

CONDENSATION HEAT AND MASS TRANSFER FROM STEAM/AIR MIXTURES TO A RETORT POUCH LAMINATE

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A system employing a stainless steel condensing block was designed and fabricated for measuring heat flux during film condensation of steam/air mixtures on a retort pouch laminate surface. The block was incorporated in a chamber that could accommodate steam/air mixtures up to 50% air, by volume. The effects of the Reynolds number (Re), and the bulk temperature (T_b) on the heat and mass transfer in the vapor-liquid condensate region were investigated, for a configuration of horizontal medium flow, with condensation taking place on the underside. The sensible fraction of the heat transferred to the vapor-liquid condensate interface was negligibly small. The data were well correlated by the equation

$$Sh = 205.7 Re^{.0746} (Sc)^{.0526}$$

The quantitative disparity observed between other investigators and this work was attributed to enhancement of heat and mass transfer due to disturbances on the condensate film surface. Generally an increase in T_b was found to result in increased heat and mass transfer coefficients and for a given T_b the heat and mass transfer coefficients could be approximated by linear functions of air content (x_a). The overall heat transfer coefficient (H) could be approximated by logarithmic functions of the surface temperature (T_w) and the temperature difference (ΔT) between the pouch surface and the steam/air medium.

INTRODUCTION

The retort pouch is a multi-layer polymer-foil pouch used in lieu of the metal can or glass jar for shelf-stable, refrigerated and frozen foods. This packaging technique is one of the significant innovations of the past two decades in the food and pharmaceutical industries. Pflug (1964) and Pflug and Borrero (1967) recognized three potential heating media for thermal processing of retort pouched food products, namely: pure steam, water/air and steam/air cooks. Despite the advantages steam/air systems offer over the alternative techniques, commercialization of the process is slow. This reluctance can be partly attributed to the meagre fundamental information available regarding heat transfer in the air content range of interest to food processors. The experimental investigation reported in this paper was prompted by the indicated paucity of data.

In addition to the physio-thermal properties of the processed product, other factors that are important to the rate of heat transfer from a steam/air medium into a processed product are air content, medium flow, bulk temperature and package orientation. This work was undertaken to investigate the heat transfer from steam/air mixtures to the underside of a retort pouch laminate with horizontal steam/air flow. This configuration was chosen, among a number of other configurations that pertain to the problem, since it was found to be least studied from both the analytical and the numerical view-point.

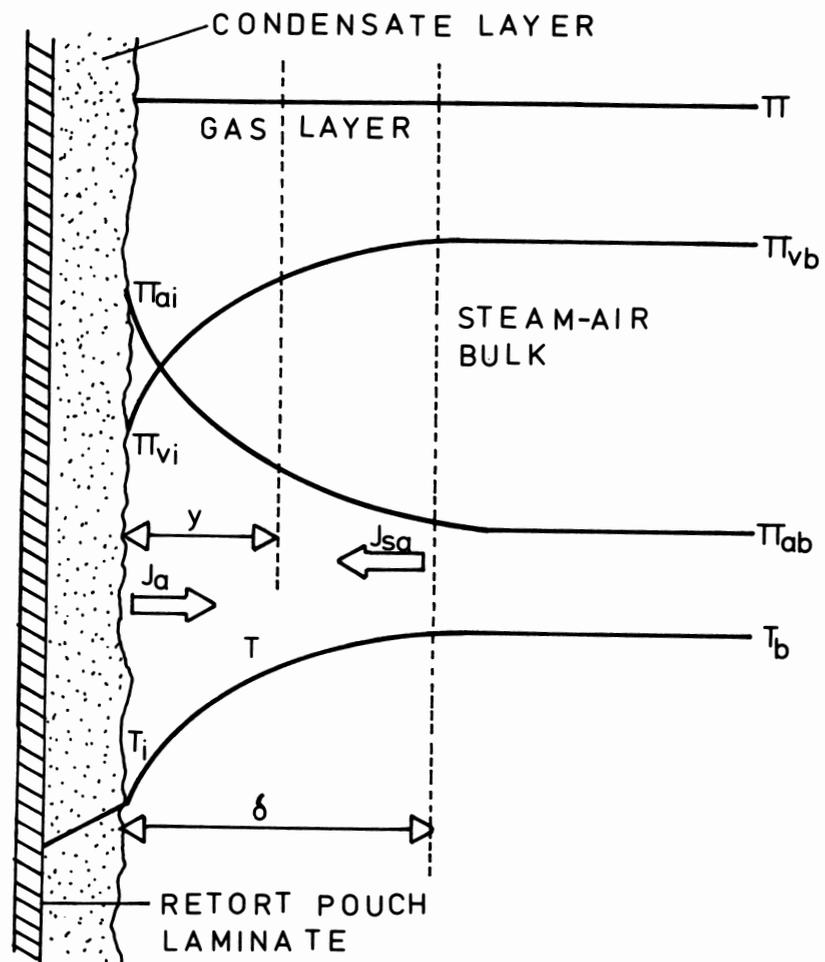


Figure 1. Steam/air condensation mechanism.

