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Flow Maldistribution in Industrial Air Heaters and its Effect on Heat Transfer

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Flow maldistribution in cross-flow heat exchangers used for industrial air heaters or recuperators is fairly common and can lead to a number of undesirable effects, one of which is deterioration in the thermal performance. The effect of gross flow maldistribution and the improvement in the thermal performance of flow correction guide vanes is investigated using a laboratory heat exchanger apparatus. The thermal performance deterioration from maldistribution was up to around 30 %. Modest improvements in the overall heat transfer coefficient and heat exchanger effectiveness when vanes were used was found for the range of air-side Reynolds numbers tested. Vanes were found to reduce the deterioration in thermal performance by up to 60 % compared to a perfectly uniform flow situation. Based on the ε-NTU correlations the thermal performance deterioration is predicted to decrease as the number of tube rows (i.e. NTU) increases. The effective heat transfer area and therefore thermal performance of industrial air heaters and recuperators can be improved by using flow correction devices, such as guide vanes.

1. Introduction

Industrial air heaters and recuperators are used in numerous industrial situations, such as for dryers and boiler air preheaters, to heat or recover heat from air or gaseous streams. Frequently, due to building space constraints, there are sudden and severe duct transitions directly before the heat exchanger that causes the air flow to become severely maldistributed and unsteady (Hoffmann-Vocke et al., 2011). Several remedies are used to correct or mitigate air-side maldistribution including boundary layer suction and injection (Freeman and Thompson, 1976), spreaders and guide vanes (Turek et al., 2012), perforated plates (Bury and Składzień, 2013), and swirl/vortex generators (Hájek et al., 2005;). Idelchick (1986) has an extended treatment regarding the relative merits of each method to improve diffuser performance. While these solutions can improve air-side distribution and reduce pressure drop, the effect of these methods on the thermal performance have not been as widely investigated as the effect on hydraulic performance (Zhang, 2009). Thermal performance has been shown to decrease with increased maldistribution (Yaïci et al., 2013), although the severity and velocity profile of the air flow is a factor in the deterioration of the thermal performance (Chin, 2013).
Improvements in thermal performance are important to understand especially in heat recovery exchangers so that the effectiveness of the heat exchanger (and therefore heat recovery) can be maximised. An experimental investigation of the thermal performance improvement of a plate-fin-and tube type heat exchanger will be reported. Implications of air-side flow maldistribution in industrial applications, especially with respect to thermal performance will also be briefly discussed.

2. Methodology

A purpose built laboratory scale industrial air heater system was used and a schematic of the system is shown in Figure 1. A backward-curved centrifugal fan controlled with a variable speed drive is used to supply air to the system and control air flow rate. A calming length is used to ensure fully developed flow before the diffuser. A plate-fin-and-tube type heat exchanger is used with face dimensions of 0.900 m by

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0.855 m. Hot water at 70 °C is circulated through the air heater by means of a fixed speed pump. Inlet and outlet temperatures of the air and hot water streams were measured and logged using Class 2 T-type thermocouples (±1 °C). The bulk air flow rate was calculated by measuring the pressure drop (using a differential pressure transducer, ±4 Pa) across a straw honeycomb screen placed in the calming length of the inlet duct. The correlation between flow and pressure drop across the screen was known.

Table 1: Heat exchanger specifications.

Parameter	
Fin Type	Continuous Plate – Wavy Edges
Fin Spacing / Fin Thickness	310 per meter / 0.0002 m
Fin / Tube Material	Aluminium / Copper
Tube Arrangement	Staggered 30° Pitch
Tube Transverse / Longitudinal Spacing	0.003175 m / 0.028 m
Tube Diameter (outside) / Wall Thickness	0.0133 m / 0.001m
Heat Exchanger Dimensions	0.900 x 0.855 x 0.110 m
Fin Area / Total Heat Transfer Area	0.921
Flow Passage Hydraulic Diameter ($D_{h,fin}$)	0.00382 m
Free Flow Area / Frontal Area (σ_{air})	0.539
Heat Transfer Area / Total Volume	565 m^2/m^3

