Title: New plates for different types of plate heat exchangers

Short running title: Plate heat exchangers

by

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Abstract

The first patent for a plate heat exchanger was granted in 1878 to Albretch Dracke, a German inventor. The commercial embodiment of these equipments has become available in 1923. However, the plate heat exchanger development race began in the 1930's and these gasketed plate and frame heat exchangers were mainly used as pasteurizers (e.g. for milk and beer). Industrial plate heat exchangers were introduced in the 1950's and initially they were converted dairy models. Brazed plate heat exchangers were developed in the late 1970's. However, copper brazed units did not start selling until the early 80's. Nickel brazing came to market around ten years later, since copper presents compatibility problems with some streams (e.g. ammonia). All-welded and semi-welded (laser weld) plate heat exchangers were developed during the 1980's and early 90's. Shell and plate heat exchangers were recently introduced in the market and can withstand relatively high pressures and temperatures, as the shell and tube does. The fusion bonded plate heat exchangers (100% stainless steel) are a technology from the 21st century, these equipments being more durable than brazed plate heat exchangers. The plates are the most important elements from the different plate heat exchangers mentioned above. This paper initially introduces the gasketed plate and frame heat exchanger and common chevron-type plates. Resorting to computer fluid dynamics techniques, the complex 3D flow in cross-corrugated chevron-type plate heat exchanger passages is visualized. Recent patents related with the plates from different plate heat exchangers are then outlined.

Keywords: Gasketed plate heat exchangers, brazed plate heat exchangers, all-welded plate heat exchangers, semi-welded plate heat exchangers, double-wall plate heat exchangers, shell and plate heat exchangers, fusion plate heat exchangers, computer fluid dynamics.

1. INTRODUCTION

Around 1850, French wine producers invited Louis Pasteur to solve a problem related with wine deterioration. Comparing samples of good wine with samples of deteriorated wine, Pasteur found several strains of micro-organisms. Some of them were predominant in the wines with good quality, while other strains were abundant in the wines with low quality. Hence, Pasteur concluded that a proper selection of micro-organisms could guarantee a consistent high quality production. In order to do that, he destroyed the micro-organisms present in the new wine of the grapes by heating it between 50º and 60 ºC and re-inoculated the new wine with wine of high quality, the latter wine containing the desirable micro-organisms. These experiments were useful to understand the role of micro-organisms in fermentations and showed that it was possible - using temperature – to control infestations provoked by harmful micro-organisms. Therefore, pasteurization was invented, this operation being widely used nowadays in the dairy and food industries. Later on, Pasteur broaden his studies to beer and, in 1857, he published one work showing that milk sours due to the presence of some micro-organisms [1].

The bacteria *Mycobacterium tuberculosis* can be present in raw milk and catalysed (in the late years of the 19^{th} century) the development of plate heat exchangers, since they are very efficient and can be easily disassembled for cleaning and sterilization to meet health and sanitation requirements [2, 3]. The first patent for a plate heat exchanger was granted, in 1878, to Albretch Dracke, a German inventor, but the commercial embodiment of these equipments has become available from APV International, England, in 1923. Around 1930, the company Alfa Laval, Sweden, launched an analogous commercial plate heat exchanger [3].

In 1996 the total market for heat exchangers in Europe amounted to USD 3.6 billion and the plate heat exchanger had a market share of 13 % (second position after the conventional shell-and-tube heat exchanger) [4]. Modern plate heat exchangers provide higher working temperatures, larger working pressures, higher resistance to chemicals, etc.. Due to this, different types of plate heat exchangers are nowadays applied in a very broad range of industrial heat exchanger needs [2-6].

2. GASKETED PLATE AND FRAME HEAT EXCHANGERS

Gasketed plate and frame heat exchangers (Fig.(**1**)) are widely used in chemical, pharmaceutical, food, dairy, pulp and paper industries, as well as in air conditioning and refrigeration systems (HVAC&R) and offshore gas and oil applications [2-10], to name a few, due to the low space requirement, low fouling tendency (high shearing forces), high efficiency, easy disassembly of the heat exchanger for cleaning and sterilization and flexibility on account of modular design, i.e., the heat transfer area can be easily varied by removing or introducing plates and the plate pattern can also be easily changed [4].

