CSIR REPORT CENG 445

PLATE HEAT EXCHANGERS

Review of transport phenomena and design procedures

W W FOCKE

CHEMICAL ENGINEERING RESEARCH GROUP – CSIR
COUNCIL for SCIENTIFIC and INDUSTRIAL RESEARCH
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SYNOPSIS

The existing thermal and hydrodynamic design information is predominantly proprietory and empirical. The understanding of the transfer mechanisms involved is sketchy and based largely on conjecture rather than on experimental proof.

The Chilton-Colburn analogy for momentum and heat transfer is not valid for plate heat exchangers. However, for the turbulent regime a correlation exists between heat transfer and energy dissipation. Turbulence models based on an energy dissipation analogy may, therefore, provide a good basis for predicting the flow and heat transfer in plate heat exchangers.

Flow distribution to plates is a critical aspect of plate heat exchanger performance, particularly where cooling of highly viscous and non-Newtonian fluids are involved and give rise to laminar flow. For this flow regime very little information is available.

Design flexibility is obtained by using chevron plates of different corrugation angles in one exchanger. The existence of optimal angles has been identified.

The development of a basic and fundamental understanding of the transfer mechanisms in plate heat exchangers is recommended. Such a scientific basis may provide the key to substantial performance improvements in future design.

KEYWORDS: Plate heat exchanger, review, design, friction factor, Nusselt number, analogy.

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PLAATHITTERUILERS
Oorsig van oordragsverskynsels en ontwerpsprosedures

W W FOCKE Februarie 1983

SINOPSIS

Die bestaande termiese en hidrodinamiese ontwerpinligting is hoofsaaklik empiries en in die hande van verskaffers. Wat die betrokke oordragsmeganismes betref, is die kennis oppervlakkig en berus meer op gissings as op eksperimentele feite.

Die Chilton-Colburn-analogie tussen hitteoordrag en wrywing geld nie vir plaathitteruilers nie; daar bestaan wel 'n analogie tussen hitteoordrag en energiedissipasie in die turbulente oordragsbereik. Turbulensiemodelle gebaseer op 'n energiedissipasie-analogie behoort dus 'n goeie grondslag vir die voorspelling van vloei en hitteoordrag in plaathitteruilers te wees.

Die vloeiverdeling na onderskeie plate is 'n kritiese aspek van die werkverrigting van plaathitteruilers, veral wanneer hoogs viskeuse en nie-Newtoniese vloeiers afgekoel word en aanleiding tot laminêre vloei gee. Vir laasgenoemde vloeiregime is min inligting beskikbaar.

In die ontwerp van plaathitteruilers word soepelheid verkry deur riffelplate van verskillende riffelhoeke in 'n plaatpak te gebruik. Die bestaan van optimale riffelhoeke is geïdentifiseer.

Die ontwikkeling van 'n fundamentele insig ten opsigte van die oordragsmeganismes in plaathitteruilers word aanbeveel. So 'n wetenskaplike grondslag het die potensiaal om beduidende verbeterings in die werkverrigting van toekomstige ontwerpe te verseker.

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1. INTRODUCTION

The first commercially successful plate heat exchanger (PHE) was developed in the late 1920's for milk pasteurization[1]. The idea was not new, however, since German patents relating to plate heat exchanger improvement date from the late 1870's[2,3]. Although with time numerous small improvements were added, the basic PHE concept has changed very little over the past century. It is still the exchanger which is almost exclusively used for hygienic duties, but the majority of today's PHE's are destined for use in chemical processing and other industrial applications[4,5]

The commercial PHE consists basically of a number of thin metal sheets onto which some corrugated pattern has been stamped or embossed. The flow channels are formed by clamping these plates together in a frame. The latter includes a fixed end plate with connecting ports and a removable cover plate with some pressing arrangement (see Figure 1). Intermediate connector plates, which allow multiple process streams to be handled in a single unit, are also available[6].

