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(54) Optimum fin designs for downflow reboilers

(57) A plate-fin heat exchanger or downflow reboiler/condenser (6) has optimum heat transfer fin (28;36) dimensions to increase the efficiency of heat transfer between evaporating and condensing fluids such as cryogens (*e.g.*, oxygen and nitrogen). The plate-fin heat exchanger has a plurality of heat transfer fins (28;36) disposed between neighbouring parting sheets of uniform fin height less than or equal to 0.20 inch (0.05 cm) and either or both (i) the fin frequency is greater than or equal to 25 in^{-1} (10 cm^{-1}) and (ii) the fin thickness is less than or equal to 0.008 inch (0.02 cm). Such optimum fins are most effectively employed at least within the first heat transfer section (28A) downstream of the hardway distributor section of the evaporating stream passages and within the entire heat transfer section (36) between the inlet and outlet distributor sections of the condensing stream passages. 8

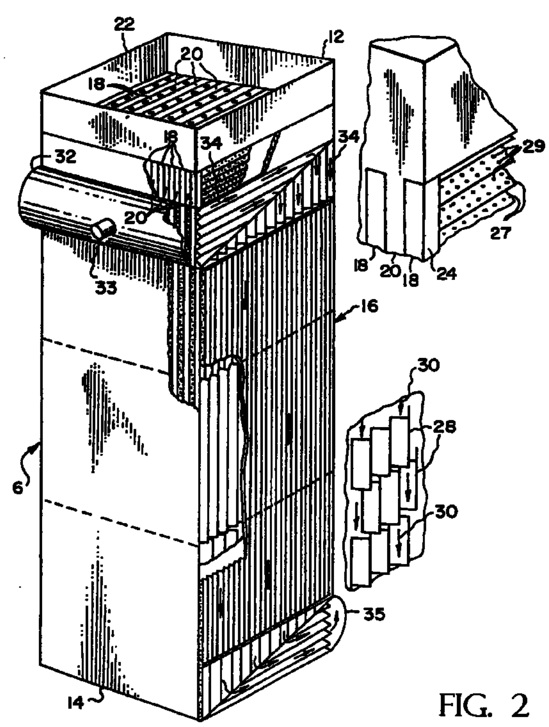


FIG. 2

EP 0 952 419 A1

Description

[0001] The invention relates to heat exchangers, and in particular plate-fin heat exchangers used as downflow reboilers in processes for cryogenic distillation of gas mixtures, such as distillation of air into its component elements.

[0002] Cryogenic separation of air is carried out by passing liquid and vapour in countercurrent contact through a distillation column. A vapour phase of the mixture ascends with an ever increasing concentration of the more volatile components (*e.g.*, nitrogen) while a liquid phase of the mixture descends with an ever increasing concentration of the less volatile components (*e.g.*, oxygen).

[0003] There are many processes for the separation of air by cryogenic distillation into its components. A generalized cryogenic air separation unit 10 is shown schematically in Figure 1. High pressure feed air 1 is fed into the base of a high pressure distillation column 2, where the air is separated into nitrogen-enriched vapour 5 and oxygen-enriched liquid 3. The oxygen-enriched liquid 3 is fed from the high pressure distillation column 2 into a low pressure distillation column 4. The low pressure distillation column 4 can be divided into multiple sections. Three such sections (4A, 4B, 4C) are shown in Figure 1 by way of example.

[0004] The nitrogen-enriched vapour 5 is passed into a reboiler condenser 6 where it is condensed to nitrogen-enriched liquid to provide reflux 7A for the high pressure distillation column 2 by exchanging heat with oxygen-enriched liquid to provide boilup to the low pressure column 4. The nitrogen-enriched liquid is partly tapped 8A and partly used as reflux 7A for the high pressure distillation column 2, and the remainder 8B is fed into the low pressure distillation column 4 as liquid reflux. In the low pressure distillation column 4, the liquid feeds (3,8B) are separated by cryogenic distillation into oxygen-rich and nitrogen-rich components. The nitrogen-rich component is removed as a vapour 9A. The oxygen-rich component is removed as another vapour 9B. Alternatively, the oxygen-rich component can be removed from a sump surrounding the reboiler/condenser 6 as a liquid. A waste stream 11 also is removed from the low pressure distillation column 4.

[0005] The present invention relates to a specific type of a plate-fin heat exchanger that is used as a reboiler/condenser 6 in such cryogenic processes. Such heat exchangers are commonly referred to as "downflow reboilers" or "falling film vaporizers", wherein heat is indirectly exchanged between a boiling or evaporating stream in one set of passages and a condensing stream in another set of passages. The terms "downflow" or "falling film" refer to the direction of flow of the evaporating stream. The usual flow of the condensing stream also is downward, although in special applications the condensing flow may be upward or cross-wise to the flow of the evaporating stream.

[0006] This type of heat exchanger can thermally link two distillation columns wherein a condensing stream containing nitrogen or argon is used as reflux at the top of one distillation column and an oxygen containing evaporating stream is used as boilup at the bottom of another distillation column, as illustrated in Figure 1 for nitrogen and discussed above.

[0007] Efforts to design and build more energy efficient air separation plants, especially of large size, have produced many advances and improvements in the performance and efficiency of distillation columns, compressors, pumps and expanders. Heat exchangers, specifically the reboilers/condensers, are a potential area for gains in energy efficiency and performance.

[0008] Plate-fin brazed aluminium heat exchangers are usually used for reboilers/condensers in air separation and other cryogenic applications. The heat transfer performance of such heat exchangers can be affected by varying the dimensions of the fins. For example, three key parameters which may be varied are fin height, fin thickness, and fin frequency.

[0009] Although downflow reboilers have been used in the cryogenic industry for many years, manufacturers of plate-fin heat exchangers have not provided optimized fin designs as taught by the present invention. Through analyses and experimentation, the present inventors have made certain surprising and unexpected discoveries pertaining to the optimum dimensions of fins for optimizing reboiler performance in air separation plants. These surprising and unexpected discoveries can be used to make the improved plate-fin heat exchanger of the present invention which, when used as a reboiler in an air separation plant, will result in a more efficient air separation process.

It is desired to have optimum fin designs which minimize the size, weight and/or cost of downflow reboilers, which will result in an air separation process more efficient and/or less expensive per unit quantity of product produced.

[0010] It is further desired to have a more efficient air separation process utilizing a downflow reboiler which is more compact and more efficient than prior art reboilers.

[0011] It also is further desired to have a method of assembling a plate-fin heat exchanger or downflow reboiler which utilizes optimized fin designs.

[0012] The present invention is based on the surprising and unexpected discovery that the use of relatively low fin height, low fin thickness, and high fin frequency provides the optimum fin design in a plate-fin heat exchanger for use as a downflow reboiler. Specifically, the present invention is a plate-fin heat exchanger or downflow reboiler which uses optimum fin dimensions as discussed below. The present invention also includes a method of assembling such a plate-fin heat exchanger and a cryogenic air separation unit having such a heat exchanger or downflow reboiler.

[0013] Although the present invention is described herein with reference to the use of the heat exchanger as a reboiler/condenser in an air separation facility, such reference is intended to point out the preferred utility; the present invention, however, can be used in any utility where such a heat exchange service is required.

[0014] In a first embodiment, the plate-fin heat exchanger has a plurality of fins disposed between neighbouring parting sheets in a substantially uniform fin frequency greater than or equal to 25 fins per inch (10 fins per cm), and each fin has a substantially uniform height that is less than or equal to 0.20 inch (0.5 cm).

[0015] In a second embodiment, the plate-fin heat exchanger has a plurality of fins disposed between neighbouring parting sheets, with each fin having a substantially uniform height less than or equal to 0.20 inch (0.5 cm) and a substantially uniform thickness less than or equal to 0.008 inch (0.02 cm).

[0016] In a third embodiment, the plate-fin heat exchanger has a plurality of fins disposed between neighbouring parting sheets in a substantially uniform fin frequency greater than or equal to 25 fins per inch (10 fins per cm), and each fin has a substantially uniform height that is less than or equal to 0.20 inch (0.5 cm) and a substantially uniform thickness less than or equal to 0.008 inch (0.02 cm). For example, in one such heat exchanger, the fin frequency is about 40 fins per inch (15 fins per cm), the fin height is about 0.100 inch (0.25 cm), and the fin thickness is about 0.004 inch (0.01 cm). In another such heat exchanger, the fin frequency is about 40 fins per inch (15 fins per cm), the fin height is about 0.100 inch (0.25 cm), and the fin thickness is about 0.008 inch (0.02 cm).

[0017] The fins may be plain, perforated, serrated, or wavy (herringbone). Typically, the fins are made of aluminium or an aluminium alloy. However, the fins also may be made of other thermally conductive materials.

[0018] A fourth embodiment is a plate-fin heat exchanger for reboiler or condenser service which comprises a parallelpipedal body that includes an assembly of a plurality of parallel parting sheets and a plurality of corrugated fins disposed between the adjacent parting sheets. The fins have a substantially uniform fin frequency greater than or equal to 25 fins per inch (10 fins per cm), and each fin has a substantially uniform height less than or equal to 0.20 inch (0.5 cm) and a substantially uniform thickness less than or equal to 0.008 inch (0.02 cm).

[0019] Another aspect of the present invention is a cryogenic air separation unit having a plate-fin heat exchanger according to the present invention. For example, the cryogenic air separation unit may have a plate-fin heat exchanger for reboiler or condenser service such as that described in the fourth embodiment above.

[0020] In yet another aspect of the present invention, there is provided a method of assembling a plate-fin heat exchanger according to the present invention, as described in the embodiments discussed above. The method of assembling such a heat exchanger comprises the following steps. The first step is to provide two substantially parallel parting sheets. The next step is to corrugate an elongate sheet to form a plurality of fins having a substantially uniform fin frequency, each fin having a substantially uniform height and a substantially uniform thickness. The final step is to dispose the corrugated sheet of fins between the parting sheets. In one embodiment of the method of assembling, the fin frequency is greater than or equal to 25 fins per inch (10 fins per cm), the fin height is less than or equal to 0.20 inch (0.5 cm) and the fin thickness is less than or equal to or equal to 0.008 inch (0.02 cm).

