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An experimental investigation of heat transfer and friction losses of interrupted and wavy fins for fin-and-tube heat exchangers

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Abstract

The present paper discusses the results of an extensive investigation about the performance of various fin configurations, carried out in the Luve Contardo experimental facilities and aimed to enhance the heat transfer capabilities of air-cooled condensers and liquid coolers. Test results here discussed are relative to 15 coil prototypes, having the same tube and fin geometry (25×21.65 mm staggered 5/8" tube banks, 2 mm fin spacing) but different fin surface geometry, from flat to wavy to louvered to "winglet". Different rates of heat transfer and pressure loss enhancement were obtained, also depending on the quality of the pressing process. General approaches to evaluate the "goodness" of one fin design with respect to another one provided questionable results: pressure loss influence on the air flow cannot be properly evaluated unless the actual fan head curve and the coil dimensions (front area and rows number) are stipulated. The performance of air-cooled condensers was therefore predicted and compared, for various fin design and for coil arrangements of practical interest. The type of fin adopted strongly influences the heat exchanger performance and louvered fins generally provide the best results. © 2001 Elsevier Science Ltd and IIR. All rights reserved.

Keywords: Heat transfer; Air; Finned tube; Fin; Geometry; Test rig

Etude expérimentale sur le transfert de chaleur et les chutes de pression par frottement pour échangeurs de chaleur à tubes à ailettes discontinues ou ondulées

Résumé

Cette communication présente les résultats obtenus dans une étude examinant la performance de diverses configurations d'ailettes. Cette étude a été effectuée dans les laboratoires de Luve Contardo ; l'objectif était d'améliorer les caractéristiques de transfert de chaleur des condenseurs refroidis par air et les refroidisseurs de fluides. Les résultats présentés ici concernent 15 prototypes de batteries avec la même géométrie de tubes et d'ailettes (25 × 21,65 mm en quinconce avec des tubes de 5/8 de pouce et des ailettes espacées de 2 mm), mais avec différentes géométries d'ailette (plate, ondulée, à per-

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siennes ou à petites ailettes). On a obtenu divers taux de transfert de chaleur et de chutes de pression accrues, selon l'efficacité du processus de fabrication. Les approches adoptées afin d'évaluer les avantages respectifs des conceptions des ailettes ont donné des résultats peu concluants : on ne peut pas évaluer l'influence des chutes de pression sur l'écoulement d'air sans connaître les caractéristiques de la courbe du ventilateur ainsi que celles de la batterie (surface de la partie frontale et nombre de rangées). On a donc prévu et calculé la performance des condenseurs refroidis à l'air, pour différentes conceptions d'ailette et dispositions de batterie. Le type d'ailette influence fortement la performance de l'échangeur de chaleur. En général, les ailettes à persiennes ont donné les meilleurs résultats.

Mots clés : Transfert de chaleur ; Air ; Tube ailetté ; Ailette ; Géométrie banc d'essai

Nomenclature

A_f	air side front area (m^2)
A_i	internal (tube side) surface (m^2)
f	friction factor
GV	volume goodness factor [Eq. (1)] (W m^{-3} K^{-1})
j	Colburn factor $= Nu\ Pr^{-1/3}\ Re^{-1}$
LMTD	logarithmic mean temperature difference (K)
Nu	Nusselt number
p	pressure (Pa)
PP	ideal pumping power [Eq. (2)] (W m^{-3})
Pr	Prandtl number
Re	Reynolds number
T	temperature (K)
U_i	global heat transfer coefficient, referred to the internal surface (W m^{-2} K^{-1})
v	air velocity at the front of the heat exchanger (m/s)
V	volume of the heat exchanger finned pack (m^3)
Δp	loss of static pressure (Pa)
ΔT_1	temperature difference at coil inlet, i.e. condensing temperature−air inlet temperature

1. Introduction

The utilization of enhanced fin design is a very effective strategy to improve the performance of fin-and-tube heat exchangers, widely used in the refrigeration and air-conditioning industry. The amelioration of the fin-side heat transfer coefficient was addressed by many authors (see for instance [1–6]) with respect to the plate fin [7,8]. Several different fin designs can be proposed: from corrugated fins (in particular, the "wavy" fin is very popular among manufacturers) to interrupted (slits, louvers, etc.). A research and development activity was carried out by Luve Contardo (a major European manufacturer of air condensers, evaporators and liquid coolers), devoted to the study of enhanced fin behavior. Particular attention was paid to louvered fins used in industrial production since the early 1990s. This paper reports the most significant results obtained during this multi-year experimental activity for fins dedicated to condensers and dry-coolers (i.e. with tight fin pitch — about 2 mm).

