A CORRELATION BETWEEN HEAT TRANSFER AND PRESSURE DROP IN HEAT EXCHANGERS OF DIFFERENT SHAPES

Summary:

The correlation proposed seems to be valid within an accuracy of ± 20%, and includes heat exchangers of very different forms. It also has many possibilities of application in other problems of heat transfer, like convection in furnaces, reciprocating engine cylinders, tube entrance effects, etc.

INTI, Buenos Aires 12/72
Ing. L.D.Porta
Ing. E. Beaumont Roveda

NOMENCLATURE:

a_o	[—]	Correction coefficient for exchangers with finite number of rows.
A	$[m^2]$	Heating surface in one hollow of the exchangers.
c_p	$[J.kg^{-1}.K^{-1}]$	Specific heat at constant pressure.
D	$[kg.s^{-1}]$	Mass flow rate.
f	[—]	Friction factor, defined by Eq. (1)
$G_{máx}$	$[kg.s^{-1}.m^{-2}]$	Mass velocity referred to the minimum cross section
h	$[J.m^{-2}.s^{-1}.K^{-1}]$	Heat transfer coefficient.
L	[m]	Flow lenght in one step of the exchanger.
P_r	[—]	Prandtl Number.
R_e	[—]	Reynolds Number.
S_{min}	$[m^2]$	Minimum flow cross section (Fig. 4.2).
v_m	$[m^3.kg^{-1}]$	Specific volume of the fluid.
p	$[N.m^{-2}]$	Pressure drop in one step of the exchanger
f	$[kg.m^{-1}.s^{-1}]$	Dynamic viscosity at mean film temperature.
$m = 1/v_m$	$[kg.m^{-2}]$	Fluid density.

INDEX:

1.- SUBJECT:

Heat exchangers theory is a very well known matter, specially on the so called compact heat exchangers (5.2.1).

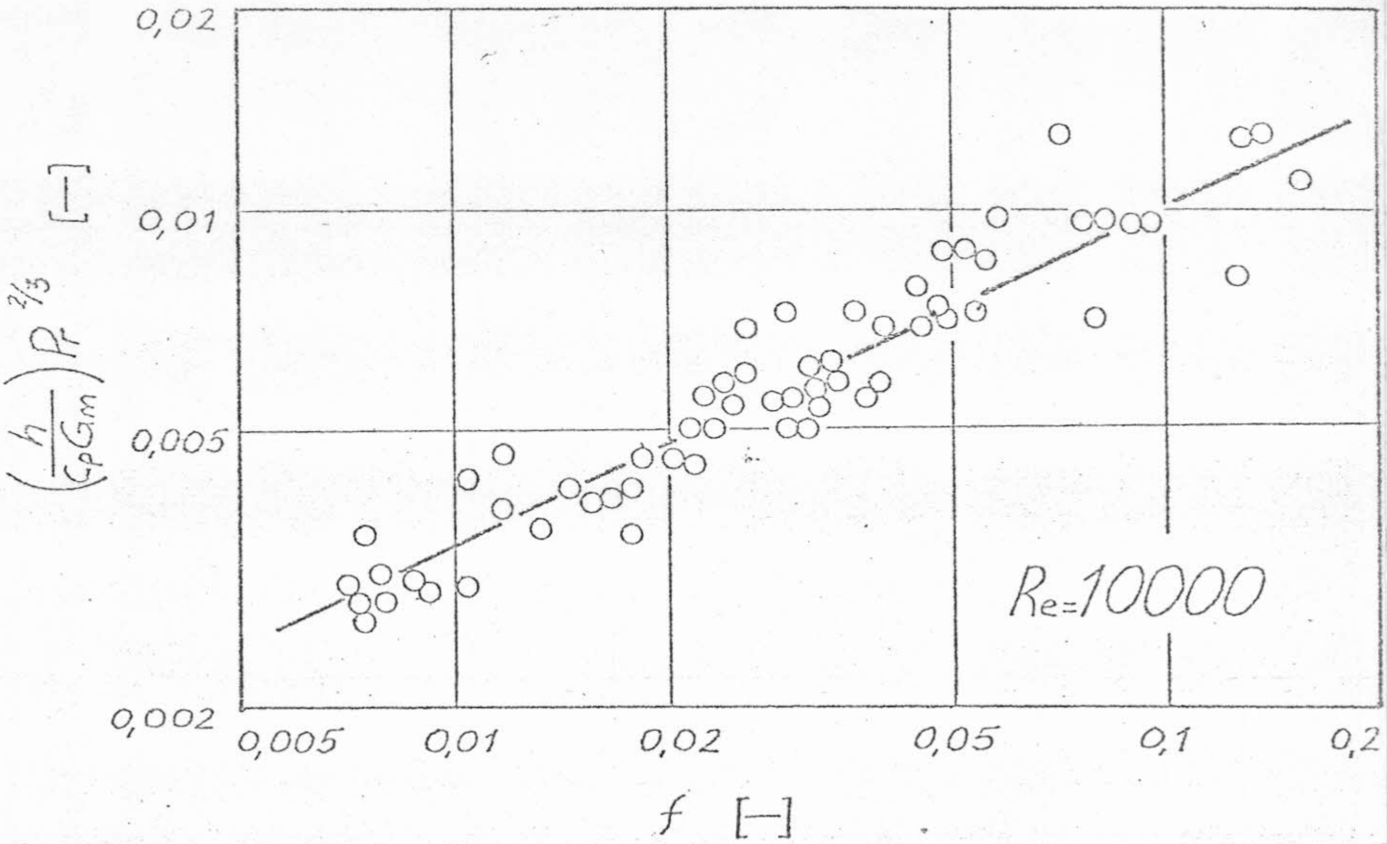
The origin of the present paper is the request of defining quantitatively heat transfer phenomena obtaining inside the cylinders of steam engines (Rankine expanders), a field of renovated interest today. Al effects attibuted to that cause have been measured up to 50% of the theoretical steam consumption.

