CHARACTERISTICS OF HEAT TRANSFER FROM PACKAGE IN REFRIGERATING ROOM WITH LAMINAR DOWNFLOW

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***Key Words*:** Heat Transfer, Heat Transfer Coefficient, Free-Forced Convection, Heat Transfer Equation, Combined Laminar Flow

To investigate the heat transfer characteristics of a package in a refrigerating room with downflow, the temperature distribution and heat transfer coefficient were measured in the case where one wall of the vertical duct was isothermally heated and gas flowed downward through the duct in the laminar flow range as a model of the package wall in a refrigerating room.

The temperature distribution is nearly uniform except near the heated surface and confront plate. The cross-sectional temperature gradient near the heated surface becomes steep with decreasing Reynolds number of the ambient fluid. The effect of free convection on average Nusselt number is large in the range of low Reynolds number and becomes small with increasing Reynolds number. An empirical equation for heat transfer was obtained within the operating conditions of this experiment.

Introduction

When a food package put in a refrigerating room for storage or freezing, the density of fluid near the package wall is decreased by the temperature rise and free convection is generated. This free convection promotes heat transfer from the package wall to ambient fluid. In the refrigerating room, cold fluid flows in the laminar flow range through the space between the packages and free convection is combined with this laminar forced convection.

Generally, the cold fluid spout is located in the upper part of the refrigerating room and consequently the cold fluid is made to flow in a downward direction

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in the space between the packages (the flow rate is usually 0.1–0.5 m/sec). In this case, free convection near the package wall flows upwardly.

If the temperature difference between the package wall and the ambient fluid is high, the free convection becomes large and cancels out the forced convection of the cold fluid. Thus it is conjectured that the heat transfer rates are greatly reduced by the stagnant flow combined with free and forced convection flows. On the other hand, if the cold flow is an upflow, its direction is the same as that of the free convection. In that case it is expected that the heat transfer rate is increased.

Many experimental and theoretical approaches have been taken to fluid flow in a refrigerating room and to similar flow systems.[1–7,9,11–13,15,16]

However, there are few works concerning combined forced-free laminar flow between the packages. Thus the pattern of heat transfer from the package wall in a refrigerating room is not yet established as to heat transfer coefficients under various operating conditions. Such information is important for energy saving.

In this study, as basic research on the effect of direction of cold fluid flow on the cooling of the package, the temperature distributions and heat transfer coefficients for a vertical duct wall with a constant temperature were measured in the case where only one side wall of the duct was heated and gas flowed downward through the duct in the laminar flow range. This is a model of the package wall in a refrigerating room in which the temperature differences between the package wall and the cold ambient fluid are the same as those between a heated wall and a flowing gas of normal temperature in the duct.

An empirical equation for the coefficients of heat transfer from the isothermally heated wall in the duct was obtained by taking account of Reynolds number, Grashof number and the geometrical conditions of the vertical duct for combined forced-free convective laminar flow. Also, the heat transfer coefficients determined in this study were compared with those in the case of pure free convection or pure forced convection, and with those in the literature.

1. Experimental Apparatus and Procedure

Figure 1 shows a schematic diagram of the experimental apparatus.

To investigate the heat transfer phenomena from a package in a refrigerating room, a vertical duct was made as a model of the system. Acrylic resin ducts of sectional areas $38 \times 12\,\mathrm{cm}^2$, $38 \times 10\,\mathrm{cm}^2$, $38 \times 8\,\mathrm{cm}^2$ and $38 \times 5\,\mathrm{cm}^2$ were used. The length of the duct was 200 cm including an entrance region. To establish the laminar velocity distribution, a calming section and a distributor with proper pressure drop were installed at the entrance of the duct.

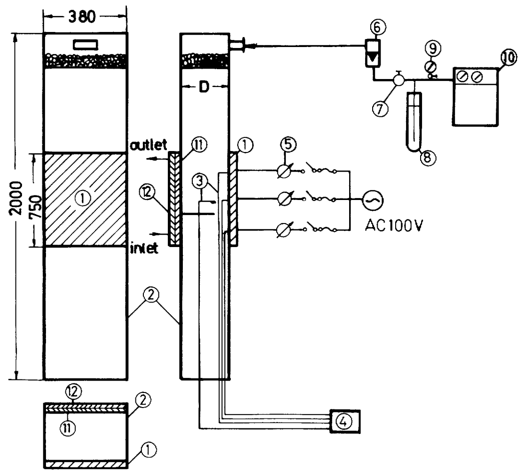

Fig. 1. Experimental apparatus

The heating section was located in the center of the duct and its area was 38 (width) $\times$ 75 (height) cm^2. The heating section consisted of a copper plate (3 mm thick) to which nicrome wires were attached as heaters in three separate places for heating severally in order to obtain isothermal conditions. On the other hand, the confront plate was not heated and a water jacket was attached to this plate in order to obtain isothermal conditions. The sectional area of the duct was changed by moving the confront plate.

