An Investigation of air side heat transfer enhancement in multi-louver fins within automotive compact heat exchangers

Patrick Fix

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An Investigation Of Air Side Heat Transfer Enhancement In
Multi-Louver Fins Within Automotive Compact Heat Exchangers

by

Patrick E. Fix III

A Thesis submitted in partial fulfillment
of the requirements for the degree of
Master of Science in Mechanical Engineering

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DECEMBER 1996
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December 16, 1996

Patrick E. Fix III
ACKNOWLEDGMENTS

Henry Beamer, from Delphi Harrison Thermal Systems in Lockport, NY, provided both professional and personal guidance. He was a point source for information on multi-louver fins, knowledge of experimental methods used to map heat exchanger performance, and data analysis techniques. Without his patience, assistance and understanding, I would have never completed this project.

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Finally, I would like to thank God. Faith in His word has led me to overcome all obstacles. He has been beside me, strengthening and molding me into the man that I am today. He has never abandoned me nor will He ever abandon anyone else who puts faith in Him.
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</tbody>
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### List of Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Total surface area on one side of a hxer (A_p + A_f)</td>
<td>m²</td>
</tr>
<tr>
<td>A_c</td>
<td>Cross-sectional area of a duct</td>
<td>m²</td>
</tr>
<tr>
<td>A_f</td>
<td>Fin surface area on one side of a hxer</td>
<td>m²</td>
</tr>
<tr>
<td>A_f_r</td>
<td>Heat exchanger frontal area</td>
<td>m²</td>
</tr>
<tr>
<td>A_o</td>
<td>Open flow area on one side of a hxer</td>
<td>m²</td>
</tr>
<tr>
<td>A_p</td>
<td>Primary surface area on one side of a hxer</td>
<td>m²</td>
</tr>
<tr>
<td>a</td>
<td>Tube wall thickness</td>
<td>m</td>
</tr>
<tr>
<td>b_c</td>
<td>Fin thickness (Air center material gage)</td>
<td>m</td>
</tr>
<tr>
<td>C</td>
<td>Heat capacity rate</td>
<td>W/°C</td>
</tr>
<tr>
<td>C_max</td>
<td>Maximum C value (air side or tube side)</td>
<td>W/°C</td>
</tr>
<tr>
<td>C_min</td>
<td>Minimum C value (air side or tube side)</td>
<td>W/°C</td>
</tr>
<tr>
<td>C*</td>
<td>Ratio of C_min and C_max</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>c_p</td>
<td>Specific Heat of a fluid at constant pressure</td>
<td>J/(kg °C)</td>
</tr>
<tr>
<td>D</td>
<td>Ideal transverse distance traveled by an air-flow [28]</td>
<td>m</td>
</tr>
<tr>
<td>D_h</td>
<td>Hydraulic Diameter</td>
<td>m</td>
</tr>
<tr>
<td>F_p</td>
<td>Fin pitch</td>
<td>m</td>
</tr>
<tr>
<td>f</td>
<td>Fanning friction factor</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>fRe</td>
<td>Fanning friction factor and Reynolds number product</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>G</td>
<td>Mass velocity</td>
<td>kg/(m² s)</td>
</tr>
<tr>
<td>g_c</td>
<td>Proportionality constant from Newton’s second law</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>H_NF</td>
<td>“No Flow Height” height of BDC</td>
<td>m</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------------------------</td>
<td>-----------</td>
</tr>
<tr>
<td>Hc</td>
<td>Center height</td>
<td>m</td>
</tr>
<tr>
<td>Hf</td>
<td>Fin height</td>
<td>m</td>
</tr>
<tr>
<td>H_{L,b}</td>
<td>Bottom louver height - measured</td>
<td>m</td>
</tr>
<tr>
<td>H_{L,c}</td>
<td>Center louver height - measured</td>
<td>m</td>
</tr>
<tr>
<td>H_{L,Dav}</td>
<td>Louver height - Davenport definition</td>
<td>m</td>
</tr>
<tr>
<td>H_{L,t}</td>
<td>Top louver height - measured</td>
<td>m</td>
</tr>
<tr>
<td>H_t</td>
<td>Tube land</td>
<td>m</td>
</tr>
<tr>
<td>h</td>
<td>Convective heat transfer coefficient</td>
<td>W/(m²°C)</td>
</tr>
<tr>
<td>ITD</td>
<td>Inlet temperature difference</td>
<td>°C</td>
</tr>
<tr>
<td>j</td>
<td>Colburn j factor</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>K_c</td>
<td>Contraction loss coefficient for flow at hxer entrance</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>K_e</td>
<td>Contraction loss coefficient for flow at hxer exit</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>L_L</td>
<td>Louver length</td>
<td>m</td>
</tr>
<tr>
<td>L_{L,full}</td>
<td>Fully developed louver length</td>
<td>m</td>
</tr>
<tr>
<td>L_p</td>
<td>Louver pitch</td>
<td>m</td>
</tr>
<tr>
<td>N</td>
<td>Actual transverse distance traveled by an air-flow [28]</td>
<td>m</td>
</tr>
<tr>
<td>N_{conv}</td>
<td>Number of convolutions across the BDC</td>
<td>convolutions</td>
</tr>
<tr>
<td>N_t</td>
<td>Number of tubes in the BDC</td>
<td>tubes</td>
</tr>
<tr>
<td>NTU</td>
<td>Number of Transfer Units</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>P</td>
<td>Perimeter of a duct</td>
<td>m</td>
</tr>
<tr>
<td>Pr</td>
<td>Prantl number</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>Q</td>
<td>Heat transfer rate in an exchanger over a specified time</td>
<td>W s</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
<td>Units</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
<td>-------</td>
</tr>
<tr>
<td>q</td>
<td>Heat transfer rate in an exchanger (“heat duty”)</td>
<td>W</td>
</tr>
<tr>
<td>q’</td>
<td>Heat transfer rate in an exchanger per unit length</td>
<td>W/m</td>
</tr>
<tr>
<td>q”</td>
<td>Heat transfer rate in an exchanger per unit surface area</td>
<td>W/m²</td>
</tr>
<tr>
<td>R</td>
<td>Heat exchanger thermal resistance</td>
<td>°C/W</td>
</tr>
<tr>
<td>R₀</td>
<td>Overall heat exchanger thermal resistance</td>
<td>°C/W</td>
</tr>
<tr>
<td>R₆</td>
<td>Heat exchanger wall thermal resistance</td>
<td>°C/W</td>
</tr>
<tr>
<td>Rbf</td>
<td>Braze fillet radius</td>
<td>m</td>
</tr>
<tr>
<td>R_conv</td>
<td>Convolution tip radius</td>
<td>m</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>ReDh</td>
<td>Reynolds number based on hydraulic diameter</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>ReLP</td>
<td>Reynolds number based on louver pitch</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>St</td>
<td>Stanton number</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>StDh</td>
<td>Stanton number based on ReDh</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>StLP</td>
<td>Stanton number based on ReLP</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>StDav</td>
<td>Stanton number based on Davenport correlation</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>t</td>
<td>Static fluid temperature</td>
<td>°C</td>
</tr>
<tr>
<td>tₘ</td>
<td>Mean static fluid temperature</td>
<td>°C</td>
</tr>
<tr>
<td>Δtm</td>
<td>Log-mean temperature difference</td>
<td>°C</td>
</tr>
<tr>
<td>Δtₘ</td>
<td>True mean temperature difference</td>
<td>°C</td>
</tr>
<tr>
<td>U</td>
<td>Mean overall heat transfer coefficient</td>
<td>W/(m² °C)</td>
</tr>
<tr>
<td>u</td>
<td>Velocity</td>
<td>m/sec</td>
</tr>
<tr>
<td>W</td>
<td>Mass flow rate</td>
<td>kg/sec</td>
</tr>
<tr>
<td>WA</td>
<td>Mass flow rate per unit frontal area</td>
<td>kg/(sec m²)</td>
</tr>
<tr>
<td>Symbol</td>
<td>Units</td>
<td></td>
</tr>
<tr>
<td>---------</td>
<td>--------------------------------</td>
<td></td>
</tr>
<tr>
<td>$W_{HDR}$</td>
<td>Header to header width - width of BDC</td>
<td>m</td>
</tr>
<tr>
<td>$x$</td>
<td>Axial distance (core depth)</td>
<td>m</td>
</tr>
<tr>
<td>$x^+$</td>
<td>Axial distance for the fluid flow problem</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>$x^*$</td>
<td>Axial distance for the heat transfer problem</td>
<td>Dimensionless</td>
</tr>
</tbody>
</table>
# List of Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \alpha_b )</td>
<td>Bottom louver angle</td>
<td>°</td>
</tr>
<tr>
<td>( \alpha_c )</td>
<td>Center louver angle</td>
<td>°</td>
</tr>
<tr>
<td>( \alpha_t )</td>
<td>Top louver angle</td>
<td>°</td>
</tr>
<tr>
<td>( \beta )</td>
<td>Mean flow angle</td>
<td>°</td>
</tr>
<tr>
<td>( \beta_{\text{max}} )</td>
<td>Maximum mean flow angle</td>
<td>°</td>
</tr>
<tr>
<td>( \gamma )</td>
<td>Form angle</td>
<td>°</td>
</tr>
<tr>
<td>( \Gamma )</td>
<td>Flow efficiency</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>( \Gamma_B )</td>
<td>Flow efficiency defined by Bellowss (1996)</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>( \Gamma_{\text{ca}} )</td>
<td>Flow efficiency defined by Cowell and Achaichia (1988)</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>( \Gamma_w )</td>
<td>Flow efficiency defined by Webb and Trauger (1991)</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>( \eta_o )</td>
<td>Total effectiveness of one side of a hxer</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>( \eta_f )</td>
<td>Fin efficiency</td>
<td>Dimensionless</td>
</tr>
<tr>
<td>( \mu )</td>
<td>Fluid dynamic viscosity coefficient</td>
<td>Pa s</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>( \rho_i )</td>
<td>Inlet fluid density on one side of a heat exchanger</td>
<td>kg/m³</td>
</tr>
<tr>
<td>( \rho_m )</td>
<td>Mean fluid density on one side of a heat exchanger</td>
<td>kg/m³</td>
</tr>
<tr>
<td>( \rho_o )</td>
<td>Outlet fluid density on one side of a heat exchanger</td>
<td>kg/m³</td>
</tr>
<tr>
<td>( \sigma )</td>
<td>Ratio of the Free Flow Area to the Frontal Area</td>
<td>Dimensionless</td>
</tr>
</tbody>
</table>
Abstract

The multi-louver fin is the primary geometric configuration used to enhance the air-side heat transfer characteristics of automotive heat exchangers today. This work presents an investigation into the effects of multi-louver fins on the air-side Stanton number ($St$) and Fanning friction factor ($f$) characteristics of a heat exchanger. Experimental $St$ data ranging between 0.0095 and 0.065 and $f$ data ranging between 0.04 and 0.55 are reported for louver pitch based Reynolds numbers ranging between 30 and 4000. Brazed core test samples with multi-louver fins are utilized to complete this study.

Previous investigators have correlated the air-side $St$ and $f$ to geometric parameters such as fin height, fin pitch, louver pitch, louver angle and louver length along with flow efficiency and Reynolds number. An example of the range of parameters tested by previous investigators is fin pitch to louver pitch ratio which ranged between 1.65 and 4.1 compared to the range of 2.1 to 5.3 in this work. In general, wider ranges of each parameter than those tested by previous investigators were chosen to produce a more robust correlation.

Three new forms of $St$ correlations are derived which can predict heat transfer more accurately than the forms previously derived in the literature. The average percent deviation was improved from $-24.37\%$ and $+2.10\%$ to $+0.59\%$ with the standard deviation of the deviations improved from $11.43\%$ and $15.04\%$ to $8.95\%$. This provided a $St$ correlation which would predict the heat transfer of a heat exchanger to within $\pm3\%$ for 60% of the data or to within $\pm9\%$ for 99% of the data.

The correlations of Cowell and Achaichia were modified by the author predict $f$ more accurately than any form examined in this work. The average percent deviation was improved from $+12.19\%$ and $-23.59\%$ to $+0.75\%$ and $-0.19\%$ with the standard deviation of the deviations improved from $53.89\%$ to $12.72\%$ for one correlation and maintained around $11\%$ for the other.
1. Introduction

This thesis presents an investigation into the effects of multi-louver fins on the air-side heat transfer and pressure drop characteristics of a heat exchanger core. The new correlations presented in this work significantly improve on the heat transfer and pressure drop prediction capability available in the open literature. Basic heat exchanger design theory and dimensionless heat transfer and pressure drop parameters are discussed and defined in Appendix A.

This work will quantify heat transfer with the Stanton number (St) and pressure drop with the Fanning friction factor (f). Twenty basic data cores (BDCs) will be tested with the results compared to the St and f correlations proposed by Davenport (1980) and Cowell & Achaichia (1988) discussed in the Chapter 2 literature review. The Cowell & Achaichia St correlation will also be integrated with the Bellows flow efficiency correlation (also discussed in Chapter 2) and compared to the experimental results.

A multiple regression analysis will be performed on each of the correlations in order to fit them to the experimental data obtained in this work. In all cases, this process will produce a correlation which is more centered on the experimental data. In most cases, the standard deviation of the percent deviations to each experimental data point is also reduced to produce a “tighter” fit.

New forms of St correlations will be derived which will attempt to integrate the work of Davenport, Cowell & Achaichia and Bellows. These correlations will predict the St more accurately than any previously suggested correlation recorder in the literature.
Finally, a new form of \( f \) correlation will be derived and found to not exceed the accuracy of the modified Cowell & Achaichia \( f \) correlations.

1.1 Automotive Applications

The focus of this work will be automotive radiators and heater cores. Information on other automotive heat exchangers can be found in Appendix B.

1.1.1 Radiator and Heater Core Applications

For a radiator or heater core, the dominant thermal resistance (discussed in Appendix A) is due to the cold-side fluid, flow and surface properties. The cold-side of a each of these heat exchangers is the air-side. For this reason, heat transfer enhancement research for radiators has been directed towards the air side.

The radiator and heater core perform the same function but for different applications. They both remove heat from the engine coolant and transfer it to the air stream that passes through their respective cores. The radiator is required to remove waste heat from the engine coolant in order to prevent premature engine failure. The heater core adds heat to the air stream that passes through its core. This heated air stream flows towards the passenger compartment of the vehicle through an HVAC module and duct work (see Appendix B).
A radiator can either be in a "crossflow" or "downflow" orientation. A crossflow radiator (depicted in Figure 1-1) is oriented to force engine coolant laterally across the grill of a vehicle. A downflow radiator (depicted in Figure 1-2) is oriented to use gravity to pull engine coolant down the grill of a vehicle. This usage of the term, “crossflow”, should not be confused with the term “crossflow” heat exchanger discussed in Appendix A.

Both crossflow and downflow radiators are crossflow heat exchangers because air travels perpendicular to the engine coolant in each. Figure 1-3 shows both types of radiators.

A heater core is essentially a smaller sized radiator. An example of a heater core can be seen in Figure 1-4.

1.2 Radiator Core and Multi-Louver Fin Geometry

Figure 1-1 has many labeled components and features of an automotive radiator. The components of concern in this paper are the air centers, tubes, core reinforcements and headers. These are the component parts of a radiator core which are used to produce the test samples for this work. The remainder of the labels in Figure 1-1 define features of the plastic radiator tanks or engine / transmission oil coolers.

Figure 1-5 shows an example of a sectioned radiator core. The tubes of the radiator contain the engine coolant and are “flat-oval” shaped. The reinforcements of the radiator are used to hold the air centers firmly against the tubes. This is necessary to ensure that all the air centers are properly joined to the tubes during the “braze” process.
Figure 1-1 Example of a Tube and Center, Crossflow Radiator
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure 1-2 Example of a Tube and Center, Downflow Radiator
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure 1-3 Two Examples of Tube and Center Radiators
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure 1-4 Example of Tube and Center Heater Core
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure 1-5 Example of a Sectioned Radiator Core
The “braze” process heats a core in a furnace to an appropriate temperature to allow cladding on the tubes to melt. The cladding flows into the joints between the tubes and the air centers of the core. When the core is removed from the furnace, the braze clad which originated on the tube solidifies to form a metallic bond between the tube and the air center. If the tubes and air centers are not firmly held together by the reinforcements, the metallic bonds will be imperfect or missing. This will cause a significant rise in the thermal resistance of the core.

The headers provide a union between the tanks and the tubes of a radiator. The tubes of the radiator are brazed to the headers. The headers are then attached to tanks through any of a variety of processes.

The air centers are multi-louver fins used to enhance the air-side heat transfer of a radiator. Figure 1-6 shows a detailed 3D diagram of a multi-louver fin. It defines the width of an air center ($W_c$), height of an air center ($H_c$), fin pitch ($F_p$), fin constant ($F_{const}$), a convolution and the number of convolutions of an air center ($N_{conv}$).

Figure 1-7 shows a detailed front view diagram of one convolution of a multi-louver fin. The air-flow travels normal to the plane of the paper in this figure. It defines the tube height (tube land, $H_t$), fin height ($H_f$), tube pitch ($T_p$), louver length ($L_L$), fully developed louver length ($L_{L,full}$), louver height ($H_L$) at the top, center and bottom of the louver, form angle ($\gamma$), convolution radius ($r_{conv}$) and braze fillet radius ($r_{bf}$).
Figure 1-6 Detailed 3D Diagram of a Multi-Louver Fin
Figure 1-7 Detailed Front View Diagram of a Multi-Louver Fin
Figure 1-8 shows a detailed louver pattern diagram of a multi-louver fin. It defines the louver height ($H_L$), louver pitch ($L_p$) and louver angle ($\alpha$). The margins, louver panels and “turn around rib” regions of a multi-louver fin are also identified. The $L_p$ is the characteristic length of the Reynolds number of the air-flow through the heat exchangers in this work. The rationale behind this is fully discussed in Chapter 2 and Appendix C.

The relationship of the dimensions defined in Figures 1-5 through 1-8 to $St$ and $f$ will be discussed throughout this work as the need arises. Chapter 2 will discuss the parameters that Davenport (1980) and Cowell & Achaichia (1988) chose to correlate to $St$ and $f$.

Chapter 2 will also discuss the definition of flow efficiency and the parameters that Cowell & Achaichia (1988) and Bellow (1996) chose to define the flow efficiency of the air-flow through the core.

