A COMPARISON OF HEAT TRANSFER AROUND A SINGLE SERRATED FINNED TUBE AND A PLAIN FINNED TUBE

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ABSTRACT

With comparable bundle geometry, serrated fin finned tube heat exchangers transfer more heat than plain finned ones. There are believed to be many factors in this behaviour, and this investigation used CFD to examine the heat transfer and fluid flow patterns around the fins to try to understand the magnitude of the differences in the processes. The study is a first step in developing a new theory-based method to predict the performance of serrated fin tube bundles.

Keywords: *serrated fins, CFD, heat recovery, furnaces, finned tube, air cooling, performance prediction*

1. INTRODUCTION

Serrated, or cut fins, are commonly used in heat recovery applications and are generally regarded as having a higher heat transfer coefficient than plain fins. They are also credited with inducing a higher pressure drop across a tube bundle.

This study is a first step into the characterisation of the flows around serrated finned tube in comparison to plain finned tube, investigating the heat flow through the fin, the heat transferred and the magnitude and effects of turbulence generated by the discontinuous fin shape. This is achieved using Computational Fluid Dynamics (CFD) to compare the flow and heat patterns using a like-for-like comparison at prescribed gas velocities.

The finned tube considered in this study is based on the geometry of an industrial tube bundle as used in furnace heat recovery applications. The fin type is a partially cut serrated type; there is a plain portion of fin and then the serrated portion above it on the fin radius.

To minimise computational requirements, and to provide a definite basis for comparison without complication from the row effect, only one tube row was modeled.

2. CFD MODELS

2.1 Bundle geometry

To reduce cycle time and minimise computer resources a modeling approach previously designed by the author [1] was used. This maximises symmetry planes and meant that only a small slice of a heat exchanger need to be represented.

The model's mesh was separated into four main blocks for both plain fin and serrated fin models; an entry, a tube section, an initial outlet and the final outlet to allow for wake renormalization. The layout of the model domain is shown in Figures 1 and 2. Table 1 describes the tube and fin geometry of the plain fin design.

The serrated geometry was the same as the plain fin described in Table 1, but with the additional data of Table 2 describing the blade geometry.

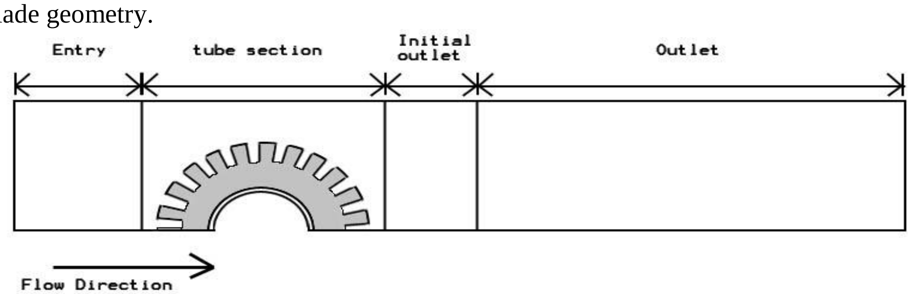

Figure 1: Layout of the CFD meshes (serrated fin shown, not to scale)

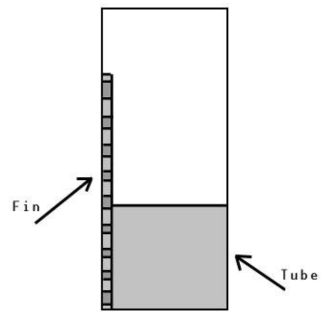

Figure 2: The gasflow approach view to the model (not to scale)

Dimensions	mm
D_f	88.9
D_o	50.07
D_i	48.07
s_f	1.2
n_f	239

Table 1: Dimensions of plain finned tube

Dimensions	mm
$\mathbf{H_{sf}}$	**6.62**
$\mathbf{W_b}$	**4.7**
$\mathbf{n_b}$	**41**

Table 2: Additional dimensions of serrated finned tube

2.2 Solver conditions

The default air physical properties in the CFD package are single values, and hence are fixed regardless of temperature. Due to the temperature variations encountered in the simulation a set of fifth order temperature dependant polynomials were used. These allowed variation in density, thermal conductivity, viscosity and specific heat capacity at constant pressure.

The k-ε Realizable turbulence model was used as per the author's previous experience. The near-wall model was used to ensure that the turbulent regions around the serrated fin's edges were computed using a turbulence model rather than using a laminar wall function approach, as is default.