From the application point of view, enhanced fins are particularly attractive, since better performance can be obtained without affecting the production cost (apart from a limited increase in the pressing tool investment cost). However, better heat transfer performance is obtained at the expense of unavoidably higher friction losses, as widely recognized by the above quoted authors. For a given heat exchanger geometry and fan head curve, elevated pressure losses decrease the air velocity and reduce the available temperature difference. The best compromise between heat transfer and pressure loss shall be found: due to the multitude of possible fin design and to the necessity of resorting to experimental (rather than numerical) investigation, the process is not straightforward. The present paper contributes to this discussion by considering the actual performance of forced-draft condensers, alternatively using different types of enhanced fins, rather than relying on general methods not fully considering the implications of the fin behavior under real operating conditions.

2. Experimental apparatus

The investigation was carried out by a wind tunnel schematically shown in Fig. 1. Air flow is driven by a variable speed fan through a square duct (500×500 mm) where the test coil is placed. The coil, fed by warm water (about 60°C), is ad-hoc manufactured to fit the air duct, with two tube rows and two parallel water feeds, to obtain a one-pass two-rows unmixed cross-flow unit.

Air temperatures are measured before and after the coil by retrieving the air from 20+20 points across the channel. Air flow is measured by means of calibrated nozzles. Water flow and temperatures are measured by a magnetic flowmeter and calibrated thermocouples. The thermal power exchanged is therefore compared on both air and water sides: the difference is typically lower than 1% (otherwise test is repeated). A fully-automated

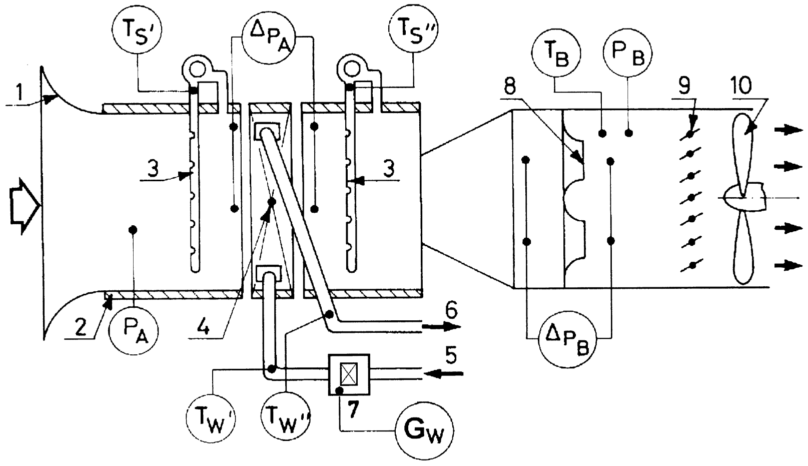

Fig. 1. Schematics of the wind tunnel. 1: inlet duct; 2: insulated wall; 3: air intakes for temperature measurement (20 points); 4: test coil; 5,6: warm water inlet/outlet; 7: water flowmeter; 8; air flow nozzles; 9: movable louvers; 10: fan.

Fig. 1. Schéma du tunnel d'essai. 1 : entrée d'air ; 2 : paroi isolée ; 3 : prises d'air utilisées pour les mesures de température (20 emplacements) ; 4 : batterie d'essai ; 5, 6 : entrée/sortie d'eau chaude ; 7 : débitmètre utilisé pour l'eau; 8 : tuyères ; 9 : persiennes ajustables ; 10 : ventilateur.

acquisition system reports the status of the thermal balance. Based on corrected LMTD, according to the peculiar flow arrangement, and on the water side heat transfer coefficient (evaluated by means of the Gnielinski correlation [9]), a fin-side heat transfer coefficient, referred to the tube internal surface, is found at various air velocities. The coil pressure loss is measured by means of $8+8$ static pressure taps, located around the tunnel wall at the coil entrance and outlet, distributed in the air duct. Pressure loss measurement is carried out in a second test session under cold conditions (zero water flow), to avoid effects related to the air density variation.

