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Alberto Cavallini has played a key role in the organization of events such as the XVth International Congress of Refrigeration of the IIR (in Venice in 1979), and "CFCs, The Day After," a conference organized by Commissions B1, B2, E1 and E2 in Padua in 1994. He was Deputy President of the International Scientific Committee of Clima2000, held in Naples in September 2001.

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Alberto Cavallini is author or co-author of over 160 scientific and technical publications, mostly published in the scientific press, and of 5 textbooks used in Italian universities. He is an adviser to several corporations in the industry, and is active in the field of design of refrigerating and air-conditioning plants in the private sector.

Heat Transfer and Energy Efficiency of Working Fluids in Mechanical Refrigeration

by

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I. CYCLE ANALYSIS

When investigating, from the point of view of thermodynamic properties, the suitability of a new fluid as a working agent for mechanical (vapour compression) refrigeration, the first step is to calculate the value of the COP reached in a simple (ideal reference) refrigerating cycle between appropriate temperature levels (condensation and evaporation). If this compares favourably with COPs achieved by the established refrigerants in the same reference cycle, the new fluid can be considered promising.

Apart from the fact that many other characteristics are required from a working fluid in mechanical refrigeration, this test cannot be considered decisive even from the sole point of view of energy efficiency; many other factors, in this respect, play key roles in practical applications. Besides, appropriate cycle modifications (staged compression, staged throttling, presence of liquid-gas internal heat exchanger) can reduce differences or change energy performance ranking among different working fluids. Nevertheless, theoretical cycle analyses can help prioritize suitable candidates and disregard those with significant lower performance.

Instead of simply considering COP values, one can refer to the exergetic efficiencies of the reference cycle, thus gaining additional insight into the intrinsic losses consequent upon the thermodynamic behaviour of a working fluid.

Figure 1 depicts the reference theoretical refrigerating cycle (1-2-3-4-5-1) in a T-s (absolute temperature–specific entropy) diagram, with constant pressure condensation and evaporation processes, isentropic compression and constant enthalpy throttling.

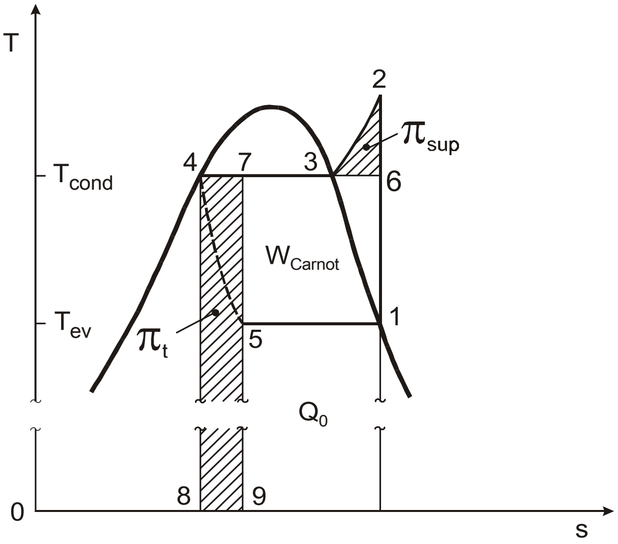

Figure 1. *Simple theoretical vapour-compression refrigerating cycle in the diagram temperature-entropy. Hatched areas represent exergy losses*

The refrigerating effect Q_0 is represented by area 1.5.9.10.1 in the diagram; ideally, the same refrigerating effect between the same temperature levels could be yielded by the Carnot cycle 1-6-7-5-1, that would require an external work equal to its area 1.6.7.5.1. This is therefore the minimum theoretical work necessary to produce the refrigerating effect Q_0. The actual work of the refrigerating cycle is given by area 1.2.3.4.8.9.5.1, exceeding the Carnot-cycle work by the two hatched areas, corresponding to irreversibility losses: in terms of exergy, π_t (area 4.7.9.8) represents the throttling exergy loss, while π_{sup} (area 3.2.6) is the superheating exergy loss. For halocarbons and hydrocarbons, usually the throttling loss far outweighs the superheating loss; Professor Gustav Lorentzen dubbed the throttling loss the cycle haemorrhage.

For ammonia, the two exergy losses are of the same order. For some fluids, such as HCFC-123, the isentropic compression process 1-2 can end up within the saturated vapour zone; consequently, the superheating exergy loss is zero; this depends on the skew of the saturation boundaries in the T-s diagram.

The ratio between the minimum (Carnot) work and the actual work is defined as the exergetic efficiency of the refrigerating cycle. The exergetic efficiency is reported in the diagram of *Figure 2* for a few refrigerants used in air-conditioning applications, referred to a fixed evaporation temperature t_{ev}=0 °C, at increasing condensation temperatures t_{cond} from 20 to 70°C.

It can be noticed that, approximately, the higher the critical temperature, the higher is the exergetic efficiency of the theoretical cycle. The critical temperature is in fact, together with the molar heat capacity, the parameter that mostly influences the theoretical energy performance of a refrigerant. Operating close to the critical temperature, in fact, greatly enhances throttling exergy losses, which are by far the prime reason for a degraded energy performance of the working fluids under scrutiny (ammonia being somewhat an exception to this general trend, as already mentioned). At the other end, operation with a low-pressure refrigerant, that is to say far from the critical point, while decreasing the throttling losses, reduces the volumetric refrigerating capacity of the fluid, making this unsuitable for use with positive-displacement type compressors.