The heat transfer surface from a gasketed plate and frame heat exchanger is constituted by a series of plates (see Figs. (**1**) and (**2a**)) containing portholes, for fluid entry and exit, in the four corners. When the plates are pressed together, the portholes form continues tunnels, leading two fluids (for instance) from the inlet into the plate package, the different fluids being distributed to alternate narrow passages, always in counter-current flow. The presence of gaskets prevents the leakage of fluid from the passages to the surrounding atmosphere [2].

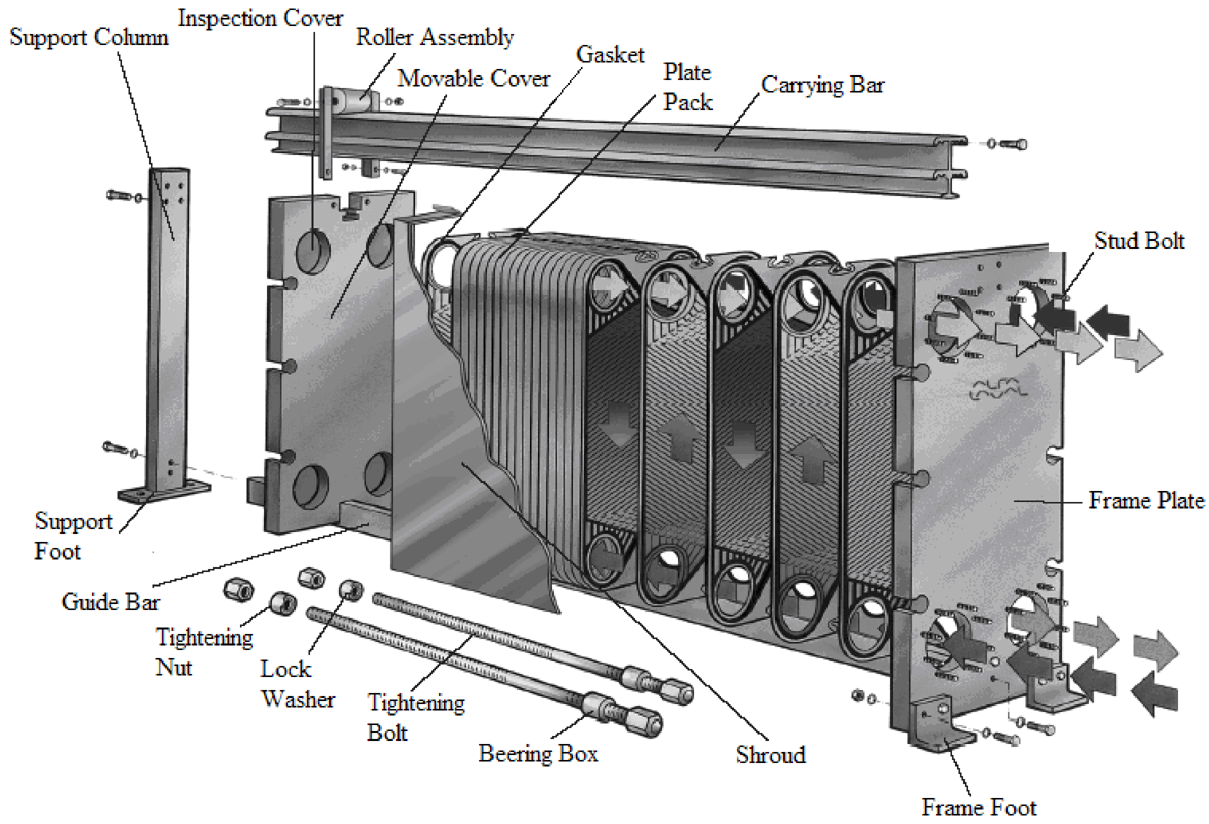


Fig. (1). Gasketed plate and frame heat exchanger (courtesy from Alfa Laval).

The most used gasketed plate and frame heat exchangers consist of plates with chevron-type corrugations [11] that have a sinusoidal shape [7-10] (Figs. (**1**) and (**2**)). The thermal-hydraulic performance of plate heat exchangers is strongly dependent on the geometrical properties of the chevron plates [2, 5, 12], namely on the corrugation angle, β, area enlargement factor, ϕ, defined as the ratio between the effective plate area and projected plate area, and channel aspect ratio (Fig. (**2**)).

