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SHORTER COMMUNICATIONS

ELECTROHYDRODYNAMIC HEAT PIPES

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NOMENCLATURE

E, electric field strength;
E_b, breakdown electric field strength of vapor phase;
g, acceleration of gravity (9·81 m/s^2);
L, heat pipe length;
$\dot{m}$, mass flow rate;
s, electrode spacing;
T^e, electric surface traction;
V, heat pipe operating voltage;
w, EHD flow structure width;
ε, dielectric permittivity;
μ, dynamic viscosity;
ρ, mass density;
ϕ, inclination angle of heat pipe.

Subscripts
a, adiabatic;
cond, condenser;
eff, effective;
evap, evaporator;
l, liquid;
max, maximum;
v, vapor.

1. INTRODUCTION

A HEAT pipe of radical design is proposed here, which substitutes polarization electrohydrodynamic force effects for capillarity to collect, guide, and pump the condensate liquid phase [1]. The basic concept is to replace the capillary wick of a "conventional" heat pipe with an electrode structure, which (i) orients the liquid phase inventory of working fluid in a heat pipe, (ii) guides this liquid as it flows axially from the cooled end (the condenser) to the heated end (the evaporator), and (iii) provides a net liquid "pumping" force to ensure true heat pipe operation.

The electrohydrodynamic heat pipe discussed here is restricted to the use of insulating dielectric liquids as working fluids. Because of the relatively poor thermal transport properties of these liquids (viz. low surface tension and low thermal conductivity), capillary heat pipes using these liquids have not been high performance devices. The employment of the electrohydrodynamic concept should enhance this performance, and help to fill the performance gap which exists in the temperature range from ~ 250°F to ~ 750°F for "conventional" capillary heat pipes.

The electrohydrodynamic heat pipe falls into a general class of dielectric liquid management systems, considered originally for propellant orientation in zero-gravity environments [2]. More recently, dielectric liquid flow structures and their possible applications have been studied [3–5]. Further, work has been done on electric field-enhanced ebullient [6] and condensation [7] heat transfer. In all the applications cited [2–7], polarization electrohydrodynamical forces play either a dominant or at least significant role. The electrohydrodynamic heat pipe represents an attractive marriage of these dielectric liquid management and heat transfer enhancement schemes.

2. THEORY OF ELECTROHYDRODYNAMIC HEAT PIPES

2.1 *Description*

The possibility of replacing the wick of a capillary heat pipe with an electrode structure was suggested at least as early as 1970 by Jones [4]. A preliminary, analytically tractible design for such a heat pipe is shown in Fig. 1. It consists of a thin-walled tube of aluminum or some other good electrical and thermal conductor, with end caps made of electrically insulating material, such as plexiglas. Stretched taut and secured in some fashion by the end caps is a thin ribbon electrode, spaced a distance s from the inside of the tube and running parallel to its axis. Upon application of a sufficient voltage, the insulating dielectric liquid in the heat pipe collects in the high electric field region shown, bridging the area between the ribbon and the grounded tube. Liquid communication between the heated and cooled ends is thus established.

2.2 *Principles of operation*

In operation, heating and the resultant evaporation of liquid causes a net recession of the dielectric liquid interface at the evaporator end of the tube, whereas cooling and the resultant condensation causes a net outward bulging of the dielectric liquid interface at the condenser end. Refer to

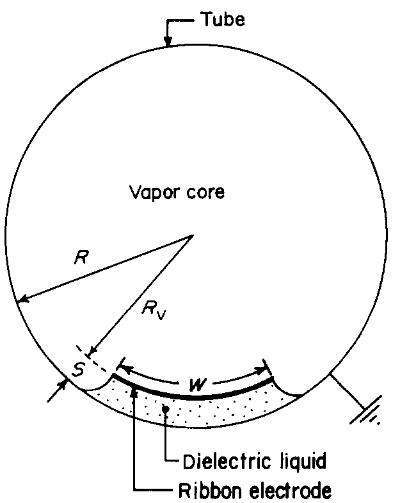

FIG. 1. Prototype electrohydrodynamic heat pipe with ribbon electrode flow structure, shown in cross section.

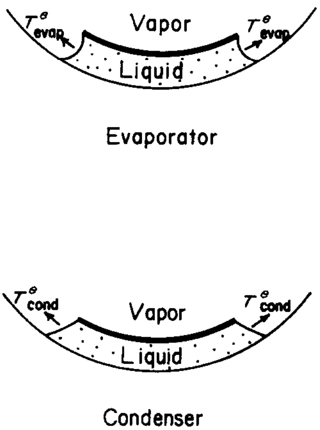

FIG. 2. Cross-sectional views of electrohydrodynamic flow structure at evaporator and condenser ends of heat pipe.