Particular attention must be paid to the prototype realization. As discussed later, the accuracy of fin pressing and mechanical expansion is very relevant to the heat transfer capabilities. All tested prototypes should be prepared by the same pressing tool, only substituting the sections devoted to the realization of the turbolators (waves, louvers, etc.). Since the pressing tools providing the highest mechanical precision are the same used for mass industrial production, the realization of prototypes requires a strict quality control as far as the collar formation and the louver cuts are concerned.

The heat transfer coefficient provided by the test sessions includes the effects of: (i) air-to-fin convective heat transfer, (ii) fin efficiency, (iii) tube-to-fin contact resistance, (iv) copper tube conductive resistance. This is useful for accurate heat exchanger design and rating calculation under standard conditions, but, to extend the results to general applications and to present them in non-dimensional form, it is necessary to separate the various effects. The air-to-fin heat transfer coefficient is evaluated by the following procedure: (i) the tube conductive resistance is calculated by very simple formulation, (ii) the tube-to-fin contact resistance is neglected: this is reasonable provided that the expansion is very accurate, (iii) the fin efficiency is theoretically evaluated by proper formulations [10] for continuous fin.[1] The details of the procedure are to be considered proprietary information. However, the test results here reported refer to the same fin-and-tube geometry and the above mentioned procedure does not affect the relative merits of the various fin configurations.

The air-to-fin heat transfer coefficient can be expressed in non-dimensional form $j=j\ (Re)$ where $j=Nu{\cdot}Pr^{-1/3}{\cdot}Re^{-1}$ and the Reynolds number Re is based on the hydraulic diameter of the elemental cell around a tube and between two fins and on the mean velocity along this flowpath. This approach provided sufficiently good results when used to extend the test results to different geometry (i.e. fin spacing, fin thickness, tube diameter, tube spacing) and/or operating conditions (i.e. air temperature or pressure, bringing about some variations in the thermodynamic properties), by repeating the same procedure in the reverse way.

The friction factor f is evaluated from the measured static pressure drop according to the same definitions of the Reynolds number and hydraulic diameter and using the row spacing as the characteristic length of the flowpath.

[1] The actual fin efficiency of interrupted fins cannot be easily estimated. Therefore, differences between the actual and the estimated fin efficiency will be attributed to the air-to-fin heat transfer coefficient.

3. Types of plate fin

The results reported here are all relative to a commonly used staggered geometry, having a 25 mm tube spacing, a 21.65 mm row spacing, with 9.52 mm tube diameter. The fin spacing is 2 mm, the fin thickness is 0.11 mm. Little variations due to slightly different fin spacing (±5%) and fin thickness (±10%) are not supposed to influence the non-dimensional terms j and f.

Table 1
List of fin configurations here addressed

Tableau 1
Liste des configurations d'ailettes

Name	Type	H[a] (mm)
P	Plain	n.a.
N	Wavy	0.9
C1	Corrugated	0.8
C2	Corrugated	0.8
L1A	Louvered	0.54
L1B	Louvered	0.54
L2	Louvered	0.60
L3A	Louvered	0.75
L3B	Louvered	0.75
L4	Louvered	0.90
X1	Extended louver	1.00
X2	Extended louver	0.75
X3	Extended louver	0.65
W1	Winglet	1.6
W2	Winglet + louver	1.6 + 0.7

[a] H, height of louvers and corrugations.

