EXPERIMENT OF NATURAL CONVECTION IN A HEMISPHERICAL CAVITY WITH DISCRETE THERMAL SOURCE

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Resumo

A transferência de calor por convecção natural no interior de uma cavidade hemisférica com fonte de calor discreta no centro de sua base foi estudada experimentalmente. Um filme termoresistivo de níquel foi usado como fonte de calor. Tal utilização é vantajosa porque foi possível determinar simultaneamente sua temperatura e troca térmica. A correlação entre os números de Nusselt e de Rayleigh na faixa de $2.5 \times 10^4 \leq Ra \leq 5 \times 10^5$ foi levantada. Os resultados e a correlação obtidos estiveram dentro de $\pm$ 10% de casos similares publicados na literatura.

Palavras-chaves: *convecção térmica, convecção natural, convecção em cavidade.*

Abstract

Natural convection heat transfer inside a hemispherical enclosure with discrete heat source at the center of its base was studied experimentally. A nickel thermoresistive film was used as heat source. Such utilization was advantageous because it allowed to simultaneously determine temperature and heat transfer. Correlation between Nusselt number and Rayleigh number over the range of $2.5 \times 10^4 \leq Ra \leq 5 \times 10^5$ was obtained. Results and the obtained correlation were within $\pm$ 10% of similar cases in the current literature.

Keywords: *heat convection, natural convection, convection inside enclosure.*

1 Introduction

In the last four decades, there has been a growing awareness that natural convection phenomena are of interest and significance in many fields of science and technology, such as meteorology, geophysics, astrophysics, nuclear reactor, scientific instrumentation, processes, fire control, chemical, food and metallurgical industries.

Although buoyancy – driven fluid flow and heat transfer in a differentially heated horizontal and vertical cavities have been investigated extensively, most of the studies consider only fully heated walls. A detailed review of literature indicates that very few efforts have been made to study natural convection from either an isolated heat source or an array of distributed heat sources on a horizontal or vertical wall of an enclosure (KEYHANI et al., 1988 and OSTRACH, 1988).

A particular case where such situation exist is the convective heat transfer inside the dome of a pyranometer which is the instrument for the measurement of global solar radiation. Since pyranometers are exposed continually to all weather conditions the sensor should be hermetically sealed inside its casing. Usually the casing consists of a hemispherical glass dome and the receiver is a small black painted sensor.

In that case the black painted sensor of such instrument is seen as a discrete heat source inside a hemispherical enclosure. It is shown in literature (ANDERSON, 1967) that the sensitivity to a given level of incident radiation and the time of response, as well as the nonlinearity and temperature coefficient, are all markedly affected by the extent and type of heat transfer inside the domes of such instruments.

There are few publications in the related literature which treat the problem of natural convection inside hemispherical enclosure. However, to the present author's knowledge, ANDERSON (1967) was the first to investigate the role of heat transfer in the design and performance of solarimeters. He was followed by CABELLI (1977) who numerically studied natural convection in inclined hemispherical cavities as an application to pyranometry. De LIMA and PARREIRA (1990) made experiment of natural heat transfer inside a hemispherical enclosure commonly used as dome of a pyranometer but the heat source was concentrated on the black painted solar sensor of the instrument. On the other hand, SHIINA et al. (1994) made a very detailed study on natural convection in a hemispherical enclosure where the bottom surface was heated in its entire extension. They obtained a correlation for a wide range of Rayleigh numbers for various fluids. In their experiment they were concerned with the problem related to safety analysis of nuclear reactors.

In the present work, instead of the usual procedure where temperature and heat flux are measured independently, we made use of a thin thermoresistive metallic film on a substrate as heat source. By this way it was possible to simultaneously determine the electric power dissipated on the sensor surface and its temperature knowing the thermal coefficient of resistance and the respective electrical resistance of the termoresistive sensor. As will be detailed further this procedure will lead to the determination of the heat transfer coefficient.

The objective of the present technical note is to experimentally study the natural thermal convection inside a hemispherical enclosure with a discrete thermal source at its center. A Nu x Ra correlation was obtained and compared to similar cases published in the literature.

2 Experimental apparatus and procedure

The scheme of the assembled experimental apparatus is shown in Fig. 1. A hemispherical glass dome with 5 cm of insulation on the upper surface is placed on a 5 cm thick polysterene plate of 20 cm x 20 cm. The glass dome has inner radius 3.5 cm and tickness 0.5 mm. At the center of the base is a heat source, a nickel resistive film was deposited on a phenolic substrate, forming a uniformly heated square of 2.10^{-4} m². Copper – constantan thermocouples were placed to read surface temperatures in locations T_2, T_c and T_1 and to read the ambient temperature T_3 as shown in Figure 1. In Figure 2 there are contours of streamlines and isotherms inside a hemisphere heated from below, as observed in studies of CABELLI (1977), SHIINA et al. (1994), and ASFIA et al. (1996). The apparatus was installed in a 4 m x 2.5 m x 3 m room. The ambient temperature trolled during the experiment.