Each plate has four corner ports which in pairs service the two channels formed on either side of the particular plate. The two process streams thus pass through alternate channels (see Figure 2). Plate thickness usually varies between 0,5 and 3 mm and the channel gap[6] is between 2 and 5 mm. Plate surface areas range from 0,3 to 2,5 m^2. The largest units[6] are capable of handling flow rates of up to 2 500 m^3/h. Units capable of withstanding pressures of up to 2,5 MPa are available.

All metals which can be cold-worked are suitable for PHE application[7]. Stainless steel is the most common material of construction, but plates made from titanium, nickel, Hastelloy, Incolloy, Monel, phosphor bronze and cupro-nickel are also available[1,2,6,7].

Sealing is accomplished by gaskets and special interlocking types are used at high differential pressures to prevent gasket blow-out[1]. The possibility of fluid intermixing (in the rare event of gasket failure) is prevented by using a double seal between the fluid streams. The interspace between the seals is also vented to the atmosphere to facilitate a visual indication of leakage[1] (see Figure 3).

Elastomeric gaskets are required limiting PHE application to relatively low process temperatures (see Table 1). The compressed asbestos fibre gaskets used at elevated temperatures contain about 6% rubber, which accounts for their temperature limit of 250 °C. Unfortunately PTFE (Teflon) cannot be used owing to its visco-elastic behaviour[7].

Compared to other exchangers the PHE has the following advantages[1,3,5,9]:

- high accessibility for inspection and cleaning
- low fouling tendencies and ease of cleaning
- compactness
- flexibility and extendability of sizes

- short retention time owing to low fluid hold-up
- economy, especially when expensive corrosion-resistant materials are required
- high thermal performance. The high degree of counter-current flow prevailing in PHE's makes temperature approaches of up to 1 °C possible. The high thermal effectiveness facilitates low grade heat recovery and high recuperation efficiencies (up to 90%)[6].

Plate heat exchangers are most suitable for liquid-liquid heat transfer duties requiring uniform and rapid heating or cooling as is often the case when treating thermally sensitive fluids. Special plates capable of handling two-phase fluids (eg, steam condensation) are available. PHE's are not suitable for erosive duties[10] or for fluids containing fibrous materials. In certain cases suspensions can be handled but to avoid clogging the largest suspended particle should be at least three times smaller than the average channel gap. Viscous fluids can be handled but extremely viscous fluids lead to flow distribution problems, especially on cooling[6].

