

Fluid Selection and Property Effects in Single- and Two-Phase Immersion Cooling

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Abstract—The governing equations for liquid and two-phase heat transfer are used to derive fluid figures of merit (FOM's) for forced and natural convection, boiling incipience, and critical heat flux in both pool and flow boiling modes. These FOM's can be used to evaluate and compare the thermal performance of several candidate immersion cooling fluids. In addition, the governing equations are used to determine the sensitivities of thermal performance to fluid property values.

NOMENCLATURE

CHF	Critical heat flux, W/m^2 .
C_p	Liquid specific heat, $J/kg \cdot K$.
D	Hydraulic diameter, m.
FOM	Figure of merit.
g	Acceleration of gravity, m/s^2 .
Gr	Grashof number, dimensionless.
h	Heat transfer coefficient, $W/m^2 \cdot K$.
h_{fg}	Latent heat of vaporization, J/kg .
k	Thermal conductivity, $W/m \cdot K$.
L	Heated length or characteristic length, m.
L'	Dimensionless length.
N_{uc}	Nucleate parameter, dimensionless.
P	Pressure, N/m^2 .
Pr	Prandtl number, dimensionless.
q''	Surface heat flux, W/m^2 .
R	Radius, m.
Re	Reynolds number, dimensionless.
R_0	Cavity radius, m.
T	Temperature, K.
ΔT_{sub}	Subcooling of liquid, K.
$(\Delta T_s)_i$	Incipience temperature superheat, K.
u	Bulk velocity, m/s.
V	Characteristic velocity, m/s.

Greek Symbols

β	Contact angle, and thermal expansion coefficient, K^{-1} .
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2ϕ	Conical angle of cavity.
σ	Surface tension, N/m.
ρ_f	Liquid density, kg/m^3 .
ρ_g	Vapor density, kg/m^3 .
μ	Dynamic viscosity, $kg/m \cdot s$.

Subscripts

crit	Property at the critical point.
f	Liquid (fluid).
FB	Flow boiling.
i	At incipience.
l	Liquid.
PB	Pool boiling.
s	At saturation.
SUB	Subcooling.
$v(g)$	Vapor(gas).
w	Wall.
ref	Reference.
sys	System or surrounding.

INTRODUCTION

THE INHERENT advantages of direct liquid cooling of high-performance electronic components have prompted a resurgence in the development of immersion cooling technology and have focused renewed attention on dielectric liquids. The stringent chemical and electrical requirements, imposed on liquids to be used in this cooling mode, have directed primary attention to the fully fluorinated fluorocarbons, such as used in the immersion-cooled Cray-2 [1]. Other liquids, notably the chlorofluorocarbons such as R-113, have also been considered.

Unfortunately, the thermal characteristics of the candidate immersion cooling liquids are generally inferior to those of more conventional coolants. Consequently, competitive thermal performance, at the system level, requires careful fluid selection and optimized fluid performance within the chosen cooling mode. The fluid figures of merit (FOM's) given herein can guide the selection and optimization of dielectric fluids for the electronics industry and be of use to both project engineers and chemists/chemical engineers involved in fluid development.

In this study, such FOM's are derived from the governing thermal transport relations for liquid and two-phase flow,

including: liquid forced and natural convection, boiling incipience, and critical heat flux for pool and flow boiling. It is to be noted that, for purposes of this study, the nucleate boiling regime is characterized by its two end-points, i.e., incipience and critical heat flux.

The derived FOM's are defined in such a way that a higher value of the FOM will always be synonymous with better heat transfer characteristics. These FOM's are then used to evaluate and compare several candidate immersion cooling fluids. Next, the governing relations are used to evaluate the sensitivity of the heat transfer rates to fluid properties. The influences of operating conditions, such as system pressure and subcooling, are also discussed.

The properties used for the computations performed in this work were obtained from various sources. The saturation data (h_{fg} , ρ_f , ρ_g) for all fluids at a pressure of 1 atm were obtained from [2]. Saturation data at pressures elevated above 1 atm were obtained from [2] for the FC liquids and from [3] for R-113. Similarly, all viscosity and specific heat data were obtained from [2], while data for R-113 were obtained from [4] for viscosity and [5] for specific heat. All surface tension data were obtained from relations of the following form:

$$\sigma = A \left(1 - \frac{T}{T_{\text{crit}}} \right)^n.$$

The table below lists the constants used in this equation, along with the references from which they were obtained, for each of the fluids considered:

Fluids	A	n	Reference
FC-72	0.042705	1.2532	[2], [6], [7]
FC-75	0.03910	1.160	[6], [7]
FC-84	0.045914	1.3282	[2]
R-113	0.055610	1.2384	[8]

SINGLE-PHASE CONVECTION

The efficacy of single-phase convection is generally expressed in terms of a heat transfer coefficient h representing the ratio of heat flux to the resulting temperature difference, i.e., $h = q''/(T_w - T_{\text{ref}})$. The higher the value of h , the more effective is the heat transfer mechanism.

