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Heat transfer in helical, curved rectangular channels—comparison of type I and type II systems

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Abstract—An experimental study of the heat transfer coefficients for flow of air in curved rectangular channels was undertaken to determine the relative advantages of Type I (aspect ratio > 1) and Type II (aspect ratio < 1) systems in a helical flow geometry. Much higher heat transfer rates (76% increase over that of an equivalent straight duct) were obtained with the Type II system than with the Type I system (6% increase over the straight duct). Although both the radius of curvature and the aspect ratio interact in a complex way to produce heat transfer enhancement, our experiments confirm that the curved channel with the tighter coil will produce the most enhancement.

INTRODUCTION

HEAT TRANSFER in curved conduits has received continuing interest over the years since the pioneering work of Dean [1, 2]. The curvature introduces secondary flow patterns which enhance mixing and heat transfer. This is especially true in the laminar flow regime, where the literature is quite extensive, for example studies in coiled tubes [3, 4] and swirl devices (twisted tapes) that impart a helical component to a primary axial flow [5], but it is also true in turbulent flow. Curved channel devices may be helical, in which there is axial progression in a corkscrew or coil spring fashion but no change in radius of curvature, or spiral, in which there is no axial progression, but rather the coil expands in radius.

Curved rectangular channels are used industrially for heat exchange devices, but much less work has been reported in this area than the foregoing. Mori *et al.* [6] used curved square channels to study enhancement in primarily laminar flow, but also reported some results for turbulent flow. Dement'eva and Telegina [7] and Dement'eva and Aronov [8] studied heat transfer and pressure drop in curved rectangular channels of different aspect ratios. Gupta and Date [9] investigated helical annular flow with the outside wall heated and a twisted tape forming part of the channel wall. Kadambi *et al.* [10] give data and correlation for a helical device with channels of rectangular cross-section.

In this work we investigate the performance of two helical heat exchangers with rectangular ducts of widely differing aspect ratio, in order to determine their relative heat transfer enhancement. One of these exchangers is a Type I system, where the duct cross-section is wider than it is deep; and the other is a Type II system, where the duct cross-section is narrower than it is deep. Type I systems have many practical advantages, such as high surface area-to-volume ratio, compact space geometry, relative ease of construction and shorter flow path leading to lower pressure drop in comparison to Type II systems. However, the most prevalent curved channel design industrially is the spiral exchanger, a Type II configuration. Heat transfer enhancement is evaluated by comparing the performance of the exchangers relative to that expected from a corresponding straight duct of the same dimensions.

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EXPERIMENTAL

Figures 1 and 2 show the geometry and construction detail of each exchanger used in this work. Figure 3 shows a schematic of the geometric variables and the definition of Type I and Type II systems according to aspect ratio.

Type I system

The Type I system, shown in Fig. 1, was constructed from flat galvanized steel sheet, 0.61 mm (0.024 inches) thick. The sheets were cut and formed into a two-fluid helical exchanger with a flat, pancake-like shape. The channels were arranged in a double-helix pattern, and all external surfaces of the exchanger were insulated. The two air streams flowed countercurrent to each other in balanced, equal flow rates measured by an air-velocity anemometer at the outlets.

Sealing the channels was done with formed U-channel sheet steel riveted to the top and bottom plates at the edges as in Fig. 1. The joining surfaces were sealed with silicone sealant. The air flow rates were measured by an orifice meter and a 'Fan-E' calibrated air flow meter from the Air Monitor Corp. Temperatures to

NOMENCLATURE

A	area [cm^2]
a	bend radius measured from center of curvature to inside wall of a curved rectangular channel ($=r_i$) [cm]
C_p	heat capacity [kJ kg^{-1} K^{-1}]
D	bend diameter, $2R$, measured from center of curvature to center of duct [cm]
d_{eq}	equivalent diameter, $4R_H$ [cm]
h	film heat transfer coefficient [W m^{-2} K^{-1}]
H	axial progression of helix in 1/2 turn [cm]
k	fluid thermal conductivity [W m^{-1} K^{-1}]
L	length of channel (from center of duct) [m]
m	mass flow rate [kg s^{-1}]
Nu	Nusselt number, hd_{eq}/k
Pr	Prandtl number, $\mu C_p/k$
Q	heat rate [W]
r	radial distance from center of curvature [cm]
Re	Reynolds number, $d_{eq}v\rho/\mu$
R_H	hydraulic radius = area cross section/wetted perimeter [cm]
r_i, r_o	inner and outer radii of curved channel [cm]
r^*	radius ratio [dimensionless]
T	temperature [K]
ΔT	temperature difference [K]
w	channel width [cm]
y	helix pitch ratio, $H/2(r_o-r_i)$ [dimensionless]
z	channel depth [cm].