Second order methods of determining the pressure-velocity relation and pressure interpolation were used, as the mesh in some regions was not well aligned with the bulk flow direction, and this technique was found to aid the resolution of the values in the non-aligned cells.

It had been found previously that by modifying the turbulent Prandtl number the heat transfer results were a better match for the experimental data; therefore the same modification was adopted for this study.

2.3 Boundary conditions

The ranges of velocities used in the simulations were 1-5m/s in 1 m/s increments. These are typical gas velocity values for 2 inch tube diameter heat recovery bundles used in heat recovery and economizer operations.

In heat recovery applications the gasflow over the tube should be a hot stream. As the difference in heat transfer performance is of interest the typical operation was reversed to a use a previously developed approach for modeling isothermally condensing air-coolers, as in like for like comparison this would be acceptable for examining the differences between the fin types. This also allowed the wide use of symmetry planes in the model, rather than accounting for cooling tubeside fluid. Hence the approach airflow temperature was set at 293^oK and the tube's inner wall set to 373^oK.

3. SIMULATION RESULTS

As the fins cannot be considered in isolation, which is away from the tube section, the results were obtained in the form of mass-weighted average values on planes running tangential to the fin surface, as shown in Figure 3. These values are therefore more representative of the heat exchangers full performance than considering the fin on its own.

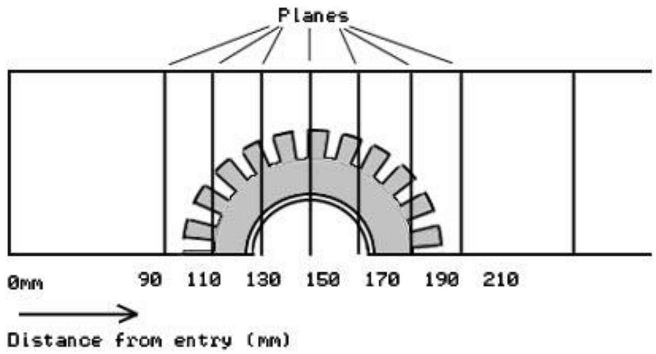

Figure 3: Position of the planes used for analysis values

3.1 Air temperature rise and pressure

The overall air temperature rise and pressure drop of the bundles is shown in Table 3. It can be seen that the pressure drop increases more rapidly for the serrated fins, whereas the difference in air temperature is nearly constant. In terms of duty this translates to an average raise of 7.9% across the approach velocity range but with an increasing pressure drop penalty varying between 0.5 to 10%.

Airflow velocity (m/s)	Pressure Drop		Temperature rise	
	Plain Fin (Pa)	Serrated Fin (Pa)	Plain Fin (^oK)	Serrated Fin (^oK)
1	2.98	2.99	17.36	19.03
2	8.25	8.41	13.01	13.74
3	16.22	17.16	10.49	11.22
4	26.62	28.86	8.89	9.72
5	38.94	43.24	7.78	8.70

Table 3: Bundle pressure drop and air temperature rise

3.2 Air temperature across fin/tube unit

The temperatures on the planes shown in Figure 3 were plotted and the results are shown in Appendix A.
It can be seen that the air temperature on all planes, except the initial one (90mm), is constantly higher for the serrated fins.

3.3 Overall turbulence

Appendix B shows the % turbulence intensity on the planes. It can be seen that the turbulence intensity is generally higher in the serrated fin models, however in each case there is similarity in the values on the final plane.

4. FLOW PATTERNS

4.1 Heat flow

The exposed surface area of the serrated fin was 95.2% that of the plain fin. Given that extended surface area is considered of paramount importance in a heat exchanger, and that the tube area is constant in both designs, the results indicate that the extra heat transfer must be due to the shape factor. Figures 4 and 5 show the temperature contours on the surface of the plain and serrated fins, respectively.

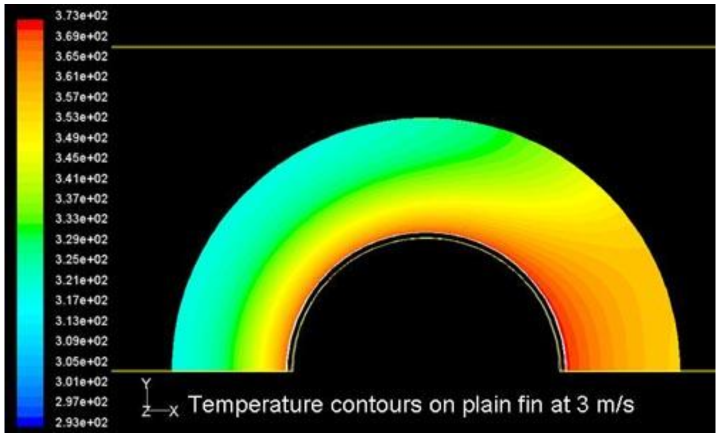

Figure 4: Temperature contours on plain fin at 3 m/s (flow left to right, scale in ^{o}K)