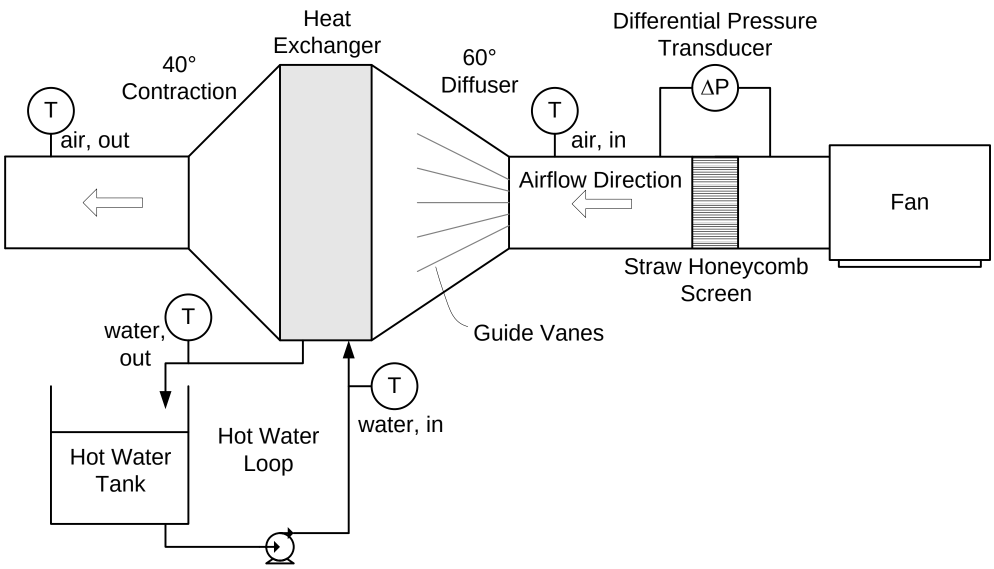

Figure 1: Schematic of the experiment heat exchanger rig (not to scale)

2.1 Heat transfer analysis

For this study the change in performance of the heat exchanger is quantified by either a change in effectiveness or a change in overall heat transfer coefficient. Both methods are valid because the water side conditions (i.e. $T_{h,i}$ and C_h) were held constant for all experimental tests, thus the water side film heat transfer coefficient is constant. Alternatively, the change in performance could have been expressed in terms of the Colburn *J*-factor although these results have not been presented in this paper. To reconcile the measured data (i.e. $Q_h = Q_c$) the air outlet temperature was calculated based on an energy balance. The energy balance error based on calculating the hot side and cold side duty independently (Q_h and Q_c) using all the measured parameters was within 5 %.

The air-side Reynolds number (Re_{air}) was calculated using Eq(1) where ρ is the air density [kg/m^3], G_{air} is the mass velocity of the air [$kg/m^2.s$], D_h is the air-side hydraulic diameter [m], and μ is the dynamic viscosity [Pa.s]. The mass velocity is calculated using Eq(2) where m_{air} is the mass flow rate of air [kg/s], A_{fr} is the frontal area of the air-side of the exchanger [m^2], and σ_{air} is the ratio of the free flow area to the frontal area. Heat exchanger effectiveness (ε) is calculated using Eq(3) where Q and Q_{max} is actual and maximum heat transfer respectively [kW], C and T is the heat capacity flow rate [kW/°C] and temperature

[°C] and subscripts min, max, *c*, *h*, *i*, and *o* indicate minimum, maximum, cold, hot and inlet and outlet respectively.