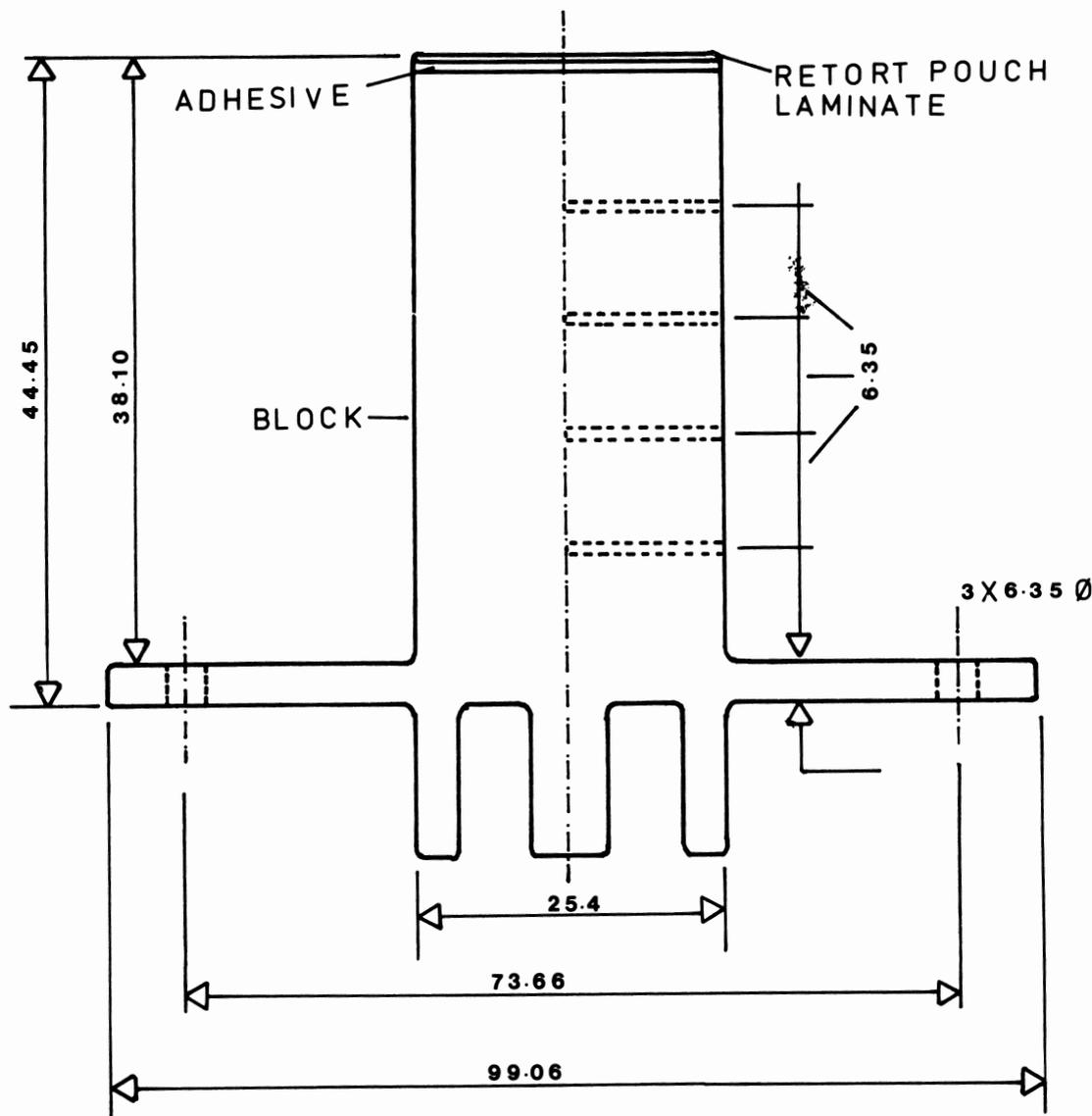


Figure 2. Condensing block and surface.

PREVIOUS WORK

The presence of small quantities of air in condensing steam has a profound influence on heat transfer. Reference to Fig. 1 shows a cold product surface in contact with the steam/air medium. Condensation occurs on the cold surface and the air is carried with the vapor to, and accumulates at, the liquid-vapor interface. The partial pressure of air at the interface increases above that in the bulk of the mixture, producing a driving force for air diffusion away from the surface. This diffusion is exactly counterbalanced by the diffusion of the steam/air mixture towards the interface. Since the total pressure remains constant, the partial pressure of vapor at the interface is lower than that in the bulk mixture which provides the driving force for vapor diffusion towards the interface. The temperature variation in the region of the interface is as illustrated in Fig. 1.

One of the earliest quantitative study of this phenomena was undertaken by Othmer (1929). He concluded that an air mass fraction as low as 0.5% caused more than a 50% reduction in the heat transfer rate. This conclusion has been confirmed recently by Minkowycz and Sparrow (1966) and Sparrow et al. (1967) for stagnant and flowing steam/air mixtures for air mass fractions ranging between 0.05 and 0.1. Their results indicated that higher heat transfer rates are associated with higher bulk temperatures and flow. In an effort to obtain heat transfer data at higher air contents, relevant to food processing, a number of investigators (Blaisdell 1963; Pflug 1964; Yamano 1967; Adams and Peterson 1982; Peterson and Adams 1982; Ramaswamy 1983) have employed lumped capacity techniques to evaluate the overall heat transfer coefficient at different air contents of steam/air media. The wide

variations in the reported "heat transfer coefficients" is not surprising, given the diverse experimental conditions and configurations employed. Also, none of these studies focused on the heat transfer mechanism at the condensation surface. Almost all the above-mentioned studies have reported increased heat transfer rates as a result of higher medium flow. Unfortunately, the data are presented in such a form that comparison with well-established heat and mass transfer correlations is impossible.

For the horizontal medium flow configuration with condensation on the underside, little has been found in the literature. The only study found was conducted by Gestermann and Griffith (1967), under stagnant conditions, who were able to conclude that heat transfer depended on gravity and surface tension only.

