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A DEVICE FOR MIXING HOT AND COLD AIR STREAMS IN SQUARE DUCTS

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SUMMARY. A device for evening out temperature maldistributions in air flows in ducts is described. Experiments have been performed on two models, one at Re$\leq 1.5\times10^5$ and the other at Re$=1.5\times10^6$. It is shown that small models can be effectively used to predict the performance of much bigger geometrically similar devices operating at significantly larger Reynolds numbers. It is also shown that the mixer has a lower pressure drop than other devices which have a similar thermal performance.

1 INTRODUCTION

While at first sight fluids at different temperatures should be quickly mixed in turbulent flow it is not uncommon for such streams to remain unmixed for long distances (Clayton, Ball and Spackman (1968)). It is the present author's experience that two air streams, one hot and the other cold, have not been mixed after passing through a centrifugal fan; the cold air flowed along the bottom of the horizontal outlet duct while the hot air flowed along the top of the same duct. It follows that if a uniform temperature distribution is required across the cross-section of a duct after two gas streams at different temperatures have been joined, or a large temperature maldistribution exists in a flowing fluid and long lengths of empty duct are not available for mixing to occur, some device for promoting mixing must be incorporated. This is particularly true of large ducts in which cost considerations and and space limitations impose severe constraints on duct lengths.

A small number of mixing devices are described in the literature, Tauscher and Streiff (1979), McKenzie (1973(a) and 1973(b) and ASHRAE Standards 33-74, 41-66 and 58-65. The available data is difficult to find and, except for proprietary lines, incomplete. In fact, the ASHRAE standards give no guidance as to how to determine the lengths of duct required after the mixer to achieve the desired degree of uniformity, nor is there any indication as to the pressure drops to be expected. It appears that the more effective the mixer (i.e the shorter the length of duct required for the desired degree of homogeneity to be achieved) the higher the pressure drop across the mixer (Tauscher and Streiff (1979)). Tauscher and Streiff's (1979) mixer for gas mixing in large ducts at high Reynolds numbers appears to have been developed from a mixer specifically designed for laminar flow (Tauscher and Schutz (1973)) and it is not suprising that they indicate that the pressure drops across their mixer are relatively high, but mixing can occur in very short duct lengths. Admittedly, the pressure drops in their device are lower than those obtained from an orifice which gives the same mixing. The mixers tested by Gibson (1973) also have unacceptably high pressure drops. Therefore there exist a need for a mixer which will correct temperature maldistributions in a short duct lengths with a smaller pressure drop across it than in the mixers mentioned above. Such a mixer is described in this paper.

2 BACKGROUND CONSIDERATIONS

No agreed method of expressing the performance of a mixing device exists. Tauscher and Schutz (1973) and Tauscher and Streiff (1979) used the relative standard deviation defined as

$$\sigma_r = \frac{\sigma}{\sigma_o}, \qquad (1)$$

as the parameter which indicates the homogeneity of the mixed steams, in which σ is the standard deviation of the temperature at any cross-section downstream of the mixer and σ_o is the standard deviation of the temperature upstream of the mixer. Unfortunately σ_r approaches zero as the distibution becomes more uniform. It is usual for a device to have a performance parameter which has a maximum of unity when the device is operating perfectly and a minimum of zero when it is not performing its function as occurs in the case of the efficiency of an engine. Thus following McKenzie (1973a) the mixer effectivness, defined as

$$\eta = 1 - \sigma_r, \qquad (2)$$

is the performance parameter used in this paper. In general a flow will be considered as homogeneous if $\eta \geq 0.99$.

There is no information in the literature on the principles which should be employed for the design of static mixers which are used for producing uniform temperatures in an air steam. The mixers discussed by Tauscher and Streiff (1979) were designed to achieve the desired temperature distribution at the outlet from the device itself with little further mixing downstream. As a result their devices lead to homogeneous temperature distributions in relatively short distances, but, as mentioned earlier, they have relatively high pressure drops. The information presented in their paper on the mixing downstream from the mixers can, however, be used to infer that the larger the "jets" issuing from the device the better the downstream mixing. Further, the pressure drop is lower in the device which produces the largest jets. Similar conclusions can be reached by examining Gibson's (1973) results (Gibson's work should not be accepted at face value as it contains a number of errors).

