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Experimental Investigation of the Effect of the Cross-Sectional Geometry on the Flow

Hasan GÜL*, Duygu EVİN

**Fırat Üniversitesi, Teknik Bilimler Meslek Yüksekokulu, 23119, Elazığ*
Fırat Üniversitesi, Mühendislik Fakültesi, Makine Bölümü, 23119, Elazığ

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ABSTRACT

In this study, flow and friction characteristics in the rectangular-to-rectangular transition channels, which have cone angles of $\phi = 4^0, 5^0, 6^0$ were investigated experimentally. The ratio of the channel inlet cross-sectional area to that of exit is 1:2. The transition ducts have different geometrical dimensions and cone angles. The duct entry and exit aspect ratios were chosen as 1.42 and 1.4, respectively. Measurements were taken at several stations, x/L, with the Reynolds number ranging from $2x10^5$ to $5x10^5$. Velocity profiles were measured starting from inlet to downstream using hot-wire anemometer. Based on experimental results, different flow characteristics were obtained. Friction coefficient decreased with increasing pipe length and increasing Reynolds number. It was seen that cross flows occurred at low Reynolds numbers.

Keyword: Asymmetric transition duct, Secondary Flow, Flow measurement

1. INTRODUCTION

Pipes having same or different cross-sectional area are used for transition ducts which are known as "transition pipes" or "transition fittings". These types of pipes can be seen on many machines and pipe line networks. For instance, transition parts of air condition channels, water and wind tunnels, entrance regions of machines pump and ventilator networks. Ducts with non-circular cross sections are also frequently encountered in industrial heat transfer equipment, compact heat exchanger, cooling channels in gas turbine blades, nuclear reactors, ventilation and turbomachinery, etc. The main flow in such ducts is influenced by the secondary motions in the plane perpendicular to the streamwise direction, commonly referred to as secondary flows of Prandtl's second-kind. The secondary motions may distort the axial flow and induce a reduction of the volumetric flow rate due to a considerable friction loss, especially in corrugated ducts. These motions are of major concern since they redistribute the turbulence kinetic energy in the cross section of the duct [1], which in turn affects the heat flux and temperature field distributions.

In general, power plants and micro turbine systems are designed to obtain high effectiveness and low pressure losses, minimum volume and weight, high reliability and low cost [2]. Applications, generally the heat exchangers contain flow channels with various cross-sectional shapes, curves, expanded-narrow channels or waves in the main flow direction, to enhance the heat transfer.

Over the two decades, the constant-extensional rate or transition channel systems have become a rapidly developing technology, finding applications in many areas of engineering and science. Burley et al. [3, 4] tested five circular-to-rectangular transition ducts including one with swirl vanes installed in the upstream of the duct inlet. These measurements were limited to values of surface static pressure, thrust ratio performance parameter, and discharge coefficient.

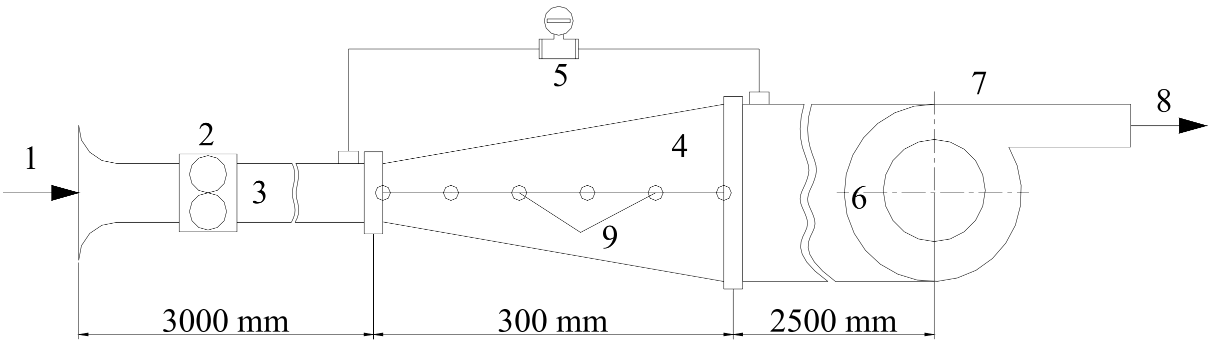
Several experimental researches have experimentally explored the aerodynamics of circular-to-rectangular transition ducts. Patrick and McCormick [5,6] recorded values of total pressure, mean velocity, and three normal Reynolds stress components at the inlet and exit planes of the two different circular-to –rectangular transition ducts. Miau et al. [7, 8] measured mean velocity and turbulence intensities at the inlet and exit of three circular-to-rectangular transition ducts. Schlichting [9] documents theories and experimental data from the pioneering works by Hagen [10].
There is enough information about the flow in pipes having different cross sectional geometry and cross-sectional area, but in most applications cross section geometry and cross-sectional area are not constant and