FIGURE 1 Plate heat exchanger assembly

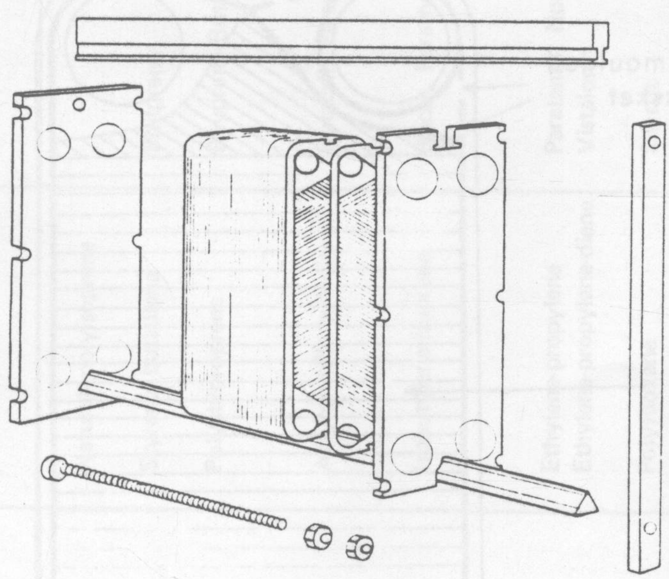


FIGURE 2 Flow pattern in a plate heat exchanger

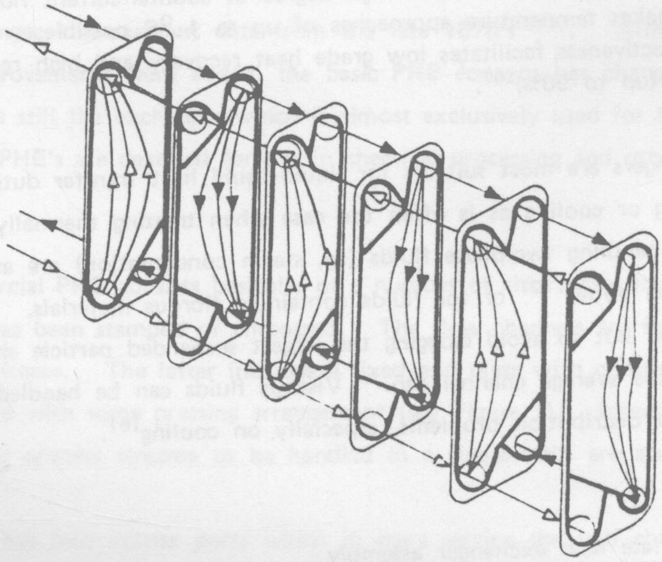


FIGURE 3 Sealing by double gaskets

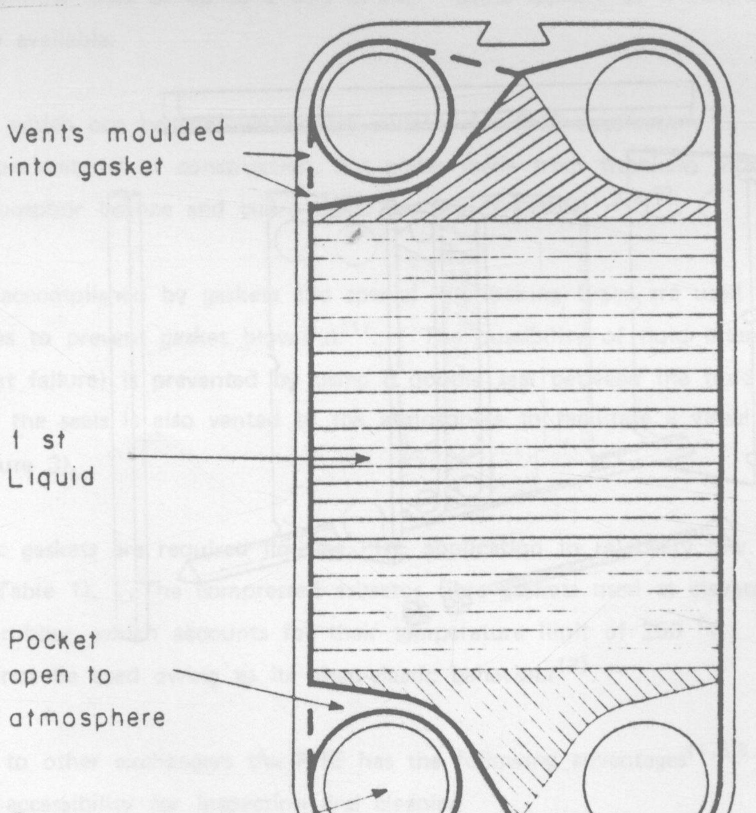


TABLE 1 Gasket materials[1,7,8]

Gasket material	Chemical name	Trade name	Maximum operating temperature (^{o}C)	Applications
Natural rubber	Natural polyisoprene		70	Oxygenated solvents, alcohols, acids
SBR	Styrene butadiene	Philprene	70	Alkalis, acids, and oxygenated solvents
Neoprene	Polychloroprene	Baypren, Butachlor	70	Alcohols, alkalis, acids, aliphatic hydrocarbon solvents
Nitrile	Acrylonitrile-butadiene	Butacril, Paracril	100 - 135	Oil and gasoline, animal and vegetable oils, alkalis, aliphatic organic solvents
Butyl (resin cured)	Isobutylene-isoprene	Bucar, Paratherm	120 - 150	Alkalis, acids, animal and vegetable oils, aldehydes, ketones, and some esters
EPDM	Ethylene-propylene Ethylene-propylene-diene	Paratemp, Nordel Vistalon	140	Alkalis, oxygenated solvents, animal and vegetable oils
Silicone	Polysiloxane	Silastic	140	Alcohols
Fluorocarbon	Fluorinated hydrocarbon	Viton, Paradur	175	Oil and gasoline, organic solvents, animal and vegetable oils
Compressed asbestos fibre		Paracaf	200 - 260	Organic solvents