Since the particular device to be designed was intended to be used to produce a uniform temperature distribution in large power station ducts for the protection of a fabric filter, the pressure drop characteristics were almost as important as the mixing performance. It was decided to design a mixer with the largest possible jet structure, which at the same time would swirl the flow downstream from the mixer. The design developed is shown in figure 1 which is drawn for the half-scale model used to test the performance of the device. The mixer consists of four identical

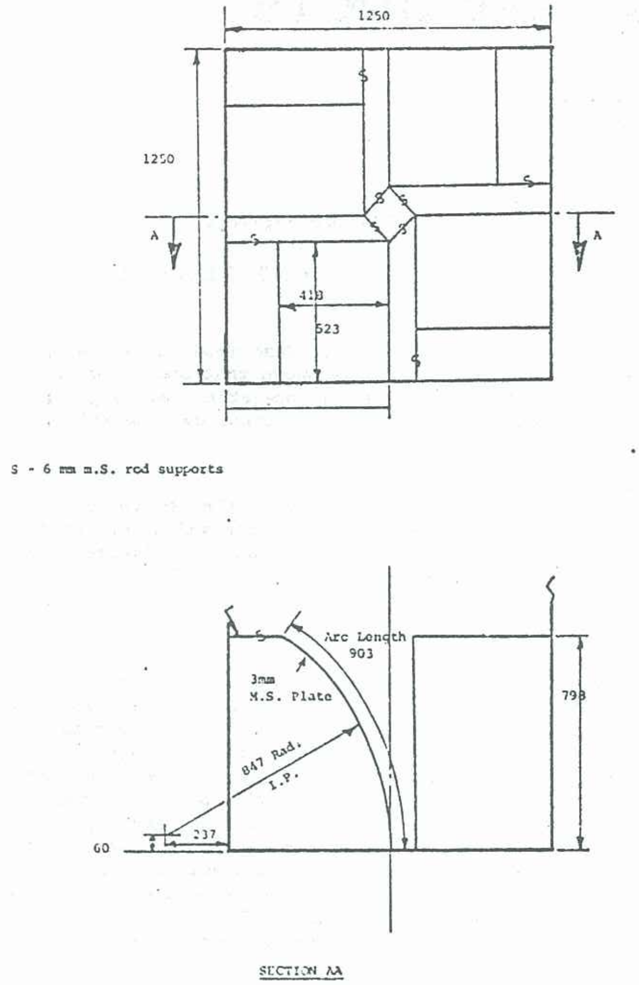

Figure 1. Mixer (looking downstream).

circular-arc plates each welded to one side of the duct and stabilised by rods to prevent vibrations.

3 TEST PROGRAM AND DESCRIPTION OF APPARATUS

An eighth-scale model of the mixer was built and tested in the rig designed by Gibson (1973) (figure 2). This consists of a long 305mm square duct which was modified by the addition of two extra temperature measuring stations downstream of the of the mixer. A temperature maldistribution was produced by a 2.4kW electric heater placed in one corner of the duct. The temperature profiles were monitored downstream of the heater and of the mixing device by twelve thermocouples at each of the stations indicated in figure 2. In the first thermocouple grid the four thermocouples which could "see" the heater were radiation shielded so as to prevent an erronous temperature being measured. The number of thermocouples is small but a larger number would have led to a partial blockage of the duct and provided extra mixing capacity. The heat input from the heater was calculated from the thermocouple readings and the velocity distribution at the second thermocouple grid downstream of the heater (G2 in figure 2). This calculated heat input differed by approximately seven percent from that obtained from the electrical energy supplied to the heater. Thus the number of thermocouples was adequate for the present purpose. The calibration of the standard nozzles was checked by integrating the velocity profile to obtain the volumetric flow rate and was found to differ by two percent from the values obtained from the pressure drop

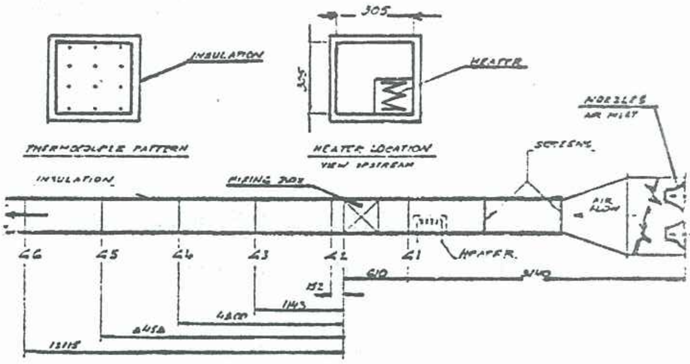

Figure 2. Schematic diagram of small test rig.

across the nozzle.