* Corresponding author, e-mail: hgul@firat.edu.tr

symmetric. Those kinds of duct transition systems need a joining-part which has variable section geometry and cross-sectional area. However, there are few works on turbulent flow in transition pipes with a rectangular cross-section, which is important for practical purposes. For this reason, the aim of this study is to investigate the flow and friction characteristics of transition channels with a cross-sectional ratio of 2:1 and equivalent conical angles of $\phi = 4,5,6$.

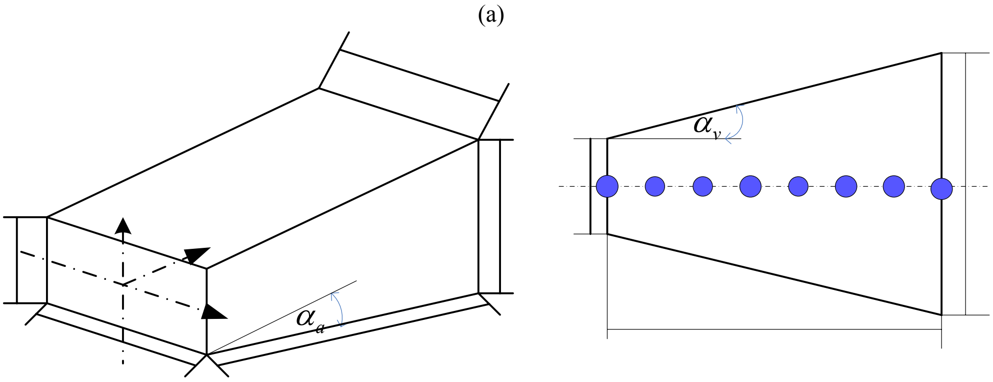
2. EXPERIMENTAL SETUP

The experimental set-up is shown in Fig.1. Air blown from a fan first passes through a channel of 3000 mm length to provide fully developed flow conditions and then through a transition duct of 300 mm length. The entry and exit cross-sectional dimensions of the transition duct are 140 mm x 200 mm and 140 mm x 200 mm, respectively. The Transition duct is symmetrical. The angle between the lateral sides and the duct axis varies as $\alpha_a=8^0$, 6^0,5^0 while the angle between the top (or bottom) side and the duct axis varies as $\alpha_v=4^0$, 3^0, 2^0. The Cone angle of the transition duct is chosen as $\varphi=6,^0 5^0,4^0$. The flow exits into the atmosphere from a duct of 3000 mm length after the transition region so as not to be effected from the ambient conditions. In the transition region the cross-sectional area increases to as much as 2 times the inlet area. The detailed test conditions at different flow rates are shown in Table 1.



