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Optimum Circuit Tube Length and Pressure Drop on the Refrigerant Side of Evaporators

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OPTIMUM CIRCUIT TUBE LENGTH AND PRESSURE DROP ON THE REFRIGERANT SIDE OF EVAPORATORS.

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ABSTRACT

A simple equation is derived to establish conditions for optimum pressure drop -- corresponding to an optimum circuit tube length and exit vapor velocity -- for evaporators. A design according to this relation will result in the *highest exit evaporator pressure* for *given heat source temperature, given heat flux and given evaporator tube type*. From a thermodynamic point of view the solution is equivalent to conditions for a minimum entropy production.

Provided that the influence of the tube length on the heat transfer and the pressure drop is known the result is general in its validity. For *smooth tubes* several well known relations exist and by using the relations of Pierre (1964, 1969) the result indicates that the optimum pressure drop (expressed as a drop of saturation temperature) is about 1/4 of the mean temperature difference between wall and refrigerant. Finally tubes with heat transfer enhancement devices are discussed briefly.

No consideration has been made to account for the influence of improper refrigerant distribution to different circuits which sometimes is a major problem in practice.

INTRODUCTION

In the design of an evaporator for a given total capacity we generally have options to design the evaporator with a suitable number of circuits in parallel (n_c). For a given tube type and given total tube length, $L_{tot} = n_c \cdot L$, the best choice of circuit length must be the one which will give us the highest exit pressure from the evaporator.

Fig. 1 illustrates in a simplified way the problem. Notice that we are considering evaporators of such capacities that it generally is not sufficient with only one circuit.

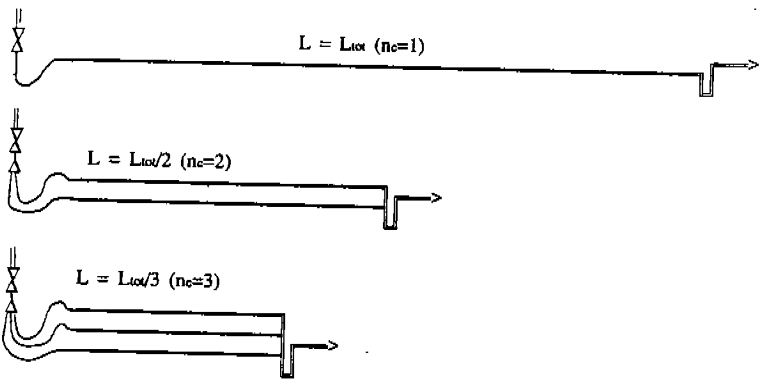


Figure 1. Illustration of three possibilities to arrange an evaporator with one, two or three tube circuits with the same total tube length. The best arrangement must be the one that results in the highest pressure at the evaporator outlet. This arrangement has circuit tube length as close to optimal tube length (L_{opt}) as possible.

For complete evaporation inside tubes the mass flow through the tube will be proportional to the tube length, if the heat flux is given. Long tubes will hence give a large mass flow through the tube which is beneficial since it increases the boiling heat transfer. However a too long tube will give excessive pressure drop, thereby reducing the refrigerant vapor pressure at the tube exit. Obviously a certain tube length can be found -- an "optimal" tube length -- which will deliver the highest possible exit pressure for a given operating condition and a given type of tube. The pressure drop corresponding to this tube length will be called an "optimal" pressure drop.

The problem has been treated in an earlier paper by the author (Granryd 1966). In that paper a somewhat different approach was applied but the results are identical for *smooth* evaporator tubes. The treatment given now is applicable also for the general case of tubes with different types of *heat transfer augmenting* devices, provided that the influence of the tube length on heat transfer and pressure drop is known.

An important practical problem is to achieve an equal refrigerant distribution in the different circuits of the evaporator. Quite often there are uneven distribution in the circuits and this will have as a result that some of the tubes are not fully utilized. In this treatment no provisions have been introduced to take into account the effects of uneven distribution.