Natural Convection

Natural convection heat transfer is governed by the Grashof and Prandtl numbers. In this mode, the heat transfer coefficient for external flow is expressible as (e.g., [9]).

$$h = \frac{k}{L} \text{Const} (\text{Gr} \cdot \text{Pr})^n$$

$$= \frac{k}{L} \text{Const} \left(\frac{\rho^2 g \beta \Delta T L^3}{\mu^2} \frac{C_p \mu}{k} \right)^n \quad (1)$$

where $\Delta T = T_w - T_{\text{ref}}$. The exponent n takes the value of 0.25 for laminar flow ($10^3 < \text{GrPr} < 10^9$) and 0.33 for turbulent flow ($10^9 < \text{GrPr} < 10^{12}$). The natural convection FOM's are found by extracting the fluid properties from these relations

$$\text{FOM}_{\text{nc,lam}} = \left(\frac{\beta \rho^2 C_p k^3}{\mu} \right)^{0.25} \quad (2)$$

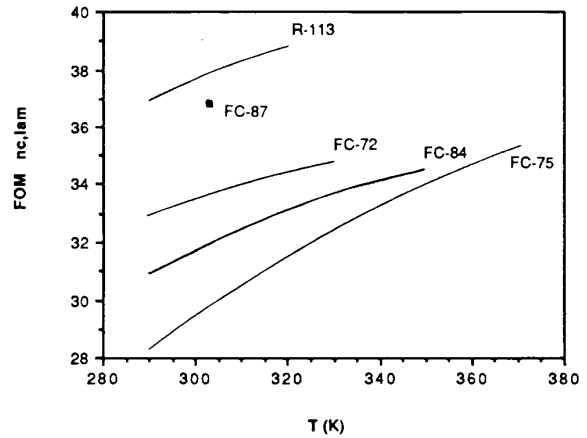


Fig. 1. FOM for natural convection, laminar external flow.

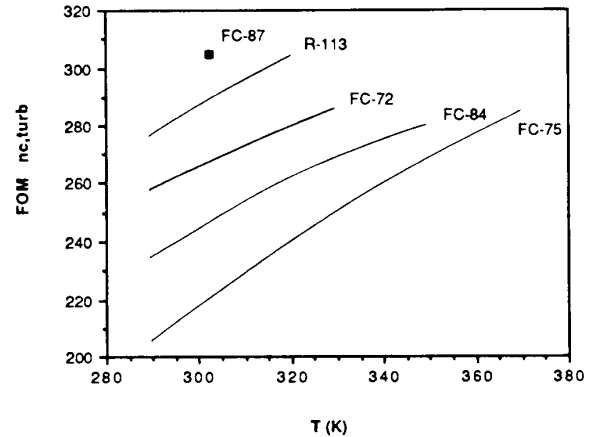


Fig. 2. FOM for natural convection, turbulent external flow.

$$\text{FOM}_{\text{nc,turb}} = \left(\frac{\beta \rho^2 C_p k^2}{\mu} \right)^{0.33} \quad (3)$$

The numerical values of these natural convection FOM's for FC-72, FC-75, FC-84, FC-87, and R-113 are shown in Figs. 1 and 2 for the 290 to 370 K temperature range. Examination of these two figures reveals that, from 290 K to the 1-atm saturation temperature of each liquid, the natural convection cooling capability of all five fluids increases modestly with temperature. Furthermore, FC-75 is seen to offer the lowest cooling capability while FC-87 and R-113 are seen to offer the highest cooling capability and, in their temperature range, to exceed the capability of the other candidate liquids by as much as 30 percent. Water is superior to the other fluids by an order of magnitude and is not shown in the figures.

Forced Convection

In forced convection, Reynolds number and Prandtl number influence the heat transfer coefficient. In this mode, the heat transfer coefficient is expressible as (e.g., [9])

$$h = \text{Const} \frac{k}{D}, \quad \text{for laminar internal flow} \quad (4a)$$

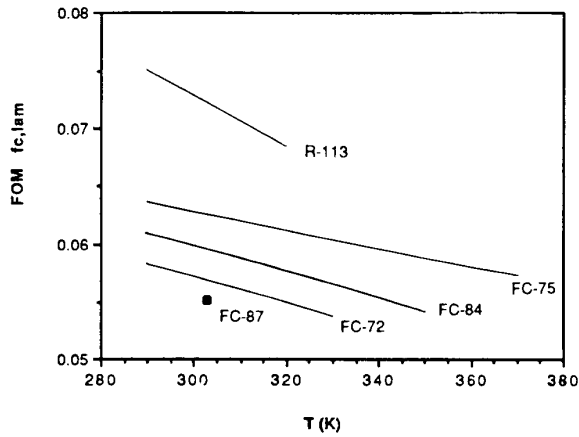


Fig. 3. FOM for forced convection, laminar internal flow.

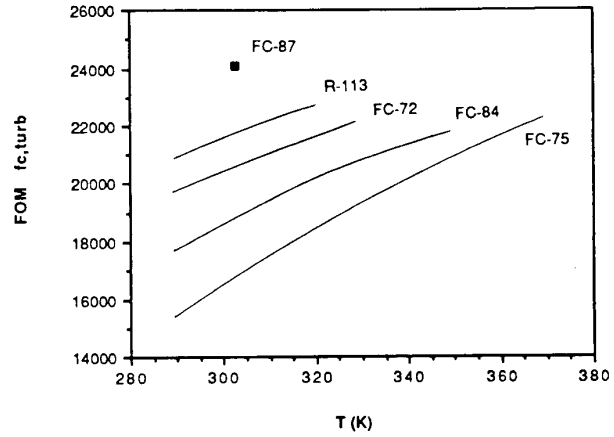


Fig. 4. FOM for forced convection, turbulent internal flow.

$$h = 0.664 \frac{k}{L} \text{Re}^{0.5} \text{Pr}^{0.33}$$

$$= 0.664 \frac{V^{0.5} \rho^{0.5} k^{0.67} C_p^{0.33}}{L^{0.5} \mu^{0.17}},$$

for laminar external flow (4b)

$$h = 0.023 \frac{k}{D} \text{Re}^{0.8} \text{Pr}^{0.4}$$

$$= 0.023 \frac{V^{0.8} \rho^{0.8} k^{0.6} C_p^{0.4}}{D^{0.2} \mu^{0.4}},$$

for turbulent internal flow (5a)

and

$$h = 0.036 \frac{k}{L} \text{Re}^{0.8} \text{Pr}^{0.33}$$

$$= 0.036 \frac{V^{0.8} \rho^{0.8} k^{0.67} C_p^{0.33}}{L^{0.2} \mu^{0.47}},$$

for turbulent external flow. (5b)

Following Mouromtseff [10], who pioneered the use of coolant FOM's, the forced convection FOM can be expressed in the form

$$M_0 = \frac{\rho^a k^b C_p^d}{\mu^e} \quad (6)$$

where the values of the four exponents are taken from (4) and (5). It may be noted that for fully developed laminar flow in pipes and ducts, this FOM is simply equal to the thermal conductivity of the liquid.