1- Air inlet, 2- flow-meter 3- flow development duct 4- transition channel 5- pressure transducer 6- exit duct 7-fan, and 8-air exit

(a)



(b)

Figure 1. (a) Experimental set up, (b) Measurement holes

To characterize the flow in the transition duct, mean velocity and local velocity profile measurements were carried out in a wide range of different length, side and cone angles. Hot wire anemometer was used for the velocity measurements along the channel and pressure changes were measured via a pressure transducer. Holes having diameters of 2 mm were drilled on the top and lateral sides of the test channel to obtain the local velocity distribution along the channel. Consequently, the local velocities along the channel were measured at 5 mm intervals vertically along $0 \le z/(a/2) \le 1$ and horizontally along $0 \le y/(b/2) \le 1$ each for x/L=0.0 , 0.20, 0.40, 0.60, 0.80, 1. The pressure drop across the channel

is measured by pressure transducer located to the inlet and outlet of the channel. In addition, surface static

pressure measurements were made through small, 2mm, tap holes whose axis oriented normal duct surface. There, 8 static pressure taps equally spaced on the duct surface in the xz-plane along x direction as shown in Fig. 1-b.

An important parameter to define the transition channels having different cross-sectional areas and geometries is the equivalent cone angel. So as to explain this parameter the following definitions are required:

The dimensionless area change along x axis can be written as;

$$\overline{A}x = \frac{Ax}{A1} = 1 + f_1(\gamma,\delta,K).\overline{x} + f_2(\gamma,\delta,K).x^{-2} \qquad (1)$$

where $K = A_2 / A_1$, $\gamma = a/b$ and $\delta = c/d$ [11]. If L indicates the length of the transition channel, $\overline{x} = \frac{x}{L}$ and $f_1(\gamma,\delta,K)$ for transition channels having different geometries can be obtained from the references of [11, 12, 13]. Consequently, $f_2(\gamma,\delta,K)$ can be determined depending on K and $f_1(\gamma,\delta,K)$ as;

$$f_2(\gamma,\delta,K) = K - 1 - f_1(\gamma,\delta,K) \qquad (2)$$

Finally, equivalent cone angle can be defined as [12];

$$\tan\phi = \frac{R_1}{2L}\frac{f(\gamma,\delta,K) + 2.f_2(\gamma,\delta,K).\overline{x}}{\sqrt{A_x}} \qquad (3)$$

where R_1 is the equivalent radius of the inlet cross-section, A_1 and A_2 are the inlet and exit cross-sectional areas of the rectangular duct, respectively.

Table 1. Experimental conditions

	$Re \times 10^{-5}$	x/L	L(m)	$\phi(^0)$	$\alpha_v(^0)$	$\alpha_a(^0)$	$A_1(m^2)$	$A_2(m^2)$
Case1	2, 3,4,5	0 - 1	500	4	2	5	0,028	0.056
Case2			750	5	3	6		
Case3			1000	6	4	8		

The Reynolds number is based on the inlet mean velocity and the transition duct inlet diameter;

$$Re = \frac{uD_h}{\nu}, \; D_h = 4A_1/P_1 \qquad (4)$$

The pressure drop is defined by

$$f = \frac{\Delta P}{\left(\frac{L}{D}\right)\rho\frac{v^2}{2}} \qquad (5)$$

where f is Darcy's friction factor.

3. RESULTS AND DISCUSSION

Local velocity distributions for four different Reynolds numbers are shown in Figures 2 to 7. The expected decrease in velocity along the axis of the channel occurred due to the area increase and the velocity decrease near the surfaces because of the developing boundary layer which can be seen clearly from these figures. α_a and α_v change with the changing length of the channel. The effects of this angular change on the vertical and horizontal velocity profiles are nearly the same for the transition channels at the lengths of L=750 [Fig. 4-5]. Although the vertical velocity profiles for the channel at length L_1 [Fig.2] look like the profiles of the channels at lengths of L_2 and L_3 [Fig.3], the profiles at horizontal velocity component comparatively differ at the channels of x/L=0.0 , 0.20, 0.40, 0.60, 0.80, 1. The velocity profiles for low Reynolds numbers are different from the profiles of the high ones. It can be said that cross flows occur at the channel length of 750. However these disorders in the flow near the surfaces get better with the increasing Reynolds number. This kind of horizontal velocity distribution is caused by the high value of α and the short length of the channel. The local velocity at the same point of the channels in different lengths differs for the same flow conditions. Although the difference in the local velocities along the velocity profiles [Figs.5–7] between the channels having the lengths of L=750 and L=1000 mm is small for the same Reynolds numbers, the difference increases at the channel having L=500 mm length. This case may be explained by the cross flows that occurs especially at the shortest channel.

