radial duct for different Re

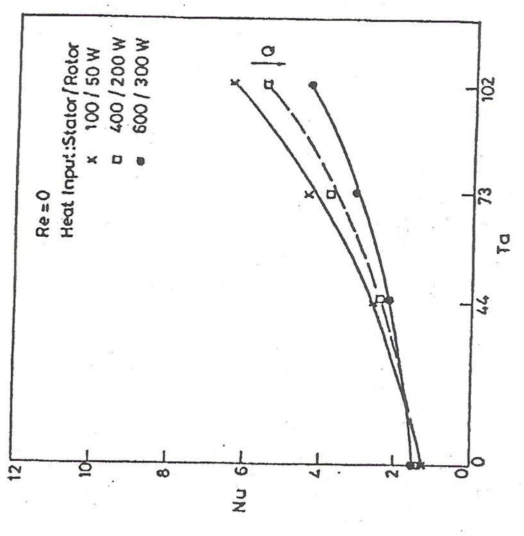

FIG.8. Nu Vs Ta for rotor radial duct for various heat inputs.

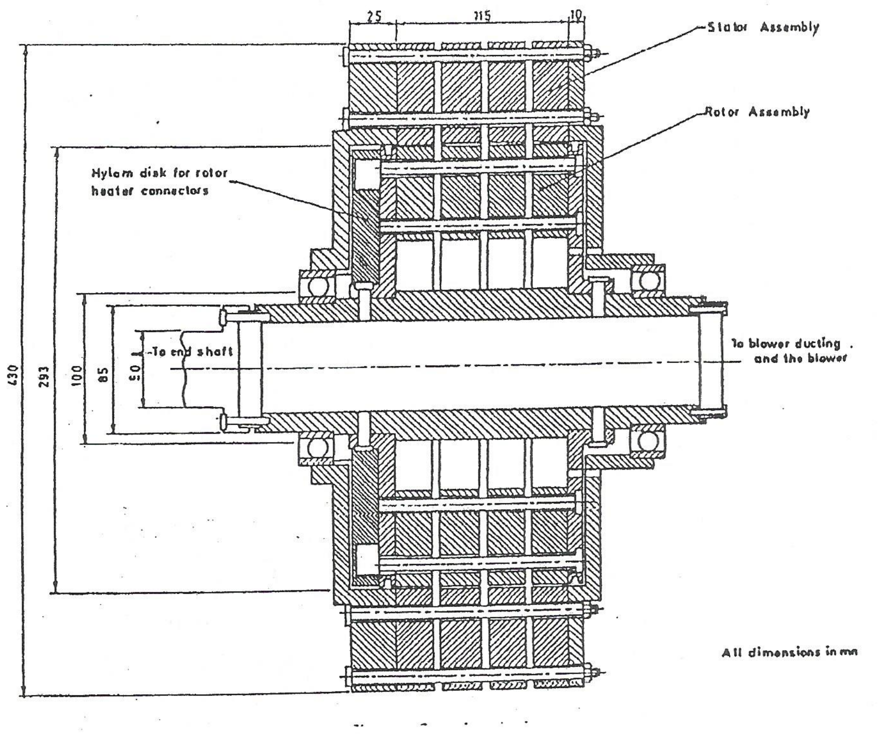

FIG.1 EXPERIMENTAL SET-UP

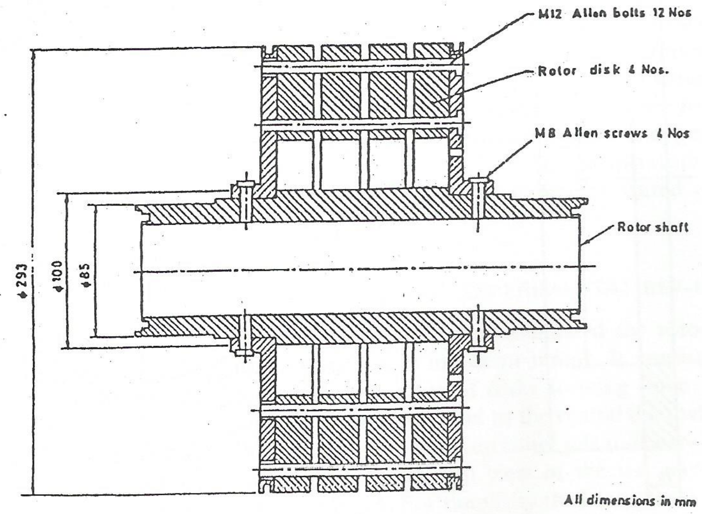

FIG.2 ROTOR ASSEMBLY