Originally it was aimed at to apply formulae derived from those used in the I.C. engine field provided the required ammendements to count for different fluid properties. Soon it was realized the inadequacy of their semi-empirical approach, and this led to think about a rational theory integrating all types of reciprocating machines.

It is well known the existence of a relationship between friction and heat transfer, yet only for a few cases in quantitative form. The first step was, therefore, to ask if it were possible a generalized correlation valid for heat exchangers of widely varying geometries, including heavily roughened tubes. And it is shown here that such a correlation is possible to a reasonable degree of accuracy, this proposed as a subject in itself and independently of the original motivations spoken of above.

It is hoped to present, in future papers, the whole treatment leading to the answer of the problems originating the present development; and also throwing light on old and inconclusive questions pertaining not only to reciprocating steam engines but also IC's, compressors, etc.

The present work does not pretend fully to answer the matter; it is only the result of the need of disposing of a tool allowing a first quantitative assessment on different questions not answered by a lenghty and aged literature enquire.



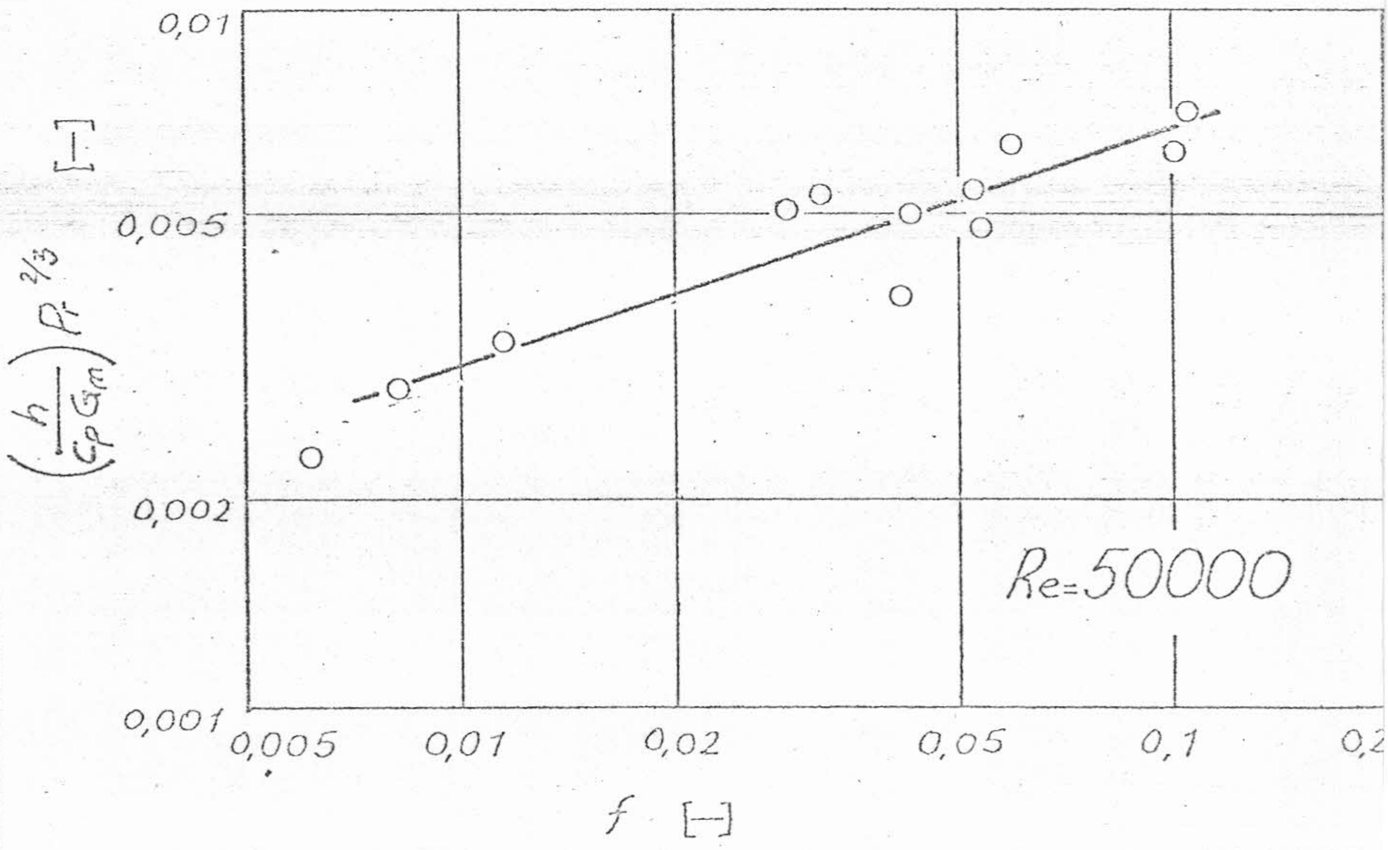


Fig. Nr 4.1. : Experimental relationship between heat transfer and pressure drop in heat exchangers.