Fig. 2. The electromechanical surface tractions [8] T^e_{evap} and T^e_{cond}, acting normal to these surfaces are then unequalized, resulting in a negative pressure gradient from the condenser to the evaporator. Net liquid flow from the condenser to the evaporator occurs. The vapor flow in the central core balances this liquid flow, carrying thermal energy stored as latent heat. A true heat pipe mode of operation is thus achieved.

As in "conventional" capillary heat pipes, the pumping mechanism of an electrohydrodynamic heat pipe depends ultimately upon the temperature difference between the two ends. The polarization force itself is not capable of pumping dielectric liquid [5]; rather, it establishes hydrostatic equilibrium, which the evaporation and condensation processes unbalance. The liquid flows continuously to reestablish this equilibrium. Electrohydrodynamic heat pipe operational principles are thus in close analogy to those of capillary devices.

Practical electrohydrodynamic heat pipes may require the use of conventional wicking structures for more effective distribution of liquid over the evaporator surface for efficient vaporization heat transfer. For devices similar to that shown in Fig. 1, such a structure might take the form of circumferential threaded grooves [1].

2.3 *Maximum flow—laminar case*

If a laminar Poiseuille flow model is chosen to approximate the liquid flow between the ribbon electrode and the tube wall, with all vapor pressure losses neglected, the flow rate for the structure is

$$\dot{m} = \frac{n\rho_l\, ws^3}{12\mu_l\, L_{eff}}(\Delta T^e - \rho_l gL \sin\phi), \tag{1}$$

where $L_{eff} = (2L_a + L_{evap} + L_{cond})/2$ is the effective heat pipe length. The term ΔT^e is the dielectric pumping head, analogous to the capillary pumping head.

$$\Delta T^e = T^e_{evap} - T^e_{cond} = \frac{(\varepsilon_l - \varepsilon_v)\, E^2_{evap}}{2} - \frac{(\varepsilon_l - \varepsilon_v)\, E^2_{cond}}{2}. \tag{2}$$

E_{evap} and E_{cond} are the (essentially tangential) electric field magnitudes experienced by the liquid interfaces at the evaporator and condenser. For normal operation, $E_{evap} > E_{cond}$, meaning that the liquid bulges out into the fringing field at the condenser and recedes into the high electric field region between the plates at the evaporator.

Under proper conditions of fluid inventory, the maximum dielectric head is:

$$\Delta T^e_{max} = \frac{(\varepsilon_l - \varepsilon_v)\, E_0^2}{2}, \tag{3}$$

where $E_0 \simeq V/s$ is the magnitude of the essentially uniform field between the ribbon electrode and the inner wall of the heat pipe tube. The maximum, so-called wick-limited mass flow rate for the flow structure may be calculated by substitution of ΔT^e_{max} for ΔT^e in equation (1). Note that a capillary distribution structure in the evaporator (or condenser), such as threaded circumferential grooves, imposes an additional constraint upon the total flow, which may result in a wicking limit lower than that defined by equations (1) and (3).

2.4 *Operating limits*

The relationship of vapor electrical breakdown to the pumping limit of the dielectric pumping head is best illustrated by considering the vertical static pumping height of various candidate dielectric liquids. This maximum height is [1]

$$X_{\max} \simeq \frac{(\varepsilon_l - \varepsilon_v) E_b^2}{2\rho_l g}, \tag{4}$$

where E_b is the vapor breakdown electric field strength of the working fluid. Note that equation (4) follows directly from equations (1) and (3), replacing $L \sin \phi$ by $X_{\max}$. Clearly if $L \sin \phi > X_{\max}$ the electrohydrodynamic heat pipe will not function at all. Table 1 tabulates $X_{\max}$ for several liquids at their atmospheric boiling points. Since E_b and thus $X_{\max}$ are relatively strong increasing functions of vapor pressure, the advisability of operating electrohydrodynamic heat pipes at internal pressures of several atmospheres is clear.

Besides the fundamental operational limit imposed by the balance of hydrostatic and hydrodynamic pressure losses with the dielectric pumping forces, additional limits for the electrohydrodynamic heat pipe have been identified [1]. Included are an entrainment limit modified by electric field forces, the conventional sonic limit, and an electrohydrodynamic surface wave celerity limit (which constrains the liquid flow velocity [4]).

