Pressure losses in transitions between square and rectangular ducts of the same cross-sectional area

E. I. Dekam and J. R. Calvert*

Experimental results are presented for the pressure loss in transitions between square and rectangular ducts where the two ends have the same cross-sectional area. The aspect ratios at the rectangular end ranged from 0.3 to 0.625, and the transition length from 1 to 2 times the hydraulic diameter. Reynolds numbers ranged from 50000 to 125000. The pressure drop may be divided into components arising from friction and velocity profile distortion. The friction component, which may be evaluated by normal pipe flow methods, accounts for the observed variation with Reynolds number. The velocity profile component increases as the aspect ratio of the rectangular end falls, and is significantly higher for rectangular to square than for square to rectangular transitions. There is an optimum length to hydraulic diameter ratio, for which the pressure loss is a minimum; it has not been found exactly, but is less than 2 and probably below 1

Keywords: *fluid flow, ducts, transitional flow*

Transition sections between ducts of different cross-sectional shapes are common in many fields of fluid dynamics. Examples include ventilation and air conditioning ducting, water and wind tunnels, aircraft engine intakes, power station intake and flue ducting, and pump and fan delivery pipework. Such transitions may be divided into five groups:

1. Transition diffusers, where the cross-sectional area increases and the mean velocity falls.
2. Transition contractions, where the cross-sectional area falls and the mean velocity rises.
3. Convergent-divergent and divergent-convergent transitions, where both accelerations and decelerations are involved and there may or may not be an overall change in cross-sectional area.
4. Sudden transitions, where there is an abrupt change of section; the cross-sectional area may or may not vary.
5. Transitions at constant cross-sectional area.

Published work on transition ducts is sparse; what there is mainly concerns[1] diffusers or, to a lesser extent, contractions, an emphasis arising largely from interest in wind tunnel components.

Our work is concerned with transitions between ducts of equal area, but different cross-sectional shape. While such transitions have frequently been used in the past, they do not seem to have been considered as separate components. Their losses have been lumped together with those of adjacent bends or changes of area (which have long been considered as separate), perhaps being classed as 'interference' effects, or considered as unspecified extra 'system' losses. Here, we separate the effects of the shape change from those of any change of area. For practical reasons, transition sections are usually constructed using straight line generators; frequently, indeed, they are made up of flat panels and conical or cylindrical surfaces. It has been shown previously[2] that such transitions usually have a maximum cross-sectional area at their longitudinal mid-point, which may be significantly larger than the end areas. Recent work, however, has shown[3] that there are transition geometries with straight line generators which have constant cross-sectional area. The sudden transitions (type 4 above), are of course a special case of this with zero overall length.

The experiments reported here are concerned with the pressure losses in transitions between square and rectangular ducts of varying length and aspect ratio, at several Reynolds numbers.

Theory

Duct geometry

Fig 1 shows the geometry of a square to rectangular transition. The area variation along the transition is

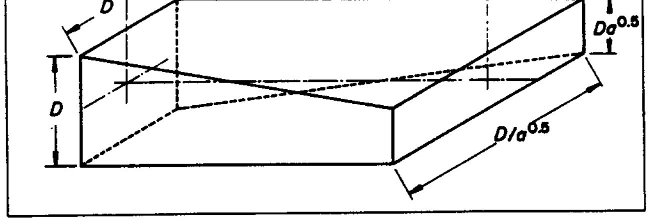

Fig 1 Transition between square and rectangular sections

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Received 4 September 1984 and accepted for publication on 14 January 1985

given[2] by:

$$A_x/A_1 = 1 + f(a)(x/L - (x/L)^2)$$

where:

$$f(a) = (1+a)/a^{0.5} - 2$$

The maximum area ratio is at $x/l = 0.5$, and is given by:

$$A_{max}/A_1 = A_2/A_1 = A_2/A_3 = 1 + f(a)/4$$

Duct flow

The flow may be treated as a diffusion from area A_1 to area A_2 followed by a contraction to area A_3 $(=A_1)$. In the diffusing half, the pressure will rise due to the diffusion, and fall due to losses. The net effect may be either a rise or a fall. In the contracting half, the pressure will fall due to both the contraction and losses. In both halves, there may be some effect arising from non-uniform velocity profiles.

Using the energy and continuity equations, the overall pressure loss coefficient C may be expressed as the sum of two parts (see Appendix):

$$C = C_v + C_f$$

where:

$$C_v = ((E_1/E_3)^2 - 1)/E_1^2$$

represents the pressure loss due to velocity profile distortion and:

$$C_f = (P_{0m1} - P_{0m3})/\tfrac{1}{2}\rho u_{av1}^2$$

represents the friction loss.