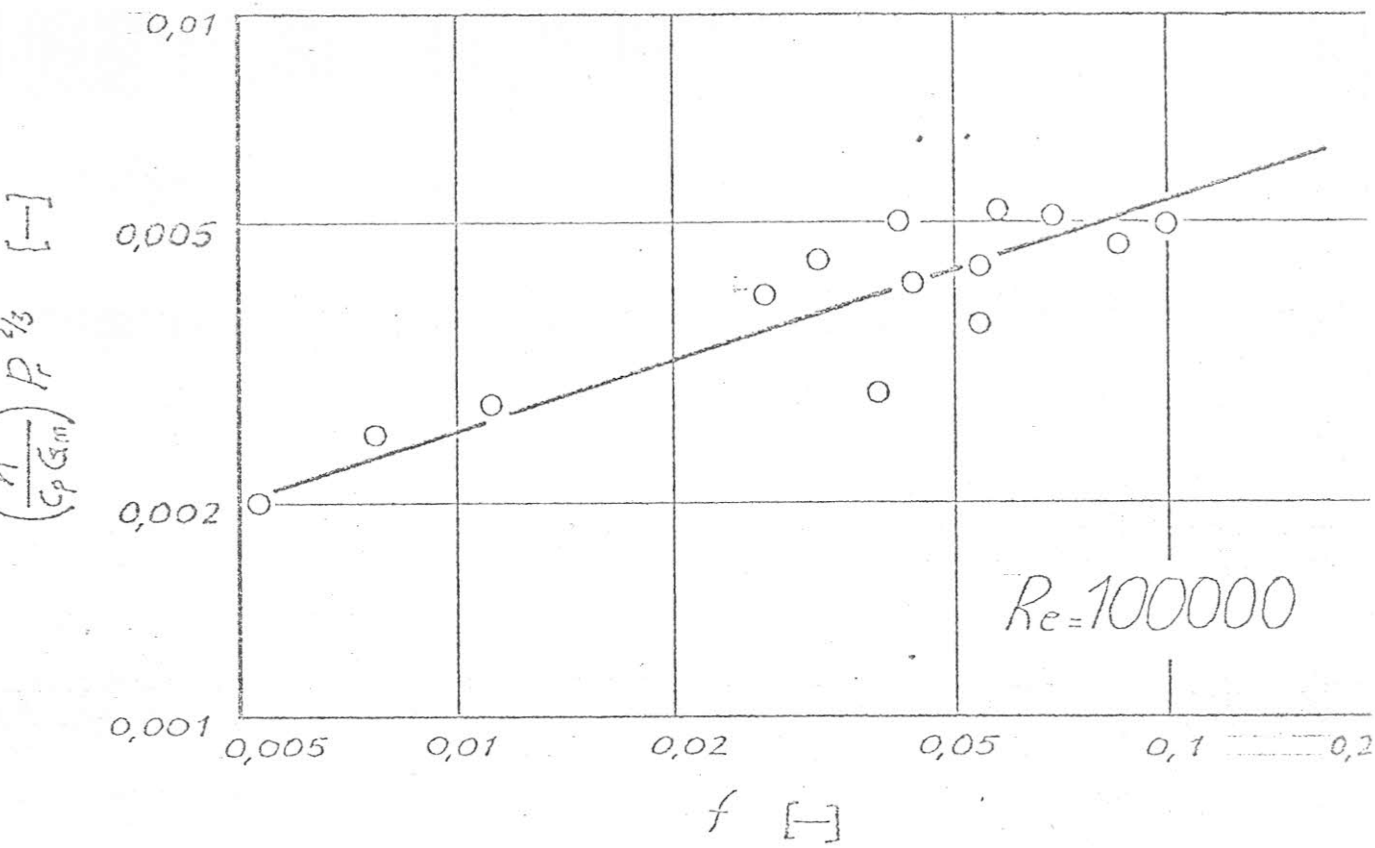


Fig. Nr. 4.1 : Experimental Relationship between heat transfer and pressure drop in heat exchangers.