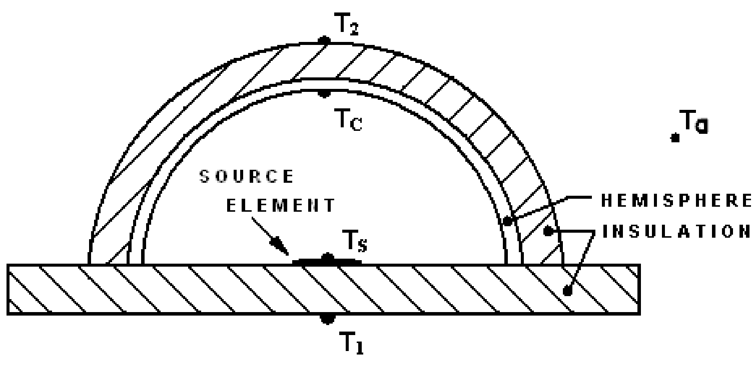

Figure 1: Scheme of the experimental apparatus.

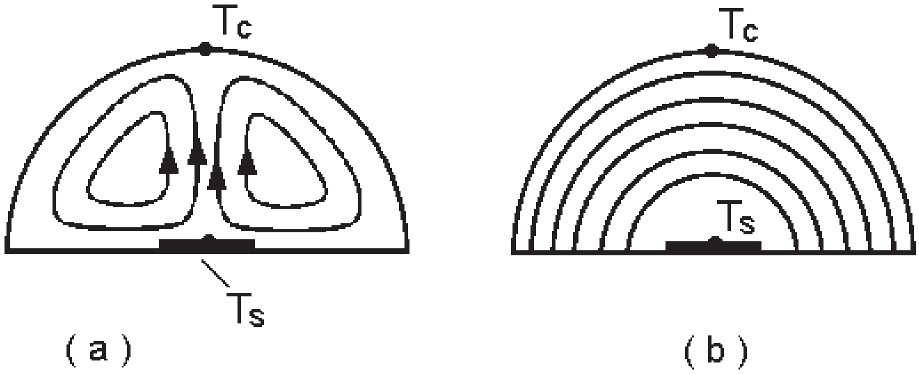

Figure 2: Streamlines (a) and isotherms (b).

The source element was connected to a variable DC power supplier. In average, three hours were necessary to attain steady state for the beginning of measurements. The thermocouple's outputs were measured by a Robert Shaw milivoltimeter to $\pm$ 10 μ V, which results in sensitivity of $\pm$ 0.25ºC. Voltage and current inputs to the heat source element were measured using a FLUKE 8050A digital multimeter with an accuracy of $\pm$ 0.3%. Power input to the heat source was then calculated using the measured voltage and current. This procedure resulted in an uncertainty of $\pm$ 1% in the calculated electrical power inputs. The uncertainty associated with the length measurements was $\pm$ 0.5 mm.

Values of all thermophysical properties of the air were obtained at the average temperature between the heat source and the internal surface of the dome. The thermophysical properties of the air were assigned an conservative uncertainty of $\pm$ 2%.

The temperature T_s of the thermoresistive source element was determined by the relation.

$$R_s = R_a\,[1+\alpha\,(T_s - T_a)] \tag{1}$$

where R_s is the electrical resistance of the heat source at temperature T_s, T_a is the ambient temperature and α is the thermal coefficient of the resistance. The resistance R_s is determined by relation $R_s = V^2/Q$.

The convection heat transfer coefficient of the heat source was calculated as

$$h = (Q - Q_{rad} - Q_{cond})/A_S\,(T_s - T_c) \tag{2}$$

where Q is the power input, Q_{rad} is the heat loss by radiation, Q_{cond} is the heat loss by conduction via phenolic base, A_s is the heat surface area, T_s is the heat source's temperature and T_c is the dome's the internal surface temperature. Q_{rad} at the largest temperature difference $(T_s - T_c)$ is about 2% of Q, Q_{cond} at the smallest temperature difference $(T_s - T_c)$ is about 8% of Q and the largest temperature difference $(T_s - T_c)$ is 2% of Q.