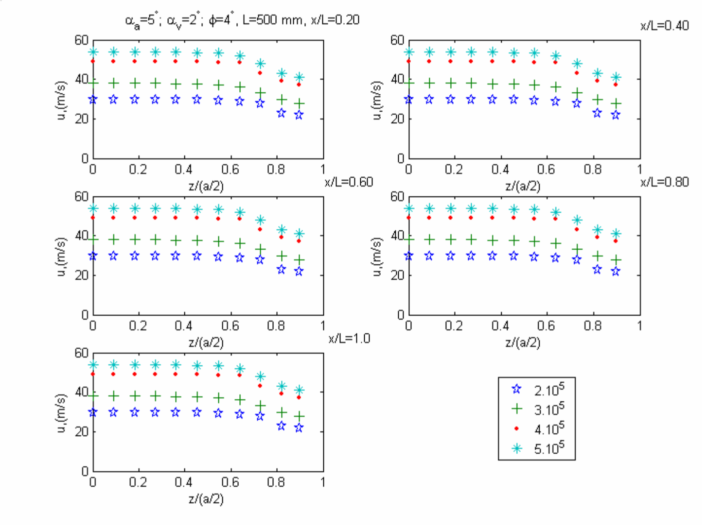


Figure 2. Vertical velocity distributions, Re=$2x10^5$-$5x10^5$

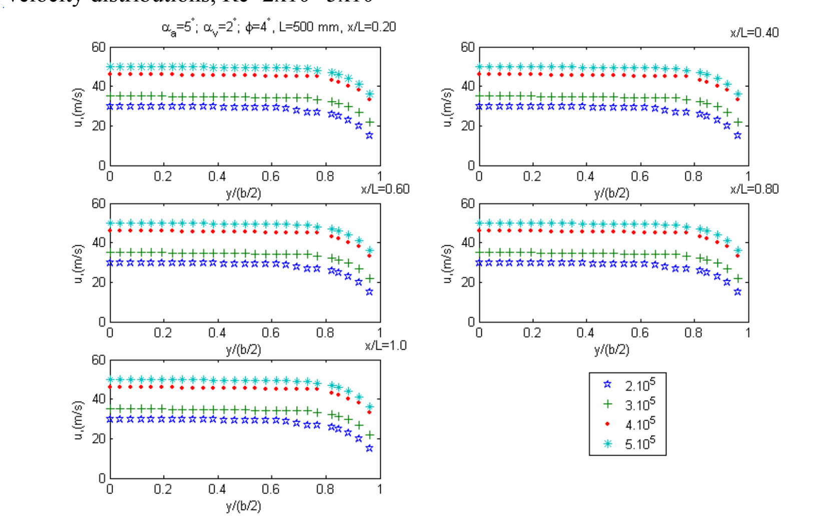


Figure 3. Horizontal velocity distributions, Re=$2x10^5$-$5x10^5$

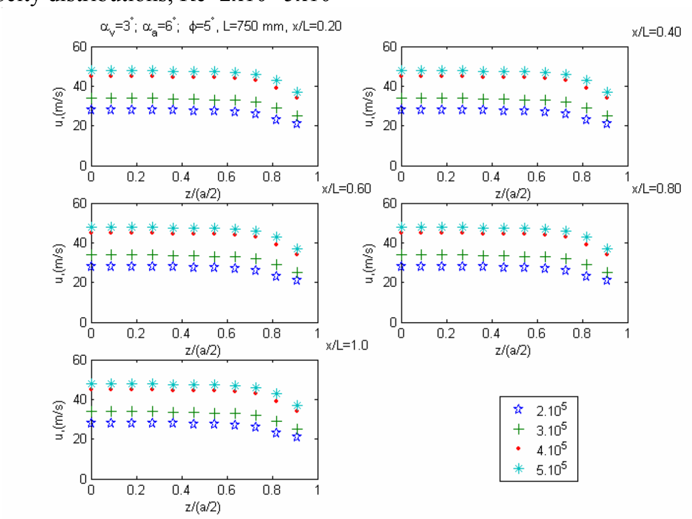


Figure 4. Vertical velocity distributions, Re=$2x10^5$-$5x10^5$

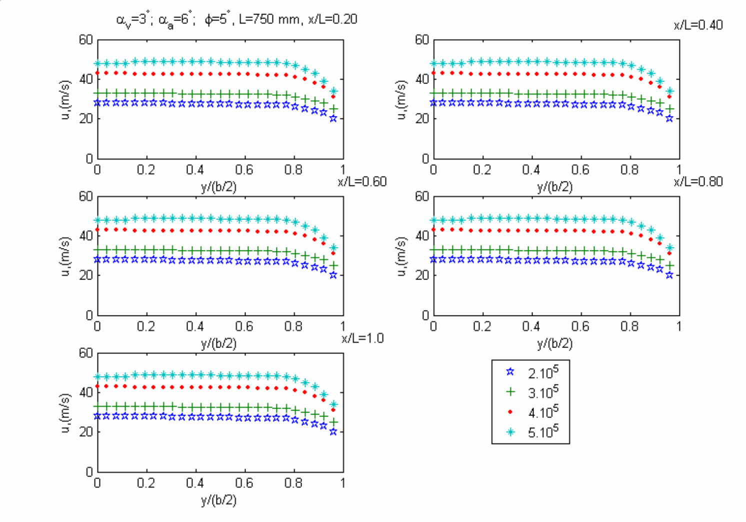


Figure 5. Horizontal velocity distributions, Re=2x10^5-5x10^5

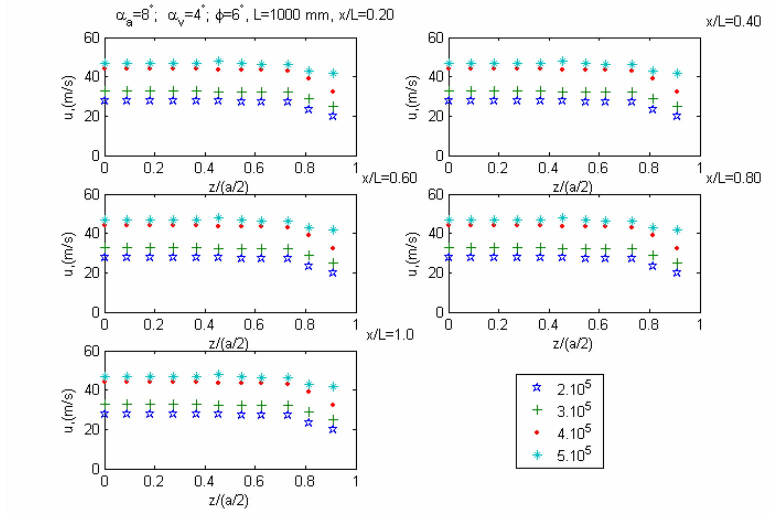


Figure 6. Vertical velocity distributions, Re=2x10^5-5x10^5

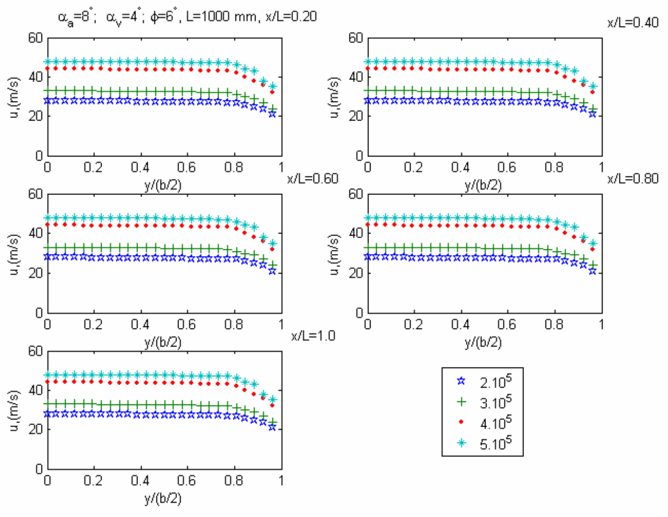


Figure 7. Measured horizontal -velocity distributions, Re=2x10^5-5x10^5

The friction factor distribution for various Re numbers can be seen in Fig.8. The highest pressure coefficient was seen for L=500 mm duct and the lowest load loss