Greek symbols

μ	fluid viscosity [Pa s]
ρ	fluid density [kg m^{-3}].

Subscripts

air	fluid
av	average
b	base, inside tube surface
eq	equivalent
f	fin
i	inner (radius)
lm	log mean
o	outer (radius).

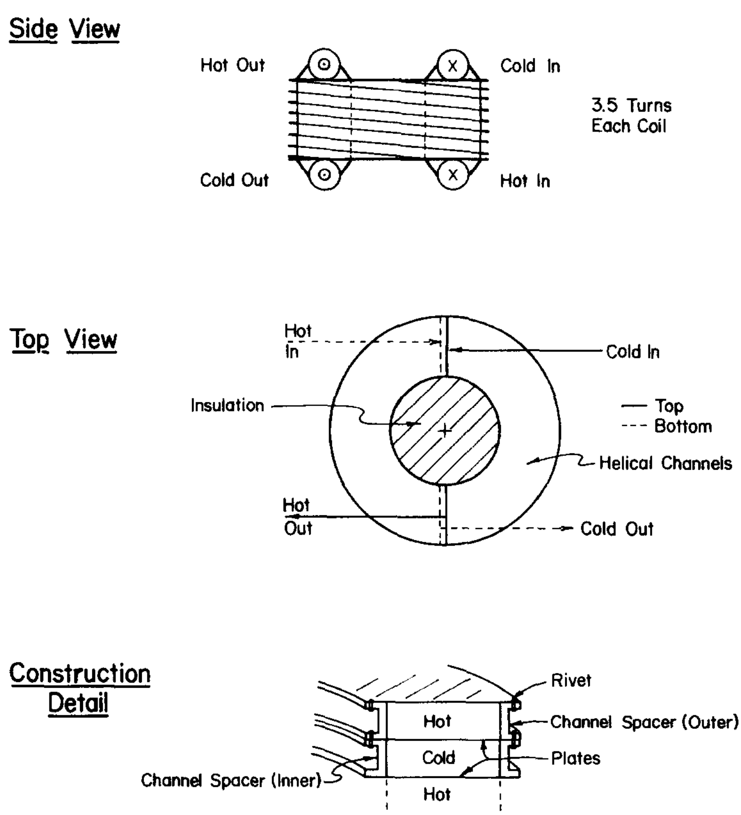

FIG. 1. Construction details of Type I system based on double helix counter flow exchanger.

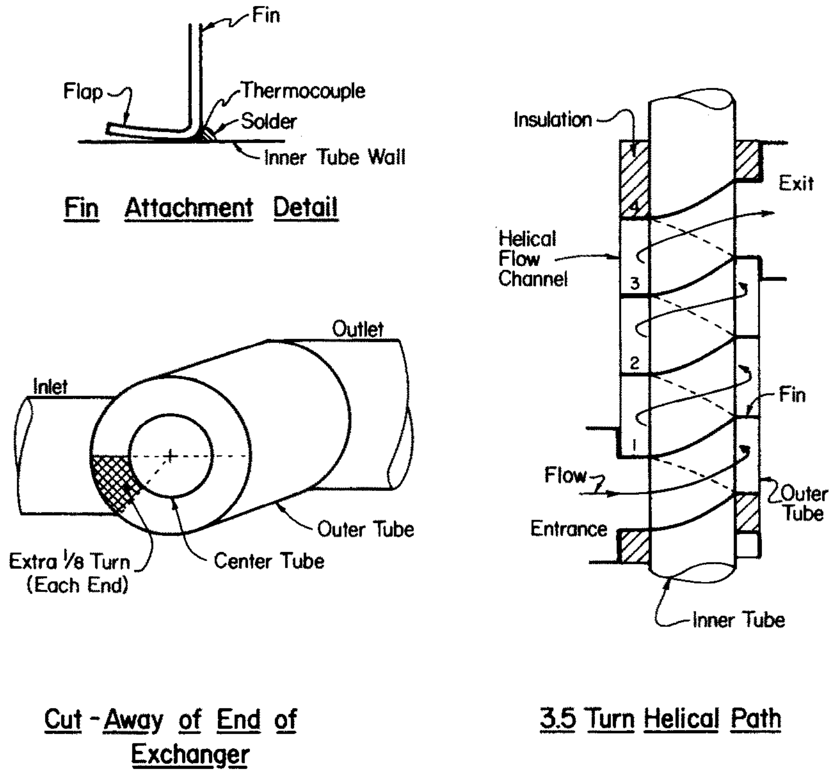

FIG. 2. Construction details of Type II system based on helical annular flow.