Values of the forced convection FOM, for the candidate fluids are shown as functions of temperature in Figs. 3 and 4 for internal laminar and turbulent flow, respectively. Interestingly, while in laminar flow the heat transfer capability diminishes modestly with increasing temperature, in turbulent internal flow the FOM increases significantly with rising liquid temperatures. R-113 appears to provide the highest cooling capability in laminar flow. FC-87 is the lowest in laminar flow and yet is the highest in heat removal capability in turbulent flow.

NUCLEATE BOILING

Incipience

Though much still remains to be learned about the effects of fluid properties and surface conditions on nucleate boiling incipience, it appears that, in most cases of practical interest, boiling incipience is linked to the activation of vapor/gas embryos trapped in microcavities on the heated surface. Because of their small contact angle and low surface tension, highly wetting liquids, such as R-113 or the FC series liquids, flood most large cavities on heated surfaces. Boiling must then commence from smaller cavities, requiring considerably higher superheat.

Following Rohsenow [11], a mechanical force balance on the gas/liquid interface suggests that the vapor pressure required to activate a bubble embryo is equal to

$$P_v(T_w) - P_l = \frac{2\sigma(T_w)}{R_0 f(\beta, \phi)} \quad (7)$$

where $R_0 f(\beta, \phi)$ is the bubble radius. By analysis [12] it can be shown that

$$f(\beta, \phi) = \left\{ \frac{(\beta_d - 2\phi)^{1/2} \beta_d^{-1/2}}{1 + \phi[2 - 3(\beta - \phi) + (\beta - \phi)^3]} \right\}^{1/3}$$

where β_d is the dynamic contact angle. The value of the incipience superheat, implicit in (7), can be determined numerically. In the use of immersion cooling, the liquid requiring the lowest incipience superheat is to be preferred and, hence, $(\Delta T_s)_i^{-1}$ can serve as the incipience FOM.

An approximation to (7), which employs the Clausius-Clapeyron equation to linearize the saturation relationship and presumes that the nucleation bubble radius R is known, is

$$(\Delta T_s)_i = \frac{2N_{uc}}{R} \quad (8)$$

where N_{uc} , the Nucleation number, is defined as

$$N_{uc} = \frac{\sigma T_s}{h_{fg} \rho_g}$$

Equation (8) can be used to define a liquid FOM, as done by Marto and Lepere [13]. However, such an FOM ignores the influence of contact angle on the embryonic bubble ra-

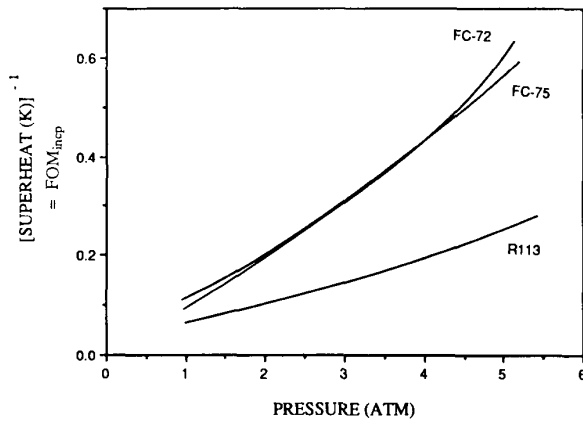


Fig. 5. Comparison of incipience superheat for several liquids at different pressures ($R_0 = 0.5 \mu\text{m}$).

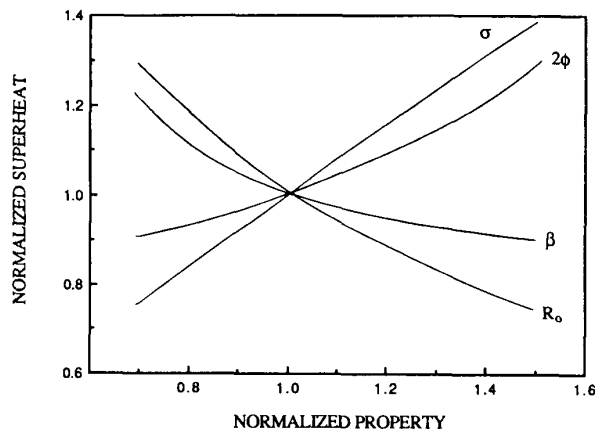


Fig. 6. Sensitivity of superheat to normalized property values for FC-72 at 1 atm.

dius and is subject to the large discrepancies introduced by the Clausius-Clapeyron equation when the incipience superheat is large. It is, therefore, preferable to rank the candidate liquids on the basis of a numerical solution of (7). Water, not shown, has an appreciable higher wetting angle, thus incipience superheat is very small and, typically, of little concern.

The contact angle effect on incipience superheat is ignored and β_d is taken to equal β , allowing computation of the incipience superheat values from (7) at different pressure levels for FC-72, FC-75, and R-113. These are shown in Fig. 5 for the conditions: $\beta = 2.0^\circ$, $2\phi = 1.0^\circ$, and $R_0 = 0.5 \mu\text{m}$.

Examination of Fig. 5 reveals that the FC liquids possess a larger incipience FOM than does R-113 in the range of 1–5 atm. The FOM_{incp} values increase with pressure for both types of fluids.

