HOW TO CHOOSE COOLING FLUID IN FINNED TUBE HEAT EXCHANGERS

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ABSTRACT

A comparison is made between an ice-slurry solution with different inlet ice mass fractions and an R404a refrigerant utilised in an evaporator with constant heat power and fixed geometry. The comparison is obtained using a parameter that represents the ratio between the total real entropy variation and the exchanged heat. This parameter, introduced by Grazzini and Ferraro (2000), shows that an ice-slurry solution is better than an R404a as refrigerant fluid, when considering a particular heat exchanger.

The heat exchanger used is of a finned tube type with an in-line tube bank. Results show that with the same heat power ice-slurry entropy variation is lower than that given by R404a.

INTRODUCTION

The need to find an alternative to common refrigerant fluids has led to the investigation of different kinds of secondary refrigerant fluid: in particular, the two-phase refrigerant fluid known as ice-slurry, composed of water, an additive and small ice crystals.

The presence of ice crystals in the solution allows more cooling energy to be transported per unit of mass than with more usual refrigerant fluids, and produces some advantages such as lower flow rate, lower pumping power, and smaller piping diameters.

In this paper, utilising a finned tube heat exchanger with an in-line tube bank, fixed geometry and constant heat power, a comparison is made between an ice-slurry solution with different inlet ice mass fractions and different inlet temperatures, and R404a refrigerant. The parameter that represents the ratio between total entropy variation and exchanged heat, is used (Grazzini and Ferraro, 2000). Results show how total entropy production is influenced by different ice-slurry inlet temperatures, different ice fraction and different ice-slurry inlet-outlet temperature differences.

1. PARAMETER EVALUATION OF AN ICE-SLURRY

Considering only one heat exchanger stream, the parameter, introduced by Grazzini and Ferraro (2000), is:

$$\frac{\Delta S}{Q} = \frac{\dot{m}}{Q}\left(c_p \ln\left(\frac{T_{out}}{T_{in}}\right) + \frac{\beta \cdot \Delta P}{\rho} \right) \tag{1}$$

where β and ρ are assumed to be constant along the length considered.

For the particular in line bank finned tube heat exchanger utilised, the geometric and physical values are reported in tables 1 and 2. A constant thermal power Q=1 kW and a constant inlet-outlet temperature difference of 10°C is assumed on the refrigerated air side. Under these conditions the external heat transfer coefficient remains constant.

From equation (1) it can be claimed that the ice-slurry entropy variation is given by two terms, where the first one is a function of the temperature (ΔS_t), and the second one is a function of pressure losses (ΔS_p):

$$\Delta S_t = \dot{m} c_p^{\,*} \ln\left(\frac{T_{out}}{T_{in}}\right) \tag{2}$$

$$\Delta S_p = \dot{m} \frac{\beta}{\rho} \Delta p \tag{3}$$

Where c_p^* is evaluated as $Q/(\dot{m}\,(T_o - T_{in}))$. The ice-slurry pressure losses (Δp) can be calculated by using the same relation as for a liquid solution, while the ice-slurry characteristic parameters, such as density (ρ_{sl}), dynamic viscosity (μ_{sl}) etc., are functions of solid and liquid phases (Grazzini and Ferraro, 1999, Bel and Lallemand, 1999, Cavallini and Fornasieri, 2000). Density and viscosity of liquid phase are calculated using Melinder equations (1997). Figure 1 shows the dependence of the ice-slurry pressure losses on ice concentration and temperature difference between inlet and outlet.

Table1: Heat exchanger geometric specification.

Width (Length of pipes) [mm]	1 200
Length [mm]	420
Height [mm]	70
Tube Material	Copper
Fin Material	Aluminium
Inside pipe diameter (D_i) [mm]	11.3
Pipe number	20
Pipe thickness [mm]	0.35
Number of fins	171
Fin thickness [mm]	0.23
Rank Type	In-line
Rank Number	10

Table2: Ice-slurry and R404a values of used to make the comparison.