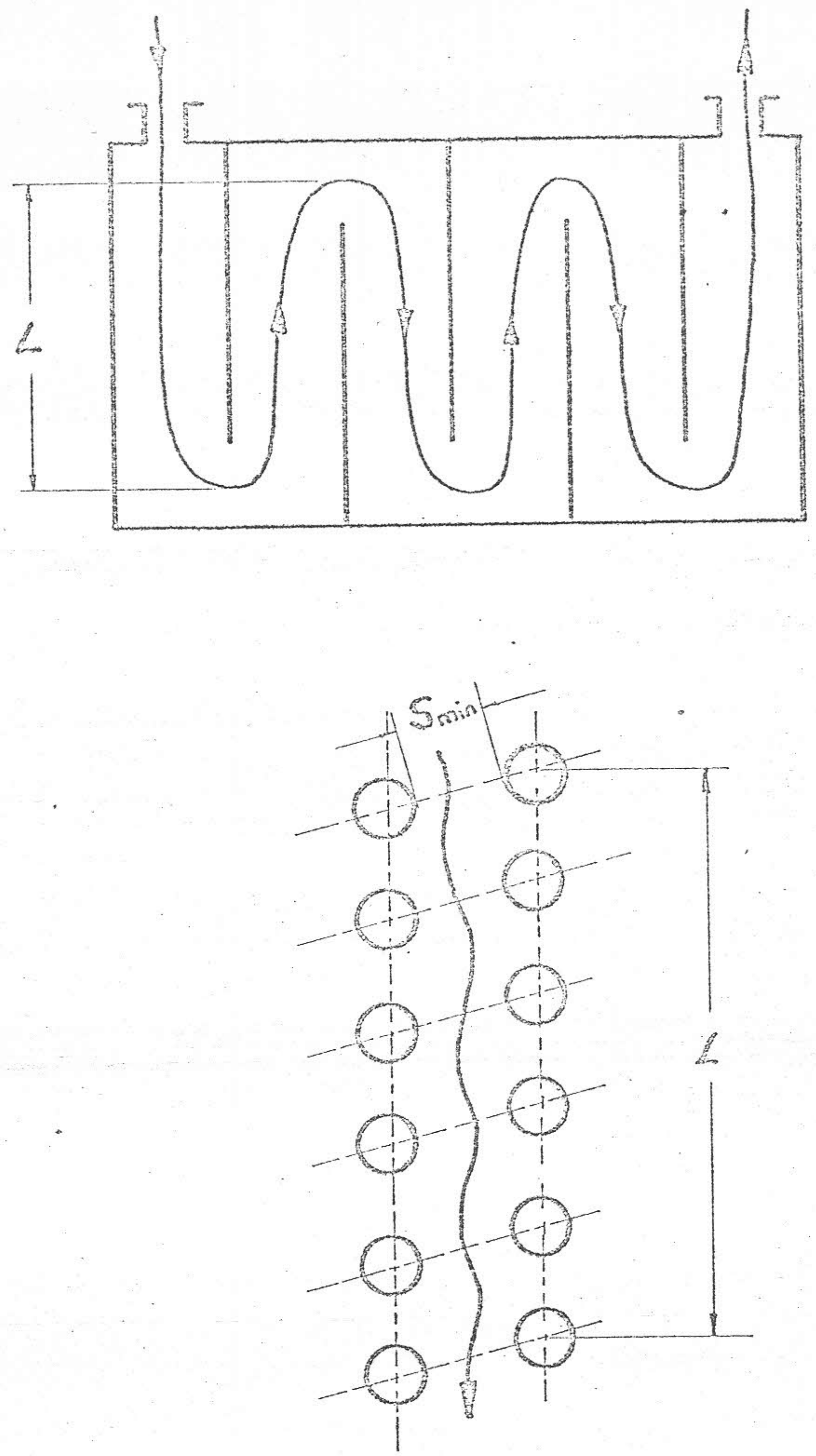
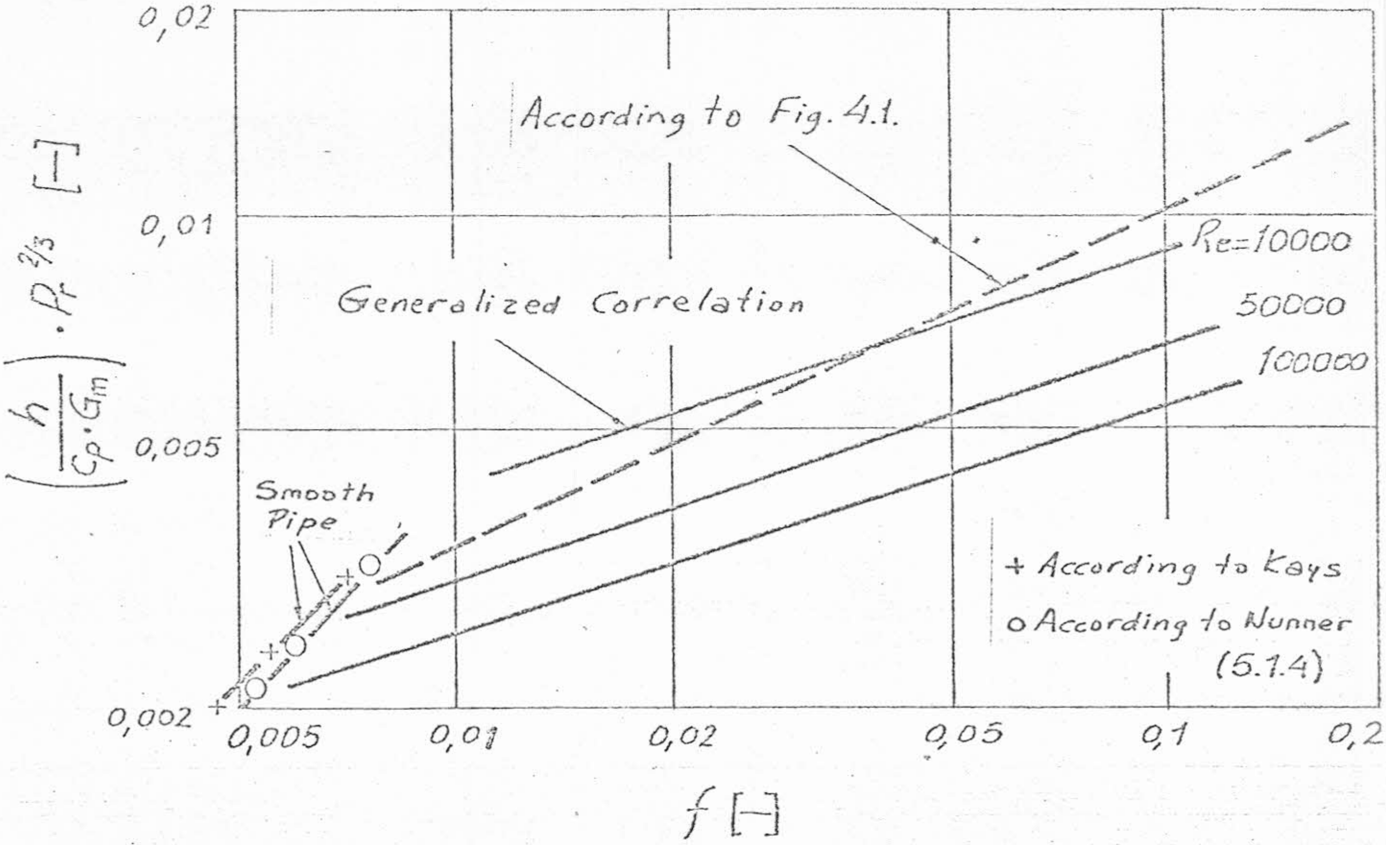


Fig. Nr 4.2. Cross Section of a heat exchanger (schematic)