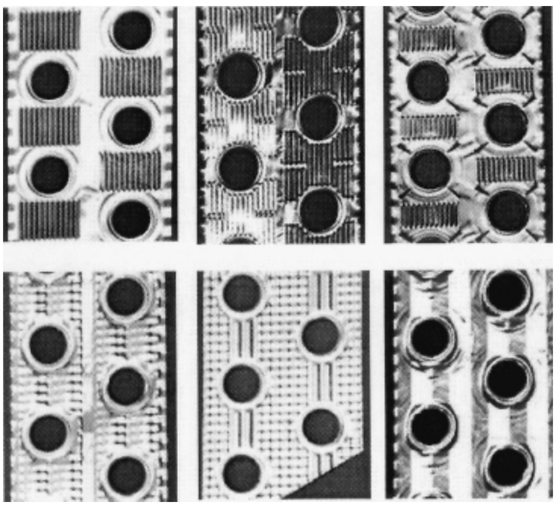

Fig. 2. Fin configurations. Louvered fins L, X and W in the upper part, corrugated fins C1, C2 and N in the lower part (see Table 1).

Fig. 2. Configurations des ailettes. Ailettes à persiennes L, X et W dans la partie supérieure, ailettes ondulées C1, C2 et N dans la partie inférieure (voir le Tableau 1).

The tested fin configurations are listed in Table 1 and shown by the pictures of Fig. 2. With more detail, fins N and L are currently used for industrial production. Fins C1 and C2 were considered as substitution of the traditional wavy fin N [1,2]. Fins X and W may represent an alternative to the louvered L configuration [3,4]: X includes a larger extension of the louvered surface, and W makes use of "winglet" type turbolators [11]. The prototype W1 differs from W2 (shown in Fig. 2) because it does not include a louvered section. For the L configuration, the louver height was varied from 0.54 to 0.9 mm, due to its relevant influence on the fin performance.

In addition, several coils were tested during previous years with 0.54 (L1) and 0.75 (L3) height: even if apparently similar, the tested performance showed significant differences, well beyond the experimental uncertainties. Therefore, L1A and L1B are, respectively, the worst and the better coil among the several tested with 0.54 louver height (the same for L3A and L3B at 0.75 height). After a detailed investigation, it was found that differences may arise from the mechanical precision and the maintenance status of the pressing tools and in particular from:

- Sharpness of the louver cutting section: if not perfect, the cross-section may result partially occluded, spoiling the beneficial effects of the interruptions.
- Correct shape of the collar: the area contacting the tube may be reduced by a larger than necessary curvature radius.
- Thickness of the neck-band collar and of the fin portion around the tube: if thinner, the fin efficiency decreases. In the worst cases, some collars may break during the tube expansion, impairing the contact between tube and fins: this occurrence cannot be accepted and the pressing machine must be revised.

4. Results of the experimental activity

The results of the lab tests are shown in non-dimensional form in Fig. 3, for heat transfer performance, and in Fig. 4, for pressure losses. The Reynolds number variation corresponds to air velocities at the front area from 1 to 3 m/s, for this particular geometry and fin pitch. The results are also shown in an easier way in Fig. 5, limited at $Re = 600$ and showing the improvement of both heat transfer and pressure loss with respect to the plain fin. We can comment that:

- Corrugated fins N, C1 and C2 provide a limited improvement of both f and j; the traditional wavy fin N remains the better solution among them; the poor results of C2 may be partially attributed to the poor quality of the prototype.

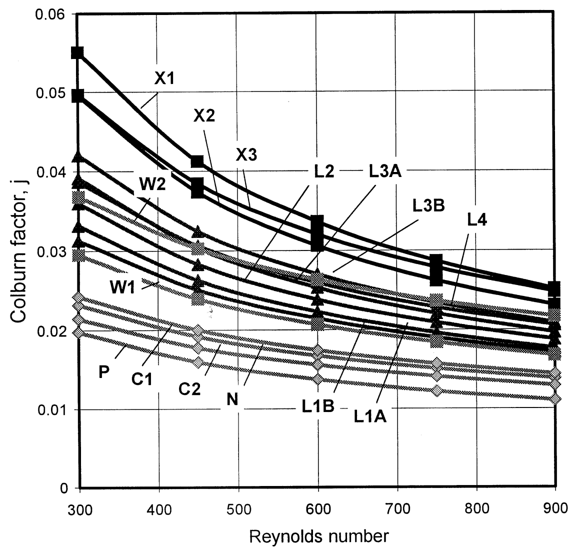

Fig. 3. Colburn factor j vs. Re for the tested fins.