Further insight into the influence of fluid properties and geometry on the incipience superheat can be obtained by probing the sensitivity of $(\Delta T_s)_i$ for FC-72 to variations in the contact angle, surface tension, cavity radius, and cavity angle. In Fig. 6 the relative change in the incipience superheat, expressed as the ratio of $(\Delta T_s)_{i,incp}$ to $(\Delta T_s)_{i,ref}$ at 1 atm, is plotted against similarly defined variations in the relevant geometric

TABLE I
INCIPIENCE SUPERHEAT SENSITIVITY OF FC LIQUIDS

	$P = 1 \text{ atm}$			
	FC-72	FC-75	FC-87	FC-84
β	-0.7193	-0.9270	-0.7281	-0.8234
σ	0.8015	0.9975	0.8039	0.9042
2ϕ	0.3711	0.4784	0.3756	0.4245
R_0	-0.3617	-0.4661	-0.3659	-0.4139
	$P = 5 \text{ atm}$			
	FC-72	FC-75	FC-87	FC-84
β	-0.9370	-0.9426	-0.9440	-0.9470
σ	0.9823	0.9787	0.9801	0.9872
2ϕ	0.4852	0.4851	0.4864	0.4903
R_0	-0.4713	-0.4730	-0.4729	-0.4746

and fluid parameters. Examination of this figure reveals that the incipience superheat, at 1 atm, is strongly affected by surface tension and cavity radius, but more moderately affected by contact angle.

The incipience superheat sensitivities, i.e., $[d(\Delta T_s)_i / (\Delta T_s)_i] / [d(\text{property}) / \text{property}]$, are shown in Table I for FC-72, -75, -87, and -84. They suggest that the trends of Fig. 6 apply to fluorocarbons other than FC-72 at 1 atm and to all the FC fluids at 5 atm.

Pool Boiling CHF

The nucleate boiling regime is limited by a maximum heat flux which is referred to as the critical heat flux, or CHF. Exceeding this heat flux results in an instability that leads to vapor blanketing of the heated surface. Since this blanketing drastically reduces the efficiency of heat transfer, its occurrence is considered to be highly undesirable from the cooling point of view.

The classic CHF relation was derived by Zuber *et al.* [14]

$$CHF_Z = \frac{\pi}{24} \sqrt{\rho_g} h_{fg} [\sigma g (\rho_f - \rho_g)]^{1/4}. \quad (9)$$

Although this equation is valid only for saturated pool boiling on large, horizontal surfaces, it is a frequently referred to benchmark. From this equation, the FOM for saturated, pool boiling on large surfaces is

$$FOM_{CHF,PB,S} = \frac{CHF_Z}{(\pi/24)} = \sqrt{\rho_g} h_{fg} [\sigma g (\rho_f - \rho_g)]^{1/4}. \quad (10)$$

This FOM is plotted in Fig. 7 as a function of pressure in the range of 1–5 atm. FC-87 and FC-84 are only plotted at 1 atm, where property data are available. It should be noted that the FOM for water is not known, as is nearly one order of magnitude larger than the value of the candidate immersion cooling liquids. The enhanced performance of water in this situation is primarily due to its larger latent heat of vaporization.

The sensitivity of CHF to variations in fluid property can be obtained by normalizing the derivative with respect to property of (9), i.e.,

$$\frac{d(CHF_Z)/CHF_Z}{d(\text{property})/\text{property}}$$

TABLE II
POOL BOILING CHF SENSITIVITY EXPRESSIONS

$\gamma = \frac{CHF_Z^2}{CHF_{I\&M}}$			
ρ_g	ρ_f	h_{fg}	σ
Saturated (9)			
$\frac{1}{2} \left(1 - \frac{1}{2} \left(\frac{\rho_g}{\rho_f - \rho_g} \right) \right)$	$\frac{1}{2} \left(\frac{\rho_f}{\rho_f - \rho_g} \right)$	1	$\frac{1}{4}$
Small heater (11)			
$\frac{1}{2} - \frac{\rho_g}{8(\rho_f - \rho_g)}$	$\frac{3}{8} \left(\frac{\rho_f}{\rho_f - \rho_g} \right)$	1	$\frac{3}{8}$
Subcooled (12)			
$\frac{1}{2} \left(1 - \frac{1}{2} \left(\frac{\rho_g}{\rho_f - \rho_g} \right) \right) - \frac{3}{4} (1 - \gamma)$	$\frac{1}{4} \left(\frac{\rho_f}{\rho_f - \rho_g} \right) + \frac{3}{4} (1 - \gamma)$	γ	$\frac{1}{4}$

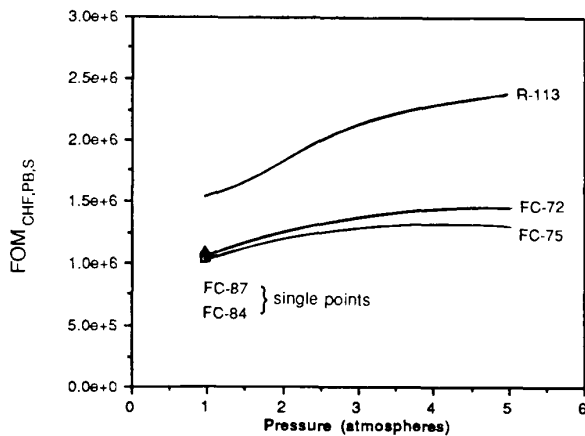


Fig. 7. CHF FOM for saturated pool boiling versus pressure.