For comparison purposes a "louver" type mixer consisting of four horizontal rows of vertical plates which alternately directed the flow to the right and left side of the duct was also tested. This device is one half of the M3 mixer tested by McKenzie (1973a) and was selected as the pressure drop was expected to be about the same as that for the new design.

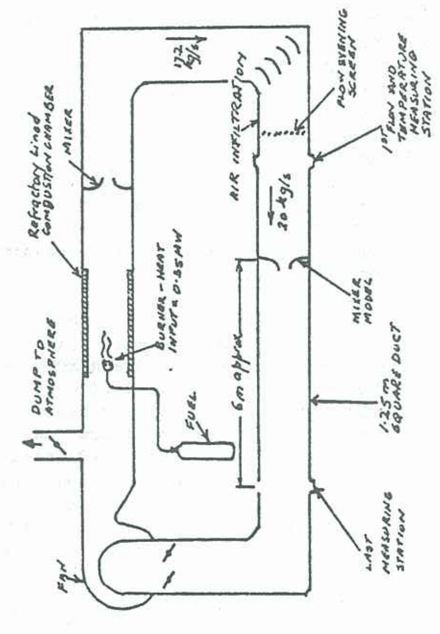

Figure 3. Schematic arrangement of large test rig.

The half scale model of the new mixer (figure 1) was tested in the apparatus shown schematically in figure 3. This consisted of a 1250mm x 1250mm duct in which the circulating air was heated to 150°C by a 350kW propane heater. The temperature distributions were measured by four thermocouple grids, in locations G1 to G4 on figure 3. Each thermocouple grid consisted of forty nine thermocouples distributed uniformly on the cross-section of the duct. The temperature maldistibution was obtained by admitting air from the atmosphere through six vertically arranged 150mm diameter holes two metres upstream of thermocouple grid G1. The quantity of hot air rejected to the atmosphere through the dump duct equalled the quantity of the infiltation air, thus permitting the propane burners to have sufficient oxygen to operate continuously and resulting in a steady flow in the duct system. The volumetric flow rate was determined from velocity traverses approximately four metres downstream from the mixer.

It was impracticable to completely calibrate each of the thermocouples in each rig as there was a total of sixty thermocouples in the small rig and 196 in the large rig. In both cases each thermocouple was calibrated at room temperature against a calibrated mercury in glass thermometer and one thermocouple in each rig was completely calibrated. Since the error was never larger than $1^{o}C$ no further calibration was necessary.

4 RESULTS AND DISCUSSION

The prototype mixer was expected to operate in the Reynolds number range $8x10^{5} < Re < 1.2x10^{6}$, however, the maximum value of the Reynolds number which could be achieved in the small rig was only of the order of $1.5x10^{5}$, approximately one tenth of the maximum value required on the prototype, whilst the Reynolds number in the larger model was $1.5x10^{6}$, approximately twenty percent greater than the desired value. This is most unusual since normally the difficulty, when testing models of large ducts, is that the model Reynolds number is much less than the Reynolds number of the prototype. No guide lines have been found in the literature on how to deal with this problem. However, it can be argued from the Colbrook-White equation (Massey 1968) that if the flow in a duct of constant cross-section is rough turbulent, $Re > 10^{5}$ say, any Reynolds number increase beyond this value will not affect the friction factor significantly. In fact, the pressure drop coefficient defined as,

$$K = \frac{\Delta p}{\frac{1}{2}\rho v^{2}}. \qquad (3)$$

in which Δp is the pressure drop, ρ is the density and v is the average velocity, is likely to be lower in the prototype than in the model. From his previous experience with modelling obstructions in ducts, the author believes that the pressure drop coefficient caused by them is also generally lower for the large prototype than the small model. The tests described in this paper provide an oportunity for testing this hypothesis. The pressure drop coefficient of the small model was found to be K=1.5 whereas for the large model K=1.4. When the small model was tested at $Re=10^{5}$ the pressure drop coefficient was 1.55, the same as at $Re=1.5x10^{5}$ within the experimental accuracy of the tests. However at $Re=5x10^{4}$ the pressure drop coefficient increased to 1.7, an increase well outside the experimental error. It follows that the usual procedure of testing a model at a lower Reynolds number than the prototype is conservative as far as pressure losses are concerned. Care should, however, be taken that the Reynolds number is sufficiently high, otherwise a gross overestimate may result.

The pressure drop coefficient in the louver type mixer at $Re=1.5x10^{5}$ was 2.1 a value forty percent greater than K=1.5 obtained with the new mixer in the same rig. Since the recommended configuration of the complete mixer consists of two such mixers in series in which one is turned through ninety degrees relative to the other (McKenzie (1973a)), the pressure drop is very high without the benefit of obtaining a uniform temperature distribution in a short distance downstream of the mixer (McKenzie (1973b)).