[0021] The following is a description by way of example only and with reference to the accompanying drawings of presently preferred embodiments of the invention. In the drawings:

Figure 1 is a schematic elevation of an air separation unit;
Figure 2 is an isometric drawing illustrating a downflow reboiler, including enlarged fragmentary details of hardway and easyway finning;
Figure 3A is a schematic representation of the evaporating stream passages of a heat exchanger used as a downflow reboiler illustrating an application of the present invention;
Figure 3B is a schematic representation of the condensing stream passages of a heat exchanger used as a downflow reboiler illustrating an application of the present invention;
Figure 4 is an exploded perspective view of a basic element or sub-assembly of a plate-fin heat exchanger;
Figures 5A-D illustrate four types of fins typically used in plate-fin heat exchangers;
Figure 6 is a schematic diagram illustrating the basic assembly of a fin between parting sheets and of the fluid streams in passages A and B, which are physically isolated from each other by the parting sheets;
Figure 7 is a schematic diagram illustrating the simplified geometry of finned passages based on an assumption that the passages are roughly rectangular;
Figure 8 is a graph which shows the dependence of [hA]* and [hAdT]* on the fin height H, at a constant fin thickness t_f of 0.008 inch (0.02 cm) and a constant fin frequency fpi of 25 in^{-1} (10 cm^{-1});
Figure 9 is a graph showing the dependence of [hA]* and [hAdT]* on fin thickness t_f, at a constant fin height H of 0.25 inch (0.65 cm) and a constant fin frequency fpi of 25 in^{-1} (10 cm^{-1});
Figure 10 is a graph which shows the dependence of [hA]* and [hAdT]* on fin frequency fpi, at a constant fin height H of 0.25 inch (0.65 cm) and a constant fin thickness t_f of 0.008 inch (0.02 cm);
Figure 11 is a graph which shows the performance characteristics of a prior art reboiler by plotting relative thermal driving force and relative duty per unit reboiler volume; and
Figure 12 is a graph which shows the performance characteristics of a reboiler according to the present invention

by plotting relative thermal driving force and relative duty per unit reboiler volume.

[0022] The downflow reboiler shown in Figure 2 is a plate-fin heat exchanger 6 having a first end 12 and a second end 14. The body 16 of the heat exchanger 6 has a generally parallelpipedal shape. The heat exchanger 6 includes a plurality of parting sheets that define a multitude of passages (18, 20) placed alternately to one another and arranged into a first group of passages 18 and a second group of passages 20 to receive different fluids. In the process described herein, for example, one group of passages receives the flow of oxygen and the other group receives the flow of nitrogen.

[0023] One group of passages 18 is adapted to receive the descending fluid from a top enclosure or open top pan-like device 22. Passages 18 are open at the top or first end 12 and at the bottom or second end 14 of the heat exchanger body 16. A side bar 24 closes the vertical ends of each passage 18. A portion of a typical side bar 24 is shown in the enlarged section to the right in Figure 2.

[0024] The passages (18, 20) each contain fins (27, 28, 36) formed by a corrugated sheet disposed between neighbouring parting sheets such that the fins are substantially perpendicular to the parting sheets. The fins are provided within the passages for fluid distribution and heat transfer. The fins and parting sheets typically are made of aluminium or an aluminium alloy, but may be made of other thermally conductive materials.

[0025] As shown in Figure 2, the passages 18 have a top portion fitted with horizontally placed fins 27 (shown in the enlarged section) containing perforations 29. This type of fin is called hardway finning and promotes even distribution of the fluid introduced through the pan-like section 22 into passages 18.

[0026] The bottom section of passages 18 includes vertically displaced heat transfer fins 28 (in the enlarged section), sometimes called easy-way fins, which receive fluid flow in the direction of the arrows 30. The heat transfer fins 28 shown in Figure 2 are serrated; however perforated, plain, herringbone (wavy) type or other similar type fins can be used.

[0027] Figures 5A-D show the four types of heat transfer fins commonly used in plate-fin heat exchangers-plain, perforated, serrated, and wavy (or herringbone).

[0028] In the embodiment shown in Figure 2, a boiling or evaporating liquid introduced into the pan-like device 22 flows downwardly through passages 18 in the heat exchanger 6 and exits by falling freely through the bottom end 14 of the heat exchanger, and is collected for other parts of the process by equipment (not shown) that is known in the art. A gas to be condensed is introduced into passages 20 of the heat exchanger via conduit 33 and header 32, and is conducted as shown by arrows 34 through a horizontal-vertical distributor, and is collected in a bottom header 35.

[0029] Figure 3A is a schematic representation of the heat exchanger 6 (shown in Figure 2) used as a downflow reboiler where a boiling/evaporating stream is introduced into the heat exchanger via the pan-like device 22 as shown by arrow 110. The boiling/evaporating stream is removed from the heat exchanger as shown by arrow 112. The incoming stream 110 is well distributed across the passages 18 by the hardway fins 27 and undergoes heat transfer in fins 28 as a falling film. Three sections of the heat transfer finning, shown as 28A, 28B and 28C, have progressively decreasing surface area, which is beneficial for wetting and good heat transfer in an evaporating stream as taught by US-A-5,122,174 (Sunder et al.).

[0030] Figure 3B is a schematic representation of the condensing passages of the heat exchanger 6 wherein a condensing stream 118 is introduced via header 32 and removed as a condensed stream 120 via bottom header 35. These passages have distribution sections 34 which are partly horizontal and partly vertical. The bulk of the heat transfer of condensation occurs as a vertically falling film in finning 36, shown in Figure 3B.

[0031] In the reboilers illustrated in Figures 2, 3A and 3B, the boiling/evaporating stream can be an oxygen-containing fluid, and the condensing stream can be a nitrogen and/or an argon-containing fluid.

[0032] The present invention pertains to the easy-way heat transfer finning which is oriented generally vertically, and is depicted as 28 in the evaporating stream passages in Figure 3A and as 36 in the condensing stream passages in Figure 3B.

[0033] Figure 4 illustrates further details of the basic sub-assembly 25 inside such plate-fin heat exchangers in terms of the evaporating stream finning 28; but the principles are also applicable to the condensing stream finning 36. A plate-fin heat exchanger includes a plurality of sub-assemblies 25 comprised of aluminium parting sheets (40,42) disposed on either side of a corrugated aluminium sheet 28, which forms a series of fins substantially perpendicular to the parting sheets. Each sub-assembly 25 is usually formed by brazing together two parting sheets (40,42) spaced apart by a fin sheet 28 with the edges enclosed by side bars (24A, 24B) as shown in Figure 4. A complete heat exchanger is assembled by brazing together a plurality of sub-assemblies 25 as well as other parts described earlier and shown in Figure 2.

[0034] Figure 6 shows the basic assembly of a fin between parting sheets. Since the streams in passages A and B are physically isolated from each other by the parting sheets, heat is exchanged only indirectly. A given heat exchanger may have many such passages, and in a downflow reboiler the streams generally will be laid out in an alternating fashion such that there is approximately one evaporating passage for each condensing passage. A fully assembled plate-fin heat exchanger has many more parts, including end- and side-bars to seal the extremities of the passages, cap sheets which cover the two outermost passages, finned distribution sections within the passages, and external headers and nozzles which serve to bring the respective process streams in a uniform manner in and out of the heat exchanger. Those features, however, are not relevant to the present invention, which optimizes the fin design in sections 28 and sections 36 of the heat exchanger represented in Figures 3A and 3B.

[0035] The key features of the present invention which achieve the optimum fin design and best reboiler performance are best discussed in terms of the analysis below. Although the present invention has more general applicability, for ease of discussion of the analysis, the evaporating and condensing fluids referred to are oxygen and nitrogen, respectively.

[0036] To help understand the heat transfer relationship between adjacent passages in the plate-fin heat exchanger, a simplified depiction of the geometry of finned passages is shown in Figure 7, where it is assumed that the passages are roughly rectangular. Each passage of stream A exchanges heat with one passage of stream B, except within the outermost passages of the heat exchanger. This end effect is minor and may be neglected because there are typically 70-100 passages for each stream in a heat exchanger.

Analysis of Heat Transfer Goodness Without Considering Pressure Drop

[0037] In the following theoretical analysis, a comparison is made of the effects of varying fin height, fin thickness and fin frequency on available heat transfer at a given flow rate of evaporating liquid into a volume that is allocated to the evaporating stream. This volume would include the space taken up by the metal in the plate-fin heat exchanger. To make comparisons of goodness, this initial superficial velocity is converted to an effective velocity for each geometry. Then, heat transfer areas and heat transfer coefficients are evaluated. Although the development and the sample calculations are shown for the evaporating side, similar results and conclusions also apply to the condensing side.

[0038] The terms used in this and following analyses are defined in the following Table.

Nomenclature		
Symbol	Description	Units
a,b	constants	
A_{eff}	effective area	Ft^2/in^3 (cm^2/cm^3) of passage A (length, width and height of multiple passages)
A_f	flow area	In^2/in (cm^2/cm) width of passage A
A_p	primary surface area	In^2/in (cm^2/cm) width and in length of passage A
A_s	secondary surface area	In^2/in (cm^2/cm) width and in length of passage A
A_{tot}	total area	Ft^2/in (cm^2/cm) width and in length of passage A
D_{eq}	equivalent diameter	In (cm)
DT	thermal driving force including effects of pressure drop	°F (K)
F	friction factor	
Fpi	fin frequency	In^{-1} (cm^{-1})
G	acceleration due to gravity	Ft/h^2 (cm/h^2)
H	fin height	In (cm)
hA xxx	hA at a reference Re_l of xxx	Btu/h °F (W/K)
[hA]* xxx	hA at the local geometry relative to hA at the reference geometry with Re_l of xxx	
h_{eff}	effective heat transfer coefficient	Btu/h ft^2 °F (W/m^2 K)
h_{Nus}	Nusselt heat transfer coefficient	Btu/h ft^2 °F (W/m^2 K)
k_l	liquid thermal conductivity	Btu/h ft °F (W/m^2 K)
k_{metal}	metal thermal conductivity	Btu/h ft °F (W/m^2 K)
ML	quantity defined in equation [14]	
P	wetted perimeter	In/in (cm/cm) width of passage A
P	stream pressure	Lbf/in^2 (kPa)
Q	superficial flow rate of stream A	$In^3/sec.\ in^2$ ($cm^3/sec.cm^2$) of space taken up by stream A (including one parting sheet per passage)

Nomenclature		
Symbol	Description	Units
Q_{eff}	effective flow rate of stream A	In^3/sec in^2 (cm^3/sec.cm^2) of flow area within passage A
Re_l	liquid Reynolds number	
Re_v	vapour Reynolds number	
T	stream temperature	°F (K)
Tanh	hyperbolic tangent function	
t_f	fin thickness	In (cm)
t_p	parting sheet thickness	In (cm)
V	vapour velocity	Ft/h (cm/h)
W	fin spacing	In (cm)
Xxx	subscript showing liquid Reynolds number for the reference geometry	
Δp	frictional pressure drop	Lbf/in^2 (kPa)
ΔT	temperature change due to frictional pressure drop	°F (K)
δ	Nusselt film thickness	T (m)
δT	thermal driving force without the effect of pressure	°F (K)
Γ	mass flow rate per unit perimeter	Lbm/ft h (Pa.s)
η	fin efficiency	
φ_{Re}	Reynolds number augmentation	
μ_l	liquid viscosity	Lbm/ft h (Pa.s)
ρ_l	liquid density	Lbm/ft^3 (kg/m^3)
ρ_v	vapour density	Lbm/ft^3 (kg/m^3)
ψ	Two phase multiplier	

[0039] Considering one passage of stream A, the following relationships exist:

$$W = 1/fpi - t_f \qquad [1]$$

$$P = 2*fpi*H + 2*(1-fpi*t_f) \qquad [2]$$

$$A_p = 2*(1-fpi*t_f) \qquad [3]$$

$$A_s = 2*fpi*H \qquad [4]$$

The parting sheets represent the primary area A_p and the fins represent the secondary area A_s.