The Nusselt and Rayleigh numbers were defined as

$$Nu = hR/k \tag{3}$$

$$Ra = g\beta(T_s - T_c)R^3\,\mathrm{Pr}/\nu^2 \tag{4}$$

where k is air thermal conductivity, R is inner radius of the hemisphere, g is acceleration due to gravity, β is thermal expansion coefficient, ν is the kinematic viscosity of the air inside the hemisphere and Pr is the Prandtl number. The overall uncertainty in the Nusselt and Rayleigh numbers varies with the power dissipated by the heat source. Smaller power input results in larger uncertainty due to the smaller difference of temperature between the heat source and the internal surface of the dome and due to energy loss by conduction. In average, the estimated uncertainties in the Nu and Ra number were 5 and 7%, respectively.

3 Results and discussion

Figure 3 presents the Nusselt number plotted as a function of the Rayleigh number. A typical correlation for the Nusselt number in natural convection inside enclosure is of the form

$$Nu = CRa^{n} \quad (5)$$

where C and n are assigned constants whose values depend on the enclosure geometry and flow regime. In Bejan (1984) a scale analysis shows that Nu is proportional to $Ra^{0.25}$, confirming experimental measurements of laminar natural convection. Therefore the value of 0.25 was assigned to n in the equation (5).

The correlation representing all the experimental values plotted in Figure 3 is of the form

$$Nu = 0.198Ra^{0.25} \quad (6)$$

Applying the experimental data to the equation (6) it was found the deviation range from –4% to 6%, being the deviation defined as

$$Deviation = [Nu_{cor} - Nu_{\exp}] / Nu_{\exp} \quad (7)$$

where Nu_{cor} and $Nu_{\exp}$ are the correlation and experimental Nusselt number, respectively.

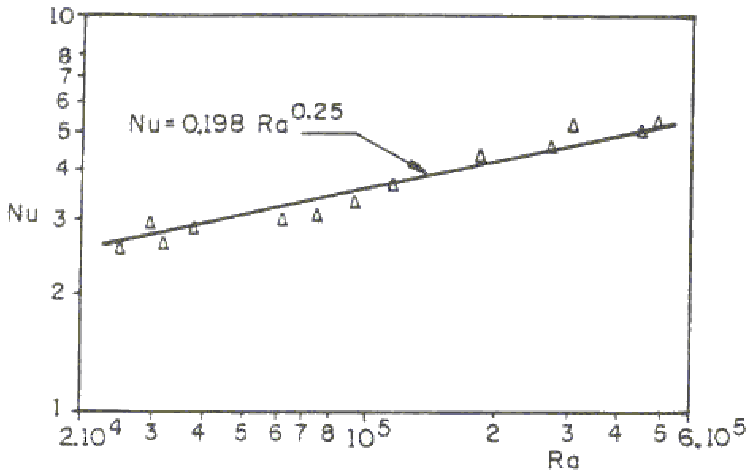

Figure 3: Measured Nu and Ra numbers and correlation.

Mikheyev (1964) presented a correlation of heat convection for various enclosed spaces including horizontal, vertical, annular, and spherical enclosures. He obtained a correlation for the Nusselt number in the form of $Nu = 0.18Ra^{0.25}$ which is about 10% less than the presented correlation. SHIINA et al. (1994) proposed a correlation for an experiment similar to the present study which is approximately 7% less than the here presented. They worked with a full heated surface. It is thought that the cause of that difference is due to the more pronounced thermal convection inside enclosure with discrete heat source.

4 Conclusion

In this technical note the problem of natural convection inside a hemispherical enclosure with discrete heat source was studied. A relatively good agreement was obtained between this experimental data and correlation with experimental data of similar cases published in the literature.

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NOMENCLATURE

A_s	surface area	[m²]
C	Coefficient in equation (5)	
g	gravitational acceleration	[m/s²]
h	heat transfer coefficient	[W/ m²K]
k	thermal conductivity	[W/mK]
n	coefficient in equation (5)	
Nu	Nusselt number	
Pr	Prandlt number	
Q	electrical power input	[W]
Q_{cond}	conduction heat transfer	[W]
Q_{rad}	radiation heat transfer	[W]
R	radius of hemisphere	[m]
Ra	Rayleigh number based on radius R	
R_a	electrical resistance at ambient temperature	[Ω]
R_s	electrical resistance at temperature T_s	[Ω]
Ta	ambient temperature	[K]
Tc	dome temperature	[K]
Ts	heat source temperature	[K]
V	electrical voltage	[V]
Greek symbols		
α	thermal coefficient of resistance	
β	thermal coefficient of volumetric expansion	
ν	kinematic viscosity	

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