The channel aspect ratio, γ, can be defined by [9]:

$$\gamma = \frac{2b}{p_x}, \qquad (1)$$

p_x being the corrugation pitch in the main flow direction and b the inter-plates distance (Fig. (**2**)).

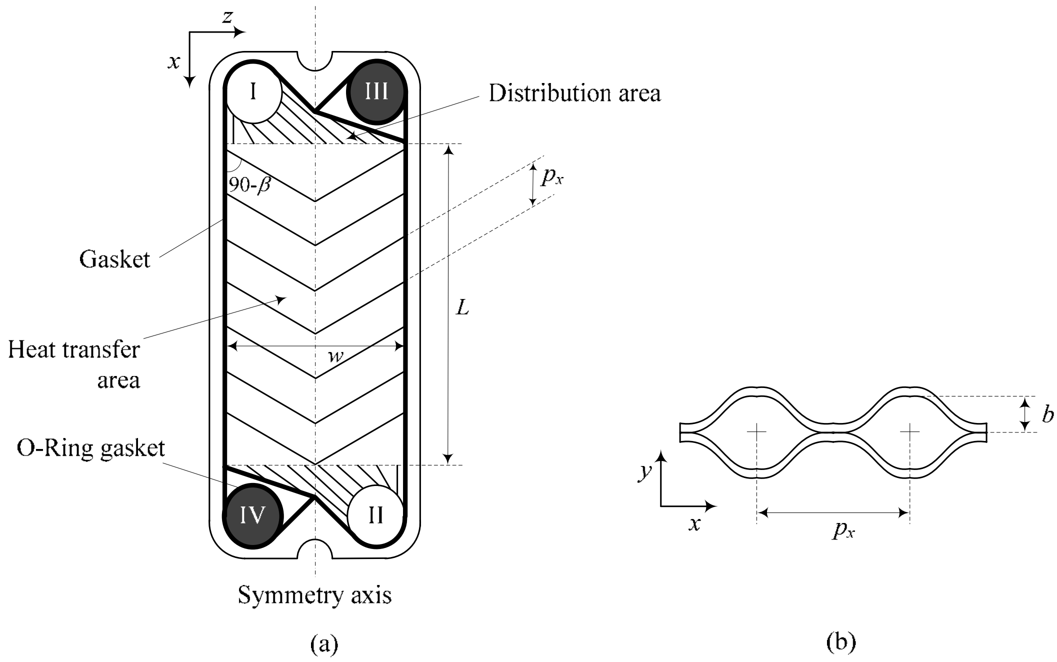


Fig. (2). (a) Schematic representation of a chevron plate. I, II, III and IV: portholes; (b) corrugation dimensions.

The area enlargement factor can be estimated by [9]:

$$\phi = \frac{1}{6}\left\{1 + \left[1 + \left(\frac{\pi}{2\cos(\beta)}\right)^2 \gamma^2\right]^{0.5} + 4\left[1 + \left(\frac{\pi}{2\sqrt{2}\cos(\beta)}\right)^2 \gamma^2\right]^{0.5}\right\}. \quad (2)$$

Typically, the area enlargement factor assumes values between 1.1 and 1.5, b normally lies in the range 2-5 mm and β is typically located in the range 22-65º [2-10, 12]. The length of the plates may vary between 0.3 and 4.3 m [2, 3], the minimum value of the ratio length/width being of the order of 1.8 [2].

The thermal-hydraulic performance of the channels formed by cross-corrugated chevron-type plates can be studied making use of computational fluid dynamics techniques [7-10]. Due to the periodicity of the flow along the width of the channel (*zz* axis in Fig (**2**)) the referred study can be carried out using unitary cells (Fig. (**3**)) [9, 10]. It is important to note that in order to obtain thermal and hydraulic fully developed flows in the main flow direction (*xx* axis), consecutive unitary cells must be used [9, 10].

