An Investigation of the Performance of Confined, Saw-Tooth Shaped Wire-on-Tube Condensers

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AN INVESTIGATION OF THE PERFORMANCE OF CONFINED, SAW-TOOTH SHAPED WIRE-ON-TUBE CONDENSERS

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ABSTRACT

This thesis presents the results of an experimental investigation of the performance of several saw-tooth shaped wire-on-tube condensers with respect to a forced air flow. The condensers were confined in a variable height wind tunnel test section and subjected to free-stream velocities ranging from 0.2 to 2.0 m/s (0.66 to 6.6 ft/s). An overall energy balance was applied to the 'refrigerant' (water for this study) giving the total heat transfer rate from the condenser. The heat loss due to radiation was estimated and accounted for, as was the fin efficiency of the wires. As a result, the air-side convection heat transfer performance was determined and compared for each of the condensers. To get a more complete understanding of the performance, the air pressure drop through the condensers also was measured, which allowed the calculation of the fan power required to cool the coils. Influences of condenser orientation (ψ = 0 for air flow normal to the wires and ψ = π/2 for air flow normal to the tube passes), saw-tooth amplitude, wire spacing, and clearance (degree of confinement) on the performance were determined in this investigation.
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NOMENCLATURE

Roman Symbols (Dimensional Parameters)

A \text{ area, m}^2
C \text{ heat capacity rate, W/K}
c_p \text{ constant pressure specific heat, J/kg-K}
D \text{ diameter, m}
FA \text{ frontal area, m}^2
h \text{ convection heat transfer coefficient, W/m}^2\text{K}
k \text{ thermal conductivity, W/m-K}
m \text{ mass, kg}
m \text{ mass flow rate, kg/s}
N \text{ number}
p \text{ pressure, N/m}^2
q \text{ heat transfer rate, W}
R \text{ thermal resistance, K/W}
S \text{ centerline-to-centerline spacing, m}
t \text{ time, s}
T \text{ temperature, K}
UA \text{ conductance, W/K}
V \text{ velocity, m/s}

Roman Symbols (Dimensionless Parameters)

C_p \text{ pressure coefficient, } \Delta p/(\frac{1}{2} \rho V^2)
D_t^* \text{ dimensionless tube diameter, } D_t/D_w
f \text{ friction factor}
F \text{ radiation view factor}
Gr \text{ Grashof number, } g\beta(T_w - T_a)D^3/\nu^2
m \text{ fin parameter, } \left(\frac{hS_t^2}{kD_w}\right)^{1/2}
NTU \text{ number of transfer units}
Nu \text{ Nusselt number, } hD/k
Pr \text{ Prandtl number, } \nu\rho c_p/k
Ra \text{ Rayleigh number, } GrPr
Re \text{ Reynolds number, } \rho V D/\mu
S_t^* \text{ dimensionless centerline-to-centerline tube spacing, } S_t/D_w
S_w^* \text{ dimensionless centerline-to-centerline wire spacing, } S_w/D_w
NOMENCLATURE (CONT.)

Greek Symbols

\( \alpha \)  angle-of-attack measured from a horizontal datum, deg
\( \beta \)  volumetric coefficient of expansion, K\(^{-1}\)
\( \Delta \)  difference
\( \varepsilon \)  heat exchanger effectiveness; also total hemispherical emissivity (for radiation)
\( \eta \)  efficiency; fin efficiency if no subscript
\( \mu \)  viscosity, kg/s-m
\( \nu \)  kinematic viscosity, m\(^2\)/s
\( \rho \)  density, kg/m\(^3\)
\( \sigma \)  Stefan-Boltzmann constant, \( 5.67 \times 10^{-8} \) W/m\(^2\)-K\(^4\)
\( \tau \)  transmissivity
\( \psi \)  yaw angle, rad where: \( \psi = 0 \): air flow \( \perp \) to wires
\( \psi = \pi/2 \): air flow \( \perp \) to tubes

Subscripts

a  air
amb  ambient
char  characteristic
conv  convection heat transfer
eff  effective
i  inner surface
in  inlet
int  internal
lm  log-mean
L  layers
max  maximum
meas  measured quantity
min  minimum
out  outlet
r  refrigerant (water)
rad  radiation heat transfer
surf  surface
sur  surroundings
t  tube
tot  total
w  wire
1. INTRODUCTION

Wire-on-tube condensers are used to reject heat from the refrigerant in domestic refrigerators. The condenser is sometimes mounted vertically on the rear of the refrigerator cabinet and cooled by natural convection. Alternatively, the condenser is located below the refrigerator cabinet, orientated horizontally, and cooled by a fan-driven forced convection. This is the position found in most modern household refrigerators and the consideration of this thesis.

A wire-on-tube condenser is made up of steel tubing and wires. The steel tubing is bent into a planar serpentine so that the tubing forms parallel passes. Steel wires are spot-welded to each side of the tubing such that they are perpendicular to the tube passes. These wires act as extended surfaces, or fins, by adding extra surface area for the convection and radiation heat loss from the refrigerant to the surroundings. Fig. 1.1 is a top view of a standard wire-on-tube condenser. Note that there is another set of wires on the opposite side of the serpentine, and these two sets of wires are usually in tandem.

![Top view of a standard wire-on-tube condenser](image)

**Figure 1.1** Top view of a standard wire-on-tube condenser

Condensers are types of heat exchangers. Their effectiveness is directly related to the heat transfer rate from the refrigerant flowing through the coil. This rate is determined by the temperature of the refrigerant, \( T_r \), the ambient temperature, \( T_a \), and the thermal resistance between the refrigerant and the environment as well as the capacity rates of the air and refrigerant streams. The thermal resistance can be broken up into an internal resistance, which includes the convective resistance between the refrigerant and the steel tubing as well as the conductive resistance of the steel tubing, and the external, or 'airs-side', resistance.
Experiments performed by Admiraal and Bullard (1993) show that the external resistance for a refrigerator condenser is at least 95% of the total resistance for the two-phase region and greater than 62% for the superheated and subcooled regions. Because a large portion of the refrigerant flowing through the condenser is in the two-phase region, it is clear that the performance of a wire-on-tube condenser can be greatly improved by decreasing the air-side thermal resistance. To reduce this resistance, the surface area of the condenser can be increased or the air-side convection heat transfer coefficient can be improved. The overall size of the condenser is somewhat limited by economic and spatial constraints. Therefore, this study primarily concentrates on optimizing the air-side convection heat transfer performance of wire-on-tube condensers.

As was previously mentioned, wire-on-tube condensers that are cooled by forced convection are generally mounted in a horizontal position in the lower portion of the refrigerator. But in studies by Hoke (1995) and Swofford (1995), the air-side convection heat transfer coefficient increased as the angle-of-attack between the plane of the condenser and the air flow is increased. Based in part on these results, Lum (1997) began to experiment with a saw-tooth shaped multi-layer configuration in which the condenser is folded into successive layers. This allows a higher angle-of-attack without giving way to dimensional restrictions. In other words, it prevents the condenser from becoming too ‘tall’. This study continues with the concept of the saw-tooth shaped multi-layer condenser. The condensers under consideration are prefabricated into the saw-tooth shape; eight wire-on-tube condensers were manufactured in this manner by Indiana Tube Corporation. Fig. 1.2 shows a side view of a typical saw-tooth condenser. The angle-of-attack, $\alpha$, is defined in this figure.

There are three main parameters that are varied for these coils. The first is the wire spacing associated with the wire-on-tube condenser. The wire spacing, along with other important parameters, is defined in Fig. 1.3. The second parameter varied in this study is

![Figure 1.2 Side view of a typical saw-tooth multi-layer condenser](image-url)
the amplitude of the saw-tooth. The amplitude is defined in Fig. 1.2. The other main parameter that is varied in this investigation is the orientation of the condenser relative to the air flow. There are two orientations that are studied: air flow perpendicular to the wires and air flow perpendicular to the tubes. In addition, the effect of varying the ‘clearance’ will be investigated. Ideally, the bottom plane of the condenser is lifted the same small distance from the floor of the wind tunnel as the distance that the top plane of the condenser is from the ceiling of the tunnel. These spacings have been termed the clearance.

The results are presented such that only one parameter (i.e. wire spacing or amplitude) is varied at a time. This gives the reader a clear understanding of how each geometric parameter effects the performance of the condenser. After these analyses are complete, graphical results comparing the performance of all eight saw-tooth condensers will be presented. Also, relevant data will be non-dimensionalized and correlated. Finally, the results from this investigation are compared with those from a previous study that focused on saw-tooth shaped wire-on-tube condensers.

![Diagram of wire-on-tube condensers](image)

**Figure 1.3** Nomenclature definitions for several parameters associated with wire-on-tube condensers
2. **LITERATURE REVIEW**

There is only a handful of published literature pertaining to wire-on-tube condensers. For years, the studies only dealt with condensers that are cooled by natural convection. There are several MS theses and technical papers related to this subject and the key findings from this literature will now be presented.

The first publications involving wire-on-tube condensers appeared in a series of MS theses by Rudy (1956), Howard (1956), and Carley (1956). Each of the authors determined the effect of one or two geometric parameters on the heat transfer performance of wire-on-tube condensers cooled by natural convection and radiation.

Rudy worked to determine an optimum wire diameter and wire spacing to maximize the heat transfer for a horizontal wire-on-tube condenser. The data in his thesis clearly show that the air side conductance, which is the product of the air-side heat transfer coefficient and the total outside area, increases with increasing wire diameter for a constant wire spacing. For increasing wire spacing, an increase in conductance is also observed. In addition, Rudy ran a series of tests to observe the performance of a condenser in a vertical configuration. He found that the average heat rate for a coil in the vertical position was only 68% of the average heat rate for the same condenser in the horizontal configuration.

Howard's study dealt with determining the effect of tube spacing on the natural convection heat transfer from a horizontal wire-on-tube condenser. The data show that the heat transfer increases with a decreased tube spacing (or an increasing number of tube passes). Howard, like Rudy, did not determine an effective outer surface area and as a result the air-side convection heat transfer coefficients were left undetermined.

The goal of Carley's thesis was to determine the effect of varying tube diameters on natural convection from horizontal wire-on-tube condensers. The results from this publication indicate that increasing the diameter of the tube increases the heat transfer. Carley did make an effort to determine a heat transfer coefficient by assuming an effective outer surface area \( A_e + \eta_w A_w \). With this analysis, he found that the air-side convection heat transfer coefficient decreases with increasing tube diameter.

Using the data from the three previous sources, Witzell and Fontaine (1957a) wrote a technical article on parameters that influence the natural convection heat transfer for horizontally orientated wire-on-tube condensers. They determined a correlation that gave the Nusselt number as a function of the Grashof number. The characteristic length used in order to form these non-dimensional numbers was
and the correlation they developed was:

\[
\text{Nu} = 0.4724 \text{ Gr}^{0.2215} \quad (2.2)
\]

Using this correlation, Witzell and Fontaine (1957b) devised a design method for wire-on-tube condensers that are cooled by natural convection and radiation. Because the correlation was developed using empirical data, they warn that extrapolation from the following requirements may produce erroneous results:

1) The condenser must be horizontal

2) The outer dimensions of the condenser should be 610 x 914 mm (24 x 36 in.) with 914 mm (36 in.) wires

3) \(0.88 \text{ mm} \leq D_w \leq 2.32 \text{ mm} \) (\(0.0348 \text{ in.} \leq D_w \leq 0.0915 \text{ in.}\))
   \(4.23 \text{ mm} \leq S_w \leq 25.4 \text{ mm} \) (\(0.167 \text{ in.} \leq S_w \leq 1 \text{ in.}\))
   \(4.76 \text{ mm} \leq D_t \leq 15.9 \text{ mm} \) (\(0.188 \text{ in.} \leq D_t \leq 0.625 \text{ in.}\))
   \(25.4 \text{ mm} \leq S_t \leq 102 \text{ mm} \) (\(1 \text{ in.} \leq S_t \leq 4 \text{ in.}\))

For his MS thesis, Papanek (1958) studied the effect of angle-of-attack, \(\alpha\), on the convection heat transfer coefficient for wire-on-tube condensers subjected to cooling via natural convection and radiation. While keeping the wires in a horizontal position, the tube passes were rotated to various angles. Using the same effective area that was used by Carley (i.e. using a fin efficiency for the wires), Papanek found that the convection heat transfer coefficient decreases for increasing \(\alpha\). By using four different wire spacings, he also found that the angular dependence of the heat transfer coefficient increases as the wire spacing decreases.

For the condensers orientated in both horizontal and vertical positions, Papanek attempted to correlate the data in the form \(\text{Nu} = f(\text{Gr})\). The same characteristic length as was used by Witzell and Fontaine (see Eq. 2.1) is incorporated in these correlations:

\[
(\text{Nu})_{\alpha=0^\circ} = 0.2714 \text{ Gr}^{0.307} \quad (2.3)
\]

\[
(\text{Nu})_{\alpha=90^\circ} = 0.0188 \text{ Gr}^{0.7556} \quad (2.4)
\]
Here the subscripts 'α = 0°' and 'α = 90°' refer to wire-on-tube condensers placed in horizontal and vertical configurations, respectively.