Apparatus and experiments

The experimental test sections were square to rectangular transitions made of perspex. The square ends were of side 158 mm. Three different aspect ratios were used at the rectangular ends: 0.3, 0.4 and 0.625, all the areas being 2.5×10^4 mm^2. Two different lengths were used, 158 mm ($L/D = 1$) and 316 mm ($L/D = 2$).

The test sections were mounted between parallel ducts 158 mm long, which in turn were attached to curved plywood entry/exit sections. The assemblies were fitted into a 310 mm square by 3 m long glass sided water channel. They could be fitted either way round, so that both rectangular to square and square to rectangular transitions could be investigated.

Mean velocities were obtained from a turbine flowmeter in the water channel pump circuit. The calibration of this instrument was confirmed from detailed velocity distributions across several of the test sections. Agreement was within 2%.

Total and static pressures were measured with pressure probes mounted on a traverse gear well downstream of the test section. Traverses were made in the approach and outlet ducts 52 mm upstream and 108 mm downstream of the working section respectively. Static pressures were also measured at wall tappings. Pressure differences were measured using a differential water manometer capable of a resolution of 0.25 mm.

A general arrangement of the apparatus is shown in Fig. 2.

Results

Static pressures were almost constant across the cross-section in all cases. The estimated maximum errors in the various measured quantities (static pressure, total pressure and flow rate) are all $\pm 1\%$ or less. The cumulative maximum errors in the derived quantities are estimated to be $\pm 3\%$, or less.

Effective area fraction

The inlet effective area fraction E_1 was approximately 0.8 for all experimental conditions. This implies a well

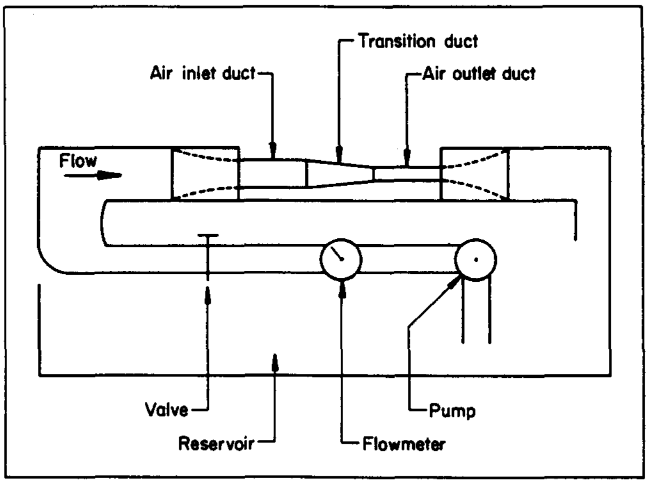

Fig 2 General arrangement of test rig

Notation

A	Cross-sectional area
A_{max}	Maximum cross-sectional area
a	Aspect ratio
C	Overall pressure loss coefficient
C_c	Pressure loss coefficient for contracting part
C_d	Pressure recovery coefficient for diverging part
C_f	Friction pressure loss coefficient
C_v	Velocity profile distortion pressure loss coefficient
D	Hydraulic diameter
E	Effective area fraction
L	Transition length
P_{0m}	Total pressure on streamline of maximum velocity
Re	Reynolds number
U	Maximum velocity
u_{av}	Average velocity
x	Axial position
ρ	Density