Table 3-1 will detail the ranges of the correlated parameters tested by Davenport (1980), Cowell & Achaichia (1988), Bellows (1996) and the author.

Finally, Chapter 4 and Chapter 5 will discuss the parameters that the author chose to correlate to $St$ and $f$. 
Figure 1-8 Detailed Louver Pattern Diagram of a Multi-Louver Fin
1.3 Objectives of the Present Work

The objectives of this work are listed below.

1. Compare the experimental results for $St$ and $f$ to those predicted by the correlations proposed by Davenport (1980). Modify the correlation coefficients to produce more accurate correlations.

2. Compare the experimental results for $St$ and $f$ to those predicted by the correlations proposed by Cowell & Achaichia (1988). Modify the correlation coefficients to produce more accurate correlations.

3. Compare the experimental results for $St$ to those predicted by the integration of the Bellows flow efficiency correlation into the correlation proposed by Cowell & Achaichia (1988). Modify the correlation coefficients to produce a more accurate correlation.

4. Develop a new form of a correlation for the prediction of $St$ in heat exchangers with multi-louver fins.

5. Develop a new form of a correlation for the prediction of $f$ in heat exchangers with multi-louver fins.
The $St$ and $f$ correlations of Davenport (1980) and Cowell & Achaichia (1988) along with the flow efficiency correlation of Bellows (1996) are detailed in the following chapter.
2. Literature Review

This chapter will provide a detailed understanding of the published correlations pertaining to the flow field behavior, pressure drop and thermal performance associated with the multi-louver fin. The Davenport (1980, 1984), Cowell & Achaichia (1988) and Bellows (1996) papers will be the only works discussed in depth. A more detailed literature review can be found in Appendix C. The summary of the literature in Section 2.1 will summarize the literature discussed in detail within this chapter and Appendix C.

2.1 Summary of the Literature

When the multi-louver fin was first introduced in the 1950's, researchers theorized that the air-flow through a core with this fin was similar to the “duct flow” exhibited in a core with plain (non-louvered) fins. This duct flow paradigm led researchers to theorize that multi-louver fins enhance heat transfer by turbulating the flow within each duct.

Some researchers tried to correlate turbulence theories to the multi-louver fin. Edwards and Alker (1974) and Russel, Jones and Lee (1982) concluded that turbulence promoters enhance heat transfer. Fiebig, Kallweit and Mitra (1986, 1991) concluded that heat transfer performance could be enhanced by 20% to 60% by delta wing turbulence promoters.
Turk and Junkhan (1986) concluded that counter-rotating vortices enhance heat transfer. Takano, Tanasawa and Nishio (1989) concluded that the enhancement of heat transfer by turbulators does not vary when angles are chosen between 45° and 90°.

As researchers attempted to enhance the heat transfer of a heat exchanger by utilizing turbulating the flow, another group of researchers theorized that louvers that are not parallel to the fins may redirect the flow through the fins of an air carrier.


Tura (1986) and Kurosaki et al. (1988) have shown that the greatest local heat transfer coefficient \( h \) is located near the leading edge of each louver in an array.

Burgers and Lemczyk (1988) concluded that louvers should span at least 70% of the fin height of a multi-louver fin. Huihua and Xuesheng (1989) concluded that the optimum louver angle is 25°.

Cowell and Achaichia (1988), Webb and Trauger (1991) and Bellows (1996) have correlated flow efficiency \( \Gamma \) as a function of louver pitch based Reynolds number \( Re_{Lp} \), fin pitch to louver pitch ratio \( F_p/L_p \) and louver angle \( \alpha \).
Davenport (1980, 1984) and Cowell and Achaichia (1988) have created empirical correlations to predict the heat transfer and pressure drop in a heat exchanger core with multi-louver fins.

Sahnoun and Webb (1992) have created an analytical model to predict the heat transfer and pressure drop in a heat exchanger core with multi-louver fins.

Once again, if greater detail is required by the reader on any of the works mentioned in this section, go to Appendix C.

2.2 Davenport $St$ and $f$ Correlations

This is the first of three sections that describe the principle works referenced by the author in this work.

Davenport (1980) performed a comprehensive study on multi-louver fins. He repeated the smoke trace flow visualization performed by Beauvais (see Appendix C) and demonstrated that the alignment of the air-flow to the louvers was largely a function of Reynolds number. He noted that at low $Re$, the alignment of the air-flow to the louvers would be slight while at high $Re$, the alignment to the louvers would be nearly complete. Davenport also noted that as fin pitch decreased, the alignment of the air-flow increased at similar $Re_{lp}$. 
Davenport suggested that the method of heat transfer within a multi-louver fin array was related to laminar flat plate theory. According to his theory, the louvers redirect the flow through the array in order to increase the average local $h$. The value of $h$ is maximized at the leading edge of the thermal boundary layer of a flat plate. The louver array provides many flat plates which “break” the thermal boundary layer created by each louver and immediately creates a new one. As the number of louvers is increased in an array, the number of thermal boundary layers is also increased. Therefore, the average local $h$ is increased.

Davenport suggested that the louver boundary layers were large enough at low $Re$ to block off the gaps between the louvers. The pressure gradient between the louvers would then be greater than the pressure gradient through the convolution (duct). In this situation, the air-flow would travel through the duct and not gain the advantage of louver enhanced heat transfer. Figure 2-1 shows a comparison of duct (fin directed) flow to flat plate (louver directed) flow (Cowell & Achaichia, 1988).

Davenport suggested that at small values of fin pitch ($F_p$), the difference in pressure gradient between traveling between the louvers and through the convolution (duct) would not be as great for similar $Re$. In this situation, a portion of the air-flow would travel between the louvers and gain the advantage of the louver enhancement to heat transfer while the remainder would travel through the duct.
Figure 2-1 Louver Directed Compared to Fin Directed Flow
Despite this observation on the relationship of $Fp$ to the flow behavior, Davenport later concluded that $Fp$ has no significant effect on heat transfer based upon his regression analysis of the heat transfer data collected for 32 cores. This conclusion is challenged in Chapter 4.

The heat transfer data that Davenport collected was used to create a correlation to predict $St$. The correlation is valid for $300 \leq Re_{Lp} \leq 4000$.

$$St = 0.313Re_{Lp}^{-0.42}H_L^{0.33}\left(\frac{L_L}{H_f}\right)^{1.1}H_f^{0.26}$$  \hfill (2-1)

Figures 1-6 through Figure 1-8 show the parameters listed in equation 2-1. $Re_{Lp}$ is the Reynolds number with the louver pitch as the characteristic length. $H_L$ is the louver height, $L_L$ is the louver length and $H_f$ is the fin height.

The pressure drop data that Davenport collected was used to create two correlations for $f$. This correlation is valid for $70 \leq Re_{Lp} \leq 900$.

$$f = 5.47Re_{Lp}^{-0.72}H_L^{0.37}\left(\frac{L_L}{H_f}\right)^{0.89}L_p^{0.2}H_f^{0.33}$$  \hfill (2-2)

This correlation is valid for $900 < Re_{Lp} \leq 4000$. 
\[ f = 0.494 \Re^{-0.39} \left( \frac{H_L}{L_p} \right)^{0.33} \left( \frac{L_f}{H_f} \right)^{1.1} H_f^{0.46} \]  

(2-3)


$L_p$ is the louver pitch which is also shown in Figure 1-7. Table 2-1 lists the 32 cores tested by Davenport. Figure 2-2 through Figure 2-10 provide the $St$ and $f$ data obtained by Davenport.

Davenport (1984) published a correlation for $j$ in a later work. This correlation was proportional to equation 2-12. The correlation was valid for $300 < \Re_{L_p} \leq 4000$.

\[ j = 0.249 \Re^{-0.42} H_L^{0.33} \left( \frac{L_L}{H_c} \right)^{1.1} H_c^{0.26} \]  

(2-4)

2.3 Cowell and Achaichia $St$ and $f$ Correlations

Cowell and Achaichia (1988) further defined the flow phenomena within multi-louver fins. The authors used methods of numerical analysis to define the mean flow angle ($\bar{\beta}$) as the average angle of the air-flow as it passes through a multi-louver fin array. Figure 2-11 shows $\bar{\beta}$ as a function of the $Re_{L_p}$ and louver angle ($\alpha$). The authors developed a correlation for flow efficiency ($\Gamma_{ca}$) which was a ratio of the mean flow angle to the louver angle.

23
<table>
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<tr>
<th>SAMPLE NO.</th>
<th>Fp (mm)</th>
<th>Lp (mm)</th>
<th>Lh (mm)</th>
<th>H (mm)</th>
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<td>1</td>
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<td>.31</td>
</tr>
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<td>.37</td>
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<tr>
<td>32</td>
<td>3.0</td>
<td>2.25</td>
<td>5.0</td>
<td>.28</td>
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H = 12.7 → Sample Nos. 1 to 19 = 

H = 7.8 → Sample Nos. 20 to 32 = 

---

Table 2-1 Davenport Fin and Louver Geometry Measurements
### Table

<table>
<thead>
<tr>
<th>Surface 10.27T</th>
<th>Surface 11.94T (Kapuane 1)</th>
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<tr>
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</tr>
<tr>
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</tr>
<tr>
<td>Fin Pitch</td>
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<tr>
<td>Ac/Afr</td>
<td>0.912</td>
</tr>
<tr>
<td>As/Al</td>
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</tr>
<tr>
<td>Hydraulic Dia.</td>
<td>3.84 mm</td>
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<tr>
<td>Fin: 0.25 mm</td>
<td>Alumimium</td>
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<table>
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<tr>
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<th>33, 12.70 mm, 3.15 mm</th>
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</thead>
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<tr>
<td>At/Afr</td>
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### Figure 2-2

Davenport St and f Data
Davenport (1980)
Figure 2-3 Davenport St and f Data
Davenport (1980)
Figure 2-4 Davenport St and f Data
Davenport (1980)
Figure 2-5 Davenport St and f Data
Davenport (1980)
Figure 2-6 Davenport St and f Data
Davenport (1980)
Figure 2-7 Davenport St and f Data
Davenport (1980)
Figure 2-8 Davenport $St$ and $f$ Data
Davenport (1980)
Figure 2-9 Davenport St and f Data
Davenport (1980)
Figure 2-10 Davenport St and f Data
Davenport (1980)
Figure 2-11 Mean Flow Angle ($\beta$) as a Function of Louver Angle ($\alpha$) and $Re_L$
Cowell & Achaichia (1988) pp. 149
The Cowell & Achaichia flow efficiency correlation was defined as:

\[
\Gamma_{ca} = \frac{\beta}{\alpha} = \left(0.936 - \frac{243}{Re_{Lp}} - 1.76 \frac{F_p}{L_p} + 0.995 \alpha\right) \alpha \tag{2-5}
\]

Cowell and Achaichia concluded that flow behavior was primarily a function of \(Re_{Lp}\), \(F_p/L_p\) and \(\alpha\), becoming independent of \(Re_{Lp}\) at high \(Re_{Lp}\).

The researchers then tested fifteen variations of cores to obtain heat transfer and pressure drop information. The data was used to correlate \(St\) to \(\Gamma_{ca}\) and \(Re_{Lp}\).

Cowell and Achaichia defined \(St\) valid for \(Re_{Lp} > 75\) as:

\[
St = \Gamma_{ca} 1.18 Re_{Lp}^{-0.58} \tag{2-6}
\]

Figure 2-12 shows a \(St\) versus \(Re_{Lp}\) plot of the Cowell and Achaichia data. The researchers note the transition from duct to flat plate flow occurs at \(70 < Re_{Lp} < 150\) due to the change in slope of the asymptote of the \(St\) versus \(Re_{Lp}\) plot from the duct flow plot to the flat plate flow plot.

The data was then used to correlate \(f\) to \(Re_{Lp}\) and various geometric parameters over two ranges of \(Re_{Lp}\).
Figure 2-12 $St$ versus $Re_{Lp}$ for Heat Exchangers with Multi-Louver Fins
Cowell & Achaichia (1988) pp. 155
For $150 < Re_{Lp} < 3000$, Cowell and Achaichia defined $f$ as:

$$f = 0.895 \left(596 Re_{Lp}^{0.318 \log Re-22.25}\right)^{1.07} F_p^{0.22} L_p^{0.25} L_L^{0.33} (H_c + H_l)^{0.26} \quad (2-7)$$

For $Re_{Lp} < 150$, Cowell and Achaichia defined $f$ as:

$$f = 10.4 Re_{Lp}^{-1.17} F_p^{0.05} L_p^{1.24} L_L^{0.25} (H_c + H_l)^{0.83} \quad (2-8)$$

2.4 Bellows Flow Efficiency Correlation

Bellows (1996) performed a dye-in-water flow visualization to challenge the flow efficiency correlations of Cowell and Achaichia (1988) and Webb and Trauger (1991). He tested 10.5:1 stereolithography models of parallel multi-louver fin arrays. His experiments were reported for $50 \leq Re_{Lp} \leq 500$. He concluded that a modified Cowell and Achaichia (1988) correlation was sufficient to describe flow efficiency through his models.

The Bellows flow efficiency correlation was defined as:

$$\Gamma_b = \left(\frac{-5 - 300}{Re_{Lp}} - 5 \frac{F_p}{L_p} + 1.34 \alpha\right) \quad \alpha$$

(2-9)
Figure 2-13 through Figure 2-16 show actual water flow streamlines through the two of Bellows multi-louver models. These photos were not easily reproducible and had to been enhanced in COREL Draw™ in order to make the streamlines more evident. Figure 2-17 shows $\Gamma_B$ compared to the actual $\Gamma$ data obtained from the models shown in Figure 2-13 through Figure 2-16.

Figure 2-13 through Figure 2-16 depict flow which is a combination of duct and flat plate flow. The figures also help the reader to visualize $\Gamma$ more easily. The figures show that as $Re_{Lp}$ is increased for a given $F_p/L_p$ ratio, the flow becomes more louver directed (flat plate) and therefore the $\Gamma$ increases. They also show that if the $F_p/L_p$ ratio is increased for a given $Re_{Lp}$ then the same occurs.
Figure 2-13 Photo of Flow through a Matrix with $F_p/L_p=1.09$ and $Re_{L_p}=60$

Bellows (1996)
Figure 2-14 Photo of Flow through a Matrix with $F_p/L_p=1.09$ and $Re_{fp}=460$
Bellows (1996)
Figure 2-15 Photo of Flow through a Matrix with $F_p/L_p=1.75$ and $Re_{L_p}=50$
Bellows (1996)
Figure 2-16 Photo of Flow through a Matrix with $F_p/L_p=1.75$ and $Re_{l_p}=490$
Bellows (1996)
Figure 2-17 $\Gamma_B$ versus $Re_{lp}$ for Bellows Test Model 4 and Model 5
Bellows (1996) pp. 37
3. Experimental Investigation

This chapter will explain the process used to obtain the experimental data used in this work. The experimental data was obtained with the use of test facilities at Delphi Harrison Thermal Systems in Lockport, NY.

The term, “basic data core”, will be used often in this chapter. Kays and London (1984), other authors, and the automotive heat exchanger industry have used this term to identify heat exchanger test samples which are tested to obtain the dimensionless basic heat transfer and pressure drop characteristics of a specific heat exchanger surface geometry. For this body of work, the basic data cores (BDCs) were tested in a programmable dissipater. The dissipater, referred to as the “Basic Data Dissipater”, was programmed to run multiple combinations of air and water flow rates. The body of test data collected was then utilized to calculate the basic air-side heat transfer and pressure drop relationships of the specific geometry of the BDC.

3.1 Outline of Investigation

A matrix of twenty BDCs were dissipated for this study. Various parameters of each test sample in the matrix was measured prior to and after dissipation.
The dissipater was programmed to run ten (10) “iso-thermal” and fourteen (14) “heat transfer” test points. Therefore, a total of 200 “iso-thermal” data points and 280 “heat transfer” data points were gathered. These test points are defined in Section 3.6.

The 280 “heat transfer” data points were used to calculate $f$ and $St$ through a “data reduction” process discussed in Section 3.8.2. The $f$ data points provided by the “iso-thermal” test runs were not used. This is because the author felt that the $f$ data points provided by the “heat transfer” test runs were acquired under conditions most similar to actual vehicle conditions and would therefore be most accurate.

The experimental $St$ and $f$ were used to calculate the percent difference of each correlation to each data point. The coefficients of each correlation were modified to provide the best fit to the experimental data and new forms of $St$ and $f$ were investigated.

### 3.2 Range of Variants Tested

Table 3-1 lists the ranges of all the geometric parameters tested in the work of Davenport (1980), Cowell & Achaichia (1988), Bellows (1996) and the present work. Values of $Re_{Lp}$ ranged between 30 and 4000. Values of $f$ ranged between 0.04 and 0.55. Finally, values of $St$ ranged between 0.0095 and 0.065.
<table>
<thead>
<tr>
<th></th>
<th>Fix</th>
<th>Davenport</th>
<th>Cowell et al.</th>
<th>Bellows&lt;sup&gt;1&lt;/sup&gt;</th>
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</thead>
<tbody>
<tr>
<td>Fin Pitch $F_p$</td>
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<td>2.01 3.35 &lt;sup&gt;2&lt;/sup&gt;</td>
<td>1.65 3.33</td>
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<td>Louver Pitch $L_p$</td>
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<td>9.3 14.2</td>
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</tbody>
</table>

<sup>1</sup> Bellows tested 10.5:1 models for flow visualization only.

<sup>2</sup> Davenport tested 30 of 32 cores w/ $F_p = 3.00 \pm 0.25$ mm.

<sup>3</sup> Davenport tested 18 of 32 cores w/ $L_p = 2.25$ mm.

Table 3-1 Range of Variants Tested

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3.3 Test Sample Selection Criteria

The BDCs used in this body of work were built from radiators that were produced by five (5) major automotive radiator manufacturers. The BDCs were either cut down from complete radiators or hand assembled from component parts.

The BDCs were selected based upon the desire to evaluate as broad a range of geometric variations as possible within the confines of available parts.

3.4 Manufacture of the Test Samples

A BDC built from a radiator utilized a core sample of approximately one foot by one foot. This type of BDC was manufactured utilizing the following process:

1. Remove the 1ft² core section from the radiator using a band-saw.
2. Prepare the tubes and centers of the core for the installation of new headers.
3. Install the new headers on each side of the core.
4. Manufacture and install the core reinforcements.
5. Manufacture and weld aluminum tanks onto each side of the core.