In an effort to correlate Papanek's data more accurately, Witzell, Fontaine, and Papanek (1959) assumed a characteristic length of:

\[ D_{\text{char}}^{1/4} = \frac{A_t D_t^{1/4} + \eta_w A_w D_w^{1/4}}{A_t + A_w} \]  

(2.5)

With this new characteristic length, the data was re-correlated to give:

\[ (\text{Nu})_{\alpha=0^\circ} = 0.905 \, \text{Gr}^{0.176} \left( \frac{S_w - D_w}{S_w} \right)^{1.2} \]  

(2.6)

\[ (\text{Nu})_{\alpha=90^\circ} = 0.034 \, \text{Gr}^{0.726} \]  

(2.7)

Cyphers, Cess, and Somers (1959) also investigated the effect of α on the convection heat transfer coefficients associated with wire-on-tube condensers that are cooled by natural convection and radiation. They looked at the cases where the wires were kept horizontal while the tube passes were rotated (ψ = 0) and where the wires were rotated while the tube passes were kept horizontal (ψ = π/2). An average convection heat transfer coefficient, \( \bar{h} \), was defined as

\[ q = \bar{h}(A_t + A_w)(\bar{T}_i - T_a) \]  

(2.8)

where \( \bar{T}_i \) is the average outer surface temperature of the condenser tube passes. \( \bar{h} \) was found to decrease with increasing α for both the ψ = 0 and the ψ = π/2 cases.

For the data obtained from the ψ = π/2 case, the convection heat transfer coefficient associated with the wires, \( h_w \), was determined as a function of α. This was done using the relation

\[ \bar{h} = h_t \frac{A_t}{A_t + A_w} + h_w \frac{A_w}{A_t + A_w} - h_{rad} \]  

(2.9)

where \( h_{rad} \) is the average radiation heat transfer coefficient. Cyphers, Cess, and Somers found that their calculated \( h_w \) are approximately equal to the \( h_w \) predicted using the theoretical Nusselt-Grashof relation.
\[ \text{Nu} = \frac{2}{\ln \left[ 1 + 5 \left( \text{Gr} \cos \alpha \right)^{-1/4} \right]} \]  \quad (2.10)

defined for all \( \alpha \).

Cyphers, Cess and Somers also performed experiments investigating the effect of confining walls on \( \overline{h} \) for wire-on-tube condensers subjected to natural convection and radiation. The vertical walls were placed on either side of the condenser, and the spacing between the wall was varied during the study. The results generally show that as the spacing between the walls was decreased, the \( \overline{h} \) associated with the wire-on-tube condenser also decreased.

A study by Collicott, Fontaine, and Witzell (1963) focused on determining radiation heat loss so that they could easily deduce the natural convection heat loss. An evacuated chamber was used to eliminate all but radiation heat transfer from the coil. A number of condensers were tested, resulting in graphs showing an effective configuration factor versus the ratios \( D_i/S_i \) and \( D_w/S_w \). The effective configuration factor, defined as

\[ F_{\text{eff}} = \frac{q_{\text{rad}}}{\varepsilon \sigma A_{\text{tot}} (T_t - T_{\text{sur}})} \] \quad (2.11)

appears to decrease with increasing values of \( D_w/S_w \). The results involving the ratio \( D_i/S_i \) are inconclusive.

The trio recognized the fact that \( F_{\text{eff}} \) is not a true view factor; in other words, this factor is dependent on the temperature distribution within the condenser wires as well as the wire-on-tube geometry. Thus, \( F_{\text{eff}} \) should only be used in situations where there is very little natural convection heat transfer. This situation corresponds to high wire fin efficiencies, \( \eta_w \). If this efficiency is relatively low, \( F_{\text{eff}} \) will overestimate the actual radiation view factor considerably.

Nevertheless, Collicott, Fontaine, and Witzell used the experimentally obtained values of \( F_{\text{eff}} \) to estimate and remove \( q_{\text{rad}} \) from the overall heat transfer rate for several wire-on-tube condensers tested at various \( \alpha \). Using eqs. (2.1) and (2.8), the Nusselt number was determined for each of the experiments. This value was compared with the published correlation

\[ \text{Nu} = 0.11 \left( \text{Gr Pr} \right)^{1/3} + \left( \text{Gr Pr} \right)^{0.1} \] \quad (2.12)
for isothermal horizontal cylinders (with identical $D_{\text{cham}}$) cooled by natural convection. Plots in the literature show that the ratio of the experimental to theoretical Nusselt number increases (to a maximum of 1.0) with increasing values of $D_i/S_i$ and decreases with increasing $\alpha$.

The aforementioned studies are important in understanding the natural convection processes associated with wire-on-tube condensers. However, the condensers in most modern domestic refrigerators are cooled by a fan-driven forced convection air flow. The first known published studies that dealt with wire-on-tube condensers exposed to forced convection and radiation appeared in a pair of MS theses by Hoke (1995) and Swofford (1995). These theses were later reprinted in a technical report by Hoke, Swofford, and Clausing (1995). The investigations focused on the convection heat transfer performance for several wire-on-tube condensers as certain parameters were varied. These varying parameters included the air velocity, $V$, the angle-of-attack of the condenser, $\alpha$, and the wire and tube geometry (namely $D_w$, $S_w$, $D_t$, and $S_t$).

Both Hoke and Swofford performed their experiments in a wind tunnel that is described in Section 3.2 - Wind Tunnel. The dimensions of the test section are 305 mm (12 in.) high, and 914 mm (36 in.) wide. This relatively large test section allows for the condensers to be tested at high $\alpha$. However, because of the size of the test section, the condensers are not confined as they would be in a household refrigerator. As a result, the convection heat transfer performance as measured in their experiments may be different than if the condensers were confined in an actual refrigerator. Also, the wind tunnel was designed to produce a spatially uniform air flow across the condenser. This, too, may be unlike the conditions experienced in an actual refrigerator where the air flow is often highly non-uniform. Although the experimental conditions may not perfectly simulate those found in a domestic refrigerator, the results of Hoke and Swofford are extremely useful. They accurately find the relative convection heat transfer performance for wire-on-tube condensers as several parameters are varied.

For these studies, a definition for the convection heat transfer coefficient associated with the wires was developed

$$h_w = \frac{q_{\text{conv}}}{\left( \frac{A_t}{\sqrt{D_t}} + \eta_w A_w \right) \left( T_t - T_a \right)}$$

(2.13)
where $D_t^*$ is the dimensionless tube diameter ($D/D_w$). This definition of the convection coefficient was used to measure the heat transfer performance of each condenser. It is similar to the convection coefficient that is used in this report, which is introduced in Section 4.3 - The Definition of $h_w$.

Hoke, Swofford, and Clausing found that $h_w$ is strongly influenced by a number of the parameters. The condensers were tested with two orientations relative to the air flow: air flow perpendicular to the wires ($\psi = 0$) and perpendicular to the tubes ($\psi = \pi/2$). As would be expected, $h_w$ increased as the air velocity, $V$, increased for both orientations. $h_w$ also was found to increase as the angle-of-attack, $\alpha$, increased. The exception to these findings involved horizontal condensers with $V < 0.5$ m/s (1.6 ft/s). For these low free-stream air velocities, the buoyant forces interact with the inertial forces in such a way as to produce a lower $h_w$ than would be observed for condensers that only experience natural convection and radiation.

By investigating seven different wire-on-tube condensers, the dependence of $h_w$ on the wire and tube geometry was established. Each condenser was tested at both the $\psi = 0$ and the $\psi = \pi/2$ orientations, and each test was run with the same predetermined set of free-stream air velocities. The data were then non-dimensionalized resulting in experimental values of $Nu_w$ and $Re_w$ with the wire diameter being the characteristic length for both quantities. A correlation was originally developed; however a dependence on the wire spacing remained. A final correlation included a non-dimensional function of the wire spacing to account for this dependence. For $Re_w > 50$ and $2.8 \leq S_w^* \leq 4.4$, the Nusselt-Reynolds relation was found to be

$$Nu_w = C Re_w^n \left[ 0.985 - 98.5 \exp(-2.32 S_w^*) \right] \quad (2.14)$$

where $S_w^*$ is the dimensionless wire spacing ($S_w/D_w$) and both $C$ and $n$ are constants based on the curve fits of the data. $C$ and $n$ are different for the two condenser orientations. For air flow normal to the wires ($\psi = 0$),

$$C = 0.274 - 0.247 \cos (|\alpha| - 4.87) \exp[-0.00234 (\alpha + 0.902)^2] \quad (2.15a)$$

$$n = 0.585 - 0.249 \cos (|\alpha| + 20.0) \exp[-0.00441 (\alpha + 1.66)^2] \quad (2.15b)$$

and for the case with air flow normal to the tubes ($\psi = \pi/2$):

9
\[
C = 0.263 - 0.235 \cos(\alpha) \exp(-0.00289 \alpha^2) \quad (2.16a)
\]
\[
n = 0.55 - 0.269 \cos(\alpha) \exp(-0.00597 \alpha^2) \quad (2.16b)
\]

Although it is not directly clear from the above equations, Hoke, Swofford, and Clausing determined that the heat transfer performance associated with the \( \psi = 0 \) condensers is slightly higher than that of the \( \psi = \pi/2 \) coils for most \( \alpha \).

In addition to the aforementioned experiments, Hoke also performed two other detailed studies. The first dealt with observing the heat transfer characteristics of a condenser without wires (i.e. an unpainted serpentine tube) at various \( \alpha \). He found that when this condenser was orientated at \( \psi = \pi/2 \) and \( \alpha > 20^\circ \), its convection coefficient closely resembles that of a cylinder in cross-flow using correlations created by Hilbert (1933) and Zhukauskas (1972). The second additional investigation by Hoke focused on the effect of the wire length, \( L_w \), on the heat transfer performance of \( \psi = \pi/2 \) wire-on-tube condensers exposed to forced convection and radiation. He found that for condensers with an angle-of-attack between \( \alpha = -5^\circ \) and \( \alpha = 0^\circ \), \( h_w \) decreases with increasing \( L_w \).

Swofford performed an additional experiment investigating the effect of \( \alpha \) on \( h_w \) associated with wire-on-tube condensers that are subjected to either natural or forced convection coupled with radiation. For these tests, a relatively small wire-on-tube condenser was used with outer dimensions of 283 x 279 mm (11.1 x 11 in.). The smaller size allowed the condenser to be tilted to large angles-of-attack in the wind tunnel test section. The results show that for a condenser exposed to natural convection and radiation, \( h_w \) decreases with increasing \( \alpha \) for both the \( \psi = 0 \) and the \( \psi = \pi/2 \) cases. For a condenser exposed to forced convection and radiation, \( h_w \) increases with increasing \( \alpha \) for both orientations. When the condenser is situated at \( \alpha = 90^\circ \), \( h_w \) is 250% higher than if the coil is positioned horizontally. The heat transfer performance was again found to be slightly higher for a condenser orientated at \( \psi = 0 \) for most \( \alpha \).

A technical article by Hoke, Clausing, and Swofford (1997) summarized the major findings of Hoke (1995) and Swofford (1995). This article focuses on the effects of \( V \), \( \alpha \), \( \psi \), and the wire and tube geometry on the relative heat transfer performance of condensers exposed to forced convection and radiation. The logic behind the definition of the convection heat transfer coefficient, \( h_w \), is also discussed in detail in the publication. The authors explain that \( h_w \) is a definition that takes into account the thermal interaction of the tubes and the wires and is extremely useful in comparing the relative performance of wire-on-tube condensers.
In his MS thesis, Lum (1997) continued with the study of wire-on-tube condensers exposed to forced convection and radiation. He recognized that the heat transfer performance of these condensers increases as the angle-of-attack increases. But since there are dimensional restrictions for condensers in actual domestic refrigerators, the coils that Lum studied were confined, saw-tooth shaped, multi-layer wire-on-tube condensers. This allows for relatively high angles-of-attack without letting the coil get too ‘tall’. Lum studied the effect of varying many of the same parameters that were investigated by Swofford and Hoke. He also studied how the individual layers performed within the multi-layer configuration. The thesis was later reprinted in a technical report by Lum and Clausing (1997).

In his investigation, Lum incorporated the same wind tunnel and basic experimental setup that was used by Hoke and Swofford. However, to simulate the tightly confined forced convection conditions that are experienced in most household refrigerators, Lum used a variable height test section in the wind tunnel. This test section is discussed in detail in Section 3.3 - Variable Height Wind Tunnel Test Section. Lum also constructed several support frames to maintain the multi-layer condensers. The support frames were necessary to help stabilize several smaller, flat wire-on-tube condensers and to keep a consistent angle-of-attack. Another major change in Lum’s study was the measurement of the air pressure drop, Δp, through the coil. This was done by placing pressure taps both upstream and downstream of the condenser. The pressure drop is an important parameter because it is directly related to the fan power that is required to do the cooling.

Lum used a slightly modified definition of $h_w$ as compared to that of Eq. (2.13). He defined the convection coefficient as

\[
h_w = \frac{q_{\text{conv}}}{\left( \frac{A_t}{D_t} + \frac{\eta_c \eta_w A_w}{\Delta T_{\text{in}}} \right) \Delta T_{\text{in}}}
\]

where:

\[
\Delta T_{\text{in}} = \ln \left( \frac{T_{\text{in}} - T_a}{T_{\text{in}} - T_a} \right)
\]

(2.18)
There are some new variables in these expressions. In Eq. (2.17), \( \eta_c \) is the effective thermal constriction efficiency, which accounts for the additional temperature drop due to the constricted heat flow path between the inside of the tube and the spot formed when the wire is welded to the tube. The definition of the log-mean temperature, Eq. (2.18), accounts for the temperature difference in each condenser layer. The subscript ‘k’ is an index referring to a specific layer number. \( T_{\text{t,in}} \) and \( T_{\text{t,out}} \) are surface temperatures at the inlet and outlet of the serpentine tube of the condenser layer, respectively.

Lum investigated the effect of the number of condenser layers, \( N_L \), and the layer spacing, \( S_L \), on \( h_w \). The results show that the \( h_w \) associated with downstream condenser layers is nearly independent of \( S_L \) when \( S_L \geq 31.2 \text{ mm} \) (1.23 in.). Lum noted that because of this, along with the finding that the \( h_w \) per layer is independent of \( N_L \), multi-layer condensers can be designed and evaluated based primarily on the performance of single layer condensers with identical wire and tube geometries.