Fig. 3. Facteur de Colborn j *comparé à* Re *pour les ailettes testées.*

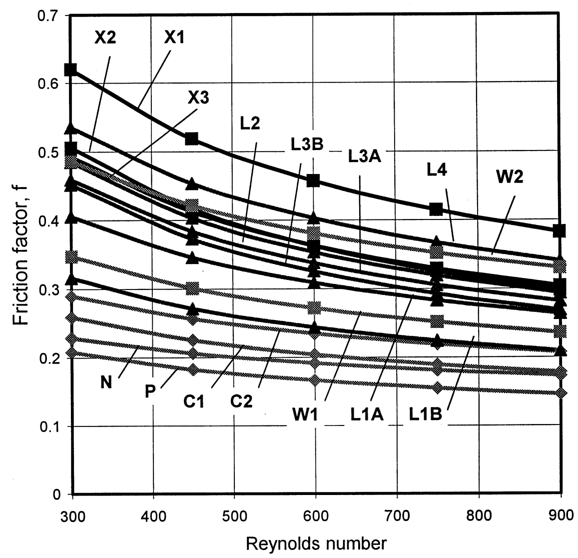

Fig. 4. Friction factor f vs. Re for the tested fins.

Fig. 4. Facteur de frottement f *comparé à* Re *pour les ailettes testées.*

- Louvered fins L provide an important improvement of the heat transfer coefficient but much higher pressure losses: the amount depends on the louver height. In particular, a louver height of 0.9 (L4) rather than 0.75 (L3) does not improve heat transfer but only pressure losses. Significant differences of both j and f were found when the louver height was varied from 0.75 (L3) to 0.54 mm (L1): louvered fins definitely require a strict dimensional control. In addition, the difference between the A (worst) and B (better) prototypes of L1 and L3 must be emphasized, especially as far as pressure losses are concerned: probably, the occlusion of the louvers is the most important effect among the ones depicted above. The heat transfer enhancement of louvered fins with respect to wavy fins is larger at low Reynolds number.
- The extended louver X prototypes provide an exceptionally high heat transfer coefficient, doubling the one of wavy fins. Except for X1 (having a higher louver with some irregularities and possible louver occlusions), pressure loss is only slightly higher than for L design with the same louver height.
- The "winglet" design W1 unexpectedly provides relatively low improvements of f and j in spite of its elevated height (1.6 mm vs. 2 mm fin pitch); when adding a louvered section (W2) f and j increase, but pressure loss becomes higher than for an L fin of comparable j. However, it must be stressed that the winglet design requires a very careful optimization (i.e. height, angle, length) and therefore the present results are not sufficient for a conclusion.

The comparison of Fig. 5 is rather useful: a fin design providing a lower f at the same j (or a higher j at the same f) is undoubtedly superior. Therefore, fins represented by a dot staying on the line at equal j/f or below that line are good fins (for instance, N, L1B, L3B, X2 and X3). Anyway, Fig. 5 does not provide any means to assess the superiority of a fin having low or high values of both f and j. This will be discussed in the next section.

5. Selection of the best fin design

5.1. General criteria

Let us select from Fig. 5 the four fin configurations N, L1B, L3B and X3, having comparable "*goodness factor*", defined as j/f (respective values at $Re = 600$ are: 0.0873, 0.0880, 0.0829, 0.0882), but very different values of j and f. One possible criterion to compare their performance for actual heat exchanger design is the evaluation of the "*volume goodness factor*" (GV) as a function of the "*ideal pumping power*" (PP) at different air front velocities (v). GV and PP are here defined as:

$$GV = U_i \cdot A_i / V \tag{1}$$

$$PP = v \cdot A_f \cdot \Delta p / V \tag{2}$$

GV represents the thermal power exchanged per unit of heat exchanger volume and per unit of ΔT, while PP represents the ideal power of a fan driving the air through the heat exchanger per unit of its volume