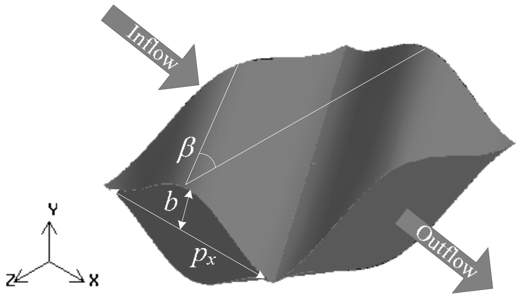
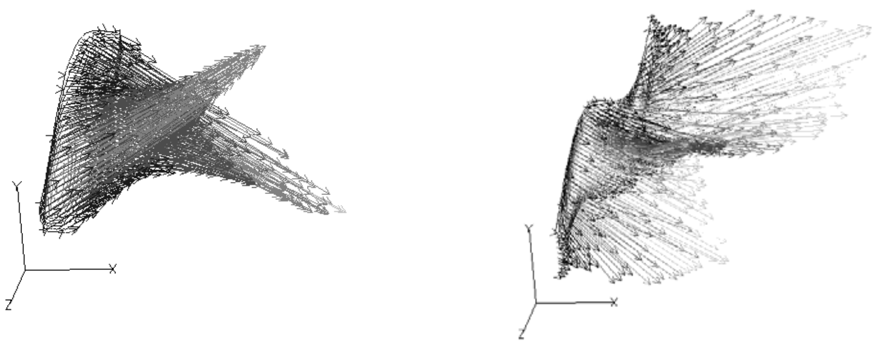
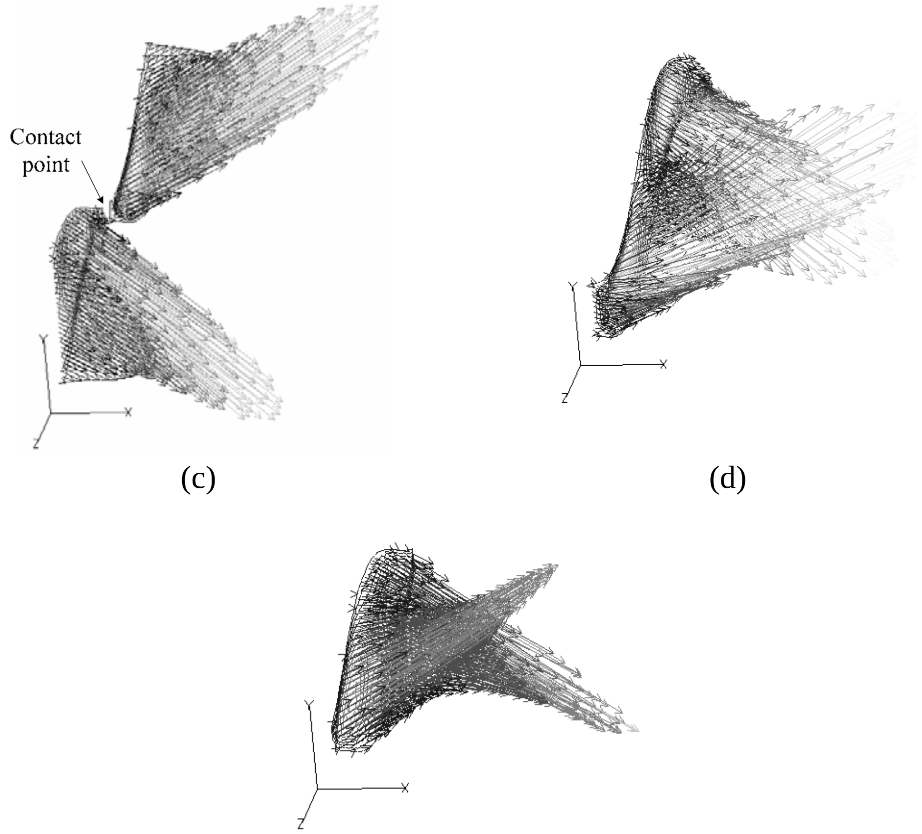


Fig. (3). Unitary cell from a channel containing chevron-type plates. The corrugation angle is 31º, $\gamma = 0.453$ and $\phi = 1.17$.

The complex laminar flow inside a cross-corrugated chevron-type plate heat exchanger channel can be observed in Fig. (**4**). In Fig. (**4**) the velocity field for different values of the normalized length *x** is presented. The inlet and outlet of the unitary cell (Fig. (**3**)) is represented by *x** = 0 and 1 (Fig. (**4**)), respectively. The plan *x** = 0.5 (Fig. (**4c**)) includes a contact point between the plates (located in the centre of the unitary cell). The maximum of the average interstitial velocity is observed for *x** = 0.25 and 0.75 (Figs. (**4b**) and (**4d**)) [10].