The subject of this study was therefore

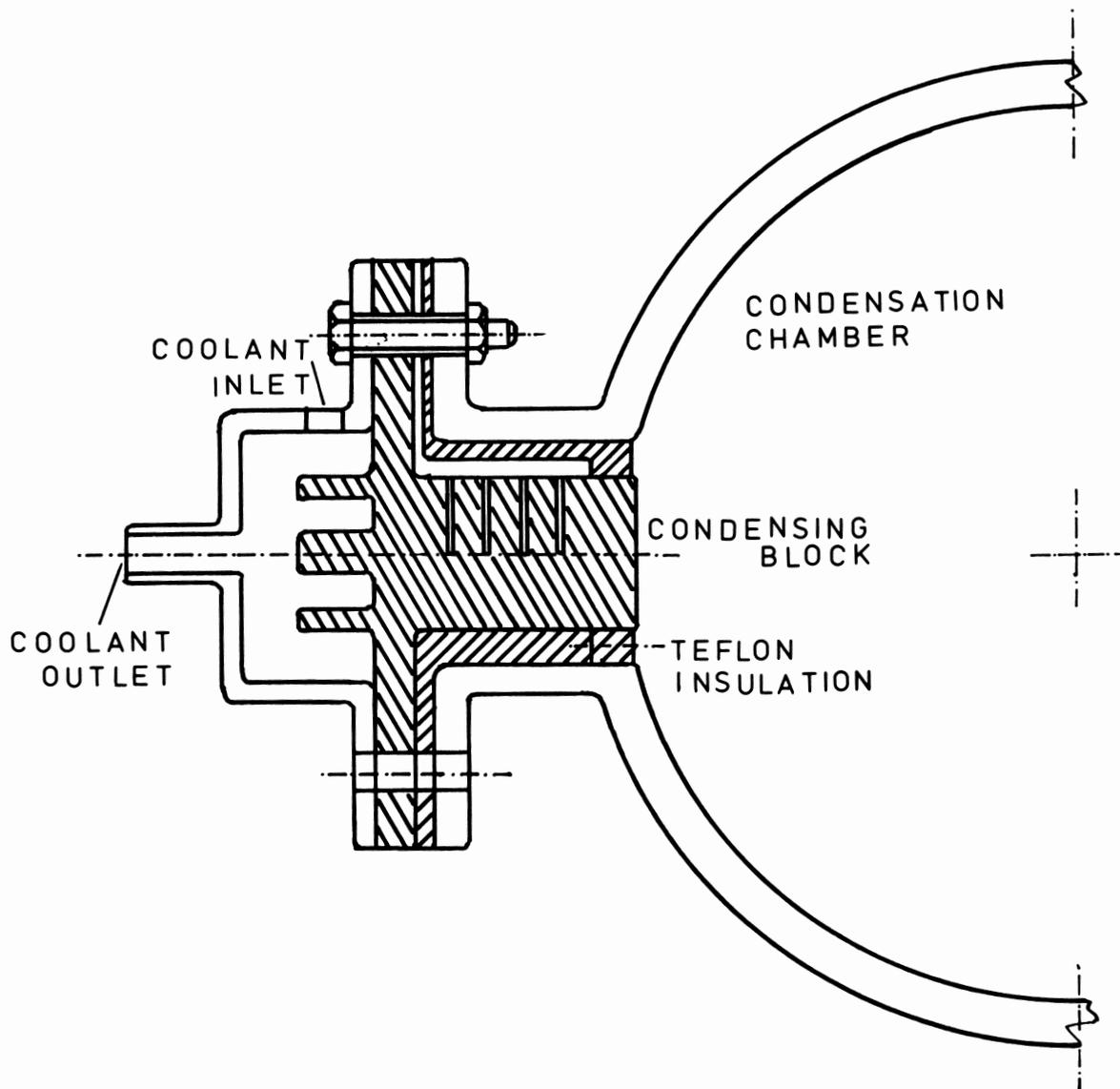


Figure 3. Section of the tubular condensation chamber with the condensing block in situ.

to investigate the influence of larger air contents and horizontal steam/air flow velocity on heat transfer rates in the region of the liquid-vapor interface, assuming a condensation mechanism as illustrated in Fig. 1, with condensation occurring on the underside of a retort pouch test surface.

THEORETICAL ANALYSIS

Referring to Fig. 1, the molar flux of air (J_a) passing through a plane parallel to and at a distance y from the interface is given by (Collier 1981):

$$J_a = J \cdot x_a + D \cdot c \cdot \frac{dx_a}{dy} \quad (1)$$

In a similar manner the molar flux of vapor is given by:

$$J_v = J \cdot x_v + D \cdot c \cdot \frac{dx_v}{dy}, \quad (2)$$

but

$$\frac{dx_v}{dy} = -\frac{dx_a}{dy}; x_v = 1 - x_a. \quad (3)$$

Therefore

$$J_v = J \cdot x_v - D \cdot c \cdot \frac{dx_a}{dy} \quad (4)$$

Eliminating J between Eqs. 4 and 1 gives,

$$J_v = -D \cdot c \cdot \frac{dx_a}{dy} \left[\frac{x_v}{x_a} + 1 \right] \quad (5)$$

Integrating the above equation between the interface ($y = 0$) and the edge of the diffusion layer ($y = \delta$) gives,

$$J_v = \frac{D \cdot c}{\delta} \ln \left[\frac{1 - x_{vi}}{1 - x_{vb}} \right] \quad (6)$$

Equation 6 can be written in terms of mass concentration flux (j_v) as,

$$j_v = K \cdot \rho \cdot \ln \left[\frac{1 - x_{vi}}{1 - x_{vb}} \right]$$

or

where

$$j_v = \frac{K \cdot \rho}{\Pi_{am}} (\Pi_{vb} - \Pi_{vi}) \quad (7)$$

$$\Pi_{am} = \frac{\Pi_{ai} - \Pi_{ab}}{\ln (\Pi_{ai}/\Pi_{ab})} \quad (8)$$

and

$$K = \frac{D}{\delta}. \quad (9)$$

The heat transfer is also affected by the occurrence of mass transfer. The total heat flux at a distance y from the interface is made up of two components, namely a sensible heat transfer (conductive) component and a sensible heat transport (convective) component as follows:

$$q_c = \lambda \frac{dT}{dy} + j_v C_p (T - T_i). \quad (10)$$

Integrating from $y = 0$ to $y = \delta_T$ and using $h = \lambda/\delta_T$ gives:

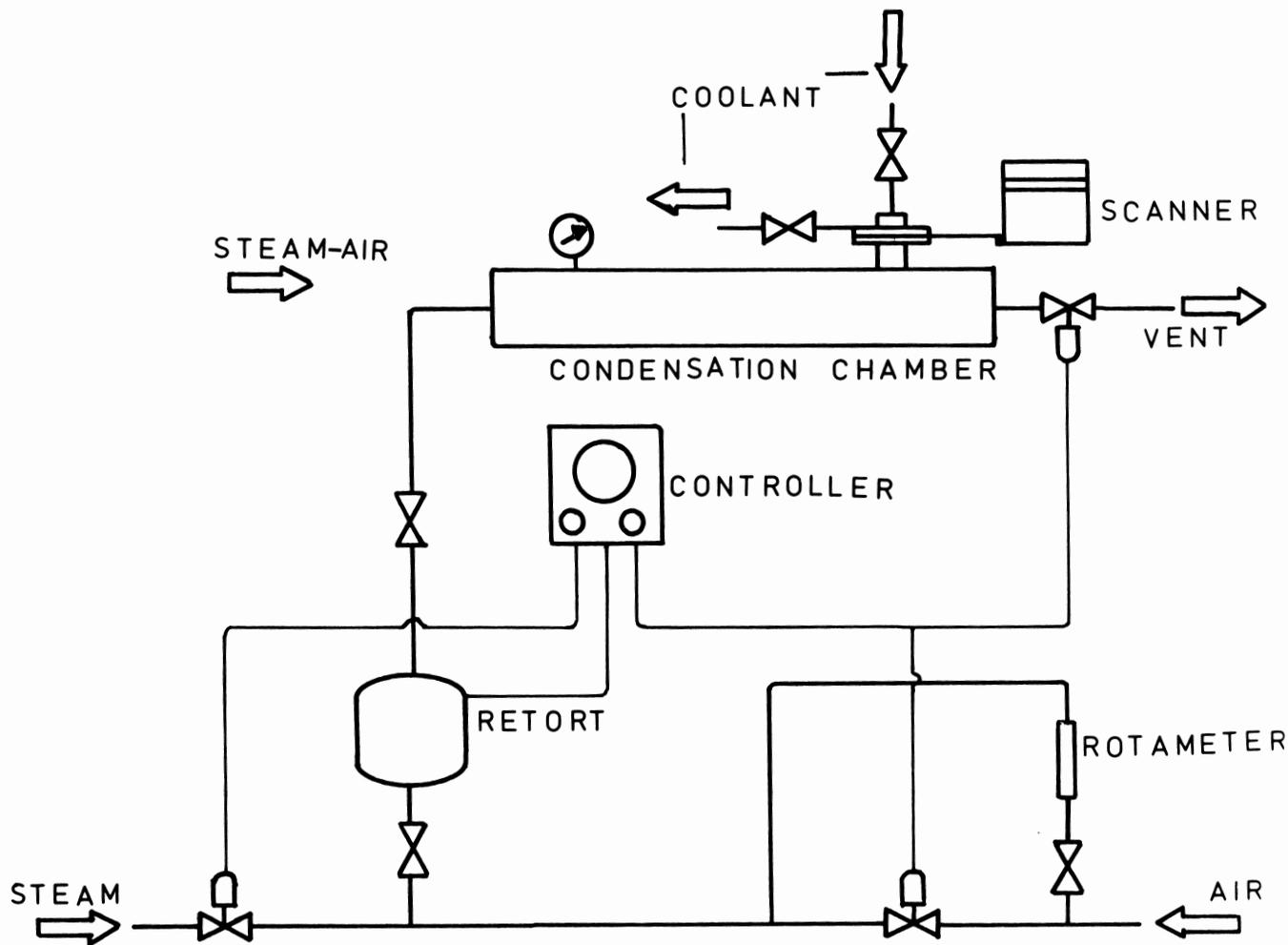


Figure 4. A flowsheet of the experimental unit.

$$q_c = \frac{n}{1 - e^{-n}} \cdot [h \cdot (T - T_i)] = h' \cdot (T - T_i), \quad (11)$$

where

$$n = (j_v \cdot C_p / h).$$

Therefore, h and h' are the sensible heat transfer coefficients with and without mass transfer, respectively. The correction in Eq. 11, that accounts for the effect of mass transfer was reported by Ackermann (1937) and Colburn and Drew (1937) to be insignificant. Colburn and Hougén (1934) proposed a heat transfer model made up of two components, namely the sensible heat transfer through the air-vapor layer to the interface and the latent heat released due to condensation of the vapor reaching the interface as:

$$q_t = q_{fg} + q_c = \frac{K \cdot h_{fg}}{\Pi_{am}} \rho \cdot (\Pi_b - \Pi_w) + h' \cdot (T_b - T_i). \quad (13)$$

For any flow regime (laminar, transition and turbulent) and geometry (tube or flat plate), a suitable correlation can be used to evaluate h' , and K can be evaluated by making use of Reynold Colburn analogy

between heat and mass transfer as follows;

$$K = \frac{h'}{\rho C_p} \left[\frac{Pr}{Sc} \right]^{2/3} \quad (14)$$

It has, however, been observed that actual heat and mass transfer coefficients are higher than those predicted by the correlations (Collier 1981), mainly as a result of unstable falling liquid films. As a result of this disturbance waves develop and grow on the interface at all Reynolds numbers. A summary of studies regarding methods for estimating this enhancement has been compiled by Collier (1981). The simplest of these was proposed by Wallis (1969) and was based on the expected analogy between heat and mass transfer and the increase found in friction factors (Hewitt and Hall-Taylor 1970). It involves multiplying h' by the ratio (f_i/f_g); the ratio can be estimated by

$$f_i/f_g = 1 + 300 \left(\frac{\delta}{d} \right) \quad (15)$$

Hewitt and Hall-Taylor (1970) have proposed the expression for (δ/d) for fully

developed flow in a tube as:

$$\frac{\delta}{d} = 43.5 Re^{-0.8} Sc^{-1/3} \quad (16)$$

Equations 16 and 15 can be employed to correct h' for the observed enhancement. In this study the corrected h' has been designated by h'' .