To measure the wall temperature in the test section, CC thermocouples were placed at twelve points in the heated plate and at three points in the confront plate. To investigate correctly the cross-sectional temperature distribution in the boundary layer and the fluid in the duct, gas temperature was measured by use of a sheathed CA thermocouple probe of very small diameter (0.2 mmϕ).

Air of ordinary temperature from a compressor was passed through a constant-pressure device and a flow meter, and then fed into the duct. After the temperature in the duct reached steady state, the sheathed thermocouple probe was put into the system from the bottom of the duct and was traversed in the cross-sectional direction by moving a travelling microscope to measure the local temperature in the duct. Gas temperature was measured at three points on the vertical center line of the heated plate ($y = 10$ cm, 37.5 cm (center) and 65 cm) and at about fourteen ($D = 5$ cm) $\sim$ twenty points ($D = 12$ cm) in one cross section, and the spacing was 1.0 mm in the boundary layer and 10.0 mm in the ambient fluid.

The experimental conditions were as follows: Temperatures of the heating and nonheating plates were in the range of 313–353 K and 286–294 K, respectively. Reynolds number was in the range of 50–700 and H/D was 6.25, 7.50, 9.38 and 15.0 [—].

2. Experimental Results

2.1 Temperature distribution

Generally, the temperature distribution and the heat transfer coefficient from the heating section in this system may be affected by variables such as the Reynolds number, Re, Grashof number, Gr, and H/D.

Figure 2 shows the temperature distribution in the cross section at the middle point of the heated surface ($y=y_M$) with Re as a parameter. From Fig. 2, the temperature distribution is nearly uniform except near the heated surface and the confront plate. Especially, one should note that as Re is decreased the cross-sectional temperature gradient near the heated surface becomes steep.

The temperature distribution shown in Fig. 2 is quite different from that with ordinary transport phenomena: as the axial velocity becomes large, the rate of heat transfer from the heated surface may decrease. This peculiar tendency was also obtained in other cases ($t_H=313$–333 K, $D=5$–12 cm) and it was notable with higher t_H or smaller D.

2.2 Effect of operating conditions on heat transfer coefficient

The relations between average Nusselt number, $\overline{Nu}$, from the heated surface in the duct and Re, Gr and H/D were investigated.

Generally, the average heat transfer coefficient, $\bar{h}$, and $\overline{Nu}$ can be calculated from the heat balance in the test section. However, in this experiment the velocity distributions in the test section were not measured. When Re was small, back-flow was observed in the exit region of the duct (bottom). Consequently, the axial velocity became negative and the average temperature could not be estimated correctly. In this study, $\bar{h}$ and $\overline{Nu}$ were calculated by Eqs. (1) and (2), based on the cross-sectional temperature gradient near the heated surface.

$$-\lambda\left(\frac{\partial t}{\partial x}\right)_{x=0}=h(t_H-t_A) \qquad (1)$$

$$\overline{Nu}=\frac{D\bar{h}}{\lambda}=\frac{D}{\lambda H}\int_0^H h\cdot dy \qquad (2)$$

where the temperature gradient, $(\partial t/\partial x)_{x=0}$, was obtained by that between the heated surface and the nearest experimental point to the heated surface.

Figure 3 shows the relation between Re and experimental $\overline{Nu}$ with Gr as a parameter. From Fig. 3, as Re is increased, $\overline{Nu}$ decreases gradually. It seems that the effect of free convection on $\overline{Nu}$ is large in the range of low Re and becomes small with increasing Re. This means that when Re is high, the upward free convective flow is weakened by the downward forced convective axial flow and becomes a stagnant flow. From these experimental results, $\overline{Nu}$ was proportional to $Re^{-0.088}$.

Figure 4 shows the relation between H/D and $\overline{Nu}$ with Gr as a parameter. From Fig. 4, when H/D is