The exergetic efficiency of a reference cycle (or even cycle COP), is often considered, with all limitations mentioned above, as a performance evaluation criterion for energy efficiency related to the use of a fluid as a working agent in mechanical refrigeration. In this event, the comparison among different working fluids is done between the same temperature levels (isobaric condensation and evaporation) of the reference cycle.

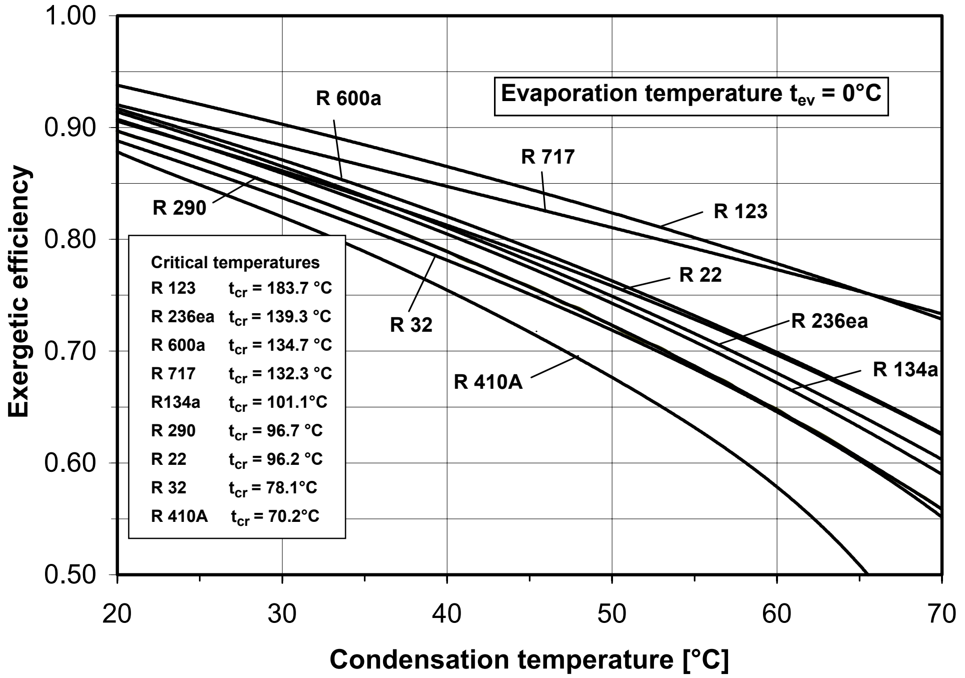

Figure 2. *Exergetic efficiency of the reference refrigerating cycle operated with different working fluids*

Generally speaking and referring to the same external constraints, improving the heat transfer performance of the heat exchangers (condenser and evaporator) of a refrigerating machine, brings about a reduction in the refrigerant mean condensation temperature, and a rise in the refrigerant mean evaporation temperature. These circumstances are both highly beneficial to the cycle energy efficiency.

This benefit can be approximately estimated by considering the relative variation of the reference cycle COP while varying one of the reference saturation temperatures, the other kept constant. Referring to an ideal reverse Carnot cycle, for the condensation temperature T_{cond}, this result can be easily obtained:

$$\left[\left(\frac{\partial COP}{\partial T_{cond}}\middle/COP\right)_{T_{ev}}\right]_{CARNOT} = -\left(T_{cond} - T_{ev}\right)^{-1} \tag{1}$$

This parameter is plotted in *Figure 3* for the ideal reverse Carnot Cycle, and for the reference cycles of the working fluids considered in the previous figure, as a function of the condensation temperature t_{cond}, for a constant evaporating temperature t_{ev}=0°C.

The same graph also holds good when considering refrigerating cycles with non ideal (non isentropic) compression, provided that the compression isentropic efficiency can be considered constant in the computation of expression (1).

A question may arise: is it possible to find out a suitable Performance Evaluation Criterion, linked to the properties of any refrigerant, able to evidence under appropriate otherwise equivalent conditions, the effects of the choice of different refrigerants on the condenser and/or evaporator thermal performance?

An attempt to positively answer this matter in the case of condensers with in-tube refrigerant flow is presented in the following of this paper.