MATERIALS AND METHODS

Experimental Apparatus

The experimental apparatus used in this work consisted of a condensing block (Fig. 2) for the measurement of the surface temperature (T_w) and total heat flux q_t . The purpose of fins was to increase the surface area in contact with the cooling medium at the cold end. The flanged block was fabricated out of stainless steel (AINS 316; 25.4 mm in diameter and 38.1 mm deep). Four 1.6-mm-diameter holes were drilled on the same dimensional plane, perpendicular and up to the center. A copper-constantan thermocouple was soldered at the bottom of each hole. The condensation surface, a retort pouch material (12 μ m Polyester/12 μ m Aluminium/70 μ m Poly-

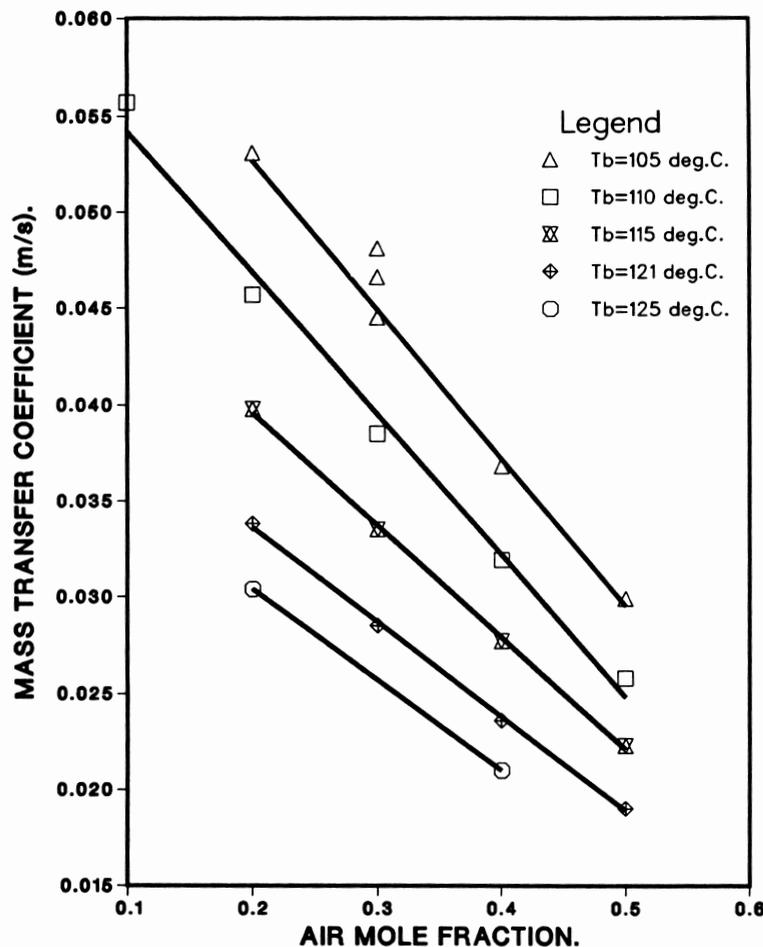


Figure 5. Mass transfer coefficient vs. air mole fraction.

olefin, was glued on the condenser block by a small amount of high conductivity adhesive (3M EC-1711). As shown in Fig. 3, the block was inserted in a tubular condensation chamber (0.1541 m diameter and 1.02 m long) 0.75 m from the steam/air inlet end of the chamber. The condensation block was insulated by Teflon, the thickness of which was such that radial heat loss was found negligible. A flow sheet showing the full experimental apparatus is presented in Fig. 4.

Experimental Procedure

The desired steam/air mixture was generated by controlling the temperature and pressure inside the retort (Fig. 4) using a Taylor Fulscope Recording Controller. The amount of air entering the system was controlled and metered through a variable flowmeter and was then mixed with the steam before entering the retort through a spreader to impart homogeneity. The homogeneous mixture was then introduced into the condensation chamber through a venting manifold located at the top of the retort. The pressures and temperatures in the chamber and the retort

were monitored separately as there were changes in mixture air content between the retort and the condensation chamber due to the loss of condensed steam, in some cases increases up to 0.1 air mole fraction were observed. The chamber and the rest of the system circuitry were insulated with one-inch-thick fibreglass to minimize this loss.

The experiments were carried out to measure T_w and q_t through the condensation block which was done at five temperature levels (105, 110, 115, 121 and 125°C) often used for thermal processing in the food industry. Air content was varied between 0.1 and 0.5 by volume and air flow into the system ranged between 4.72×10^{-3} to 14.5×10^{-3} m³/sec. The experimental design consisted of a total of 41 runs. Each run consisted of six experiments with varying cooling water flow-rates (36.5, 45.6, 52.7, 78.5 and 78.7 mL/sec) past the fined end of the block, set by a calibrated valve. By varying the cooling water flow it was possible to vary the heat flux through the block. After setting the temperature and pressure controller at the desired values and allowing the system to achieve steady state, thermocouple tem-

perature readings were recorded at 2-min intervals and averaged by means of a Kaye Ramp II Scanner/Processor system. After the cooling water flow-rate was changed to the next desired value, the system was allowed to stabilize and the thermocouple reading was recorded. The procedure was repeated until all desired flow-rates were obtained.

Calculations

The air mole fraction was calculated using Dalton's Law of Partial Pressures:

$$x_a = 1 - \Pi_{vb}/\Pi$$

and the accuracy of pressure measurements was ± 0.0345 bars (± 0.5 psi), resulting into a 4.5% uncertainty in x_a .

At steady-state conditions, with negligible heat loss through the Teflon block insulation, uni-dimensional heat conduction through the block was assumed as it was not a source of significant error ($q \leq 1\%$ q_t). Therefore, q_t was calculated by Fourier's law of heat conduction as follows:

$$q_t = -\lambda_{xx} \cdot dT/dy \quad (17)$$

where dT/dy is the thermal gradient along the block and was established by regression analysis of the thermocouple temperature measurements with respect to distance. The coefficient of correlation for all the data was equal to or better than 0.99. T_w was obtained by extrapolation.

The properties Π_v , ρ , η , λ , C_p and h_{fg} for steam were obtained from tabulations by Schmidt (1979) and the properties ρ , C_p , η and λ for air were taken from National Bureau of Standards tables (1955) and air was treated as a perfect gas. The thermal conductivities of the mixtures were calculated using the equation of Lindsay and Bromley (1950). Typically errors in the calculated values do not exceed 1–3%. Values of ρ and C_p for the mixtures were calculated by standard additive procedures. D was calculated by Gilliland's correlation found in Sherwood and Rigford (1952). The mixture viscosities were calculated using the approximation by Wilke (1950), for which average deviations between calculated and experimental values have been reported to be 1.9%.