TEMPERATURES ALONG AN EVAPORATOR TUBE

Consider the temperature along an evaporator tube as illustrated in Fig. 2a - b. Fig. 2a shows a simplified case with a constant temperature, t_B, of the outside fluid (*b*rine). The saturation temperature of the refrigerant will decrease along the tube due to the refrigerant flow pressure drop. The outside fluid will generally also change its temperature along the tube. In Fig. 2b cases of co current and counter current flow are illustrated as well as the special case of a constant temperature for the outside fluid. The penalty of the temperature drop of the refrigerant (due to the pressure drop) may be different in these cases and results of a treatment for this will be shown. (Similar considerations can be made for the use of Non Azeotropic Refrigerant Mixtures.)

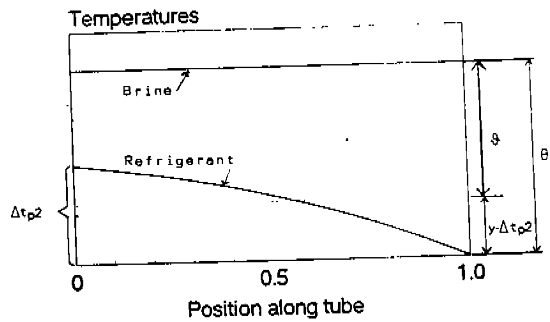


Figure 2a.
Temperature distribution along an evaporator tube, simplified case with constant external fluid temperature.

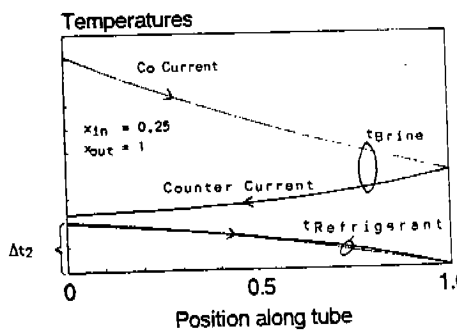


Figure 2b.
Temperature distributions for different arrangements with co-current and counter current flow.

The temperature difference of significance for the compressor operation of a heat pump or a refrigerating plant is the temperature difference, θ, between the outside fluid and the saturation temperature *corresponding to the pressure at the evaporator tube exit* (see Fig. 2a). We can express the temperature difference θ by the following equation:

$$\theta = q/U + y \cdot \Delta t_{p2} \qquad (1)$$

where q = heat flux

U = overall coefficient of heat transfer of the evaporator

Δt_{p2}= saturation temperature drop caused by the pressure drop in the evaporator tube

$y = \Delta t_{p2m}/\Delta t_{p2}$ = factor introduced to take into account the shape of the pressure drop curve along the tube, (see Fig. 2a).

The *overall heat transfer coefficient*, U, is related to the *refrigerant side* heat transfer coefficient, α, and to the heat transfer resistant, R, of the *wall* and the *external fluid*. It can be expressed as:

$$1/U = 1/\alpha + R \qquad (2)$$

where here, for simplicity, the thermal resistance, R, will be assumed to be constant.

The *factor y* can be estimated from the following equation, which is evident from the definition (t_B denoting external fluid temperature, t_R refrigerant temperature, t_{Rout} refrigerant saturation temperature at evaporator exit):

$$y = \frac{\Delta t_{p2m}}{\Delta t_{p2}} = \frac{\int_0^A (t_B - t_R) \cdot dA}{\int_0^A (t_B - t_{Rout}) \cdot dA} \qquad (3)$$

The factor y hence indicate the portion of the the pressure drop which is "lost" from a heat transfer point of view. A few different temperature profiles along the evaporator tube are shown in Fig. 2b. Values of the factor y has been calculated by numerical integration along the heat exchanger, with results as given in Fig. 3. (The integration is done under