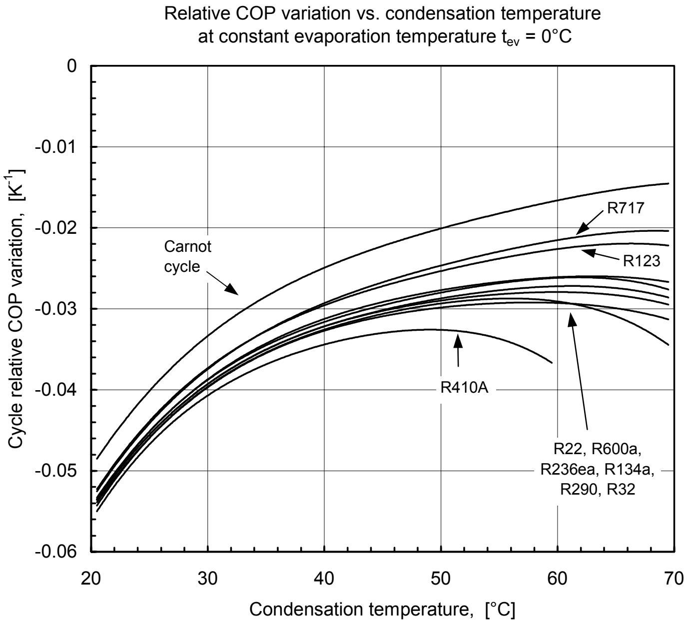

Figure 3. Relative variation of cycle COP with condensation temperature at fixed evaporation temperature t_{ev}=0°C

II. THE ROLE OF HEAT TRANSFER

In a real application, the temperature levels relevant to energy efficiency at fixed external constraints are the refrigerant saturation temperatures at compressor suction and discharge. They in fact determine the required compressor temperature lift (that is, the compressor power). If saturation temperature variations associated with refrigerant pressure drops in suction and discharge lines can be neglected, the relevant temperatures to be considered are those of saturation at condenser inlet and evaporator outlet.

Figure 4 depicts the qualitative saturation temperature profile of a refrigerant (pure fluid or azeotropic mixture) in a counter-current condenser (with tube-side condensation) of a refrigerating machine; the temperature profile of the external cooling agent (air, water…) is also plotted in the same graph. The refrigerant saturation temperature is not constant along the condenser, because the refrigerant saturation temperature drop, consequent upon the refrigerant pressure drop, must be considered in this situation. In fact, it cannot usually be neglected, especially if the goal is to compare different refrigerants in a given exchanger geometry. Contrary to single phase flow, pressure drop in two-phase flow affects the temperature profile of the fluid in the exchanger, with consequences in the mean effective temperature difference.

For simplicity's sake, reference can be made to the mean temperatures of both fluids in the condenser, as shown in the figure. Hence one can approximately conclude that the refrigerant inlet saturation temperature is affected by the sum of the required overall heat transfer driving temperature difference DT_{do}, and by half of the refrigerant saturation temperature drop DT_{sr} in the exchanger. This follows from the consideration that only about one half of the refrigerant saturation temperature drop is actually lost as heat transfer driving potential.

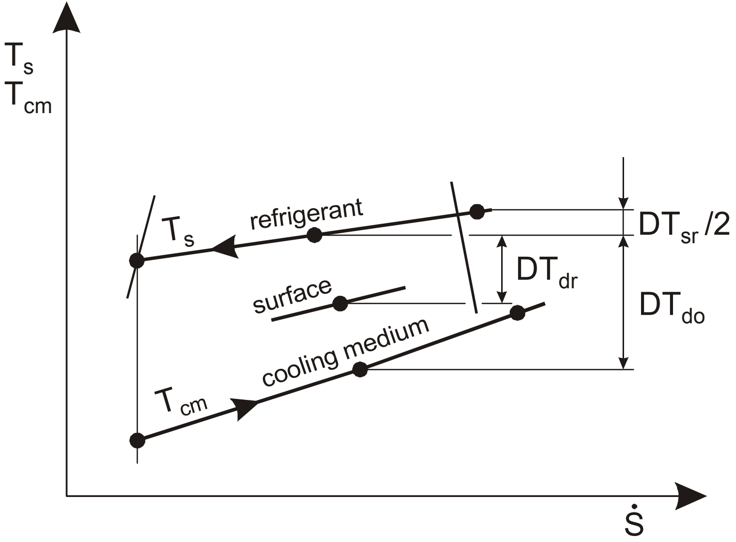

Figure 4. *Temperature profiles in a condenser of a refrigerating machine*

By minimizing, under appropriate constraints, the above sum involving the two penalising components, an optimal condition can be established for the condenser thermal performance.

Under fixed operating conditions for the cooling agent of a condenser of definite size and overall surface geometry and amount of heat exchange surface area, and for a prefixed thermal duty, the choice of different refrigerants influences both DT_{do} and DT_{sr}.

Same heat exchange area and same heat duty mean same average heat flux, that in turn means same average external component (tube wall + external fins, if present + cooling medium + scale, if applicable) of the overall driving temperature difference. Therefore, in this situation, relevant to the optimization of the condenser thermal performance is only the condensing refrigerant side driving temperature difference DT_{dr} (from saturated refrigerant to inner pipe wall). This only depends, under the condition of fixed mean heat flux, on the refrigerant convective heat transfer coefficient α_i.

The second term penalizing the condenser performance, DT_{sr}, depends on the frictional pressure gradient experienced by the refrigerant in the exchanger circuits; the pressure recovery due to the acceleration pressure gradient is in fact usually negligible. To be noted that DT_{sr} is to be computed for total vapour condensation in the condenser circuit, down to vapour quality $x=0$, this being the most common condition in real circumstances.