Since the conditions at the liquid-vapor interface were unknown, it was assumed that T_i corresponded to the saturation temperature equivalent to the partial pressure of the vapor at the interface. With the knowledge of T_w , T_b and q_t an iterative procedure was employed to estimate T_i ($T_w < T_i < T_b$) such that Eq. 13 was satisfied. The evaluation of h was based on Hansen's

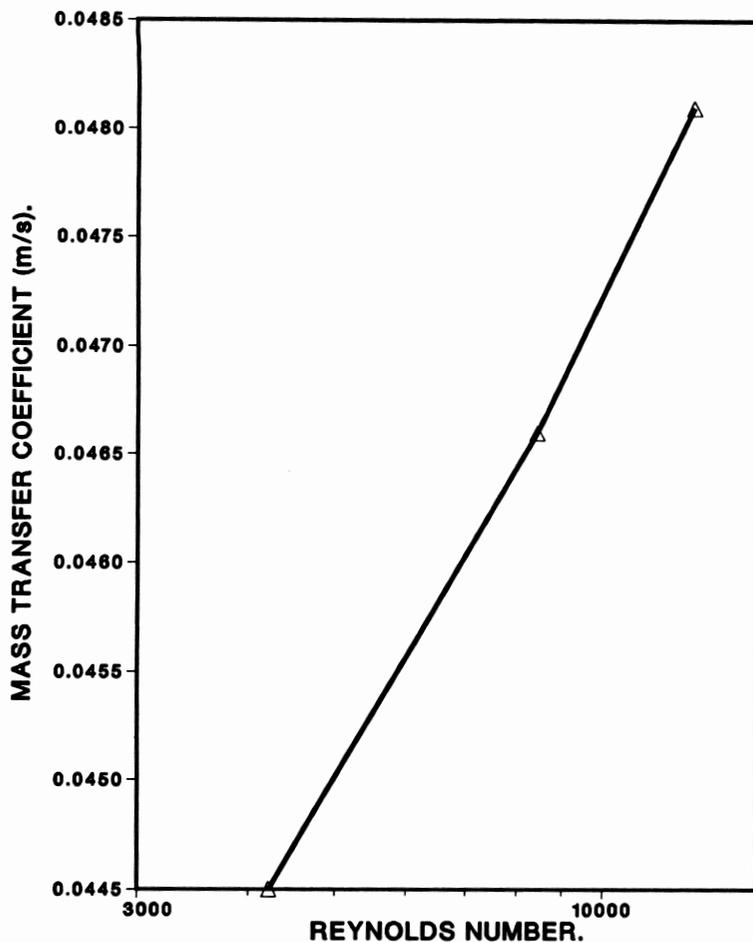


Figure 6. Mass transfer coefficient vs. Reynolds number for a steam/air temperature of 105°C and air mole fraction of 0.3.

equation recommended by Knudsen in Perry and Chilton (1973).

RESULTS AND DISCUSSION

The sensible heat transfer rate through the diffusion layer has been found to be a very small component of the total heat transfer to the interface. The fraction q_{fg}/q_t ranges between 0.996 for the lowest x_a and 0.950 for the highest x_a . Therefore, the discussion of the influence of the process parameters on h'' has been considered of small significance.

Reynolds Number and Bulk Temperature Effects

Empirical relationships were sought to show the dependence of K on T_b and x_a . The data obtained have been plotted (K against x_a for constant T_b), and plots of this type are shown in Fig. 5. For a given T_b , K can be approximated by a linear function of x_a . However, for the plot of $T_b = 105^\circ\text{C}$, a number of points, which correspond to higher Re in the region of 8 400–12 700 lie off the plot. This confirms the well-known association of high mass transfer coefficients (K) with high Reynolds numbers

(Re). Plotting the values of K against Re for $x_a = 0.3$ and $T_b = 105$ further confirms the influence of Re on K as shown in Fig. 6. A relationship between K and x_a that does not take Re into account has been found inadequate.

Mass Transfer Data Corrections

The Chilton–Colburn (1934) mass transfer analogy which agrees well with experimental data is:

$$\frac{Sh}{Re} Sc^{-1/3} = \Psi(Re) = J_d \quad (18)$$

Figure 7 shows a plot of J_d against Re , and these results are in qualitative agreement with the experimental results of Gilliland and Sherwood (1934) and Linton and Sherwood (1950). However, their experiments were conducted in vertical so called wetted-wall towers. All their data are empirically correlated by the following equation:

$$Sh = 0.023 Re^{0.83} Sc^{0.44} = C Re^m (Sc)^n. \quad (19)$$

By taking the logarithm of both sides of Eq. 19 and performing a multiple regression analysis using a Triangular Regres-

sion Package (Le and Tenisci 1978), the parameters C , m and n were estimated for the data in this study. And they were found to be equal to 205.7, 0.0746 and 0.0526 with standard deviations (roots of variances) of 1.0, 0.0021 and 0.0113, respectively. The correlation coefficient was found to be equal to 0.9769.

For condensation on the underside of a horizontal surface, Gerstmann and Griffith (1957) observed enhancement of heat and mass transfer coefficients and attributed this to the disturbances they observed on the condensate film surface as waves. However, the results of other investigators (Stern and Volta 1968; Onda et al. 1970; Deo and Webb 1983) have confirmed the absence of waves during condensation of vapor in presence of non-condensables for configurations where the condensate is flowing on a vertical surface. Their results were in close agreement with the film theory predictions using correlations of heat and mass transfer coefficients which did not take into account any enhancement. Therefore, given the horizontal configuration employed in this work, the cause of the quantitative disparity between this study and those that employed the vertical configuration (Gilliland and Sherwood 1934; Linton and Sherwood 1950) has been attributed to the disturbances on the condensate film surface.

The applicability of the proposed correlation in this study has some disadvantages that the influence of x_a is implied indirectly through the dimensionless parameters that are x_a dependent.

Bulk Temperature and ΔT Effects on the Lumped Heat Transfer Coefficient

There are various methods employed to estimate the sterilizing capacity of heat processes (Stumbo 1973). These evaluations are based on heat penetration measurements at a location or locations of interest in a packaged food product.