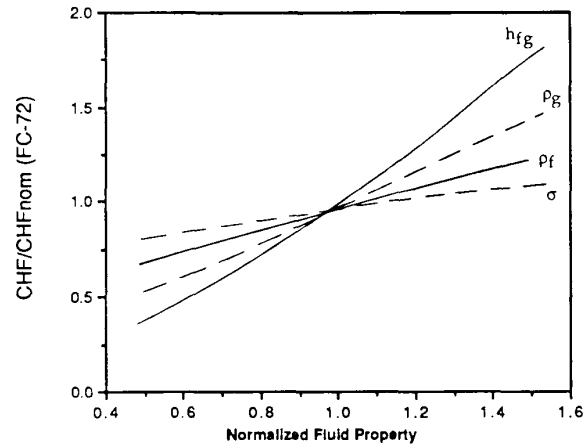


Fig. 8. Sensitivity of normalized Zuber-based CHF to fluid property variation (FC-72).

The resulting expression for sensitivity is presented in Table II along with similar sensitivity expressions obtained from relations (to be presented below) applicable to the subcooled and small heater cases. Fig. 8 presents a graphical description of the sensitivity of CHF to fluid properties via a plot of CHF versus a specific fluid property. Both the CHF and the fluid property are normalized to a 1-atm saturated condition. The plot indicates that CHF is most sensitive to latent heat and vapor density in the large heater, pool boiling mode.

When the heater size is considerably reduced, a substantial increase in CHF can be attained. It has been experimentally demonstrated that below a critical size, CHF increases with decreasing heater size in a fashion correlatable by an expression attributed to Lienhard and Dhir [15]

$$CHF_{L\&D} = \frac{1.4}{\sqrt[4]{L'}} CHF_Z \quad (11)$$

where

$$L' = L \sqrt{\frac{\sigma}{g(\rho_f - \rho_g)}}$$

This equation is merely a correction to (9) and is used when the heater size is smaller than a critical value which, at 1 atm, is <6 mm for most of the candidate fluids considered here. Despite the changes in CHF sensitivity resulting from this correlation (see Table II), a fluid FOM based on (11) can be shown to display liquid rankings and pressure variations virtually identical to those shown in Fig. 7.

The most substantial increase in pool boiling CHF can be effected by subcooling the liquid below its saturation temperature. This subcooling effect has been correlated by Ivey and Morris [16], in the following form:

$$CHF_{I\&M} = CHF_Z \left\{ 1 + 0.1 \left(\frac{\rho_g}{\rho_f} \right)^{1/4} \cdot \left[\frac{c_p \rho_f (T_s - T_{sys})}{h_{fg} \rho_g} \right] \right\} \quad (12)$$

Fig. 9 shows the effect of subcooling on the CHF as computed

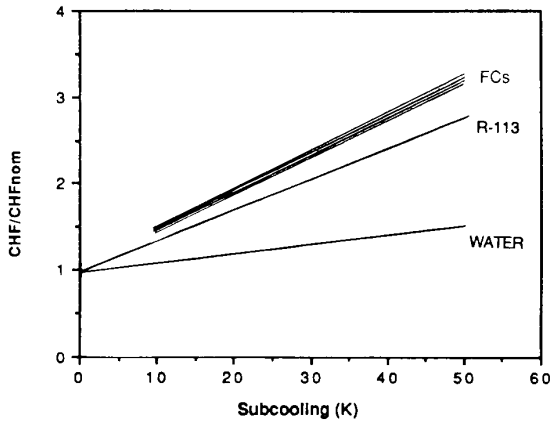


Fig. 9. Normalized CHF versus subcooling (pool boiling).

by (12). CHF is seen to be very sensitive to subcooling in the FC fluids, with 20° of subcooling resulting in a twofold increase in CHF. This effect is somewhat reduced for R-113 and is substantially diminished for water. These trends are a function of the parameter $\rho_f^{3/4} c_p / \rho_g^{3/4} h_{fg}$ which may be viewed as an FOM for subcooling effects on pool boiling CHF.

The sensitivity of CHF to fluid properties as a function of subcooling is obtained from (12) (see Table II). Due to the form of the CHF_{I&M} relation, the sensitivity of the critical heat flux to specific fluid properties varies significantly with the degree of subcooling and the operating pressure. The sensitivity of pool boiling CHF to fluid properties is tabulated in Table III for FC-72, -75, -84, and -87 at 1 atm and FC-72 and -75 at 5 atm. Figs. 10 and 11 reproduce these values in graphical form for FC-72 at 1 and 5 atm, respectively. Previously, it was shown that at low (or zero) subcoolings, CHF can be improved most dramatically by increasing the latent heat of vaporization. However, at large subcoolings, CHF is increased most effectively by increasing the specific heat or liquid density. For example, at a subcooling of 50 K, CHF is nearly twice as sensitive to the liquid density and the specific heat as it is to the latent heat. A similar, though far weaker, trend is apparent at 5 atm, where the sensitivity of CHF to latent heat drops substantially from its value of unity as subcooling increases, and CHF is found to be nearly equally sensitive to latent heat, liquid density, and specific heat at subcoolings of 50 K.

Flow Boiling CHF

Lee *et al.* [17] recently developed a subcooled, flow boiling CHF correlation which may be used in liquid cooling of small heated regions such as electronic devices. The correlation, developed for heated lengths from 0.25 to 5.0 mm and bulk velocities from 1 to 15 m/s, takes the form

$$CHF = 0.0742 \left(\frac{u^{0.27}}{L^{0.365}} \right) (\rho_g^{0.239} \rho_f^{0.396} \sigma^{0.365} h_{fg}) \cdot \left[1 + 0.952 \left(\frac{\rho_f^{0.118} C_p^{1.414}}{\rho_g^{0.118} h_{fg}^{1.414}} \right) \Delta T_{sub}^{1.414} \right]. \quad (13)$$