One degree of freedom is left to the condenser geometry moving from one refrigerant to another: within the constraint of a fixed total length, the number of circuits in parallel can vary. Thus, length and refrigerant mass velocity for any single circuit can vary, allowing optimization according to operating conditions and refrigerant heat transfer characteristics. *Figure 5* clearly illustrates how different number of circuits can be obtained in a condenser of fixed basic design.

The above external constraints can be used to compare the thermal performance for tube-side condensation of different refrigerants; the appropriate objective function in this case is the value of the refrigerant saturation temperature at condenser inlet: the lower, the better performing refrigerant.

It is to be recognised that other objective functions might be considered as well, associated with different constraining conditions. For example, different objectives (keeping appropriate parameters fixed) might be:

- to reduce heat transfer surface area for fixed operating temperatures;
- to increase heat exchange capacity for fixed amount of heat exchange surface area and operating temperatures.

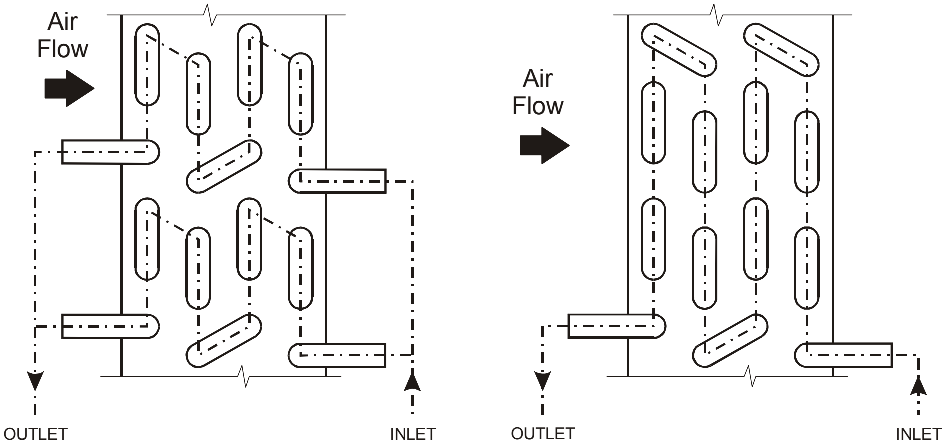

Figure 5. Different circuitry for a finned-coil condenser

III. THE PENALTY FACTOR

Recently, Cavallini et al[1] presented a new predictive model for heat transfer coefficient and frictional pressure drop applicable to halogenated refrigerants condensing inside horizontal smooth tubes in the inner-diameter range 3-21 mm. It was shown that this model, contrary to other well-established predicting procedures appeared in the open literature before the introduction into the market of the new substitute HFC refrigerants, can be confidently used even with the new synthetic fluids. Some of the new working fluids, such as R410A and R32, are characterized by a higher operating pressure than that of the old refrigerants (such as R22) they replace, and therefore are often dubbed ‘high-pressure’ working fluids.

During condensation inside horizontal tubes, the prevalent flow pattern encountered is forced convection annular flow, dynamically dominated by the interfacial shear stress at liquid vapour interface, while gravity forces can be neglected. Toward the end of the condensation process, with vapour quality approaching zero, the influence of gravity becomes more and more important, and the flow pattern changes to stratified configurations. A sound predicting procedure for heat transfer and pressure drop during in-tube condensation must of course appropriately consider the type of flow pattern locally present.

The new Cavallini et al. model, in the annular flow regime applies the Von Karman analogy between heat and momentum transfer to compute heat transfer coefficients from pressure drop data; other models apply the same approach.

This establishes a close link between heat transfer coefficient and frictional pressure gradient in forced convection condensation inside tubes. Through the application of the Clausius-Clapeyron equation, the refrigerant saturation temperature gradient can by calculated from the frictional pressure gradient and fluid properties as follows:

$$\left(\frac{\mathrm{d}T_s}{\mathrm{d}z}\right)=\left(\frac{\mathrm{d}p_f}{\mathrm{d}z}\right)\frac{T_s}{h_{lv}}\left(\frac{1}{\rho_v}-\frac{1}{\rho_l}\right) \qquad (2)$$

The combination of the above equation with the expression Cavallini et al suggested for the computation of the frictional pressure gradient in this circumstance, and considering the energy balance in an elementary length dz of the exchanger tube, yields:

$$PF=\left(\frac{\mathrm{d}T_s}{\mathrm{d}x}\right)\cdot DT_{dr}=DT_{sr}\cdot DT_{dr}=T_s\left(\frac{1}{\rho_v}-\frac{1}{\rho_l}\right)\Phi_{LO}^2\cdot f_{LO}\frac{G^3}{2\rho_l\cdot\alpha_i} \qquad (3)$$

In the above expressions h_{lv} is the latent heat of condensation, α_i is the convective heat transfer coefficient, ρ_v and ρ_l are the densities of vapour and liquid phases respectively, G is the refrigerant mass velocity, Φ^2_{LO} and f_{LO} are the two-phase frictional multiplier and the friction factor respectively, both referred to the liquid phase with total flow rate.