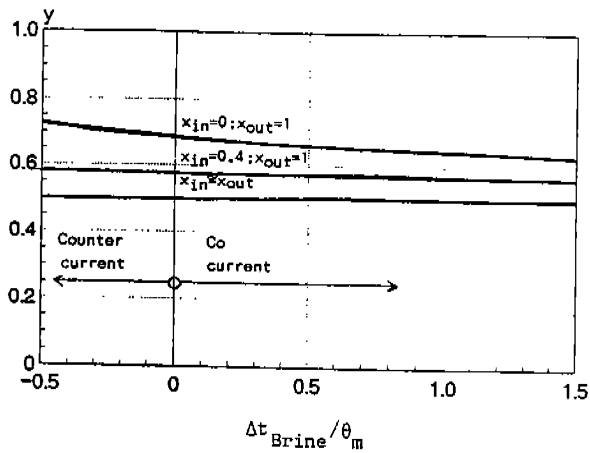


Figure 3.
Factor y achieved by numerical integration. For most practical cases one can use y ≈ 0,6.

the simplifications that the heat transfer coefficient is assumed constant along the tube, that the specific heat of the outside fluid is constant and that the pressure drop is approximately proportional to the vapor quality in each section of the tube.) A low value of y is beneficial, since it means that a smaller portion of the pressure drop really is lost from the heat transfer point of view. Fig. 3 gives curves for different inlet vapor quality. The horizontal axis indicates cases with different temperature change on the brine side.

Co-current flow arrangements are defined as giving positive difference (t_{Bin}-t_{Bout}) see Fig. 2b. As is seen from Fig. 3 the larger the temperature change is on the brine side in relation to the mean temperature difference the smaller y-value will result, but the influence is quite small. The vapor inlet quality will have a significant influence but for most cases the inlet vapor quality will be between 0 and 0,4 for which results are given in Fig. 3. As is seen one can quite often use a value of $y \cong 0{,}6$.

BASIC RELATIONS

For given parameters, (given heat flux, tube diameter, refrigerant properties etc), we can write:

Coefficient of heat transfer:

$$\alpha \cong \text{Constant}_1 \cdot L^{n\alpha}$$

Pressure drop:

$$\Delta p \cong \text{Constant}_2 \cdot L^{np}$$

The pressure drop can be converted into an equivalent loss of saturation temperature, Δt_2. Clapeyrons equation gives

$$\partial t/\partial p = T \cdot \frac{v''-v'}{r}$$

by which we can write:

$$\Delta t_2 = \Delta p \cdot T_2 \cdot \frac{v''-v'}{r} \qquad (4)$$

where T_2 = evaporating temperature (in absolute temperature scale, K)
v'' = spec volume of saturated vapor
v' = spec volume of saturated liquid
r = latent heat of vaporization.

The results of these equations can obviously be expressed in the following convenient form,

$$\alpha = \alpha_o \cdot (L/L_o)^{n\alpha} \qquad (5)$$

$$\Delta t_2 = \Delta t_{2o} \cdot (L/L_o)^{np} \qquad (6)$$

where α_o and Δt_{2o} are the heat transfer coefficient and the temperature change in the evaporator tube due to the pressure drop for a *reference case with tube length $L=L_o$.*

Introducing the expressions 2, 5 and 6 into eq. 1 we can write:

$$\theta = \frac{q}{\alpha_o \cdot (L/L_o)^{n\alpha}} + q \cdot R + y \cdot \Delta t_{2o} \cdot (L/L_o)^{np} \qquad (7)$$

where $q \cdot R$ will be regarded as constant.

Fig. 4 illustrates for an example how the tube length L influences the temperature difference θ by using eq. 7 (external thermal resistance is neglected, $R = 0$). The example is calculated for a case with a heat flux on the refrigerant side of $q = 5\ kW/m^2$ and with tube inner diameter $d_i = 11$ mm. Let us assume that we want to design an evaporator for a total

Figure 4.
Exit evaporator temperature difference for an example.
Data: R22; t_2 = 0 °C;
d = 11 mm; x_{in} = 0,25;
x_{out}=1; y = 0,6;
F = 0.02; q = 5 kW/m^2.