Antifreeze	Methanol		
Ice fraction (X_s)	0.2 - 0.15 - 0.10		
Ice-slurry (T_{out}-T_{in}) [°C]	0.5 - 1- 1.5 – 2		
h_e [W/m^{2}K]	43		
Warm fluid T_{in} [°C]	0	-2	-5
R404a T_{in} [°C]	-11	-13	-16
Ice-slurry T_{in} [°C]	-10.4	-12.4	-15.4

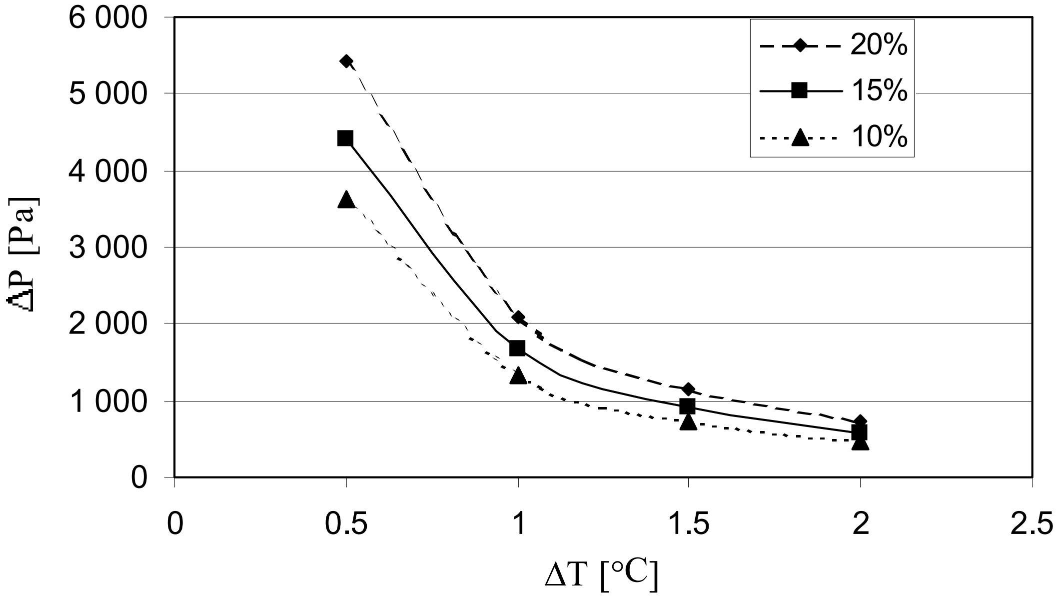

Figure 1 – Ice-slurry pressure losses versus inlet-outlet temperature difference with different X_{sl}

Evaluating the flow rate as:

$$\dot{m} = \frac{Q}{r\Delta X_{sl} + c_p\left(T_{out} - T_{in}\right)} \tag{4}$$

then equation (1) becomes:

$$\frac{\Delta S}{Q} = \frac{c_p^{\ *} \ln\left(\frac{T_{out}}{T_{in}}\right) + \frac{\beta}{\rho}\Delta p}{r\Delta X_{sl} + c_p\left(T_{out} - T_{in}\right)} \tag{5}$$

Equation (5) shows that, when ice-slurry has a fixed inlet ice fraction and constant exchanged thermal power, an increase in the inlet-outlet temperature difference leads to a decrease of ΔS/Q (fig. 2). While, under the same conditions, a reduction of the ice-slurry inlet temperature leads to an increase of ΔS/Q (fig. 3).

The ice-slurry heat transfer coefficient can be calculated by the correlation given by Christensen and Kauffeld (1997):

$$Nu_{sl} = Nu_{fl}\left(1 + 0.103 X_s - 2.003 \operatorname{Re}_{sl}^{\ -0{,}192(30-i)/30}\ X_s^{\ 0{,}339(\operatorname{Re}_{sl}/1000)}\right) \qquad X_{sl} > 5\% \tag{6}$$

$$Nu_{sl} = Nu_{fl} \qquad X_{sl} < 5\% \tag{7}$$

Using ice-slurry thermal conductivity λ_{sl}, the heat transfer coefficient is:

$$h_{sl} = \frac{Nu_{sl}\lambda_{sl}}{D_i} \tag{8}$$

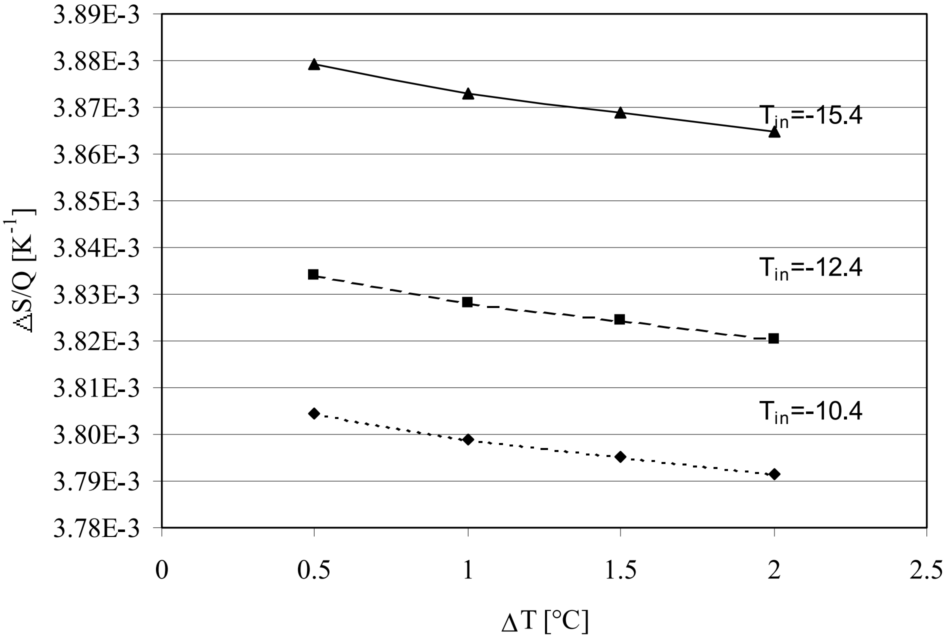

Figure 2 - ΔS/Q versus ice-slurry inlet-outlet difference temperature, with fixed inlet ice fraction X_{sl} = 0.20 and temperatures

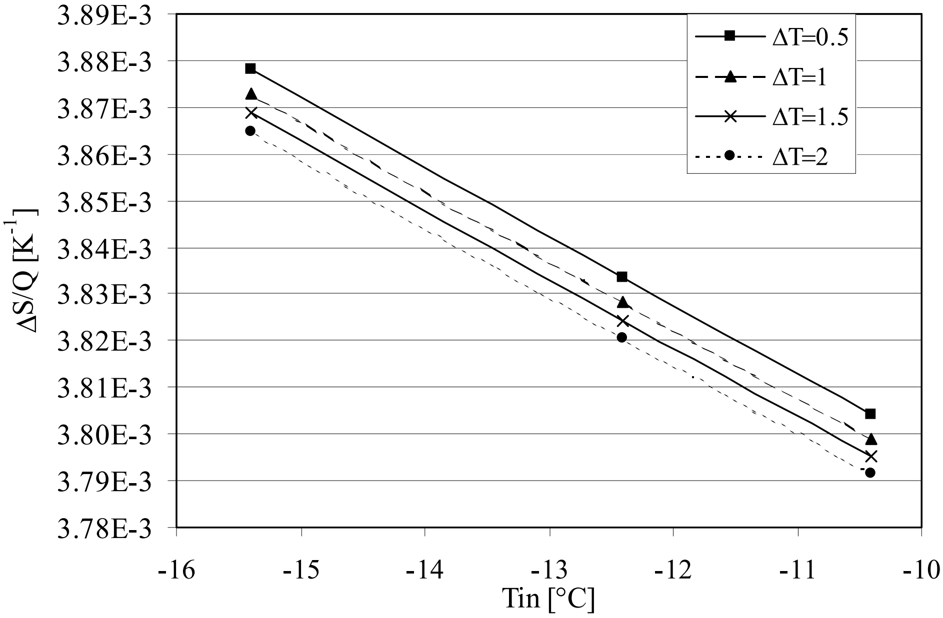

Figure 3 - ΔS/Q versus ice-slurry inlet temperature, with fixed inlet ice fraction X_{sl} = 0.20 and temperatures