Hoke, Clausing, and Swofford (1997) concluded that the heat transfer performance improves as the angle-of-attack increases when a wire-on-tube condenser is exposed to forced convection and radiation. In his study, Lum confirmed this general trend. The results show that for condensers that are orientated at \( \psi = \pi/2 \), there is an increase in \( h_w \) as \( \alpha \) increases. Lum concluded that condensers with this orientation should be designed such that they have large angles-of-attack.

In an effort to observe the effect of the number of condenser layers on \( \Delta p \), Lum tested condensers with varying \( N_L \) and measured the air pressure drop through each multi-layer condenser. The data show that the \( \Delta p \) is linearly dependent on the number of condenser layers. Lum points out that this simplifies the design process for multi-layer condensers, since the \( \Delta p \) for a particular multi-layer condenser can easily be predicted from that of an identically oriented single layer condenser with the same wire and tube geometry.

In his results, Lum finds that the \( h_w \) associated with the wire-on-tube condensers orientated at \( \psi = 0 \) is slightly higher than that of the \( \psi = \pi/2 \) coils. This is consistent with the findings of Hoke, Clausing, and Swofford (1997). However, Lum advises against the design and use of condensers with the air flow perpendicular to the wires (\( \psi = 0 \)). This is because the \( \Delta p \) per layer associated with multi-layer condensers in this orientation are significantly higher than those of condensers at \( \psi = \pi/2 \). The data show that the percent differences between the \( \Delta p \) are much greater than the percent difference in \( h_w \) obtained from the two orientations.

Lum went on to correlate his results in the form of a Nusselt-Reynolds relation. The characteristic length used for both dimensionless numbers is the wire diameter.
the Reynolds number, Lum uses a maximum velocity, \( V_{\text{max}} \) (based on the average air flow through the minimum flow area), rather than the free-stream air velocity. The minimum flow area was determined by subtracting the frontal projected area of a single set of wires (the wires on one side of a condenser layer) and the tube passes of a particular condenser layer from the cross-sectional area of the test section. Eq. (2.19) defines \( V_{\text{max}} \) as a function of the wire-and-tube geometry, the wind tunnel duct dimensions, and the free-stream air velocity:

\[
V_{\text{max}} = \left( \frac{H_{\text{duct}}}{H_{\text{duct}} - N_t D_t} \right) \left( \frac{W_{\text{duct}}}{W_{\text{duct}} - (1/2) N_w D_w} \right) V
\]  

(2.19)

Lum’s definition of the maximum velocity is different from the one used to reduce the data in this investigation, which is introduced in Section 5.6 - Other Results. However, the resulting maximum velocities when implementing the two definitions for the same condenser are quite close.

Lum devised separate correlations for each condenser orientation relative to the air flow. They both take the form

\[
\text{Nu}_w = C \text{Re}_{w,max}^{0.5744}
\]  

(2.20)

where:

\[
C = \begin{cases} 
0.2591 & \text{for } \psi = 0, 45^\circ \leq \alpha \leq 90^\circ \\
0.502 \sin(\alpha) \exp(-1.014 \alpha + 0.3775 \alpha^2) & \text{for } \psi = \pi/2, 45^\circ \leq \alpha \leq 90^\circ 
\end{cases}
\]  

(2.21)

(2.22)

In addition, Lum non-dimensionalized the pressure drop data by introducing a drag coefficient, \( C_D \). This quantity was correlated with the Reynolds number based on the maximum velocity. Again, the results were strongly dependent on the angle-of-attack.

\[
C_D = D_1 + D_2 \text{Re}_{w,max}^{-0.06333}
\]  

(2.23)

where:

\[
C_D = \frac{\Delta p}{(1/2) \rho V^2}
\]  

(2.24)

\[
D_1 = -0.7856 \sin(\alpha) \exp(1.177 \alpha - 0.3229 \alpha^2)
\]  

(2.25)
\[ D_2 = 2.451 \sin(\alpha) \exp(0.2858 \alpha) \] (2.26)

This correlation is only relevant for condensers orientated at \( \psi = \pi/2 \) with \( 45^\circ \leq \alpha \leq 90^\circ \). Lum discusses that the drag coefficient is highly dependent on the locations of the wires on one side of the condenser with respect to those on the other side of the condenser for \( \psi = 0 \). There was not enough data to validate a \( C_D \) correlation for this orientation given the wide range of \( \alpha \).
3. EXPERIMENTAL APPARATUS AND PROCEDURE

The equipment used in conducting these experiments was previously used in the research conducted by Hoke (1995), Swofford (1995), Rasmussen (1997), and Lum (1997). The apparatus primarily consists of an induced flow wind tunnel with a variable height test section and a constant temperature water supply system. For the current study, numerous wood forms were created to match the required height of the condensers being tested. The following section discusses the experimental apparatus, instrumentation, and procedure.

3.1 Constant Temperature Water Supply System

The primary purpose of the constant temperature water supply system is to deliver relatively hot water (used as the refrigerant for these experiments) to the inlet of the condensers at a consistent pre-set temperature. This system was designed and developed by Swofford (1995). It has two main portions, a preheating section and an accurate temperature regulation section. A schematic of the entire system is shown in Fig. 3.1.

Water is used as the refrigerant for the purposes of this study. There are several reasons for using water as opposed to standard refrigerant. First, water is abundant, inexpensive, and

![Figure 3.1 Schematic of the constant temperature water system. From Lum (1997).](image)
safe. Leaks may occur in the circulation system, however water is much easier to clean up than refrigerant and there are not any health related issues. Also, the thermal and transport properties of water have been clearly established and the inside resistance due to convection with water can be evaluated accurately. Although the internal resistance of the water flowing through the condensers is not the same as that of a two-phase refrigerant, the focus of this investigation deals with air-side convection. Because the air-side thermal resistance is separated in the determination of \( h_w \), the use of water allows for a suitable comparison of relative air-side convection coefficients.

As seen in Fig. 3.1, the water comes from the city water supply and passes through a filtration system to remove any impurities. After passing through a pressure regulator, the water enters a domestic water heater where it is preheated to a temperature of 322.1 ± 5.6 K (120 ± 10 °F). However, the temperature of the water exiting the domestic water heater is not constant, so the water temperature has to be regulated more accurately. This is accomplished by passing the water through an isothermal bath and ensuring that the water exits the bath at a specified temperature.

The isothermal bath contains approximately 42 gallons of propylene glycol. Propylene glycol has superb thermal properties. Thus, when the temperature controller is set to a specified temperature, the 4 kW (13600 Btu/hr) heater that is immersed in the bath heats the propylene glycol to an accurate, spatially uniform temperature. A 610 mm (24 in.) stainless steel sheathed thermocouple probe is inserted into the bath to monitor the temperature. The temperature controller uses an ON/OFF controlling method. When the temperature of the bath exceeds the desired temperature by 0.06 °C (0.1 °F), the 4 kW heater is deactivated. Once the temperature of the bath drops 0.06°C (0.1 °F) below the specified temperature, the heater turns back on and the cycle continues throughout the experiment. There are two 15 W (1/50 hp) immersion pumps located in the bottom of the bath which serve to circulate the propylene glycol, minimizing temperature stratification.

As the water leaves the domestic water heater and enters the isothermal bath, it travels through a pair of plate-fin evaporator coils immersed in the bath. This causes the water to experience heat transfer with the propylene glycol. The heat exchanger effectiveness of the evaporators is 0.99; hence, the water leaves the bath at essentially the same temperature as the propylene glycol. Once the water leaves the isothermal bath, it travels down a length of insulated tubing to the wind tunnel test section, where it enters the condenser being studied.

### 3.2 Wind Tunnel

The wind tunnel used in conducting the experiments was developed by Hoke (1995). It simulates the forced convection air flows that are experienced by wire-on-tube condensers in many
domestic refrigerators. The wind tunnel is capable of producing uniform air flows in the range of 0.15 to 2 m/s (0.5 to 6.6 ft/s). There are distinct portions of the wind tunnel as seen in Fig. 3.2. These include the flow conditioning section, the test section where the condensers are mounted, and the flow exhaust section. The air flow is induced by a backward inclined centrifugal fan which is powered by a 560 W (3/4 hp) variable speed DC motor.

The air that is drawn into the wind tunnel by the fan first travels through a 150 mm (6 in.) honeycomb flow straightener, followed by a series of five nylon screens. This minimizes any turbulence or vortices that may occur before the air flow enters the test section. The test section is constructed out of 13 mm (0.5 in.) thick Plexiglas which is supported by an aluminum frame. The interior of the test section has the following dimensions: 305 mm (12 in.) in height, 914 mm (36 in.) in width, and 762 mm (30 in.) in depth. Within this part is the variable height wind tunnel test section, which is the topic of Section 3.3 - Variable Height Wind Tunnel Test Section. After leaving the test section, the air flows through a 2.44 m (8 ft) converging, square to round, galvanized sheet metal duct. The air exits at the 254 mm (10 in.) circular fan inlet, which is connected to the fan by a 200 mm (8 in.) long flexible duct section before being exhausted into the room.

The air flow has been measured to have very little turbulence and instability due to the converging section as well as the flow straighteners and screens. The flow uniformity has been measured to be 2.5% across the test section and the flow remains steady to within 2.5%. In
addition, the turbulence at \( V = 2 \text{ m/s} \) (6.6 ft/s) is below 1%. These measurements were made using a TSI IFA 100 hot wire anemometer.

3.3 Variable Height Wind Tunnel Test Section

A variable height test section that fits within the original wind tunnel test section has been developed by Rasmussen (1997). As the name suggests, the variable height test section allows the height of the test section to be adjusted with relative ease. This is important because it helps better simulate the tightly confined, forced convection situation found in domestic refrigerators. Also, with the difference in sizes of the saw-tooth condensers being investigated in this study, the variable height test section allows the wind tunnel to be adjusted so that the clearance is equal for each condenser. As will be seen, it is important to maintain a consistent clearance for all the condensers so that performance can be fairly compared.

The top and bottom panels of the test section are 6.35 mm (0.25 in.) thick sheets of acrylic attached in a plane to a thin, flexible sheet of polycarbonate. Also attached to each of the panels are four slotted acrylic tabs that can be bolted to the side panels, forming a rectangular duct. The slotted tabs allow the duct height to be varied based on where they are bolted to the side panels. Once the slotted tabs have been bolted to the side panels, the flexible sheets of polycarbonate are carefully bent and bolted down so that they are flush against the top and bottom of the side panels. Fig. 3.3 is an illustration of the assembled variable height test section.

![Variable height wind tunnel test section](image)

**Figure 3.3** Variable height wind tunnel test section set at (a) maximum height and (b) minimum height. From Lum (1997).

The test section is positioned in the wind tunnel so that the portion on the right side of Fig. 3.3 is facing the flow conditioning section. The width of the variable height test section is less than the width of the original test section so that it can be easily positioned in the wind tunnel. Because of this, the portions of the original wind tunnel that extend beyond the width of the variable height test section are blocked off using foam barriers so that the flow is forced entirely through the new test section. After the air flow goes through the conditioning section, it is
contracted by the curved, smooth polycarbonate sheets. The top and bottom acrylic panels, as well as the side panels, define the new test section's cross sectional area.

As was previously mentioned, the height of the wind tunnel is varied by the location that the slotted tabs are bolted to the side panels. As seen in figs. 3(a) and 3(b), the height of the test section can be adjusted from 50.8 mm (2 in.) to 152 mm (6 in.). The width of the test section is fixed at 762 mm (30 in.). The depth (acrylic portions only) measures 622 mm (24.5 in.). These dimensions are sufficient for the testing of all the saw-tooth condensers relevant to this study.

Because the width of the test section is considerably greater than that of the condensers being investigated, wood forms are used to reduce the width of the wind tunnel. These wood forms have to be carefully constructed so that they match the contour of the test section. The wood forms are made of half-inch thick plywood which remains rigid and stable when positioned in the test section. Since the multi-layer condensers vary in amplitude, several sets of wood forms have been constructed so that the clearance for each coil is consistent. Attached to the top and bottom of the wood forms are strips of foam, which serve two purposes. First, the foam prevents air from escaping in case there are any voids between the wood and the test section. Secondly, by using varying densities of foam, the wood forms can be manipulated so that they will fit in a slightly taller test section. In other words, the test section height can be altered by as much as a half-inch by using different foam configurations. An illustration of a typical wood form is presented in Fig. 3.4.

![Figure 3.4 Side view of a typical wood form](image-url)
3.4 Data Acquisition System

Once the condenser is situated in the wind tunnel and the temperature of the hot water bath has reached steady-state, it is time to take some measurements that are required to perform the data reduction. These measurements include: the free stream air velocity (V), the air temperature upstream of the condenser (T_{a,in}), the water temperatures at the inlet and outlet of the condenser (T_{r,in} and T_{r,ou}), the mass flow rate of the refrigerant (m_r), and the pressure drop through the multi-layer condenser (∆p). This section discusses how each of these quantities are measured or calculated, if necessary.

The velocity of the air, V, is measured directly upstream of the wire-on-tube condenser. A probe from a TSI 8355 Air Velocity Meter is brought up through a hole from underneath the wind tunnel. Care is taken to make sure that the sensor on the probe is positioned precisely in the center of the cross-section of the test section. This anemometer has been calibrated prior to use, and has an absolute uncertainty of ± 0.03 m/s (0.1 ft/s). The measured velocity must be adjusted to account for the ambient conditions. This is done using the equation

$$V = V_{\text{meas}} \frac{T_{a,in}}{294.25} \left( \frac{760 \text{ mm Hg}}{P_{\text{amb}}} \right)$$

where \(V_{\text{meas}}\) is the measured velocity and \(P_{\text{amb}}\) is the ambient pressure.