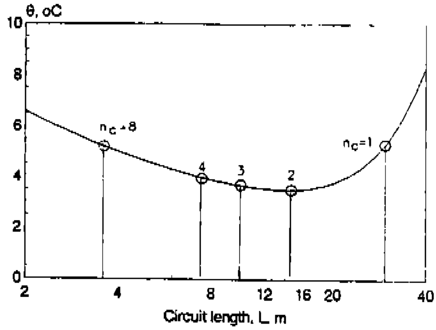
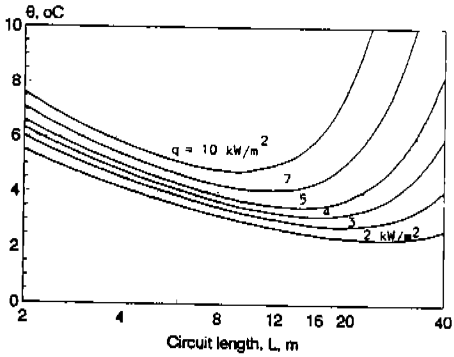


Figure 5.
Exit evaporator temperature differences for evaporator tubes for the same example as in Fig. 4 but with different heat flux.



capacity of Q_2 = 5 kW. This means that (with heat flux 5 kW/m^2) a total tube length of L_{tot} ≈ 29 m will be necessary. As is seen an arrangement with 2 circuits, each with 14,5 m of length will result in the smallest exit temperature difference, θ. This will be very close to the optimum and a further division into more circuits will not be beneficial.

In Fig. 5 curves are shown for the same example but with curves different heat flux. As expected the optimum tube length will be shorter if larger heat flux is used.

RELATIONS FOR PRESSURE DROP CORRESPONDING TO OPTIMUM TUBE LENGTH

It is easy to find an optimum tube length from eq. 7, by setting ∂θ/∂(L) = 0. The result can in the simplest and most general from be expressed as the equivalent temperature drop, $\Delta t_{p2_{opt}}$:

$$\Delta t_{p2_{opt}} = \frac{\vartheta}{y}\left(\frac{n\alpha}{n_p}\right) \qquad (8)$$

where $\vartheta = q/\alpha$ = the refrigerant side heat transfer temperature difference at optimum point conditions (tube wall to refrigerant)

$\left(\frac{n\alpha}{n_p}\right)$ = ratio of exponents defined by equations 5 and 6

y = "shape" factor, most often ≈ 0,6, see eq 3 and Fig. 3.

SMOOTH TUBES

For smooth tubes several different relations for the coefficient of heat transfer, α, and for the pressure drop, Δp, can be found in the literature. One of the most commonly used relations, giving average values for the heat transfer coefficient in tubes are those given by Pierre (1964, 1969). These relations can give us values for the constants of eq.s 5 and 6. The Pierre relations can be summarized as

$$\alpha = \frac{\lambda}{d} \cdot 0{,}01 \cdot Re^{0{,}8} \cdot K_f^{\,0{,}4} \qquad (9)$$

$$\Delta p = F \cdot \left(\frac{\dot{m}}{A}\right)^2 \cdot v_m \cdot \frac{L}{d} \qquad (10)$$

where λ = refrigerant liquid thermal conductivity
d = tube diameter
L = tube length
$Re = \frac{\dot{m} \cdot d}{A \cdot \mu}$ = Reynolds number (μ dynamic viscosity of liquid)
$K_f = \left(\frac{\Delta h}{g \cdot L}\right)$ = Pierre boiling number

F = a total friction factor (including influence of wall friction in tubes, tube bends and acceleration)
v_m = average vapor specific volume (for practical cases $v_m \cong 4{,}4 \cdot \frac{d^{0{,}25}}{L^{0{,}5}} \cdot v''$)