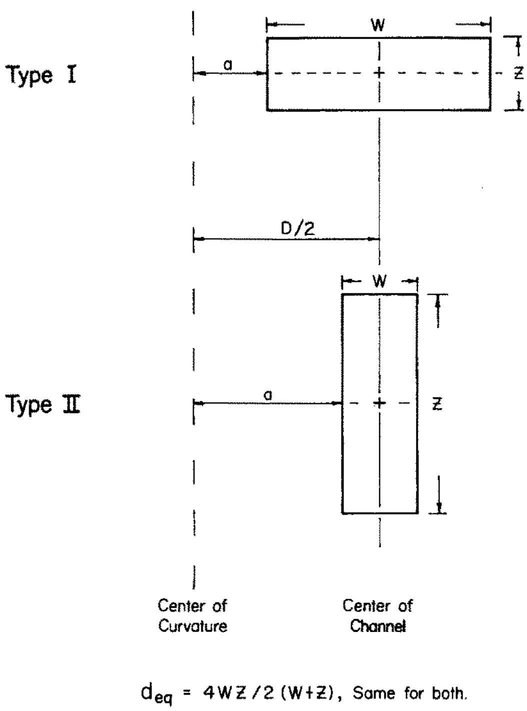

FIG. 3. Comparison of Type I and Type II curved rectangular channels—geometrical parameters.

$\pm 0.1^\circ$C were measured with copper–constantan thermocouples and a digital readout.

To reduce the data to Nusselt form, the overall heat transfer coefficient (U) was first calculated using the log-mean temperature difference formulation. The wall resistance (wall thickness/thermal conductivity) was subtracted from the overall resistance ($A\Delta T_{lm}/Q$) and the result multiplied by two to obtain the film coefficient (h). A is the heat transfer surface area and Q is the heat flow rate. Since each film would have the same resistance at the same air flow rate, the flows on each side needed to be the same for this technique to work. The difference between LMTD and average ΔT driving force was negligible, because the maximum temperature rise for both fluids was small (about $16^\circ\text{C} \pm 0.2$). In a heat balance the temperature rise of the cold air should equal the temperature loss of the heating air. This occurred in all runs to within $\pm 0.4^\circ$C.

Type II system

The Type II system (Fig. 2) was primarily a commercial heat reclaimer unit using annular flow to reclaim heat from a flue gas flowing in the central pipe. We modified this unit to obtain a helical flow path in the annulus by fitting a copper fin (0.030 inches thick) helically wound along the length of the inside tube and extending through the annular space to the outside tube. The annular helical duct went through 3.5 turns from inlet to outlet. The fin was soldered to the inside tube to form a continuous metal contact and seal. The fin collar winding was a flap of fin material about 1.9 cm (3/4 inch) wide, cut about every 3 cm (1.25 inches) and bent over to fit the tube in the construction process (see detail in Fig. 2). This acted as a support for the fin and was not removed after the fin was soldered into place.

Because the outlet ports were much larger than the helical channel cross sectional area, unwanted flow from channels other than the last or first was excluded by blocking the exit/entrance so that only flow from the first or last turn entered/exited the exchanger. In the calculations, an allowance of an extra 1/4 turn was used to account for the extra area contacted by the annular fluid upon entrance and exit from the exchanger (see detail in Fig. 2).

The inside surface of the outside tube was covered with a layer of closed-cell polyethylene foam, 1.5 mm (1/16 inch) thick, for sealing purposes. The outside of the assembly was wrapped in fiberglass insulation. Hot air flowed in the center tube at about 82°C (180°F), and cool air to be heated entered the annular helical channel at room temperature (about 23°C). Thus, air entering the helical annular channel was heated by both the center tube and the fin.

Inlet and outlet temperatures were measured by copper–constantan thermocouples. Other thermocouples measured wall temperatures along the inside tube (5 thermocouples) and the copper fin (tip, base and midpoint at three locations—entrance, middle and exit—of the exchanger). The average temperatures of the inside tube and the fin were used to calculate the heat transfer coefficient at various flow rates to reduce the data to Nusselt format.