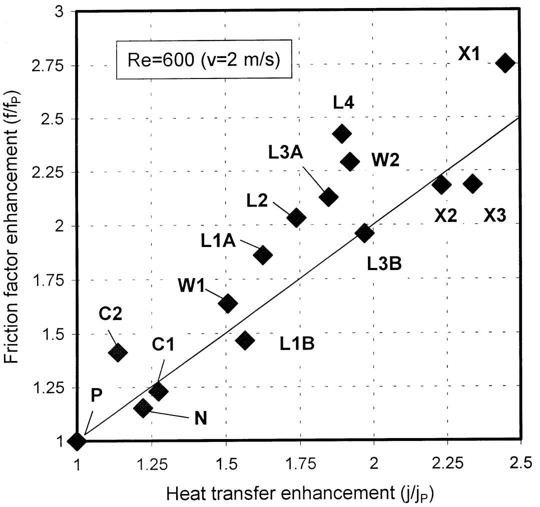

Fig. 5. Heat transfer and friction factor enhancement of the tested fins with respect to the plain fin, at $Re = 600$, corresponding to a front area velocity of 2 m/s. Fin configurations are described in Table 1 and Fig. 2.

Fig. 5. Transfert de chaleur et amélioration du facteur de frottement des ailettes testées en comparaison avec une ailette simple, pour Re = *600, ce qui correspond à une vitesse de 2 m/s sur la superficie avant. Les configurations des ailettes sont décrites dans le Tableau 1 et la Figure 2.*

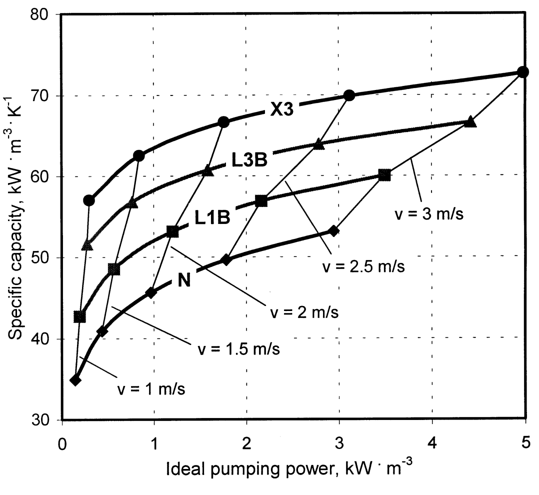

Fig. 6. Volume goodness factor criterion applied to fin configurations having very different j and f.

Fig. 6. Critère d'efficacité appliqué à la configuration des ailettes possédant des facteurs j *et* f *très différents.*

(reproducing with some modification the approach proposed by Shah [12]). To provide a more realistic situation, a tube-side heat transfer coefficient of 5000 W m^{-2} K^{-1} was assumed in the evaluation of U_i, as often found for R22 condensers with micro-finned tubes. For this comparison, the air-side heat transfer coefficient keeps into account the fin efficiency (whose variations among the various fins were not considered when addressing j in Figs. 3 and 5).

Fig. 6 shows the GV–PP relation for the four selected fins, at various air velocities. At the same pumping power, GV clearly increases for enhanced fins, even if a lower air velocity is required to compensate the higher friction factor. This behavior can be easily explained if one thinks that PP depends on the velocity with a 2.5–2.75 power, vs. a 0.3–0.45 power for the heat transfer coefficient: therefore, a slight air velocity reduction overrides large f differences with very limited effects on the heat transfer capabilities. Even a fin design having a poor j/f but an elevated j (for instance, L4) would perform better than a wavy fin if reported in the GV–PP plot. Therefore, according to the indications of Fig. 6, it is sufficient to adopt a slightly larger front area to take full advantage of enhanced fins without affecting the pumping power.

However, the results provided by the GV–PP criterion must be carefully considered. In the design practice, the adoption of fins having a better heat transfer coefficient would increase the heat exchanger capacity at the same volume and cost, but the friction factor augmentation would reduce the air flow when coupling the coil to a real (not "ideal"!) fan having a given head–flow curve. The air flow reduction has two detrimental effects: (i) the lower air velocity reduces the heat transfer coefficient, (ii) the air temperature increases while crossing a condenser unit, thus reducing the LMTD at the same initial temperature difference. The second effect is not taken into account by the GV–PP analysis, which makes reference to the capacity exchanged per unit of ΔT between the fluids (i.e. per unit of LMTD). Therefore, a loss of air flow does not affect the indications of the GV–PP criterion, but represents a very large detrimental effect in the applications.[2] Thus, the relative merits of various fins cannot be estimated soundly by a general criterion, as we will stress in the next paragraph.