TABLE III
POOL BOILING CHF SUBCOOLING SENSITIVITIES

	<i>P</i> = 1 atm				<i>P</i> = 5 atm	
	FC-72	FC-75	FC-84	FC-87	FC-72	FC-75
Subcooling = 0°C						
ρ_f	0.25	0.25	0.25	0.25	0.26	0.26
ρ_g	0.50	0.50	0.50	0.50	0.49	0.49
σ	0.25	0.25	0.25	0.25	0.25	0.25
C_p	0.00	0.00	0.00	0.00	0.00	0.00
h_{fg}	1.0	1.0	1.0	1.0	1.0	1.0
Subcooling = 10°C						
ρ_f	0.51	0.51	0.53	0.51	0.38	0.38
ρ_g	0.24	0.24	0.22	0.24	0.37	0.37
σ	0.25	0.25	0.25	0.25	0.25	0.25
C_p	0.35	0.34	0.37	0.34	0.16	0.16
h_{fg}	0.65	0.66	0.63	0.66	0.84	0.84
Subcooling = 20°C						
ρ_f	0.64	0.63	0.65	0.63	0.46	0.47
ρ_g	0.11	0.12	0.10	0.12	0.29	0.28
σ	0.25	0.25	0.25	0.25	0.25	0.25
C_p	0.51	0.50	0.53	0.50	0.27	0.27
h_{fg}	0.49	0.50	0.47	0.50	0.73	0.73
Subcooling = 30°C						
ρ_f	0.71	0.70	0.72	0.70	0.53	0.53
ρ_g	0.04	0.05	0.03	0.05	0.22	0.22
σ	0.25	0.25	0.25	0.25	0.25	0.25
C_p	0.61	0.60	0.63	0.60	0.35	0.36
h_{fg}	0.39	0.40	0.37	0.40	0.65	0.64
Subcooling = 40°C						
ρ_f	0.76	0.75	0.77	0.75	0.58	0.58
ρ_g	-0.01	0.00	-0.02	0.00	0.17	0.17
σ	0.25	0.25	0.25	0.25	0.25	0.25
C_p	0.67	0.66	0.69	0.66	0.42	0.42
h_{fg}	0.33	0.34	0.31	0.34	0.58	0.58
Subcooling = 50°C						
ρ_f	0.79	0.78	0.80	0.78	0.61	0.62
ρ_g	-0.04	-0.04	-0.05	-0.03	0.14	0.13
σ	0.25	0.25	0.25	0.25	0.25	0.25
C_p	0.72	0.71	0.73	0.71	0.47	0.47
h_{fg}	0.28	0.29	0.27	0.29	0.53	0.53

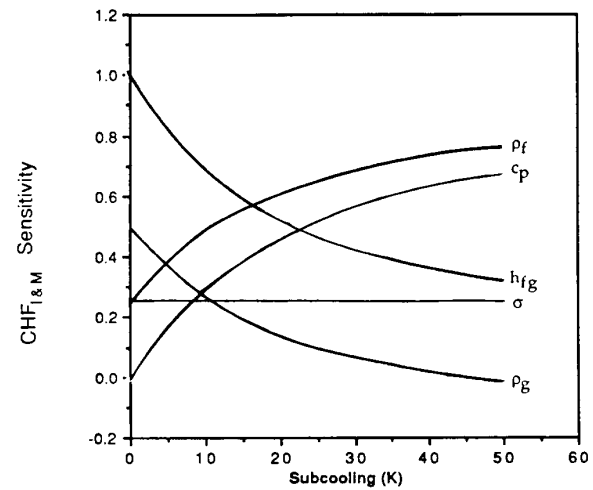


Fig. 10. Sensitivity of pool-boiling CHF to liquid properties at varying subcooling (*P* = 1 atm, FC-72).

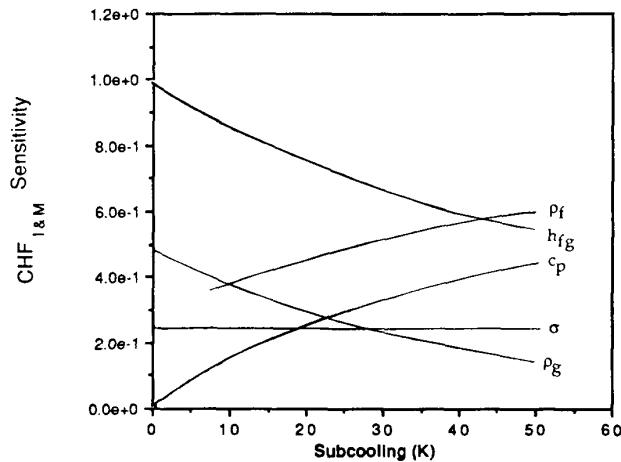


Fig. 11. Sensitivity of pool boiling CHF liquid properties at varying subcooling ($P = 5$ atm, FC-72).

A fluid FOM for saturated flow boiling, critical heat flux can be extracted from this relation, by ignoring the geometric, velocity, and subcooling effects, to yield

$$FOM_{CHF,FB,S} = \rho_g^{0.239} \rho_f^{0.396} \sigma^{0.365} h_{fg}. \quad (14)$$

The specific fluid properties contributing to the influence of subcooling on CHF can be grouped into a second FOM as

$$FOM_{CHF,FB,SUB} = \frac{\rho_f^{0.118} C_{pl}^{1.414}}{\rho_g^{0.118} h_{fg}^{1.414}}. \quad (15)$$

It is to be noted that alternative CHF correlations, notably [18], initially developed for relatively large heated areas, are available in the literature. While the use of (13) is to be preferred in the evaluation of liquids for immersion cooling of electronic chips and components, use of these other relations is not expected to substantially alter the fluid rankings or their sensitivity to pressure and subcooling.

As in other heat transfer regimes, the value of the saturated, flow boiling CHF figure of merit for water is higher than that for the candidate immersion cooling liquids. As a consequence, water was omitted from the Fig. 12 display of the variation of $FOM_{CHF,FB,S}$ with pressure, over the range of 1–5 atm. For the candidate fluids, this figure of merit is seen to vary only modestly with pressure, with R-113 attaining nearly twice the value of the FC liquids.