The parameter PF is dubbed Penalty Factor, as it is the product of the two components penalising the condenser inlet saturation temperature, as discussed above. It depends on the very same variables the heat transfer coefficient depends on, that is (in shear dominated condensation), on the fluid local thermophysical properties, inside pipe diameter d, refrigerant mass velocity G and local vapour quality x.

To compare different refrigerants, in respect to their potential thermal performance in condensers, on the basis of values of penalty factors at equal mass velocity (besides tube diameter, reference vapour quality and saturation temperature) is meaningless, because equal mass velocity is an arbitrary constraint, never enforced in real applications.

A plot of the penalty factor against the heat transfer coefficient provides by far a much better representation of the potential performance of different refrigerants in condensation heat transfer. This is done in the graph of *Figure 6*, which refers to a smooth tube with inner diameter 8 mm, with condensation of refrigerants under forced convection (annular flow pattern) at 45°C saturation temperature and vapour quality x=0.5. To evidence the meaning of the penalty factor PF, the graph can be read in the following way. When a fixed duty condenser is operated with different refrigerants at the same condensation heat transfer coefficient, the dissipative saturation temperature drop (for complete condensation) is in proportion to the penalty factor of the refrigerant. The penalty factor PF can therefore be considered a Performance Evaluation Criterion of a refrigerant as far as its behaviour in condensation heat transfer within the considered geometry is concerned: the lower the penalty factor PF, the better the potential performance of the refrigerant is. The same ranking among the examined refrigerants as it appears in *Figure 6* is maintained when referring to different smooth tube diameters and different condensation temperatures, within the ranges of practical interest.

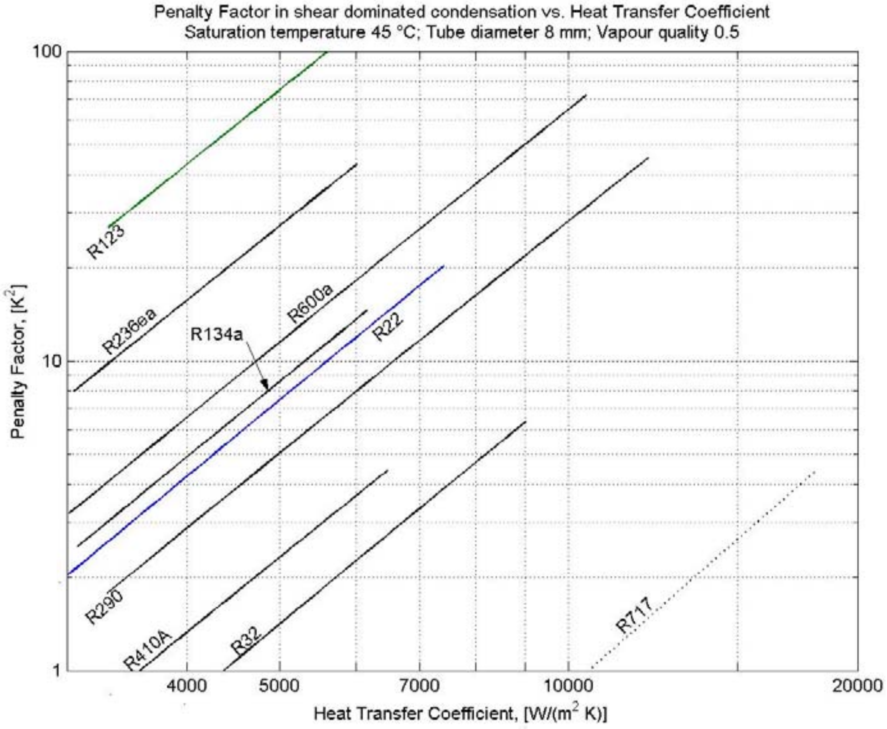

Figure 6. *Penalty Factor vs. Heat Transfer Coefficient for different working fluids*

From the graph in *Figure 6*, it can be observed that the ranking of the refrigerants goes almost exactly in reverse order of their critical temperatures. The low critical temperature fluids (high-pressure refrigerants) are the best performers in this respect, contrary to what happens for the reference cycle COP (or exergetic efficiency). The only noteworthy exception to this rule is R717 (ammonia), which displays excellent properties both with respect to cycle COP and condensation penalty factor PF.

Attention must be called to the fact that all the penalty factors plotted in *Figure 6* have been calculated using the predicting model recently proposed by Cavallini et al, as earlier discussed. This model has been extensively validated for HCFC/HFC refrigerants, and proved to work also with the HC fluids here considered. On the contrary, for ammonia no reliable experimental data is available at present (at least in the open literature) for the validation of the calculation procedure here employed. While ammonia data reported in *Figure 6* can perhaps be questioned quantitatively, there cannot be any question that the trend depicted for this fluid is the correct one.

IV. OPTIMIZING CONDENSER DESIGN THROUGH THE PENALTY FACTOR

As shown in the graph of *Figure 6,* the representation of penalty factor PF vs. heat transfer coefficient α_i for different refrigerants in a log-log diagram appears as a series of almost-parallel straight lines. The same happens for other temperatures, tube diameters and vapour qualities, within ranges of practical interest, as long as fully forced convection condensation is considered.