(a) (b)



(c) (d)

(e)

Fig 4. Velocity vectors in different planes. (a) $x^* = 0$; (b) $x^* = 0.25$; (c) $x^* = 0.5$; (d) $x^* = 0.75$; (e) $x^* = 1$.

As happens in granular beds [13-16], the flow in chevron-type passages is highly tortuous (see Fig. (**4**)). Fernandes et al. [9] estimated the tortuosity coefficient in chevron-type plate heat exchanger passages, this coefficient being used by plate heat exchanger producers (CIAT) [17] in order to determine friction factors and convective heat transfer coefficients. The model from CIAT [17] was developed resorting to an analogy [18] developed for fixed beds. The similitude between the flow and heat transfer in chevron-type plate heat exchangers passages and granular beds was also emphasized by Edwards et al. [19].

3. NEW PLATES FOR PLATE HEAT EXCHANGERS

In order to obtain a higher number of transfer units (NTU-VALUE) than that from conventional chevron-type plates (Figs. (**2**) and (**3**)), Bojesen [20] disclosed the inclusion of a series of dents/indents, recesses/protrusions and/or additional corrugations in the surface of conventional chevron plates (Fig. (**5**)). In Fig. (**5**) it is shown the location of protrusions (16) and recesses (17) and contact points (15), these points being generated by the conventional corrugations ((12) and (13)) when two plates are pressed together to form a channel. The projections (16) and recessions (17) are located between the contact points (15).

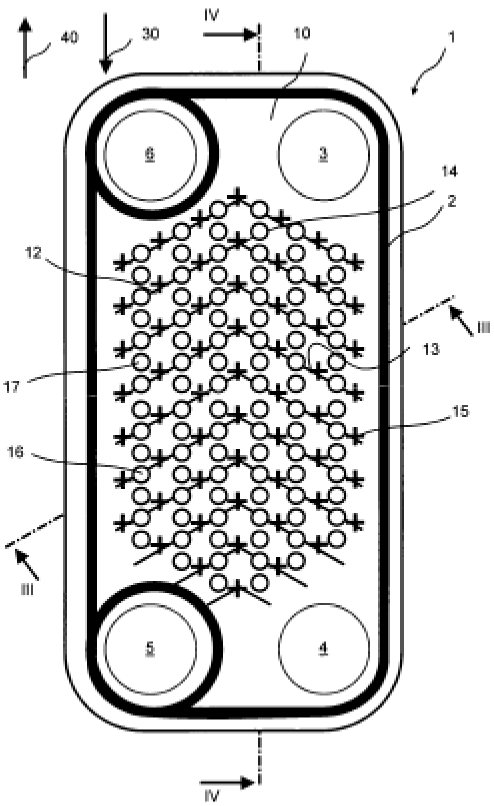


Fig 5. Schematic top view of a heat exchanger plate (for the complete description of the numbers see [20]).

The difference between conventional chevron-type plate heat exchangers channels and the channels obtained with the disclosed plates may be better seen in Fig. (**6**), these figure showing a schematic sectional view in the direction of arrow 30 (main flow direction), along line IV-IV (Fig. (**5**)). Observing Fig. (**6c**) it can be induced that the fluid passing through the disclosed passage (Fig. (**6a**)) has to change its flow direction much more (higher tortuosity coefficient [9]) than by flowing through a conventional passage (Fig. (**6b**)). This happens due to the presence, in the disclosed passage, of protrusions (26´) and recesses (27´). Due to this, more turbulences are generated and the NTU-VALUE increases. In the patent [20] it is referred that by providing modifications as shown in Fig. (**6**) the NTU-VALUE (a measure of the heat transfer surface area requirements for a given heat duty or size of the heat exchanger [3]) can be surprisingly increased (more than 5%).