After the BDC was constructed, the pre-dissipation measurements (described in Section 3.5.1) were obtained.
A hand assembled or “hand-stacked” BDC was built using tubes and air centers instead of a core sample from an existing radiator. A “hand-stacked” BDC was built according to the following process:

1. Collect the required number of tubes and air centers.
2. Place the tubes and air centers in alternating order on a surface.
3. Align the tubes and air centers.
4. The tubes and centers are “crushed” down to a predetermined dimension. This dimension is determined from the part dimensions and the tube pitch. This process insures contact between the tubes and centers for braze.
5. Install the headers on each side of the core.
6. Manufacture and install the core reinforcements.
7. Send the core through a braze oven to “bond” the core.
8. Manufacture and weld aluminum tanks onto each side of the core.

As with the other process, after the BDC was constructed, the pre-dissipation measurements (described in Section 3.5.1) were obtained.

Figure 3-1 shows an example of a basic data core used in this work.
Figure 3-1 Example of a Basic Data Core
3.5 Test Sample Geometric Variables

Each BDC had various geometric dimensions measured before and after the dissipation process. Dimensions measured before the BDC was dissipated were obtained in a non-destructive process. Dimensions measured after the BDC was dissipated were obtained in a process which removed core samples from the BDC.

3.5.1 Pre-Dissipation Measurements

Each BDC had the following measurements obtained prior to dissipation:

1. No-Flow Height \( (H_{NF}) \)
2. Header to Header Width \( (W_{HDR}) \)
3. Center Height \( (H_c) \)
4. Tube Land \( (H_t) \)
5. Number of Convolutions across the BDC \( (N_{conv}) \)

The \( H_{NF} \) and \( W_{HDR} \) are displayed in Figure 3-1. The remaining dimensions are displayed on Figure 1-6 and Figure 1-7.
The No-Flow Height ($H_{NF}$) refers to the distance between the reinforcements of the BDC. This is the height of the heat transfer surface of the BDC which includes the tubes and air centers. The Header to Header Width ($W_{HDR}$) refers to the distance between the headers of the BDC. This is the width of the heat transfer surface of the BDC. The $H_{NF}$ and $W_{HDR}$ dimensions were measured with metal scales marked in decimal inches. The error for these devices was $\pm 0.01"$. The error for the measurements due to the equipment was calculated to be from $\pm 0.014"$ to $\pm 0.017"$. The location of the start and finish of the $H_{NF}$ and $W_{HDR}$ dimensions had to be determined by a human operator. This also added some error into the measurements. The overall error was estimated to be $\pm 0.05"$.

The Center Height ($H_c$) refers to the height of the air centers within the BDC and the Tube Land ($H_t$) refers to the height of the tubes within the BDC. The values for the $H_c$ and $H_t$ were measured with dial calipers. The error for this device was $\pm 0.0005"$. These dimensions were measured ten times per core and averaged. This method should have captured the dimensional variation throughout the part and averaged out the factor of human error. Therefore, the error for this measurement should be close to $\pm 0.0005"$.

The Number of Convolutions across the BDC ($N_{conv}$) refers to half the total number of fins per air center within the BDC. The $N_{conv}$ was counted manually from three (3) separate air centers in each BDC. The values were averaged to obtain an average convolution count for the BDC. The error for this measurement was $\pm 0.6$ convolutions.
3.5.2 Post-Dissipation Measurements

Each BDC was destructively analyzed once the dissipation test runs were completed. The BDCs were destroyed in order to verify the quality of the BDC (see Section 3.8.1) and to retrieve samples of the BDC for dimensional analysis.

A band-saw was used to remove the BDC samples from the BDC core. Four (4) sections were cut out of each BDC and were encased in an epoxy mount. Once the samples were sealed in this coating, a band-saw was used to cut through the samples.

Samples were cut through the top, center and bottom of the air center louver panels while traveling parallel to the tubes. Next, three (3) samples were polished using fine grades of sandpaper in order to remove any scratch, gouge marks or burrs that may interfere with measurements obtained with an optical comparator.

The following dimensions were measured on these three (3) potted samples:

1. Top Louver Angle ($\alpha_t$)
2. Center Louver Angle ($\alpha_c$)
3. Bottom Louver Angle ($\alpha_b$)
4. Louver Pitch ($L_p$)

These dimensions are displayed in Figure 1-8. 3D diagrams of these samples are shown in Figure 3-2 through Figure 3-4.
Figure 3-2 Example of an Epoxy Sample Used to Verify $\alpha_r$.
Figure 3-3 Example of an Epoxy Sample Used to Verify $\alpha_e$
Figure 3-4 Example of an Epoxy Sample Used to Verify $\alpha_9$. 
The $\alpha_t$, $\alpha_c$ and $\alpha_b$ refer to the average angles that the louvers of a fin were cut and formed relative to the fin. The angles were calculated based upon optical measurements of the slope of the louvers relative to a datum created along the fin.

The optical comparator had an error of $\pm 0.0005^\circ$. The Louver Pitch ($L_p$) refers to the average width of a louver within a louver panel on a fin. This dimension was also measured with the optical comparator.

There was concern over the measurements taken utilizing this process. Care had to be taken in order to prevent the data from being skewed due to burring caused by air bubbles in the epoxy of the polished samples. The air bubbles were formed by the shrinkage of the epoxy as it pulled away from the air center or by the epoxy never completely penetrating the core before curing. If an air bubble was located near a section of the air center that the band-saw passed, the air center section would deform into the air bubble. Because of this concern, samples were carefully checked for the presence of air bubbles and the process of encasing the samples was adjusted to prevent their formation. However, small air bubbles may cause "micro burrs" which may affect the data read off a precise machine like the optical comparator. Human error from having to position the sample and select start and finish points for dimensions also affected the accuracy of the data. The overall error, including the potential for human error, was estimated to be $\pm 0.001^\circ$ on length measurements and $\pm 1^\circ$ for angle measurements.
The fourth epoxy mounted sample was cut parallel to the BDC frontal area. The sample was then carefully ground to between the 1st and 2nd louver. This allowed the following variables to be measured:

1. Braze Fillet Radius \( (r_{bf}) \)
2. Convoluted Fin Tip Radius \( (r_{conv}) \)
3. Convoluted Fin Angle \( (\gamma) \)
4. Center Height \( (H_c) \)
5. Louver length \( (L_i) \)
6. Fully Developed Louver Length \( (L_{i,full}) \)
7. Bottom Louver Height \( (H_{ib}) \)
8. Center Louver Height \( (H_{ic}) \)
9. Top Louver Height \( (H_{it}) \)

These dimensions are displayed in Figure 1-7.

The Braze Fillet Radius \( (r_{bf}) \) refers to the radius formed between the tube and air center during the braze process. The Convolution Tip Radius \( (r_{conv}) \) refers to the radius between two adjacent fins within an air center. The Form Angle \( (\gamma) \) refers to the angle between two adjacent fins within an air center. The Louver length \( (L_i) \) and the Fully Developed Louver Length \( (L_{i,full}) \) are defined in Figure 1-7 as length variables of a louver within an air center along direction of the fin height \( (H_f) \). The Bottom Louver Height \( (H_{ib}) \), Center Louver Height \( (H_{ic}) \) and Top Louver Height \( (H_{it}) \) is defined in Figure 1-7.
As before, the samples were carefully checked for the presence of air bubbles and the process of encasing the samples was adjusted to minimize the measurement error which may have been caused by their presence.

3.6 Test Sample Dissipation Procedure

Each BDC was tested in the Basic Data Dissipater once the pre-dissipation measurement process had been completed.

For this study, the Basic Data Dissipater was programmed to run ten (10) “iso-thermal” test points and fourteen (14) “heat transfer” test points.

The run of iso-thermal test points was first. During this test run, there was no water pumped through the BDC. The dissipater was programmed to test each point at an increased air flow rate over the previous test point. The data obtained was used to calculate the Fanning friction factor at various Reynolds numbers at ambient temperatures (~80°F).

The run of heat transfer test points followed the iso-thermal test run. During this test run, heated water was pumped through the BDC. The temperature of the water was maintained near 180°F. This is the highest practical temperature that the system can run without the danger of boiling within the flow. Once again, the dissipater was programmed to test each point at an increased air flow rate over the previous test point.
The water-side Δt was only allowed to operate within a limited temperature range (~2°F). It was kept small in order to maintain a low tube-side resistance to heat transfer and to keep a tight control over the fluid properties of the water. The water-side Δt was also maintained above a minimum temperature in order to minimize the error in the measurements obtained by the thermocouples of the dissipater.

When the dissipater detected a water-side Δt that was outside this range, it re-ran the point with the same air flow and an increased water flow rate. By increasing the water flow rate, the water-side Δt was lowered.

This basic data test algorithm resulted in the fourteen (14) heat transfer points mentioned previously. However, since some air flow rates were tested twice during the test program due to the presence of large water-side Δt values, some of the fourteen (14) points were duplicated. Therefore, for the average BDC, only data for ten (10) air flow rates were obtained.

A schematic of the Basic Data Dissipater is shown in Figure 3-5. The schematic only details the air-side of the dissipator. The air-flow is drawn through the dissipator by either a large or small fan on the outlet side of the BDC. The water-side of the dissipator has little complexity. Both sides consist of piping or duct work, a pump or fan, flow meters, thermocouples and pressure transducers.
Figure 3-5 Schematic of the Basic Data Dissipater
Courtesy of Delphi Harrison Thermal Systems
3.7 Test Sample Dissipation Data Reduction

The term “data reduction” is used to describe the process of utilizing experimental test data to formulate dimensionless relationships which can be used to predict the behavior of known heat exchanger geometries under any condition within the range of conditions tested. In this case, temperature, flow, and pressure data for the air-side of a BDC is used to calculate Nusselt number \((Nu)\) and Fanning friction factor \((f)\) and Reynolds number \((Re)\) product \((fRe)\) relationships versus various characteristic lengths for each BDC. Air flow points between those tested on the dissipater are predicted utilizing linear or, preferably, non-linear interpolations of the dimensionless \(Nu\) and \(fRe\) parameters.

Predicting the air-side heat transfer and pressure drop values of a heat exchanger geometry can be very accurate so long as the correct interpolation method is used. Heat exchangers with geometric variables that have been tested in the past are predicted with the most accuracy.

Interpolation of dimensionless values for an untested geometry which is between two known geometric configurations can be risky and potentially inaccurate. The amount of risk is highly dependent on which geometric variable is changed and the magnitude of that change. Prediction of geometric configurations outside known geometric configurations is extremely risky or even impossible.
3.8 Validation of Experimental Results

The experimental results obtained for each BDC were validated using two methods. First, each BDC was destroyed in order to check the quality of its construction. This verified that the each BDC tested was of quality construction. Second, the data reduction results were used to predict the results of each of the fourteen (14) heat transfer test points tested in the Basic Data Dissipater for each BDC.

3.8.1 Test Sample Destructive Analysis

Each BDC underwent destructive analysis after it had been dissipated in order to measure the multi-louver geometric variables discussed in Section 3.5.2 and to detect whether or not the BDC had adequate tube to air center “braze bond”.

The term “braze bond” identifies the union between a tube and a convolution of an air center. In order to form a proper bond, the BDC must be brazed or heated to an appropriate temperature to allow cladding on the tube to melt and flow into the joint between the tube and the air center. When the BDC is removed from the furnace, the braze clad which originated on the tube solidifies to form a metallic bond between the tube and the air center. The absence of this bond between a tube and air center will greatly increase the thermal resistance of wall of the BDC and skew any dissipation data. The thermal resistance of the BDC consists of conductive resistance for this analysis only. The remainder of the resistances are lumped into the surface characteristics.
In order to verify the quality of the tube to air center braze bond of a core, a band-saw was used to cut through the air centers in five (5) equally spaced cuts parallel to the tubes of the core. The blade of the band-saw would tear the air centers away from the tubes instead of splitting the air centers due to their small material gage.

A good braze bond between a convolution of an air center and a tube would leave a braze fillet on the tube once the air center was torn away. A poor braze bond would leave little or no braze fillet. The tube to air center bond for each convolution was evaluated. The percentage of good tube to air center braze joints versus the total number of tube to air center braze joints was calculated. If the percentage of good braze bonds exceeded 99%, the BDC was considered acceptable. If poor braze bond was found to permeate the BDC, the results obtained using that BDC were considered invalid.

Figure 3-6 shows examples of acceptable and unacceptable tube to air center braze bond. Convolutions 1, 2, 4 and 7 would be acceptable. Convolutions 3, 5, 6 and 8 would not be acceptable.

3.8.2 Prediction of Dissipation Test Results

The $Nu$ and $fRe$ relationships obtained from the reduction of the data from each BDC were used to predict the heat transfer and pressure drop values for each of the fourteen (14) “heat transfer” test points. The results of these predictions were compared to the actual test data for each BDC to validate the results.
Figure 3-6 Acceptable and Unacceptable Tube to Air Center Braze Bond
The results were predicted with a proprietary software routine that interpolates values of $Nu$ and $fRe$ based upon eight (8) of the fourteen (14) basic data heat transfer test points. Eight (8) points were the maximum number allowed within the software routine. Accurate prediction of the experimental results verified that the eight (8) points selected out of the fourteen (14) for the two relationships were adequate over the air and water flow ranges tested.

As mentioned previously, if interpolation is used to predict the performance of a core using data obtained with that core geometry, the results are accurate. Interpolation used to predict a different core geometry becomes incrementally more hazardous.
4. Data Analysis

This chapter discusses the correlation methods used to analyze the data obtained by the experimental procedure detailed in Chapter 3. Due to the proprietary nature of the correlation work performed in this chapter, the leading coefficients and exponential coefficients for all the non-published correlations have been omitted. Scatter plots of $St$ versus $Re_{Lp}$ and $f$ versus $Re_{Lp}$ and error analyses are provided for all correlations to evaluate the goodness of fit for each.

4.1 Overall Methodology

Twenty BDCs were tested to provide 280 data points for this work. As mentioned previously, these BDCs were selected to be representative of cores used in automotive radiator applications today. Table 3-1 details the range of fin pitch, louver pitch, fin pitch to louver pitch ratio, louver angle, louver height, louver length, fin height, louver length to fin height ratio, tube land and tube pitch tested for this work as well as those tested by Davenport (1980), Cowell and Achaichia (1988) and Bellows (1996).

The results of the data reduction process described in Section 3.7 provided 280 data points consisting of $Re_{Dh}, f, j, x^*, fRe, x^*, Nu$ and $Pr$ information. $Re_{Lp}$ and $St$ were calculated for each data point based on $D_h$ and $L_p$ information obtained when the BDCs were measured and the $Re_{Dh}, Nu$ and $Pr$ information obtained from the data reduction. This data was used as the "experimental" data for all correlation fits and error analyses.
Figure 4-1 is a scatter plot of all the $St$ versus $Re_{lp}$ data points obtained in this work. $St$ was chosen instead of $j$ as the dimensionless parameter to correlate in this work because Davenport (1980) and Cowell and Achaichia (1988) had chosen it. This eliminates the need to convert predicted Davenport and Cowell & Achaichia $St$ values to $j$. Plus, since the $Pr$ does not vary much from 0.7 on the air-side of an automotive heat exchanger, the $St$ need only be multiplied by 0.79 to calculate $j$.

Inspection of Figure 4-1 reveals a change in the average slope of the data around $Re_{lp}=150$. This change in slope is sometimes referred to as the “knee” in the plot. The location of this knee is important to heat exchanger designers because it pinpoints the $Re_{lp}$ at which a greater rate of increase in the air-side thermal resistance occurs. The change in behavior of the thermal resistance can be attributed to both a change from flat plate to duct flow and to a potential increase in the occurrence of experimental errors at low flow rates.

The change from flat plate to duct flow was not physically observed in this experiment because of the scale of the tested surfaces but can be seen in the work of Beauvais (1965), Wong and Smith (1973), Davenport (1980, 1984), Cowell and Achaichia (1988), Webb and Trauger (1991) and Bellows (1996). The knee is evidence of the presence of a flow efficiency described by Cowell and Achaichia (1988), Webb and Trauger (1991) and Bellows (1996).
Figure 4-1 St vs Re - Present Data
The knee in the experimental data is situated well before the knee situated at $Re_{Lp}=1050$ in the Webb and Trauger (1991) flow efficiency correlation but is nearly predicted by both the Cowell and Achaichia (1988) and Bellows (1996) flow efficiency correlations.

Figure 4-2 is a scatter plot of all the $f$ versus $Re_{Lp}$ data obtained in this work. This figure displays the typical behavior of $f$. The data for $f$ tends to scatter more as the $Re_{Lp}$ approaches 0. A slight knee in the data can be observed in the range of $900 < Re_{Lp} < 1000$.

Correlations were plotted versus $Re_{Lp}$ and compared to the experimental $S_t$ and $f$ data. The correlations of Davenport (1980) and Cowell & Achaichia (1988) were examined over the entire range of $Re_{Lp}$ tested in this work and over the range of $Re_{Lp}$ suggested by the author(s) of each correlation. The percent deviation of each predicted correlation data point was calculated for each of the experimental values of $S_t$ and $f$. Statistical information on the magnitude of the percent deviation values were captured. The statistical data included the average, standard deviation, maximum value, minimum value and the overall range of deviations. A summary of the statistical deviation data for each correlation to the experimental data can be found in Table 4-1 and Figures 4-3 and 4-4.
Figure 4-2 $f$ vs $Re$ - Present Data
<table>
<thead>
<tr>
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<tr>
<td></td>
<td>Average</td>
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<tr>
<td>1</td>
<td>Davenport vs St</td>
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<tr>
<td>2</td>
<td>Davenport Fit vs St</td>
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<tr>
<td>3</td>
<td>Davenport vs St 300&lt;Re&lt;4000</td>
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<tr>
<td>4</td>
<td>Davenport Fit vs St 300&lt;Re&lt;4000</td>
</tr>
<tr>
<td>5</td>
<td>Davenport vs f 70&lt;Re&lt;900</td>
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<tr>
<td>6</td>
<td>Davenport Fit vs f 70&lt;Re&lt;900</td>
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<td>Davenport vs f 1000&lt;Re&lt;4000</td>
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<tr>
<td>28</td>
<td>Fix vs f Re&gt;150</td>
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</tbody>
</table>

Table 4-1 Correlation % Deviation Analysis
Average % Deviation

Figure 4-3 Average % Deviation
Figure 4-4 Standard Deviation of % Deviations
After each correlation was compared to the experimental data, they were modified or "fit" to the experimental data using a method of multiple regression analysis. The leading coefficients and exponential coefficients for each correlation were optimized using the Microsoft Excel\textsuperscript{TM} Version 5.0 Solver function. Once again, the new coefficients are not listed in this work and are simply represented as \( A', A'' \), etc. The multiple regression process yielded correlations that were more centered on the experimental data and with less overall variation.