Therefore the dependence of the penalty factor on the condensation heat transfer coefficient (in annular flow pattern) can be represented by an equation in the general form:

$$PF = C_{Ref} \cdot (\alpha_i)^m \tag{4}$$

where the leading coefficient C_{Ref} depends on the refrigerant properties, value of vapour quality and tube diameter considered; the exponent m comes out to be a very weak function of the same parameters. *Table 1* lists the best-fit values of C_{Ref} and m for the refrigerants considered in Figure 6, in the same ranges of the plot.

Table 1. Leading coefficient C_{Ref} and exponent m in eq. (4) to calculate the Penalty Factor PF of some refrigerants at saturation temperature T_s=45°C, vapour quality x=0.5, for forced convection condensation in a smooth tube with inner diameter d=8 mm. SI units to be used

Refrigerant	***C_{Ref}***	***m***	***Refrigerant***	***C_{Ref}***	***m***
HCFC R123	$4.922 \cdot 10^{-8}$	2.482	HFC R32	$5.023 \cdot 10^{-10}$	2.555
HCFC R22	$2.821 \cdot 10^{-9}$	2.547	HC R290	$3.071 \cdot 10^{-9}$	2.491
HFC R236ea	$1.727 \cdot 10^{-8}$	2.487	HC R600a	$6.207 \cdot 10^{-9}$	2.505
HFC R134a	$3.936 \cdot 10^{-9}$	2.524	(R717)	$1.057 \cdot 10^{-11}$	2.729
HFC R410A	$1.246 \cdot 10^{-9}$	2.506			

Let's now consider a condenser such as an air-cooled finned coil, of defined overall geometry, fixed thermal duty, and fixed operating conditions for the cooling medium (mass flow rate and inlet/outlet temperatures). The only action left open to the designer for the optimization of such a heat exchanger is to vary the number of circuits in parallel for the refrigerant. This can be obtained using different ways of connecting the single tubes to each other on the sides of the apparatus, and with the inlet and outlet headers; the scheme in *Figure 5* clearly illustrates the problem. A similar condition can regard a shell-and-tube condenser with in-tube condensation, where the number of tube passes can be varied. It is assumed here that all circuits/tube passes are of equal length and operate under equivalent thermal conditions.

The total mass flow rate of refrigerant entering the condenser is considered fixed, as the total length of the tubes is, and therefore the refrigerant mass flux feeding any circuit is proportional to its length (that is, inversely proportional to the number of circuits in parallel). The longer the single circuit (that is, the less the number of circuits in parallel), the higher the mean condensation heat transfer coefficient α_i achieved and consequently the lower the necessary driving mean temperature difference DT_{dr}. At the same time, the longer the circuit, the higher is the refrigerant frictional pressure drop (both because of the longer path and the higher mass flux), and therefore the higher is the penalty component DT_{sr}.

Under the stated design constraints, an optimal length of the single circuit must exist, which results in the minimum saturation temperature at the condenser inlet, this being the objective of this optimization process.

This occurrence is well depicted by the graphs in *Figure 7*, where (top diagram) the refrigerant saturation temperature $(t_{sc})_{in}$ at condenser inlet is plotted against tube length, under the discussed design constraints and operating conditions specified in the figure itself. It is shown that, in this case, the optimal tube length is L_{opt}=14 m.