$$A_f = H^*(1\text{-}fpi^*t_f) \quad [5]$$

$$D_{eq} = 4^*A_f/P \quad [6]$$

[0040] Since comparisons are made with equal superficial flow rates Q into the evaporating side volume of the heat exchanger, a conversion is made into an effective flow rate in order to calculate the relative performance, as follows.

$$Q_{eff} = Q^*[(H+t_p)/H]/(1\text{-}fpi^*t_f) \quad [7]$$

$$\Gamma = (Q_{eff}{}^*3600/1728)^*\rho_l{}^*A_f/(P/12) \quad [8]$$

$$Re_l = 4^*\Gamma/\mu_l \quad [9]$$

$$\delta = (3^*\mu_l{}^*\Gamma/\rho_l{}^2 g)^\wedge 1/3\rho \quad [10]$$

$$h_{Nus} = k_l/\delta \quad [11]$$

$$h_{eff} = h_{Nus}{}^*\varphi_{Re} \quad [12]$$

$$\varphi_{Re} = a^*\ Re_l{}^\wedge b \quad [13]$$

[0041] The basic equation for heat transfer through a falling laminar film was first derived by Nusselt in 1916. This is shown in equations [10] and [11]. In practice, higher values are obtained due to waves or turbulence which can be modelled using correction factors which involve the Reynolds number and the Prandtl number of the liquid film, as well as due to the shear stresses generated by the concurrent vapour flow which can be modelled by the thinning of the falling film. As comparisons will be made by taking a ratio within two different geometries under identical operating conditions, the last two terms will effectively cancel out, and only the Reynolds number term will remain. Based on some results reported in the literature, the "a" term is of the order of 0.7 and the "b" term is of the order of 0.1. This overall simplification introduces a small error in the fin efficiency calculation, which is discussed further below. This is a second order effect, however, and is therefore neglected. Further, when the flow rate in an evaporating stream is below a critical value, dry patches will appear that will reduce the heat transfer coefficients. This will limit the applicability of the teachings herein to only some portions of the evaporating side, as discussed further below. But such degradation does not occur within the condensing passages, and therefore the teachings herein apply to the entire condensing side.

$$mL = (H/2)^*[2^*h_{eff}/(t_f{}^*k_{metal}{}^*12)]^\wedge 0.5 \quad [14]$$

$$\eta = \tanh(mL)/mL \quad [15]$$

$$A_{tot} = (A\rho + \eta^*A_s)/144 \quad [16]$$

$$A_{eff} = A_{tot}/(H+t_p) \quad [17]$$

$$hA_{xxx} = h_{eff}{}^*A_{eff} \quad [18]$$

$$[hA]^*_{xxx} = hA_{xxx}/hA_{ref} \quad [19]$$

Sample Calculations

[0042] The sample calculations below pertain to the main reboiler/condenser of a double column air separation plant. The evaporating stream contains 99.5% pure oxygen at 25 psia (170 kPa). The properties of the liquid are: density = 70 lbm / ft^3 (1120 kg/in^3), thermal conductivity = 0.085 Btu / h ft F (0.147 W/m K) and dynamic viscosity = 0.42 lbm / ft h (38 Pa.s). The thermal conductivity of aluminium is 100 Btu / h ft F (173 W/m K). The parting sheet thickness is 0.041 inch (0.105 cm). The condensing side stream would be nearly pure nitrogen. However, the calculations below are specific to the oxygen stream side only.

[0043] The range of fin dimensions typically used in downflow reboilers are:

H = 0.20 - 0.30 in (0.5 — 0.75 cm)
t_f = 0.008 - 0.012 in (0.02 — 0.03 cm)

fpi = 15 - 25 in^{-1} (6 — 10 cm^{-1}).

These ranges are partly limited by the commercial availability.

[0044] To demonstrate the teachings of the invention, sample calculations are shown over a much wider range of dimensions as described below. Many of the dimensional combinations that would result at the extremes of the ranges below are not commercially available at present.

H = 0.10 - 0.40 in (0.25 — 1.0 cm)
t_f = 0.002 - 0.014 in (0.005 — 0.035 cm)
fpi = 10 - 40 in^{-1} (4 —16 cm^{-1})

[0045] Figure 8 shows the calculated dependence of [hA]* on the fin height H, at a constant fin thickness t_f of 0.008 inch (0.02 cm) and a constant fin frequency fpi of 25 in^{-1} (10 cm^{-1}). The underlying calculations were performed at the same superficial oxygen velocity such that the liquid Reynolds numbers at the central point within this range of fin heights are 100 and 500 for the two different curves. For each curve, the liquid Reynolds number will be different at all points away from the central point, but the effect of this variation is accounted for in this analysis. The measure of volume goodness, [hA]*, increases monotonically with a decrease in fin height H in the entire range considered. This suggests that employing lower fin heights would be beneficial in downflow reboilers.

[0046] Figure 9 shows the dependence of [hA]* on fin thickness t_f, at a constant fin height H of 0.25 inch (0.65 cm) and a constant fin frequency fpi of 25 in^{-1} (10 cm^{-1}) at two different superficial velocities, such that the liquid Reynolds numbers are 100 and 500 at the central point As shown, [hA]* increases monotonically with an increase in fin thickness in the entire range considered. This suggests that employing thicker fins would be beneficial in downflow reboilers.

[0047] Figure 10 shows the dependence of [hA]* on fin frequency fpi, at a constant fin height H of 0.25 inch (0.65 cm) and a constant fin thickness t_f of 0.008 inch (0.02 cm) at two different superficial velocities, such that the liquid Reynolds numbers are 100 and 500 at the central point. As shown, [hA]* increases monotonically with an increase in fin frequency in the entire range considered. This suggests that employing more fins per inch (fins per cm) would be beneficial in downflow reboilers.

Analysis of Heat Transfer Goodness Considering Pressure Drop

[0048] The following analysis includes the effect of pressure drop on heat transfer goodness. The available thermal driving force in a downflow reboiler is reduced by the frictional pressure drop. While the heat transfer characteristics are determined primarily by the liquid phase, the pressure drop is determined primarily by the vapour phase. If a plate-fin passage is thought of as analogous to a small pipe, the frictional pressure gradient may be expressed as

$$\Delta p/\Delta z = [(2f^*\rho_v V^2)/g^*D_{eq}]^*\Psi \qquad [20]$$

[0049] The constants to balance the dimensions in the above equation are not shown. It is well known that the friction factor fin the turbulent regime is proportional to the vapour Reynolds number or the velocity raised to -0.2 power. Since comparisons are made of different heat exchangers under similar operating conditions over any distance z within the exchanger where the relative liquid and vapour conditions would be similar, the dependence of equation [20] can be related back to the effective liquid flow rate yielding the following result (which only works for finding relative values between the different cases). Thus

$$\Delta p/\Delta z \;\alpha\; Q_{eff}^{\;1.8}/D_{eq} \qquad [21]$$

[0050] Further, the temperature and pressure of the streams in a downflow reboiler are not independent of each other. Both the evaporating and condensing side streams follow their saturation relationship of temperature versus pressure. From basic thermodynamics it can be shown that

$$\Delta T/\Delta z \;\alpha\; (T^2/p) * (\Delta p/\Delta z) \qquad [22]$$

[0051] Further, if the thermal driving force in a heat exchanger is δT without frictional pressure drop, and a reduction of ΔT occurs due to pressure drop then the average available driving force dT may be approximated as

$$dT = \delta T - (\Delta T/2) \qquad [23]$$

[0052] From equations [20-23] it follows that the available thermal driving force with two different geometries operating at the same superficial conditions may be stated as

$$dT/dT_{ref} = [1 - (\Delta T/2^*\delta T)]/[1-(\Delta T/2^*\delta T)_{ref}] \qquad [24]$$

with

$$(\Delta T/\Delta T_{ref}) = [Q_{eff}^{\ 1.8}/D_{eq}]/[Q_{eff}^{\ 1.8}/D_{eq}]_{ref} \qquad [25]$$

[0053] To evaluate the heat transfer goodness in the presence of pressure drop the following expressions are used.

$$hAdT_{xxx} = h_{eff}{}^*A_{eff}{}^*dT \qquad [26]$$

$$[hAdT]^*_{xxx} = hAdT_{xxx}/hAdT_{ref} \qquad [27]$$

Sample Calculations

[0054] Some sample calculations of [hAdT]* are shown in figures 8-10. For these calculations, the term $[1-(\Delta T/2^*\delta T)_{ref}$ in equation [24] is assumed to be 0.9 (at the reference condition) within each curve.

[0055] Figure 8 shows the dependence of [hAdT]* on the fin height H, at a constant fin thickness t_f of 0.008 inch (0.02 cm) and a constant fin frequency fpi of 25 in^{-1} (10 cm^{-1}). These calculations were performed at the same superficial oxygen velocity such that the liquid Reynolds numbers at the central point within this range of fin heights are 100 and 500 for the two different curves. For each curve, the liquid Reynolds number will be different at all points away from the critical point, but the effect of this variation is accounted for in this analysis. [hAdT]*, which is a measure of volume goodness, increases with a decrease in fin height H. But in a clear contrast to the [hA]* curves discussed earlier, the trend is not monotonic in the entire range considered. Rather, an optimum is indicated towards the lower end of the range considered. This suggests that employing relatively low fin heights would be beneficial in downflow reboilers.

[0056] Figure 9 shows the dependence of [hAdT]* on fin thickness t_f, at a constant fin height H of 0.25 inch (0.65 cm) and a constant fin frequency fpi of 25 in^{-1} (10 cm^{-1}) at two different superficial velocities, such that the liquid Reynolds numbers are 100 and 500 at the central point. As shown, [hAdT]* increases with an increase in fin thickness. But in a clear contrast to the [hA]* curves discussed earlier, the trend is not monotonic in the entire range considered. Rather, an optimum is indicated towards the upper end of the range considered. This suggests that employing fin thicknesses in this upper range would be beneficial in downflow reboilers.

[0057] Figure 10 shows the dependence of [hAdT]* on fin frequency fpi, at a constant fin height H of 0.25 inch (10 cm^{-1}) and a constant fin thickness t_f of 0.008 inch (0.02 cm) at two different superficial velocities, such that the liquid Reynolds numbers are 100 and 500 at the central point. As shown, [hAdT]* still increases monotonically with an increase in fin frequency in the entire range considered, although the rate of increase is somewhat reduced relative to the [hA]* curves. This suggests that employing a high fin frequency would be beneficial in downflow reboilers. Evaluation Of Good Solutions

[0058] To design efficient downflow reboilers, it is necessary to seek an optimized set for all of the three fin parameters (H, t_f and fpi). The results of the calculations above, shown in Figures 8-10, indicate that the selection of low fin height, medium to high fin thickness, and high fin frequency increases heat transfer goodness. When the effect of pressure drop is included, the three parameters are expected to be relieved away from the extreme values to obtain high heat transfer goodness. The worst heat transfer goodness is expected by the combination of the parameters at the opposite ends of each of the three ranges.

[0059] Some sample calculations of hA* and hAdT* are shown in Table 1.