The inlet air temperature, \(T_{a,in}\), is measured with copper-constantan (type T) thermocouple junctions that are located in the flow conditioning section of the wind tunnel. By sealing each of the thermocouple reference junctions in a Kay Instruments ICE POINT reference, each thermocouple is ensured to be referenced at 0.0 °C (32 °F). The differences in the temperatures between the junctions of the thermocouples generates an emf, which is read directly from a Fluke digital voltmeter with a resolution of 1 μV. The thermocouples have all been calibrated in an isothermal bath and a thermometer with a resolution of 0.05 K (0.09 °F). There are five thermocouples that are used to determine the air temperature, and the precision limit of each ranges from 0.04 K (0.06 °F) to 0.06 K (0.11 °F). In an effort to improve uncertainty, each thermocouple is read three times during a test run and the results are averaged, with an absolute uncertainty of ± 0.07 K (0.12 °F) for each averaged measurement. Four of these averaged temperatures are then used to obtain \(T_{a,in}\), and the absolute uncertainty of the resulting quantity is estimated to be ± 0.06 K (0.10 °F).

The temperatures of the water at the inlet and outlet of the condenser are measured using mixing cups at the two ends of the coil. A schematic of a mixing cup is shown in Fig. 3.5. It consists of copper fittings that are connected to form a sudden expansion followed by a sudden
contraction. Two type T thermocouple probes with a diameter of 1.02 mm (0.040 in.) are inserted approximately 25 mm (1 in.) into the flow in order to measure the temperature of the water as it exits the mixing cup. Thermopiles have four thermocouple junctions, as opposed to two junctions for standard thermocouples, so the measured temperatures have a much greater resolution. Again, a Kay Instruments ICE POINT reference is used to ensure that the thermopiles are referenced to 0.0 °C (32 °F). The emf differences are read on the same Fluke digital voltmeter with a resolution of 1 μV. These thermopiles have also been calibrated in the constant temperature bath and thermometer that were used to calibrate the previously mentioned thermocouples. It is estimated that the absolute uncertainty associated with these measurements is ± 0.07 K (0.12 °F).

**Figure 3.5** Schematic of a mixing cup for water temperature measurement

Another quantity that needs to be accurately determined is the flow rate of the water, \( \dot{m}_w \). The flow rate can be adjusted by changing the water supply pressure, which is controlled using a pressure regulator. Once the test begins, the water flowing from the outlet of the condenser is gathered in a bucket, which sits on a Scientech SG 5000 Electronic Balance. The balance was calibrated using a 2000 ± 0.005 g Electronic Balance Calibration Mass and has an absolute uncertainty of ± 0.10 g (0.00022 lbm). Concurrently, a stopwatch with an absolute uncertainty between ± 0.0063 s and ± 0.0088 s is started and runs throughout the test. The water is collected until either 180 seconds passes, or until 3000 g (6.6 lbm) of water accumulates, whichever takes longer, at which point the stopwatch is halted and the mass flow rate is calculated. Assuming operator biases of ± 0.03 s and ± 8.00 g (0.018 lbm) for the stopwatch and the balance respectively, the absolute uncertainty of the mass flow rate is ± 0.05 g/s (0.397 lbm/hr). However, the precision limit based on the unsteadiness of the water flow rate has been measured to be ± 0.27 g/s (2.14 lbm/hr), so this will be the assumed absolute uncertainty of the water's flow rate.

The pressure drop through the condenser is also measured during the experimentation.
An Omega PX 653 pressure transducer is used to measure $\Delta p$. There are a pair of pressure taps located both upstream and downstream of the condenser that are connected to the transducer via Tygon™ tubing. The output from the transducer is displayed on an Omega digital voltmeter with a resolution of 1 mV. Calibration data for the transducer was supplied by the manufacturer. The absolute uncertainty of the pressure drop measured by the transducer is estimated to be $\pm 0.065$ Pa ($0.0003$ in. H$_2$O) by the manufacturer.

### 3.5 Experimental Procedure

Before the experiment can start, the condenser must be situated in the wind tunnel. Based on the size of the condenser and the desired clearance, the variable height test section is adjusted and appropriate wood forms are selected. The coil is placed on several small, low-conductivity blocks that lift it a slight distance off the floor of the test section (this distance is the clearance). Once the test section is bolted down and the constant temperature water supply system has reached steady-state, the tests can begin.

The wind tunnel is adjusted so that the air flow is very close to 0.20 m/s (0.66 ft/s), and the system is allowed to come to steady-state for about one minute. The output on the digital voltmeter for each thermocouple is read three times and averaged. Also, the air pressure drop through the condenser is read on a separate digital voltmeter. Finally, the mass flow rate of the water is recorded at which point the velocity of the wind tunnel is increased and the process starts all over. The other air velocities tested are 0.25, 0.35, 0.50, 0.75, 1.0, 1.25, 1.5, 1.75, and 2.0 m/s (0.82, 1.2, 1.6, 2.5, 3.3, 4.1, 4.9, 5.7 and 6.6 ft/s).
DATA REDUCTION

The following sections discuss the data reduction assumptions and methodology. Multi-layer wire-on-tube condensers have a somewhat complicated geometry, requiring careful data reduction in several steps. In addition, the fact that the wires are not isothermal surfaces adds complexity to the process. But by combining a heat exchanger methodology with an extended surface analysis, the data reduction can be done in an accurate and meaningful fashion.

4.1 Preliminary Analysis

There are several quantities that are acquired directly from the experimental work and used as inputs for the data reduction. These include the air velocity, \( V \), the inlet air temperature, \( T_{a,in} \), the refrigerant inlet and outlet temperatures, \( T_{r,in} \) and \( T_{r,out} \), and the mass flow rate of the refrigerant, \( \dot{m}_r \). It should be mentioned that if these experiments involved actual refrigerant, additional measurements would be necessary. Water is being used as the refrigerant in this investigation for the reasons discussed in Section 3.1 - Constant Temperature Water Supply System.

The total heat transfer rate from the entire wire-on-tube condenser can be calculated by performing an energy balance on the fluid flowing through the serpentine tube of the condenser. This is accomplished using the equation

\[
q_{tot} = C_r (T_{r,in} - T_{r,out})
\]

where \( C_r \) is the heat capacity rate of the refrigerant. This is the product of the refrigerant mass flow rate and the specific heat of the refrigerant, \( c_{p,r} \).

To get a heat transfer rate that can be used for calculation of the air-side convection coefficient, \( h_w \), the radiative portion of the total heat transfer rate must be subtracted:

\[
q = q_{tot} - q_{rad}
\]

The estimation of the radiation heat loss will be discussed in Section 4.4 - Radiation. From this heat rate, \( q \), the outlet air temperature can be determined by using another energy balance

\[
T_{a,out} = T_{a,in} + \frac{q}{C_a}
\]
where \( C_a \) is the heat capacity rate of air and is the product of the air's mass flow rate and specific heat, \( c_{p,a} \). Implicit in Eq. (4.3) is the assumption that the heat loss due to radiation goes to the air.

### 4.2 Heat Exchanger Analysis

The wire-on-tube condensers that were studied are heat exchangers; hence, the continuously changing temperature difference between the two streams must be accounted for in the data reduction process. There are two orientations for these condensers as defined by the yaw angle, \( \psi \). \( \psi = 0 \) refers to air flow perpendicular to the wires, whereas \( \psi = \pi/2 \) radians corresponds to air flow normal to the tubes. Each of these orientations is being modeled as a distinct type of heat exchanger. The \( \psi = 0 \) orientation is modeled as a shell-and-tube heat exchanger with an even number of tube passes. This is a good model because the refrigerant flows in a number of tube passes and alternates between flowing with the air flow and against the air flow, as seen in Fig. 4.1. Because the number of tube passes is large, the error induced if there are an odd number of tube passes is small.

On the other hand, the \( \psi = \pi/2 \) orientation is modeled as being either a counterflow or a parallel flow heat exchanger, depending on where the refrigerant enters the condenser. If the coil is connected in such a way that the refrigerant enters the condenser downstream in the air flow, it is considered a counterflow heat exchanger. This is because the refrigerant is continuously flowing against the direction of the air flow as it is cooled. A schematic of this setup is shown in Fig. 4.2. Alternatively, if the refrigerant enters the condenser in the upstream location, it is viewed as a parallel flow heat exchanger because the refrigerant proceeds in the same direction as the air flow. A counterflow heat exchanger arrangement was used for virtually all cases with air flow perpendicular to the tube passes (\( \psi = \pi/2 \)). For a pure condenser, the type of heat exchanger (i.e. counterflow, parallel flow, shell-and-tube, etc.) is immaterial and the results will be the same.

With the heat rate associated with air-side convection determined from Eq. (4.2), the next step is to calculate the effectiveness of the heat exchanger

\[
\varepsilon = \frac{q}{q_{\text{max}}} = \frac{q}{C_{\text{min}} (T_{\text{r,in}} - T_{\text{a,in}})}
\]

(4.4)

where \( C_{\text{min}} \) is the minimum of the two heat capacity rates, \( C_r \) and \( C_a \). The effectiveness is the ratio of the actual heat transfer rate for a heat exchanger to the maximum possible heat transfer rate. This maximum rate is the product of the minimum heat capacity rate (either the air or the refrigerant) and the maximum possible temperature change. The maximum temperature change is what would be seen in a counterflow heat exchanger of infinite length.
Figure 4.1 Condenser modeled as a shell-and-tube heat exchanger

Figure 4.2 Condenser modeled as counterflow heat exchanger
The number of transfer units (NTU) is a measure of the size of the heat exchanger where:

\[
NTU = \frac{UA}{C_{\min}} \tag{4.5}
\]

This quantity, like the effectiveness, is a dimensionless number used in the analysis of heat exchangers. The NTU is a function of the heat exchanger effectiveness, the ratio of the heat capacity rates (the minimum over the maximum), and the geometry of the heat exchanger. There are different formulas for calculating the NTU for each heat exchanger geometry: counterflow, parallel flow, shell-and-tube, etc.

For a counterflow heat exchanger

\[
NTU = -\frac{1}{C_r - 1} \ln \left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right) \tag{4.6}
\]

where:

\[
C_r = \frac{C_{\min}}{C_{\max}} \tag{4.7}
\]

and for a shell-and-tube heat exchanger with an even number of tube passes

\[
NTU = -\left(1 + C_r^2\right)^{-1/2} \ln \left(\frac{E - 1}{E + 1}\right) \tag{4.8}
\]

where:

\[
E = \frac{2\varepsilon - (1 + C_r)}{(1 + C_r)^{1/2}} \tag{4.9}
\]

With the NTU known for a specific heat exchanger, it is possible to calculate the overall conductance, UA, from Eq. (4.5). In order to get the surface resistance of the heat exchanger, the internal resistance from the refrigerant flowing through the tube passes must be subtracted from the total resistance:

\[
R_{\text{surf}} = \frac{1}{UA} - R_{\text{int}} \tag{4.10}
\]
To determine this internal resistance, the refrigerant-side heat transfer coefficient is calculated using the Gnielinski correlation

\[
\text{Nu}_D = \frac{(f/8)(\text{Re}_D - 1000)\text{Pr}}{1 + 12.7(f/8)^{1/2}(\text{Pr}^{2/3} - 1)}
\]  

(4.11)

where:

\[
f = \left(0.79\ln(\text{Re}_D - 1.64)\right)^{-2}
\]  

(4.12)

and:

\[
\text{Re}_D = \frac{\text{VD}_{t,i}}{\nu_r}
\]  

(4.13)

Now, the convection coefficient from the refrigerant can be found using the equation

\[
\text{h}_r = \frac{\text{Nu}_D k_r}{D_{t,i}}
\]  

(4.14)

and the internal resistance is simply the inverse of the product of \(\text{h}_r\) and \(A_{t,i}\):

\[
R_{\text{int}} = \frac{1}{\text{h}_r A_{t,i}}
\]  

(4.15)

With Eq. (4.10), the surface resistance is now known, and this is the first step towards determining a convection coefficient for the wires of the condensers. The process of defining this coefficient is the topic of the following section.

### 4.3 The Definition of \(h_w\)

The convection coefficient from the wires, \(h_w\), is one of the main parameters used to compare different condensers. The following analysis has been used extensively in the study of wire-on-tube condensers, and a more detailed derivation is given by Hoke, Clausing, and Swofford (1997). To begin, one must realize that the wires are not isothermal surfaces. To account for this, the area of the wires will be replaced by an effective area, which is the wire area times the fin efficiency, \(\eta\). In addition, the fact that the heat transfer from the tubes cannot be
neglected adds complexity to the problem. The result is a definition of the surface thermal resistance

\[ \frac{1}{R_{\text{surf}}} = A_t h_t + \eta A_w h_w \quad (4.16) \]

with the fin efficiency following from the analysis of an extended surface:

\[ \eta = \frac{\tanh m}{m} \quad (4.17) \]

and:

\[ m^2 = \frac{h_w S_t^2}{k_w D_w} \quad (4.18) \]

The dimensionless fin parameter, \( m \), indicates the importance of the temperature gradients in the wires, and the spacing of the tubes (\( S_t \)) plays a key role in the ultimate determination of \( h_w \). \( m^2 \) is the ratio of the internal conductive resistance of the wire to the external convective resistance between the wire and the surrounding air.