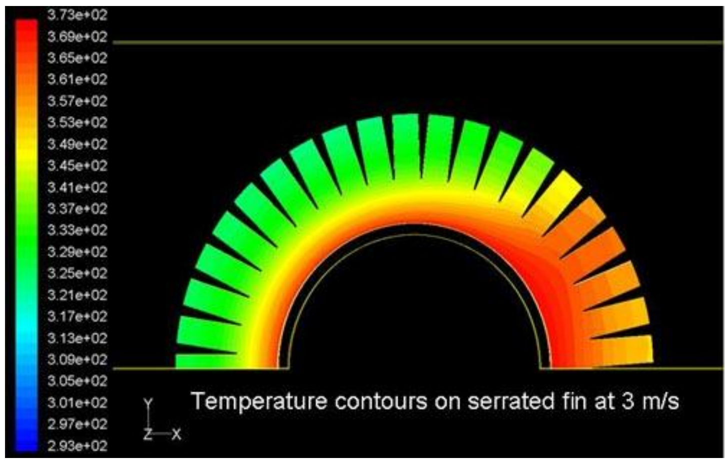

Figure 5: Temperature contours on serrated fin at 3 m/s (flow left to right, scale in ^{o}K)

It can be seen that on the leading edge of the serrated fin that the temperature is higher at approximately 329^{o}K than on the same region on the plain fin, which is approximately 317^{o}K. The temperature in the wake region in Figure 5 and immediately around the tube can be seen to be noticeably higher than that of the plain fin case, indicating a higher heat flow through the fin. The blades in the wake region of Figure 5 show a significantly higher temperature, and given that this is over a reasonable proportion of surface area this region alone will contribute significantly to the improved heat transfer, even though the effect is partially created by recirculating flow.

The comparisons for 1, 2, 4 and 5 m/s are similar, and hence are not shown.

4.2 Airflow characteristics

The reduced surface area of the serrated fin will result in reduced skin friction drag, but with the irregular shape of the blade protrusions into the oncoming flow the results would indicate that this was overcome by an increase in excrescence drag. In investigations into the local velocities around the fin surface it was seen that separation on the blade at the front of the fin was pronounced when compared to the separation that occurs over the plain fin leading edge. This would promote increased drag, and hence overall pressure drop, as seen in Table 3. The flow simply entered the gaps between blades, and then separated. There was very little evidence of recirculation behind the fins around the middle of the tube and none of flow crossover between the opposite sides of the blade, rather the flow stagnated behind the blades.

Due to the irregular shapes in the flow and the edges of the blades there are more contrasting flows local to the serrated fin surface and hence more turbulence was generated than by plain fins, as shown in Figure 5. Figure 5 also shows that there is significantly more turbulence over the middle section of the serrated fin than over the plain fin. This is due to the highly channeled flows between the blades on the front and the blades on the mid section of the fin causing separations, and therefore adding disturbances to the largely isotropic flow over the front of the tube region.

At the rear of the tube the turbulence of both fin types is comparable, which indicates that the turbulence is almost reducing for the serrated case, rather than either maintaining the difference or increasing it. The reasons for this are unclear, but may lie with the recirculation and wake effects developing over the rear of the tube being a much larger driving force, and that the extra turbulence generated by serrated fins becomes homogenised with the bulk flow, as with the plain fin example.

5. CONCLUSIONS

Though the single row heat exchanger is a special case, it can be seen that serrated fins improve the heat transfer although the mechanism may not be as amenable to a straightforward treatment as reported by previous workers, such as Weierman [2,3] and Ganapathy [4]. The results suggest that the heat transfer increase due to increasing turbulence that is often attributed to serrated fins may only be effective in certain regions, and would suggest that the access of the airflow further into the fin material is a larger driver in the heat transfer process. A further study with differing geometries may answer this question more fully.