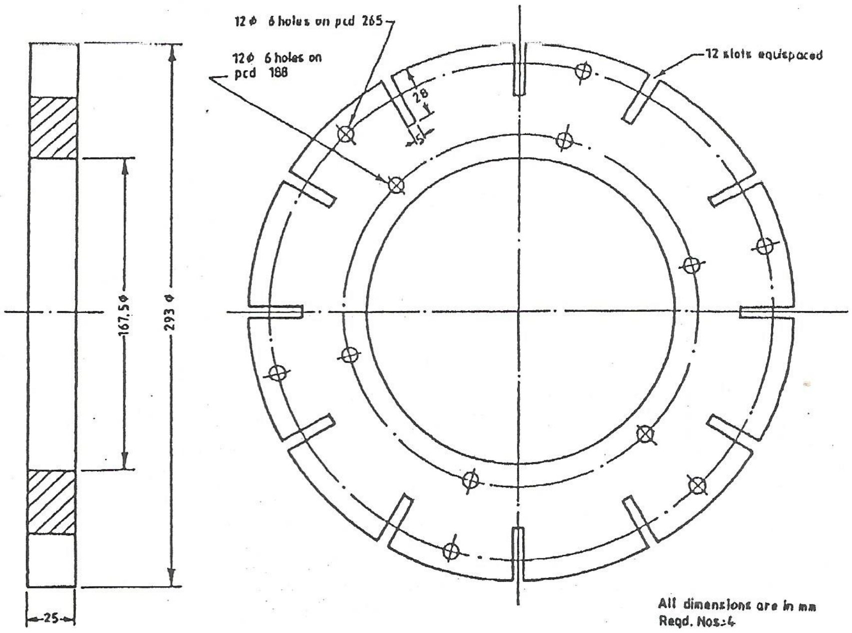

FIG.3 ROTOR DISKS

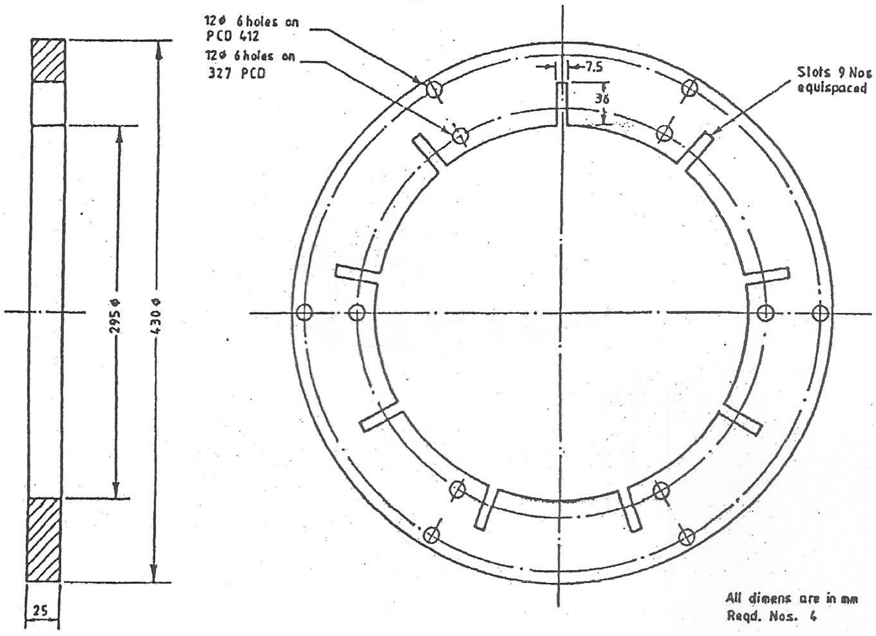

FIG.4 STATOR DISK

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Transient free convective heat transfer from co-rotating concentric disks