Eq. (4.16) can be rewritten in a form defining \( h_w \) explicitly as:

\[ h_w = \frac{1}{A_t \frac{h_t}{h_w} + \eta A_w} R_{\text{surf}} \quad (4.19) \]

The goal is to solve for \( h_w \) in Eq. (4.19), but \( h_t \) (or \( h_t / h_w \)) is unknown. It seems unlikely to be able to determine the convection coefficient from the tubes, \( h_t \), but it may be possible to estimate the quotient \( h_t / h_w \). Published correlations involving natural convection from a horizontal cylinder \((10^{-2} < Ra < 10^2)\) and forced convection from a cylinder in cross flow \((40 < Re < 4000)\) both reveal that \( h \propto D^{-n} \) where \( n \) is approximately equal to 0.5. So, if these correlations are assumed to be applicable to both the tubes and the wires, the ratio of the convection coefficients can be approximated as

\[ \frac{h_t}{h_w} = (D_t^*)^{-0.5} \quad (4.20) \]

where \( D_t^* = D_t / D_w \).
This estimate will be used for the following reasons. First, the wire area accounts for at least 72% of the total area in the wire-on-tube condensers being tested for this study. In addition, the convective heat transfer coefficients for the wires are expected to be considerably larger than those for the tubes. Thus, unless the fin efficiency is very low, the second term in the denominator of Eq. (4.19) will be substantially larger than the first. And Hoke, Clausing, and Swofford (1997) report that most current condenser designs appear to operate with values of $\eta$ greater than 0.85. Therefore, using Eq. (4.20) seems to be a reasonable step to eliminate $h_t$. Replacing $h_t / h_w$ in Eq. (4.19) yields:

$$h_w = \frac{1}{\left( \frac{A_t}{\sqrt{D_t}} + \eta A_w \right) R_{\text{surf}}}$$  \hspace{1cm} (4.21)

It is clear from eqs. (4.17), (4.18), and (4.21) that a transcendental equation must be solved to determine $h_w$ because $\eta$ is a function of $h_w$. An independent solution of $\eta$ is not necessary, however it is instructive to examine this data following the calculation of $h_w$. Note that the geometry of the multi-layer condenser is such that the wires and tubes, as well as the separate layers, interact extensively. Therefore, the value of $h_w$ will be different than a value of $h$ that is determined from a single wire that is similarly situated in cross-flow. The key point is that Eq. (4.21) is a simply a definition that is to be used in this experimental study. This definition has been shown by Hoke, Clausing, and Swofford (1997) to be useful in comparing the relative performance of the condensers and ultimately to correlate the results.

Once $h_w$ is accurately determined, some other definitions of performance can be discussed. For example, the definition of $h_A$ to be used in this report uses the ‘effective areas’ that have just been introduced; that is,

$$h_A = h_w \left( \frac{A_t}{\sqrt{D_t}} + \eta A_w \right)$$  \hspace{1cm} (4.22)

is the definition of $h_A$ that is used for the presentation of results. This quantity appropriately discounts the wire area by $\eta$ (accounting for non-isothermal wires) and the tube area is discounted to account for the lower convection heat transfer from the tubes.

Also, it can be deduced from Eq. (4.10) that the total conductance, $UA$, can be expressed more explicitly as:
The total conductance is an important quantity that will be presented in the results. It is a very good measure of the performance of a heat exchanger.

### 4.4 Radiation

In order to get an accurate measure of the convection coefficient $h_w$, it is necessary to subtract the heat loss due to radiation from the total heat transfer rate. The method taken here is to determine view factors based on the wire-and-tube geometry of the condenser, as well as imaginary surfaces. Once these factors are found, they can be used in general formulas to estimate radiation heat loss.

To begin, the view factor from consecutive wires within the same condenser layer should be determined. Fig. 4.3 is a schematic of this situation. This factor, $F_{w-w'}$, can be estimated by using published formulas involving infinitely long parallel cylinders:

$$ F_{w-w'} = \frac{1}{\pi} \left\{ \left( \frac{S_w}{D_w} \right)^2 - 1 + \sin^{-1} \left( \frac{D_w}{S_w} \right) - \left( \frac{S_w}{D_w} \right) \right\} $$  \hspace{1cm} (4.24)

![Figure 4.3 View factor between two wires in the same condenser layer. From Lum (1997).](image)

Now, a view factor can be developed from a single wire to an imaginary surface running tangent to the wires, as seen in Fig. 4.4. This is the same as the view factor from all of the wires to the imaginary surface. This view factor, Eq. (4.25), is determined using simple enclosure rules and the previously determined $F_{w-w'}$:
With this view factor established, the next step is to find the view factor from this imaginary surface that runs tangent to the wires to all the wires on the condenser layer. This is accomplished using the reciprocity rule because the areas of the surfaces are known:

\[
F_{\text{surf-w}} = F_{\text{w-surf}} \left( \frac{A_w}{A_{\text{surf}}} \right) = F_{\text{w-surf}} \left( \frac{\pi D_w}{S_w} \right) \quad (4.26)
\]

Since the tube passes can also be approximated as infinitely long parallel cylinders, the view factors involving them are identical to those of the wires, with the subscript ‘w’ replaced by ‘t’. That is:

\[
F_{t-t} = \frac{1}{\pi} \left\{ \sqrt{\left( \frac{S_t}{D_t} \right)^2} - 1 + \sin^{-1}\left( \frac{D_t}{S_t} \right) - \left( \frac{S_t}{D_t} \right) \right\} \quad (4.27)
\]

\[
F_{t-surf} = \frac{1 - 2F_{t-t}}{2} \quad (4.28)
\]

\[
F_{\text{surf-t}} = F_{t-surf} \left( \frac{A_t}{A_{\text{surf}}} \right) = F_{t-surf} \left( \frac{\pi D_t}{S_t} \right) \quad (4.29)
\]

Now it is necessary to calculate the view factors between the wires and the tube passes of the same layer. This is done by using the view factors involving the imaginary surfaces. For the view factor from all the wires on one side of a condenser layer to all the tube passes of the same layer:

\[
F_{w-t} = (F_{\text{w-surf}})(F_{\text{surf-t}}) \quad (4.30)
\]
The view factor from all the tube passes of a condenser layer to all the wires on one side of the same layer can be calculated using Eq. (4.31):

\[
F_{t-w} = 2 \left( F_{t-surf} \right) \left( F_{surf-w} \right)
\]  

The individual view factors from the wires and tubes to the surroundings must be established. For the view factor between the wires and the surroundings, the wires on opposite sides of the tube passes must be accounted for, as seen in Eq. (4.32):

\[
F_{w-surr} = 1 - F_{w-(wires \ on \ same \ side)} - F_{w-(wires \ on \ opposite \ side)} - F_{w-t}
= 1 - 2F_{w-w} - \left( F_{w-surf} \right) \left( 1 - F_{surf-t} \right) F_{surf-w} - F_{w-t}
\]  

This expression uses the rule that all view factors in an enclosure sum to unity. Likewise, the view factor from the tube passes to the surroundings can be expressed as:

\[
F_{t-surr} = 1 - 2F_{t-t} - F_{t-w}
\]

To obtain useable expressions for the view factors from the condenser to the surroundings, all layers of the condenser must be accounted for. The above expressions are for single layers; hence, an expression must be developed which includes all layers of the condenser. First a view factor from one layer to the surroundings (represented as an imaginary surface) will be determined. This situation is illustrated in Fig. 4.5. Then the view factor will be expanded from the single layer to account for all layers in the condenser.

As seen in Fig. 4.5, the cross-strings method can be used to determine a view factor from a single layer (surface '1') to the imaginary surface that acts as the surroundings (surface 's').

Using the cross-strings rule gives Eq. (4.34):

![Figure 4.5 View factor from one layer to the surroundings](image-url)
Note from the figure that this view factor also corresponds to the cosine of the angle-of-attack, that is:

\[ F_{1-s} = \frac{L_{\text{wave}}}{2L_{\text{layer}}} = \cos \alpha \]  \hspace{1cm} (4.35)

Now an expression can be developed for the view factor from the entire coil to the surroundings. This is accomplished using Eq. (4.35) and the number of layers, \( N_L \), in an algebraic expression:

\[ F_{\text{coil-s}} = \frac{1 + (N_L - 1)F_{1-s}}{N_L} \]  \hspace{1cm} (4.36)

This serves to increase the view factor from the coil as the number of layers is increased. Note that if the angle-of-attack were zero, the view factor is always unity.

With all the layers accounted for in Eq. (4.36), it is necessary to make separate formulas for the view factors from the wires and tubes to the surroundings. This is done not only by realizing the direct radiation, but also accounting for the layers that absorb energy from other layers. In Eq. (4.37), the first term deals with the direct radiation while the second term accounts for that energy transmitted to other layers:

\[ F_{w-s} = \left( \frac{1 + (N_L - 1)F_{1-s}}{N_L} \right) F_{w-\text{surr}} + \left( \frac{(N_L - 1)(1 - F_{1-s})\tau}{N_L} \right) F_{w-\text{surr}} \]  \hspace{1cm} (4.37)

This equation can be simplified using algebraic manipulation and substitution of Eq. (4.35):

\[ F_{w-s} = \frac{F_{w-\text{surr}}}{N_L} \{1 + (N_L - 1)(1 - \tau)\cos \alpha + (N_L - 1)\tau\} \]  \hspace{1cm} (4.38)

The view factor from the tubes to the surroundings for the entire multi-layer condenser is the same expression with the subscript 'w' replaced with 't':

\[ F_{t-s} = \frac{F_{1-\text{surr}}}{N_L} \{1 + (N_L - 1)(1 - \tau)\cos \alpha + (N_L - 1)\tau\} \]  \hspace{1cm} (4.39)
It should be noted that the transmissivity, \( \tau \), of the coil has been taken to be 0.3 for all experiments for the purpose of data reduction.

With the view factors from the condenser to the surroundings now determined, the heat loss due to radiation can be calculated. The only quantities left to determine are the temperatures of the coil and of the surroundings. The temperature of the coil is taken to be the average of the refrigerant flowing through the condenser,

\[
T_{\text{coil}} = \frac{T_{r,\text{in}} + T_{r,\text{out}}}{2} \tag{4.40}
\]

while the temperature of the surroundings is assumed to be equal to the average air temperature:

\[
T_s = \frac{T_{a,\text{in}} + T_{a,\text{out}}}{2} \tag{4.41}
\]

Eq. (4.42) is the formula that is used to determine the radiation heat loss:

\[
q_{\text{rad}} = \sigma \varepsilon \left( A_t F_{t-s} + \eta A_w F_{w-s} \right) \left( T_{\text{coil}}^4 - T_s^4 \right) \tag{4.42}
\]

Here \( \sigma \) is the Stefan-Boltzmann constant (5.67 x 10^{-8} \text{ W/m}^2\cdot\text{K}^4) and \( \varepsilon \) is the emissivity of the coil. The emissivity of the coil has been taken to be 0.80 in all experiments for the data reduction process. Note that the fin efficiency of the wires has been included in the formula for the radiation heat loss.

A FORTRAN program has been developed that implements the previously discussed assumptions and methodology to reduce the data. With the input of some of the aforementioned variables, there are several output variables of interest, including \( h_w \). A copy of the program is included in Appendix B - Data Reduction Program.
5. RESULTS AND DISCUSSION

There are two main objectives in this investigation of multi-layer wire-on-tube condensers. The first is to maximize the heat transfer from the coil, a large percentage of which is accomplished by convection from the wires or extended surfaces. The wire convection coefficient, \( h_w \), was introduced in Section 4.3 - The Definition of \( h_w \), and this quantity will be used often to compare the condensers. An additional expression frequently used to compare the heat transfer performance is acquired by multiplying \( h_w \) by a weighted area, which produces a definition of \( h_A \). The other main objective in the optimization process is to minimize the fan power required to do the aforementioned cooling. The pressure drop through the condenser, \( \Delta p \), is a measure of how much the air flow is inhibited as it proceeds through the coil. The fan power is proportional to both the pressure drop and the frontal area of the condenser.

The three main parameters that are varied for these condensers are: 1) the orientation of the coil relative to the air flow, 2) the amplitude of the saw-tooth, and 3) the wire spacing. In addition, some experiments varying the clearance, or the open space above and below the condenser, have been performed. The performance of the condensers will be discussed for each case and any advantages will be pointed out and explained. Other relevant data are also presented and discussed.

Table 1 shows the geometrical parameters of the eight wire-on-tube condensers under study. Note that each coil has a symbol assigned to it in the table. These are the symbols that will represent data points for its respective condenser in the graphical results. This convention is for the convenience of the reader and will be strictly followed, except where indicated. It should be noted that the open symbols represent condensers where the

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Sw (in.)</th>
<th>Sw (mm)</th>
<th>Amp (in.)</th>
<th>Amp (mm)</th>
<th>( \psi ) (rad)</th>
</tr>
</thead>
<tbody>
<tr>
<td>○</td>
<td>0.188</td>
<td>4.78</td>
<td>2.22</td>
<td>56.4</td>
<td>0</td>
</tr>
<tr>
<td>□</td>
<td>0.227</td>
<td>5.77</td>
<td>2.22</td>
<td>56.4</td>
<td>0</td>
</tr>
<tr>
<td>△</td>
<td>0.188</td>
<td>4.78</td>
<td>3.20</td>
<td>81.3</td>
<td>0</td>
</tr>
<tr>
<td>▽</td>
<td>0.227</td>
<td>5.77</td>
<td>3.10</td>
<td>78.7</td>
<td>0</td>
</tr>
<tr>
<td>●</td>
<td>0.188</td>
<td>4.78</td>
<td>2.57</td>
<td>65.3</td>
<td>( \pi/2 )</td>
</tr>
<tr>
<td>▼</td>
<td>0.227</td>
<td>5.77</td>
<td>2.53</td>
<td>64.3</td>
<td>( \pi/2 )</td>
</tr>
<tr>
<td>▲</td>
<td>0.188</td>
<td>4.78</td>
<td>3.37</td>
<td>85.6</td>
<td>( \pi/2 )</td>
</tr>
<tr>
<td>▼</td>
<td>0.227</td>
<td>5.77</td>
<td>3.38</td>
<td>85.9</td>
<td>( \pi/2 )</td>
</tr>
</tbody>
</table>
air flow is normal to the wires ($\psi = 0$) whereas closed symbols correspond to condensers with air flow perpendicular to the tube passes ($\psi = \pi/2$). All eight saw-tooth condensers have a relatively high angle-of-attack of $\alpha = 60^\circ$.