Finally, lessons learned from examining the modified Davenport, Cowell & Achaichia and Cowell & Achaichia with \( \Gamma_b \) correlations were used to formulate new correlation forms to more accurately predicted \( St \) and \( f \).

4.2 Comparison to the Davenport \( St \) and \( f \) Correlations

The Davenport correlations for \( St \) and \( f \) were the first to be compared to the experimental data. The equations for the correlations in this chapter are numbered in the same order as they appear in Table 4-1 for consistency:

The Davenport \( St \) Correlation with no \( Re_{Lp} \) restrictions:

\[
St = 0.313Re_{Lp}^{-0.42}H_f^{0.26}\left(\frac{L_L}{H_f}\right)^{1.1}H_f^{0.26}
\]  
(4-1)
Figure 4-5 shows this correlation compared to the experimental data. Inspection of the figure yields that the Davenport correlation does not exhibit a knee and clearly under predicts $St$. The slope of the Davenport correlation closely resembles the slope of the experimental data after the knee ($Re_{LP} > 150$). The average deviation of the correlation to the experimental data is -24.55%.

The lack of the knee in the Davenport correlation can be attributed to its form. Davenport realized that his equation has this characteristic and bounded his correlation as valid for $300 \leq Re_{LP} \leq 4000$.

The Davenport correlation under predicts $St$ for a number of reasons. The most probable reason for this is that Davenport did not test cores with values of $L_p$ as small as those which are tested in this work. His smallest $L_p$ of 1.5 mm was larger than the largest $L_p$ of the present work (1.15 mm). Davenport also tested 18 of his 32 cores with a 2.25 mm $L_p$. The present data contained a wide spread of $L_p$ which ranged between 0.73 and 1.15 mm. The large concentration of cores with 2.25 mm louver pitch in his data may have contributed to the deviation.
Figure 4-5: St vs Re (Lp) Restrictions and Present Data

\[
St = 0.313 R e_{Lp}^{-0.42} H_L^{0.33} \left( \frac{L_L}{H_f} \right)^{1.1} H_f^{0.26}
\]
The Davenport multi-louver fin did not have a similar “turn-around rib” to those found in the present BDCs. The Davenport rib was “V-shaped” and looked as though two half louvers were formed together to create it. The ribs in the present BDCs looked more “pan” shaped as depicted in Figure 1-8. These differences may have accounted for a percentage of the Davenport under prediction.

**The Modified Davenport St Correlation with no \( Re_{Lp} \) restrictions:**

\[
St = A' \left( \frac{L_L}{H_f} \right)^{A''} \left( \frac{L_L}{H_f} \right)^{A'''} H_f^{A''''} 
\]

(4-2)

Figure 4-6 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Davenport St correlation still does not exhibit a knee but predicts St more accurately than before. The average deviation of the correlation to the experimental data has improved to +7.48% from -24.55%.

The scatter plot shows the modified Davenport St correlation slightly over predicting at high \( Re_{Lp} \). The slope of the modified Davenport St correlation was flattened because of the presence of the knee in the range of correlated data. If the knee were removed from the range, the correlation would have been fit to the data in the higher \( Re_{Lp} \) range with greater accuracy. This fact makes it not surprising that Davenport limited the valid \( Re_{Lp} \) range of his correlation to \( 300 \leq Re_{Lp} \leq 4000 \) which are to the right of the knee.
\[ St = A \left( \frac{L_t}{H_t} \right)^{A''} \left( \frac{L_t}{H_t} \right)^{A'''} \]

Figure 4-6 St vs Re
Modified Davenport w/ No Re(Lp) Restrictions and Present Data
For reference, Figure 4-7 shows the Davenport and modified Davenport $St$ correlations plotted versus $Re_{lp}$. This plot shows the shift in the correlation created by the new correlation coefficients. Note the distinct change in slope created by the Solver trying to fit the Davenport correlation to the entire range of experimental data.

**The Davenport St Correlation over $300 \leq Re_{lp} \leq 4000$:**

$$St = 0.313Re_{lp}^{-0.42} H_L^{0.33} \left( \frac{L_t}{H_f} \right)^{1.1} H_f^{0.26}$$  \hspace{1cm} (4-3)

Figure 4-8 shows this correlation compared to the experimental data. This figure is identical to Figure 4-5 with the exception of the removal of values of Davenport $St$ correlation data below $Re_{lp}=300$.

Inspection of the figure yields that the Davenport correlation still clearly under predicts $St$. The average deviation of the correlation to the experimental data is -24.37\% over this range. This average deviation is slightly better than the deviation for the Davenport correlation plotted over the entire measured range of $Re_{lp}$.

**The Modified Davenport St Correlation over $300 \leq Re_{lp} \leq 4000$:**

$$St = A' Re_{lp}^A H_L^{A'} \left( \frac{L_t}{H_f} \right)^{A''} H_f^{A'''}$$  \hspace{1cm} (4-4)
Figure 4-7 St vs Re
Davenport w/ No Re(Lp) Restrictions and Modified Davenport w/ No Re(Lp) Restrictions
St versus Re(Lp)

- Experimental
- Davenport 300<Re<4000

\[ St = 0.313 Re^{-0.42} H_f^{0.33} \left( \frac{L_L}{H_f} \right)^{1.1} H_f^{0.26} \]
Figure 4-9 shows this correlation compared to the experimental data. This figure clearly shows the best Davenport $St$ correlation fit to the experimental data. The average deviation of the correlation to the data has improved to $+1.29\%$ from $-24.37\%$. The slope of the modified correlation closely matches the slope of the data in the correlated range. However, the correlation still does not capture all values of $St$ for different cores at a given $Re_{lp}$. This is evidence of a missing parameter from the form of the correlation.

During his regression analysis, Davenport discounted the effect of the fin pitch on $St$. This observation was based upon a limited range of fin and louver pitch. His range of fin pitch was 2.01 to 3.35 mm with 30 of 32 cores having fin pitch equal to $3.00\pm0.25$ mm and 18 of 32 cores with the louver pitch equal to 2.25 mm.

For reference, Figure 4-10 shows the Davenport and modified Davenport $St$ correlations over the range of $300 \leq Re_{lp} \leq 4000$ plotted versus $Re_{lp}$. This plot shows the shift in the correlation created by the new correlation coefficients. The modified Davenport correlation contains much less scatter and predicts too small of a range of $St$ for each $Re_{lp}$. This is evidence of a missing parameter from the form of the correlation. Hence, the increased standard deviation is still present. A parameter may be missing from the form of the correlation. The missing parameter may be the fin pitch and/or flow efficiency. This will be examined later in this chapter.
Figure 4-9 St vs Re
Modified Davenport 300<Re<4000 and Present Data
Figure 4-10 St vs Re
Davenport 300<Re<4000 and Modified Davenport 300<Re<4000

\[ \text{St} = 0.313 \text{Re}^{0.42} \left( \frac{L_L}{H_L} \right)^{0.11} \left( \frac{H_f}{H_L} \right)^{0.26} \]
The standard deviation of the deviations from the experimental data for equations 4-3 and 4-4 were ~12%. The standard deviation of the deviations from the experimental data for equations 4-1 and 4-2 were ~14% and ~19%. Therefore, equations 4-3 and 4-4 were “tighter” fits to the data.

The Davenport $f$ Correlation over $70 \leq Re_{Lp} \leq 900$:

$$f = 5.47 Re_{Lp}^{-0.72} H_L^{0.37} \left( \frac{L_L}{H_f} \right)^{0.89} L_p^{0.2} H_f^{0.23}$$

(4-5)

Figure 4-11 shows this correlation compared to the experimental data. The average deviation of the correlation to the experimental data is -44.86%. The standard deviation of the deviations from the experimental data is 11.00%. The Davenport $f$ correlation for this range of $Re_{Lp}$ under predicts $f$ in much the same way as Davenport under predicted $St$.

The Davenport $f$ correlation for $70 \leq Re_{Lp} \leq 900$ seems to have a different slope than the experimental data exhibits. The cause of the different slope may also be due to the limited range of variables tested by Davenport.

The Modified Davenport $f$ Correlation over $70 \leq Re_{Lp} \leq 900$:

$$f = A' Re_{Lp}^{A''} H_L^{A''' \prime} \left( \frac{L_L}{H_f} \right)^{A'''} L_p^{\dddot{A}}} H_f^{\dddot{A}}}$$

(4-6)
Figure 4-11 f vs Re
Davenport 70<Re<900 & 1000<Re<4000 and Present Data
Figure 4-12 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Davenport $f$ correlation for $70 \leq Re_{LP} \leq 900$ predicts $f$ more accurately than before. The average deviation of the correlation to the experimental data has improved to +5.99% from -44.86%. The standard deviation of the deviations from the experimental data is 16.77%.

This modified Davenport $f$ correlation is centered on the experimental data but does not map the scatter of the experimental data well. This Davenport $f$ correlation seems to predict too small of a range of $f$ for each $Re_{LP}$. This is evidence of a missing parameter from the form of the correlation. Hence, the increased standard deviation is still present.

The Davenport $f$ Correlation over $900 < Re_{LP} \leq 4000$:

\[ f = 0.494 \cdot Re^{-0.39} \left( \frac{H_L}{L_p} \right)^{0.33} \left( \frac{L_L}{H_f} \right)^{1.1} H_f^{0.46} \]  

(4-7)
Figure 4-12 f vs Re
Modified Davenport 70<Re<900 & 1000<Re<4000 and Present Data
Figure 4-11 shows this correlation compared to the experimental data. This figure also shows equation 4-5. Inspection of the figure yields the break in the Davenport correlations at $Re_{Lp}=900$ matches the slight knee in the experimental data. The Davenport $f$ correlation for this range of $Re_{Lp}$ under predicts $f$ in much the same way as Davenport under predicted $St$. This Davenport $f$ correlation seems to predict too small of a range of $f$ for each $Re_{Lp}$. This is evidence of a missing parameter from the form of the correlation. The average deviation of the correlation to the experimental data is -42.93%. The standard deviation of the deviations from the experimental data is 9.75%.

The Modified Davenport $f$ Correlation over $900 < Re_{Lp} \leq 4000$:

$$f = A' Re_{Lp}^{A''} \left( \frac{H_L}{L_p} \right)^{A'''} \left( \frac{L_L}{H_f} \right)^{A''''} H_f^{A''''}$$  \hspace{1cm} (4-8)

Figure 4-12 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Davenport $f$ correlation for $900 < Re_{Lp} \leq 4000$ predicts $f$ more accurately than before. This modified Davenport $f$ correlation is centered on the experimental data but does not map the scatter of the experimental data well.
This Davenport \( f \) correlation seems to predict too small of a range of \( f \) for each \( Re_{L_p} \). This is evidence of a missing parameter from the form of the correlation. Hence, the increased standard deviation is still present. The average deviation of the correlation to the experimental data is now +1.75%. The standard deviation of the deviations from the experimental data is 13.17%.

For reference, Figure 4-13 shows the Davenport and modified Davenport \( f \) correlations over the range of \( 70 \leq Re_{L_p} \leq 900 \) and \( 900 < Re_{L_p} \leq 4000 \) plotted versus \( Re_{L_p} \). This plot shows the shift in the correlation created by the new correlation coefficients. The modified correlation for \( 70 \leq Re_{L_p} \leq 900 \) does not predict the range of \( f \) for each \( Re_{L_p} \) that the original correlation does. The modified correlation for \( 900 < Re_{L_p} \leq 4000 \) predicts the range of \( f \) for each \( Re_{L_p} \) better than the original correlation. The modified correlations over both ranges of \( Re_{L_p} \) have slopes which are different from the original correlations.

### 4.3 Comparison to the Cowell and Achaichia Correlation

The Cowell and Achaichia correlations for \( St \) and \( f \) were the next correlations compared to the experimental data. The equations for the correlations in this chapter are numbered in the same order as they appear in Table 4-1 for consistency:

**The Cowell & Achaichia \( St \) Correlation with no \( Re_{L_p} \) restrictions:**

\[
St = \Gamma_{ca} 1.18 Re_{L_p}^{-0.58} \tag{4-9}
\]
Figure 4-13 f vs Re

Davenport 70<Re<900 & 1000<Re<4000 and Modified Davenport

\[ f = 5.47 \cdot Re^{-0.72} \cdot H_L^{0.37} \left( \frac{L_p}{H_f} \right)^{0.89} \cdot L_p^{0.2} \cdot H_f^{0.23} \]

\[ f = 0.494 \cdot Re_{Lp}^{0.39} \left( \frac{H_L}{L_p} \right)^{0.33} \left( \frac{L_L}{H_f} \right)^{1.1} \cdot H_f^{0.46} \]

\[ f = A' \cdot Re_{Lp}^{A''} \cdot H_L^{A'''(A''')} \cdot L_p^{A''''(A''''')} \cdot H_f^{A'''''} \]

\[ f = A' \cdot Re_{Lp}^{A''} \left( \frac{H_L}{L_p} \right)^{A''} \left( \frac{L_L}{H_f} \right)^{A''} \cdot H_f^{A''} \]
\( \Gamma_{ca} \) has been defined by equation 2-5.

Figure 4-14 shows this correlation compared to the experimental data. Inspection of the figure yields that the Cowell & Achaichia correlation exhibits a knee for only some of the data and does not under predict \( St \) as the Davenport correlation did. The slope of the Cowell & Achaichia correlation closely resembles the slope of the experimental data after the knee (\( Re_{lp} > 150 \)). The average deviation of the correlation to the experimental data is +6.44%. The standard deviation of the deviations from the experimental data is 22.25%.

The knee in the Cowell & Achaichia correlation occurs in only the data points of some cores because of the Cowell & Achaichia flow efficiency (\( \Gamma_{ca} \)).

The Cowell & Achaichia correlation predicts the \( St \) better than the Davenport \( St \) correlation because they tested cores which were more similar in louver geometry to those tested in this body of work. The correlation may also be more robust than the Davenport correlation.

**The Modified Cowell & Achaichia \( St \) Correlation with no \( Re_{lp} \) restrictions:**

\[
St = \Gamma_{ca} A' Re_{lp}^{A''}
\]  
(4-10)
Figure 4-14 St vs Re
Cowell w/ No Re(Lp) Restrictions and Present Data
Figure 4-15 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Cowell & Achaichia correlation still exhibits a knee for only some of the data. The slope of the modified Cowell & Achaichia correlation was flattened because of the presence of the knee in the range of correlated data.

If the knee were removed from the range, the correlation would have been fit to the data in the higher $Re_{Lp}$ range with greater accuracy. For this reason, Cowell & Achaichia limited the valid $Re_{Lp}$ range of their correlation to $Re_{Lp} \geq 75$. The average deviation of the correlation to the experimental data with no $Re_{Lp}$ restrictions has improved to $+3.92\%$ from $+6.44\%$. The standard deviation of the deviations from the experimental data is $15.47\%$.

For reference, Figure 4-16 shows the Cowell & Achaichia and modified Cowell & Achaichia $St$ correlations plotted versus $Re_{Lp}$. This plot shows the shift in the correlation created by the new correlation coefficients. Note the distinct change in slope created by the Solver trying to fit the Cowell & Achaichia correlation to the entire range of experimental data.

**The Cowell & Achaichia $St$ Correlation for $Re_{Lp} \geq 75$:**

$$St = \Gamma_{ca} \cdot 1.18 Re_{Lp}^{-0.58}$$

(4-11)
Figure 4-15 St vs Re
Modified Cowell w/ No Re(Lp) Restrictions and Present Data

\[ St = \Gamma \alpha A \cdot Re_{Lp}^{\lambda} \]
Figure 4-16 St vs Re
Cowell w/ No Re(Lp) Restrictions and Modified Cowell w/ No Re(Lp) Restrictions
Figure 4-17 shows this correlation compared to the experimental data. This figure is identical to Figure 4-14 with the exception of the removal of values of Cowell & Achaichia St correlation data below $Re_{lp}=75$.

Inspection of the figure yields that the Cowell & Achaichia correlation for $Re_{lp} \geq 75$ predicts $St$ rather well. The average deviation of the correlation to the experimental data is $+2.10\%$. This average deviation is much better than the deviation for the Cowell & Achaichia correlation plotted over the entire measured range of $Re_{lp}$. The standard deviation of the deviations from the experimental data is $15.04\%$.

The correlation still does not capture the full range of $St$ for different cores at a given $Re_{lp}$. This is evidence of a missing parameter from the form of the correlation.

**The Modified Cowell & Achaichia St Correlation for $Re_{lp} \geq 75$:**

$$St = \Gamma_{ca} A' Re_{lp}^{A''} \quad (4-12)$$

Figure 4-18 shows this correlation compared to the experimental data. This figure clearly shows the best Cowell & Achaichia $St$ correlation fit to the experimental data. The average deviation of the correlation to the data has improved to $+1.22\%$ from $+2.10\%$. 
Figure 4-17 St vs Re
Cowell Re>75 and Present Data
98
The correlation still does not capture all values of $St$ for different cores at a given $Re_{Lp}$. This is evidence of a missing parameter from the form of the correlation. The missing parameter may be the louver length to fin height ratio that Davenport had previously investigated. The slope of the correlation is slightly flattened due to the range of $Re_{Lp}$ between 75 and 150 that the Microsoft EXCEL™ Version 5.0 Solver is trying to fit. Limiting the range of $Re_{Lp}$ greater than 150 may provide a better fit. This will be attempted in Section 4.5.

For reference, Figure 4-19 shows the Cowell & Achaichia and modified Cowell & Achaichia $St$ correlations for $Re_{Lp} \geq 75$ plotted versus $Re_{Lp}$. This plot shows the shift in the correlation created by the new correlation coefficients.