Since the turbulence level is increased with an increase in Reynolds number, it was expected that the larger Reynolds number in the prototype would lead to a slightly more uniform temperature distribution downstream of the mixer than that obtained in the model. This hypothesis was initially tested in the small rig by carrying out two further test, one at a Reynolds number of 10^{5} and the other at $5x10^{4}$. Although these test could not be conclusive, the results would indicate whether variations in Reynolds number had a significant effect.

4.1 Small Rig Temperature Results

The temperature distribution just upstream of the mixer is shown in figure 4. Since the heater occupies only the bottom right hand side of the duct (when looking downstream), the majority of the air just downstream of the heater (i.e. just upstream of the mixer see figure 1) is at room temperature. This distribution is reasonably difficult to even out as may be seen from the bare duct results in figure 5. Homogeneity has not been achieved by forty hydraulic diameters downstream of the first temperature measuring grid which confirms the observations of Clayton, Ball and Specman (1968) that turbulent mixing may require a large distance to complete.

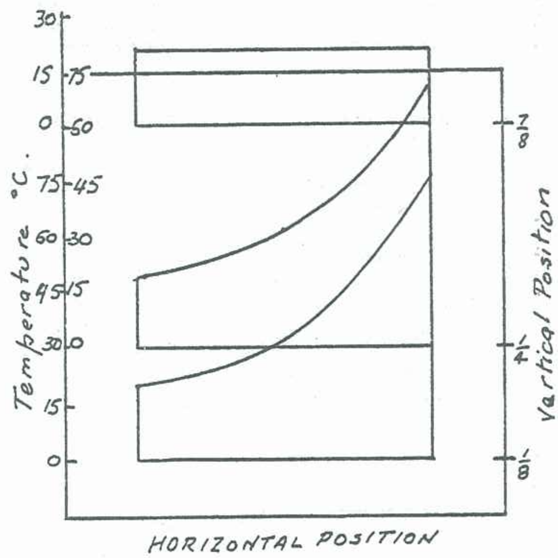

Figure 4. Temperature distribution upstream of the mixer in the small rig (looking downstream).

The louver mixer does promote mixing in that a uniform temperature distribution is achieved twenty hyrdaulic diameters downstream of the mixer. This is a quite unrealistic distance in terms of very large ducts. The temperature distribution which was used was particularly unfavourable for this type of mixer since the way in which it was used promoted side-to-side mixing only. Whilst the addition of the other half of the mixer (McKenzie (1973a)) does indeed improve the performance of the mixer as now only about ten hydraulic diameters are required for homogeneity, this is still too large for practical use. This, together with the high pressure drop (K=4.9), makes this type of mixer of doubtful value as a means of obtaining uniform temperature distributions in ducts.

As may be seen in Figure 5, there is a Reynolds number effect in that the performance of the mixer is significantly lower at $Re=5x10^{4}$ than at $Re=1.5x10^{5}$. The difference between the perfomance at $Re=10^{5}$ and at $Re=1.5x10^{5}$ is very much less, but, nevertheless noticeable in terms of thermal performance (figure 5).

4.2 Large Rig Temperature Results

The distribution of temperature just upstream of the mixer in the large rig may be seen in figure 6. The fact that cold air was infiltated from the atmosphere on the right hand side (looking downstream) is readily apparent. The increase in Reynolds number has not

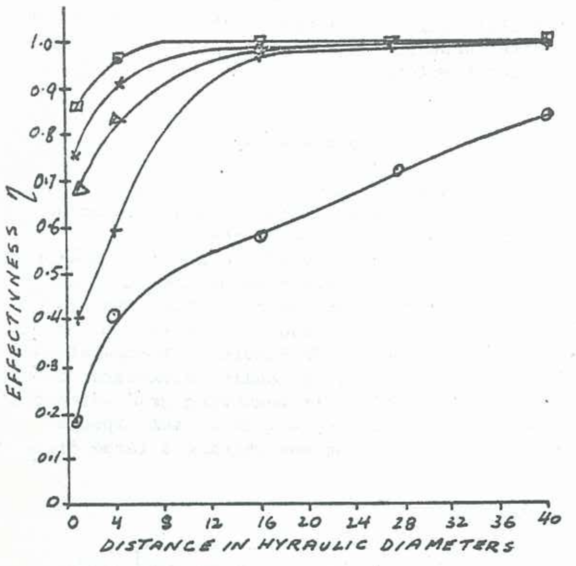

Figure 5. Temperature distribution development in the small rig. ⊙ bare duct ⊡ new mixer $Re=1.5\times10^5$, × new mixer $Re=10^5$, △ new mixer $Re=5\times10^4$, + louver type mixer $Re=1.5\times10^5$.

altered the turbulent mixing to any large extent as may be seen from a comparison of figures 5 and 7 for the bare duct results. Whilst it is true that the mixing is slightly better in the large rig than in the small rig, the difference is not sufficiently large to permit the use of a bare duct as a mixing agent even at very high values of Reynolds number.