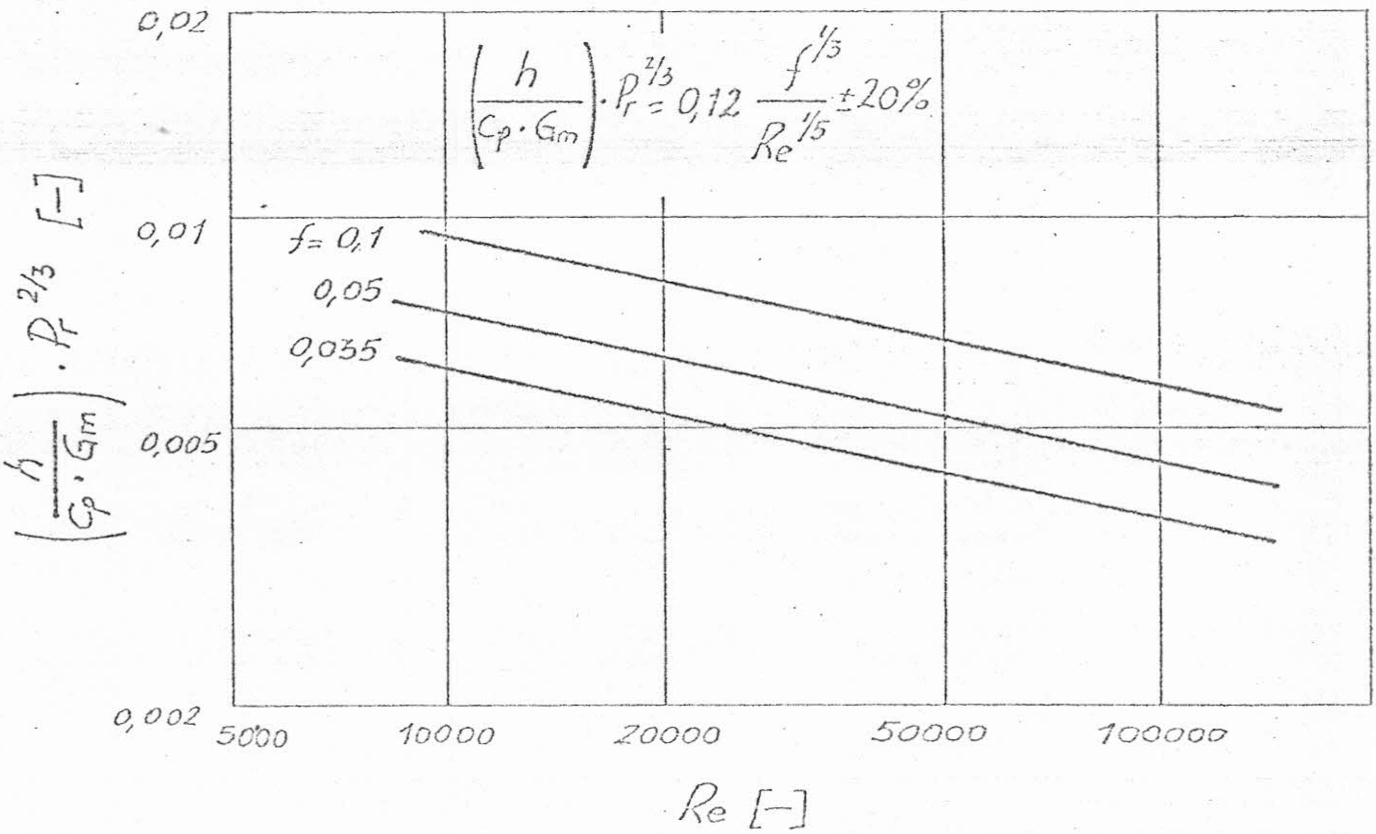


Fig. Nr 4.3.: Proposed Correlation between heat transfer and pressure drop for a heat exchanger of any shape.

2.- THEORY:

2.1.- Exchangers with infinite number of rows:

Kays (5.2.1) gives a great amount of experimental data referred to exchangers of very different shapes, which is condensed in fig. 4.1. (The values were obtained experimentally with air; for other fluids, see app. 5.1).

This data is referred to Reynolds numbers not greater than R_e = 10 000. The extension for other values of R_e, may be made using the classical Grimison experimental data.

Following Mc Adams (5.2.2), it is:

$$\Delta P = f \cdot \frac{G_{máx}^2 \cdot A}{2 \cdot S_{min} \cdot \rho_m} \quad [N \cdot m^{-2}] \qquad (1)$$

Defining Reynolds number as:

$$R_e = \frac{4 \cdot S_{min} \cdot L \cdot G_{máx}}{A \cdot \mu_f} = \frac{4 \cdot L \cdot D}{A \cdot \mu_f} \quad [-] \qquad (2)$$

and considering that

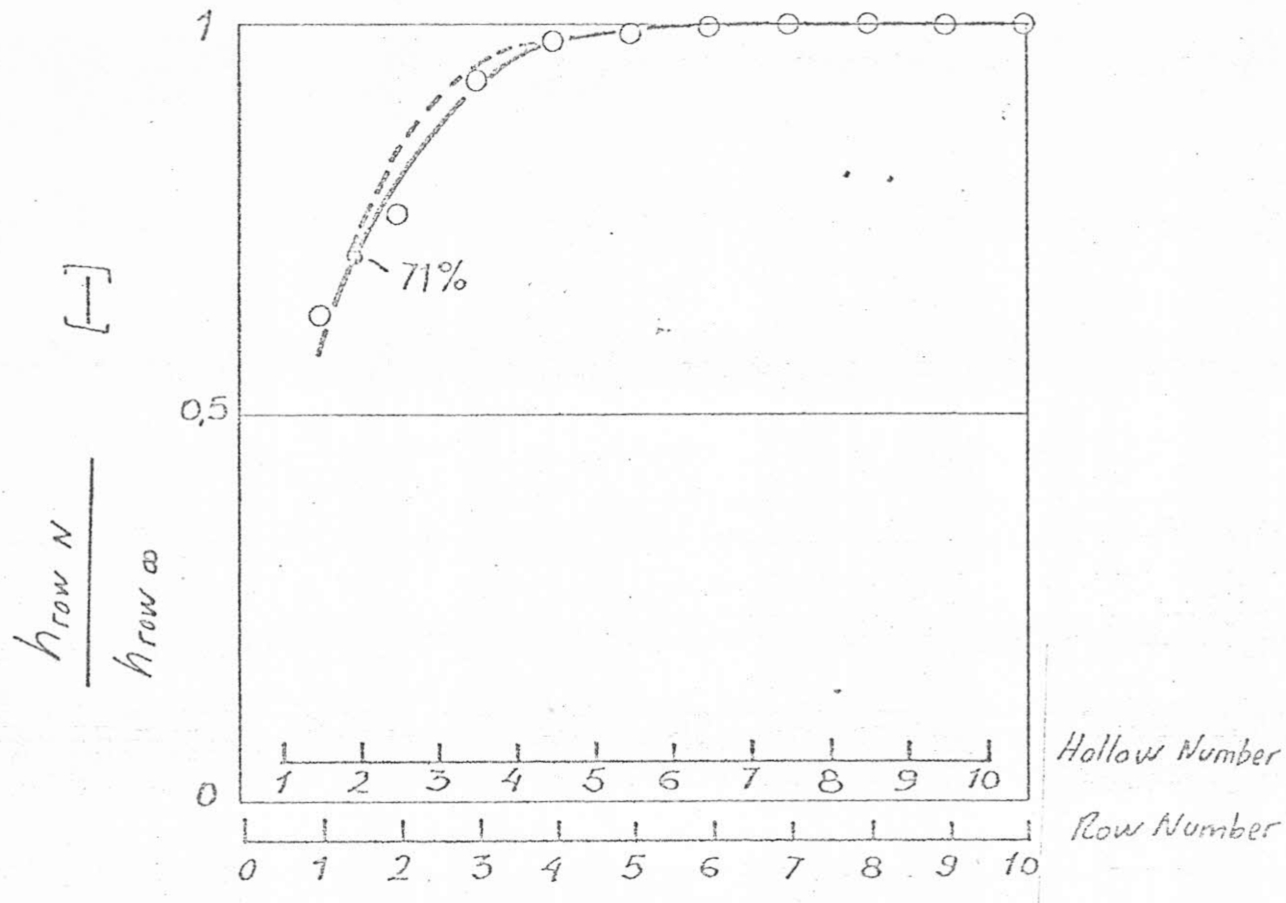
$$D = G_{máx} \cdot S_{min} \quad [kg \cdot s^{-1}] \qquad (3)$$