BISWAJIT BANERJEE, K. V. CHALAPATHI RAO and V. M. K. SASTRI
Department of Mechanical Engineering, Indian Institute of Technology, Madras 600 036, India

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Abstract—Experimental investigations are carried out for the determination of transient heat transfer coefficients between parallel co-rotating and concentrically stationary disks with internal heat generation. The experimental model simulates a typical induction motor with unequal losses in the stator and rotor at the start-up condition. The data collected cover a range of Taylor numbers for various heat inputs. Quantitative assessment is made for the increase in heat transfer with speed of rotation under free convective ambient conditions. Temperatures along the axial, radial and tangential directions are measured. Unsteady heat transfer coefficients to the ambient air are evaluated in the radially diverging section. The results of the corresponding R-C network are obtained on a digital computer. The predicted values of temperatures at the corresponding nodal points are compared with the measured values and found to be in good agreement. The results are potentially very useful in the thermal design of electrical machines, more specifically, radially ventilated induction motors.

INTRODUCTION

PREDICTION of the temperature distribution in an electrical machine, both for steady and unsteady state conditions, is a subject matter of great interest to researchers and engineers. It is essential to know the magnitudes of the highest temperatures and their locations and deviations from the average value because of their bearing on the design of the machines. The problem assumes greater complexity if one has to predict these values under transient conditions. The temperature distribution also depends upon the variation in type and location of the heat source, the ventilation system and the transient nature of rotor thermal loading.

The electrical analogue approach is one of the most commonly used methods for the prediction of temperature distribution for both steady and unsteady state conditions.

The need for transient analysis arises due to several electrical design factors such as reactive overload capacity, negative phase sequence, etc.

Several authors [1–3] reported work on heat transfer from disks with or without enclosure and for the case of parallel disks [4] without rotation. Mochizuki and Yang [5] reported work with co-rotating parallel disks but with steam as the heating fluid. To the authors' knowledge, no single paper has so far been reported for the determination of the heat transfer coefficient for the case of co-rotating parallel disks together with stationary concentric parallel disks, a configuration that closely resembles the rotor–stator of an electrical machine, with forced or free convection taking place between the disks. The present work aims at determining the heat transfer coefficients under free convective conditions.

SCOPE OF THE PRESENT WORK

A model of a stator and rotor of a typical induction motor with radial duct cooling is simulated. Typical loss distribution in the stator and rotor will be considered and simulated by heat sources at suitable locations. Even though there is forced flow of air, the present analysis is limited to the no-flow condition, i.e. free convection is predominant, with rotation of the rotor superimposed. This may simulate the condition of a fan failure wherein heat loads are to be met by self-ventilation. The heat transfer coefficients, obtained experimentally for different Taylor numbers and different heat losses in the stator and rotor, are used to calculate the convective resistances for the analytical model. The analytical model is solved for the transient and steady state temperature distributions for comparison with the measured values of temperatures.