Seeing that ice-slurry is a combination of a pure solid phase, ice, and a liquid phase, λ_{sl} is normally correlated to the ice fraction and thermal conductivity of both phases.
Thermal conductivity λ_{sl} is obtained by Jeffrey equation (1973):

$$\lambda_{sl} = \lambda_l (1 + 3\varphi_s \beta + 3\varphi_s{}^2 \beta^2 \gamma) \tag{9}$$

with:

$$\gamma = 1 + \frac{\beta}{4} + \frac{3\beta}{16}\left(\frac{\alpha+2}{2\alpha+3}\right) \quad ; \quad \beta = \frac{\alpha-1}{\alpha+2} \quad ; \quad \alpha = \frac{\lambda_g}{\lambda_l} \tag{10}$$

2. COMPARISON BETWEEN ICE-SLURRY AND R404a

The comparison is made by using ice-slurry solution with different inlet ice fractions, ($X_{s\,l}$= 0.20 $X_{s\,l}$= 0.15, $X_{s\,l}$= 0.10) and inlet-outlet temperature differences (ΔT=0.5, ΔT=1, ΔT=1.5, ΔT=2).
The classical equations:

$$Q = \dot{m}\left(c_p\left(T_{out} - T_{in}\right) + r\Delta X_s\right) \tag{11}$$

$$\frac{1}{UA} = \frac{1}{h_i \pi n D_i L} + \frac{1}{h_e \pi n D_e L \eta} \tag{12}$$

$$Q = UA\Delta T_{ml} = UA\left[\frac{\left(T_{g\,in} - T_{sl\,out}\right) - \left(T_{g\,out} - T_{sl\,in}\right)}{\ln\frac{\left(T_{g\,in} - T_{sl\,out}\right)}{\left(T_{g\,out} - T_{sl\,in}\right)}}\right] \tag{13}$$

are used to evaluate ice-slurry temperatures and the global heat transfer coefficient, which is calculated neglecting the thermal resistance of the pipe wall; η = 9.594 is the finned surface effectiveness, defined as the heat exchanged by the finned surface over the unfinned one.
Thermophysical properties of R404a are calculated using the Refprop 5 program (Huber et al. 1996), while local heat transfer coefficient using empirical correlation by Duminil and Vrinat (1991) and pressure losses using empirical correlation by Dukler (ASHRAE, 1997). Once the inlet and outlet temperatures for both fluids are known, parameter ΔS/Q is calculated by using equation (4). Figures 4 and 5 show the entropy ratio values $(\Delta S/Q)_{R404a}/(\Delta S/Q)_{sl}$.

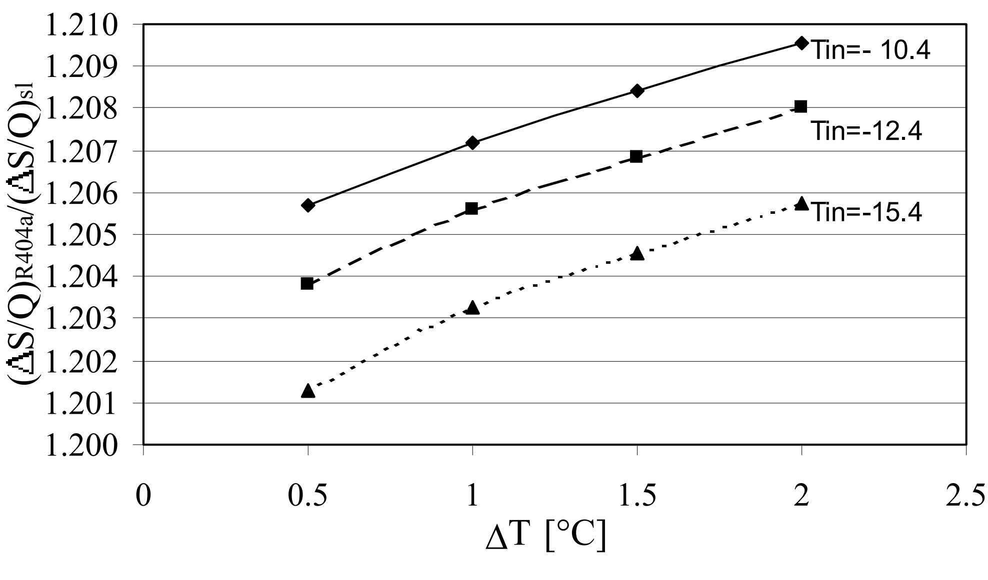

Figure 4 - $(\Delta S/Q)_{R404a}/(\Delta S/Q)_{sl}$ versus ΔT for ice-slurry with ice fraction X_{sl}=0.2 and different inlet temperatures