5.2. Application to condensers

The use of different fin designs certainly affects the performance of air-cooled condensers: for industrial applications, the best fin is the one allowing for the lower possible cost of the condenser per unit of thermal power exchanged under a selected ΔT_1.[2] However, its selection is not straightforward, since manufacturers propose a whole range of products, modularly assembled. Each module is characterized by (i) various fan rotational velocity and/or diameter, determining the

[2] For air-cooled forced-draft condensers (but the same holds for evaporators), the relevant matter is the thermal power exchanged per unit of ΔT_1 (i.e. condensing temperature–ambient temperature) and not per unit of LMTD. The larger the air flow, the lower the air outlet temperature, the higher the capacity under the same operating conditions and $U \cdot A$.

noise level and the air flow, (ii) various heat exchanger sizes: front area and number of rows. Therefore, the designer should consider all the different situations encountered in the practice to perform a comprehensive comparison of the fin performance.

The result of such an investigation is reported in Fig. 7. It considers several combinations of coil front area and rows number, when coupled to a 800 mm fan driven by six or 12 poles motors. We can comment that:

- For two rows, fins providing the higher heat transfer obtain better results, especially with the six poles motor, at any front area. X3 is definitely the best fin, as suggested in Fig. 6.
- For three rows, the same holds for six poles (but the percentage performance gain from N to X3 is reduced), while for 12 poles all fin designs provide approximately the same capacity. The higher pressure loss of louvered fins yields to reduced air flows (much more with three rather than two rows) and LMTD, counterbalancing the effect of higher heat transfer coefficient.
- For four rows, all louvered fins have the same performance at six poles, while wavy fins performs