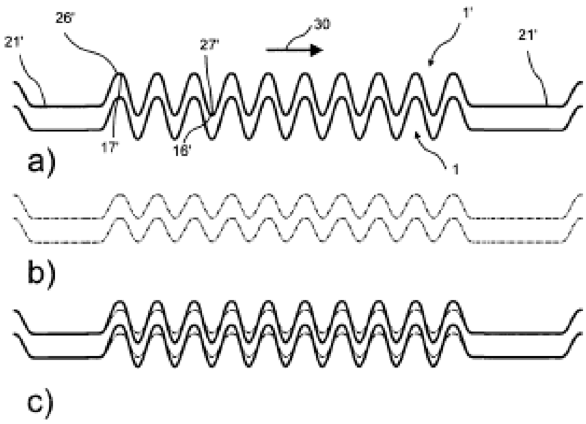


Fig 6. Schematic sectional view along line IV-IV from Fig. (**5**). (a) embodiment of the invention; (b) conventional chevron-type passage; (c) difference between the embodiment of the invention and the conventional chevron-type passage (for the complete description of the numbers see [20]).

In order to promote the production of a turbulent flow through all, or the major part, of a passage defined by two heat exchanger plates, Rausing [21] disclosed the use (see Fig. (**7**)) of turbulent-promoting protrusions (spherical, ellipsoid, waves or grooves), these protrusions (4) containing a surface profile (6) that also promotes turbulence. The surface profile consists of spherical or ellipsoid segments, concavely or convexly arranged relative to the protrusions. When the protrusions are hemispherical and the surface profile is concavely arranged, this complex may be compared to the surface of a golf ball [21].

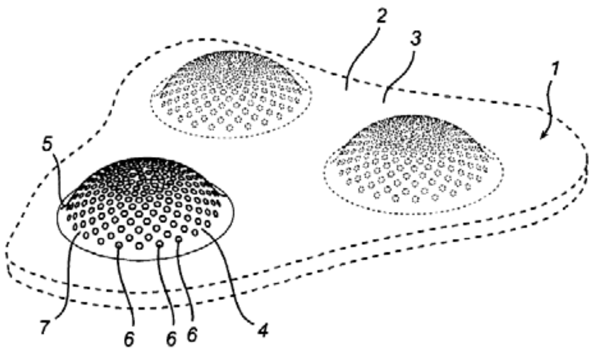


Fig 7. Schematic protuberances (4) and turbulent-promoting surface profile (6) (for the complete description of the numbers see [21])

The fact that the surface profile consists of spherical or ellipsoid segments contributes to the heat exchanger plate not having any sharp edges or corners that can create dead spaces which conventional cleaning methods cannot reach [21]. The author emphasizes that soft geometric transition is most important from the aspect of hygiene since plate heat exchangers when used in the food industry require frequent and very careful cleaning. Any sharp geometric transitions can form growth zones for bacteria and other organisms Due to this, the author also recommends the use of a smooth geometrical transition between the flat part of the plate ((1) in Fig. (**7**)) and the protrusions (4). Besides promoting turbulence, the presence of a surface profile generates higher plates surface area [21].

In distilleries, sugar mills, paper industry, textile industry, food industry, pharmaceutical industry, etc., the fluids processed in plate heat exchangers can be very viscous and contain particles, fibres or other difficult components [22]. Due to this, Gustafsson [22] developed a porthole (see Figs. (**2**) and (**8**)) that mitigates the attachment of particles or fibres to the porthole edge ((19) in Fig. (**8**)). Close to the porthole (18), in the area between the gasket groove (23) and the porthole edge there is a corrugation which forms a wall towards the gasket groove which ensures that the gasket is held in place in the gasket groove. It is important to note that the referred corrugations are, normally, also present in the portholes without O-Ring gasket (porthole I and II in Fig. (**2**)) and that conventional portholes contain the referred corrugation in the entire perimeter. In order to reduce the risk that particles or fibres get attached to the porthole edge, Gustafsson [22] designed a porthole which is characterized in that the first edge portion ((19') in Fig. (**8**)) has a corrugation and that the second edge portion (19'') has a substantially flat shape.