2. GEOMETRY

A large number of proprietary corrugated patterns have been developed. Some of the more common plate patterns are depicted in Figure 4. The corrugations improve the rigidity of the plates and provide mechanical support to the system. They also lead to enhanced heat transfer rates, particularly by increasing the available surface area and also by inducing turbulence at low Reynolds numbers.

The two major configurations in general use are[1]:

- *Intermating troughs* (washboard design). The corrugations are pressed deeper than the plate spacing and the plates nestle into one another when the plate pack is assembled. Pips or dimples (500 to 750 per square metre) provide interplate contact and maintain the channel gaps.
- *Herringbone or chevron design.* These consist of troughed plates with corrugations inclined at an angle to the flow direction. Plates are assembled with the patterns pointing in opposite directions, thereby producing a three-dimensional flow passage of almost constant overall cross-sectional flow area. This design can have a high density of contact points, thus enhancing its capability to resist unbalanced pressures or pressure fluctuations.

Plate geometry determines, to a large extent, the thermo-hydraulic performance of the particular plate. The major parameters that affect performance (see Figure 5) are:

- trough pitch p
- trough height (or depth) H
- channel spacing h
- corrugation inclination to flow direction β
- base angle ϕ or corrugation shape.

At this stage it is still not possible to predict, from basic principles, the performance that will be achieved by a particular plate pattern although claims have been made[11] that computer-aided evaluation is possible.

For correlating performance of a family of geometrically similar plates in non-dimensional form, at least one characteristic length dimension of the particular pattern is required. The one that is most commonly used is the hydraulic diameter which, for ducts of constant (2-D) cross-sectional area, is defined as:

$$d_h = \frac{4 \times \text{cross-sectional area}}{\text{wetted perimeter}} \qquad (1)$$

In turbulent flow the heat transfer and pressure drop characteristics of all such ducts fall in a narrow band when the hydraulic diameter is used as the characteristic length dimension(12). For a rectangular duct of low aspect ratio:

$$d_h = \frac{2Wh}{W + h} \tag{2a}$$

$$\approx 2h \tag{2b}$$

For three-dimensional, variable area ducts an analogous definition for an effective hydraulic diameter (or equivalent diameter) could be(13):