DESCRIPTION OF EXPERIMENTAL SET-UP

The experimental set-up simulated the rotor and stator of a typical induction motor. It consisted of essentially four pairs of disks forming three radial ducts. Attention was paid to the central duct whereas the remaining two ducts on either side took care of the end effects. A sectional view of the test section is shown in Fig. 1. For simplicity the laminations of the motor are idealized to be a simple disk of solid steel. The disks were held in position by means of tie rods and end flanges which were suitably insulated to minimize axial conduction. The rotor was fixed to two end flanges with a hollow shaft through which thermocouples and power leads were taken out. To simulate the heat generation due to copper losses in

NOMENCLATURE

A heat transfer area, $(2\pi N/4)(D_2^2-D_1^2)+\pi NL(D_2+D_1)$ [m^2]
B spacing between disks [m]
D disk diameter: D_1, inner; D_2, outer; D_m, mean
D_H hydraulic diameter, $2B$ [m]
h average heat transfer coefficient [W m^{-2} K^{-1}]
L axial dimension of disks [m]
N number of disks
n rotational speed of rotor [rps]
Q heat input [W]
T temperature [°C]
Ta Taylor number, $B^2\Omega/\nu$.

Greek symbols
ν kinematic viscosity of air [m^2 s^{-1}]
Ω angular velocity of rotor, $2n$ rad s^{-1}.

Subscripts
∞ ambient
av average
R rotor
S stator.

the conductor, rectangular heater elements each of 150 W capacity, were inserted into the slots made in the periphery of the rotor and stator disks. The stator assembly was similar and concentric with the rotor. After assembly the rotor–stator air gap was about 1 mm. Both rotor and stator surfaces were chrome-plated to prevent rust formation and maintain a clean surface. A power slip ring with brushes was used for power supply to the heating elements. Precautions were taken to prevent slipping and short circuiting of heating elements even at high rotational speeds. A variable speed drive was connected to the rotor shaft through a pulley mechanism for varying the speeds.

A number of copper–constantan thermocouples were fixed at different locations on the surface of the rotor and stator disks (complete details are given in ref. [6]). The experiments were carried out for different heat inputs and at various Taylor numbers. Temperatures in the axial, tangential and radial directions were measured to an accuracy of 0.1°C.

EXPERIMENTAL RESULTS AND DISCUSSION

The heat transfer performance for free convection through the radial ducts of the combined rotor–stator system with internal heat generation is expressed in terms of the average heat transfer coefficient h. Experiments were performed for rotor speeds of 0–700 rpm, which correspond to a Taylor number range of 0–102, where the Taylor number is defined as

$$Ta = B^2\Omega/\nu.$$

The air-side heat transfer coefficient is defined as

$$h = Q/A\Delta T$$

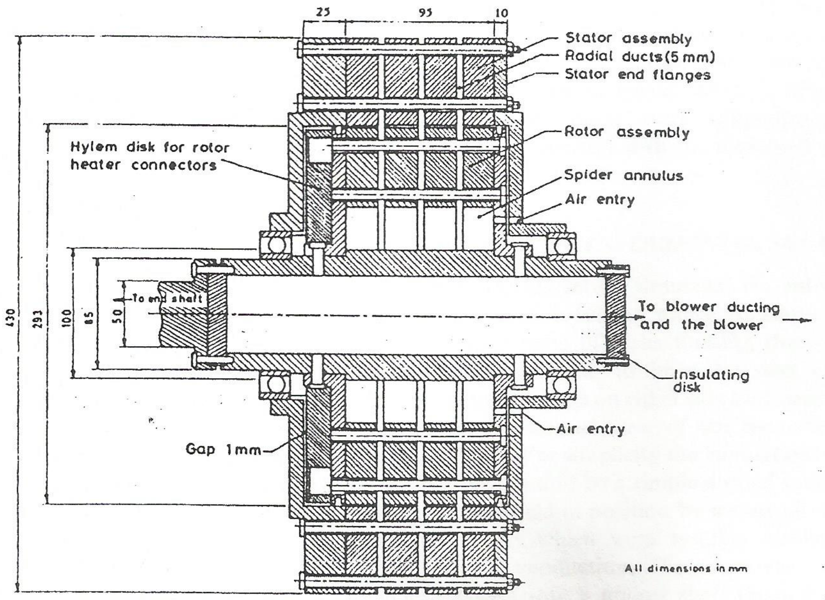

FIG. 1. Sectional view of the test section.