Table 1: Heat transfer goodness as a function of fin dimensions at equal superficial velocities

Case	H		t(f)		fpi		Re(l)	[hA]*100	[hAdT]*100	Re(l)	[hA]*300	[hAdT]*300	Re(l)	[hA]*500	[hAdT]*500	
	in	cm	In	cm	In^{-1}	cm^{-1}	-	-	-	-	-	-	-	-	-	
1	0.100	0.255	0.004	0.010	10	4	174	0.667	0.697	523	0.636	0.665	871	0.624	0.652	
2	0.100	0.255	0.004	0.010	25	10	100	1.259	1.227	301	1.212	1.182	502	1.192	1.162	
3	0.100	0.255	0.004	0.010	40	16	71	1.898	1.663	212	1.837	1.609	353	1.811	1.586	
4	0.100	0.255	0.008	0.020	10	4	178	0.671	0.697	534	0.636	0.661	890	0.622	0.646	
5	0.100	0.255	0.008	0.020	25	10	104	1.277	1.184	311	1.217	1.129	518	1.193	1.106	
6	0.100	0.255	0.008	0.020	40	16	73	1.938	1.358	219	1.853	1.298	365	1.818	1.273	
7	0.100	0.255	0.012	0.030	10	4	182	0.661	0.682	545	0.626	0.645	909	0.611	0.630	
8	0.100	0.255	0.012	0.030	25	10	107	1.252	1.067	320	1.189	1.014	534	1.164	0.992	
9	0.100	0.255	0.012	0.030	40	16	76	1.898	0.514	227	1.807	0.490	378	1.769	0.479	
10	0.250	0.635	0.004	0.010	10	4	204	0.401	0.432	611	0.399	0.430	1019	0.399	0.430	
11	0.250	0.635	0.004	0.010	25	10	99	0.839	0.864	296	0.854	0.880	493	0.862	0.888	
12	0.250	0.635	0.004	0.010	40	16	65	1.301	1.252	195	1.337	1.287	325	1.355	1.304	
13	0.250	0.635	0.008	0.020	10	4	206	0.454	0.488	618	0.446	0.479	1031	0.443	0.476	
14	0.250	0.635	0.008	0.020	25	10	100	1.000	1.000	300	1.000	1.000	500	1.000	1.000	Ref.
15	0.250	0.635	0.008	0.020	40	16	66	1.586	1.342	198	1.599	1.353	330	1.605	1.357	
16	0.250	0.635	0.012	0.030	10	4	209	0.477	0.510	626	0.465	0.497	1043	0.459	0.492	
17	0.250	0.635	0.012	0.030	25	10	101	1.074	1.022	304	1.061	1.010	507	1.056	1.005	
18	0.250	0.635	0.012	0.030	40	16	67	1.722	0.960	201	1.713	0.955	335	1.709	0.952	
19	0.400	1.015	0.004	0.010	10	4	215	0.272	0.295	646	0.276	0.299	1077	0.278	0.302	
20	0.400	1.015	0.004	0.010	25	10	98	0.575	0.599	294	0.597	0.621	490	0.609	0.634	
21	0.400	1.015	0.004	0.010	40	16	63	0.891	0.873	190	0.933	0.914	317	0.956	0.937	
22	0.400	1.015	0.008	0.020	10	4	217	0.327	0.354	651	0.330	0.357	1086	0.332	0.359	
23	0.400	1.015	0.008	0.020	25	10	99	0.732	0.743	297	0.756	0.767	495	0.767	0.779	
24	0.400	1.015	0.008	0.020	40	16	64	1.160	1.017	192	1.208	1.058	320	1.231	1.079	
25	0.400	1.015	0.012	0.030	10	4	219	0.360	0.388	657	0.361	0.389	1095	0.362	0.390	
26	0.400	1.015	0.012	0.030	25	10	100	0.831	0.808	300	0.849	0.826	499	0.858	0.835	
27	0.400	1.015	0.012	0.030	40	16	65	1.332	0.823	194	1.374	0.849	323	1.393	0.861	
					Max			1.938	1.663		1.853	1.609		1.818	1.586	
					Min			0.272	0.295		0.276	0.299		0.278	0.302	

[0060] Two of the three parameters, fin height H and fin frequency fpi, are considered in their entire original range. But the third parameter, fin thickness t_f, is considered in a slightly smaller range than before for two reasons. First, there is a practical limit on how thin fins can be formed and brazed into a plate-fin heat exchanger. Second, far more extreme process combinations occur when the three parameters are changed simultaneously over wide ranges. In order to maintain symmetry about the central point, a range of 0.004 - 0.012 inch (0.01 - 0.02 cm) was used for the calculations.

[0061] The calculations were performed at constant superficial velocities within each case wherein the liquid Reynolds numbers at the central reference point are 100, 300 or 500. As shown in Table 1, the worst combinations occur with high fin height H, low fin thickness t_f and low fin frequency fpi (case 19). The combinations for the best performance are unexpected-low fin heights, low to medium fin thicknesses, and high fin frequencies (cases 3 and 6). This is surprising, considering the fact that a low fin thickness yields low fin efficiency which affects heat transfer goodness. When pressure drop is included, the best performance is obtained with very thin fins.

Summary Of Calculation Results And Limitations

[0062] The above analysis demonstrates the unexpected result that the use of low fin height, low fin thickness, and high fin frequency is the optimum fin design in a plate-fin heat exchanger for use as a downflow reboiler. However, some qualifications apply, as discussed below.

[0063] The overall shape of a downflow reboiler, in terms of its length, width, and height, often is limited by the available space to fit the reboiler into in a given air separation plant. Due to this, the use of the fins taught herein may not be employed in the entire heat exchanger. (Further criteria are discussed below so that the current invention can be applied where the benefit is the maximum.) As an evaporating film descends down a vertical surface, its liquid Reynolds number decreases continuously while its vapour Reynolds number increases simultaneously. It is well known that evaporating films tend to form dry patches when the film gets very thin resulting in degradation of heat transfer. For this reason, the teachings herein are most useful near the entrance and within the upper part of the evaporating side in the heat transfer section of a downflow reboiler. This is the section represented by finning 28A in Figure 3A. For regions further below (finning 28B, 28C), the fin frequency (fpi) may be progressively reduced. Further, for both mechanical and thermal reasons, the fin thickness (t_f) may be increased simultaneously with this decrease in fin frequency (fpi). As the geometry of the plate-fin heat exchanger does not allow an easy change of the fin height, the optimum design would maintain a low fin height (H) in the entire length of the heat exchanger (27, 28A, 28B, 28C).

[0064] It should be noted that although the above description has been made in terms of three finned sections 28A, 28B and 28C, the most general application of the current invention would be to the first heat transfer section of the evaporating side passages which may contain one or more types of finned sections (of progressively decreasing surface area).

[0065] Although the above calculations were shown with respect to the evaporating side stream, similar calculations will lead to the same conclusions for the condensing side stream. In other words, low fin height, low fin thickness, and high fin frequency also will be beneficial for the condensing side passages. Also, the teachings herein may be applied to the entire condensing side stream passages (36) because the phenomenon of heat transfer degradation by dry patches does not occur in condensation. It is, however, possible to change the fin density in the opposite fashion from that of the evaporating side, so that the fin frequency may go from low near the inlet at the top to high approaching the outlet at the bottom when the heat exchanger space is severely restricted.

Experimental Results

[0066] To test the validity of the above analysis, two downflow reboilers were constructed and tested. Some dimensions of these heat exchangers are:

Case	Prior art	Present invention
Length, in (cm)	42 (107)	42 (107)
Width, in (cm)	20 (51)	18 (46)
Stack height, in (cm)	5.25 (13.3)	5.25 (13.3)
Evaporating side		
Fin height H, in (cm)	0.281 (0.715)	0.160 (0.405)
Fin frequency fpi, in^{-1} (cm^{-1})	18.2 (7.15)	28 (11.0)
Fin thickness t_f, in (cm)	0.008 (0.02)	0.006 (0.015)
Condensing side		
Fin height H, in (cm)	0.281 (0.715)	0.160 (0.405)
Fin frequency fpi, in^{-1} (cm^{-1})	20.2 (7.95)	28 (11.0)
Fin thickness t_f, in (cm)	0.010 (0.025)	0.006 (0.015)

[0067] All three fin dimensions (H, t_f, fpi) of the prototype of the present invention were changed relative to the prior art in the direction taught by the present invention as optimum for downflow reboilers. Thus, a lower fin height, a higher fin frequency, and a lower fin thickness were used in that prototype as compared to the fin dimensions of the prior art reboiler. Although these values are not at the extremes shown in the sample calculations above, substantial performance improvement has been verified, as discussed below. Except for the dimensional differences of the heat transfer fins noted above, the two heat exchangers are similar in all other features. This includes the specific types of heat transfer fins and all the other details, such as distributor fins, headers, and nozzles Experiments were carried out using evaporating nitrogen against condensing nitrogen in a closed loop system. The results are shown in Figures 11 and 12.

[0068] Figure 11 shows the performance characteristics of the prior art reboiler. The relationship between the duty and the external thermal driving force is shown as a function of the quality at the outlet of the reboiler. The quality refers to the fraction that is vapour relative to the total flow on the evaporating side. Figure 12 shows similar plots for the reboiler according to the present invention. At equivalent thermal driving force conditions, the present invention achieves about 1.5 times more duty (per unit volume). Stated differently, for the same duty, the present invention requires less external thermal driving force by the reciprocal of the above ratio (*i.e.*, 2/3) in comparison with the prior art reboiler. These results are consistent with the analysis presented above.

Claims

1. A plate-fin heat exchanger (6) having a plurality of heat transfer fins (28;36) disposed between neighbouring parting sheets in a substantially uniform fin frequency, each fin (28;36) having a substantially uniform height, wherein

   the fin height is less than or equal to 0.5 cm (0.20 inch), and either or both

   (i) the fin frequency is greater than or equal to 10 cm^{-1} (25 in^{-1}) and.
   (ii) each fin has a substantially uniform thickness of less than or equal to 0.02 cm (0.008 inch).

2. A heat exchanger as claimed in Claim 1, wherein the fin frequency is greater than or equal to 10 cm^{-1} (25 in^{-1}).

3. A heat exchanger as claimed in Claim 2, wherein each fin has a substantially uniform thickness less than or equal to 0.02 cm (0.008 inch).

4. A heat exchanger as claimed in Claim 1, wherein each heat transfer fin (28;36) has a substantially uniform thickness of less than or equal to 0.02 cm (0.008 inch).

5. A heat exchanger as claimed in Claim 1, wherein each heat transfer fin (28;36) has a substantially uniform height and a substantially uniform thickness, wherein

the fin thickness is less than or equal to 0.02 cm (0.008 inch),
the fin height is less than or equal to 0.5 cm (0.20 inch), and
the fin frequency is greater than or equal to 10 cm^{-1} (25 in^{-1}).

6. A heat exchanger as claimed in Claim 5, wherein

the heat transfer fin thickness is about 0.01 cm (0.004 inch),
the fin height is about 0.25 cm (0.100 inch), and
the fin frequency is about 16 cm-1 (40 in-1).