$$Re_{air} = \frac{\rho G_{air} D_h}{\mu} \tag{1}$$

$$G_{air} = \frac{m_{air}}{A_{fr}\sigma_{air}} \tag{2}$$

$$\varepsilon = \frac{Q}{Q_{\max}} = \frac{C_h(T_{h,i} - T_{h,o})}{C_{\min}(T_{h,i} - T_{c,i})} = \frac{C_c(T_{c,o} - T_{c,i})}{C_{\min}(T_{h,i} - T_{c,i})} \tag{3}$$

For the cross-flow configuration where both fluids are unmixed the *ε*-NTU relationship is given by Eq(4) and the NTU value has to be found implicitly. The term *C** in Eq(4) is known as the capacity ratio and is calculated by Eq(5).

$$\varepsilon = 1 - \exp\left[\frac{NTU^{0.22}}{C^*}\left[\exp\left(-C^* NTU^{0.78}\right) - 1\right]\right] \tag{4}$$

$$C^* = \frac{C_{\min}}{C_{\max}} \tag{5}$$

From the definition of NTU the overall heat transfer coefficient, *U*, [W/m^2°C] can then be calculated. The deterioration in the performance of the heat exchanger due to flow maldistribution can be expressed as a exchanger thermal performance deterioration factor (τ) as in Eq(6) proposed by Chiou (1978), where ε_{uni} and ε_{mal} is the effectiveness of the uniform and maldistributed situation respectively.

$$\tau = \frac{\varepsilon_{uni} - \varepsilon_{mal}}{\varepsilon_{uni}} \tag{6}$$

3. Results and discussion

3.1 Velocity profile

The local air velocity (*u*) directly in front of the heat exchanger at the end of the diffuser was measured using a thermal anemometer (TSI 8386-M-GB). A total of 7 points vertically were measured at each flow rate, with and without vanes. The local velocities were then normalised by the average bulk velocity (u_{bulk}), thus a perfect uniform velocity profile would be equal to unity. Figure 2 shows normalised velocity profiles for two different flow conditions, both with and without vanes taken along the centre of the heat exchanger face along line AB (*z* or vertical direction) as shown in Figure 2. As clearly shown in Figure 2 there is acute maldistribution occurring when there is no flow correction, with a large jet in the centre diffuser. The local velocities are over seven times greater than the average bulk velocity at the highest Reynolds number and almost five times at the lowest Reynolds number tested. When vanes are introduced there is still some degree of maldistribution, however the severity is greatly reduced and the maximum normalised velocity is only around two times the average bulk velocity. It should be remembered that maldistribution in these situations is a three dimensional phenomenon and the flow can be highly unsteady, although the flow regime is dependent on the diffuser geometry and flow rate (Freeman and Thompson, 1974).

3.2 Heat transfer coefficient and heat exchanger effectiveness

The overall heat transfer coefficient was calculated for three situations: i) no vanes; ii) vanes; and iii) uniform velocity profile. An average of the correlations found in Figure 10-91 and 10-92 from Kays and London (1998) for similar geometry plate-finned heat exchangers was used to calculate the overall heat transfer coefficient for the uniform velocity profile case (U_{uni}). The air-side film heat transfer coefficient was corrected for the number of tube rows by using the correction factor found in Figure 7-7 of Kays and London. The heat exchanger effectiveness was also calculated for these three situations. Figure 3 shows the overall heat transfer coefficient and the heat exchanger effectiveness for the three situations for a range of air-side Reynolds numbers. There was a slight improvement in both heat transfer coefficient and effectiveness for all Reynolds numbers tested when vanes were used for flow correction. The uniform situation was still predicted to have the highest heat transfer coefficient and effectiveness.