Pflug et al. (1965) indicated that the rate of conduction heating of a food product in a container is a function of the geometry of the container, the physio-thermal properties of the food, the heat transfer characteristics of the container and the heating medium. They further indicated that in a system for processing food where the overall heat transfer coefficient (H) is large enough such that the Biot number (Bi) is greater than 10, the heating rate of the product becomes independent of H , and is a fraction of the thermal properties of the food and the container dimensions. This implies that for situations where $Bi \gg 10$, the internal resistance to heat penetration is much greater than the external resis-

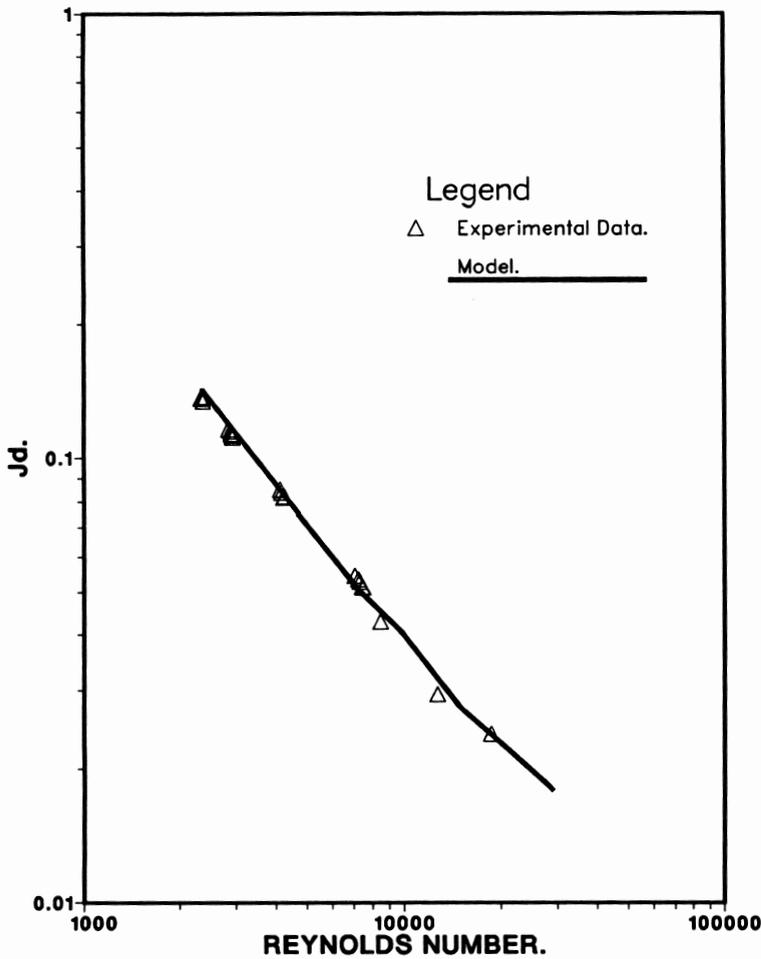


Figure 7. J_d vs. Reynolds number.

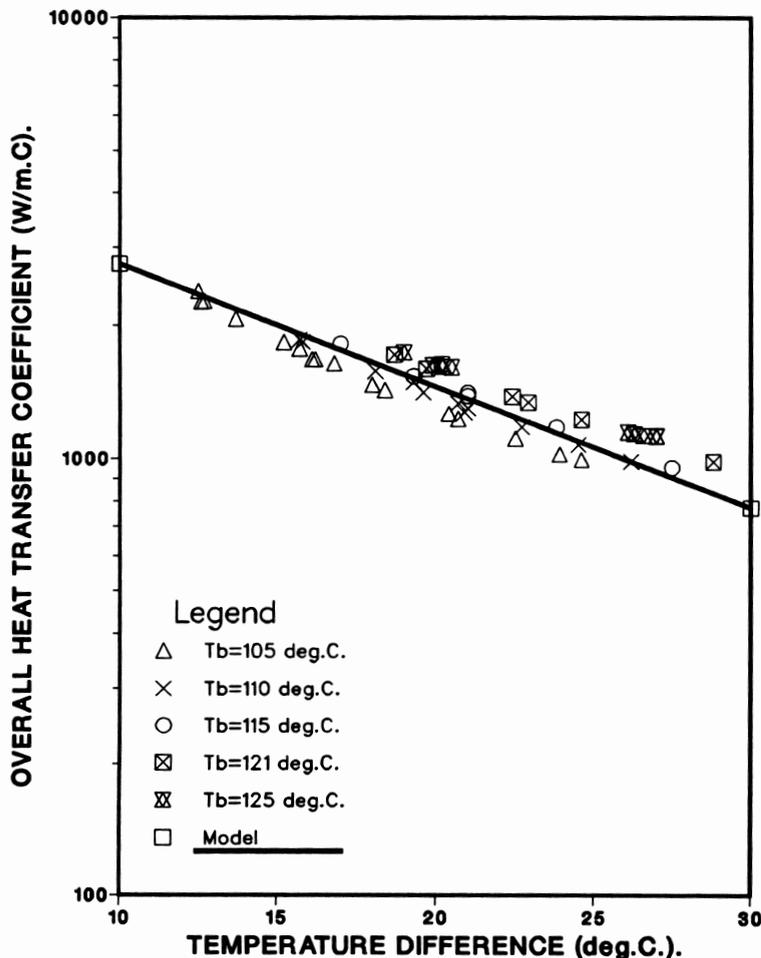


Figure 8. Overall heat transfer coefficient vs. block surface temperature.

tance and, therefore, the surface temperature (T_w) of the food can be assumed to be equal to the heating medium temperature (T_b). For situations where $Bi \ll 10$, the internal resistance to heat penetrations would be considerably smaller than the surface resistance such that the lumped parameter approach can be used to describe the heating of the food product. In this case there would be an appreciable temperature difference (ΔT) between the food product surface and the heating medium at the beginning of the process, which would reduce gradually as the food product average temperature approaches the process target temperature.

The effects of steam/air T_b , T_w and ΔT on the heat penetration rate are desirable for thermal process evaluation. By lumping up the thermal resistances (refer to Figs. 1 and 2) due to the retort pouch material, the condensate and the gas layer into a combined thermal resistance ($1/H$), relationships between H , ΔT and T_w were sought. As shown in Figs. 8 and 9, H can be approximated by logarithmic functions of T_w and ΔT , in the region considered, of the forms:

$$H = \alpha_1 e^{\alpha_2 T_w} \quad (20)$$

$$H = \beta_1 e^{\beta_2 \Delta T} \quad (21)$$

where α_1 , α_2 , β_1 and β_2 are parameters. The data shown in Fig. 8 were pooled together to estimate the values of β_1 and β_2 and they were found equal to 5200.2 and -0.06356 with standard deviations of 105.1 and 0.0010, respectively. The estimates of α_1 and α_2 are shown in Table I.