7. A heat exchanger as claimed in Claim5, wherein

the fin thickness is about 0.2 cm (0.008 inch),
the fin height is about 0.25 cm (0.100 inch), and
the fin frequency is about 16 cm^{-1} (40 in^{-1}).

8. A plate-fin heat exchanger as claimed in any one of the preceding claims, wherein the fins are made of aluminium or an aluminium alloy.

9. A plate-fin heat exchanger (6) as claimed in any one of the preceding claims for reboiler or condenser service, wherein the exchanger comprises a parallelpipedal body (16) including an assembly of a plurality of parallel parting sheets and a plurality of corrugated fins (28) disposed between adjacent parting sheets to provide the said heat transfer fins of substantially uniform height.

10. A heat exchanger as claimed in any one of the preceding claims which is a downflow reboiler (6) having a generally parallelpipedal body formed by an assembly of substantially parallel vertically extending passages adapted to receive a first fluid introduced into a first group of passages (18) and a second fluid introduced into a second group of passages (20), the passages in the second group of passages (20) alternating in position with the passages in the first group of passages (18), the first group of passages (18) having a plurality of fins (27,28) disposed between neighbouring parting sheets, the fins including hardway fins (27) for fluid distribution of the first fluid and easyway heat transfer fins (28) downstream of the hardway fins, the easyway heat transfer fins (28) forming one or more "first group" heat transfer sections (28A,28C,28B) with progressively decreasing surface area, at least the heat transfer fins in the or the first of the first group heat transfer section (28A) constituting the said heat transfer fins of substantially uniform height.

11. A downflow reboiler as claimed in Claim 10, wherein the succeeding heat transfer sections following the first heat transfer section have a progressively decreasing fin frequency.

12. A downflow reboiler as claimed in Claim10 or Claim 11, wherein the succeeding heat transfer sections following the first heat transfer section have a progressively increasing fin thickness.

13. A heat exchanger as claimed in any one of the preceding claims, which is a downflow reboiler (6) having a generally parallelpipedal body (16) formed by an assembly of substantially parallel vertically extending passages adapted to receive a first fluid introduced into a first group of passages (18) and a second fluid introduced into a second group of passages (20), the passages in the second group of passages (20) alternating in position with the passages in the first group of passages (18), the second group of passages (20) having a plurality of fins (34,36) disposed between neighbouring parting sheets, the fins including inlet and outlet distribution fins (34) for uniform flow of the second fluid into and out of the second group of passages (20) and heat transfer fins (36) forming at least one heat transfer section between the inlet and outlet distribution fins (34), at least the heat transfer fins (36) in the or one "second group" heat transfer section constituting said fins of substantially uniform height.

14. A cryogenic air separation unit having a heat exchanger (6) as claimed in any one of the preceding claims.

15. A method of assembling a heat exchanger as claimed in any one of the preceding claims, comprising the steps of:

providing two substantially parallel parting sheets;
corrugating an elongate sheet to form a plurality of said heat transfer fins; and
disposing the corrugated sheet of fins between the parting sheets.

16. A column of an air separation plant having a downflow reboiler (6) as claimed in any one of Claims 10 to 13 in which, in operation, a liquid oxygen containing stream passes through the first group of passages (18) in parallel flow to a nitrogen and/or argon containing stream in the second group of passages (20).

17. The use of fins as defined in any one of Claims 1 to 8 to increase the efficiency of heat transfer between evaporating and condensing fluids in a plate fin heat exchanger.