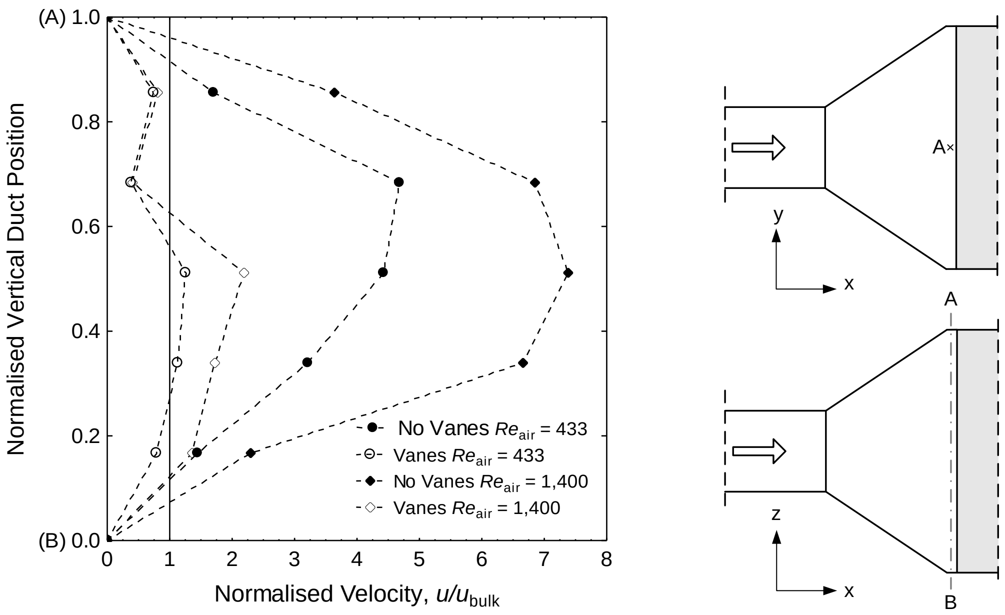

Figure 2: Normalised velocity profiles at the end of the diffuser directly prior to the heat exchanger along vertical line AB with and without flow correction vanes at two different air-side Reynolds numbers

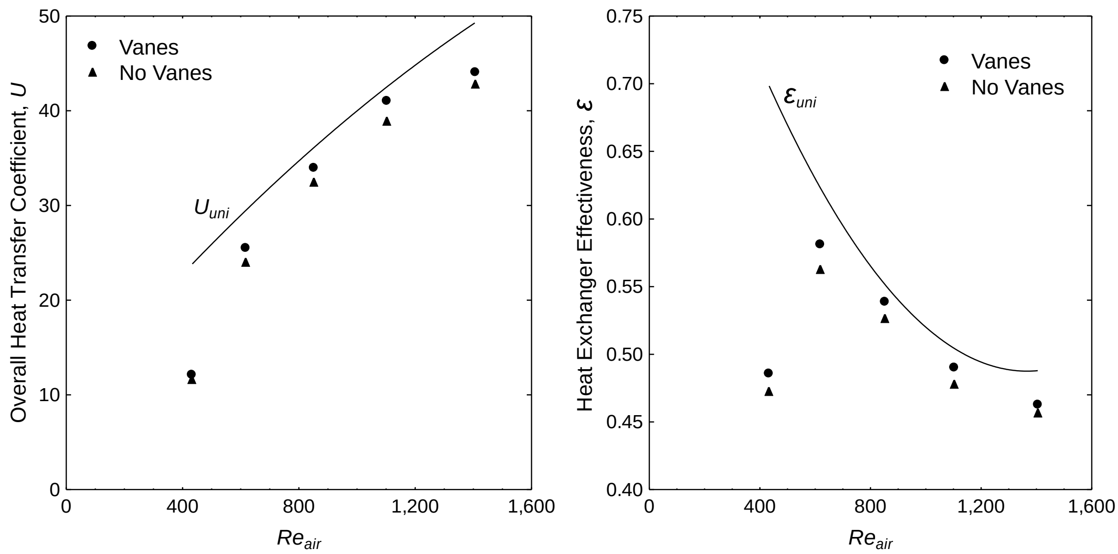

Figure 3: The overall heat transfer coefficient (left) and the heat exchanger effectiveness (right) for a range of air-side Reynolds numbers with vanes, without vanes, and the theoretical uniform case (solid line)

The single set of vanes used in the laboratory tests were not optimised for the geometry, leaving a degree of flow maldistribution across the heat exchanger, although the severity of the maldistribution was greatly reduced, as shown previously in Figure 2. Due to the scale of the laboratory heat exchanger, wall effects still have a significant impact on the flow profile and therefore heat transfer, with these effects more pronounced on a laboratory scale, while on larger industrial scale units these effects become relatively insignificant.

