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Heat transfer in square duct fitted diagonally with angle-finned tape—Part 1: Experimental study☆

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ABSTRACT

The paper presents an experimental study on turbulent flow and heat transfer characteristics in a square duct fitted diagonally with 30° angle-finned tapes. The tested duct has a square section and uniform heat-fluxed walls and the flow rate of air used as the test fluid is presented in terms of Reynolds number from 4000 to 23,000. The angle-finned straight tape in the present work is newly invented without previous investigations available. The insertion of the finned tape is performed with three ratios of fin pitch to duct height ($PR=P/H$) at the fin attack angle of 30° with respect to the main flow direction. The finned tape inserted diagonally in the duct is expected to generate a longitudinal vortex flow pair through the heated duct. Influences of five fin-to-duct height ratios ($BR=b/H=0.1$–0.3) for each fin pitch on thermal and flow friction characteristics of the inserted duct are investigated. The experimental result shows that at smaller fin pitch spacing, the finned tape with $BR=0.3$ provides the highest heat transfer and friction factor but the one with $BR=0.2$ and $PR=1.0$ yields the best thermal performance. The thermal performance of the newly invented finned tape turbulator is found to be much higher than that of the wire coil/twisted tape turbulator.

1. Introduction

Vortex/swirl flows have been commonly used for increasing convection heat transfer coefficients in several engineering applications such as heat exchangers, drying processes and vortex combustors. There are many types of vortex generators employed in the heat exchanger ducts such as helical/twisted tapes [1–3], coiled wires [4–6], ribs/fins/baffles [7–9] and winglets [10,11]. Most vortex flow devices mentioned above are effectively applied to circular tubes while the rib/baffle/fin and winglets are suitably employed for the channels or flat surface ducts. In high performance heat exchanger duct systems, periodic ribs/baffles/fins have been widely applied in many industrial applications. The baffled/finned duct successfully prevents the development of thermal boundary layer, and therefore augments the heat transfer performance and results in much better heat transfer efficiency than that in smooth duct with no baffle/fin. Because of the practical importance, the heat transfer and flow characteristic in ducts with rib/baffle/fin turbulators have attracted many investigators.

Han et al. [12,13] studied experimentally the heat transfer in a square channel with ribs on two walls for nine different rib configurations. Average heat transfer and friction factor were reported for the $P/e=10$ and $e/H=0.0625$ rib by heating either only one of the ribbed walls or both of them, or all four channel walls. The heat transfer augmentations and the friction factor were highest for the 60° orientation amongst the angled ribs. Liou and Hwang [14,15] used a real time Laser Holographic Interferometry to measure the local as well as average heat transfer coefficient of square, triangular and semi-circular ribs and found that the square ribs give the best heat transfer performance among them. This is contrary to the experimental result of Ahn [16] reported that the triangular rib is better than the square one. Taslim et al. [17] conducted measurements of the heat transfer in a square channel with three e/H ratios ($e/H=0.083$, 0.125 and 0.167) and a fixed $P/e=10$ using a liquid crystal technique. Various staggered rib configurations were studied, especially for the angle of 45° and experimental data showed a significant increase in average Nusselt number with increasing the e/H ratio. Chandra et al. [18] studied heat transfer behaviors in a square channel with continuous ribs on four walls where ribs were placed superimposed on walls and found that the heat transfer increases with raising the number of ribbed walls and with reducing Reynolds number while the friction factor increases with both cases.

Kwankaomeng and Promvonge [19], and Promvonge et al. [20] studied numerically the laminar periodic flows over 30° and 45° angled baffles repeatedly mounted only on one wall of a square channel, respectively. They noted that the heat transfer enhancement for the 45° angled baffle with $BR=0.4$ was about 2–3 fold higher than that for the 90° baffle while the friction loss was some 10–25% lower. In addition, they found that a single streamwise main vortex flow created by the angled baffles/fins can help to induce impingement jets on the upper, lower and baffle trailing end side walls of the channel. The appearance of vortex-induced impingement (VI) flows led to drastic

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Nomenclature

A	convection heat transfer area of duct, m^2
AR	aspect ratio of duct, (W/H)
b	fin height, m
BR	fin blockage ratio, (b/H)
C_p	specific heat capacity of air, J/kgK
D	hydraulic diameter of duct, $(=H)$, m
e	rib height, m
f	friction factor
H	duct height, m
h	average heat transfer coefficient, W/m^2K
I	current, A
k_a	thermal conductivity of air, W/mK
L	length of test duct, m
$\dot{m}$	mass flow rate of air, kg/s
Nu	Nusselt number, (hD/k_a)
P	fin pitch spacing (axial length of spacing), m
Δp	pressure drop, Pa
PR	fin pitch to duct height ratio, (P/H)
Pr	Prandtl number
Re	Reynolds number, (UD/ν)
Q	heat transfer, W
T	temperature, K
TEF	thermal performance enhancement factor
t	thickness of fin, m
U	mean velocity, m/s
V	voltage, V
$\dot{V}$	volumetric flow rate, m^3/s
W	width of duct