the obtained correlation from the represented values, not very different from that of Grimison values (5.2.3), may be expressed as:

$$\left(\frac{h}{c_p \cdot G_{máx}}\right) \cdot P_r^{2/3} = 0,12 \cdot \frac{f^{1/3}}{R_e^{1/5}} \pm 20\,\% \; [-] \qquad (4)$$

valid between R_e = 20 000 and 150 000, and f = 0,01 and 0,1.

It is necessary to note that ΔP is the pressure loss due only to friction, i.e., not taking into account stream acceleration. Also the entrance and exit effects are not included and ΔP is referred to exchangers with infinite number of rows.



—o Alternative Pipes 5.2.1. Table 10-7; 5.2.2. p. 275

--- In line pipes 5.2.1. Table 10.6; 5.2.2. p. 274

Fig. Nr. 4.4. : Heat Transfer in different rows of an exchanger.

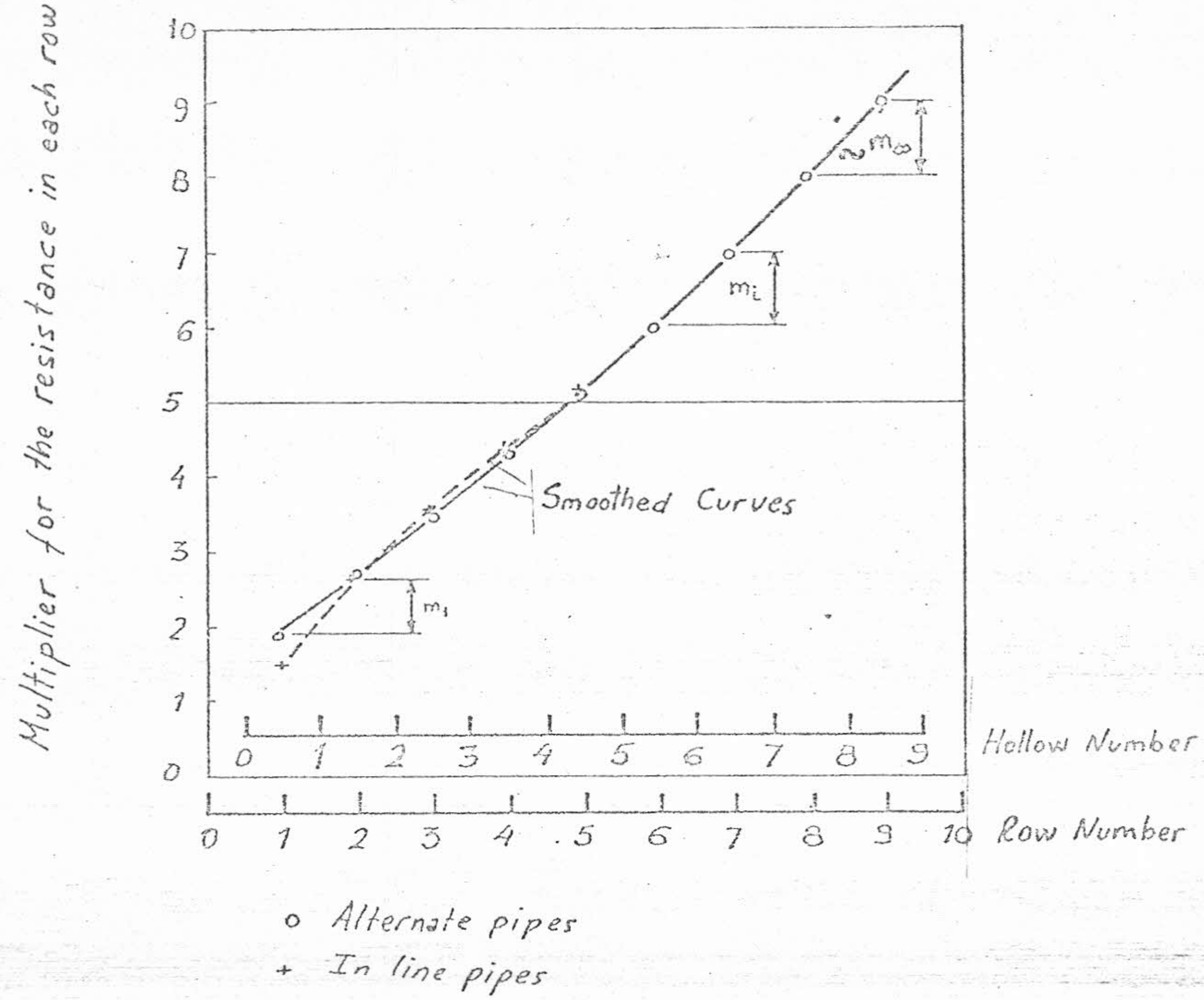


Fig. Nr. 4.5: Pressure drop variation along a bundle of pipes.