The percentage improvement in both heat transfer coefficient and heat exchanger effectiveness is shown on the left in Figure 4 for the range of air-side Reynolds numbers. The percentage increase was greater for the heat transfer coefficient and the trend in both heat transfer coefficient and effectiveness was a decrease as Reynolds number increased. The thermal performance deterioration factor is shown in the right of Figure 4 and was calculated using Eq(6) and the uniform effectiveness shown in Figure 3.

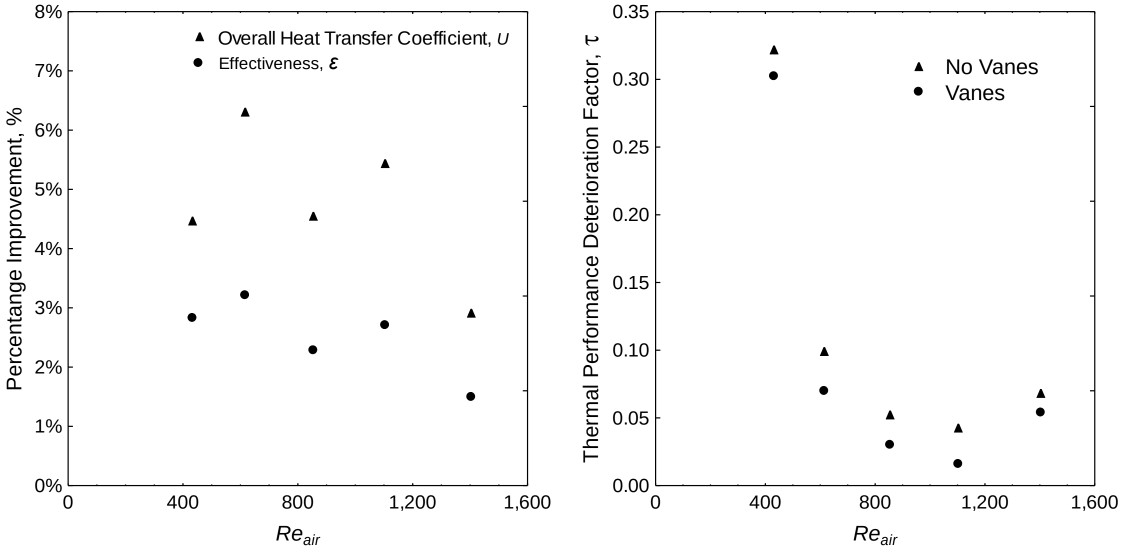

Figure 4: The percentage improvement in overall heat transfer coefficient and effectiveness by the addition of vanes (left) and the thermal performance deterioration factor with and without vanes (right) for a range of air-side Reynolds numbers

One of the goals of flow correction devices such as guide vanes is to reduce the deterioration in thermal performance and one method to determine the effectiveness of a design or to compare multiple designs or devices is to examine the percentage reduction in the thermal performance deterioration factor. With the measure a "perfectly" designed device would yield a 100 percent reduction in the deterioration factor. For the vanes tested in this study the deterioration factor was reduced by up to around 60 %, as shown on the left in Figure 5. Using the ε-NTU formulas, the uniform and measured maldistributed overall heat transfer coefficients (Figure 3), the thermal performance deterioration factor can be estimated as a function of the number of tube rows. The heat exchanger used in these tests had four tube rows. The predicted thermal performance deterioration factor for different tube rows is shown on the right in Figure 5 for the range of air-side Reynolds numbers tested. In all cases the deterioration factor reduced dramatically as the number of tube rows increased.