The average fin temperature was computed by fitting the hyperbolic cosine function [11] to the measured average temperature profile (a function of radius), then integrating numerically for a circumferential fin shape. The hyperbolic cosine function is the solution to the general differential equation describing heat transfer in a fin when the fin tip is insulated, as is the case here. The temperature at the fin mid-point was measured experimentally and agreed with that predicted by the function to $\pm 5\%$, so we believe the fin temperature profile was very accurate. An arithmetic average was used to calculate the average wall (base) temperature.

The fin temperature driving force and the base temperature driving force were used with their appropriate area terms and the heat rate, calculated by the inlet and outlet temperatures of the air and its flow rate, to obtain the experimental value of the film heat transfer coefficient, according to equations (1) and (2):

$$h = Q/(A_b \Delta T_b + A_f \Delta T_f) \tag{1}$$

and

$$Q = mC_p \Delta T_{air} \tag{2}$$

where h is the film heat transfer coefficient, Q is the heat rate, m is the mass flow rate of air, ΔT_{air} is the temperature rise of the air, C_p is the air heat capacity, A_b is the heat transfer area of the inner tube (minus a correction for the fin collar winding), A_f is the total fin area for 3.5 turns of the fin (plus a correction for the fin collar winding), ΔT_b is the temperature difference between the average tube wall temperature and the average bulk temperature of the air being heated, and ΔT_f is the corresponding driving force for the fin (average fin temperature minus bulk average air temperature). The bulk average air temperature was the arithmetic average of inlet and outlet temperatures.

The driving forces were measured directly, and therefore fin efficiency was not used. The joining material (solder) at the base is a low thermal conductivity material and acts as a 'contact resistance' for the fin, thus rendering the fin efficiency method ineffective.

A summary of the geometrical dimensions of the two exchangers is given below in Table 1.

RESULTS AND DISCUSSION

The results of both exchanger types are shown in Fig. 4, where the heat transfer performance is compared to that of an equivalent straight channel. The Type I system showed only minor enhancement, about 6%. The Type II system showed an average enhancement of 76% over the straight duct performance calculated by the Seider–Tate formulation:

$$Nu_{sd} = 0.023 Re^{0.8} Pr^{0.33}. \tag{3}$$

These results are for the turbulent flow regime. The laminar flow regime has been studied much more extensively elsewhere, and our equipment was unsuitable for such low flow rates. Typically, enhancement in the laminar regime would be much more, because

Table 1. Summary of geometrical dimensions for helical duct heat exchangers with rectangular channels

	Type I	Type II
width, w [cm (in)]	29.85 (11.75)	3.81 (1.5)
depth, z [cm (in)]	2.22 (0.875)	8.89 (3.5)
aspect ratio, w/z	13.5	0.429
eq diam, d_{eq} [cm (in)]	4.13 (1.626)	5.33 (2.1)
bend diam, D [cm (in)]	90.17 (35.5)	19.05 (7.5)
d_{eq}/D	0.046	0.28
turns	3.5	3.5
curv rad, a [cm (in)]	31.12 (12.25)	7.62 (3.0)
L/d_{eq}, dimensionless	243	39.3

Note: d_{eq} is the equivalent diameter of the duct computed in the usual way ($4 \times$ hydraulic radius); D is the bend diameter (twice the bend radius measured from center of curvature to the center of the duct); a is the bend radius from center of curvature to the inside wall of the rectangular channel; w is the duct width, and z is the duct height (or depth). Figure 3 shows the variables in relationship to the geometry.

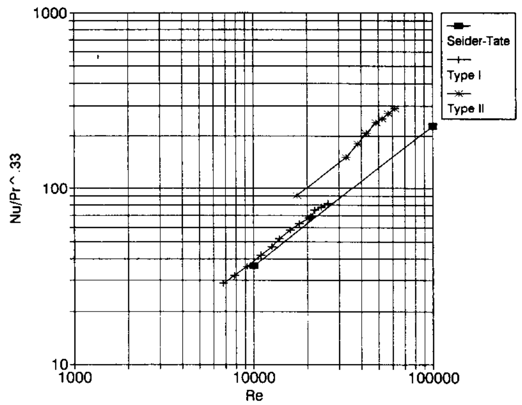

FIG. 4. Heat transfer results for Type I and Type II systems compared to the Seider–Tate correlation for turbulent flow in straight ducts.

the curved duct induces a secondary swirl pattern in the flow, maintaining a turbulent character. Our Type I data show less scatter than the Type II data because of the way in which the heat transfer coefficients were calculated. The scatter in the Type II data amounts to about $\pm 10\%$, typical for heat transfer studies of this type.