The raw data corresponding to the graphical results presented in the following sections and in Appendix E - Additional Plots are tabulated in Appendix D - Tabular Data. This allows the interested reader to reduce the data using any method, should the need or desire arise.

5.1 Verification of Experimental Data

In an effort to instill confidence in the results that are to be presented, this section briefly discusses the validity of the data acquisition and reduction. As will be shown in Section 5.5 - Effect of Clearance on Performance, varying the clearance can have a profound effect on the performance of the condenser. For this reason, a clearance of 0.25 in. (6.4 mm) has been selected to compare all eight coils. Each condenser has been tested at least twice at this clearance to verify that the experimental results are valid. Some of these tests were taken months apart, and the similar results will give confidence in the long-term repeatability of the data.

Fig. 5.1 is a plot of $h_w$ for two different saw-tooth condensers, coils 2 and 8, tested over the course of roughly seven months. The data sets of each condenser seem to be highly repeatable. The $h_w$ for each condenser follows a $h_w \propto V^n$ trend. The value of $n$ for Coil 2 is 0.75 for the July test and 0.77 in February. Coil 8 reveals a bit of a flatter curve, with $n = 0.57$ for the experiment in July and $n = 0.59$ in March. Coils 2 and 8 are in different orientations; Coil 2 has the air flow perpendicular to the wires whereas Coil 8 has the air flow normal to the tube passes, and this accounts for their distinct curve shapes. There is an average of a 3.3 percent difference in the magnitudes of $h_w$ for Coil 2 and a 3.2 percent difference for Coil 8. These are due to slight changes in the setup of the wind tunnel and the uncertainty in the experimental equipment. However, the magnitude of the differences are small, and these results give the reader confidence that the data has been taken and reduced in an accurate and repeatable fashion.

The air pressure drop through the condenser, $\Delta p$, is a measured quantity during the experimentation. $\Delta p$ is an important quantity because it is proportional to the fan power required to cool the coil during forced convection. Fig. 5.2 is a plot of the pressure drop for the same test runs of coils 2 and 8. The differences in magnitude of the pressure drop for the two condensers are very small and the tests appear to be extremely repeatable. The average percent difference for both Coil 2 and Coil 8 is 3.1%. These small differences are
Figure 5.1 Repeatability of the dependence of \( h_w \) on \( V \)

Figure 5.2 Repeatability of the dependence of \( \Delta p \) on \( V \)
again due to slight changes in experimental setup and the uncertainty in the pressure transducer.

5.2 Effect of Orientation on Performance

Two condenser orientations relative to the air flow are under consideration. $\psi$ is the yaw angle that defines the coil orientation. $\psi = 0$ corresponds to air flow perpendicular to the wires, and $\psi = \pi/2$ radians corresponds to air flow perpendicular to the tube passes. Of the eight coils being tested, there are four condensers with each orientation. Within these groups of four condensers, there are two separate amplitudes: nominally 2 in. (51 mm) and 3 in. (76 mm). However, as can be seen in Table 5.1, the actual amplitudes are greater than this in all cases. For simplicity in the discussion, the condensers will be referred to as the smaller and larger amplitude coils.

Very different results are seen when comparing orientations at the two separate amplitudes. Because of this, the results comparing the performance based on the orientation will be presented for both the smaller and larger amplitudes. The smaller amplitude condensers will be discussed first. This means that coils 1 and 5, as well as coils 2 and 6 can be compared. The only difference between these two pair of condensers is the wire spacing; the former (coils 1 and 5) has a wire pitch of 0.188 in. (4.78 mm) as opposed to 0.227 in. (5.77 mm) for coils 2 and 6. Since the two sets of condensers yield similar results, only data comparing coils 1 and 5 are presented.

To begin, the air-side convection coefficient, $h_w$, will be compared for coils 1 and 5. The results are presented graphically in Fig. 5.3. It is clear that the $h_w$ is much higher for coil 1 ($\psi = 0$) than for coil 5 ($\psi = \pi/2$). Both condensers follow a $h_w \propto V^n$ trend, with $n$ slightly higher for Coil 1 at 0.76 versus 0.71 for Coil 5. But the magnitude of $h_w$ is as much as 64% higher for Coil 1 with the air flow perpendicular to the wires.

For the small amplitude, clearly the $\psi = 0$ orientation is preferred from a heat transfer point of view. But the pressure drop of the air through the condenser should be examined because it gives an indication of how much fan power will be required to do the cooling. Fig. 5.4 is a plot of the pressure drop, $\Delta p$, versus air velocity for Coils 1 and 5. For both coils, the pressure drop follows a relation $\Delta p \propto V^n$, where $n$ is approximately 1.45. In addition, there is very little difference in magnitude of pressure drop for the two condensers at each velocity.

Another quantity that should be examined is the total convection heat transfer rate, $h_A$. This was first introduced in Section 4.3 - The Definition of $h_w$. It involves multiplying $h_w$ by the wire and tube areas that are discounted due to various complexities
Figure 5.3 Effect of orientation on $h_w$ for smaller amplitude

Figure 5.4 Effect of orientation on $\Delta p$ for smaller amplitude
of the wire-on-tube condenser:

\[ h_A = h_w \left( \frac{A_t}{\sqrt{D_t}} + \eta A_w \right) \]  \hspace{2cm} (5.1)

It is instructive to plot \( h_A \) versus the required fan power to do the cooling, which is a very important quantity in the consideration of domestic refrigerators. This fan power is proportional to the pressure drop through the condenser. However, it also is proportional to the duct area (\( A_{\text{duct}} \)), so a condenser that has a larger frontal area will require more fan power to do the necessary cooling. The power is a more practical measure to compare the performance of these condensers than the pressure drop, since they vary in height and width. The fan power is given by

\[ \text{Power} = \frac{\Delta p V A_{\text{duct}}}{\eta_{\text{fan}}} \]  \hspace{2cm} (5.2)

where \( V \) is the free-stream air velocity and \( \eta_{\text{fan}} \) is the fan/duct efficiency. \( \eta_{\text{fan}} \) has been conservatively set to 0.10 for all experiments. In actuality, this efficiency could be 0.30 or greater.

One gets a sense of the overall condenser performance by plotting \( h_A \) versus fan power, as is done in Fig. 5.5 for coils 1 and 5. In order to maximize heat transfer while minimizing fan power, the data points should fall as high and as far to the left as possible on the plot. As would be expected based on the results already presented, the overall performance of Coil 1 (with the \( \psi = 0 \) orientation) is superior to that of Coil 5. It is clear from Fig. 5.5 that to attain an \( h_A \) of 40 W/K (76 Btu/hr-°F), Coil 5 requires a fan power of nearly 8 W (27 Btu/hr) whereas Coil 1 requires less than 1/4 this power.

The data points corresponding to low and intermediate air velocities are congested in the lower left portion of Fig. 5.5. These results are likely to be of high interest, because the use of free-stream air velocities under 1 m/s (3.3 ft/s) is advisable to minimize dust buildup. In an effort to show these results more clearly, Fig. 5.6 shows only the data with free-stream velocities up to 1 m/s (3.3 ft/s). In this region of relevance for household refrigerators, the condenser with the air flow normal to the wires (Coil 1) again proves to be the better performer. At the 3 W (10 Btu/hr) fan power mark, the \( h_A \) for Coil 1 is roughly 46 W/K (87 Btu/hr-°F) versus 32 W/K (61 Btu/hr-°F) for Coil 5. Hence, for an equal fan power, the condenser orientated at \( \psi = 0 \) results in an \( h_A \) that is nearly 50% higher.
Figure 5.5  Overall performance for the two orientations at the smaller amplitude

Figure 5.6  Overall performance for the two orientations at the smaller amplitude
\( (V \leq 1.0 \text{ m/s}) \)
Similar findings regarding the effect of orientation for the smaller amplitude condensers can be seen by observing the results of coils 2 and 6. These results are presented graphically in Appendix E - Additional Plots (figs. E.1 - E.4).

Much different results come from comparing orientations for the larger amplitude saw-tooth condensers. Referring to Table 1, it is clear that either coils 3 and 7 or coils 4 and 8 should be compared. Once again, the only difference between these two pair of condensers is the wire spacing. For this discussion, the results from coils 3 and 7 will be presented.

Coil 3 is in the $\psi = 0$ orientation whereas Coil 7 is orientated at $\psi = \pi/2$. The results from the smaller amplitude saw-tooth condensers show that the coil with the air flow perpendicular to the wires ($\psi = 0$) has much greater heat transfer performance. For the larger amplitude condensers, however, this advantage is not nearly as evident. Fig. 5.7 is a plot of the convection coefficient, $h_w$, as a function of air velocity for coils 3 and 7. The convection coefficient is slightly higher for the $\psi = 0$ case, but only by as much as 11%. As was seen with the smaller amplitude, both condensers follow a $h_w \propto V^n$ relationship, with $n = 0.69$ for Coil 3 and $n = 0.63$ for Coil 7. This is consistent with the fact that the data associated with the $\psi = \pi/2$ orientation reveals a flatter trend when fit with a curve. Although the magnitude of $h_w$ is noticeably greater for Coil 3 at the higher air velocities, the

![Figure 5.7 Effect of orientation on $h_w$ for larger amplitude](image-url)
data show the convection coefficient of the two condensers to be very close for velocities less than 1 m/s (3.3 ft/s). These are likely to be the velocities used in domestic refrigerators, as household refrigerator condenser fans are capable of producing $\Delta p = 10$ Pa (Lum).

With this in mind, Fig. 5.8 plots the pressure drop versus air velocity for coils 3 and 7. As was seen in the previous case, the condensers follow a $\Delta p \propto V^n$ pattern, with $n = 1.37$ for both coils. The magnitude of pressure drop for Coil 7 is slightly higher than that for Coil 3, especially at higher velocities. It is seen that the pressure drop is only below 10 Pa for velocities less than 1.5 m/s. Although Coil 3 is superior from a heat transfer standpoint, it is not so much at these lower velocities. Nevertheless, Coil 3, with the air flow normal to the wires, does appear to have a slight overall performance advantage over Coil 7 for the larger amplitude. But the differences are nowhere near as great as those seen when comparing the smaller amplitude condensers, coils 1 and 5.

To further investigate the overall performance of these two larger amplitude sawtooth condensers, $h_A$ is plotted against the required fan power in Fig. 5.9. At each air velocity tested, Coil 3 shows slightly better heat transfer performance but also requires a larger fan power. For example, at $V = 2$ m/s (6.6 ft/s), $h_A$ is 7.1% higher for Coil 3 but its required fan power is also 13% higher. This highlights the importance of looking at the fan

![Figure 5.8 Effect of orientation on $\Delta p$ for larger amplitude](image-url)
power and not simply the pressure drop, since Coil 7 actually displays a higher \( \Delta p \) at this air velocity, as seen in Fig. 5.8. But Eq. (5.2) shows that the fan power is also proportional to the duct area. Although coils 3 and 7 have similar amplitudes, the condensers with the \( \psi = 0 \) orientation are considerably wider; thus the frontal area of these condensers is greater than that of the \( \psi = \pi/2 \) coils. This is the reason for the higher fan power for Coil 3 at each air velocity. This topic will be discussed in more detail in Section 5.3 - Effect of Amplitude on Performance. Nevertheless, Coil 3 still appears to be the better performer of the pair when comparing the data of Coil 3 at \( V = 1.75 \) m/s (5.7 ft/s) with Coil 7 at 2 m/s (6.6 ft/s). The heat transfer performance is similar, with \( h_A \) for Coil 3 only 2.4\% higher, but the required fan power is a considerable 28\% lower.

It is useful to look at the data by limiting the free-stream air velocities to 1 m/s (3.3 ft/s), and this plot is presented in Fig. 5.10. The difference in overall performance is not nearly as great as was seen for the smaller amplitude coils. At \( V = 1 \) m/s (3.3 ft/s), \( h_A \) is 7.4\% higher for Coil 3 while the required fan power is 19\% higher. These differences are even smaller for the data corresponding to the lower air velocities. At an equal fan power of 3 W (10 Btu/hr), \( h_A \) is approximately 50 W/K (95 Btu/hr-°F) for Coil 3 compared to 49 W/K (93 Btu/hr-°F) for Coil 7.

Very similar results are seen when comparing the performance of coils 4 and 8.
These are also larger amplitude condensers that differ only by their orientation relative to the air flow. They have a slightly wider wire spacing than that of coils 3 and 7. The data that compare coils 4 and 8 are presented graphically in figs. E.5 - E.8.