NOMENCLATURE

$\mathbf{D_f}$	Fin tip diameter	m
$\mathbf{D_o}$	Outer tube diameter	m
$\mathbf{D_i}$	Inner tube diameter	m
$\mathbf{H_{sf}}$	Height of solid component of fin	m
$\mathbf{n_b}$	number of blades per fin	-
$\mathbf{n_f}$	Fin frequency	-
$\mathbf{\Delta P}$	Pressure drop	Pa
$\mathbf{s_f}$	Mean fin thickness	m
$\mathbf{W_b}$	Width of blade	m

REFRENCES

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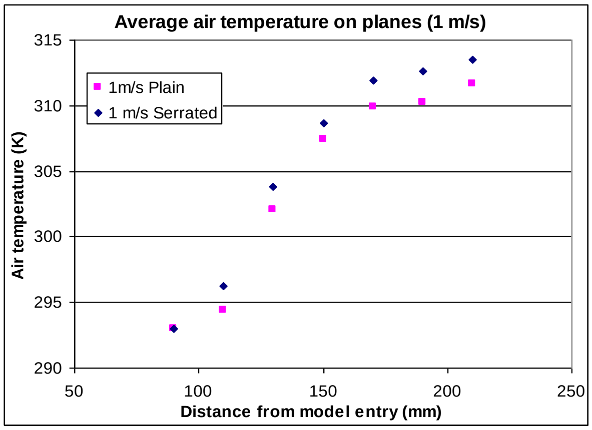
APPENDIX A - Comparison of air temperature on planes



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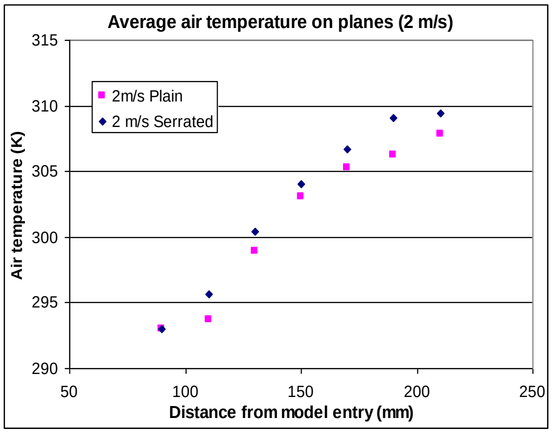

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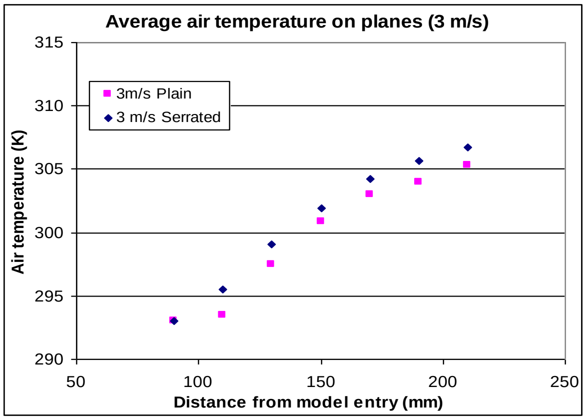

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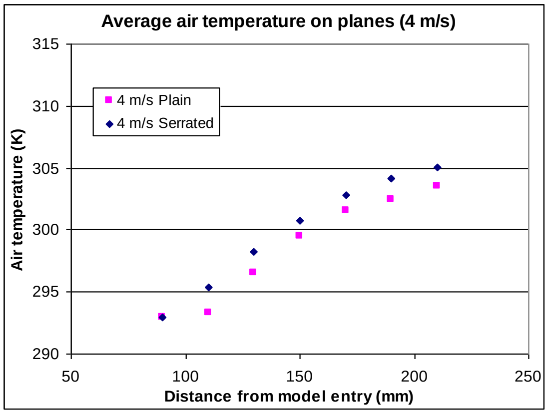

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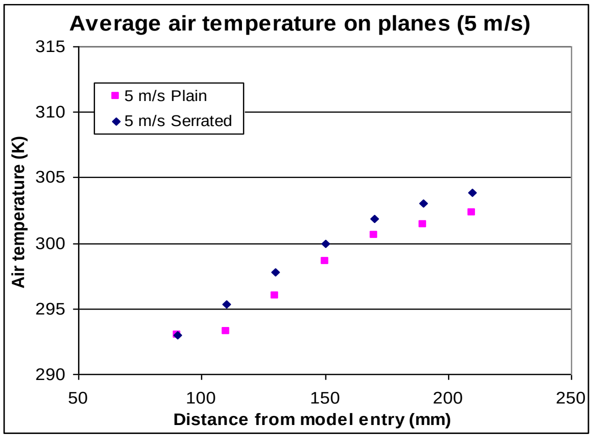