3. SPECIAL FEATURES

The electrohydrodynamic heat pipe offers some unusual features which may prove advantageous.

3.1 *Electrohydrodynamic heat transfer enhancement*

A properly designed electrode structure in an electrohydrodynamic heat pipe may promote evaporative heat transfer at the heated end and condensation at the cooled end, and thereby improve the relatively poor heat transfer coefficients realized with dielectric liquids. Boiling heat transfer is enhanced by a non-uniform electric field because of the tendency of the polarization force to agitate the non-uniformly heated liquid and to expel vapor bubbles from high electric field regions [6]. Condensation heat transfer is likewise enhanced by a non-uniform electric field because the condensate is collected off cooled surfaces into high field regions promoting pseudo-dropwise condensation [7].

3.2 *Priming*

Many high performance artery heat pipes exhibit an unfortunate tendency toward unreliable priming. The electrohydrodynamic heat pipe has no such drawback; priming occurs *with 100 per cent reliability*, by application of the operating voltage. This is because the electrode flow structures do not rely on uncertain wick-fluid wetting conditions.

3.3 *Bubble ejection from flow structure*

The same dielectrophoretic force which promotes boiling heat transfer in the evaporator of an EHD heat pipe should also help to eject vapor bubbles caught in the axial flow structure. These bubbles can affect the performance of capillary arteries, but they should be less of a problem in electrohydrodynamic flow structures because of this automatic ejection mechanism.

3.4 *Liquid friction factor*

The net viscous friction of an electrohydrodynamic flow structure will generally be very small in comparison to that of a capillary wick of equal cross section. The result of this should be the elimination of axial flow resistance as a factor in wick-limited operation for dielectric fluid heat pipes.

3.5 *Voltage-controlled heat pipe*

The electrohydrodynamic heat pipe enters the milieu of heat pipe control with a capability of direct voltage control of the pumping mechanism, loosely analogous to varying the surface tension in a capillary device. Using Marcus's classification [9], it should be called a *liquid flow control* technique, but not of any type previously envisioned. The liquid flow path is not obstructed by changing the voltage, rather the pumping head itself is controlled. For constant temperature evaporators, this approach leads to thermal control, continuous from *very close* to zero to maximum throughput. The purely electrical control linkage holds clear advantage.

3.6 *Other possibilities*

It may be possible to use electric field force effects to prime capillary arteries and/or to expel trapped vapor bubbles. Abu-Romia has recently suggested the use of electro-osmotic forces to assist in the pumping of liquid in a capillary heat pipe [10]. Another possible electrohydrodynamic pumping mechanism worth consideration might be the ion–drag effect in dielectric liquids [11].

Table 1. Maximum dielectric liquid static height of rise for several insulating dielectric liquids at atmospheric boiling point

Fluid	$X_{\max}$(m)	E_b (kV/m)	T_b (deg K)
Freon-113	0·098	15600	391
Freon E-3	0·189	19400*	425
Dowtherm A	~0·48	~20000*	530

*Estimated values.

4. CONCLUSION

The newly proposed electrohydrodynamic heat pipe offers interesting prospects. Preliminary considerations indicate the possibility of an enhanced status for dielectric fluid heat pipes as high performance devices, thus filling the performance gap in the temperature range between water and liquid metal heat pipes.

Against this and the other features discussed in Section 3 must be weighed the disadvantages of the greater complexity of the electrohydrodynamic heat pipe, with its attendant reliability questions. A high voltage, low current power supply is required. The need for high voltage insulation in reases fabrication costs. Further, the long-term degradation of dielectric fluids in the presence of corona or intermittent arcing may be a significant factor.

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TURBULENT TRANSPORT COEFFICIENTS FOR COMPRESSIBLE HETEROGENEOUS MIXING

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I. INTRODUCTION

Within the past decade considerable amount of work has been done in the area of turbulent wakes and free mixing of coaxial jets due to the large interest in the wakes of high speed missiles and re-entry vehicles, various base injection techniques, cooled plasma jets, decaying exhaust plumes, supersonic combustion, etc. However, the problem is far from being solved yet. One of the main difficulties stems from the lack of information about the dependence of the turbulent transport coefficients on the flow properties. Although many models for the turbulent diffusivity [1] and viscosity [1–5] have been proposed, none has been found to be satisfactory so far. Therefore, an effort is made here to obtain theoretically the expressions for the turbulent transport coefficients on the basis of the general behavior characteristics of flow variables in the wake.

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