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DIRECT CONTACT CONDENSATION WITH A MULTI-COMPONENT MIXTURE OF NON CONDENSABLE GASES

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NOMENCLATURE*

C_p,	specific heat of condensate;
D_j,	"effective binary diffusivity" of the j constituent in pure vapour;
k,	condensate thermal conductivity;
$\dot{m}$,	local condensation mass flux;
M,	molecular weight;
Pr'	condensate Prandtl number;
P^*,	$Pr'(\mu\rho'/\mu'\rho)$;
Sc_j,	$\mu/\rho D_j$;
T,	local temperature;
T^*,	$(T - T_0)/(T_\infty - T_0)$;
u_0,	constant free stream velocity in [1];
w,	dimensionless mass fraction of the noncondensable gas;
w^*,	w/w_∞;
x,	longitudinal coordinate, equation (4); mole fraction, equation (3);
α',	condensate thermal diffusivity;
λ,	latent heat of vaporization;
λ^*,	$\lambda\rho/C'_p\rho'(T_\infty - T_0)$;
μ,	absolute viscosity;
ρ,	density;
η,	heat-transfer efficiency.

Subscripts

0,	at $x = 0$;
∞,	in the bulk;
i,	at the interface;
j,	for noncondensable species j;
v,	condensing vapour;
',	liquid.

THE EFFECT of noncondensable gas on condensation was mostly studied for the case where a single noncondensable phase is present in the condensing vapour [1–5]. In the system of air where N_2 and O_2 are the main constituents a single phase analysis is indeed a reasonable approach because of the similarity in the transfer characteristics of oxygen and nitrogen ($Sc_{O_2-H_2O} = 0{\cdot}59$, $Sc_{N_2-H_2O} = 0{\cdot}61$ at atmospheric pressure). If, however, the vapour contains non similar gases a multi-component analysis is necessary.

In general the presence of multi-noncondensable phases will cause considerable complication in the analysis. However, for the particular case described in [1] (and near the "leading edge") this extension is straightforward. The purpose of this note is to draw attention to such a possibility.

In this probelm we are interested in the direct contact condensation rates of vapour on a laminar stream in the presence of multicomponent noncondensable gases. Our aim is to predict the concentrations distribution at the vapour stream interface for the noncondensable species and their total effect on the heat-transfer efficiency. Since the concentration of the noncondensable gases is generally small the diffusion coefficient of each component is very close to its value for the binary mixture with the pure vapour [6]. It can, therefore, be estimated by the Gilliland's method [7]. Under this restriction and considering the additional assumptions made in [1] one may notice that the dimensionless diffusion equation describing the vapour gas boundary layer will be identical for each non-condensable species, namely,

$$u^* \frac{\partial w_j^*}{\partial x^*} + v^* \frac{\partial w_j^*}{\partial y^*} = \frac{P^*}{Sc_j} \frac{\partial^2 w_j^*}{\partial y^{*2}} \qquad (1)$$

also the non-permeable conditions of the noncondensable gases at the interface

$$-\frac{P^*}{Sc_j} \frac{\partial w_j^*}{\partial y^*} + w_j^* v^* = 0 \qquad (2)$$

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* The nomenclature given here is sufficient for practical purposes. For details see [1].

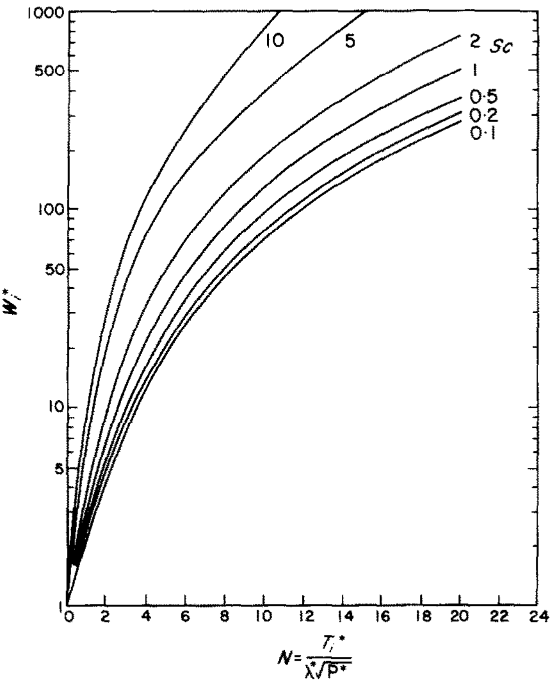

FIG. 1. Generalized curves of w_i^* as a function of N.

is valid for every species. The general solution shown in Fig. 1† can therefore be applied for the system containing a multi-component mixture of noncondensable constituents.