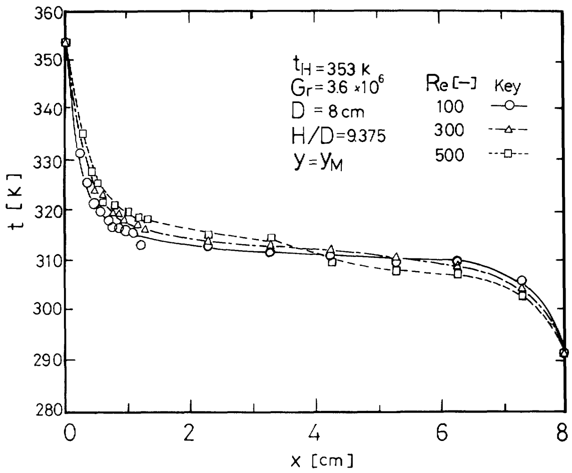

Fig. 2. Effect of Re on experimental temperature profile

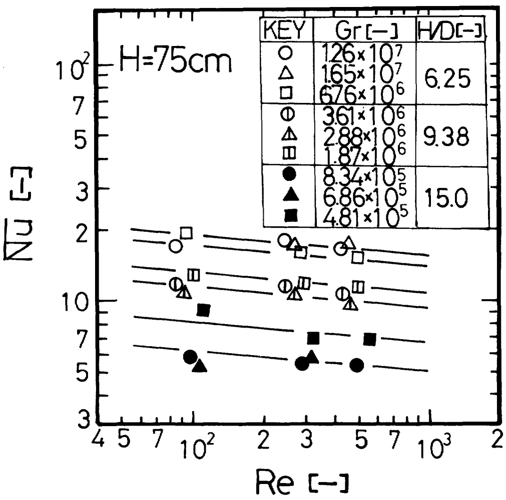

Fig. 3. Relation between experimental $\overline{Nu}$ and Re

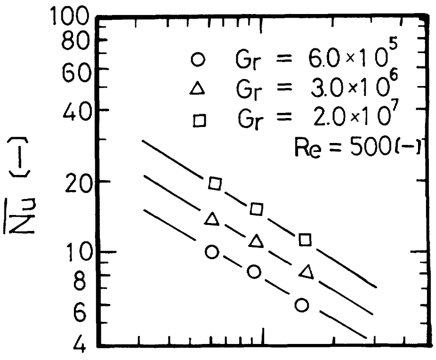

Fig. 4. Relation between experimental $\overline{Nu}$ and H/D

increased, $\overline{Nu}$ decreases. This means that $\overline{Nu}$ is affected sensitively by H/D and is also decreased with decreasing D. $\overline{Nu}$ was proportional to $(H/D)^{-0.60}$. Furthermore, $\overline{Nu}$ was also affected by Gr and was proportional to $Gr^{0.14}$.

Figure 5 shows the relationship between Gr and $\overline{Nu}(Re)^{0.088}(H/D)^{0.60}$. From Fig. 5, the following empirical equation was obtained for the heat transfer coefficient in a vertical duct with an isothermal heating section.

$$\overline{Nu}=9.2Re^{-0.088}Gr^{0.14}(H/D)^{-0.60} \tag{3}$$

In Eq. (3), the constant was decided from all the experimental data by the least-squares method. The applicable ranges of Eq. (3) are from 100 to 500 for Re, 4×10^5 to 2×10^7 for Gr and 6.25 to 15.0 for H/D.

3. Discussion

From Fig. 3, the rate of heat transfer from the package wall in a refrigerating room is decreased with increasing cold fluid in the downflow in this operating condition. These experimental facts may indicate the validity of the assumptions for stagnant flow combined with the free and forced convection as mentioned in the introduction to this paper.

Figure 6 shows the average heat transfer coefficient for the combined flow in this study and for pure free or forced convective flow in the system used by the authors. The solid line shows values of $\bar{h}$ obtained by the present Eq. (3), and the broken line and the dotted line show those obtained in the case of pure laminar forced convective flow[8] and pure free convective flow[6,10,14] respectively.

It seems that since the direction of the forced convective flow is opposite that of the free convective flow, the latter is weakened by the former and consequently the present experimental result (Eq. (3)) is obtained.

Furthermore, since from Fig. 6 the forced convective flow may become larger than the free convective flow in the range of high Re, it is conjectured that the stagnant flow is swept away by the forced convective flow with very high Re and then the heat transfer rate may change to an increase. However, this must be left for future study.

Figure 7 shows previous results for the Nusselt number for a vertical heated surface in a flow system similar to that used by the authors[3,4,11,13].

From Fig. 7, it appears that the heat transfer rate in this downflow system is considerably smaller than that in previous upflow-system results, and there are