The Cowell & Achaichia $f$ Correlation for $Re_{Lp} < 150$:

\[
f = 10.4 Re_{Lp}^{-1.17} F_p^{0.05} L_p^{1.24} L_c^{0.25} (H_c + H_l)^{0.83}
\]  

(4-13)

Figure 4-20 shows this correlation compared to the experimental data. Inspection of the figure yields the break in the Cowell & Achaichia correlations at $Re_{Lp}=150$ matches the location of the increased amount of scatter in the experimental data. The Cowell & Achaichia correlations also handle the slight knee in the data at $Re_{Lp}=900$ quite well.
Figure 4-19 St vs Re
Cowell Re>75 and Modified Cowell Re>75
Figure 4.20 $f$ vs $Re(L_p)$

Experimental
Cowell $Re < 150$
Cowell $Re > 150$

\[
f = 10.4 Re_{L_p}^{-1.17} P_r^{0.05} L_p^{1.24} L_{L_p}^{0.25} (H_c + H_r)^{0.83}
\]

\[
f = 0.895 (596 Re_{L_p}^{0.318 \log Re - 2.25})^{1.07} P_r^{0.22} L_p^{0.25} L_{L_p}^{0.33} (H_c + H_r)^{0.26}
\]
The average deviation of the correlation to the experimental data is +12.19%. The standard deviation of the deviations from the experimental data is 53.89%. The Cowell & Achaichia $f$ correlation for this range of $Re_{L_p}$ under predicts $f$ in much the same way as Davenport under predicted $f$.

The Cowell & Achaichia $f$ correlation for $Re_{L_p} < 150$ has a different average slope than the experimental data exhibits and a large amount of scatter.

**The Modified Cowell & Achaichia $f$ Correlation for $Re_{L_p} < 150$:**

$$f = A' Re_{L_p}^{X} F_p^{X'} L_{p}^{X''} L_{L}^{X'''} (H_c + H_t)^{X''''}$$  \hspace{1cm} (4-14)

Figure 4-21 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Cowell & Achaichia $f$ correlation for $Re_{L_p} < 150$ predicts $f$ more accurately than before. The average deviation of the correlation to the experimental data has improved to 0.75% from +12.19%. The standard deviation of the deviations from the experimental data is 12.72%.

This modified Cowell & Achaichia $f$ correlation is centered on the experimental data and maps the range of the experimental data better. This Cowell & Achaichia $f$ correlation may not be missing a parameter since it captures the phenomena.
Figure 4-21 f vs Re
Modified Cowell Re<150 & Re>150 and Present Data
The Cowell & Achaichia $f$ Correlation for $150 < Re_{Lp} < 3000$:

$$
f = 0.895 (596 Re_{Lp}^{(0.318 \log Re - 2.25)})^{1.07} F_p^{0.22} L_p^{0.25} L_L^{0.33} (H_c + H_t)^{0.26} \quad (4-15)
$$

Figure 4-20 shows this correlation compared to the experimental data. This figure also shows equation 4-13. Inspection of the figure yields the break in the Cowell & Achaichia correlations at $Re_{Lp}$=150 matches the location of the increase of scatter in the experimental data. The Cowell & Achaichia correlations also handle the slight knee in the data at $Re_{Lp}$=900 quite well. The Cowell & Achaichia $f$ correlation for this range of $Re_{Lp}$ under predicts $f$ in much the same way as Davenport under predicted $f$. The average deviation of the correlation to the experimental data is -23.59%. The standard deviation of the deviations from the experimental data is 10.60%.

The Modified Cowell & Achaichia $f$ Correlation for $150 < Re_{Lp} < 3000$:

$$
f = A' \left( A'' Re_{Lp}^{(A''' \log Re - A''')} \right)^{A''''} F_p^{A'''''} L_p^{A'''''} L_L^{A'''''} (H_c + H_t)^{A''''''} \quad (4-16)
$$
Figure 4-21 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Cowell & Achaichia $f$ correlation for $150<Re_{Lp}<3000$ predicts $f$ more accurately than before. The modified Cowell & Achaichia $f$ correlation is centered on the experimental data. The average deviation of the correlation to the experimental data has improved to -0.19% from -23.59%. The standard deviation of the deviations from the experimental data is 10.95%.

For reference, Figure 4-22 shows the Cowell & Achaichia and modified Cowell & Achaichia $f$ correlations over the range of $Re_{Lp}<150$ and $150<Re_{Lp}<3000$ plotted versus $Re_{Lp}$. This plot shows the shift in the correlation created by the new correlation coefficients. The modified correlations over both ranges of $Re_{Lp}$ have slopes which are different from the original correlations.

It is interesting to note that had Cowell and Achaichia chosen the same $Re_{Lp}$ break point for $St$ as they did for $f$ ($Re_{Lp}=150$), they would have created a more accurate $St$ correlation. It is also logical to place the low end of the $Re_{Lp}$ valid range for both $St$ and $f$ at the same $Re_{Lp}$ because the observed knee in the data may be flow phenomena based. They both should be governed by the behavior of the flow phenomena in the core.
Figure 4-22: $f$ vs $Re(L_p)$

- Cowell $Re < 150$
- Modified Cowell $Re < 150$
- Cowell $Re > 150$
- Modified Cowell $Re > 150$

$$f = 10.4 Re_{L_p}^{-1.17} F_p^{0.05} L_p^{1.24} L_L^{0.25} (H_c + H_t)^{0.83}$$

$$f = 0.895 (596 Re_{L_p}^{0.318} \log Re - 2.25)^{1.07} F_p^{0.22} L_p^{0.25} L_L^{0.33} (H_c + H_t)^{0.26}$$

$$f = A' \left( Re_{L_p}^{A''} \log Re - A''' \right)^{A''''} F_p^{A'''''} L_p^{A'''''} L_L^{A'''''} (H_c + H_t)^{A''''''}$$

$$f = A' \left( Re_{L_p}^{A''} \log Re - A''' \right)^{A''''} F_p^{A'''''} L_p^{A'''''} L_L^{A'''''} (H_c + H_t)^{A''''''}$$
4.4 Comparison to the Cowell and Achaichia St Correlation with $\Gamma_B$

The Cowell and Achaichia correlation for $St$ was integrated with the flow efficiency determined by the Bellows (1996) correlation. This was done to determine whether the use of $\Gamma_B$ (see equation 2-9) would enhance the ability of the Cowell & Achaichia correlation to predict $St$. $\Gamma_B$ was not used to enhance the Cowell and Achaichia correlations for $f$ since they do not contain a flow efficiency parameter. The equations for the correlations in this chapter are numbered in the same order as they appear in Table 4-1 for consistency:

**The Cowell & Achaichia St Correlation with $\Gamma_B$ and no $Re_{lp}$ restrictions:**

$$St = \Gamma_B 1.18 Re_{lp}^{-0.58}$$  (4-17)

Figure 4-23 shows this correlation compared to the experimental data. Inspection of the figure yields that the Cowell & Achaichia correlation with $\Gamma_B$ exhibits a knee for most of the data and under predicts $St$ for many data point strings.

The correlation does not predict all of the cores within a reasonable amount of error. The slope of the Cowell & Achaichia correlation with $\Gamma_B$ closely resembles the slope of the experimental data after the knee ($Re_{lp}>150$).
Figure 4-23 St vs Re
Cowell (Bellows) w/ No Re(Lp) Restrictions and Present Data
The average deviation of the correlation to the experimental data is -28.41%. The standard deviation of the deviations from the experimental data is 26.46%.

The Cowell & Achaichia \( St \) Correlation with \( \Gamma_B \) does not predict \( St \) well for all cores because of the Bellows flow efficiency. Bellows (1996) tested cores with \( F_p/L_p \) ratios ranging from 2.18 to 3.50. The \( F_p/L_p \) ratios of the present work ranges from 2.10 to 5.29. The \( F_p/L_p \) ratios of the Bellows work ranged from 2.18 to 3.50. The cores tested in the present work with the large \( F_p/L_p \) ratios which were outside Bellows’ range of correlation were responsible for the under predicted cores that are visible in the plot.

The Modified Cowell & Achaichia \( St \) Correlation with \( \Gamma_B \) and no \( Re_{L_p} \) restrictions:

\[
St = \Gamma_B A' \left( Re_{L_p} \right)^{\alpha'}
\]  

(4-18)

Figure 4-24 shows this correlation compared to the experimental data. Inspection of the figure yields that the modified Cowell & Achaichia correlation with \( \Gamma_B \) still exhibits a lack of correlation of \( St \) for some cores. The slope of the modified Cowell & Achaichia correlation with \( \Gamma_B \) was flattened because of the presence of the knee in the range of correlated data.
Figure 4-24 St vs Re
Modified Cowell(Bellows) w/ No Re(Lp) Restrictions and Present Data
If the knee were removed from the range, the correlation would have been fit to the data in the higher \( Re_{Lp} \) range with greater accuracy. This fact makes it not surprising that Cowell & Achaichia limited the valid \( Re_{Lp} \) range of his correlation to \( 75 \leq Re_{Lp} \). The average deviation of the correlation to the experimental data has improved to -13.59\% form -28.41. The standard deviation of the deviations from the experimental data is 30.67\%.

For reference, Figure 4-25 shows the Cowell & Achaichia and modified Cowell & Achaichia \( St \) correlations with \( \Gamma_B \) plotted versus \( Re_{Lp} \). This plot shows the shift in the correlation created by the new correlation coefficients. Note the distinct change in slope created by the Solver trying to fit the Cowell & Achaichia correlation with \( \Gamma_B \) to the entire range of experimental data.

The Cowell & Achaichia \( St \) Correlation with \( \Gamma_B \) for \( Re_{Lp} \geq 75 \):

\[
St = \Gamma_B \, 1.18 \, Re_{Lp}^{-0.58}
\]

(4-19)

Figure 4-26 shows this correlation compared to the experimental data. This figure is identical to Figure 4-23 with the exception of the removal of values of Cowell & Achaichia \( St \) correlation with \( \Gamma_B \) data below \( Re_{Lp} = 75 \).
Figure 4.25 St vs Re

St versus Re(Lp)

- Cowell (Bellows) w/ no Re(Lp) Restrictions
- Modified Cowell (Bellows w/ no Re(Lp) Restrictions

\[ St = \Gamma_B 1.18 Re_{L_p}^{-0.58} \]

\[ St = \Gamma_B A' Re_{L_p}^{A''} \]
Figure 4-26 St vs Re
Cowell(Bellows) Re>75 and Present Data

$St = \Gamma_b 1.18 Re_{Lp}^{-0.38}$
Inspection of the figure yields that the Cowell & Achaichia correlation with $\Gamma_B$ for $Re_{Lp} \geq 75$ still does not predict $St$ rather well. The average deviation of the correlation to the experimental data is -29.74%. This average deviation is actually worse than the deviation for the Cowell & Achaichia correlation with $\Gamma_B$ plotted over the entire measured range of $Re_{Lp}$. The standard deviation of the deviations from the experimental data is 22.98%.

The Modified Cowell & Achaichia $St$ Correlation with $\Gamma_B$ and for $Re_{Lp} \geq 75$:

$$St = \Gamma_B A' Re_{Lp}^{A''}$$  \hspace{1cm} (4-20)

Figure 4-27 shows this correlation compared to the experimental data. This figure clearly shows the best Cowell & Achaichia $St$ correlation with $\Gamma_B$ fit to the experimental data, yet, the overall correlation is still unacceptable. The average deviation of the correlation to the data is now -16.31%. The standard deviation of the deviations from the experimental data is 27.09%.

For reference, Figure 4-28 shows the Cowell & Achaichia with $\Gamma_B$ and modified Cowell & Achaichia with $\Gamma_B St$ Correlation for $Re_{Lp} \geq 75$ plotted versus $Re_{Lp}$. This plot shows the shift in the correlation created by the new correlation coefficients.
Figure 4-27 St vs Re
Modified Cowell(Bellows) Re>75 and Present Data

\[ St = \frac{\Gamma_A}{\psi} \cdot \frac{Re_{lp}}{\psi} \]
Figure 4.28: St vs Re for Cowell (Bellows) Re > 75 and Modified Cowell (Bellows) Re > 75

\[ St = \Gamma_b 1.18 Re_{lp}^{-0.58} \]

\[ St = \Gamma B A' Re_{lp} A'' \]
4.5 New Correlations for $St$ and $f$

New correlations for $St$ and $f$ were the next correlations to be created and compared to the experimental data. The equations for the correlations in this chapter are numbered in the same order as they appear in Table 4-1 for consistency:

$St$ Correlation Based Upon the Davenport Correlation Parameters and $F_p/L_p$ with no $Re_{L_p}$ Restrictions:

$$St = A' Re_{L_p}^{A'} H_L^{A''} \left( \frac{L_L}{H_f} \right)^{A'''} H_f^{A''''} \left( \frac{F_p}{L_p} \right)^{A'''''}$$

(4-21)

The form of this correlation was chosen in order to examine how well the Davenport correlation would have fit the experimental data if Davenport had not dismissed the influence of the fin pitch on $St$.

Figure 4-29 shows this correlation compared to the experimental data. Inspection of the figure yields that this $St$ correlation does not exhibit a knee but predicts $St$ more accurately than the modified Davenport correlation. The average deviation of the correlation to the experimental data is now $+7.48\%$. The standard deviation of the deviations from the experimental data is 18.70\%.
Figure 4-29 St vs Re
Davenport w/ Fp/Lp w/ No Re(Lp) Restrictions and Present Data
119
The scatter plot shows this $St$ correlation slightly over predicting at high $Re_{lp}$. The slope of this $St$ correlation was flattened because of the presence of the knee in the range of correlated data. If the knee were removed from the range, the correlation would have been fit to the data in the higher $Re_{lp}$ range with greater accuracy.

**$St$ Correlation Based Upon the Davenport Correlation Parameters and $F_p/L_p$ for $Re_{lp} \geq 150$:**

$$St = A' Re_{lp}^\alpha H_L^\alpha \left( \frac{L_L}{H_f} \right)^\alpha H_f^\alpha \left( \frac{F_p}{L_p} \right)^\alpha$$  \hspace{1cm} (4-22)

Figure 4-30 shows this correlation compared to the experimental data. This figure clearly shows the best $St$ correlation fit to the experimental data that does not include flow efficiency and shows that Davenport would have produced a better correlation if he had retained the fin pitch. The average deviation of the correlation to the data is now -0.68%. The standard deviation of the deviations from the experimental data is 10.15%.

One core can not be predicted accurately by this correlation. Therefore, correlation still may not capture all values of $St$ for different cores at a given $Re_{lp}$. This is evidence of a missing parameter (possibly flow efficiency) from the form of the correlation.
Figure 4-30 St vs Re
Davenport w/ Fp/Lp Re>150 and Present Data

\[ St = A' Re^{\alpha} \left( \frac{E}{F} \right)^{xH_L H_f L} \]
St Correlation Based Upon the Davenport Correlation Parameters, $F_p/L_p$ and $\Gamma_{ca}$

with no $Re_{L_p}$ Restrictions:

$$St = A' \cdot Re_{L_p}^{A'} \cdot H_f^{A''} \left( \frac{L_f}{H_f} \right)^{A'''} \cdot H_f^{A''''} \left( \frac{F_p}{L_p} \right)^{A'''''} \cdot \Gamma_{ca}^{A'''''}$$ (4-23)

The form of this correlation was chosen in order to improve on the results obtained by equation 4-22. The addition of the flow efficiency parameter to the form of the correlation seemed to be the next logical choice.

Figure 4-31 shows this correlation compared to the experimental data. Inspection of the figure yields that this $St$ correlation exhibits a slight knee and predicts $St$ more accurately than the modified Davenport correlation. The average deviation of the correlation to the experimental data is $+3.45\%$. The standard deviation of the deviations from the experimental data is $15.28\%$.

The scatter plot shows this $St$ correlation slightly over predicting at high $Re_{L_p}$. The slope of this $St$ correlation was flattened because of the presence of the knee in the range of correlated data. If the knee were removed from the range, the correlation would have been fit to the data in the higher $Re_{L_p}$ range with greater accuracy.
Figure 4-31 St vs Re
Davenport w/ Fp/Lp & Cowell w/ No Re(Lp) Restrictions and Present Data

\[ St = \frac{L_L}{H_L} \left( \frac{F_p}{H_f} \right)^{\gamma - 1} \]
St Correlation Based Upon the Davenport Correlation Parameters, $F_p/L_p$ and $\Gamma_B$

with no $Re_{Lp}$ Restrictions:

$$St = A' \cdot Re_{Lp}^{A'} \cdot H_L^{A''} \left( \frac{L_L}{H_f} \right)^{A''} \cdot H_f^{A'''} \left( \frac{F_p}{L_p} \right)^{A''''} \cdot \Gamma_B^{A'''''}$$  \hspace{1cm} (4-24)

Figure 4-32 shows this correlation compared to the experimental data. Inspection of the figure yields that this $St$ correlation exhibits a slight knee and predicts $St$ more accurately than the modified Davenport correlation. The average deviation of the correlation to the experimental data is $+6.15\%$. The standard deviation of the deviations from the experimental data is $20.00\%$. The Bellows flow efficiency does not improve the correlation over the correlation which includes the Cowell & Achaichia flow efficiency when the entire range of $Re_{Lp}$ is considered.

The scatter plot shows this $St$ correlation slightly over predicting at high $Re_{Lp}$. The slope of this $St$ correlation was flattened because of the presence of the knee in the range of correlated data. If the knee were removed from the range, the correlation would have been fit to the data in the higher $Re_{Lp}$ range with greater accuracy.
$St = A' \cdot Re_{Lp}^{A''} \cdot H_L^{A'''} \cdot \left( \frac{L_L}{H_f} \right)^{A'''} \cdot H_f^{A''''} \cdot \left( \frac{F_p}{L_p} \right)^{A'''''} \cdot \Gamma_B^{A'''''}$
\( St \) Correlation Based Upon the Davenport Correlation Parameters, \( F_p/L_p \) and \( \Gamma_{ca} \) for \( Re_{Lp} \geq 150 \):

\[
St = A' Re_{Lp}^A H_L^A \left( \frac{L_L}{H_f} \right)^{A'''} H_f^{A'''} \left( \frac{F_p}{L_p} \right)^{A'''} \Gamma_{ca}^{A''''}\tag{4-25}
\]

Figure 4-33 shows this correlation compared to the experimental data. This figure clearly shows one of the best \( St \) correlation fit to the experimental data. The average deviation of the correlation to the data is now +1.24%. The standard deviation of the deviations from the experimental data is 9.01%. This is only one of two correlations that were found to have a standard deviation of the percent deviations below 10%.