The sensitivity of the saturated flow boiling critical heat flux to variations in fluid properties can be found by partial differentiation of the governing equation (13). The results of this operation are shown in the first section of Table IV and seen to generally parallel the fluid-property sensitivities previously encountered for saturated, pool boiling CHF, i.e., the predominant influence of the latent heat.

To take maximum advantage of possible liquid subcooling, the candidate liquid possessing the highest value of $FOM_{CHF,FB,SUB}$ should be chosen for use in an immersion cooling system. While use of this FOM, which contains the latent heat in the denominator in (15), would reverse the order shown in Fig. 12, a closer examination of (13) and Fig. 13

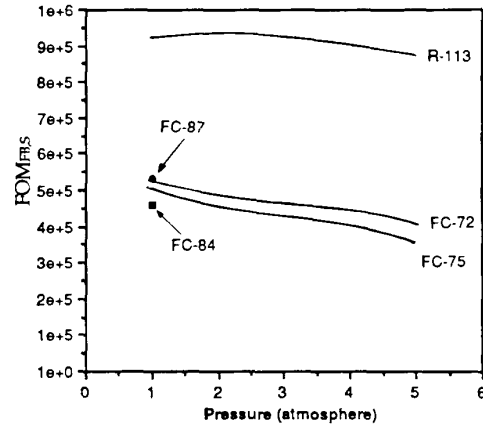


Fig. 12. Comparison of pressure effect on flow boiling FOM for saturated boiling CHF.

TABLE IV
SENSITIVITY OF FLOW BOILING CHF FOR FC SERIES

	$P = 1$ atm			$P = 5$ atm		
	FC-72	FC-75	FC-84	FC-87	FC-72	FC-75
$\Delta T_{sub} = 0^\circ C$						
ρ_f	0.40	0.40	0.40	0.40	0.40	0.40
ρ_g	0.24	0.24	0.24	0.24	0.24	0.24
σ	0.36	0.36	0.36	0.36	0.36	0.36
C_{pl}	0.00	0.00	0.00	0.00	0.00	0.00
h_{fg}	1.00	1.00	1.00	1.00	1.00	1.00
$\Delta T_{sub} = 10^\circ C$						
ρ_f	0.41	0.41	0.41	0.41	0.41	0.41
ρ_g	0.23	0.23	0.23	0.23	0.23	0.22
σ	0.36	0.36	0.36	0.36	0.36	0.36
C_{pl}	0.11	0.13	0.13	0.11	0.15	0.18
h_{fg}	0.89	0.87	0.87	0.89	0.85	0.82
$\Delta T_{sub} = 30^\circ C$						
ρ_f	0.43	0.43	0.43	0.43	0.44	0.44
ρ_g	0.21	0.20	0.20	0.21	0.20	0.19
σ	0.36	0.36	0.36	0.36	0.36	0.36
C_{pl}	0.40	0.44	0.45	0.39	0.49	0.57
h_{fg}	0.60	0.56	0.55	0.61	0.51	0.43
$\Delta T_{sub} = 50^\circ C$						
ρ_f	0.45	0.45	0.45	0.45	0.46	0.46
ρ_g	0.19	0.18	0.18	0.19	0.18	0.17
σ	0.36	0.36	0.36	0.36	0.36	0.36
C_{pl}	0.61	0.67	0.68	0.60	0.72	0.81
h_{fg}	0.39	0.33	0.32	0.40	0.28	0.19

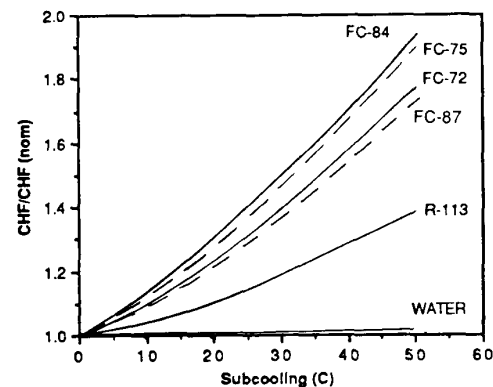


Fig. 13. Subcooling effect on flow boiling CHF at 1 atm.

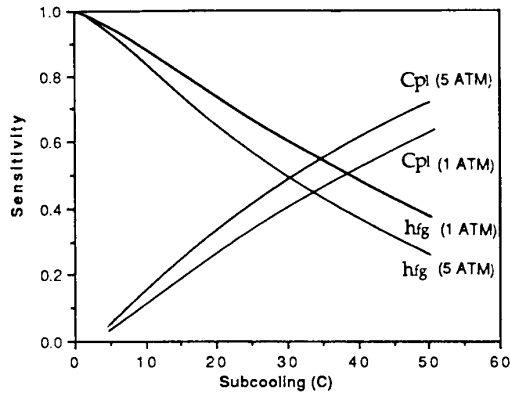


Fig. 14. Pressure and subcooling effects on flow boiling CHF sensitivity for FC-72.

suggests that subcoolings in excess of 60 K for FC's and 100 K for R-113 would be required to overcome the dominance of the saturated flow boiling term in the flow boiling CHF correlation. Since electronic cooling systems are typically required to operate with chip temperatures below 100°C and are sensitive to energy costs, it is unlikely that subcoolings substantially greater than 50 K would be encountered. In this range of subcoolings, the fluid rankings based on $FOM_{CHF,FB,S}$ can be expected to be valid.

While the presence of subcooling may not alter the choice of immersion cooling fluid, the Lee *et al.* [17] correlation does show subcooling to have a major influence on the flow boiling critical heat flux. This is displayed most clearly in Fig. 13, where a 50 K subcooling (at 1 atm) is seen to double CHF for FC-84, to increase FC-72 and FC-87 CHF by nearly 75 percent and R-113 by approximately 40 percent.