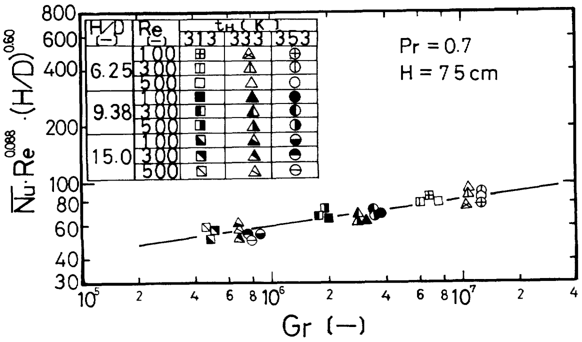

Fig. 5. Relation between $\overline{Nu}Re^{0.088}\ (H/D)^{0.60}$ and Gr

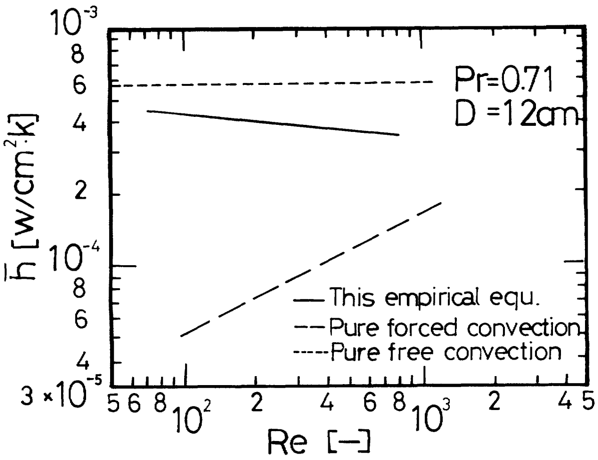

Fig. 6. Comparison of $\bar{h}$ with present study and some pure convection results

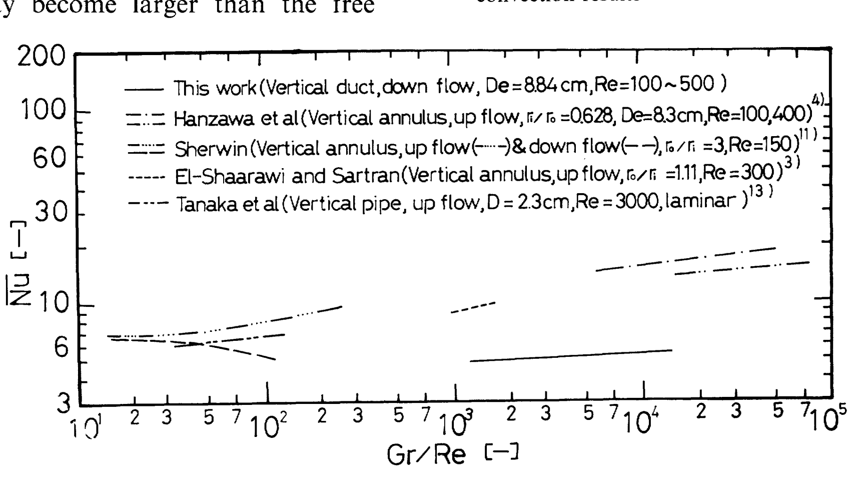

Fig. 7. Average Nu compared with some previous results

reasonable values in the comparison of $\overline{Nu}$ between the downflow system of Sherwin and the present one.

Conclusion

To investigate the heat transfer characteristics of a package in a refrigerating room, the temperature distribution and heat transfer coefficient were measured in the case where one wall of the vertical duct was isothermally heated and gas flowed downward through the duct in the laminar flow range as a model of the package wall in a refrigerating room.

The following results were obtained.

(1) The cross-sectional temperature gradient was large in the vicinity of the heated surface and nearly uniform in the ambient fluid.

(2) The temperature gradient near the heated surface becomes steep with decreasing Re.

(3) The heat transfer coefficient was affected by Re, Gr and H/D, and the following empirical equation was obtained within the limits of this experiment.

$$\overline{Nu}=9.2Re^{-0.088}Gr^{0.14}(H/D)^{-0.60}$$

(4) It was proved that the rate of heat transfer from the package wall in the refrigerating room decreased with increasing cold fluid in the downflow.

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Nomenclature

C_p	= specific heat of fluid	[J/g·K]
D	= plate spacing	[cm]
D_e	= equivalent diameter	[cm]
Gr	= Grashof number, $D^3g\beta(t_H-t_A)/v^2$	[—]
g	= gravitational acceleration	[cm/s²]
H	= height of heating plate	[cm]
h	= heat transfer coefficient	[W/cm²·K]
Nu	= Nusselt number, hD/λ	[—]
Pr	= Prandtl number, $C_p\mu/\lambda$	[—]
Re	= Reynolds number, Dv/ν	[—]
r_i	= radius of inner tube in annulus	[cm]
r_0	= radius of outer tube in annulus	[cm]
t	= temperature	[K]
v	= velocity in y-direction	[cm/s]
x	= horizontal coordinate	[cm]
y	= axial coordinate	[cm]
β	= volumetric coefficient of expansion of fluid	[K⁻¹]
λ	= thermal conductivity of fluid	[W/cm·K]
μ	= viscosity of fluid	[Pa·s]
ν	= kinematic viscosity of fluid	[cm²/s]
ρ	= density of fluid	[g/cm³]

⟨Subscript⟩

A	= inlet zone
H	= heating zone
M	= middle point of heating zone
W	= wall

⟨Superscript⟩

—	= average

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