\( St \) Correlation Based Upon the Davenport Correlation Parameters, \( F_p/L_p \) and \( \Gamma_B \) for \( Re_{Lp} \geq 150 \):

\[
St = A' Re_{Lp}^A H_L^A \left( \frac{L_L}{H_f} \right)^{A'''} H_f^{A'''} \left( \frac{F_p}{L_p} \right)^{A'''} \Gamma_{B}^{A''''}\tag{4-26}
\]

Figure 4-34 shows this correlation compared to the experimental data. This figure clearly shows one of the best \( St \) correlation fit to the experimental data. The average deviation of the correlation to the data is now +0.59%. The standard deviation of the deviations from the experimental data is 8.95%.
Figure 4-33 St vs Re

\[ St = A' Re_{L_p}^\alpha H_L^\beta \left( \frac{L_L}{H_f} \right)^\gamma H_f^\delta \left( \frac{F_p}{L_p} \right)^\epsilon \Gamma_{ca}^{\gamma} \]
Figure 4-34 St vs Re
Davenport w/ Fp/Lp & Bellow Re>150 and Present Data
128
This is the other of the two correlations that were found to have a standard deviation of the percent deviations below 10%. The Bellows flow efficiency does improve the correlation over the correlation which includes the Cowell & Achaichia flow efficiency when the range of $Re_{lp}$ ≥ 150 is considered.

This correlation provides the most accurate prediction of $St$ over the entire data set. Therefore, the author suggests using this correlation when attempting to correlate $St$. It provides a $St$ which would predict the heat transfer of a heat exchanger to within ±3% for 60% of the data or to within ±9% for 99% of the data.

**$f$ Correlation:**

\[
f = A' Re^{n'} F_p^{n''} L_p^{n'''} L_L^{n''''} (H_c + H_t)^{n'''''} H_L^{n''''''}
\]  

(4-27)

The form of this correlation was chosen in order to improve on the results obtained by the Cowell & Achaichia $f$ correlation. The louver height ($H_L$) parameter was added to the form of the correlation for this purpose.

Figure 4-35 shows this correlation compared to the experimental data. This figure clearly shows that about six cores fall below and one above the experimental data. The form of this correlation can not be used to predict $f$ accurately. The average deviation of the correlation to the data is -14.70%. The standard deviation of the deviations from the experimental data is 32.01%.
**f Correlation for \( \text{Re}_{Lp} \leq 150 \):**

\[
f = A' \text{Re}_{Lp}^\alpha F_p^\beta L_p^\gamma L_L^\delta (H_c + H_t)^\epsilon H_L^\zeta
\]  

(4-28)

Figure 4-36 shows this correlation compared to the experimental data. This figure clearly shows that about six cores fall below and one above the experimental data. The form of this correlation cannot be used to predict \( f \) accurately. The average deviation of the correlation to the data is -12.54%. The standard deviation of the deviations from the experimental data is 32.49%.

**f Correlation for \( \text{Re}_{Lp} \geq 150 \):**

\[
f = A' \text{Re}_{Lp}^\alpha F_p^\beta L_p^\gamma L_L^\delta (H_c + H_t)^\epsilon H_L^\zeta
\]  

(4-29)

Figure 4-36 also shows this correlation compared to the experimental data and the same number of cores fall below and above the experimental data. The form of this correlation cannot be used to predict \( f \) accurately over this range either. The average deviation of the correlation to the data is -9.18%. The standard deviation of the deviations from the experimental data is 28.02%.

The form of the above correlations is not adequate to calculate \( f \) in multi-louver fins. Therefore, louver angle must not have a significant impact on \( f \).
Figure 4-36 f vs Re
Cowell w/ H(L) Re<150 & Re>150 and Present Data

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Other forms of the $f$ correlation were tried. Each iteration of form never matched up to the results of the Cowell & Achaichia correlation. The modified Cowell & Achaichia correlations (equations 4-15 & 4-16) provided much more accuracy and should be used to predict $f$. 
5. Conclusions

This work studied a larger range of values of several parameters to create correlations to predict the air-side $St$ and $f$ within a core with multi-louver fins than any study found in the literature cited by the author. The $St$ and $f$ correlations proposed by Davenport (1980) and Cowell & Achaichia (1988) were significantly improved in this work. Table 4-1 on page 71 lists all the percent deviation statistics discussed in this chapter.

1. The unmodified Davenport $St$ correlation does not accurately predict $St$ for cores with the multi-louver fins used today. A multiple regression analysis was performed on the correlation to create a new, modified Davenport $St$ correlation that predicts $St$ with greater success. The average deviation was improved from -24.55\% to +7.48\% over the entire $Re_{Lp}$ range and was improved from -24.37\% to +1.29\% for $300 \leq Re_{Lp} \leq 4000$.

The modified Davenport equation did not predict the range of experimental data for each $Re_{Lp}$. It is assumed that a parameter was missing from the correlation. The limited range of fin pitch in the experimental data collected by Davenport may have led him to conclude that fin pitch does not significantly effect the values of the $St$ or $f$. This was proven false for the larger range of fin pitch in this work.
The unmodified Davenport $f$ correlations do not accurately predict $f$ for within this study. A multiple regression analysis was also performed on the correlations to create new, modified Davenport $f$ correlations that can be used to predict the $f$ with greater success. The average deviation was improved from -44.86% to +5.99% for $70 \leq \text{Re}_{Lp} \leq 900$ and was improved from -42.93% to +1.75% for $1000 \leq \text{Re}_{Lp} \leq 4000$.

2. The unmodified Cowell & Achaichia $St$ correlation also does not accurately predict $St$ for the cores within this study. A new, modified Cowell & Achaichia $St$ correlation was created that predicts the $St$ with greater success. The average deviation was improved from +6.44% to +3.92% over the entire $Re_{Lp}$ range and was improved from +2.10% to +1.22% for $75 \leq Re_{Lp}$. The standard deviations of the percent deviations was improved from 22.25% to 15.47% and 15.04% to 12.72% respectively.

The unmodified and modified Cowell & Achaichia $St$ correlations have been shown to predict $St$ more accurately than the unmodified and modified Davenport correlations, respectively. However, the modified Cowell & Achaichia equation did not predict the range of experimental data for each $Re_{Lp}$. It is assumed that a parameter was missing from the correlation. The omission of the louver length to fin height ratio stands out as a possible parameter. The Cowell & Achaichia correlation would have been more accurate if $Re_{Lp} \geq 150$ had been chosen instead of $Re_{Lp} \geq 75$. 
The unmodified Cowell & Achaichia $f$ correlations also do not accurately predict $f$ for all the cores in this study. A multiple regression analysis was also performed on these correlations to create new, modified Cowell & Achaichia $f$ correlations that can be used to predict the $f$ with greater success. The average deviation was improved from $+12.19\%$ to $+0.75\%$ for $Re_{Lp}\leq 150$ and was improved from $-23.59\%$ to $-0.19\%$ for $150\leq Re_{Lp}$.

The modified Cowell & Achaichia correlations are highly recommended over all other correlations tested for the prediction of $f$. The standard deviations of the percent deviations was improved from $53.89\%$ to $12.72\%$ for $150\leq Re_{Lp}$. The standard deviations of the percent deviations was maintained around $10.5\%$ for $150\leq Re_{Lp}$.

3. The Cowell & Achaichia $St$ correlation integrated with the Bellows flow efficiency ($\Gamma_B$) has been shown to predict the experimental $St$ worse than any correlation discussed body of work. A multiple regression analysis was performed on the equation to create a new, modified Cowell & Achaichia with $\Gamma_B$ $St$ correlation which was still unacceptable. This correlation is not recommended for the prediction of air-side $St$. The average percent deviation of the best of these correlations was improved from $-28.41\%$ to $-13.59\%$ with the standard deviation of the deviations maintained degrading to $27.09\%$ from $22.98\%$. 

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4. New correlations for $St$ were developed which utilized the parameters used by Davenport (1980) such as Reynolds number based on the louver pitch, the louver height, the louver length and the fin height. The fin pitch and the Cowell & Achaichia (1988) or Bellows (1996) flow efficiency parameters were integrated with the Davenport parameters to create a correlation which is superior to those proposed by Davenport (1980) and Cowell & Achaichia (1988). The average percent deviation of this correlation for $150 \leq Re_{lp}$ was $+0.59\%$ with a standard deviation of the deviations of $8.95\%$ to the experimental data.

5. The forms of the proposed $f$ correlations were not adequate to calculate $f$ in multi-louver fins. The modified Cowell & Achaichia $f$ correlations provided greater accuracy and should be used to predict $f$. The average percent deviation of the best of these correlations was $-9.18\%$ with a standard deviation of the deviations of $28.02\%$.

The new $St$ correlation which utilizes the Davenport parameters, fin pitch to louver pitch ratio and Bellows’ flow efficiency predicted $St$ more accurately than any form of correlation examined in this work. The average percent deviation of the best of these correlations was $+0.59\%$ with a standard deviation of the deviations of $8.95\%$.

This provides a $St$ which would predict the heat transfer of a heat exchanger to within $\pm3\%$ for $60\%$ of the data or to within $\pm9\%$ for $99\%$ of the data.
The modified Cowell and Achaichia correlations predict $f$ more accurately than any form of correlation examined in this work. This includes new forms examined by the author. The modified Cowell and Achaichia $f$ correlation valid over $Re_{Lp}<150$ had an average percent deviation of $+0.75\%$ and standard deviation of $12.72\%$. The modified Cowell and Achaichia $f$ correlation valid over $150<Re_{Lp}<3000$ had an average percent deviation of $-0.19\%$ and standard deviation of $10.95\%$.

With the use of experimental data which contained a wide range of fin pitch (2.00 - 4.00 mm), the author has created correlations which more accurately predict $St$ and $f$ than any correlation other correlation tested in this work. Fin pitch, louver pitch, louver height, louver length, fin height and the Reynolds number based on the louver pitch were identified as parameters which are required to accurately predict both air-side heat transfer and pressure drop in automotive radiator cores with multi-louver fins.
6. Recommendations for Future Work

This investigative work should guide future researchers into the following topics of work:

1. Use a larger data set to create more accurate correlations for heat transfer and pressure drop which are centered and have a standard deviation of the deviations to the experimental data below 5%. Determine which parameters are missing from the correlations and what forms the correlations must take in order to reach this goal.

2. Investigate the accuracy of the Sahnoun & Webb (1992) analytical model for multi-louver fins. Verify that it can predict heat transfer and pressure drop as accurately as current empirical correlations.
References


Appendix A - Basic Heat Exchanger Theory

This appendix provides the reader with a basic understanding of heat exchanger design principles and basic heat transfer and pressure drop parameters discussed throughout this work.

A.1 Heat Exchangers

Shah (1991) defines a heat exchanger as “a device which is used for transfer of internal thermal energy between two or more fluids at differing temperatures”. Heat exchangers are used whenever there is a need to recover or reject heat from a system.

An example of a heat recovery application is a heat exchanger which employs hot gases billowing out of a smoke stack to heat water destined for a boiler. A heat exchanger in this application collects waste heat and uses it to reduce the cost of producing steam.

An example of a heat rejection application is a heat exchanger employed to remove heat from a mechanical system in order to prevent a premature mechanical breakdown. Automotive radiators perform this function.
A.2 Heat Exchanger Design

The topic of heat exchanger design can be divided into two categories: rating and sizing. Rating refers to the calculation of both the heat transfer and pressure drop performance of an existing heat exchanger. Sizing refers to the determination of the size and geometry of a heat exchanger to transfer a required amount of heat between two or more fluids. In industry, this required amount of heat to be transferred is referred to as the “heat duty” of the heat exchanger.

The sizing process is used more often in industry than the rating process since designers usually are given the required heat duty of an application and asked to provide a heat exchanger to meet it.

A methodology of the heat exchanger sizing design process is shown in Figure A-1. Kays and London (1984) use this model to point out three major inputs which are required to design a heat exchanger. These inputs are the problem specifications, physical properties of the fluids within a heat exchanger and the surface characteristics of a heat exchanger.

The problem specification provides the required heat duty of a heat exchanger and estimates the input fluid and flow conditions to a heat exchanger.
Figure A-1 Methodology of Heat Exchanger Design
The physical properties of the fluids (density, viscosity, etc.) can be determined from the fluid conditions (temperature, pressure, etc.) provided by the problem specification.

The surface characteristics of a heat exchanger determine the ability of a heat exchanger surface geometry to transfer heat or resist fluid flow. These abilities are quantified in non-dimensional parameters and are determined through experimental study.

The importance of knowing the surface characteristics of a given heat exchanger geometry is now evident. A knowledge of these characteristics is necessary to accurately size a heat exchanger for a given heat duty.

**A.3 Heat Exchanger Thermal Circuit**

Heat flow through a heat exchanger can be compared to the travel of an electric current through a circuit. Heat transfer is energy driven to motion due to a difference in temperature between two points. Electricity is energy driven to motion due to a difference in voltage between two points. Heat transfer and electricity are both inhibited by some form of resistance. This analogy is seen by comparing the equation for heat transfer to Ohm's Law for electricity.

\[
q = UA\Delta t = \frac{\Delta t}{(UA)^{-1}} = \frac{\Delta t}{R_e}
\]  

(A-1)
Electricity: \[ i = \frac{\Delta E}{R} \] (A-2)

The parameters in equations A-1 and A-2 are defined in Table A-1. Note that the current, potential, resistance and conductance parameters are all analogous.

This analogy allows heat transfer to be depicted using a circuit diagram. Figure A-2 shows a circuit diagram of heat transfer within a heat exchanger (Kays & London, 1984).

The thermal circuit is verified by the following energy conservation equation:

\[ dq = q' dA = -C_h dt_h = -C_c dt_c \] (A-3)

where \[ C = Wc_p \] (A-4)

and \( C \) is the heat capacity rate of the fluid, \( c_p \) is the specific heat of a fluid at constant pressure, \( t \) is the temperature of the fluid, and the \( h \) and \( c \) subscripts refer to the hot and cold fluids, respectively. The heat capacity rate is the “amount of heat in Joules that must be added to or extracted from the fluid stream per second to change its temperature by 1 \(^\circ\)C” (Shah, 1991). The integration of equation A-3 yields:

\[ q = \int_A q' dA = C_h (t_{h,i} - t_{h,o}) = C_c (t_{c,o} - t_{c,i}) \] (A-5)

Subscripts \( i \) and \( o \) refer to the inlet and outlet of the heat exchanger, respectively.
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Heat Transfer</th>
<th>Electricity</th>
</tr>
</thead>
<tbody>
<tr>
<td>current</td>
<td>$q - W, \text{BTU/hr}$</td>
<td>$i - \text{ampere, A}$</td>
</tr>
<tr>
<td>potential</td>
<td>$\Delta t - ^\circ C, ^\circ F$</td>
<td>$E - \text{volts, V}$</td>
</tr>
<tr>
<td>resistance</td>
<td>$R_0 - ^\circ C/W, (^\circ F \text{hr})/\text{BTU}$</td>
<td>$R - \text{ohms, } \Omega, \text{V/A}$</td>
</tr>
<tr>
<td>conductance</td>
<td>$UA - W/^\circ C, \text{BTU/(hr }^\circ F)$</td>
<td>$G - \text{mhos, S, A/V}$</td>
</tr>
</tbody>
</table>

Table A-1 Analogy Between Heat Transfer and Electricity
Figure A-2 Heat Exchanger Thermal Circuit
Equation A-1 can be written as:

\[ q'' = \frac{dq}{dA} = U \Delta t \]  \hspace{1cm} (A-6)

therefore

\[ \int \frac{dq}{\Delta t} = \int UdA \]  \hspace{1cm} (A-7)

finally

\[ q = U_m A \Delta t_m \]  \hspace{1cm} (A-8)

where \( U_m \) is the mean overall heat transfer coefficient and \( \Delta t_m \) is the effective mean temperature difference. Since \( U \) is considered constant in most literature, \( U_m \) often is written as \( U \). From equation 1-7, \( U \) and \( \Delta t_m \) are defined as:

\[ U_m = \frac{1}{A} \int UdA \]  \hspace{1cm} (A-9)

\[ \Delta t_m = \frac{1}{q} \int \frac{dq}{\Delta t} \]  \hspace{1cm} (A-10)

In Figure A-2, note that there are three resistances to heat transfer shown in the circuit. is

The hot-side resistance due to fluid and flow properties and surface geometry is \( R_h \), the cold-side resistance due to fluid and flow properties is \( R_c \), and surface geometry and the thermal resistance of the heat exchanger wall is \( R_w \).
Shah (1991) includes the resistance due to the collection of foreign material on the hot-side and/or cold-side surfaces of the heat exchanger. This foreign material is referred to as “fouling”. Fouling is not included here for simplicity. Its effect can be incorporated through an effective heat transfer coefficient on each side of the heat exchanger.

The heat transfer rate equation for the thermal circuit is:

\[ q = \frac{t_h - t_{w,h}}{R_h} = \frac{t_{w,h} - t_{w,c}}{R_w} = \frac{t_{w,c} - t_c}{R_c} \]  \hspace{1cm} (A-11)

where (for \( R_h \) & \( R_c \))

\[ R = \frac{1}{\eta_o hA} \]  \hspace{1cm} (A-12)

where \( \eta_o \) is the total surface temperature effectiveness of an extended fin surface and \( h \) is the convective heat transfer coefficient. As with an electric circuit, the thermal circuit can be written in terms of the overall resistance of the circuit:

\[ q = \frac{t_h - t_c}{R_o} = UA(t_h - t_c) \]  \hspace{1cm} (A-13)

where

\[ R_o = R_h + R_w + R_c \]  \hspace{1cm} (A-14)

and

\[ R_o = \frac{1}{UA} = \frac{1}{(\eta_o hA)_h} + R_w + \frac{1}{(\eta_o hA)_c} \]  \hspace{1cm} (A-15)
Equation A-15 clearly shows that if any one of \( R_h, R_w \) or \( R_c \) is significantly higher than the other two, that resistance would account for a high percentage of \( R_o \). Therefore, that resistance would be labeled the controlling or "dominant" resistance. The dominant resistance of a heat exchanger is always chosen as the first target for enhancement by a designer. By choosing to decrease or modify the dominant resistance, the designer gets the largest possible decrease in overall resistance.

**A.4 \( \varepsilon \)-NTU Method**

This method quantifies the total heat transfer in a heat exchanger as follows:

\[
q = \varepsilon C_{\text{min}} (t_{h,i} - t_{c,i})
\]

where \( \varepsilon \) is the effectiveness of a heat exchanger and \( C_{\text{min}} \) is the smaller of the \( C_h \) or \( C_c \) values. The effectiveness will be shown as a function of \( C^*, \) NTU and the heat exchanger flow arrangement in this section.