The sensitivities to property variations of the subcooled, flow boiling CHF of the candidate immersion cooling liquids, as derived from (13), are tabulated in the last three sections of Table IV, for both atmospheric and 5-atm operating pressures. Examination of Table IV reveals a constant CHF sensitivity to surface tension (due to its absence from the subcooling term in (13)) and a near-constant sensitivity to liquid and vapor density across the tabulated range of liquids, subcoolings, and operating pressures. However, as may be seen in Fig. 14 for FC-72, this is not the case for the sensitivity of $CHF_{FB,SUB}$ to latent heat and liquid-specific heat. The CHF sensitivity to these two fluid properties displays opposite trends, with h_{fg} dominating near-saturation conditions and $C_{p,f}$ dominating at high subcoolings. Parity for FC-72 is attained near 30 and 40 K for 5 and 1 atm, respectively.

CLOSING REMARKS

In the preceding sections, liquid FOM's that can be used in the selection and development of immersion cooling fluids, were identified for the modes of cooling that are relevant to the thermal control of electronic equipment. Five candidate immersion cooling fluids, including four fully fluorinated fluorocarbons (FC-72, -75, -84, -87) and one chlorinated fluorocarbon (R-113), as well as water, were evaluated and compared using these parameters. The influence of system tem-

perature and pressure on heat transfer rates in each of the heat transfer regimes was studied with the aid of commonly used empirical correlations.

Fluid Selection

Not surprisingly, water displays significantly better thermal performance than the candidate fluorocarbon liquids in all the heat transfer regimes and is clearly the coolant of choice, in its range of operation, when only thermal considerations need be addressed. Furthermore, the chlorinated fluorocarbon, R-113, is nearly always found to offer better heat removal capability than the FC liquids. Electrical and chemical considerations have, however, focused much of the electronic industry's attention on the inert, fully fluorinated fluorocarbons. The comparison of the thermal performance of these liquids to water and R-113 was the primary motivation for the present study.

In single-phase liquid convection, R-113, generally displays superior performance, but the FOM's for FC-87 are within 10 percent of the R-113 values for all but fully developed, internal, forced convection, laminar flows. Increasing the operating temperature, in the range of 10 to 100°C, was generally found to improve liquid convective heat transfer rates for all the candidate liquids by 5 to 20 percent except for the case of forced convection laminar internal flow.

In selecting a fluid for ebullient cooling, attention must be paid to the superheat required to initiate boiling and to the maximum heat flux at which nucleate boiling can be sustained (CHF).

Comparisons based on boiling incipience show that the fully fluorinated liquids display higher incipience superheat FOM values than the chlorinated fluorocarbon, R-113, value. Incipience superheats become small (and the FOM's rise) as pressure is elevated; and incipience superheat is shown to be strongly affected by surface tension and cavity size.

Fluid comparisons based on the critical heat flux FOM's reveal that the four fully fluorinated liquids display nearly identical performance for saturated and subcooled pool boiling, and saturated flow boiling conditions and are only slightly differential at subcooled flow boiling conditions. Consequently, if CHF is the design parameter of concern, an immersion cooling system can be designed for a "generic" FC fluid, leaving the choice of saturation, and hence component surface temperature a free variable.

Evaluation of the candidate liquids suggests that R-113 may provide approximately 50 percent higher CHF in pool boiling and as much as 90 percent higher CHF in flow boiling than the FC liquids. Increasing system pressure, in the range of 1 to 5 atm, results in a predicted pool boiling CHF improvement of approximately 20 percent for the FC's and 50 percent for R-113. In saturated flow boiling, a 1-5 atm pressure increase results in a modest (~10-percent) decrease in CHF for the fully fluorinated fluorocarbons and an even weaker decrease for R-113.

Subcooling has a very pronounced influence on CHF, especially for the FC liquids. Subcooling the liquid by 20 K may increase CHF for the FC's by as much as a factor of 2 in pool boiling and by 1.3 in flow boiling. While similar pool boiling improvements are predicted for R-113 (20 K roughly

doubling CHF), in 20 K subcooled flow boiling, R-113 can be expected to experience only a 10-percent CHF improvement. These results suggest that the FC liquids may approach the thermal performance capability of the chlorinated fluorocarbons in electronic systems employing highly subcooled pool and flow boiling.

Fluid Development

The chemical and electrical attractiveness and, yet, inherently poor heat removal capability of the fully fluorinated fluorocarbons, offers a significant challenge and opportunity to chemists involved in the development or application of new liquids. The FOM's can be used to compare the performance of newly synthesized (or theoretically possible) liquids to currently available immersion cooling fluids. The results of the sensitivity analyses can serve to focus attention on fluid property enhancements that are most likely to improve thermal performance.

As is no doubt apparent, the choice of heat transfer regime will dictate the fluid properties to be addressed. In single-phase liquid convection, the exponents on the fluid properties in (6) will prioritize such efforts. As an example, an FC liquid, designed primarily for turbulent, forced convection heat transfer, would benefit from a significant increase in liquid density and thermal conductivity, and a decrease in viscosity, in that order.

Fluids designed for minimum superheat upon incipience of boiling would benefit from a higher surface tension. This benefit would be seen as a reduced superheat for the initiation of boiling in a particular cavity size as well as an ability to boil from larger cavities without flooding the sites.

To improve the saturated, pool boiling CHF of the fully fluorinated fluorocarbons, it is desirable to increase the latent heat of vaporization, vapor density, liquid density, and surface tension, in that order. If, however, primary attention is to be focussed on saturated flow-boiling CHF, fluid properties should be increased in a slightly different order; latent heat of vaporization, the liquid density, the surface tension, and the vapor density. As previously noted, development of immersion cooling liquids to be used primarily in a highly subcooled

boiling mode would benefit most from attention to the liquid density and liquid specific heat and only modest attention need be paid to the other properties.

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