$$d_e = \frac{4 \times \text{volume}}{\text{developed surface area}} \tag{3}$$

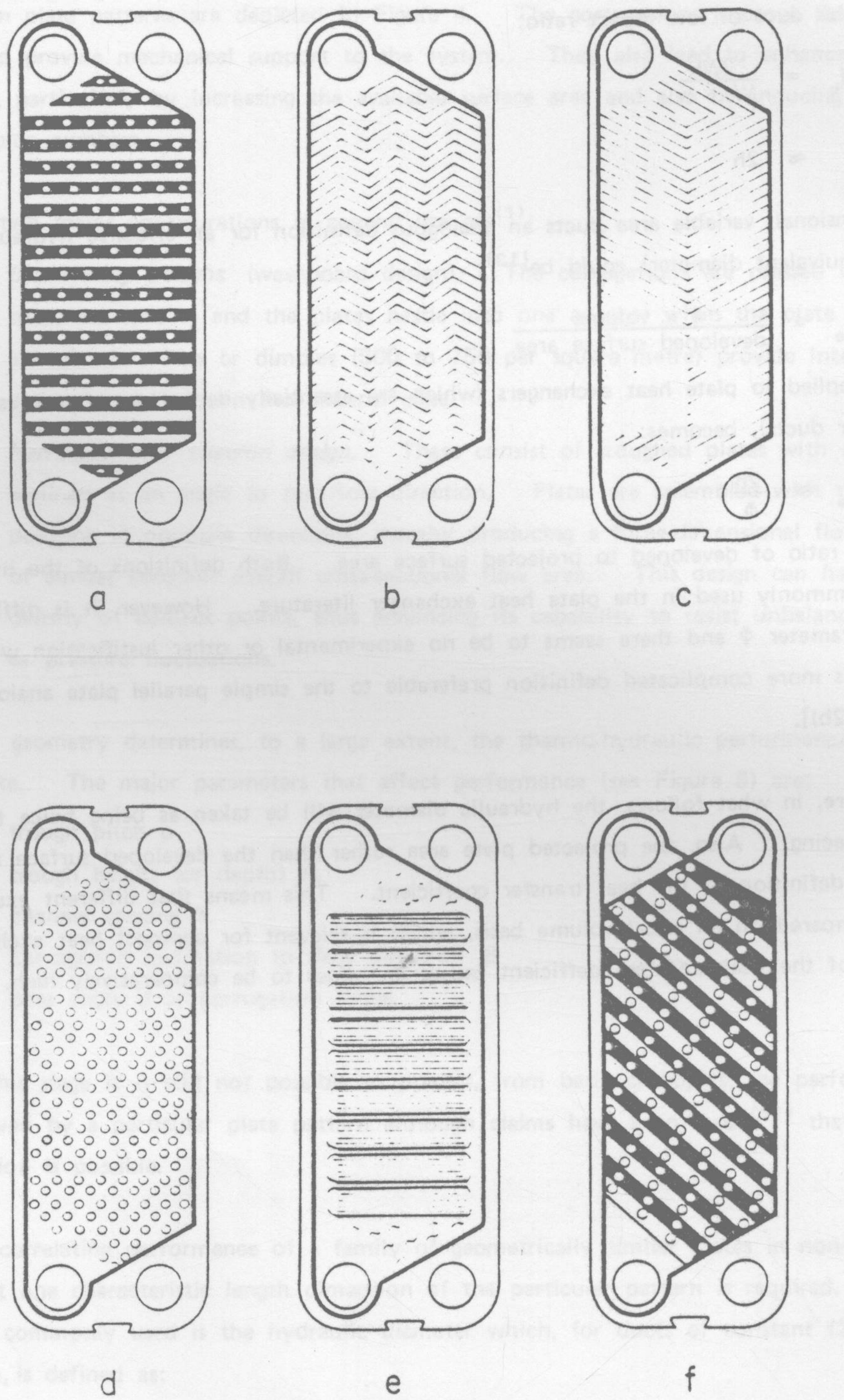
which, when applied to plate heat exchangers (which are essentially derivatives of low aspect ratio rectangular ducts), becomes:

$$d_e \approx \frac{2h}{\Phi} \tag{4}$$

where Φ is the ratio of developed to projected surface area. Both definitions of the hydraulic diameter are commonly used in the plate heat exchanger literature. However, it is difficult to measure the parameter Φ and there seems to be no experimental or other justification which would make this more complicated definition preferable to the simple parallel plate analogy(13) [see Equation (2b)].

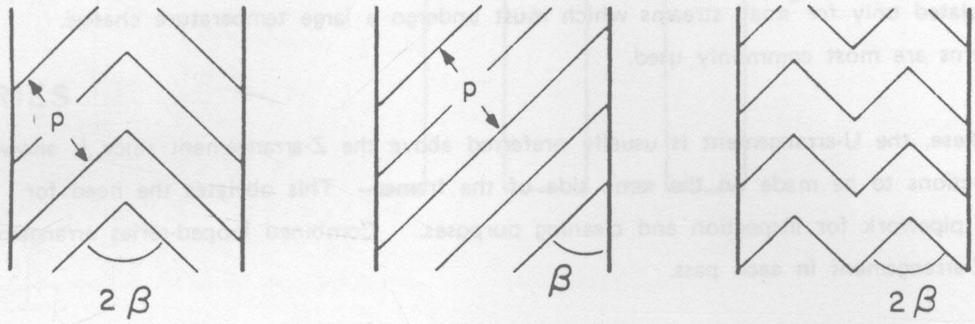
Therefore, in what follows, the hydraulic diameter will be taken as being twice the average plate spacing. Also, the projected plate area rather than the developed surface area will be used in the definition of the heat transfer coefficient. This means that different exchanger patterns are compared on an equal volume basis, which is relevant for compact heat exchangers. This definition of the heat transfer coefficient causes its values to be comparatively high.