On the basis of table 2 and figures 4 and 5 some considerations are possible. To obtain the same required temperatures, an ice-slurry solution can work with a higher inlet temperature than R404a (Table 2). When the entropy ratio is greater than one, the ice-slurry solution is preferable to R404a. The ice-slurry solution can exchange the same thermal power with different ice mass fractions and inlet-outlet temperature differences as figures 4 and 5 show. In particular, an entropy ratio increase is obtained by increasing the inlet temperature, with constant inlet-outlet temperature difference or, constant inlet-outlet ice mass fraction difference. Lower contribution causes ice fraction increase, figure 5, in comparison with inlet temperature increase.

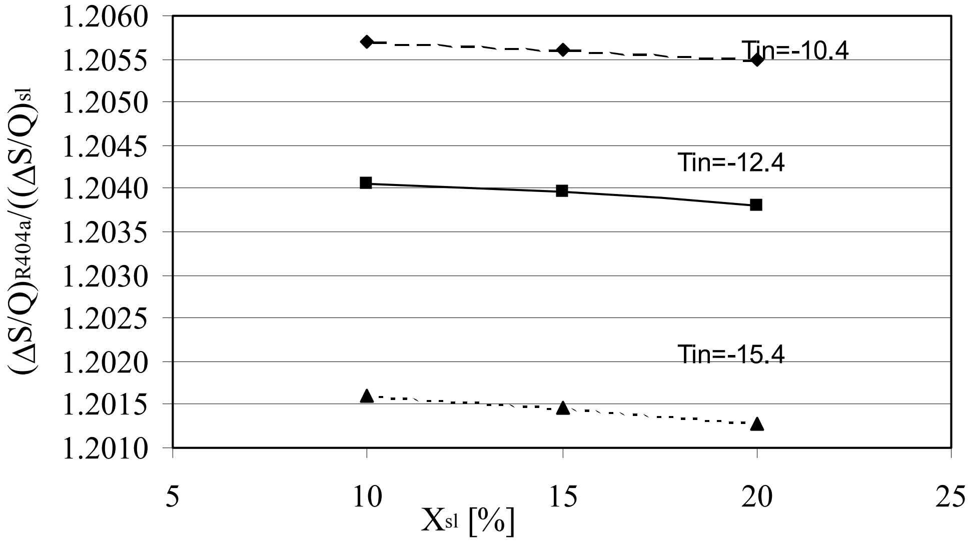

Figure 5 - $(\Delta S/Q)_{R404a}/(\Delta S/Q)_{sl}$ versus X_{sl} for ice-slurry solution with $\Delta T=0.5$ inlet-outlet temperature difference

The entropy production parameter proposed gives a simple criterion for choosing between different fluids when a fixed geometry heat exchanger is used and shows that ice-slurry is preferable to R404a, even if the ice mass fraction changes. On this basis it can be seen that, an ice mass fraction change does not influence operating conditions, for a particular range of values. In other words ice-slurry makes the plant more stable.

NOMENCLATURE

Latin

A	heat exchanger surface [m^2]
c_p	specific heat [J $kg^{-1}K^{-1}$]
D_i	inside pipe diameter [m]
D_e	external pipe diameter [m]
h	heat transfer coeff. [$WK^{-1}m^{-2}$]
L	pipe length [m]
$\dot{m}$	mass flow rate [kg/s]
n	fins number
p	pressure [Pa]
Q	heat power [W]
r	heat of fusion [J kg^{-1}]
Re	Reynolds number
S	entropy [J K^{-1}]
T	temperature [K]
U	global heat transfer coeff. [W $K^{-1}m^{-2}$]
V	specific volume [m^3 kg^{-1}]
X	ice mass fractions

Greek symbols

β	volumetric expansion coeff. [K^{-1}]
η	finned surface effectiveness
λ	heat conductivity [$WK^{-1}m^{-1}$]
φ	volumetric ice fraction
ρ	density [kgm^{-3}]

Subscripts

c	cold
e	external
fl	fluid
g	gas
h	hot
i	internal
in	inlet
out	outlet
s	ice
sl	ice-slurry

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