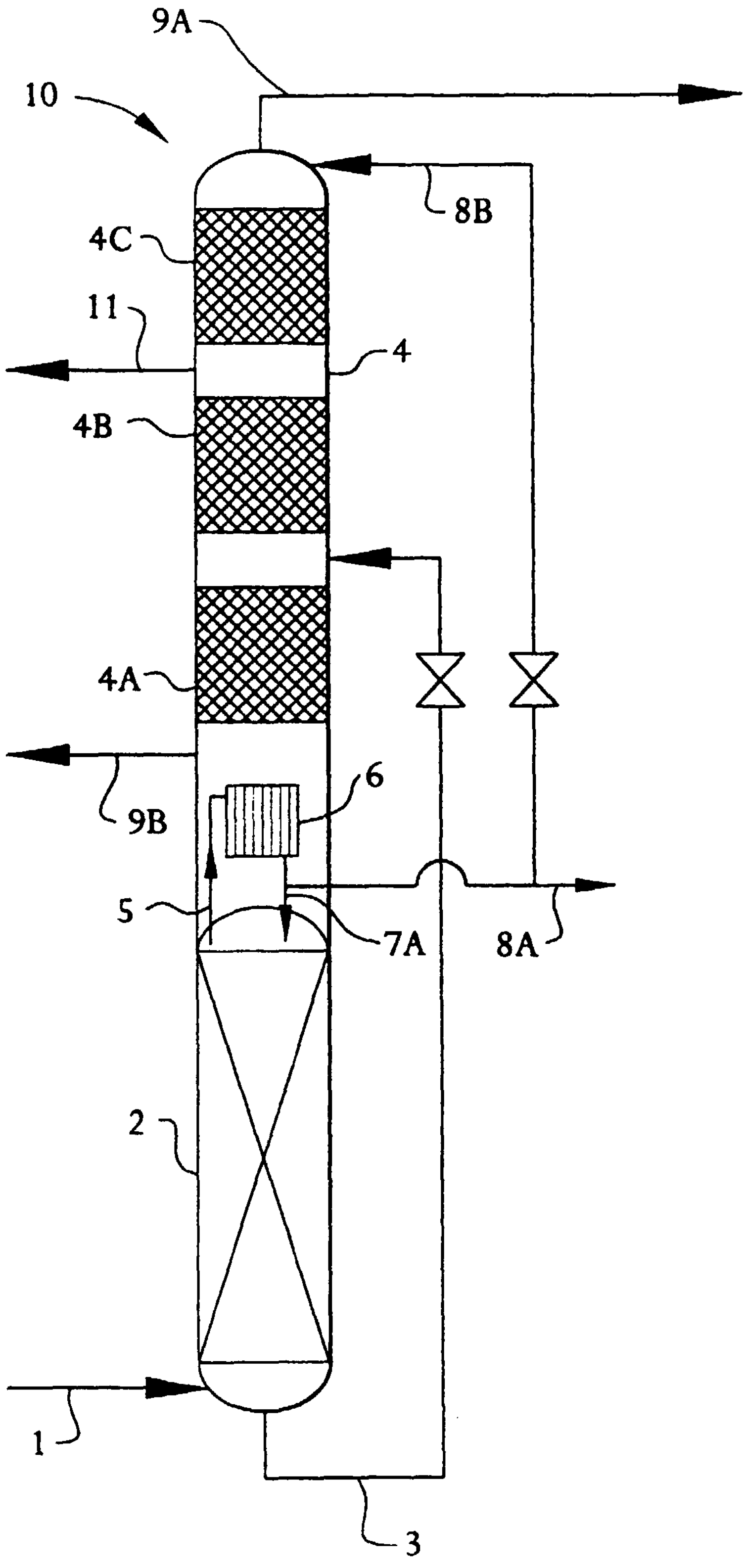


FIG. 1

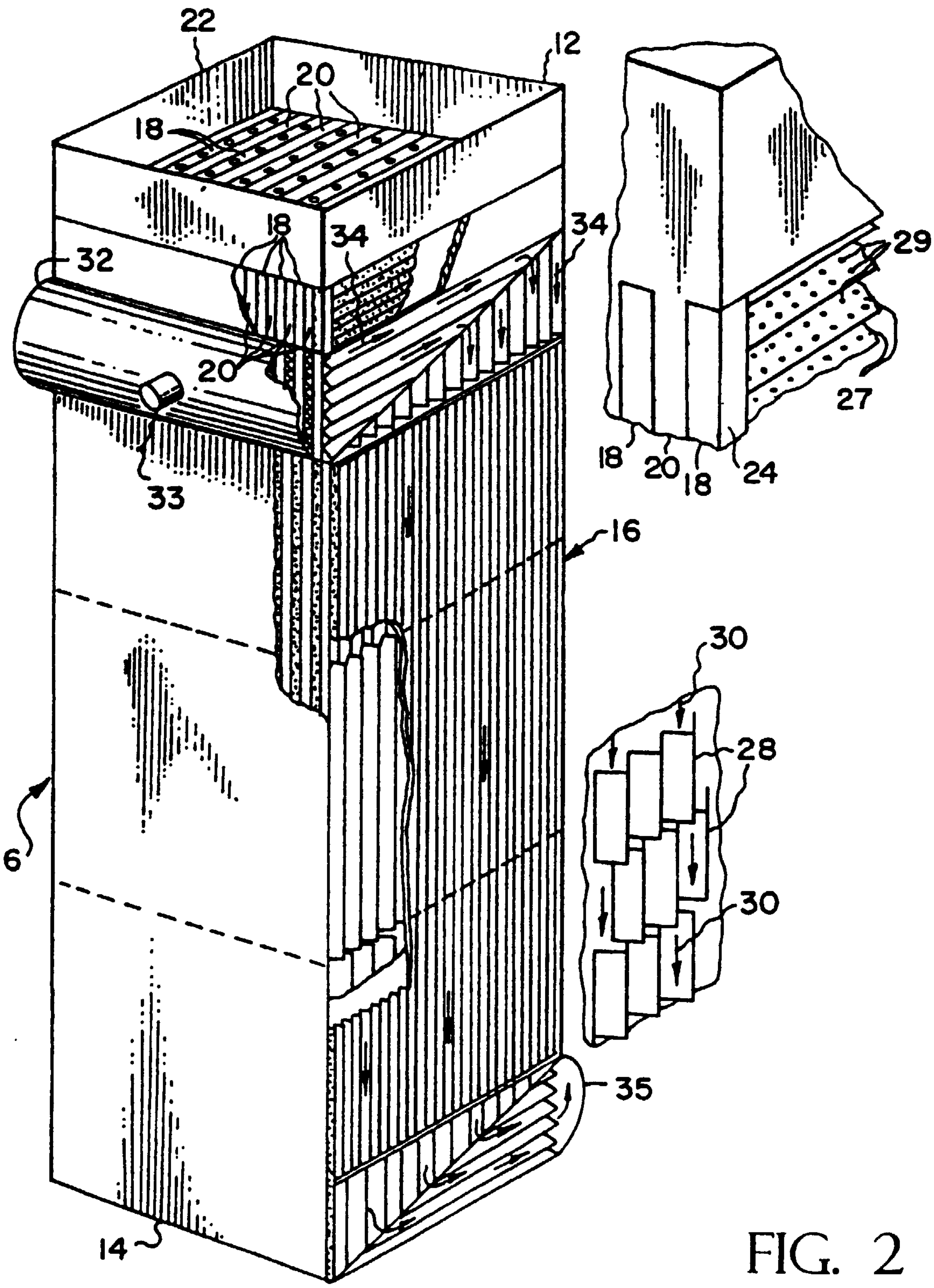


FIG. 2

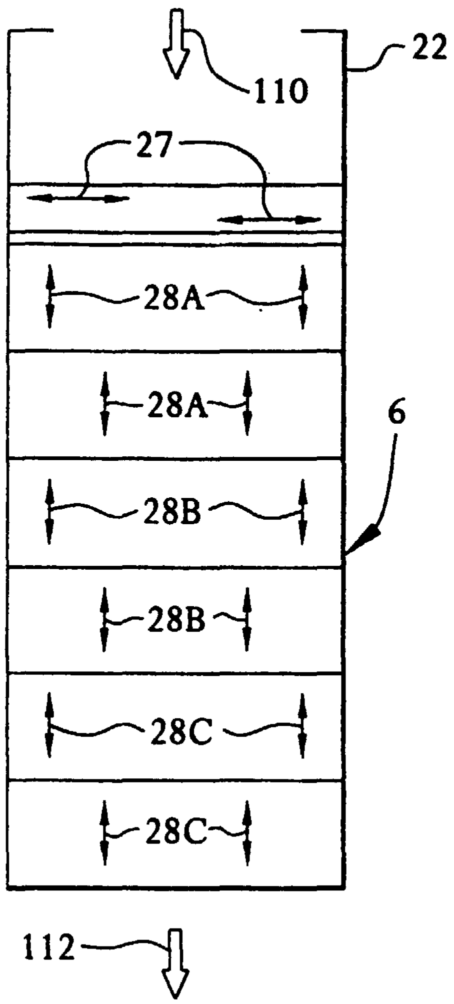

110
22
27
28A
28A
28B
6
28B
28C
28C
112

FIG. 3A

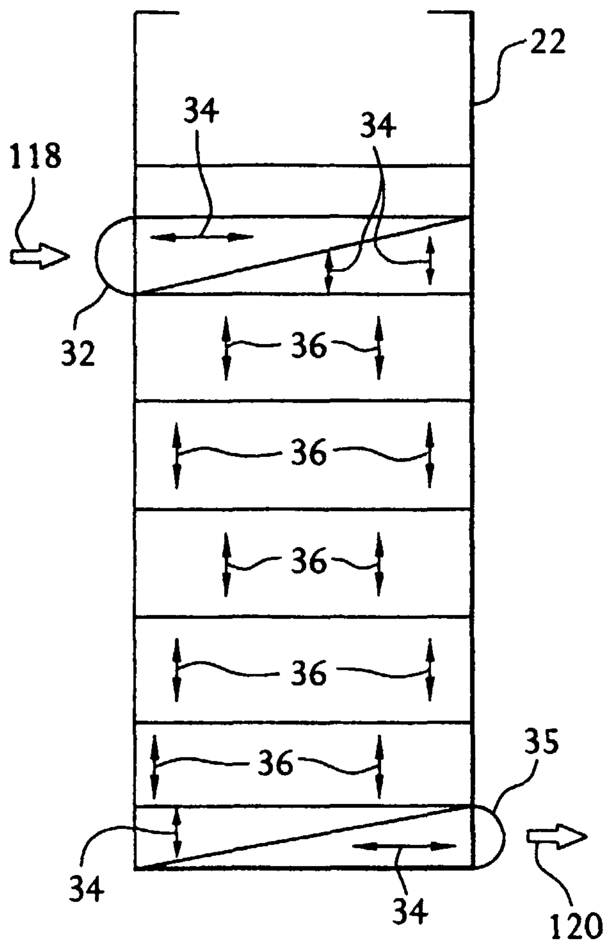

22
34
34
118
32
36
36
36
36
36
35
34
34
120

FIG. 3B

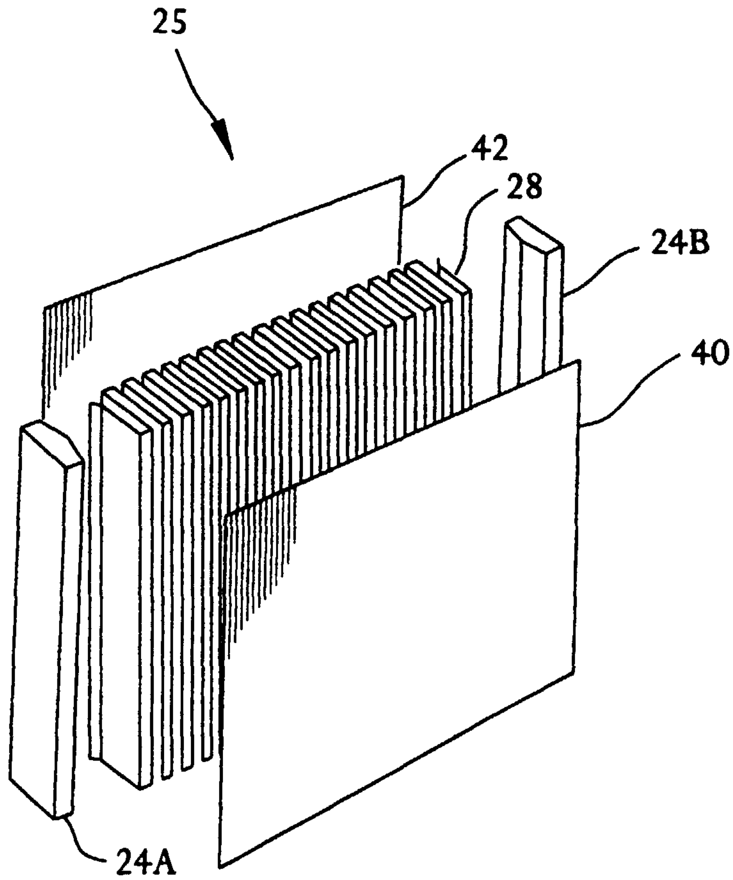


FIG. 4

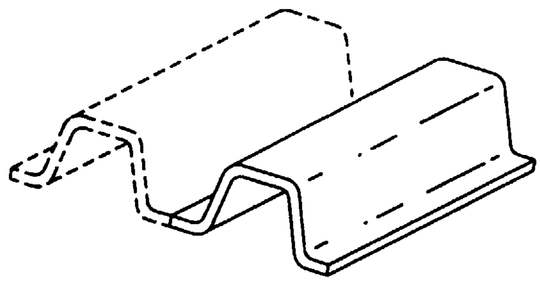


FIG. 5A

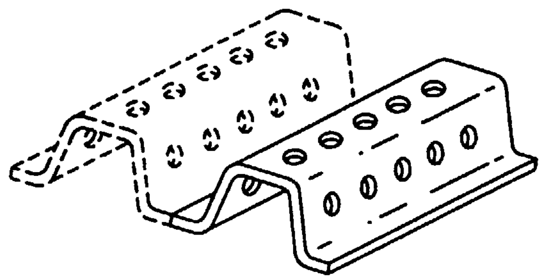


FIG. 5B

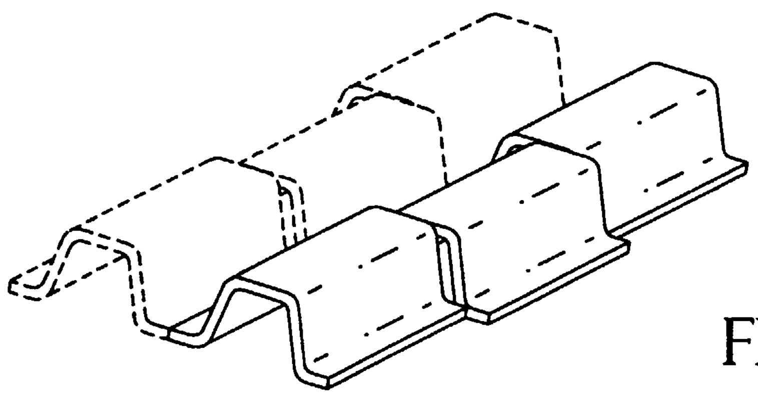


FIG. 5C