**A.4.1 Effectiveness of a Heat Exchanger (\( \varepsilon \))**

The effectiveness of a heat exchanger is defined as the ratio of the actual heat transfer through the heat exchanger to the maximum possible heat transfer.

\[
\varepsilon = \frac{q}{q_{\text{max}}}
\]
where
\[ q_{\text{max}} = C_{\text{min}} (t_{h,i} - t_{c,i}) = C_{\text{min}} ITD \] (A-18)

and \( ITD \) stands for the “inlet temperature difference” of a heat exchanger. Combining equations A-17 and A-18 yields:

\[ \varepsilon = \frac{C_h (t_{h,i} - t_{h,o})}{C_{\text{min}} (t_{h,i} - t_{c,i})} = \frac{C_c (t_{c,o} - t_{c,i})}{C_{\text{min}} (t_{h,i} - t_{c,i})} \] (A-19)

Combining equations A-8 and A-18 yields:

\[ \varepsilon = \frac{UA \Delta t_m}{C_{\text{min}} ITD} \] (A-20)

A.4.2 Heat Capacity Rate Ratio (\( C^* \))

The heat capacity rate ratio, \( C^* \), is defined as the ratio of \( C_{\text{min}} \) to \( C_{\text{max}} \).

\[ C^* = \frac{C_{\text{min}}}{C_{\text{max}}} \] (A-21)

This ratio will always range between 0 and 1. Therefore, from equation A-5, the fluid with the minimum heat capacitance rate will always undergo a greater temperature change than the fluid with the maximum heat capacitance rate.
A.4.3 Number of Transfer Units (NTU)

The Number of Transfer Units (NTU) is defined as the ratio of the overall conductance to the smaller heat capacity rate as follows:

\[ NTU = \frac{UA}{C_{\min}} = \frac{1}{C_{\min}} \int_{A} U dA \]  \hspace{1cm} (A-22)

Substituting \( UA \) from equation 1-22 into equation 1-15 yields:

\[ NTU = \frac{1}{C_{\min}} \left[ \frac{1}{\left( \eta_o h A \right)_h} + \frac{1}{R_w} + \frac{1}{\left( \eta_o h A \right)_c} \right] \]  \hspace{1cm} (A-23)

The \( NTU \) designates the “thermal size” of a heat exchanger while the surface area designates the “physical size” of a heat exchanger. \( NTU \) takes the physical size, the surface characteristics and the heat capacity rates (dependent on fluid and flow rate) of a heat exchanger into account. Therefore, a large heat exchanger with normal surface characteristics may not have as high of an \( NTU \) as a smaller heat exchanger with enhanced surface characteristics at identical heat capacity rates.
A.4.4 Heat Exchanger Flow Arrangements

The $\varepsilon$ versus $NTU$ relationship has been calculated for many flow arrangements. Shah (1991) lists fourteen (14) relationships in his course notes. For this work, only the $\varepsilon$-NTU relationship for a crossflow, unmixed fluid heat exchanger will be discussed. This flow arrangement was chosen because it is the flow arrangement used in many automotive radiators and condensers. These products are discussed in Appendix B.

The definition of a crossflow heat exchanger is one in which one fluid flows perpendicular to the path of another fluid. The unmixed fluids term describes the fact that each fluid in the heat exchanger does not mix with the entire fluid flow on any given side of the heat exchanger. This is accomplished by forcing the fluids to flow through ducts or channels.

Kays and London (1984) define the $\varepsilon$-$NTU$ relationship for a crossflow heat exchanger with unmixed fluids as follows:

$$\varepsilon = 1 - e^{-NTU}$$

(A-24)

Figure A-3 shows a plot of the $\varepsilon$-$NTU$ relationship for various values of $C^*$. 
Figure A-3 ε-NTU Relationship for Crossflow Heat Exchangers with Unmixed Fluids

\[ N_{tu} = \frac{AU}{C_{min}} \]
A.5 Air-Side Convective Heat Transfer Coefficient

The convective heat transfer coefficient, $h$, must be obtained in order to calculate the thermal resistance due to the hot or cold side of a heat exchanger (see Equation A-12). The value of $h$ can be obtained using a variety of dimensionless parameters. These dimensionless parameters include the Nusselt number ($Nu$), Stanton number ($St$) and Colburn $j$ factor ($j$).

Shah and Bhatti (1988) define the Nusselt number ($Nu$) as "the ratio of the convective conductance $h$ to the pure molecular thermal conductance $k/D_h$":

$$
Nu = \frac{hD_h}{k} = \frac{q''D_h}{k(t_w - t_m)}
$$

where $k$ is the thermal conductive coefficient of the fluid, $q''$ is the heat flux (see equation A-6), $t_w$ is the wall temperature of the heat exchanger and $t_m$ is the mean temperature of the fluid. $D_h$ is the hydraulic diameter of the duct through which the fluid flows.

The hydraulic diameter is a characteristic length. A characteristic length is a dimension chosen to create the dimensionless relationships used to predict the heat transfer and pressure drop in a heat exchanger. The choice of the characteristic length is extremely important for predicting the dimensionless parameters properly and is based on the behavior of the flow.
For flow through a duct, the chosen characteristic length is the $D_h$. For flow over a flat plate, the chosen characteristic length is the length of the plate in the direction of the flow. Duct and flat plate flow will be fully discussed later in this chapter.

The air-side $D_h$ for a heat exchanger with a tube and center core (Shah & Bhatti, 1988) is:

$$D_h = \frac{4A_c}{P} = \frac{F_p H_c}{2F_p + 2\sqrt{(F_p)^2 + \left(\frac{1}{2} H_c\right)^2}}$$  \hspace{1cm} (A-26)

where $A_c$ is the cross-sectional area and $P$ is the perimeter of the duct.

Shah and Bhatti (1988) define the Stanton number ($St$) as "the ratio of the convective heat transfer (per unit duct surface area) to amount virtually transferable". $St$ does not depend on any characteristic length.

$$St = \frac{h}{Gc_p} = \frac{Nu}{RePr}$$  \hspace{1cm} (A-27)

Shah and Bhatti (1988) define the Reynolds number ($Re$) as "proportional to the ratio of the flow momentum rate (inertia force)". It is considered the flow modulus and is used as a dimensionless parameter to characterize the flow.

$$Re = \frac{\rho m u_m D_h}{\mu} = \frac{GD_h}{\mu}$$  \hspace{1cm} (A-28)
where $\rho_m$ is the mean density of the fluid, $u_m$ is the mean fluid velocity, $\mu$ is the viscosity of the fluid and $G$ is the mass velocity.

\[ G = \rho_m u_m \quad (A-29) \]

Shah and Bhatti (1988) define the Prandtl number ($Pr$) as “the ratio of momentum diffusivity to thermal diffusivity of the fluid”. It is considered the fluid property modulus.

\[ Pr = \frac{\mu c_p}{k} \quad (A-30) \]

Shah and Bhatti (1988) define the Colburn $j$ factor ($j$) as “the modified Stanton number to take into account the moderate variations in the Prandtl number for $0.5 \leq Pr \leq 10.0$ in turbulent flow”.

\[ j = StPr^{\frac{\nu}{5}} = \frac{Nu}{RePr^{\frac{\nu}{5}}} \quad (A-31) \]
St and j are commonly correlated to various geometric variables and Re in order to predict heat transfer in a heat exchanger.

### A.6 Pressure Drop

The air-side pressure drop through a heat exchanger is dependent on the geometry of the heat exchanger core. Shah (1991) defines the pressure drop through the air-side of a tube and center heat exchanger as:

\[
\Delta p = \frac{G^2}{2g_c\rho_l} \left[\left(1 - \sigma^2 + K_c\right) + 2\left(\frac{\rho_l}{\rho_o} - 1\right) + f \left(\frac{4W_c\rho_l}{D_h\rho_m} - (1 - \sigma^2 - K_c)\frac{\rho_l}{\rho_o}\right)\right]
\]  

(A-32)

where \(g_c\) is the proportionality constant from Newton's second law and is equal to 1 for SI units. The ratio of the free flow area \(A_o\) to the frontal area \(A_{fr}\) of a heat exchanger is \(\sigma\).

\[
\sigma = \frac{A_o}{A_{fr}} = \frac{H_{NF}W_{HDR} - N_fH_fW_{HDR} - 2N_{conv}H_f b_c (N_t + 1)}{H_{NF}W_{HDR}}
\]  

(A-33)

where \(H_{NF}\) and \(W_{HDR}\) are the height and width of the core, respectively. They are depicted in Figure 3-1. \(N_{conv}\) represents the number of convolutions in the air center from one header to the opposite header of the core. This parameter is defined in Figure 1-6. \(N_t\) is the number of tubes in the heat exchanger core. \(b_c\) is the material thickness of the multi-louver fin.
In equation A-32, $K_c$ and $K_e$ are the entrance and exit flow contraction loss coefficients, respectively. Values for these parameters can be obtained from Figure A-4 for tube and center heat exchangers with triangular convoluted fins (Shah, 1991). In Figure A-4, $N_R$ is the symbol Shah uses for Reynolds number ($Re$).

$W_c$ is the width of the air center. The width of the air center is synonymous with the depth of the core. $W_c$ is shown in Figure 1-5 and Figure 1-7.

Finally, $f$ is the Fanning friction factor. Shah and Bhatti (1988) define $f$ as “the ratio of the wall shear (skin frictional) stress to the flow kinetic energy per unit volume”. The Fanning friction factor is commonly correlated to various geometric variables and $Re$ to predict the pressure drop penalty in a heat exchanger.
Figure A-4 Pressure Loss Coefficients for a Core with Triangular Ducts  
A.7 Recommended Sources for Further Information

This appendix has briefly discussed heat exchanger design principles and basic heat transfer and pressure drop parameters. If a more thorough discussion on these topics is desired by the reader, the author suggests reading “Compact Heat Exchangers” by Kays and London (1984) or “Heat Exchanger Design” by Shah (1991).
Appendix B - Automotive Applications

This appendix provides the reader with an understanding of the types of heat exchangers used in automotive applications. Radiator and heater core applications are detailed in Chapter 1.

B.1 Classifications of Heat Exchangers in Automotive Applications

A heat exchanger can be classified by the following: whether or not the fluids within the heat exchanger contact each other, the number of fluids it utilizes, its surface compactness, its construction type, its fluid flow arrangement and its heat transfer mechanisms. Figure B-1 shows various classifications of heat exchangers.

Heat exchangers in automotive applications tend to be indirect contact heat exchangers. These types of heat exchangers are sometimes labeled “recuperators” (Shah, 1991). Indirect contact heat exchangers are simply heat exchangers which do not allow the hot and cold fluids to mix. In this case, air is not allowed to mix with either engine coolant or refrigerant.
Figure B-1 Classification of Heat Exchangers
Shah (1991) pp. 3
All automotive heat exchangers utilize only two fluids. In radiators, heater cores and condensers, the cold fluid is air while the warm fluid is either engine coolant or refrigerant. In evaporators, the cold fluid is refrigerant and the warm fluid is air. Figure B-2 depicts an example of an engine cooling and air conditioning system of an automobile. The heat exchanger components of these systems will be described in detail in later sections.

Automotive heat exchangers tend to be compact. By definition, a heat exchanger is compact if the ratio of the total available heat transfer surface area to the heat exchanger volume is greater than or equal to 700 m²/m³. Figure B-3 depicts the “compactness” of heat exchangers in various applications. It shows that automotive radiators fall in the middle of the “compactness” spectrum.

Automotive heat exchangers are usually extended surface with plate and fin geometry, or extended surface with tube and fin geometry.

An extended surface heat exchanger is one which uses protrusions from the primary heat exchanger surface to enhance heat transfer by providing more available surface area to displace heat. These protrusions are often called “fins.”
Small Car System
FIG. 11

Figure B-2 Example of the Engine Cooling and HVAC Systems of an Automobile
Courtesy of Delphi Harrison Thermal Systems, Lockport NY
"COMPACTNESS?" A MATTER OF DEGREE

**Heat Transfer Surface Area Density Spectrum of Applications**

Shah (1991) pp. 8

- **COMPACT**
  - Cryogenic H.E.
  - Gas Turbine Rotary Regenerators
  - Automotive Radiators
  - Matrix Types, Wire Screen Sphere Bed, Corrugated Sheets
  - Human Lungs

Flow

Plain Tubular, Shell-and-Tube H.E.

Strip-Fin and Louvered-Fin H.E.

Plate Heat Exchangers

Hydraulic Diameter $D_h$, mm

Heat transfer surface area density $\beta$, $m^2/m^3$

- $\beta = \frac{2\pi}{X_c X_L D}$
- For $X_c X_L = 1.89$,
  - $\beta = \frac{3333}{D_h (mm)}$, $m^2/m^3$

- $\beta = \frac{4a}{D_h}$ and $\sigma = 0.833$
  - $= \frac{3333}{D_h (mm)}$, $m^2/m^3$
Plate and fin heat exchangers use convoluted fins assembled between parallel plates. Many times, flat tubes with rounded corners are used to sandwich the fins. Figure B-4 shows examples of plate and fin heat exchangers. Automotive radiators, heater cores and condensers can be of tube and center construction shown in the figure.

Tube and fin heat exchangers are constructed with flat fins and round, rectangular or oval tubes which are forced through the fins. Figure B-5 shows examples of tube and fin heat exchangers. Automotive radiators and condensers can be of tube and fin design also.

Automotive heat exchangers can be of cross-flow one pass, two pass or multi-pass designs. Heater cores are usually two-pass. Condensers are usually multi-pass. The term "pass" describes how many times the fluid within the heat exchanger tube (coolant or refrigerant) "passes" the air flow to transfer heat. The number of passes is regulated by baffles or separators which are placed within the heat exchanger tanks. The number of passes can also be regulated by how many times a "serpentine" tube forces the refrigerant to pass the air flow by bending 180° at each end of the condenser core. If the number of passes is increased within a core, the tube-side fluid velocity increases accordingly which in turn will increase the heat transfer and pressure drop.
Figure B-4 Examples of Plate and Fin Heat Exchangers
Shah (1991) pp. 30
Figure B-5 Examples of Tube and Fin Heat
Finally, automotive heat exchangers are either single-phase or two-phase heat exchangers. Radiators and heater cores are exclusively single phase / single phase heat exchangers since the air remains in the gaseous state and the coolant remains in the liquid state as they pass through the heat exchanger. Condensers and evaporators are single phase / two phase heat exchangers since the air remains in the gaseous state but the refrigerant undergoes a phase change between the gaseous state and the liquid state as they pass through the heat exchanger.

Table B-1 summarizes the material presented in this section.

### B.2 Condenser and Evaporator Applications

In order for an air conditioning (A/C) system to function, it must contain the following components: a compressor, condenser, expansion valve (orifice) and evaporator. A schematic representation of an A/C system can be found in Figure B-6.

The system functions as follows: Low temperature, low pressure gaseous refrigerant is compressed to a high pressure inside the compressor which raises the temperature of the gas. Then, high temperature, high pressure gaseous refrigerant enters the condenser. The refrigerant is condensed into the liquid state and cooled within the condenser. The high pressure, low temperature liquid refrigerant then travels through an expansion valve which lowers both the pressure and the temperature of the refrigerant.
Climate Control System

Figure B-6 Air Conditioning (Climate Control) System
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
<table>
<thead>
<tr>
<th></th>
<th>Radiator</th>
<th>Heater Core</th>
<th>Condenser</th>
<th>Evaporator</th>
</tr>
</thead>
<tbody>
<tr>
<td>indirect/direct contact</td>
<td>indirect</td>
<td>indirect</td>
<td>indirect</td>
<td>indirect</td>
</tr>
<tr>
<td>indirect/direct heat transfer</td>
<td>direct</td>
<td>direct</td>
<td>direct</td>
<td>direct</td>
</tr>
<tr>
<td>number of fluids.</td>
<td>2</td>
<td>2</td>
<td>2</td>
<td>2</td>
</tr>
<tr>
<td>surface compactness</td>
<td>compact</td>
<td>compact</td>
<td>compact</td>
<td>compact</td>
</tr>
<tr>
<td>construction type(s)</td>
<td>ext. surface tube &amp; fin or ext. surface tube &amp; fin</td>
<td>ext. surface tube &amp; center or ext. surface tube &amp; fin</td>
<td>ext. surface tube &amp; center or ext. surface tube &amp; fin</td>
<td>ext. surface plate and fin drawn cup</td>
</tr>
<tr>
<td>number of passes (tube-side)</td>
<td>1 or 2 pass</td>
<td>2 pass</td>
<td>multi-pass</td>
<td>1 or 2 pass</td>
</tr>
<tr>
<td>heat transfer mechanisms</td>
<td>1 Phase/1 Phase</td>
<td>1 Phase/1 Phase</td>
<td>1 Phase/2 Phase</td>
<td>1 Phase/2 Phase</td>
</tr>
</tbody>
</table>

Table B-1 Summary of the Classifications of Various Automotive Heat Exchangers
The low temperature and low pressure liquid refrigerant enters the evaporator. The refrigerant is evaporated into the gaseous state by absorbing heat from the air stream. The refrigerant then enters an "accumulator / dehydrator" (A/D) which removes any remaining liquid refrigerant and other substances from the flow. Finally, the cycle repeats itself.

The condenser is necessary to remove heat from the compressed gaseous refrigerant of the system. Figures B-7, B-8 and B-9 show examples of tube and center, serpentine tube and center and tube and fin condensers, respectively. All three types of condensers are depicted in Figure B-10.

The condenser is placed before the radiator at the front end of a vehicle (for reference see Figures B-2 and B-6). The heat removed from the gaseous refrigerant as it condenses is carried by the air flow which passes through the condenser to the radiator. The positioning of these two heat exchangers forces a designer to consider balance the two designs for optimum heat flow in both systems.
Figure B-7 Example of a Tube and Center Condenser
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure B-8 Example of a Serpentine Tube and Center Condenser
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure B-9 Example of a Tube and Fin Condenser
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY

B-15
Figure B-10 Three Examples of Plate and Fin Condensers
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
The evaporator is placed inside the Heating Ventilation and Air Conditioning (HVAC) module of the vehicle. Figure B-11 shows an example of an evaporator. Figures B-12 shows three examples of an evaporator with different pass configurations. Figure B-13 depicts how an HVAC module regulates air flow through the evaporator and/or heater core (described in Chapter 1) and delivers the cooled/warmed air to the passenger compartment of a vehicle. Figures B-14 and B-15 show a detailed HVAC module schematic and a photo of an actual module, respectively. The evaporator is necessary to remove heat from the air stream which travels through its core. The cooled air stream is redirected to the passengers of the vehicle through the HVAC module and its duct work.