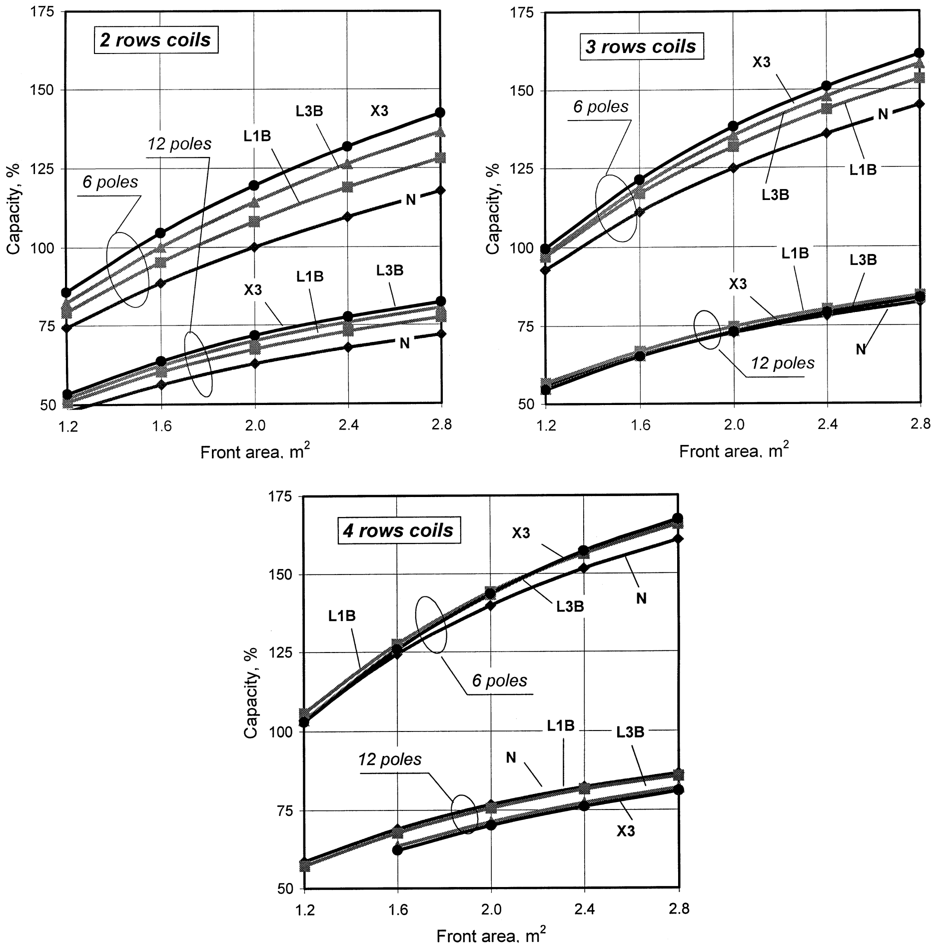

Fig. 7. Thermal capacity of air-cooled condensers driven by one 800 mm fan with six or 12 poles motor, as a function of the coil area. Two, three of four rows of tubes and four different fin designs (Table 1) are considered. For all cases, thermal capacity is referred to the one of two rows, 2 m^2 front area, wavy fin N, six poles motor. Capacities were obtained for R22 condensation at 40°C, 25°C superheating, no subcooling, inlet air at 25°C, by using micro-finned tubes with an optimized number of parallel feedings for each case.

Fig. 7. Puissance thermique des condenseurs refroidis par air entraînés par un ventilateur de 80 mm avec moteurs à 6 ou 12 pôles, selon la superficie des batteries. Deux, trois ou quatre rangées de tubes et quatre conceptions d'ailette différentes (Tableau 1) sont examinées. Dans tous les cas, la capacité thermique est exprimée pour l'une de deux rangées, surface frontale de 2 m^2, ailette ondulée N, moteur à 6 pôles. Les puissances thermiques sont obtenues pour la condensation de R22 à 40°C, 25°C, avec surchauffe et sans sous-refroidissement, pour une température d'air à l'entrée de 25°C, utilisant des tubes à microailettes avec un nombre optimal de points d'alimentation en parallèle dans chaque cas.

better at 12 poles. Comparing three to four rows, one can notice that with 12 poles capacity does not increase for the N design and decreases for X3 and L3B: the adoption of four rows is definitely "bad design". Even with the six poles fan the capacity improvement with respect to three rows is rather limited for the X3 fin, but significant for the N fin (air flow remains large due to limited pressure loss of N fin).
- For all cases, the X3 fin allows for the best performance, excluding cases with no practical interest (four rows, 12 poles).

In general, the performance improvement made possible by enhanced fins does not depend (or very slightly) on the coil front area, but mostly on the rows number and air flow. It can be said that large improvement can be obtained if the base case (N fin) has a low or moderate heat exchanger effectiveness: otherwise there is little to gain with a higher U_i, especially if accompanied by an air flow reduction.

Fig. 7 does not help in selecting the most appealing solution for the condenser module:[3] a cost estimation is necessary to determine whether the increased capacity of solutions with larger front area and rows number coils may override the higher coil cost (which is only a part of the total cost for a forced draft condenser). This exercise is more an industrial rather than scientific matter and will not be treated here. However, it can be anticipated that, based on present cost structure, the three rows arrangements provide better cost/capacity than for two or four rows solutions. Comparing the results of Fig. 6 (volume goodness factor) to those of Fig. 7 — three rows, the superiority of the X3 solution is confirmed, but at a much lower rate than suggested in Fig. 6 (as a matter of fact, for 12 poles all the fin designs provide the same performance, while Fig. 6 suggested a superiority of louvered fins at any air velocity).

6. Conclusions

A large number of fin configurations were tested to assess their heat transfer and pressure loss capabilities. For louvered fins it was found that the louver height largely influences its performance and that the quality of the pressing process is essential to the achievement of the best results.

When comparing fin configurations having comparable j/f but different j and f, the situation cannot be fully depicted by general criteria, not keeping into account the real interaction between the heat exchanger and the fan driving the air flow. By considering real design cases, it was found that enhanced fins are particularly useful for large air flows (high speed fan) and/or reduced coil depth; with an elevated row number, their superiority vs. traditional fins is questionable.

[3] The two fan speeds here addressed are not to be compared: six poles is the best solution as far as investment costs and space requirement (footprint) are concerned, but 12 poles has very reduced power consumption and, mostly, much lower noise level. Selection is therefore dependent on the application and on customers' requirements.

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