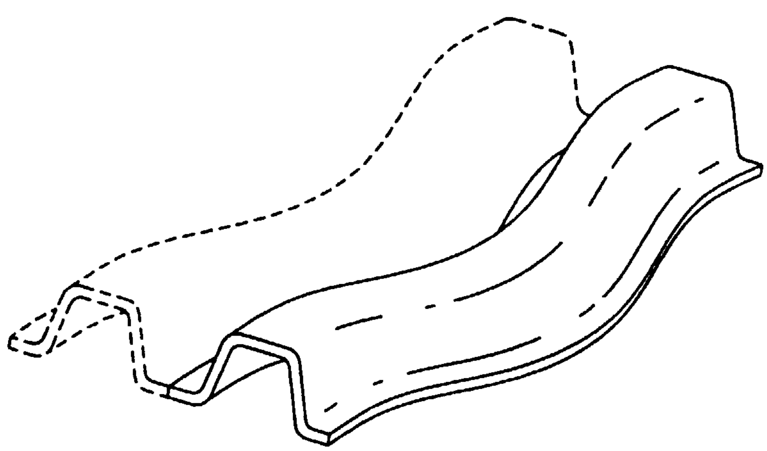


FIG. 5D

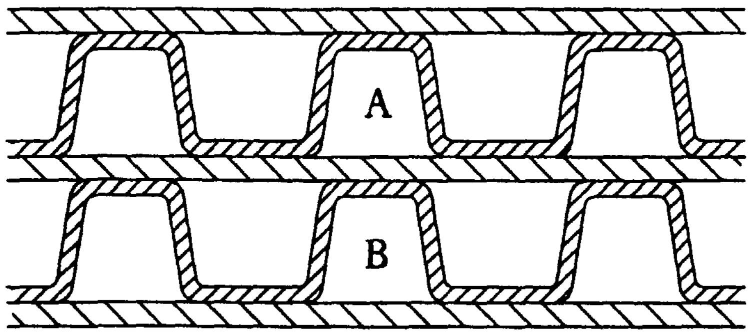


FIG. 6

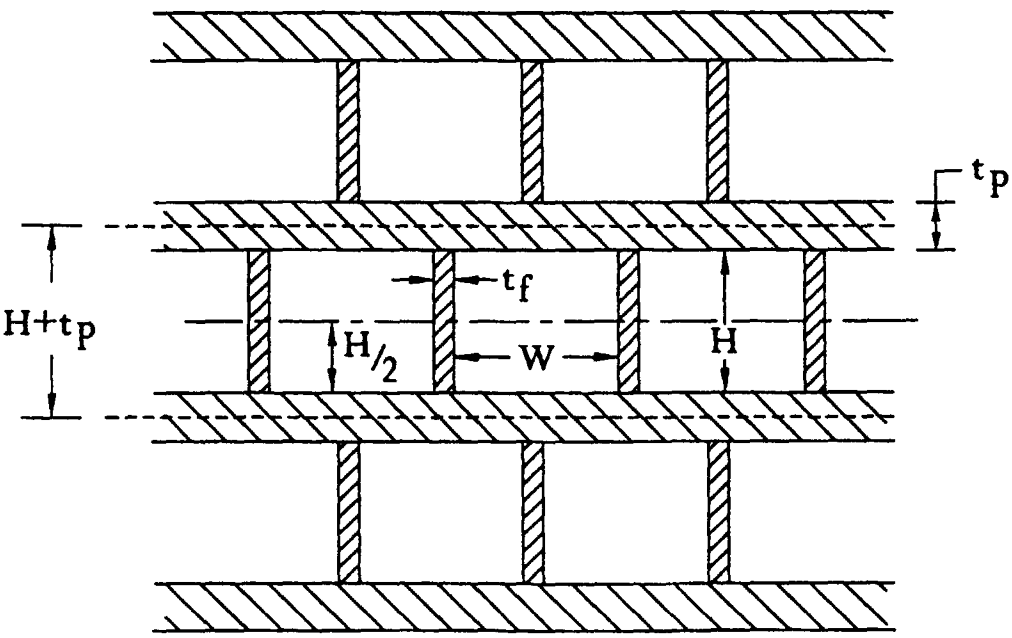


FIG. 7

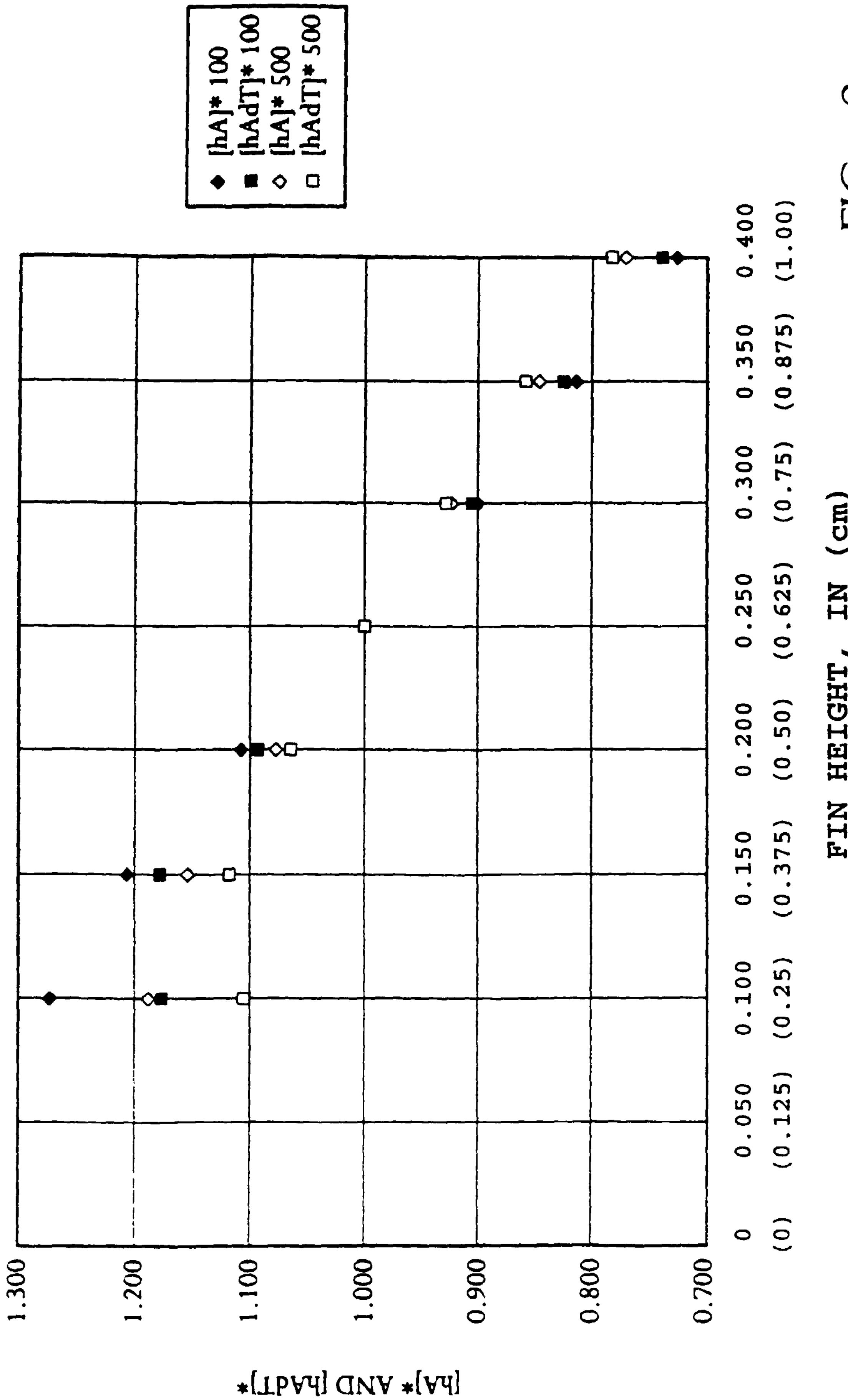

[hA]* 100
[hAdT]* 100
[hA]* 500
[hAdT]* 500
1.300
1.200
1.100
1.000
0.900
0.800
0.700
[hA]* AND [hAdT]*
0 (0)
0.050 (0.125)
0.100 (0.25)
0.150 (0.375)
0.200 (0.50)
0.250 (0.625)
0.300 (0.75)
0.350 (0.875)
0.400 (1.00)
FIN HEIGHT, IN (cm)

FIG. 8

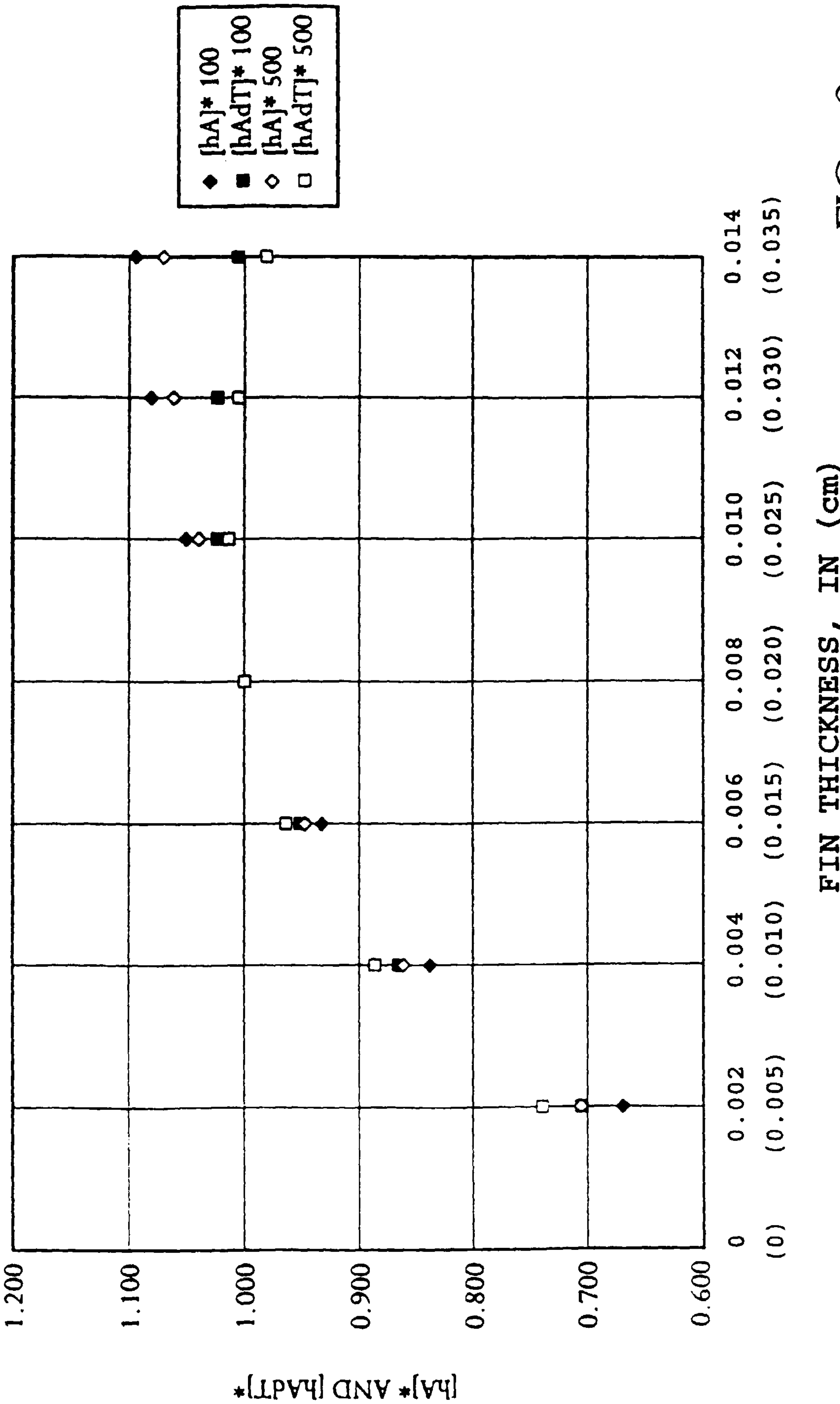
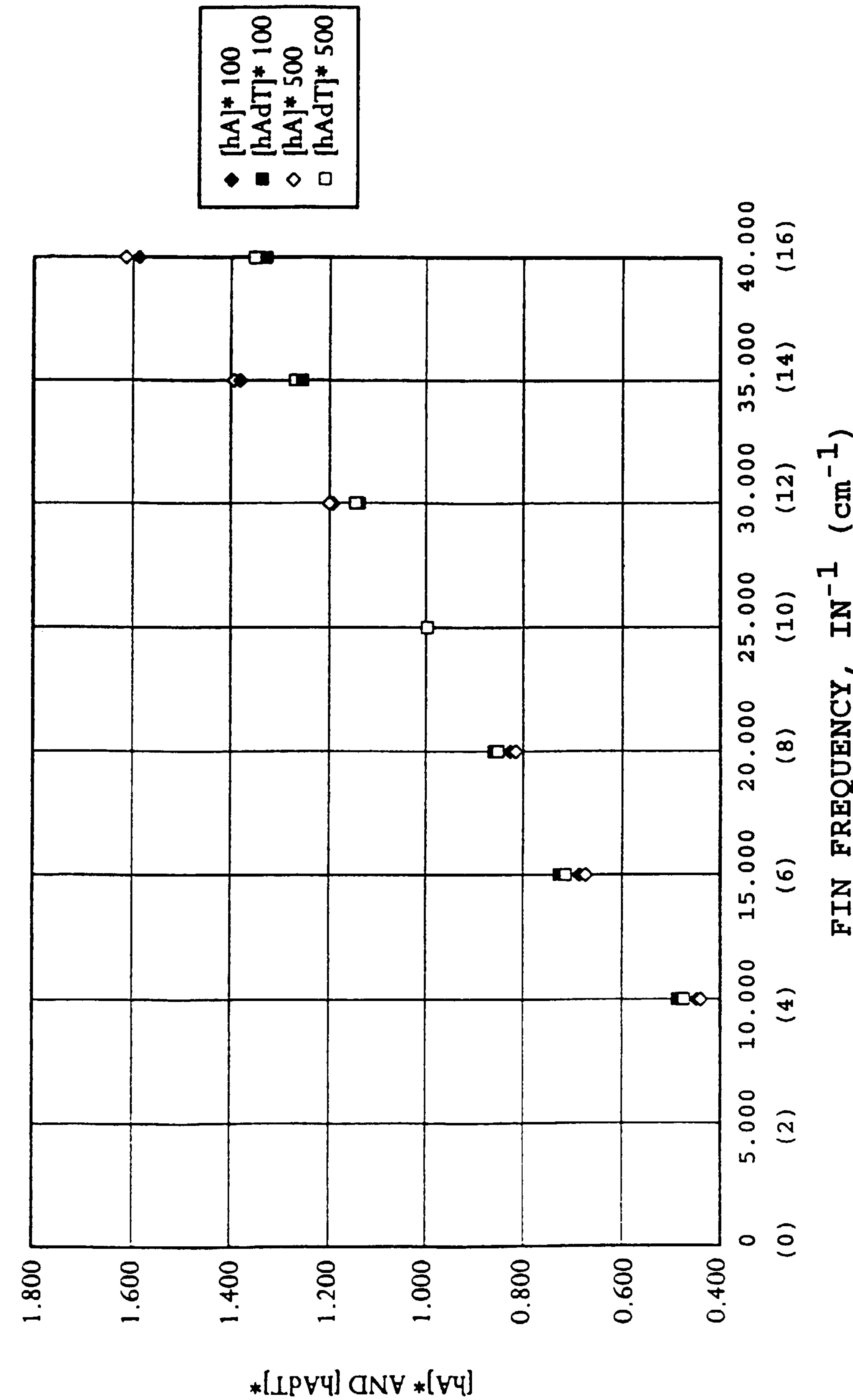