The difference in results for the separate saw-tooth amplitudes may come as a surprise. The reason for this difference is hypothesized to be caused by varying blocking of the air flow as it proceeds through the condenser. Because of the way these condensers were bent to create the saw-tooth design, the peaks are somewhat flattened out. This is especially true for the case where the air flow is perpendicular to the tube passes (the \( \psi = \pi/2 \) orientation). The tubes directly block the air flow at the peaks for this orientation. This causes the air flow to channel down the middle of the condenser, and the entire condenser is not properly utilized. For the smaller amplitude condensers, there are a greater number of peaks because the number of peaks is proportional to \((1/\text{amplitude})\). In addition, the peaks are flatter and more elongated for these smaller condensers. On the other hand, the larger condensers have fewer peaks and they tend to be sharper. Hence, it is likely that the larger coils in the \( \psi = \pi/2 \) orientation experience a more uniform air flow through the wire-and-tube matrix.
5.3 Effect of Amplitude on Performance

Although the effect of varying amplitude can be deduced implicitly from the previous section, it is worth discussing in slightly more detail here. For a condenser with the air flow normal to the wires, there is not a huge difference in performance. The comparison of \( h_w \) for coils 2 and 4 is presented in Fig. 5.11. These coils are both at the \( \psi = 0 \) orientation and are identical with the exception of their amplitudes. Similar findings come from comparing coils 1 and 3, however the results from these condensers were already shown in Section 5.2 - Effect of Orientation on Performance.

The convection coefficient is slightly higher for the larger amplitude condenser (Coil 4) at each velocity tested. At \( V = 0.2 \text{ m/s} \) (0.66 ft/s), \( h_w \) is 36% higher for Coil 4. The difference decreases as the air velocity increases, and at \( V = 2 \text{ m/s} \) (6.6 ft/s), the convection coefficient is only 8.9% higher for Coil 4. The reason for this slight advantage is again due to the fact that there is a more uniform air flow through the wire-and-tube matrices of the larger amplitude coils. When the condensers are confined as they are in the testing and as they would be in an actual refrigerator, the air flow is slower near the confining walls due to boundary layer effects. With the taller 'duct' for the larger amplitude, the air velocity is higher for a larger percentage of the condenser than it would be for the smaller amplitude condenser and there is going to be a slightly higher convection cooling rate. Also, as was explained earlier, the smaller amplitude condensers have a number of elongated, flattened out peaks. These regions act as condensers with an angle-of-attack of \( \alpha = 0^\circ \). This is another reason that the smaller amplitude coils show a smaller \( h_w \), as Hoke, Clausing, and Swofford (1997) determined that \( h_w \) decreases with \( \alpha \).

As can be seen in Fig. 5.12, the air pressure drop across the condensers is slightly lower for the larger amplitude condenser, Coil 4. This makes sense because this condenser has six layers whereas Coil 2 has eight layers. Although the two coils have the same total area and are identical before being bent into a saw-tooth shape, bending Coil 4 into a larger amplitude saw-tooth requires it to have fewer layers. With this configuration, the air flow does not experience as much resistance as it does for Coil 2. However, because the condensers follow a \( \Delta p \propto V^n \) pattern, with \( n \) approaching 1.5, one would expect that there would be a lesser difference in pressure drop for the lower air velocities, which are of interest in domestic refrigerators. This is verified by Fig. 5.12.

Although the pressure drop is higher for Coil 2, the frontal area of Coil 4 is greater due to its larger amplitude. This is important because the required fan power is proportional to the duct area, as seen in Eq. (5.2). Table 5.2 gives the frontal dimensions for all eight condensers under study. In this table 'FA' refers to the frontal area of the saw-
**Figure 5.11** Effect of amplitude on $h_w$ for $\psi = 0$

**Figure 5.12** Effect of amplitude on $\Delta p$ for $\psi = 0$
tooth condenser. This is approximately equal to the duct area, although the frontal area is slightly smaller due to the allowance for clearance. From Table 5.2, it can be seen that the frontal area of Coil 4 is 40% larger than that of Coil 2.

Table 5.2 Frontal dimensions and areas for the condensers

<table>
<thead>
<tr>
<th></th>
<th>Amplitude (m)</th>
<th>Width (m)</th>
<th>FA (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coil 1</td>
<td>0.0564 (2.22 in.)</td>
<td>0.711 (28.0 in.)</td>
<td>0.0401 (62.2 in²)</td>
</tr>
<tr>
<td>Coil 2</td>
<td>0.0564 (2.22 in.)</td>
<td>0.711 (28.0 in.)</td>
<td>0.0401 (62.2 in²)</td>
</tr>
<tr>
<td>Coil 3</td>
<td>0.0813 (3.20 in.)</td>
<td>0.711 (28.0 in.)</td>
<td>0.0578 (89.6 in²)</td>
</tr>
<tr>
<td>Coil 4</td>
<td>0.0787 (3.10 in.)</td>
<td>0.711 (28.0 in.)</td>
<td>0.0560 (86.8 in²)</td>
</tr>
<tr>
<td>Coil 5</td>
<td>0.0653 (2.57 in.)</td>
<td>0.582 (22.9 in.)</td>
<td>0.0380 (58.9 in²)</td>
</tr>
<tr>
<td>Coil 6</td>
<td>0.0643 (2.53 in.)</td>
<td>0.582 (22.9 in.)</td>
<td>0.0374 (57.9 in²)</td>
</tr>
<tr>
<td>Coil 7</td>
<td>0.0856 (3.37 in.)</td>
<td>0.582 (22.9 in.)</td>
<td>0.0498 (77.2 in²)</td>
</tr>
<tr>
<td>Coil 8</td>
<td>0.0859 (3.38 in.)</td>
<td>0.582 (22.9 in.)</td>
<td>0.0499 (77.4 in²)</td>
</tr>
</tbody>
</table>

This difference in frontal area is a key factor in the required fan power to cool the condensers. Fig. 5.13 compares the overall performance of coils 2 and 4 by plotting $h_A$ versus fan power. Note that for each air velocity tested, the fan power is somewhat higher for Coil 4. Although the air pressure drop through this condenser is actually lower, the larger frontal area increases the necessary fan power. But the larger amplitude condenser is the better overall performer; to attain an $h_A$ of 60 W/K (114 Btu/hr-°F), Coil 2 requires a fan power of roughly 14 W (48 Btu/hr) whereas Coil 4 requires only about 10 W (34 Btu/hr).

Fig. 5.14 is a plot of $h_A$ versus fan power for the same two condensers, with the free-stream air velocity being limited to 1 m/s (3.3 ft/s). The slight overall performance advantage for the larger Coil 4 is clear from this plot. Comparing the data at the 3 W (10 Btu/hr) fan power mark, it is determined that Coil 4 has superior heat transfer. At 48 W/K (91 Btu/hr-°F), $h_A$ is 4.4% higher for the larger amplitude condenser with the $\psi = 0$ orientation.

Appendix E - Additional Plots includes graphical results comparing the performance of coils 1 and 3 (see figs. E.9 - E.12). Although these results were presented separately in Section 5.2 - Effect of Orientation on Performance, it may be instructive to compare the data on the same plots.

One can conclude that for condensers with air flow perpendicular to the wires, amplitude does not play a pivotal role in the overall performance. However, as was seen
Figure 5.13 Overall performance comparison for the different amplitudes at $\psi = 0$

Figure 5.14 Overall performance comparison for the different amplitudes at $\psi = 0$

$(V \leq 1.0 \text{ m/s})$
earlier, when the saw-tooth condenser is situated in the $\psi = \pi/2$ orientation, amplitude plays a huge role in the performance. To demonstrate this, the data from coils 6 and 8 will be presented and discussed. This pair of condensers is identical to coils 5 and 7 with the exception of their wire spacings. Coils 5 and 7 show extremely similar results, however the data from these condensers were also presented in Section 5.2 - Effect of Orientation on Performance.

Fig. 5.15 compares the convection coefficient for coils 6 and 8. When fit with a curve, the data from both condensers follow a $h_w \propto V^n$ pattern, with $n = 0.68$ for Coil 6 and $n = 0.57$ for Coil 8. For air velocities over 1 m/s (3.3 ft/s), the convection coefficient is roughly 50% higher for the larger Coil 8. For the low to intermediate air velocities that are of interest for domestic refrigerators, there is an even larger percent difference, although the magnitudes of $h_w$ are smaller.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure5.15.png}
\caption{Effect of amplitude on $h_w$ for the $\psi = \pi/2$ orientation}
\end{figure}

From a heat transfer standpoint, the advantage gained by using a larger amplitude saw-tooth condenser in the $\psi = \pi/2$ orientation is obvious. And as is seen in Fig. 5.16, the pressure drop through coils 6 and 8 is nearly identical at each air velocity. Both coils follow the familiar $\Delta p \propto V^n$ pattern, with $n$ approaching 1.4. The comparability in the magnitude of the pressure drop at each air velocity is somewhat surprising. Because the
larger amplitude condenser has fewer layers, one would expect it to have a lower overall pressure drop as was seen when comparing the condensers with the $\psi = 0$ orientation. The reason $\Delta p$ for Coil 6 is lower than expected is indirectly related to the blockage of air flow by the tube passes. If the air is to flow through the wire-and-tube matrix for these smaller $\psi = \pi/2$ saw-tooth condensers, it is restricted to channel down the center of the coil due to the blockage at the peaks. Because of this, much of the air flows above and below the condenser (in the clearance area). Also, for the coils with the $\psi = \pi/2$ orientation, there are wire-free tube bends on the sides of the condenser and these regions permit much of the air flow. With much of the air not flowing directly through the coil, the pressure drop is somewhat relieved. This topic is discussed in more detail in Section 5.4 - Effect of Clearance on Performance.

\[ \text{Figure 5.16 Effect of amplitude on } \Delta p \text{ for the } \psi = \pi/2 \text{ orientation} \]

Since the pressure drop through each condenser is essentially equal, and the frontal area of the larger Coil 8 is 34% greater than that of Coil 6, it comes as no surprise that the fan power is higher for Coil 8 at each air velocity tested. But, as was seen in Fig. 5.15, the heat transfer data is dominant for Coil 8. With this in mind, results showing that the overall performance is superior for the larger amplitude $\psi = \pi/2$ saw-tooth condenser are not unexpected. Fig. 5.17 is a plot of $h_A$ versus fan power for coils 6 and 8. Coil 6
Figure 5.17 Overall performance comparison for the different amplitudes at $\psi = \pi/2$

requires 10 W (34 Btu/hr) of fan power to reach an $h_A$ of 40 W/K (76 Btu/hr-°F) while Coil 8 requires 1/5 that power, only 2 W, to attain an $h_A$ of 40 W/K.

Fig. 5.18 shows the data from Fig. 5.17 that correspond to free-stream air velocities up to 1 m/s (3.3 ft/s). The curves depicting $h_A$ as a function of fan power give evidence to the much higher magnitude of heat transfer for the larger amplitude condensers when the air flow is perpendicular to the tubes. At an equal fan power of 3 W (10 Btu/hr), $h_A = 45$ W/K (85 Btu/hr-°F) for Coil 8 compared to $h_A = 33$ W/K (63 Btu/hr-°F) for Coil 6.

Although the data exhibiting the performance of coils 5 and 7 are presented separately in Section 5.2 Effect of Orientation on Performance, they are repeated on the same plots in figs. E.13 - E.16 for the convenience of the reader. These figures show the same trends that have just been observed for coils 6 and 8.

In conclusion, the amplitude appears to have a significant influence on the performance of saw-tooth condensers that are orientated with the air flow perpendicular to the tubes. The tube passes directly block the flow of air, and for the smaller amplitude condensers, the blockage is more substantial. The result is that the air flow is channeled down the center of the condenser, and around the condenser, and the heat transfer potential
5.4 Effect of Wire Spacing on Performance

Referring to Table 5.1, there are two different wire spacings being analyzed. They are 0.188 in. (4.78 mm) and 0.227 in. (5.77 mm). There are more total wires on the condensers with the smaller wire spacing (220 vs. 182). Since the diameter of the wires is identical for all condensers, the total wire area is greater for those coils with the larger number of wires.

Fig. 5.19 is a plot comparing the convection coefficient for Coils 3 and 4, which vary only by wire spacing. It is seen that \( h_w \) is slightly higher at each air velocity for Coil 4, which has the wider wire spacing.

Although it has a lower \( h_w \), Coil 3 has the greater number of wires, and therefore a greater total area. Thus, it is instructional to investigate the total convection heat rate, \( h_A \), as a function of air velocity for these two condensers. Fig. 5.20 clearly shows that the increased area due to the additional wires results in a higher \( h_A \) for the tighter wire spacing.
Figure 5.19 Effect of wire spacing on $h_w$

Figure 5.20 Effect of wire spacing on $h_A$
(Coil 3). At $V = 2$ m/s (6.6 ft/s), $h_A$ is 13% higher for Coil 3. This difference is not as great for the low to intermediate velocities, where domestic refrigerators would be operating. But this plot shows that the total convection rate is at least as high for Coil 3, even though $h_w$ for Coil 4 slightly higher. It should be pointed out that the added wires due to the narrower wire spacing mean additional materials cost.

Despite the fact that the condensers with the tighter wire spacing perform at least as well from a heat transfer standpoint, the pressure drop through these coils is seen to be slightly higher, especially at the higher air velocities. This is not surprising, since the tighter wire-and-tube matrix results in a higher maximum velocity ($V_{\text{max}}$) through the coil, and $\Delta p \propto V_{\text{max}}$. The topic of $V_{\text{max}}$ will be discussed in more detail in Section 5.6 - Other Results. It is important to note that the difference in pressure drop is seen to be nearly negligible for the lower air velocities tested, which are of primary interest for domestic refrigerators. Also, as seen in Fig. 5.20, there is not a large advantage in heat transfer performance for either condenser at these lower air velocities.

With this in mind, Fig. 5.21 compares the overall performance of these condensers with the different wire spacings. This plot only includes low to intermediate air velocities, $V \leq 1.0$ m/s (3.3 ft/s). Clearly, there is not a large difference in performance between coils 3 and 4. In fact, for $V \leq 0.5$ m/s (1.6 ft/s), the data for these two condensers virtually

![Figure 5.21](image)

**Figure 5.21** Overall performance comparison based on the wire spacing ($V \leq 1.0$ m/s)
coincide. At an equal fan power of 3 W (10 Btu/hr), the heat transfer performance for these two coils is nearly identical: $h_A = 50\, \text{W/K (95 Btu/hr-}^\circ\text{F)}$ for Coil 3 and $h_A = 48\, \text{W/K (91 Btu/hr-}^\circ\text{F)}$ for Coil 4.