The following procedure should be adopted for calculating the interfacial concentrations distribution, temperature and condensation rate. We guess an interfacial temperature T_i, find the corresponding saturation partial pressure of the condensing vapour ($p - T$ diagram) and calculate the mole fraction $x_v = p_v/p_{\text{total}}$. From Fig. 1 we compute w_j^* at the interface for each noncondensable species according to its Schmidt number for binary mixture. The mole fraction is then computed from:

$$x_j = \frac{w_j^* w_{\infty,j}/M_j}{\sum \frac{w_j^* w_{\infty,j}}{M_j} + \frac{w_v}{M_v}} \tag{3}$$

By trial and error we find the correct T_i and w_j which satisfy $\sum x_j + x_v = 1$. When T_i is known the local condensation mass flux is calculated from:

$$\dot{m} = \frac{k(T_0 - T_i)}{\lambda} \sqrt{\frac{u_0}{\pi \alpha' x}}\,. \tag{4}$$

The heat-transfer efficiency η defined as the ratio between the actual flux divided by the maximum flux obtained in the condensation of pure vapour ($w_j = 0$) is given by:

$$\eta = \frac{T_i - T_0}{T_\infty - T_0} = T_i^*. \tag{5}$$

As an example we consider the condensation of water vapour at 760 mmHg and 55 mmHg on a laminar water stream with $w_\infty = 0{\cdot}001$ mass fraction of CO_2 and H_2. The temperature difference between the saturated vapour and the stream is 20°C. The following table summarizes other physical data and results obtained after few trials:

As would be expected the relative concentration (w_j^*) of the lighter species is smaller than the heavier one owing to the difference in the diffusion coefficients. It should be noted, however, that the difference in the interfacial concentration between H_2 and CO_2 is somewhat less than might be expected on the basis of the difference between their diffusion coefficients. Namely, $D_{H_2-H_2O}/D_{CO_2-H_2O}$ is about 3·6 and $w_{i,CO_2}/w_{i,H_2}$ is only about 1·5. One may also notice that the effect of the noncondensable gases is accentuated at lower pressures.

Table 1

	λ^*	p^*	Sc_j		w_i	$T_i - T_0$(°C)	η(%)
Pressure condensation	0·01	135	H_2 CO_2	0·23 0·84	0·016 0·027	15·8	79
Vacuum condensation	0·0015	1370	H_2 CO_2	0·21 0·74	0·068 0·100	10·0	50

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† This figure is reported in [1] and is given here for convenience.

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EXPERIMENTELLE ERGEBNISSE MIT EINEM DOCHTFREIEN ZENTRIFUGAL-WÄRMEROHR

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EINFÜHRUNG

Die Kühlung rotierender Körper bietet oft Schwierigkeiten, wenn diese aus konstruktiven Gründen gegen die Umgebung abgekapselt sind. Ein Beispiel hierfür sind die Rotoren elektrischer Maschinen. Experimente zur Umgehung dieser Schwierigkeiten werden nachfolgend beschrieben.

VERSUCHSAUFBAU

Nur die beiden Wellenenden verbinden den Rotor konstruktiv mit der Umgebung einer elektrischen Maschine. Über diese Wellenenden könnte man also zusätzlich Rotorverlustwärme unter Umgehung des Motor-Innenraums direkt an die Aussenluft ableiten, wenn die Wärmeleitfähigkeit der Welle entsprechend erhöht wird. Hierzu bietet sich z.B. die Ausgestaltung der Welle als Wärmerohr an [1].

Zu diesem Zweck wurde die Welle eines Versuchsmotors (4 kW tefc Käfigläufer-Asynchronmaschine, 1500 U/min) als abgeschlossenes Rohr ausgeführt. Auf die für ein Wärmerohr sonst wesentliche Auskleidung des Rohrs mit einem Docht kapillarer Struktur wurde verzichtet. Dieser Docht soll einerseits die Arbeitsflüssigkeit des Wärmerohrs an der Wand festhalten und somit gegen dem Dampfraum abtrennen, andererseits sorgt er durch die Wirkung einer Kapillardruckdifferenz für den Rückfluss des Kondensats vom Kondenser zum Verdampfer. Diese beiden Funktionen können aber bei einem schnell um die Längsachse rotierenden Wärmerohr durch die Zentrifugalkraft übernommen werden. Diese presst die Flüssigkeit an die Wand und sorgt für den Rückfluss des Kondensats, denn dieses kann sich auf der Kondenserseite nicht anhäufen.

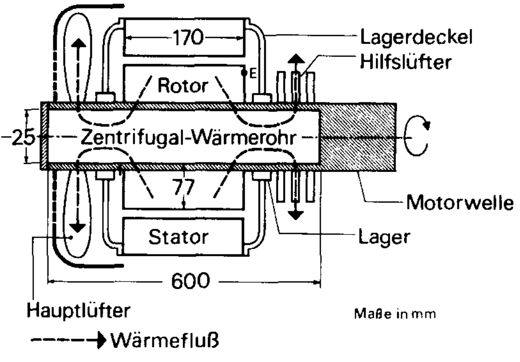

Abb. 1. Übersichtsbild des Versuchsmotors für Rotorkühlung mit Zentrifugal-Wärmerohr.

Mit dem Docht entfällt auch der Strömungswiderstand in den Kapillaren und die Konstruktion wird wesentlich