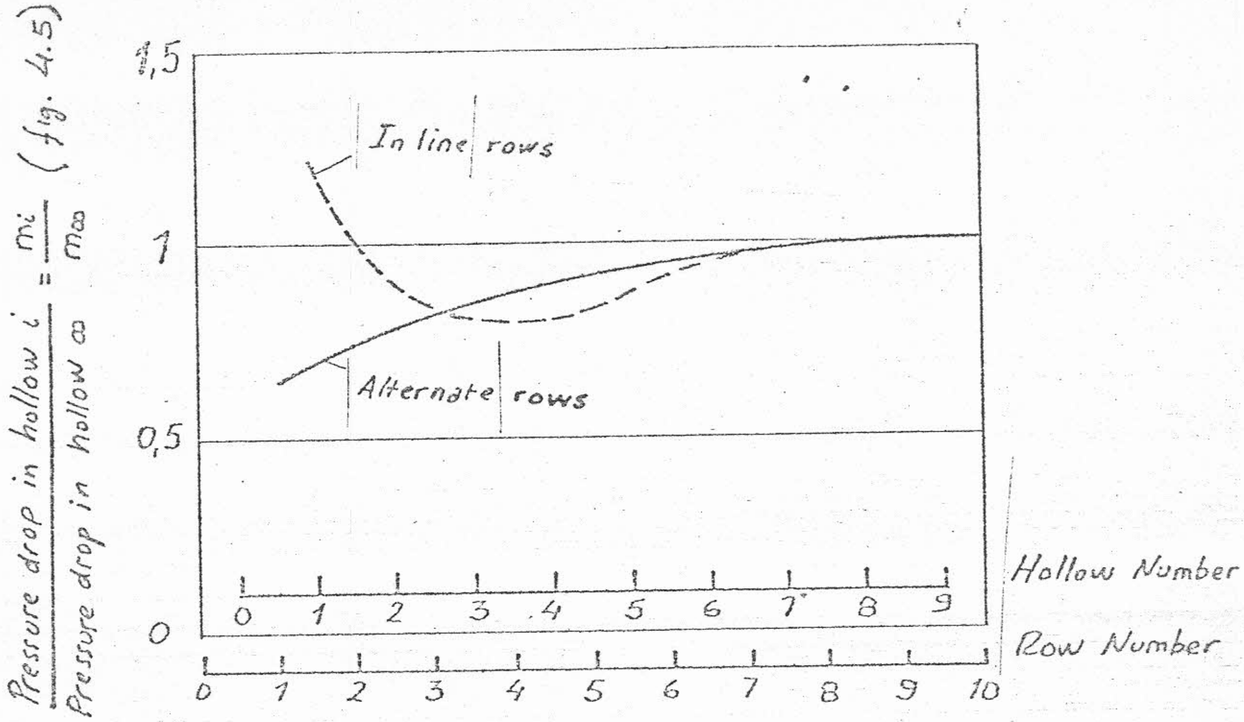


Fig. Nr. 4.6. Relationship between pressure drops in the hollows i & ∞

2.2.- Finite number of rows corrections:

It is well known that in heat exchangers heat transfer is lower in the first rows than that corresponding to rows placed far from the fluid entrance (flowing outside the tubes). This is due to the fact that the later tubes are subjected to a stream embodying the turbulence generated by the precedent ones.

Fig. 4.4 shows such variation according to data given in (5.2.2).

In that figure it has been indicated either for tubes in line or in alternating arrangement - heat transfer figures obtained along succesive "hollows" giving the latter name to the space existing between two succesive rows. It is seen that heat transfer is different in every hollow because of the turbulence of the incoming stream.

Neither the pressure drop is uniform because the greater turbulence gives a different pressure loss in each hollow.

Ledinegg (5.2.5) gives multipliers of the pressure loss in each row according to the experiment of Ter Linden (5.2.6) as reported in the following table:

TABLE 2.2.1

Multipliers for the resistance in each row (Ter Linden 5.2.6)

Rows	Pattern	
	Alternate	In line
1	1,85	1,47
2	1,34	1,33

3	1,15	1,19
4	1,08	1,09
5	1,02	1,03
6	1,00	1,00
...	...	...
10 and more	1,00	1,00

The total resistance of a tube bundle, in arbitrary units, is the product of the multipliers times the mumber of rows, as it shown in fig. 4.5.

In that figure, it have been drawn smoothed curves. Also in arbitrary units, the difference between the resistances occassioned at every row gives the pressure drop obtained in each hollow as indicated in fig. 4.6.

All the above can be considered incorporating a coefficient a_o in the formula given in section 2.1.

According to the figure 4.4., the heat transfer in the first hollow is, for alternating rows, 71% of that corresponding after the sixth one. Also, the pressure drop is only 70% of that of the same rows (Fig. 4.6).

Since a part of the difference is already taken into account because of the lesser pressure drop and for a given pressure drop in that first row of alternating tubes, the heat transfer in the first hollow can be described with a coefficient,

$$a_o = \frac{0,71}{\sqrt[3]{0,70}} = 0,80$$

of that corresponding to those rows from the sixth one.