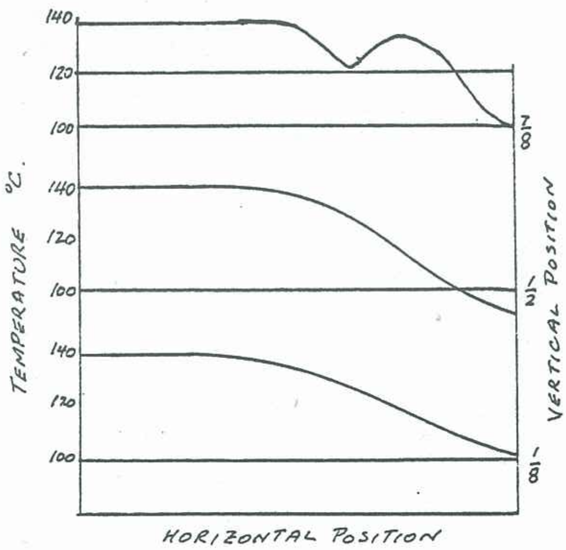

Figure 6. Temperature distribution upstream of the mixer in the large rig (looking downstream).

The results for the development of the temperature profile in the large rig for the new mixer (figure 7) are different at short distances downstream of the mixer from those obtained on the model at $Re=1.5\times10^5$ (figure 5). At distances greater than two hydraulic diameters downstream of the mixer the results in figure 5 at $Re=1.5\times10^5$ and those in figure 7 are in excellent agreement. The disparity near the mixer is most probably due to the very different inlet

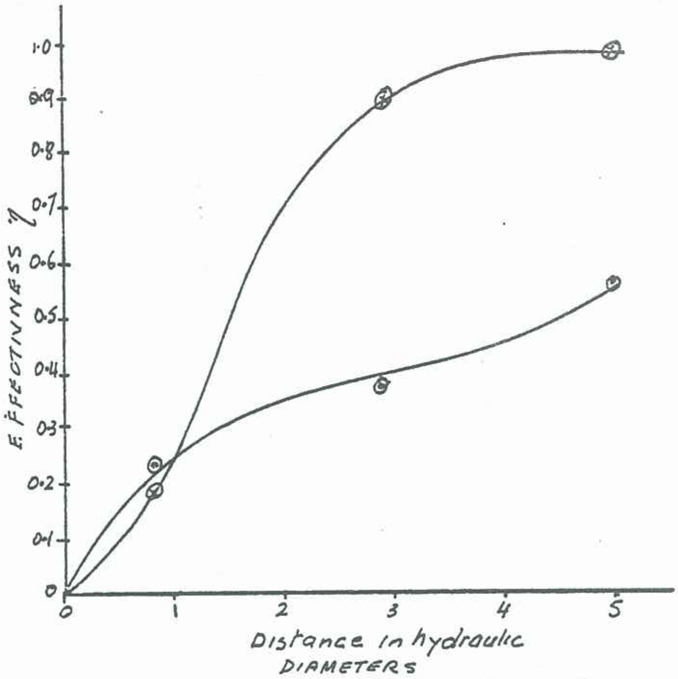

Figure 7. Temperature distribution development in the large rig $Re=1.5\times10^6$. ⊙ Bare duct, ⊗ New mixer.

temperature profiles used in the two cases, compare figures 4 and 6.

It may be concluded from figures 5 and 7 that at distances greater than two hydraulic diameters downstream of the mixer, the uniformity of the temperature distribution is independent of the inlet temperature profile and that a uniform temperature distribution is achieved in about five hudraulic diameters with a pressure drop coefficient of 1.4. The device described by Tausher and Streiff (1979) for a similar thermal performance has a pressure drop coefficient of aproximately 3 and is much more complex and expensive.

5 ACKNOWLEDGMENTS

James Howden (Aust) Pty. Ltd. is hereby thanked for permission to use the results obtained in the large rig.

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