Subscripts

1	At transition entry
2	At transition centre
3	At transition exit
x	At position x

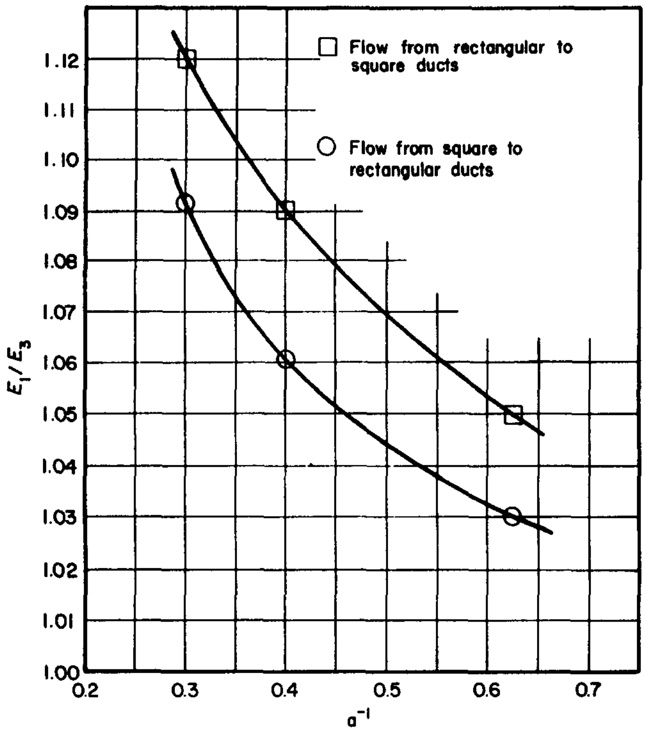

Fig. 3 Effect of aspect ratio on effective area fraction

developed, although probably not fully developed, turbulent flow at entry. It is equivalent to an inlet boundary layer displacement thickness of about 8 mm.

Fig 3 shows the outlet effective area fraction E_3 as a function of aspect ratio. C_v is almost independent of L/D, as shown below, and E_1 is the same so these results are valid for both duct lengths. The ratio E_1/E_3 falls as aspect ratio rises (the duct becomes more square), whichever the flow direction. These results imply an outlet boundary layer displacement thickness of about 9 mm for the square to rectangular direction, and about 11 mm for the rectangular to square. This increase is comparable to that which would be expected for the normal growth of a turbulent boundary layer in zero pressure gradient.

Effect of Reynolds number

Results of experiments at four different Reynolds numbers (based on mean velocity and hydraulic diameter) ranging from 50000 to 125000 are shown in Fig 4. There is a consistent fall in loss coefficient C as Reynolds number rises which is consistent with what is found in straight hydraulically smooth pipes. The maximum value of friction loss coefficient C_f was about 0.05 (for $a=0.3$, $L/D=2$, $Re=50000$) and the minimum about 0.02 ($a=0.625$, $L/D=1$, $Re=125000$). These correspond to pipe friction factors f of about 0.0057 and 0.0049 respectively and are close to the expected hydraulically smooth values.

Effect of duct length

The loss coefficient C is higher for the longer sections than the shorter. The differences arise almost completely from the frictional loss term C_f, which is approximately doubled. The velocity term C_v is virtually unchanged.

While these experiments show lower loss for a shorter duct, this obviously cannot be universally true. An abrupt transition (zero length) must have a high loss, and there must be an optimum length between $L/D=0$ and $L/D=2$. It has not been found in this work, but it probably lies below 1.

Effect of aspect ratio

The effects of aspect ratio are shown, for two different Reynolds numbers, in Figs 5 and 6.

There is a significant increase in pressure loss coefficient as the aspect ratio falls. This arises partly from the friction coefficient C_f, due to the increase in surface area, and partly from the velocity profile distortion. At lower aspect ratio there will be a larger deviation of the flow direction. On the diverging pair of sides this will lead to a more rapid boundary layer growth, which will probably more than cancel out the rather slower rate of growth on the converging pair.

Effect of direction of flow

In Figs 3 to 6 it is obvious that flow from the rectangular to the square end involves approximately 30% higher pressure loss than flow from square to rectangular. This difference must arise principally from the velocity profile distortion, since the surface areas, perimeters, etc, which mainly govern the friction loss, must be the same in the two cases.