However, it should be emphasized that these data can only be useful for situations where $Bi \gg 10$, for cases with $Bi \ll 10$, T_w rises suddenly so that ΔT is very small for the largest part of the process period.

CONCLUSIONS

A systematic experimental investigation has been carried out to study the influence of process parameters (T_b , x_a and Re) on the heat transfer in the vapor-liquid region during film condensation on the underside of a horizontal surface. The results suggest the following:

(i) The fraction of the sensible heat transferred to the interface is a very small component of the total heat transferred making h'' an insignificant variable.

(ii) An increase in x_a results in a decrease of both h'' and K and an increase in T_b and Re results in an increase in both h'' and K and the heat and mass transfer parameters are well correlated by an equation of the form $Sh = C Re^m (Sc)^n$.

(iii) Heat and mass transfer correla-

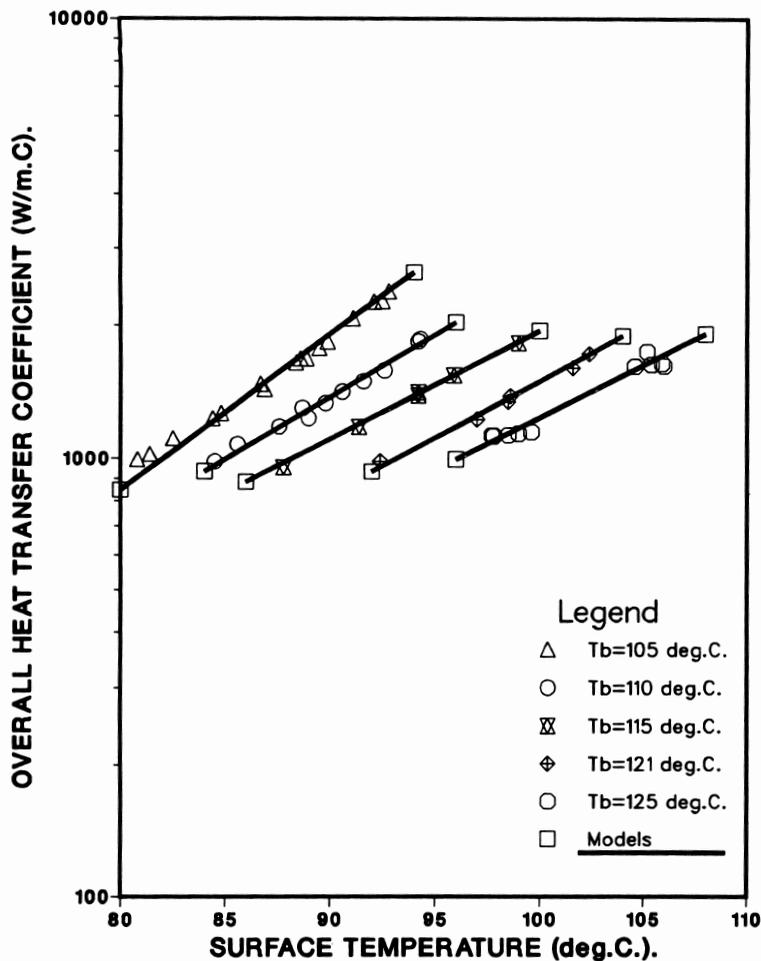


Figure 9. Overall heat transfer coefficient vs. temperature difference (ΔT) between the block surface and the bulk steam/air mixtures.

TABLE 1. ESTIMATED PARAMETERS, α_1 AND α_2

Temp. (°C)	α_1 (W/m ² C)	α_2 (1/C°)	SD† of α_1	SD of α_2
105	1.2686	0.0813	0.1157	0.0010
110	4.0383	0.0648	0.4896	0.0013
115	6.9137	0.0564	0.7155	0.0011
121	4.1374	0.0589	0.6526	0.0016
125	5.3205	0.0545	1.0895	0.0020

†SD is the standard deviation.

tions which do not take into account any enhancement due to disturbances on the condensate film surface should not be used in predicting heat and mass transfer coefficients.

(iv) For a given steam/air temperature, the lumped heat transfer coefficient can be determined empirically from exponential functions of T_b or ΔT .

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NOMENCLATURE

c	= total molar concentration (mol/mol)
C	= constant
C_p	= specific heat capacity (J/(kg·K))QL
d	= pipe diameter (m)
D	= molar diffusion coefficient (m ² /sec)
f_g	= interfacial friction factors
f_i	= friction factor based on air alone
H	= overall heat transfer coefficient (W/(m ² ·K))QL
h, h', h''	= sensible heat transfer coefficient (W/(m ² ·K))QL
h_{fg}	= latent heat of vaporization (J/kg)

J	= drift flux of bulk fluid towards interface (m/sec)
K	= mass transfer coefficient (m/sec)
m	= constant
n	= constant
q_t, q_{fg}, q_c	= heat flux (W/m ²)
v	= linear velocity (m/sec)
T	= temperature (K)
x	= mole fraction
y	= distance measured from a reference point (m)

Dimensionless Groups

Bi	= hy/λ , Biot number
Nu	= hd/λ , Nusselt number
Pr	= $\eta C_p/\lambda$, Prandtl number
Re	= $\rho v d/\eta$, Reynolds number
Sc	= $\eta/\rho D$, Schmidt number
Sh	= Kd/D , Sherwood number

Greek symbols

α, β	= parameters
δ	= thickness of diffusion layer (m)
ρ	= density (kg/m ³)
ψ	= designates function
Π	= pressure (bars)
λ	= thermal conductivity (W/(m·K))
η	= viscosity (kg/m.s)

Subscripts

a	= air
b	= bulk
i	= interface
sa	= steam/air
ss	= stainless steel
m	= log mean
v	= vapor
w	= wall

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