The primary reason for the greater enhancement of the Type II system over the Type I system is the larger Dean curvature parmeter, d_{eq}/D, of the Type II system. In general, Type II systems by their geometry have greater potential for heat transfer enhancement, because the coil can be wound tighter, giving larger Dean parameters, than in the Type I case.

The enhancement factor, Nu/Nu_{sd}, due to curvature of the duct is often reported [12] as 3.54 d_{eq}/D, which predicts enhancement of 16% for the Type I exchanger and 99% for the Type II exchanger of this work. These do not correspond exactly with our results, but the trends are reflected accurately. One difficulty quoted by Dement'eva and Aronov [8] is that the handbook recommendation does not include an aspect ratio parameter. They propose a different formulation, $3.66(d_{eq}/D)^{1.22}(w/z)^{0.22}$, which gives slightly different results, 15% for the Type I system and 64% for the Type II system. These results fit our data better.

Other literature data are compared with ours in Table 2. Table 2 shows that Type II systems give higher enhancement, in general, than Type I systems. Gupta and Date [9] report results for three Type II systems of different geometrical dimensions, and they all showed quite high enhancement (30–100%). Kadambi *et al.* [10] report heat transfer results on two Type I systems; they show enhancement of 20–40%. This is less than the Gupta and Date [9] results and our Type II results, but higher than our Type I results.

However, a few anomalies do exist (Mori [6], KLN "A" [10] and Gupta and Date "2" [9]). In all cases the data for these three situations showed a slope of *Nu* vs *Re* quite removed from 0.8 in the power of the Reynolds number. Mori *et al.*'s data [6] were taken for less than one full turn of the channel. In all other cases, there were at least two full turns of the coil, so the differences seen in Mori *et al.*'s data [6] may be due to a relatively large entrance contribution.

In apparent contrast to the conclusions of Table 2, Dement'eva and Aronov [8], and Dement'eva and Telegina [7] claim better enhancement from Type I systems. Their data were not reported in Nusselt format and could not be translated into the form of Table 2. Some of their data were for ducts of equal w/z and the same d_{eq}/D, but different a/w or r^*. The Type I system shows better enhancement for their experimental conditions, not because it is Type I, but rather because their Type I channels had a lower a/w or r^* (as in Fig. 3), hence tighter coil and larger enhancement.

In general, designs with low pitch ratio (y) and tight coils will have better enhancement as confirmed by Gupta and Date [9]. However, the aspect ratio also

Table 2. Comparison of enhancement in curved, rectangular channels from literature data

Literature source (Type)	Dean param d_{eq}/D	aspect ratio w/z	coil tightn a/w	radius ratio r^*	pitch ratio y	Enhancem experim Nu/Nu_{sd}
Mori (sq.)	0.036	1.0	12.85	0.93	0	1.42
KLN "A" (I)	0.049	2.43	5.5	0.85	<1	1.32
KLN "C" (I)	0.027	2.43	10.4	0.91	<1	1.19
G&D "1" (II)	0.462	0.047	1.56	0.61	5.04	1.31
G&D "2" (II)	0.794	0.047	0.70	0.41	5.30	1.28
G&D "3" (II)	0.761	0.095	0.70	0.41	2.66	2.04
This work (I)	0.046	13.5	0.96	0.49	0.04	1.06
This work (II)	0.28	0.423	2.2	0.69	0.58	1.76

Note: Literature sources were Mori *et al.* [6], Kadambi *et al.* [10] (KLN) and Gupta and Date [9] (G&D). In all cases except Mori, data were presented in standard Nusselt format. All experiments were done with air. Most were with constant heat flux, except this work, which were two-fluid heat exchange operations. Parameters listed in the table had to be derived from geometries described and presented in their own way by the authors. The variables r^* and y are related to our geometry as follows: $r^* = a/(a+w)$, and $y = z/4w$ (Type II) or $z/2w$ (Type I).

plays an important role as our data and those of Dement'eva and Aronov [8] show. This is especially true in curved duct heat exchangers where the axial progression of the helix is small (low pitch ratio). In twisted tapes, such as the annular type used by Gupta and Date [9], the pitch ratio is large, easily varied, and would play a much more important role.

CONCLUSIONS

In experiments with two helical path heat exchangers of widely differing aspect ratios, the Type II system, with aspect ratio (width/depth) less than 1, gave much better heat transfer enhancement than the Type I system, with aspect ratio greater than 1. In an analysis of other literature data, this would seem to be generally true, because Type II systems can be coiled tighter (low radius ratio) giving higher swirl effects and higher heat transfer as a result.

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