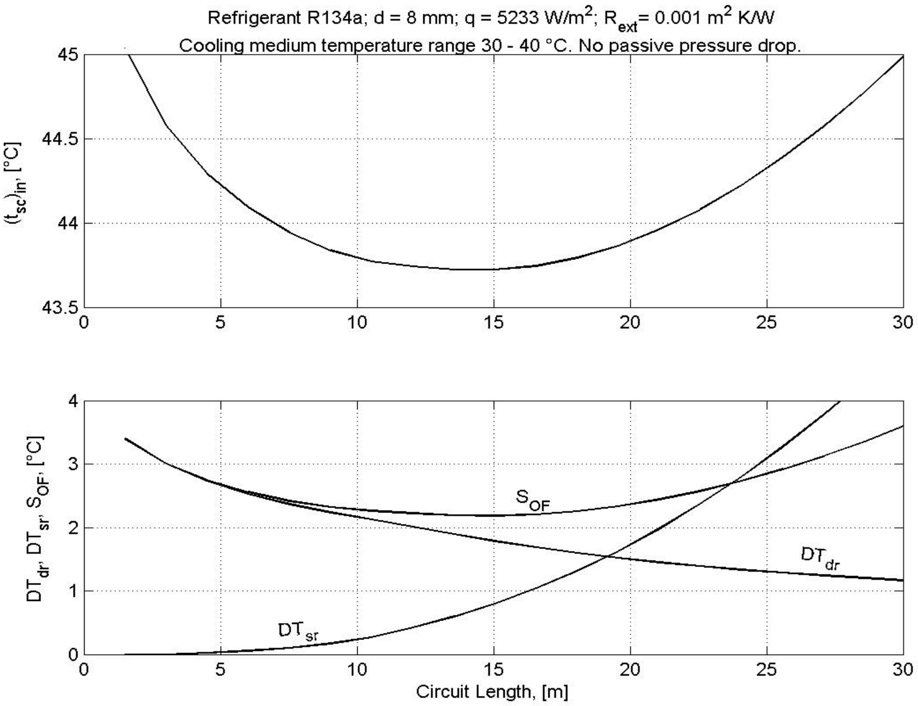

Figure 7. *Length optimization of condenser circuits*

It can also be appreciated from the same plot that strictly sticking to the optimal value of the tube length is not a critical condition for an optimized design, as the trend of the condenser inlet saturation temperature around the optimal value is rather flat. This is a general occurrence. It should also be considered that in the design of a real condenser, possible lengths of circuits go stepwise, and the practical length closer to the optimal one should be chosen.

The plots in the bottom part of *Figure 7* illustrate the trends, with tube length, of the two components affecting the refrigerant saturation temperature at condenser inlet, DT_{dr} and DT_{sr}. The additional plot on the same graph shows that the optimum for the circuit length closely matches the condition:

$$S_{OF}=(DT_{dr}+0.5\cdot DT_{sr})\Rightarrow \text{minimum} \tag{5}$$

as discussed in relation to *Figure 4*.

Under some simplifying assumptions, the optimal operating condition for a condenser subject to the above design constraints can be determined analytically involving the penalty factor PF of the refrigerant considered.

It is assumed that the operating conditions of a condenser can be treated and expressed in terms of suitable bulk average parameters referred only to the condensation process from vapour quality x=1 to x =0; the relevant parameters are calculated at the average condition x =0.5.

In this case the design constraints are:

- Average heat flux q constant;
- External heat transfer resistance (pipe wall + fins, if present, + cooling medium convective term + scale) R_{ex} constant;
- Fixed temperatures (inlet and outlet) of the cooling medium.

Then one can write (the meaning of the multiplying factor Bf will be clarified later; for the moment, consider Bf=1, therefore non-influential in the results):

$$DT_{dr}=q\cdot\alpha_i^{-1};\quad DT_{sr}=Bf\cdot PF/\,DT_{dr}=Bf\cdot C_{Ref}\cdot q^{-1}\cdot\,\alpha_i^{m+1} \tag{6}$$

The objective function S_{OF} can therefore be expressed as a function of the condensation heat transfer coefficient α_i, and the optimal value $(\alpha_i)_{opt}$ immediately found be equaling to zero the first derivative $\left(\partial S_{OF}/\partial\alpha_i\right)_q$. This procedure yields:

$$(\alpha_i)_{opt}=\left[\frac{q^2}{0.5\cdot(m+1)\cdot Bf\cdot C_{Ref}}\right]^{\frac{1}{m+2}} \tag{7}$$

The calculation of the design optimal value for the condensation heat transfer coefficient allows then computing the optimal value for all the other relevant design parameters, such as for example the optimal design driving temperature difference DT_{dr} (optimal mean temperature difference between saturated condensing refrigerant and pipe inner wall):

$$(DT_{dr})_{opt}=\left[0.5\cdot(m+1)\cdot Bf\cdot C_{Ref}\cdot q^m\right]^{\frac{1}{m+2}} \tag{8}$$

Furthermore, from the optimal design value of the condensation heat transfer coefficient one can calculate, given pipe diameter and saturation temperature, the refrigerant mass flux G_{opt} in the tube necessary to achieve the desired value of $(\alpha_i)_{opt}$; then the calculation of the optimal length of the circuit is straightforward.
Worth mentioning is the resultant optimized value for the ratio $DT_{dr}/\,DT_{sr}$:

$$\left(\frac{DT_{dr}}{DT_{sr}}\right)_{opt}=0.5\cdot(m+1) \tag{9}$$

As discussed before, the value of m is very similar for all refrigerants. Therefore the simple general rule can be established: an optimal design calls for a frictional saturation temperature drop about one half of the refrigerant-to-wall driving temperature difference. To be noted that, in an earlier study of the same problem related to evaporators, Granryd[2] reached the conclusion that the optimum frictional saturation temperature drop should be one quarter of the refrigerant-side driving temperature difference.

It is now possible to compare the thermal performance of different refrigerants condensing in the tube-side of exchangers of equal size and overall geometry, at same thermal duty and conditions for the cooling medium. Comparing the value of the objective function S_{OF} can do this, when for each refrigerant the optimized condenser layout as far as the number of circuits is considered. The relevant expression is:

$$\left(S_{OF}\right)_{opt} = \left[0.5\cdot(m+1)\cdot\left(\frac{m+3}{m+1}\right)^{m+2}\cdot Bf\cdot C_{Ref}\cdot q^{m}\right]^{\frac{1}{m+2}} \tag{10}$$

The optimized value of the objective function $(S_{OF})_{opt}$ given by the expression (10) is plotted in *Figure 8* (Bf= 2.5, to account for passive pressure loss. See last paragraph) against heat flux q for all the refrigerants considered before; the graph refers to the refrigerant saturation temperature t_s=45°C and the pipe inside diameter d=8 mm. As obvious, the same ranking among the refrigerants is established as given by the penalty factors PF present in the above expressions through the values of the leading parameter C_{Ref} and the exponent m in equation (4).