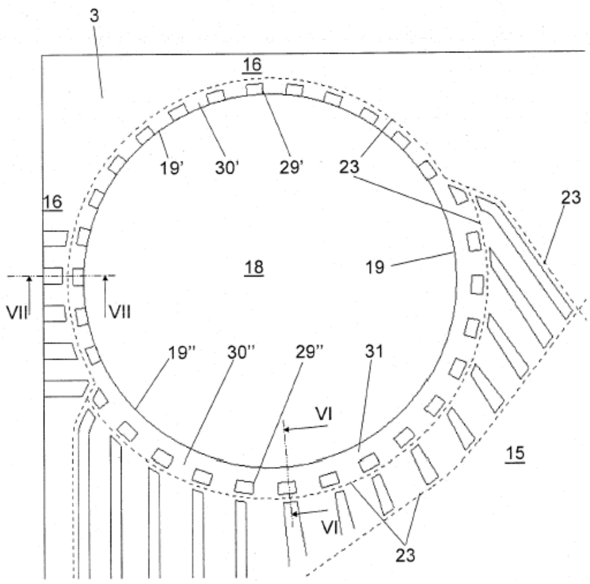


Fig 8. Schematic representation of a porthole (for the complete description of the numbers see [22]).

After the introduction of a heat exchange fluid from the inlet porthole (porthole I in Fig. (**2a**), for instance), the distribution area promotes the expansion of the flow to the entire width of the heat transfer area [23]. When the fluid is effectively spread to every corner of the plate, the heat exchange performance is improved [23] and the plate is less prone to heat transfer fouling [24] or particulate fouling [25], due to the absence of low velocity regions.

The distribution area contains a small amount of contact points between the two plates from a plate heat exchanger passage. On the transition region between the distribution area and heat transfer area, contact points are scarcer. In the heat transfer area the number of contact points (or support points) increase with the increase of the corrugation angle (see Fig. (**2a**)), the number of contact points from a chevron-type plate with β = 65º being approximately the double than that from a plate with β = 25º [26]. Due to this, a passage with low corrugation angle can be mechanically unstable, i.e., the inter-plates distance can vary during the operation. In order to improve the strength of a passage containing chevron plates with low corrugation angle (25º for instance) Blomgren and Krantz [26] disclosed a passage (see Fig. (**9**)) containing two corrugated transition areas (58), located between the distribution areas (34) and heat transfer area (33). In this invention, the transition areas - in Fig. (**9**) is only shown one of two the transition areas - contain corrugations with a high value of β (65º for instance) in order to obtain a high number of contact points (64) between the plates and, therefore, improved strength.

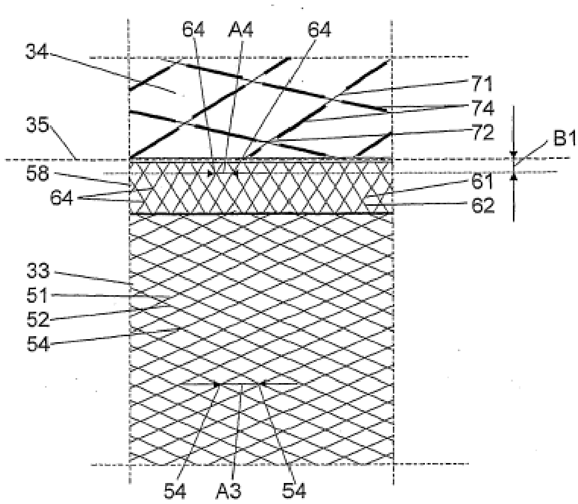


Fig 9. Schematic top view of two adjacent plates (for the complete description of the numbers see [26]).

The above mentioned inventions from Gustafsson [22] and Blomgren and Krantz [26] can be applied to gasketed plate and frame heat exchangers (Fig. (**1**)) as well as to brazed plate heat exchangers [27, 28] and welded plate heat exchangers [29].

Brazed plate heat exchangers were developed in the late 1970's. However copper brazed units did not start selling until the early 80's. Nickel brazing came to market around ten years later, since copper presents compatibility problems with some streams (e.g. ammonia). Copper or nickel foils are placed in between each of the stainless steel plates and a pile of plates is placed in a furnace just above the melting temperature of the foils. Capillary forces draw the copper to the contact points between the plates, connecting the plates at the edges and at a large number of contact points across the heat transfer area. Because the plates are brazed to each other and there are no gaskets (temperature sensitive elastomeric materials), brazed plate heat exchangers allowed the increase of maximum operating temperatures and pressures. Typical maximum operating temperatures and pressures from gasketed plate and frame plate heat exchangers (160 ºC and 25 bar) are lower than that from brazed plate heat exchangers (225 ºC and 30 bar) [3, 6]. Due to the absence of frames, brazed plate heat exchangers are also characterized by very low weight [3].