B.2.1 Condenser and Evaporator Heat Transfer Enhancement

Every year, the automotive industry places stricter space and mass constraints on all components used on a vehicle. Heat transfer enhancement has become the primary tool used to meet the reductions in these parameters required by the demands of the automotive industry. For a condenser, the dominant thermal resistance (discussed in Section 1.3) is due to the hot-side fluid, flow and surface properties. The hot-side of a condenser is the refrigerant side. For this reason, heat transfer enhancement research for condensers has been primarily directed towards the refrigerant side. Enhancements considered on the refrigerant side include optimization of the number of tubes in each pass of a multi-pass condenser and modifications to the condenser internal tube geometry.
Figure B-11 Example of a Plate and Fin Evaporator
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Figure B-12 Three Examples of Plate and Fin Evaporators
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY
Air Mix System

Figure B-13 Schematic of the Functions of an HVAC Module
 Courtesy of Delphi Harrison Thermal Systems, Lockport, NY

B-20
Figure B-14 Detailed Schematic of an HVAC Module
Courtesy of Delphi Harrison Thermal Systems, Lockport, NY

B-21
Figure B-15 Photo of an HVAC Module

Courtesy of Delphi Harrison Thermal Systems, Lockport, NY

B-22
Appendix C - Detailed Literature Review

This appendix provides the reader with a history of the theories presented in the literature to describe the dimensionless heat transfer and pressure drop parameters of the surface of a core containing multi-louver fins. The works of Davenport (1980), Cowell & Achaichia (1988) and Bellows (1996) will not be discussed here since they have already been discussed in Chapter 2.

C.1 Multi-Louver Fin Correlation History

Multi-louver fins are used in automotive heat exchanger applications in order to minimize the space and mass requirements of the heat exchanger. This improves the cost, performance and manufacturability of an automobile.

The genesis for the multi-louver fin was the plain fin. The air-side heat transfer and pressure drop characteristics of the plain fin could easily be calculated by treating the air-side as a series of triangular ducts. The characteristic length used to determine all of the air-side dimensionless parameters ($Re, f, St, j, Nu$) is the hydraulic diameter.
When the multi-louver fin was first introduced in the 1950's, researchers theorized that the air-flow through a core with this fin was similar to the "duct flow" exhibited in a core with plain fins. This duct flow paradigm led researchers to theorize that multi-louver fins enhance heat transfer by merely turbulating the flow within each duct. This theory was challenged by the work of Beauvais (1965), Wong & Smith (1974) and Davenport (1980).

C.2 Air-Side Heat Transfer Enhancement Using Vortex Generators

Despite the work of Beauvais in 1965 (described in Section 2.6), researchers continued to develop turbulator theories related to various types heat exchanger fins. Some researchers tried to correlate turbulence theories to the multi-louver fin. This section will summarize the experiments and theories of researchers within this circle.

Edwards and Alker (1974) investigated the effect of various protrusions into the air-side flow on heat transfer. The protrusions were attached to the lower wall of a test tunnel in order to turbulate the air-flow. The lower wall of the test tunnel was heated to 69°C and the inlet air was maintained at 24°C. The local surface temperature of the lower wall was measured by taking spot temperature readings using a luminescent phosphorous technique. Effective $h$ values were calculated from the average wall and air temperatures from each test run. These $h$ values were used to form a ratio to $h$ values calculated with no protrusions in the test tunnel.
Interpolation of the data presented by the authors in the plot of \( j \) versus \( Re \) showed an increase in \( j \) of 50\% to 60\% in test samples that had the protrusions. The enhancement was dependent on the position and size of the protrusion.

Russel, Jones and Lee (1982) attempted to increase the local heat transfer coefficients of tube and fin heat exchangers by turbulating the flow. Figure C-1 shows an example of their test sections with louvers cut parallel to the fin. Flow visualization was used to view the vortices created by the test sections. The test sections were then heated and placed into an air-flow to obtain heat transfer data. Figure C-2 shows the most productive louver orientation which had louver height values that were half the distance between the fins.

The conclusion of this study was that louvers cut and formed parallel to the fin can enhance heat transfer by 30\% to 60\% by turbulating the flow.

Fiebig, Kallweit and Mitra (1986) tested two types of delta wing vortex generators. Figure C-3 shows the delta wing configurations. Figure C-4 shows the theorized vortex structure generated by a delta wing. Figure C-5 shows an example test section. As in previous experiments, the test sections were heated in a test tunnel. The local heat transfer coefficients were measured using unsteady liquid crystal thermography and flow visualization techniques were used to view the flow vortices created by the delta wings.
Figure C-1 Example of Test Sample Louvers
Russel, Jones and Lee (1982) pp. 284
Figure C-2 Most Productive Louver Enhancement
Different types of small aspect ratio delta wing vortex generators; (a) delta wing (b) delta winglet-pair.
Leading edge vortex structure generated by a delta wing at angle of attack

Figure C-4 Leading Edge Delta Wing Vortex Structure
Fiebig, Kallweit and Mitra (1986) pp. 2909
Test section with three identical plate-fins representing a fin cascade. The positions of the test equipment for vortex flow visualization and fluid force measurements are also shown: \( d = 40\, \text{mm}; \quad L = 480\, \text{mm}; \quad x_v = 150\, \text{mm} \)
The conclusions of this study verified that delta wing protrusions into a flow field turbulate the flow. The heat transfer performance of the heat exchanger was enhanced by 20% to 60% for fin plate areas that are 60 times the area of the delta wing. Figure C-6 graphically represents the heat transfer results of this study. The researchers also concluded that the highest $j$ to $f$ ratios were found at low delta wing angles.

Turk and Junkhan (1986) studied the affect of “blade pair” vortex generators on heat transfer. Figure C-7 shows an example of the “blade pair” vortex generator. The blade pair consists of two vortex generators which produce counter-rotating vortices.

Figure C-8 shows an example of the test section used in this study. A test tunnel forced cool air through the vortex generators and over heater foils. The free stream and heater foil temperatures were measured with thermocouples.

The conclusions of this study verified that vortex generator alignments which produce counter-rotating vortices enhance heat transfer. Turk and Junkhan found that there is a correlation between the blade spacing to height ratio and the amount of heat transfer enhancement gained.

Takano, Tanasawa and Nishio (1989) studied the effect of a turbulence promoter placed at 45°, 60° and 90° to a surface on heat transfer. The turbulator was placed on a heated wall of a test tunnel. Figure C-9 shows the turbulence promoter.
Local heat transfer coefficient \( \alpha \) along the centerline for a fin with a delta and a rectangular wing at different angles of attack. The largest relative increase is achieved at small angles of attack; \( Re = 1815; \ l = 40 \ mm \).

Figure C-6 Local Heat Transfer Coefficient versus Distance
Fiebig, Kallweit and Mitra (1986) pp. 2911
Vortex generator schematic for two blade pairs set to produce counter rotating vortices.

Figure C-7 Vortex Generator Schematic for Two Blade Pairs
Turk and Junkhan (1986) pp. 2903
Plate construction and coordinate system.

Figure C-8 Vortex Generator Test Plate
Turk and Junkhan (1986) pp. 2904
Figure C-9 Turbulence Promoter
The researchers concluded that the average heat transfer coefficient does not vary much within the range of angles chosen. Figure C-10 shows how the Nusselt number plot for each turbulator nearly overlays the Nusselt plots for the other turbulators. Pressure drop, however, was greatly impacted by increasing turbulator angle of attack. This can be seen in Figure C-10 in the form of the Fanning friction factor.

Finally, Fiebig, Kallweit, Mitra and Tiggelbeck (1991) compared a delta wing, rectangular wing, delta wing-let pair and rectangular wing-let pair to determine which geometry would provide the greatest heat transfer enhancement.

Figure C-11 shows the four geometric configurations tested. The experimental methods used by these researchers included flow visualization by a laser light sheet, the determination of local heat transfer coefficients with unsteady liquid crystal thermography and the measurement of wing drag with a balance. Figure C-12 shows an example of their test section.

The conclusions of this study added to the previous work of Fiebig, Kallweit and Mitra (1986). The delta wings were found to be the most effective enhancement device per unit vortex generator and out-performed the rectangular wings by 20%.
Figure C-10 Takana, Tanasawa and Nishio Experimental Results
Takano, Tanasawa and Nishio (1989) pp. 16
Figure C-11 Four Types of Turbulator Wings Tested by Fiebig et. al
Fiebig, Kallweit, Mitra and Tiggelbeck (1991) pp. 105
Figure C-12 Example of a Test Section Used by Fiebig et al.
Fiebig, Kallweit, Mitra and Tiggelbeck (1991) pp. 106
C.3 Multi-Louver Fin Laminar Flow Theory

As researchers attempted to enhance the air-side heat transfer of a heat exchanger by turbulizing the flow, another group of researchers theorized that louvers that are not parallel to the fins may redirect the flow through the fins of a core. This group would prove that the air-side flow is laminar and a combination of duct and flat plate flow. This section will summarize the experiments and theories of researchers within this circle.

Beauvais (1965) used methods of smoke trace flow visualization on large scale models to demonstrate that louvers actually attempt to realign the air-flow in a direction parallel to their own angle. His work was the foundation of much of the work performed in multi-louver fin laminar flow theory.

Wong and Smith (1973) built upon the work of Beauvais by verifying that large scale models can be used to evaluate louvered surfaces. Figure C-13 and Figure C-14 show the comparisons between the large scale and full scale models for pressure drop and heat transfer, respectively. Local air velocity measurements also confirmed the flow directing properties of the louvers reported by Beauvais.

Tura (1986) studied the effect of the louver angle and length of the turn around rib (see Figure 1-8) on multi-louver fin performance. He used a phase change paint technique on 4:1 scale models of triangular fins to measure $h$. 
Figure C-13 Large Scale Compared to Full Scale Model Δp
Wong and Smith (1973) pp. 3
Figure C-14 Large Scale Compared to Full Scale Model Nu
Wong and Smith (1973) pp. 4
Tura concluded that a high local heat transfer coefficient was present on the leading edge of each louver. Figure C-15 maps the convective heat transfer coefficient measured by Tura. The figure shows developing flow in the first few louvers and a region of degraded performance just beyond the turn-around rib. Tura also noted vortex shedding at high $Re_{Lp}$ (~1400). This $Re_{Lp}$ is well beyond the operating point of automotive heat exchangers and therefore causes no concern.

Burgers and Lemczyk (1988) attempted to develop a 2D analytical model for a system with multi-louver fins. The model was applied to optimize the ratio of the louver length ($L_l$) to the fin height ($H_f$). The researchers concluded that the optimal louver length to fin height ratio is greater than 0.7. Their analysis also demonstrated the importance of a good tube to center braze bond (see Section 3.8.1).

Kurosaki, Kashiwagi, Kobayashi, Uzuhashi and Tang (1988) determined the $Nu$ for parallel louvered fins. They used laser holographic interferometry to study heat transfer in a variety of fin configurations. Their work verified that the location of the greatest local convective heat transfer coefficients is present on the leading edge of each louver.
Figure C-15 Convective Heat Transfer Coefficient Map
Tura (1986)
Aoki, Shinagawa and Suga (1989) tested a new device that may be used to measure the heat transfer coefficients of individual louvers in multi-louver fins. Their process utilized a sensing device made of nickel film that is evaporated onto the individual louvers. The authors claim that such a device will aid in optimizing louver arrays but no one has used it. The other methods of analysis used by Davenport (1980, 1984), Tura (1986), Cowell and Achaichia (1988), and Kurosaki et al. (1988) are preferred.

Huihua and Xuesheng (1989) performed a two dimensional experimental analysis on a multi-louver fin array. They concluded that the angle of the louvers within a multi-louver fin is optimal at 25°. This value agrees with the current range of values of α used in industry today.


Ikuta, Sasaki and Tanaka (1990) used CFD analysis to investigate the heat transfer and pressure drop within heat exchangers with multi-louver fins. They simplified their analysis by assuming a 2D problem. The results of the analysis confirmed that CFD software can predict the performance of multi-louver fins when they are parallel to each other in a core (i.e. tube and fin core).
Webb (1990) and Webb and Trauger (1991) conducted flow visualization experiments using a dye injection technique. The range of $Re_{Lp}$ from 400 to 4000 was studied. The researcher compared measured $\Gamma$ to $\Gamma_{ca}$ in Figure C-16 and determined that a revised correlation for flow efficiency ($\Gamma_w$) was necessary. Figure C-17 show the Webb and Trauger definition of $\Gamma_w$.

$$\Gamma_w = \frac{N}{D} = 0.95 \left( \frac{L_p}{F_p} \right)^{0.23} - 37.17 \times 10^{-6} \left[ Re_{Lp} - 828 \left( \frac{2\alpha}{\pi} \right)^{-0.34} \right]^{1.1} \left( \frac{L_p}{F_p} \right)^{-1.35} \left( \frac{2\alpha}{\pi} \right)^{-0.61} \tag{C-1}$$

$N$ is the actual transverse distance traveled by the air-flow and $D$ is the ideal distance.

Figure C-18 shows flow efficiency values which are a function of $Re_{Lp}$. A direct comparison of $\Gamma_w$ to $\Gamma_{ca}$ is shown in Figure C-19.

Webb and Trauger concluded that $\Gamma$ increases with decreasing $F_p$ to $L_p$ ratio at a constant $Re_{Lp}$. $\Gamma$ also increases with increasing $\alpha$ at a constant $F_p$ to $L_p$ ratio and $Re_{Lp}$. Finally, they concluded that vortex shedding may begin at $Re_{Lp} = 200$. 

C-24
Figure C-16 Measured Flow Efficiency (Γ) Compared to Γ_{re}
Webb (1990) pp. 8
Figure C-17 Flow Efficiency ($\Gamma_w$) as Defined by Webb
Webb (1990) pp. 6
Figure C-18 Webb and Trauger's Flow Efficiency ($\Gamma_\psi$) Measurements and Fit
Webb and Trauger (1991)
Figure C-19 $\Gamma_w$ Compared to $\Gamma_{ea}$
Webb (1990) pp. 16
Sunden and Svantesson (1991) tested four louvered fin geometrical configurations (shown in Figure C-20 through Figure C-23) in order to discover a surface which performs better that the multi-louver fin. They concluded that the multi-louver fin provided the most heat transfer enhancement when compared to the other fins. They also mentioned that the geometry shown in Figure C-22 performed nearly as well as the multi-louver for \( Re_{Lp} > 1100 \).

Sahnoun and Webb (1992) modeled the heat transfer and pressure drop through a multi-louver fin by dividing the air-side heat transfer area of a multi-louver fin into four distinct segments. The segments included the portion of the entrance and exit margins (see Figure 1-8), louver panels and “turn around rib” in the direction of the air flow bounded vertically by the \( L_L \) (see Figure 1-7 for \( L_L \)). The fourth section accounted for the flow which bypassed the louvers near the top and bottom of the convolutions.

The authors assumed fully developed flow over the louver panels and proceeded to create an analytical model for \( h \) and \( f \). \( \Gamma_w \) was used to calculate the portion of flow which passed through the louver panels. The characteristic length for the \( Re \) was redefined for each segment of the multi-louver fin. Characteristic length parameters considered were the width of the entrance and exit margins, louver pitch, width of the “turn around rib” and a modified hydraulic diameter.
Figure C-20 Sunden & Svantesson Modified Multi-Louver Fin

Figure C-21 Sunden & Svantesson Multi-Louver Fin
Sunden & Svantesson (1991) pp. 115
Figure C-22 Sunden & Svantesson L3 Louvered Fin
Sunden & Svantesson (1991) pp. 115
Figure C-23 Sunden & Svantesson L4 Louved Fin
Sunden & Svantesson (1991) pp. 116
The authors then compared the results of their model against the Davenport (1984) experimental data. The model under-predicted the 12.7 mm fin height data by 8% (mean value) and over-predicted the 7.8 mm fin height data by 8% (mean value). The authors concluded that the deviations from the Davenport data were due to error in the data. This was based upon core to core comparisons. The authors also concluded that their analytical model, which included no empirical constants, could predict the heat transfer and pressure drop performance of multi-louver fins with only moderate inconsistencies.

Cowell, Heikal and Achaichia (1993) compared the multi-louver fin to other fin designs by plotting the ratios of $D_h$ versus pumping power. They concluded that multi-louver fins should out perform offset strip fins for tube and fin type heat exchangers.

C.4 Summary of the Literature

When the multi-louver fin was first introduced in the 1950’s, researchers theorized that the air-flow through a core with this fin was similar to the “duct flow” exhibited in a core with plain fins. This duct flow paradigm led researchers to theorize that multi-louver fins enhance heat transfer by turbulating the flow within each duct.
Some researchers tried to correlate turbulence theories to the multi-louver fin. Edwards and Alker (1974) and Russel, Jones and Lee (1982) concluded that turbulence promoters enhance heat transfer. Fiebig, Kallweit and Mitra (1986, 1991) concluded that heat transfer performance could be enhanced by 20% to 60% by delta wing turbulence promoters.

Turk and Junkhan (1986) concluded that counter-rotating vortices enhance heat transfer. Takano, Tanasawa and Nishio (1989) concluded that the enhancement of heat transfer by turbulators does not vary when angles are chosen between 45° and 90°.

As researchers attempted to enhance the heat transfer of a heat exchanger by utilizing turbulating the flow, another group of researchers theorized that louvers that are not parallel to the fins may redirect the flow through the fins of a core.


Tura (1986) and Kurosaki et al. (1988) have shown that the greatest local $h$ is located near the leading edge of each louver in an array.
Burgers and Lemczyk (1988) concluded that louvers should span at least 70% of the fin height of a multi-louver fin. Huihua and Xuesheng (1989) concluded that the optimum louver angle is 25°.

Cowell and Achaichia (1988), Webb and Trauger (1991) and Bellows (1996) have correlated flow efficiency ($\Gamma$) as a function of $Re_{Lp}$, $F_p/L_p$ and $\alpha$.

Davenport (1980, 1984) and Cowell and Achaichia (1988) have created empirical correlations to predict the heat transfer and pressure drop in a heat exchanger core with multi-louver fins.

Sahnoun and Webb (1992) have created an analytical model to predict the heat transfer and pressure drop in a heat exchanger core with multi-louver fins.