FIGURE 4 Plate patterns

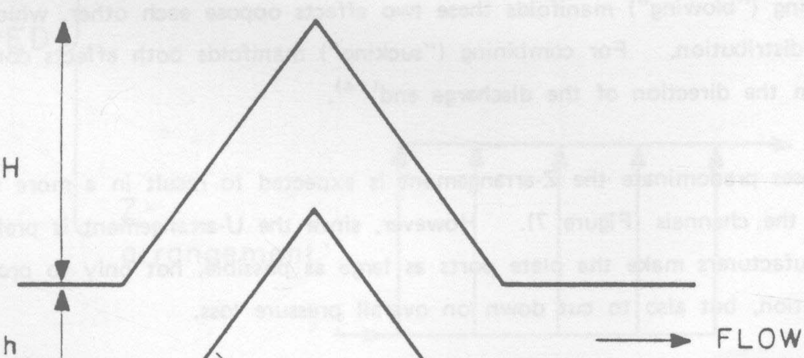


a : Washboard ; b : Zig-zag ; c : Chevron or Herringbone ; d : Protrusions and depressions ; e : Washboard with secondary corrugations ; f : Oblique washboard

FIGURE 5 Plate characteristic geometric parameters



OBLIQUE CORRUGATIONS



INTERMATING TROUGH

3. HYDRODYNAMICS

3.1 Flow patterns

Basic flow patterns inside PHE's are shown in Figure 6. The type of flow arrangement required depends on the available pressure drop, the minimum fluid velocity allowed, the flow rate ratio of the two process streams and the heat transfer duty involved[6]. Series flow is not very effective since it always entails a measure of co-current flow of the exchanging fluids. A large portion of the available pressure drop is wasted in series flow by the many flow reversals required. It is contemplated only for small streams which must undergo a large temperature change. Looped patterns are most commonly used.

Of these, the U-arrangement is usually preferred above the Z-arrangement since it allows all the connections to be made on the same side of the frame. This obviates the need for disconnecting pipework for inspection and cleaning purposes. Combined looped-series arrangements require the Z-arrangement in each pass.

3.2 Flow distribution to channels

Equal flow distribution to channels in the same pass is usually preferred. Actual flow distributions achieved may deviate significantly from this ideal and are determined primarily by the pressure profiles in the manifolds. These in turn depend on the fluid friction losses incurred in the manifold and the momentum changes which result from changes in the fluid velocity owing to branching. For dividing ("blowing") manifolds these two effects oppose each other, which tends to even out the pressure distribution. For combining ("sucking") manifolds both effects contribute towards a pressure drop in the direction of the discharge end[14].

Where friction losses predominate the Z-arrangement is expected to result in a more even flow distribution between the channels (Figure 7). However, since the U-arrangement is preferred for practical reasons, manufacturers make the plate ports as large as possible, not only to promote a more even flow distribution, but also to cut down on overall pressure loss.

3.3 Flow distribution in a channel

The two primary port arrangements in a plate give rise to two possible flow arrangements[1], viz, diagonal and vertical flow (see Figure 8). Diagonal flow requires two different plate geometries to make up a plate pack, whereas the vertical flow arrangement requires one plate type only. The two configurations required to make up a pack of the latter are simply obtained by rotating every other plate through 180 degrees[1].

The flow distribution in older designs was not very uniform[15] but more modern patterns exhibit a fairly uniform distribution apart from a channelling effect (see Figure 9) close to the gaskets[16,17]. This channelling edge effect is caused by the lack of profilation of the plates in the vicinity of the gasket groove. It may have a detrimental effect on the heat transfer and residence time distribution but fortunately its influence is usually small since only a small fraction of the fluid is involved. Rubber strips may be used to improve the flow distribution[15].