The all-welded and semi-welded plate heat exchangers were developed during the 1980’s and early 90’s. All-welded plate heat exchangers use plates similar to those in gasketed plate and frame heat exchangers, the laser welds being applied along the edges, in the plane of the plates [3, 4]. Since the plate edges are all sealed by welded joints, this allows the increase of typical maximum operating temperatures and pressures (350 ºC and 40 bar). Brazed and all-welded plate heat exchangers has a same drawback (when compared with the gasketed versions) since they cannot be dismantled for mechanical cleaning and, therefore, their use is restricted to applications where fouling does not occur [4, 5].

In the semi-welded plate heat exchangers it is built a sealed (laser beam) passage (a plate cassette [5] containing two welded plates) for the aggressive media while the service medium flows outside the welded plate cassette, in a conventional gasketed plate heat exchanger passage. In the plate cassettes the ports have to be sealed with O-Ring gaskets, but despite of this, there is a reduction of 90% of the gasket length necessary for the aggressive media [4, 5]. The semi-welded plate heat exchanger is therefore partially dismountable, the maximum operating temperatures and pressures being at the same level of the conventional gasketed plate and frame heat exchangers [3-5].

The double-wall plate heat exchanger contains double plates, sealed by conventional gaskets, which replace the single plate that usually separates two streams. They are many times designed for a reacting media and if this aggressive media reacts with the surface of the double-wall plates, the leakage is directed in the passages between the double plates [3].

In the invention from Rehberg [30] it is disclosed a plate heat exchanger comprising a stack of double-walled heat exchanger plates with which the search for leaks is facilitated (see Fig. (**10**)). The two plate members ((33) and (34)) are in close mutual engagement in the heat transfer area (35) and this is continued in an edge portion (36) which is upwardly bent with the respect of the heat transfer area (35). Neighbouring double-plates ((31) an (32), for instance) are connected by soldering material (39) in the area of their upwardly bent portions (36). Since there is a spacing (37) between the outer edges ((33a) and (34a)) of a double-plate, any fluid getting between the two plate members ((33) and (34)) will become visible (region (38)) from outside and, therefore, be seen by monitoring or operating staff [30].

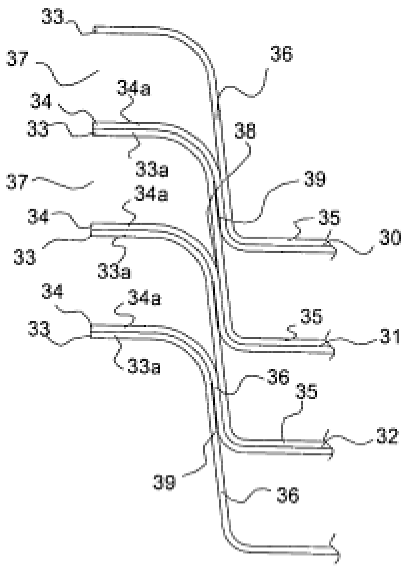


Fig 10. Schematic sectional view of an edge portion of a plate heat exchanger (for the complete description of the numbers see [30]).

4. CURRENT & FUTURE DEVELOPMENTS

Shell and plate heat exchangers [5, 31-33] were recently introduced in the market, being composed by round welded plates inserted in a shell [5, 31-33]. These equipments combine the advantages of shell and tube and plate and frame technologies, i.e., the high mechanical integrity of shell and tube (up to and beyond 400 bar and 800 ºC [34]) and the superior thermal performances of plate and frame [5, 31]. Kontu and Virtanen [32] disclosed several new uses for shell and plate heat exchangers. In order to achieve that, the referred authors suggested the use of plates, shell and inlet/outlet passages made of carbon steel, a cheap and abundant material.

The already rich selection guides [3] of plate heat exchangers will continue to be enriched with products such as the 100% stainless steel fusion bonded plate heat exchangers, these equipments being much more durable compared to brazed plate heat exchangers.

In years to come, with increased choice of better heat exchange equipment, the impetus of adopting equipment that a different industry already uses will decrease [34].

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