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Review

Counterflow heat exchanger with core and plenums at both ends

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ABSTRACT

This paper illustrates the morphing of flow architecture toward greater performance in a counterflow heat exchanger. The architecture consists of two plenums with a core of counterflow channels between them. Each stream enters one plenum and then flows in a channel that travels the core and crosses the second plenum. The volume of the heat exchanger is fixed while the volume fraction occupied by each plenum is variable. Performance is driven by two objectives, simultaneously: low flow resistance and low thermal resistance. The analytical and numerical results show that the overall flow resistance is the lowest when the core is absent, and each plenum occupies half of the available volume and is oriented in counterflow with the other plenum. In this configuration, the thermal resistance also reaches its lowest value. These conclusions hold for fully developed laminar flow and turbulent flow through the core. The curve for effectiveness vs number of heat transfer units (N_{tu}) is steeper (when $N_{tu} < 1$) than the classical curves for counterflow and crossflow.

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Contents

1. Introduction

Heat exchangers are a central topic in thermal science and engineering because of their essential role across the landscape of technology, from geothermal and fossil power generation to refrigeration, desalination, and air conditioning [1–4]. The literature on heat exchangers is voluminous and continues to be active today (e.g., Refs. [5–10]). The field covers two main aspects of this class of flow systems: fluid flow and heat transfer performance, and ways (criteria) of evaluating performance [11–15]. The general trend in the field is to develop heat exchangers that offer better performance. This trend is universal in evolution [16], and unites heat exchangers with other evolutionary flow systems, bio, non bio, and manmade.

The starting point for the present paper is the observation that all performance criteria change, and hopefully improve, when one changes and chooses a better flowing architecture. This, the free

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Nomenclature

A	area, m^2
C	capacity rate, W/K
C_D	drag coefficient
C_{th}	thermal conductance, W/K
D	diameter, m
F	force, N
f	friction factor
H	height, m
h	heat transfer coefficient, W/m^2 K
K_l	factor, m^2/s, Eq. (3)
K_t	factor, m^3/kg, Eq. (8)
L	length, m
$\dot{m}$	mass flow rate, kg/s
n	number of tubes
N_{tu}	number of heat transfer units
q	heat transfer, W
R_f	fluid flow resistance
R_{th}	thermal resistance
Re	Reynolds number
T	temperature, K
U	overall heat transfer coefficient, W/m^2 K
U_c	core fluid velocity, m/s
ν	kinematic viscosity, m^2/s
U_p	plenum fluid velocity, m/s
W	depth, m
x	fraction of the volume occupied by the plenum

Subscripts

c	cold, core
e	entrance
h	hot
in	inlet
l	laminar
out	outlet
t	turbulent
max	maximum
min	minimum
p	plenum

Greek symbols

ΔP	pressure drop, Pa
ρ	density, kg/m^3
ν	kinematic viscosity, m^2/s

evolution of the flow architecture is captured by the law of physics of evolution [17]: it is the essence of constructal design [18], and serves as unifying method for all evolutionary design phenomena. Here, we illustrate this approach by analyzing a morphing two-stream counterflow heat exchanger with one plenum at each end. The key architectural feature to be discovered is how much of the total volume is allocated to the counterflow core and the plenums.

2. Model

Two streams ($\dot{m}_1$, $\dot{m}_2$) flow in counterflow through parallel tubes of diameter D, in a core situated between two plenums. As shown in Fig. 1, each stream arrives into a plenum (xL) by flowing across tubes that carry the second stream. At the other end, the second stream arrives into a plenum by flowing across tubes that carry the first stream.

The elemental volume of the heat exchanger has the longitudinal length L (one plenum + the flow length of one stream) and the width $H = nD$, shown in the vertical direction in Fig. 1. The third dimension of this element is D, and is not shown in Fig. 1. The number of $H \times L$ elements stacked in the third dimension is not important. The size of the element is dictated by the area $A = HL$, which is fixed, and the tube diameter, which is also fixed.

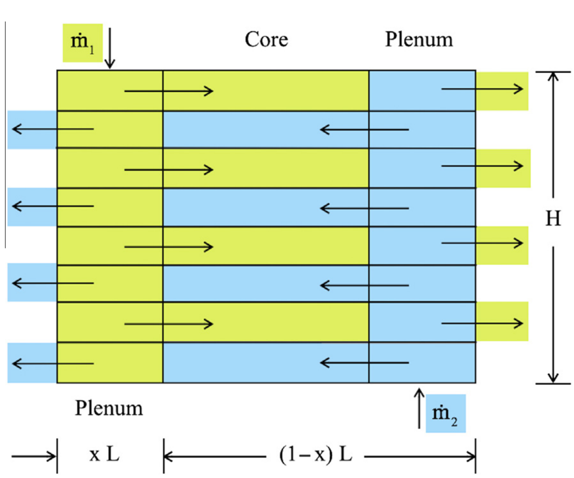

Fig. 1. Two streams in counter flow between two plenums in cross flow.

The challenge is to determine the most effective configuration, which is represented by the shape parameter (H/L) and the volume fraction occupied by one plenum (x). The element has two functions, flow with low pressure drop along each stream, and low thermal resistance between the streams.

3. Pressure drop

The overall pressure drop experienced by one stream $\dot{m}_1$ is due to contributions from the plenum (p) and the core flow (c), through tubes of diameter D and length $(1-x)\,L$,

$$\Delta P = \Delta P_p + \Delta P_c \tag{1}$$

The plenum has the flow cross section xLD, flow length H (vertical in Fig. 1), and average fluid velocity $U_p = \dot{m}_1/(\rho xLD)$. The drag force experienced by each tube of length xL (or frontal area xLD) inside the plenum is $F_1 = C_D xLD\frac{1}{2}\rho U_p^2$, where we regard C_D as of order 1, based on the assumption that the Reynolds number based on D inside the plenum is greater than 10^2. From the vertical force balance on the plenum, $\Delta P_p xLD = nF_1$, we deduce

$$\Delta P_p = \frac{n\dot{m}_1^2}{2\rho(xLD)^2} \tag{2}$$

For the in-tube flow, we start with the assumption that the flow is in the fully developed laminar regime. The mass flow rate through one tube is $\dot{m}_1/n$. The pressure drop along the duct of length $(1-x)\,L$ is

$$\Delta P_c = K_l \frac{\dot{m}_1}{n}\frac{(1-x)L}{D^4} \tag{3}$$

where $K_l = 128\nu/\pi$. Next, we combine Eqs.(1)–(3), replace n with H/D, and then replace H with A/L. The resulting expression for the overall pressure drop is

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