FIGURE 6 Basic flow patterns in PHE

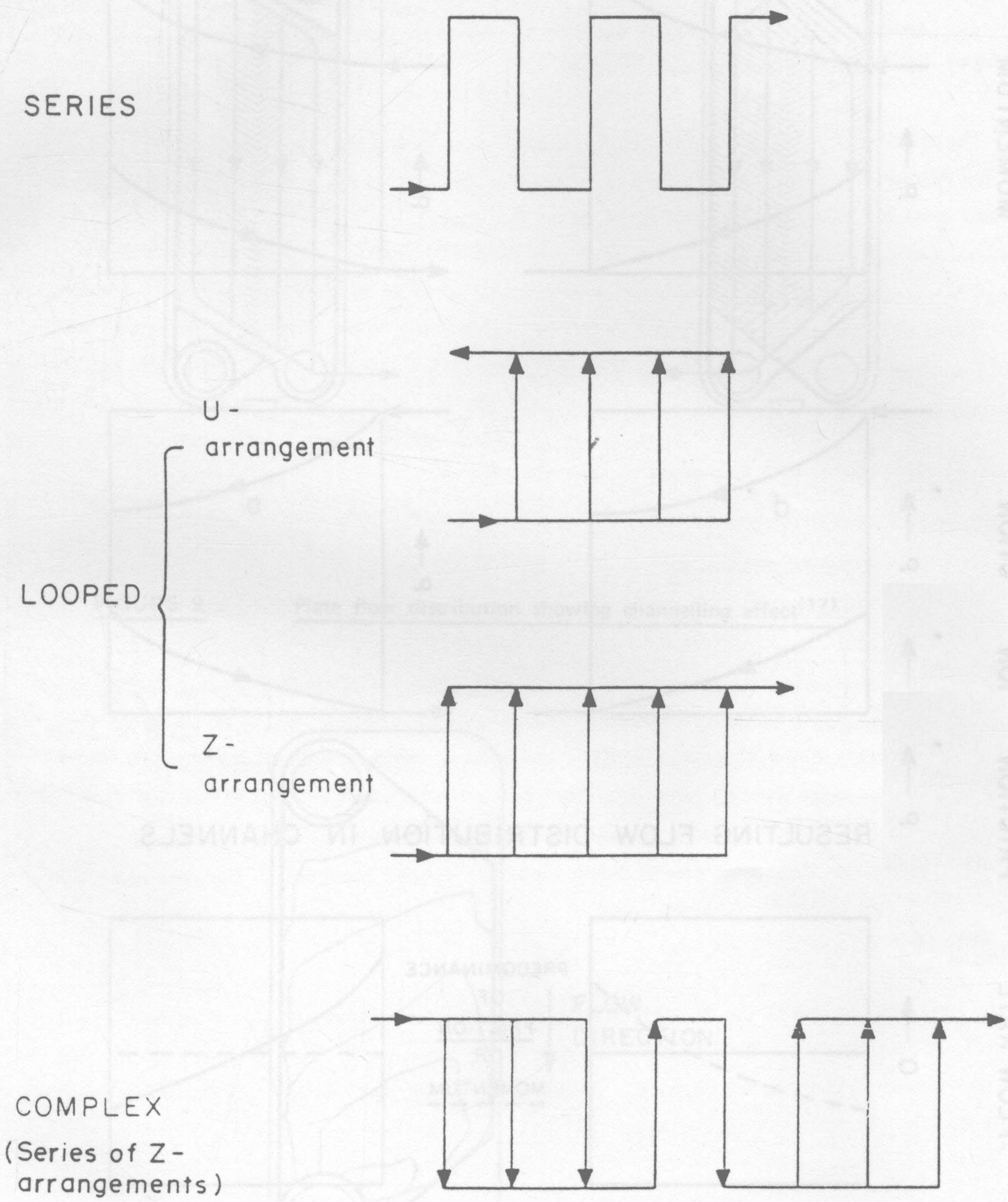


FIGURE 7 Flow distributions in looped flow arrangements