Greek letters

α	attack angle of fin, °
ρ	density of air, kg/m^3
ν	kinematics viscosity, m^2/s

Subscripts

b	bulk
0	smooth duct
conv	convection
i	inlet
o	out
pp	pumping power
s	duct surface

increase of the thermal performance of the channel. In comparison, the 30° baffle/fin performs better than the 45° one due to lower pressure loss. Promvonge et al. [21,22] again investigated numerically the laminar flow structure and thermal behaviors in a square channel with 30° and 45° inline baffles on two opposite walls. Two streamwise counter-rotating vortex flows were created along the channel and VI jets appeared on the upper, lower and baffle leading end side walls while the maximum thermal performance factor was found for the 30° inline baffle case but the 45° inline baffle provides higher heat transfer rate.

From the literature review cited above, the use of wire coil/twisted tape inserts in the square duct may not be suitable due to the flow leakage in the duct corners leading to lower strength of the vortex flow. The works in Ref. [19–22] triggered the present work to investigate the heat transfer enhancement in a square duct inserted diagonally with a tape with two-sided angled fins attached. In a square duct with one wall roughened by repeated angled baffles (or fins), the baffle-induced secondary flows (or vortex flows) accompanied by enhanced turbulence intensity provide a drastic increase in heat transfer due to VI effects as reported in Ref. [20,21]. In the present work, another type of inserted devices has been developed by insertion of a 30° angle-finned tape into a square duct that has never been found in the literature. This inserted device stems from the concept that the square duct can be divided in diagonal into two isosceles triangular ducts and then the angled fins can be mounted repeatedly on the common base (diagonal side of the duct) of the two imaginary isosceles triangular ducts. Therefore, the diagonally inserted tape in the square duct is assumed to be the common base of the two isosceles triangular ducts. The present inserted device has been developed from a combination of the merits of rib, baffle, winglet and twisted tape turbulators. This means that the present inserted device will provide a drastically high heat transfer rate like baffles, low pressure drop like angled ribs, swirl/vortex flow as winglets and ease of practical use like twisted tapes. Therefore, a new 30° finned tape insert is proposed and expected to provide higher thermal performance than wire coil/twisted tape inserts due to the VI effect. The experimental results using air as the test fluid for the 30° angle-finned tape inserted diagonally in the square duct are presented for turbulent flows in a *Re* range of 4000 to 23,000 in the current work.

2. Experimental setup

A schematic diagram of the experimental apparatus is presented in Fig. 1 while the detail of a tape with double-sided angled fins, inserted diagonally into the square duct is shown in Fig. 2. In Fig. 1, a circular pipe used for connecting a high-pressure blower to a settling tank was attached by an orifice flow meter to measure the flow rate while a square duct including a calm section (2000 mm) and a test section (1000 mm) was employed following the settling tank. The square duct configuration was characterized by the duct height, H of 45 mm while three fin-pitch to duct-height ratios ($PR = P/H = 1$, 2 and 3) and the fin attack angle of 30° were used for the finned tape. The overall length of the duct was 3000 mm. The tested square duct made of 3 mm thick aluminum sheets has a cross section of 45×45 mm^2 and 1000 mm length (L). The diagonal straight tape was made of aluminum with its dimension of $63 \times 1500 \times 0.5$ mm^3. The five fin strip sizes were 4.5, 6.75, 9, 11.25 and 13.5 mm high (b) with 0.3 mm thickness (t). The angled fins made of a 0.3 mm aluminum strip were attached on the two sides of the aluminum tape with hot superglue. To assure a fully developed periodic flow throughout the test duct, the tape inserted diagonally into the duct was extended by about 11D upstream of the test section. The test section consisted of the four heating walls. The AC power supply was the source of power for the plate-type heater, used for heating all walls of the test section in order to maintain a uniform surface heat flux.

Air as the test fluid in both heat transfer and pressure drop experiments, was directed into the systems by a 1.45 kW high-pressure blower. The operating speed of the blower was varied by using an inverter to provide desired airflow rates. The flow rate of air in the systems was measured by using an orifice plate system pre-calibrated by using hot wire and vane-type anemometers. The pressure drop across the orifice was measured using an inclined manometer. In order to measure temperature distributions on the upper and lower wall and a sidewall, thirty thermocouples were fitted into the outer duct walls. The thermocouples were installed in holes drilled from the rear face and centered of the duct walls with the respective junctions positioned within 1.5 mm of the inside wall and axial separation was 100 mm apart. To measure the inlet and outlet bulk temperatures, two sets of two thermocouples were positioned upstream and downstream of the test duct. All thermocouples were type K, 1.5 mm diameter wire. The thermocouple voltage outputs were fed into a data acquisition system (Fluke 2650A) and then recorded via a personal computer.