[hA]* 100
[hAdT]* 100
[hA]* 500
[hAdT]* 500
1.200
1.100
1.000
0.900
0.800
0.700
0.600
[hA]* AND [hAdT]*
0
(0)
0.002
(0.005)
0.004
(0.010)
0.006
(0.015)
0.008
(0.020)
0.010
(0.025)
0.012
(0.030)
0.014
(0.035)
FIN THICKNESS, IN (cm)

FIG. 9



[hA]* 100
[hAdT]* 100
[hA]* 500
[hAdT]* 500
[hA]* AND [hAdT]*
1.800
1.600
1.400
1.200
1.000
0.800
0.600
0.400
0 (0)
5.000 (2)
10.000 (4)
15.000 (6)
20.000 (8)
25.000 (10)
30.000 (12)
35.000 (14)
40.000 (16)
FIN FREQUENCY, IN^{-1} (cm^{-1})

FIG. 10

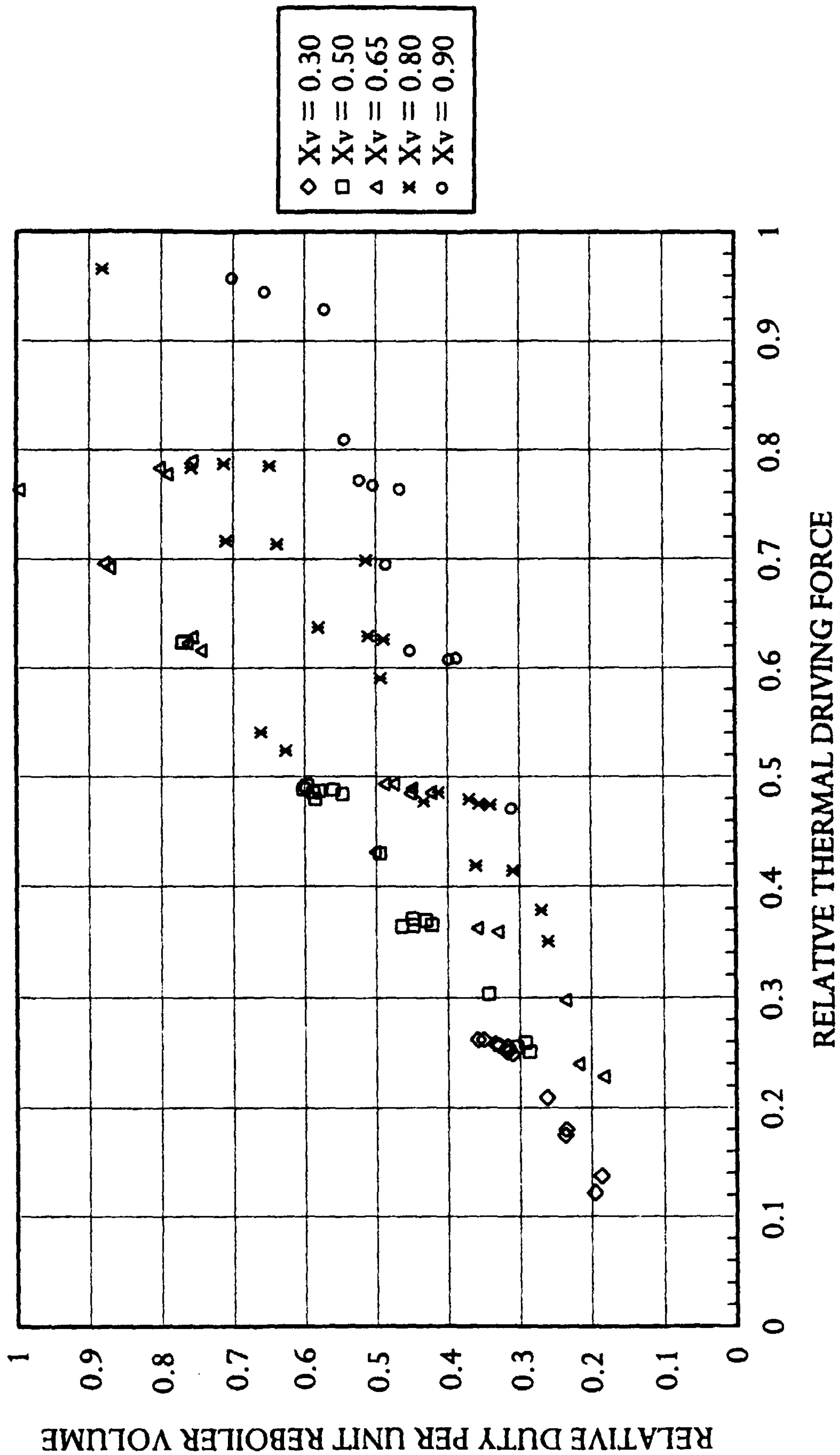
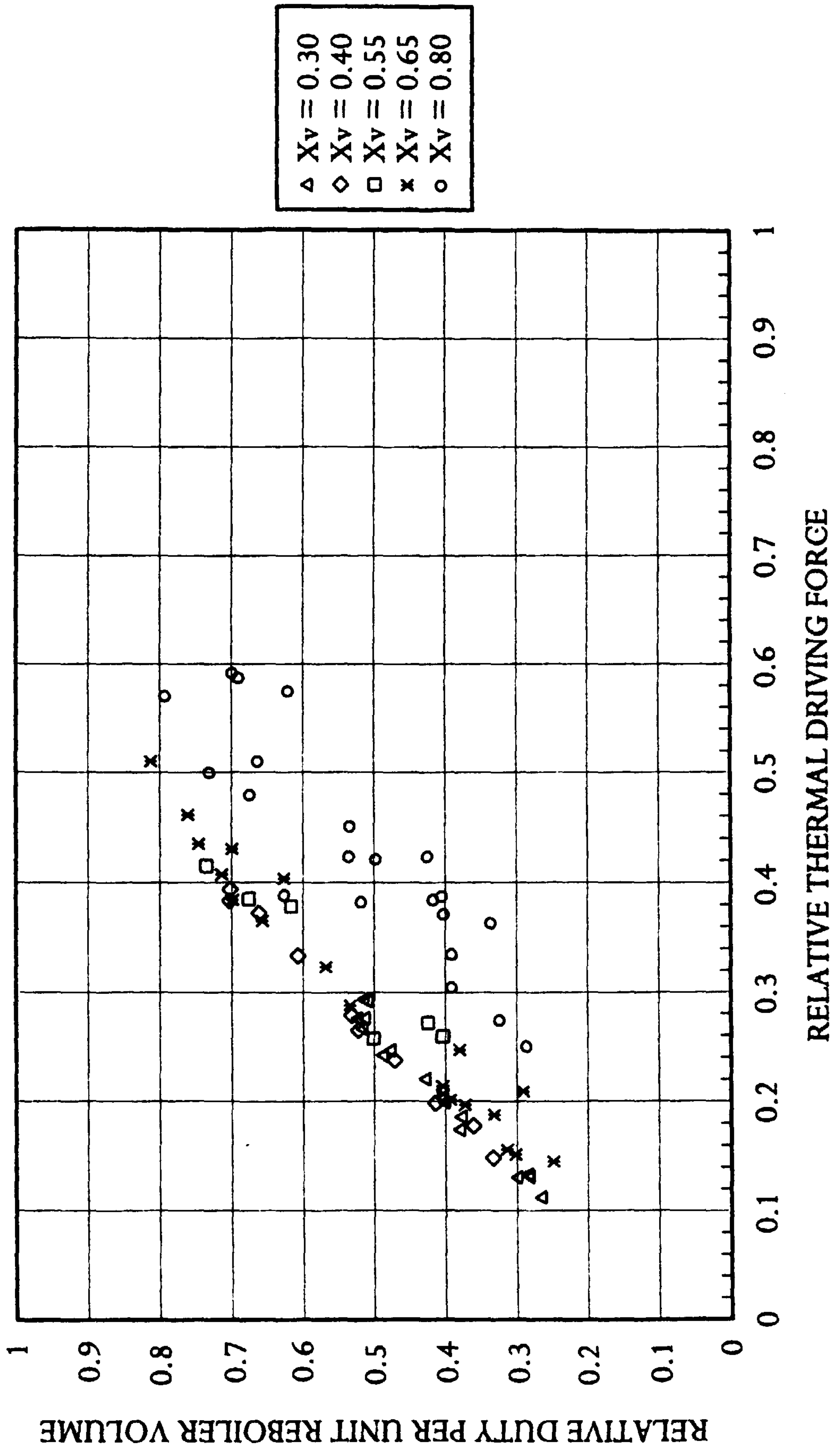

◇ Xv = 0.30
□ Xv = 0.50
△ Xv = 0.65
✖ Xv = 0.80
○ Xv = 0.90
RELATIVE DUTY PER UNIT REBOILER VOLUME
RELATIVE THERMAL DRIVING FORCE
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1

FIG. 11



Xv = 0.30
Xv = 0.40
Xv = 0.55
Xv = 0.65
Xv = 0.80
RELATIVE DUTY PER UNIT REBOILER VOLUME
RELATIVE THERMAL DRIVING FORCE
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1

FIG. 12

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 99 30 2930

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X	US 4 715 433 A (SCHWARZ ET AL) 29 December 1987 (1987-12-29) * column 3, line 30 - column 4, line 5; figures 1-3 *	1-5,8,9	F28D9/00 F28F3/02 F25J3/00
Y		10,13-17	
Y	US 5 122 174 A (SUNDER ET AL) 16 June 1992 (1992-06-16) * column 4, line 47 - column 5, line 60; figure 1 *	10	
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Place of search	Date of completion of the search	Examiner
THE HAGUE	26 August 1999	Beltzung, F

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