4. Industrial observations and implications

Gross air-side flow maldistribution is a common occurrence in industrial air heaters and recuperators, primarily due to large diffuser angles (40-90 degrees included angle) being used to reduce the physical space of duct transitions and poor upstream duct design. Incident flow profiles can also be heavily influenced by highly skewed fan exit profiles, with ranges in velocity of over an order of magnitude commonly encountered. The heat exchanger used in the above laboratory tests was effectively a mixed flow configuration on the water side due to the multiple passes horizontally across the duct itself. As such this is not a true reflection of a typical industrial air heater, where the liquid/condensing vapour side of the heat exchanger is typically configured with single flow passes across the duct. Therefore in a typical industrial heater the impact of air-side flow maldistribution would be anticipated to be greater than these laboratory results outlined above. Numerous industrial case studies have been found to reinforce this point and have led to ongoing research into this phenomenon.

5. Conclusions

A flow correction device in the form of guide vanes was used to improve the thermal performance of a cross-flow plate-finned heat exchanger with severe air-side flow maldistribution. The thermal performance of the heat exchanger deteriorated by over 30% at the worst case due to air-side flow maldistribution. Improvements in the overall heat transfer coefficient and heat exchanger effectiveness by up to around 6.5 % and 3.2 % was achieved with the use of guide vanes for flow correction. The effective heat transfer area of recuperators can be improved by the use of flow correction devices to minimise air-side flow maldistribution and improve heat recovery. The thermal performance deterioration factor is predicted to decrease as the number of tube rows (and therefore NTU) increases.

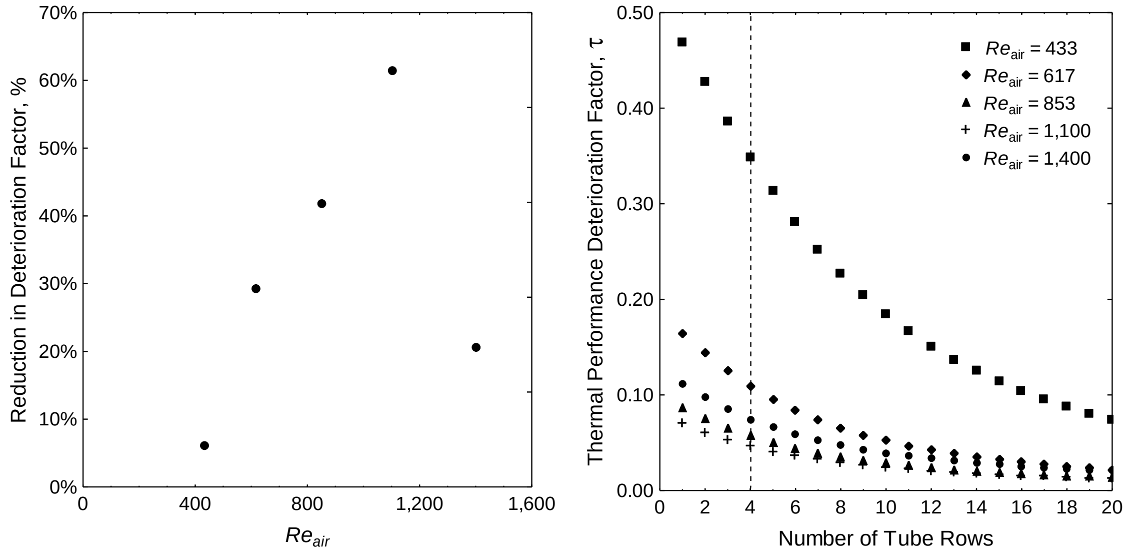

Figure 5: Percentage reduction in the thermal performance deterioration factor with the addition of vanes for a range of air-side Reynolds numbers (left) and the predicted effect of the number of tube rows on the thermal performance deterioration factor for each air-side Reynolds number tested (right)

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