$\dot{m}$ = mass flow rate in the tube
A = tube cross sectional area ($= \pi \cdot d^2/4$)
Δh = refrigerant enthalpy change along the tube
g = acceleration due to gravity, (g = 9,81 m/s^2 in SI-units)

It is important to notice that the mass flow rate, $\dot{m}$, in a tube or a channel of an evaporator is coupled with the heat flux, $\dot{q}$, and the tube length L. The cooling capacity, per circuit of the evaporator, $\dot{Q}_2/n_c$, is given by:

$$\dot{Q}_2/n_c = \dot{m} \cdot \Delta h = \dot{q} \cdot \pi \cdot d \cdot L \qquad (11)$$

where Δh = enthalpy change for the refrigerant
d = tube (or channel hydraulic) diameter

By using these relations for smooth tubes (disregarding the influence of Reynolds number on the friction factor F for calculating the pressure drop in the tube) the exponents of equations 2 and 3 are

$$n_\alpha = 0{,}8 - 0{,}4 = 0{,}4 \qquad (12)$$

$$n_p = 2 + 1 - 0{,}5 = 2{,}5 \qquad (13)$$

It should be mentioned that the Pierre relation originally was developed from tests with R12, R22 and R502. However they have been checked in many different tests and seem to be applicable for most refrigerants in use. As an example it can be mentioned that results recently were published from tests with R134a, Hambraeus (1991), and equivalent agreements have been achieved with R152a.

"OPTIMUM PRESSURE DROP" FOR SMOOTH TUBES

The data in the preceding paragraph will give $\left(\frac{n_\alpha}{n_p}\right) \approx 0{,}4/2{,}5 \approx 0{,}16$. If we further use the shape factor y = 0,6 then eq. 8 will result in the following very simple expression for the optimal pressure drop:

$$\Delta t_{p2_{opt}} = 0{,}27 \cdot \vartheta \qquad (14)$$

To exemplify: This means that if the temperature difference due to heat transfer on the refrigerant side is 5 degrees C, then the evaporator tube length should be chosen in such a fashion that the pressure drop corresponds to about $0{,}27 \cdot 5 \approx 1{,}3$ °C.

As a rule of thumb: Design for a pressure drop of about 1/4 of the temperature difference on the refrigerant side for smooth tubes.

OPTIMUM CIRCUIT TUBE LENGTH, AND ASSOCIATED EXIT VAPOR VELOCITY

The result derived and expressed by equation 14 can be transformed to an expression for the equivalent, *optimal tube length.* By using the equations 9 and 10 the result will be as follows.

$$L_{opt} = C_L \cdot \frac{d^{1,29} \cdot \Delta x^{0,83}}{q^{0,62} \cdot (y \cdot F)^{0,344}} \quad [m] \qquad (15)$$

where: d = tube diameter (m)
q = heat flux (W/m^2)
F = a total friction factor for the evaporator tube. With oil-free refrigerants one can often use $F = 0{,}02\text{-}0{,}03$ while for cases with oil $F = 0{,}04 - 0{,}06$.
Δx = vapor quality change in evaporator tube

$$C_L = 0{,}56 \cdot \left[\frac{\mu^{0,8} \cdot r^{3,4}}{\lambda \cdot (v''-v') \cdot v'' \cdot T_2}\right]^{(1/2,9)}$$

For a few commonly used refrigerants values for the constant C_L are given in Fig. 6. The cooling capacity per circuit will be

$$Q_{2\cdot opt} = \pi \cdot d \cdot q \cdot L_{opt} \qquad (16)$$

and we can use this result to express the velocity corresponding to the optimum tube length. This will give:

$$w_{opt} = C_W \cdot \frac{q^{0,38} \cdot d^{0,29}}{\Delta x^{0,17} \cdot (y \cdot F)^{0,344}} \quad [m/s] \qquad (17)$$

where $C_W = C_L \cdot 4 \cdot v''/r$.