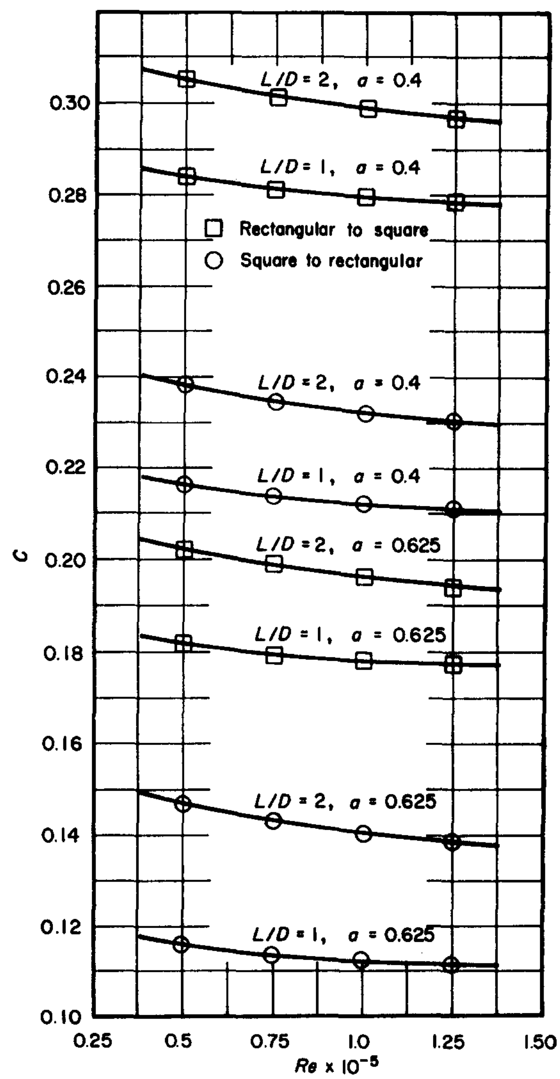

Fig 4 Effect of Reynolds number on pressure loss coefficient

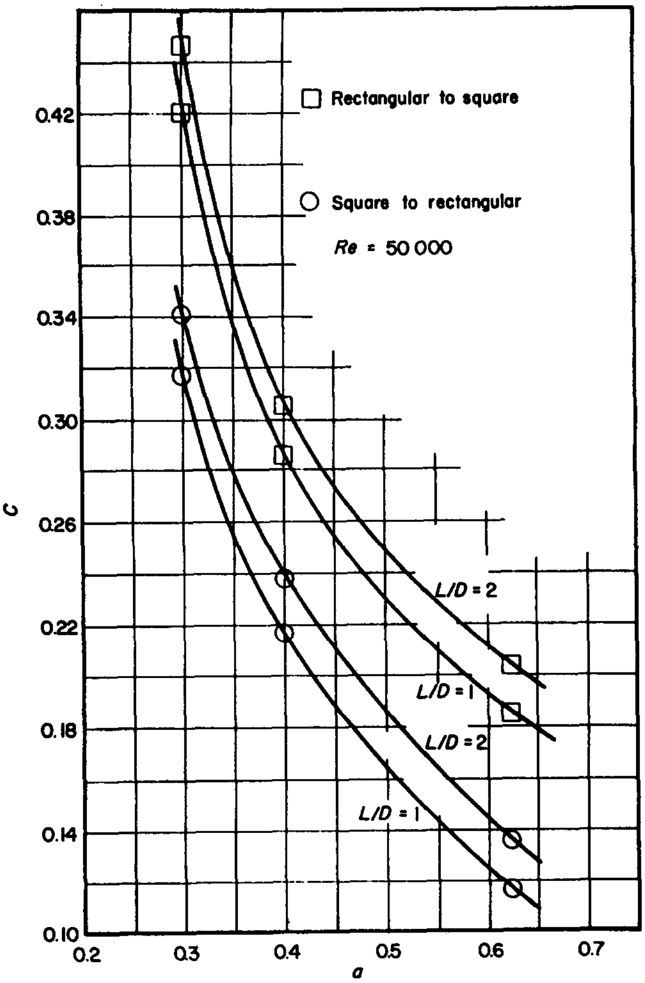

Fig 5 Effect of aspect ratio on pressure loss coefficient, Re=*50 000*

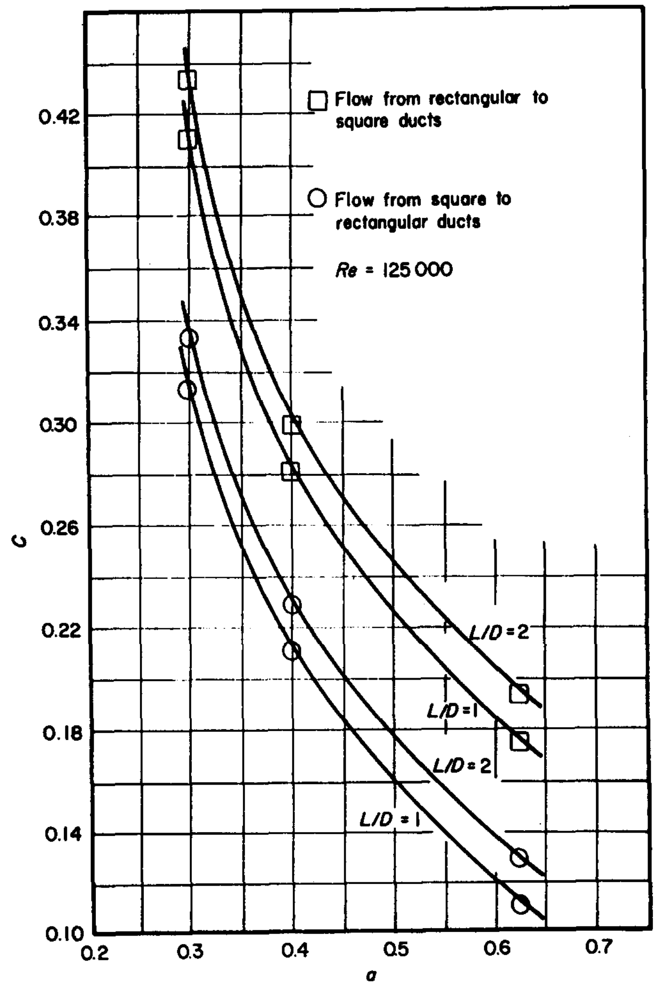

Fig. 6 Effect of aspect ratio on pressure loss coefficient, Re=*125 000*

Conclusions

The pressure loss in a transition between square and rectangular ducts of the same cross-sectional area may be divided into two components arising from friction and velocity profile distortion. Measurements of static pressure and total pressure on the centre line allow both these components to be obtained.

Over the range of Reynolds numbers (50 000 to 125 000) and geometries tested (aspect ratios 0.3 to 0.625 and length/width ratios of 1 to 2), the frictional component is that expected for a hydraulically smooth straight pipe. The higher pressure loss in longer ducts can be accounted for in terms of the additional skin friction alone.

The effect of reducing the aspect ratio of the rectangular end is to increase the pressure loss. This arises from both frictional and velocity profile effects.

Flow from a rectangular to a square section has significantly higher pressure loss than in the opposite direction. This must arise from velocity profile effects, but has not been fully explained. There must be an optimum length to width ratio, for which the pressure loss is minimum. This is at a ratio less than 2, but has not been located more precisely.

References

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A: Analysis of pressure loss

The geometry of the transition shows a divergence up to the mid-point, followed by a convergence[2]. For analysis of the pressure loss, these two parts may be treated separately.

Divergence

At any point on the streamline of maximum velocity, the total pressure is:

$$P_{0m} = P + \tfrac{1}{2}\rho U^2$$

Assuming that the static pressure is uniform across any cross-section, the actual pressure rise between any two sections is:

$$P_2 - P_1 = \tfrac{1}{2}\rho U_1^2(1-(U_2/U_1)^2) - (P_{0m1} - P_{0m2})$$