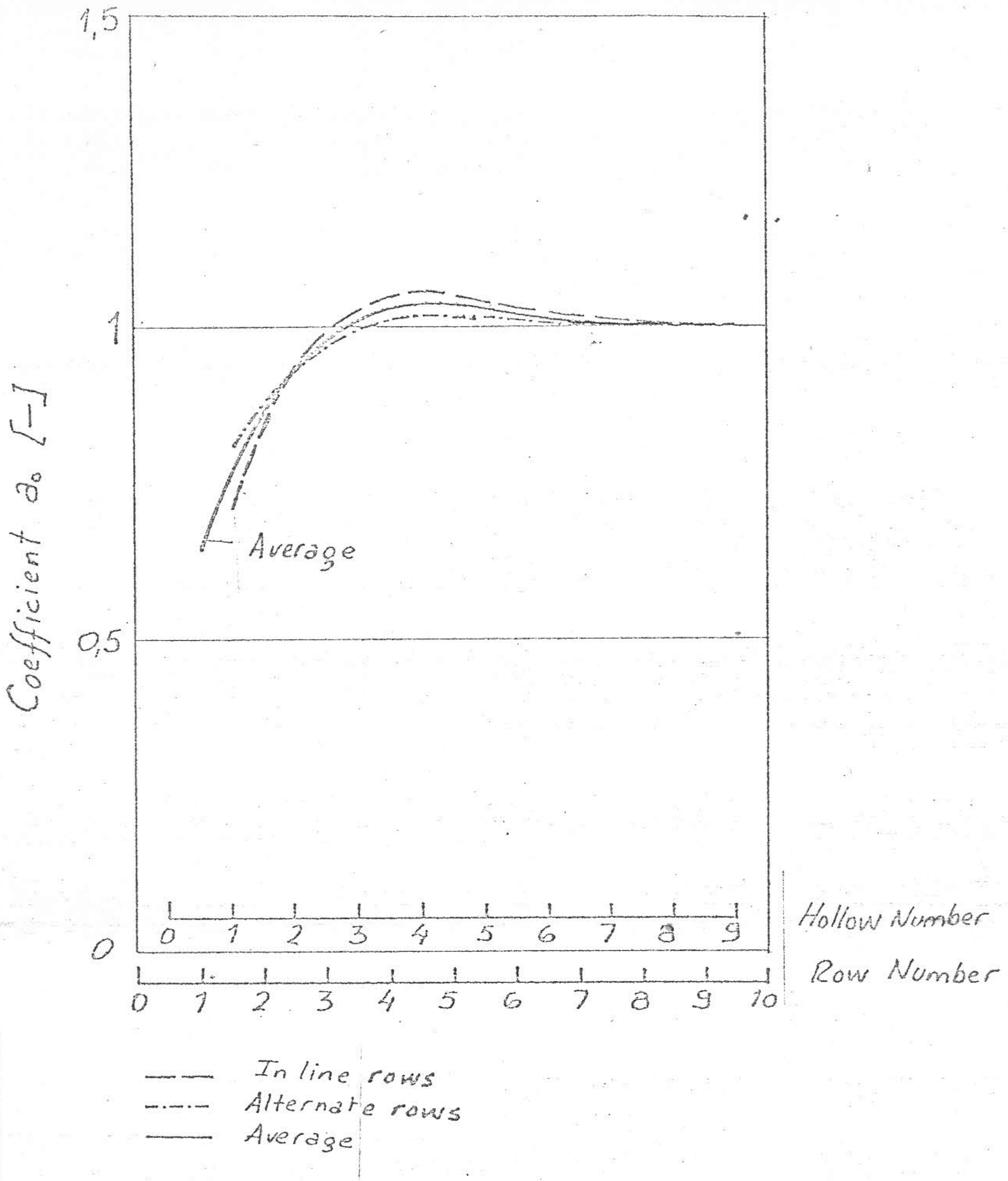


Fig. Nr. 4.7. : Heat Transfer in different hollows for a pressure drop equal to that of hollow Nr. 6... 7... 8 and followings.

By an analogous calculation it is obtained the curve given in Fig. 4.7. Similarly, a curve is obtained for in line arrangements and a mean curve of both is finally proposed.

3.- COMMENTS:

It can be observed, in Fig. 4.1, that the scatter is about ± 10% with a few points reaching ± 20%, which is considered very satisfactory in order to assess the usefulness of the proposed correlation. This is also verified within ±10% in the careful experiences reported in Ref. 5.2.8.

5.- APPENDIX:

5.1.- Corrections for $P_r \gg 1$

The correlation given in Sec. 2.1 is based on data given in Mc Adams (5.2.2) and Kays & London (5.2.1) in which the influence of P_r is included by the factor $P_r^{2/3}$. This is not important for gases, but it is not so for liquids with $P_r \gg 1$.

Deissler (5.2.7) has proposed, for flow inside tubes, a correlation in which the influence of P_r is given by $P_r^{0.55}$ in the range $P_r = 1 \ldots 10$. It is proposed to accept that correlation ot be extended to the field here dealt with and according to the best available knowledge.

Therefore, and in order to not changing the familiar 2/3 figure, the factor:

$$\left(\frac{P_r}{0,70}\right)^{1/9}$$

is added.

The figure 0,70 is to be understood as P_r for air since all experimental points given in Fig. 4.1 have been obtained using air as medium (Kays & London, Ref. 5.2.1, Page 2)

$$\left(\frac{h}{c_p \cdot G_{máx}}\right) \cdot P_r^{2/3} = 0,12 \cdot \frac{f^{1/3}}{R_e^{1/5}} \cdot \partial_o \cdot \left(\frac{P_r}{0,70}\right)^{1/9} \quad [-]$$

5.2.- References:

5.2.1.- W.M.KAYS & A.A. LONDON: "Compact heat exchangers" Mc Graw Hill Book Co. Inc., N.Y. (1958).

5.2.2.- MC ADAMS, W.H.: "Heat transmission" Mc. Graw Hill, (1954)

5.2.3.- GRIMISON: Transactions ASME 59 (1937) 583-94.

5.2.4.- NUNNER: "Wärmeubertragung und Drucksbfall in rahmen Rohren" VDI Forschungheft 455 (1956).

5.2.5.- LEDINEGG: "Dampferzeugung" Springer Verlag, Wien, (1952) p. 221.

5.2.6.- TER LINDEN: "Der Strömungswiderst and eines Rohrbundels" Wärme, (1939), p.319.

5.2.7.- DEISSLER: "The analysis of turbulent heat transfer, mass transfer and friction in smooth tubes at high Prandtl and Schmidt numbers" Harnett. Recent Advances in Heat and Mass Transfer. Mc Graw Hill, p. 253-310.

5.2.8.- M. CLER, D. SWETCHINE, S. VIANNAY et A. PIROVANO: "Essais comparativs de faisceaux de tubes à ailettes extérieures hélicoidales. Exchange thermique. Perte de pression". Rev. Gen. Therm., Fr., N° 133, Janvier 1973, p.23.