The diagram of Fig. 7 give values of C_W for a few refrigerants.

Example:

Assume a tube inner diameter, $d_i = 0{,}011$ m (11 mm); a heat flux of $q = 5000$ W/m^2; and an inlet vapor quality of 0,25 to the evaporator (which gives $\Delta x = 0{,}75$). Let us further assume that the application allow us to use a factor $y \approx 0{,}6$ (see Fig. 3) and a total friction factor $F \approx 0{,}02$.
Refrigerant R22 is used, with evaporating temperature $t_2 = 0$°C. For this refrigerant Fig. 6 and 7 gives $C_L \approx 2{,}64 \cdot 10^5$ and $C_W \approx 0{,}24$.
These numbers used in equation 15 give:

$$L_{opt} \approx 14{,}4 \text{ m}$$

while eq 17 gives:

$$w_{opt} \approx 8 \text{ m/s}$$

which will be the exit velocity if the circuit length = L_{opt} is used and if the other conditions of the example prevails.

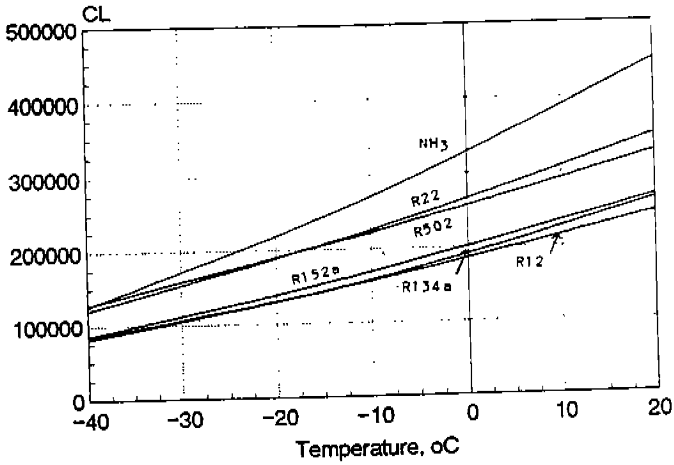


Figure 6. The refrigerant property dependant factor CL to be used in equation 15 to estimate the optimum tube length.

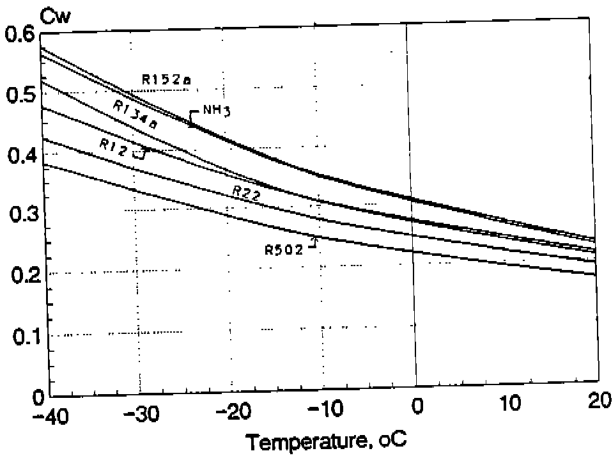


Figure 7. Factor Cw for different refrigerants to be used in eq 17 to estimate vapor velocity at the exit for optimum tube length circuits.

The simple equation 8 is valid also for this case. However, for enhancement structures the exponents of equations 5 and 6, probably are different compared to cases of smooth tubes. Correct values for exponents $n\alpha$ and n_p are difficult to establish without experimental data. Examples of experimental results are given in a recent publication, Agrawal & Varma (1991). An analysis of their data indicate that a somewhat larger exponent $n\alpha$ and a somewhat smaller exponent n_p than for the case of smooth tubes, although the presentation does not permit the comparison to be conclusive. (The conditions are not quite comparable.)