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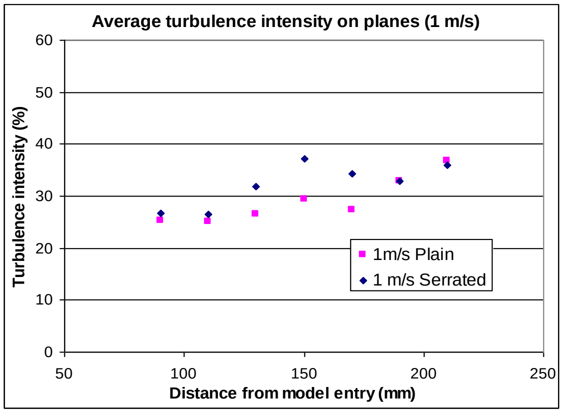
APPENDIX B - Comparison of flow turbulence on planes



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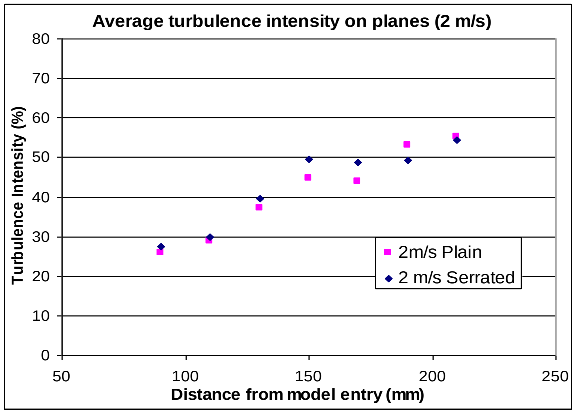

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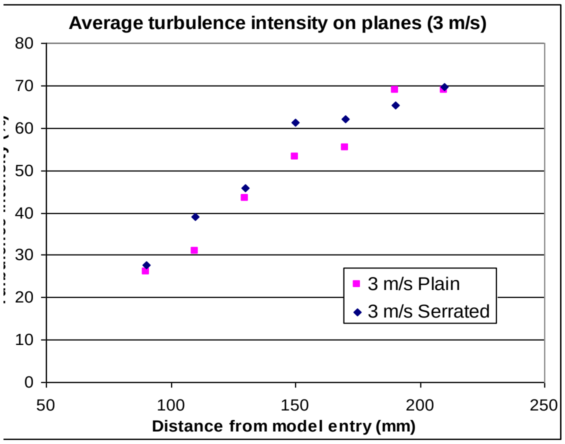

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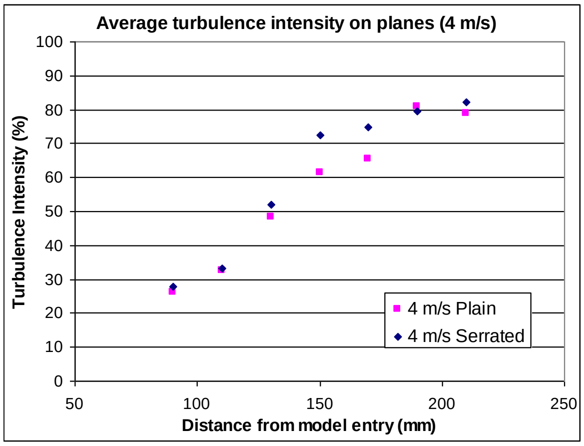

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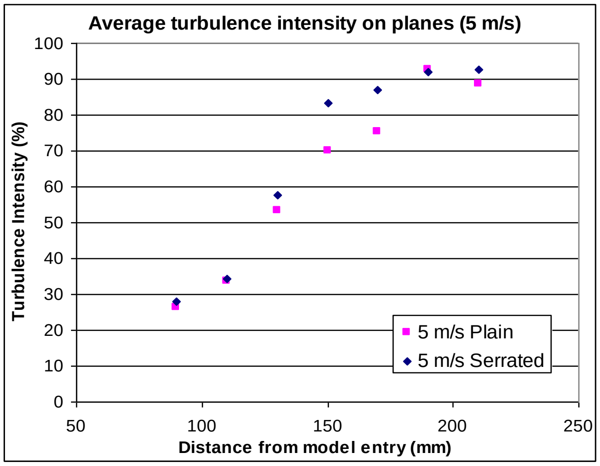

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BIOGRAPHY:

Dr. Stuart Mcilwain received a B.Eng (Hons) degree in Automotive Engineering from the University of Hertfordshire. After a period working for the UK National Engineering Laboratory he received a Ph.D. in mechanical engineering from the University of Strathclyde. Currently, he is a lecturer at the University of the West of Scotland. His research interests include fluid mechanics and thermal engineering. He is also a retained consultant to a number of local and international companies on fluid and thermal design issues