Basing a pressure recovery coefficient on the dynamic head corresponding to the average velocity at inlet, and noting that the effective area fraction is defined by:

$$E = u_{av}/U$$

at any section, the pressure recovery coefficient is:

$$C_d = \frac{(1-(E_1A_1/E_2A_2)^2)}{E_1^2} - \frac{(P_{0m1}-P_{0m2})}{\frac{1}{2}\rho u_{av1}^2}$$

Convergence

Using a similar procedure to that for the divergence gives a pressure drop coefficient for the converging part:

$$C_c = \frac{(1-(E_3A_3/E_2A_2)^2)}{E_3^2} - \frac{(P_{0m2}-P_{0m3})}{\frac{1}{2}\rho u_{av3}^2}$$

Since $A_1 = A_3$, and $u_{av1} = u_{av3}$ (by continuity), the overall pressure loss coefficient is:

$$\begin{aligned} C &= C_c - C_d \\ &= \frac{((E_1/E_3)^2-1)}{E_1^2} + \frac{(P_{0m1}-P_{0m3})}{\frac{1}{2}\rho u_{av1}^2} \\ &= C_v + C_f \end{aligned}$$

where C_v represents the effect of velocity profile distortion, and can be obtained from inlet and outlet velocity measurements, and C_f represents frictional loss, and can be obtained from total pressure measurements on the streamline of maximum total pressure.

BOOK REVIEW

Heat Transfer in Enclosures

Eds R. W. Douglas and A. F. Emery

This symposium volume is a collection of papers presented at two technical sessions at the 1984 Winter Annual Meeting of ASME. Both experimental (five papers) and analytical (eight papers) works are included covering a broad range of geometrics and practical motivations for the work. The first nine papers deal with cavities or containers of various shapes, while the remaining papers deal with annuli of one kind or another. Applications mentioned range from solar collectors to building heat transfer to electrostatic precipitator heat transfer. The session organizers believe that these papers, while certainly not an exhaustive compilation of current research, nonetheless offer a representative cross-section of the status of heat transfer in enclosures.

Contents

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Published, price $24, by ASME, 345 East 47th Street, New York, NY 10017, USA