For a "limiting" case, which might be applicable to an evaporator surface with a turbulating structure, let us for the purpose of illustration assume that the exponents $n\alpha \cong 0.8$ and $n_p \cong 2.8$. This would correspond to turbulent convective heat transfer and pressure drop relations (experiment of the Reynolds number equivalent to $n\alpha$). Using, as previously, $y = 0.6$, then eq. 8 will give the result:

$$\Delta t_{p2_{opt}} \cong 0.48 \cdot \vartheta \qquad (18)$$

If we know the influence of the enhancement element in the tube on the heat transfer and the pressure drop we may consider to use this relation as a tool for choosing the enhanced tubes in the most effective way. This is best illustrated by means of an example:

Example:

Let us assume that we have a method to enhance the heat transfer in an evaporator tube which, compared to a smooth tube (with heat transfer coefficient α_o^*), will increase the heat transfer coefficient by a factor e_h, to:

$$\alpha = \alpha_o^* \cdot e_h$$

but also increases the pressure drop from Δp_o^* for the smooth tube by a factor e_f, to:

$$\Delta p = \Delta p_o^* \cdot e_f$$

The resulting exit temperature difference is

$$\theta = \frac{q}{\alpha_o^* \cdot e_h \cdot (L/L_o)^{n\alpha}} + q \cdot R + y \cdot \Delta t_{po}^* \cdot e_f \cdot (L/L_o)^{n_p}$$

For an example with

$$e_h = 2; \quad e_f = 4$$

the result is illustrated in Fig. 8, together with some equivalent examples for smooth tubes.

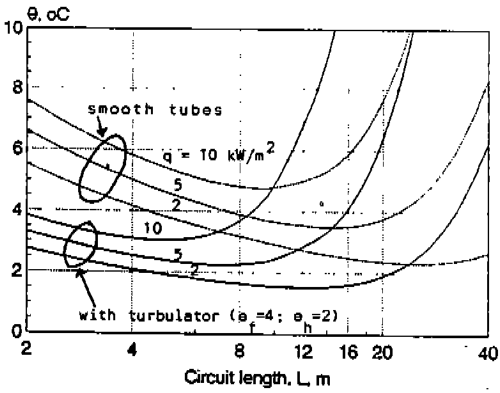


Figure 8.
Exit temperature difference for configuration with and without heat transfer augmenting devices for an example.

The same base values (ϑ_o^* and Δt_{po}^*) as for the smooth tube have been used; however the values of the exponents $n\alpha$ and n_p, are changed for the enhanced tubes to values as previously discussed ($n\alpha$=0,8; n_p = 2.8).

As is seen optimum conditions occur at a shorter tube length than for the case with the *smooth* tube. As is shown the optimum tube length for the previous example with the smooth tube and heat flux of 5 kW/m^2 was 14,8 m. If that length would be used also for the augmented evaporator design then this will give *no* improvement, see Fig. 8. An optimal design for the augmented design would be to use a tube length of about 8 m for this example, as seen in Fig. 8. One can show that the optimum conditions as illustrated in Fig.s 5 and 8 satisfies eq. 14 (or eq.15) and 18, respectively.

CONCLUSION AND DISCUSSION

A simple equation is derived to establish conditions for optimum tube length -- or expressed in another way -- optimum pressure drop, for evaporators. This condition will result in the highest *exit* evaporator pressure for given temperature on the heat source and given heat flux. From a thermodynamic point of view the solution is equivalent to a solution for a minimum entropy production.

The result as expressed in eq. 8 is valid for any kind of evaporator, but requires that one knows the exponents $n\alpha$ and n_p (defined in eq.s 5 and 6). However it can be shown that the final result is rather insensitive to deviations from the strict optimum conditions. Results are exemplified in Fig.s. 5 and 8. Expressions for the tube length and exit vapor velocity equivalent to the pressure drop are given for smooth tubes in eq. 15 and 17. Required data for different refrigerants are given in Fig.s 6 - 7 for quick calculations.

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