coefficient was obtained for the longest pipe L_3=1000 mm. All these behaviours can be explained by the flow corrections by means of increasing pipe length and pressure loss effects. Pressure coefficients for vertical axis are bigger than horizontal axis for these transition parts. This behaviour arises from the bigger angle of the side expansion angle and a bigger horizontal boundary static pressure than vertical one. Static pressure increases because of the transition part expansion throughout the pipe length.

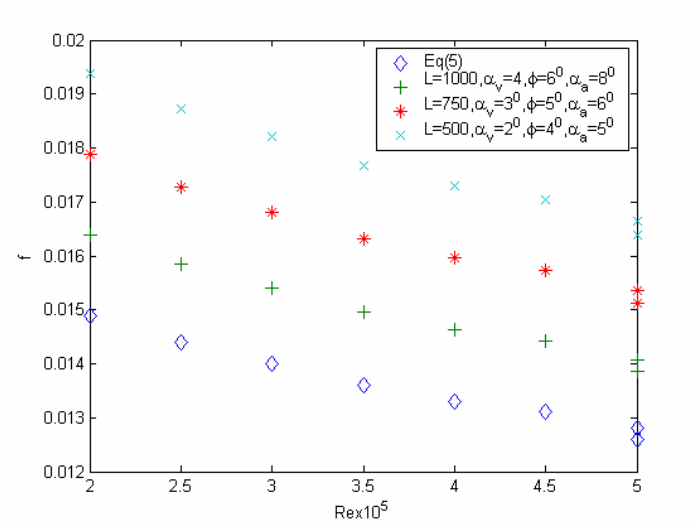


Figure 8. Friction factor distribution for various Re numbers

NOMENCLATURE

A_1 : inlet cross sectional area of the transition channel, m^2
A_2 : exit cross sectional area of the transition channel, m^2
D_h : hydraulic diameter of the channel, m
f : friction factor
g: gravitational acceleration,m/s^2
L: length of the channel, m
P_1 : inlet periphery of the channel, m
Q: volume flow rate of water through the tube, m^3/s
Re: Reynolds number ($u_{av}D_h/\nu$)
u : velocity component in flow direction, m/s
ΔP: pressure gradient across the tube, N/m^2

Greek symbols

μ: dynamic viscosity of air, kg/ms
ν: kinematics viscosity of air, m^2/s
ρ: density of air, kg/m^3
α_v:The angle between the top (or bottom) of the channel and x axes
α_a : The angle between the sides of the channel and x axes
φ: cone angle, ^{0}
γ : a/b
δ : c/d

Subscript

a,b: the dimensions of the duct at the entrance
c,d: the dimensions of the duct at exit
v: measurement taken at vertical direction
a: measurement taken at axial direction

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