In addition, the plot in *Figure 8* allows evaluation in a quantitative way the gain or loss in the optimum saturation temperature at condenser inlet that can be expected by choosing different alternative refrigerants, in relation to the design constraints considered in this case. It is to be noted that the external heat transfer resistance R_{ex} plays no role in this respect.

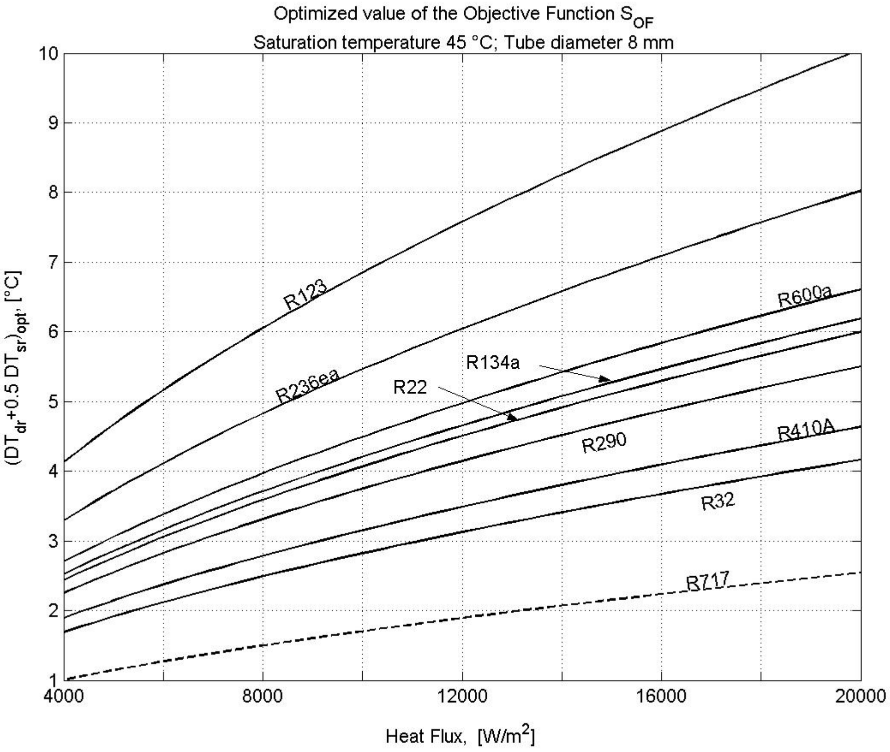

Figure 8. *Optimized value of the objective function, equation (5), for different refrigerants*
The diagram is for passive pressure loss Bf *= 2.5*

The good behaviour of the high-pressure halogenated refrigerants is once again evidenced, together with the superiority of ammonia as compared to all the other working fluids.

V. PASSIVE PRESSURE DROP

In the above development, only the refrigerant frictional pressure drop that takes place along the heat transfer portions of the circuit is considered in the computation of the refrigerant saturation temperature drop. This can be dubbed the active pressure drop, inasmuch as it is the necessary penalty paid for achieving the required value of the heat transfer coefficient.

The total refrigerant frictional pressure drop includes also a passive additional contribution, which takes place in circuit components such as return bends and header connections extraneous to active heat transfer. This term cannot usually be neglected, while it plays a role in the mean effective temperature difference between the condensing refrigerant and the pipe inner wall.

Under the same driving temperature difference DT_{dr}, the effect of the passive pressure drop in the condenser circuits is to increase the saturation temperature drop DT_{sr} (as only calculated from the active pressure drop term, as in the penalty factor PF) proportionally to the ratio of the total actual frictional pressure drop to the active pressure drop. It follows then that the optimization of the condenser in this case can be dealt with in the same manner as done above, simply by referring to an effective value of the refrigerant penalty factor increased in the same proportion as the frictional pressure drop.

This is obtained by introducing a bend factor Bf larger than 1 in all the expressions developed in the previous paragraph, as a multiplier to the leading coefficient of the penalty factor C_{Ref}. By using the established procedure to calculate the frictional pressure drop of pipe fittings in terms of equivalent length, the bend factor Bf can be estimated as the ratio of the total equivalent length of a circuit, to its actual heat transfer-active length.

The effects of the passive pressure drop on the optimized values of the relevant design parameters can now be easily established by considering the correlations developed in the previous paragraph. The differences among thermal performances of different refrigerants are enhanced by the passive pressure drop, as shown by expression (10), while the design criterion given by equation (9) remains unaffected and of general validity.

VI. CONCLUSIONS

The parameter Penalty Factor for in-tube condensation of refrigerants has been defined; why it can be regarded as an evaluation criterion of the potential thermal performance of a working fluid in tube-side condensation have been discussed. When compared at equal heat transfer coefficient, the penalty factor establishes a ranking among the different working fluids in this respect: the high-pressure refrigerants come out as the best performers, and ammonia far outweighs all the possible alternatives. Further, the penalty factor has been used to optimize the circuitry of in-tube condensers.

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