The trend of a higher $h_w$ but lower $h_A$ for increased $S_w$ is observed for all the condensers. Figs. E.17 - E.19 compare the performance of coils 1 and 2. These sawtooth condensers are at $\psi = 0$, but the same general trends also apply for the $\psi = \pi/2$ coils.

These results seem to imply that there is little dependence on wire spacing with regard to overall condenser performance. However, only two wire spacings were studied in this investigation. Swofford (1995) and Hoke (1995) report that as the dimensionless wire spacing, $S_w$ (i.e. $S_w/D_w$), is decreased, the performance of the coil decreases. So it is likely that if wire pitch became too small, the performance of the condensers would suffer.

5.5 Effect of Clearance on Performance

The clearance has been found to have a strong influence on performance for sawtooth shaped condensers. The clearance is the open space above and below the condenser as it is situated in the wind tunnel. The condenser is lifted off the floor of the wind tunnel a specified distance using a set of small, low-conductivity blocks. As was explained in Section 3.3 - Variable Height Wind Tunnel Test Section, the wind tunnel being used for the testing can be adjusted so that its height varies from 2 in. (50.8 mm) to 6 in. (152 mm). Thus, the height of the wind tunnel can be adjusted in such a fashion as to allow a spacing between the top of the condenser and the ceiling of the wind tunnel that is equal to the aforementioned spacing underneath the condenser. This situation is illustrated in Fig. 5.22.

Extreme care has been taken to make sure that the clearance is equal for all the condensers as they were tested. This is because, as will be seen, any variance in the clearance can cause the performance parameters (notably $h_w$ and $\Delta p$) to change appreciably. Thus, for each of the coils tested, there is a 0.25 in. (6.4 mm) clearance above and below the condenser. All of the results presented thus far have been with this 1/4 inch clearance in an effort to get consistent results. The exception to this is those tests that were performed to examine the effect of varying the clearance. It is these experiments that are the focus of this section.

To see how changing the clearance can affect the convection coefficient, refer to Fig. 5.23. This is a plot of $h_w$ versus air velocity for two sets of experiments on Coil 3. For one of the tests, the clearance was set to the normal 1/4 inch level. But clearly for the experiment where the clearance is reduced to 1/8 in. (3.2 mm) there is a marked increase in
Figure 5.22 Definition of clearance with a side view of the wind tunnel test section.

Figure 5.23 Effect of varying the clearance on $h_w$ for Coil 3.
the convection coefficient. In fact, the $h_w$ associated with the smaller clearance is at least 10% higher at each velocity. More importantly, the difference is even greater at the lower air velocities. For instance, the convection coefficient is 30% higher for the more confined condenser at the lowest velocity tested, $V = 0.2$ m/s (0.66 ft/s). It should be mentioned that in Fig. 5.23 and in some of the following plots, the symbol notation for the graphical results that was introduced earlier has been altered slightly. The closed symbols refer to the coil with the larger clearance (1/4 inch).

It appears that making the clearance as small as possible would be beneficial based on the heat transfer results. But the drawback to reducing the clearance is that the pressure drop is increased considerably. Fig. 5.24 shows the air pressure drop through Coil 3 for the same two clearance levels. As has been seen in other results, the pressure drop difference becomes greater as the air velocity gets higher. The reason for this can be seen when the data are fit with a power curve, with both cases following a $\Delta p = C V^n$ trend where $C$ and $n$ are constants. With the 1/8 inch clearance, $C = 7.39$ and $n = 1.46$. The curve corresponding to the 1/4 inch clearance is somewhat flatter, with $C = 5.77$ and $n = 1.37$. Referring to Fig. 5.24, for the highest velocity tested, $V = 2.0$ m/s (6.6 ft/s), the pressure drop for the smaller clearance is 25% higher. However, for the lower air velocities, and especially for free-stream velocities less than 0.75 m/s (2.5 ft/s), the

![Figure 5.24 Effect of varying the clearance on $\Delta p$ for Coil 3](image-url)
pressure drop for the two experiments is quite close.

The reason the performance results are so dependent on the magnitude of the clearance has to do with the pattern of the air flow through and around the condenser. For the larger clearances, some of the air flows above and beneath the coil. On the other hand, for the smaller clearances, not as much air flows around the coil because the confining walls are too close to the top and bottom of the condenser. The no-slip rule of viscous fluid flow insures that the velocity is zero at the wall and cannot be too high very close to the wall. In other words, for the condensers confined with a smaller clearance, there is more air flow being forced through the actual wire-and-tube matrix of the coil. This explains why $h_w$ is higher, since more of the air flow is directly cooling the wires. Also, because more of the air flow is being forced through the actual coil, it makes sense that the pressure drop increases because the air flow faces more resistance.

There is a considerable difference in the amount of free area in the wind tunnel test section for the two clearance levels. Table 5.3 gives the ratio of free area to actual condenser frontal area for both clearances. Note that the free area due to the open tube bends on the $\psi = \pi/2$ condensers has been accounted for, increasing the ratios for coils 5-8.

<table>
<thead>
<tr>
<th>1/4 in. (6.4 mm) Clearance</th>
<th>1/8 in. (3.2 mm) Clearance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Free Area (m²)</td>
<td>% Free Area / FA</td>
</tr>
<tr>
<td>Coil 1</td>
<td>0.00903</td>
</tr>
<tr>
<td>Coil 2</td>
<td>0.00903</td>
</tr>
<tr>
<td>Coil 3</td>
<td>0.00903</td>
</tr>
<tr>
<td>Coil 4</td>
<td>0.00903</td>
</tr>
<tr>
<td>Coil 5</td>
<td>0.0115</td>
</tr>
<tr>
<td>Coil 6</td>
<td>0.0114</td>
</tr>
<tr>
<td>Coil 7</td>
<td>0.0124</td>
</tr>
<tr>
<td>Coil 8</td>
<td>0.0125</td>
</tr>
</tbody>
</table>

If the pressure drop were much higher for the more confined condenser at the low air velocities, then it would be advisable to find an ideal clearance level to maximize heat transfer while still keeping the required fan power to a minimum. But the pressure drop at low air velocities, which are of interest for use in household refrigerators, is comparable for the two clearance levels tested. With this in mind, Fig. 5.25 compares the overall...
performance of Coil 3 for these two clearances. This is a plot of $h_A$ versus fan power, with the free-stream air velocities limited to 1 m/s (3.3 ft/s). It is seen that the performance of Coil 3 with the smaller 1/8 inch clearance is superior for the lower air velocities. However, with the air velocity approaching 1 m/s, this advantage is lessened as the curves fitting the data begin to converge. At an equal fan power of 3 W (10 Btul/hr), the benefit of the smaller clearance is minimal: $h_A = 52 \text{ W/K (99 Btul/hr-}^\circ\text{F)}$ versus $50 \text{ W/K (95 Btul/hr-}^\circ\text{F).}$

![Graph showing performance comparison for Coil 3 with varying clearance](image)

**Figure 5.25** Overall performance comparison for Coil 3 with varying clearance ($V \leq 1.0 \text{ m/s}$)

Similar results are seen when varying the clearance level of Coil 4. These data are presented graphically in Appendix E - Additional Plots, figs. E.20 - E.22. Based on these findings, there does not seem to be a reason to make the clearance anything more than 1/8 in. (3.2 mm), assuming the fan driven air velocity is less than 1 m/s (3.3 ft/s). It is possible that if the air velocities are kept very low, there may be advantages to making the clearance even less than 1/8 inch. However, a small clearance is advisable to avoid any vibration noise that may occur during operation.

This is a good time to discuss another topic that is somewhat related to clearance and the pattern of air flow. It deals with the condensers with the air flow normal to the tube.
passes (the $\psi = \pi/2$ orientation) and the wire-free tube bends that are found on the sides of these coils. The way these condensers were fabricated and bent, there is about 1.25 in. (31.8 mm) of open tube bends on either side of the condenser. Because of the logistics of the wind tunnel, the easiest way to test these condensers was to put the wood forms to the sides of the coil and let the air flow through these regions. A top view of the wind tunnel set up in this fashion is presented in Fig. 5.26.

![Figure 5.26 Top view of the wind tunnel as originally tested for $\psi = \pi/2$ coils](image)

The results from testing these condensers in this configuration have already been presented in the previous sections. The lack of performance of the smaller $\psi = \pi/2$ coils provides motivation to try to modify these condensers in some way to improve their performance. One way to do this is to block off the wire-free portions on the sides of the condenser, as seen in Fig. 5.27. Because there are no wires on these areas, a sizable fraction of the air flow is certain to flow through the channel-like passes. By blocking off these channels, the entire condenser is effectively covered in wires and the air is forced to flow through the wire-and-tube matrix. The method taken to accomplish this blockage was to construct a set of two smooth balsa wood boards and punch holes to accommodate the tube bends that are jutting out. Then these boards are snugly fit onto the sides of the condenser and any remaining holes in the balsa wood are patched to prevent air from
escaping. Note that the balsa wood boards are gently curved so that the air entering the test section is free of instabilities.

Figure 5.27 Top view of the wind tunnel after modification for testing

Results from these tests are promising. As anticipated, the convection coefficient increased noticeably when the wind tunnel is modified in this way. Fig. 5.28 compares \( h_w \) for Coil 6 with both the sides open and the sides blocked. There is as much as a 27% difference between the two cases. The clearance was the same for these two experiments, and the only difference was installing the balsa wood boards to manipulate the air flow. For Coil 6, blocking the open tube bends reduces the free area from 0.0114 m\(^2\) (17.7 in.\(^2\)) to 0.00661 m\(^2\) (10.3 in.\(^2\)). Thus, it can be concluded that there was a lot of air passing through the tube bend areas for the smaller \( \psi = \pi/2 \) coils.

Of course, forcing the air flow through the wire-and-tube matrix is going to increase the air pressure drop through the condenser. This can be seen in Fig. 5.29, which compares \( \Delta p \) for these two experiments. Whenever the air flow is forced through the condenser there is going to be an increase in \( \Delta p \) in addition to the improvement in heat transfer performance. For the lower air velocities, the difference in pressure drop is not overwhelming, but the heat transfer gained by blocking the sides is also relatively small.
Figure 5.28 Effect of blocking the sides of Coil 6 on $h_w$.

Figure 5.29 Effect of blocking the sides of Coil 6 on $\Delta p$. 
With this in mind, it is a good idea to investigate whether blocking the sides of the smaller \( \psi = \pi/2 \) condensers greatly improves the overall performance in the velocity regimes that are relevant to household refrigerators. It should be mentioned that although the pressure drop is increased by blocking open tube bends, the area of the duct is reduced slightly and this has the effect of reducing the fan power. \( h_A \) is plotted against fan power for these two experiments in Fig. 5.30, with the free-stream air velocity limited to 1 m/s (3.3 ft/s). These data are fit with power curves, which make it clear that as the air velocity increases, removing the open sides of Coil 6 leads to better heat transfer performance. For an equal fan power of 3 W (10 Btu/hr), \( h_A \) is 24% higher when the tube bends of Coil 6 are blocked.

![Figure 5.30 Effect of blocking the sides of Coil 6 on overall performance (V ≤ 1.0 m/s)](image)

The other method to try to improve the performance of Coils 5 and 6 is to reduce the clearance when testing these condensers. It was observed earlier in this section that lowering the clearance for the \( \psi = 0 \) condensers resulted in improved performance. When reducing the clearance to 1/8 inch for Coil 6, there is a noticeable gain in the magnitude of the convection coefficient, \( h_w \). This is shown in Fig. 5.31, which compares the two clearance levels for Coil 6. The gain is not as great at the higher air velocities as was
observed in Fig. 5.28 with the open sides of Coil 6 blocked, but it is appreciable nonetheless. Also, the gain in heat transfer for the low to intermediate velocities is very similar to that seen when blocking the sides of Coil 6.

![Graph showing effect of varying clearance on heat transfer](image)

**Figure 5.31** Effect of varying the clearance on $h_w$ for Coil 6

Considering that lowering the clearance for the $\psi = 0$ coils resulted in a substantial gain in pressure drop, investigating the pressure drop while varying the clearance for Coil 6 ($\psi = \pi/2$) yields surprising results. As can be seen in Fig. 5.32, there is not much increase in pressure drop for any of the air velocities when reducing the clearance to 1/8 inch. In fact, for the lower velocities, the pressure drop is nearly identical for the two clearances tested. This is likely due to the fact that a good portion of the air flow is still channeling down the open sides where there are no wires. But by reducing the clearance, the air flow going above and below the condenser is lessened, and more of the air is flowing near the all important wires, thereby increasing $h_w$.

The overall performance for Coil 6 at the two clearances is shown in Fig. 5.33. These data are limited to the low to intermediate air velocities that are relevant to domestic refrigerators. The data have been fit with power curves with the relationship $h_A = C \cdot (\text{Power})^n$ where 'C' and 'n' are constants. $n$ is approximately equal to 0.3 for each case, and as a result the curves have very similar shapes. However, the magnitude of $h_A$ is
Figure 5.32 Effect of varying the clearance on Δp for Coil 6

Figure 5.33 Overall performance comparison for Coil 6 with varying clearance (V ≤ 1.0 m/s)
higher for the 1/8 inch clearance, because C is 21\% higher. Thus, as the air velocity
increases, the heat transfer advantage for the smaller clearance becomes greater. At the 3 W
(10 Btu/hr) power mark, hA is 21\% higher for Coil 6 with the 1/8 inch clearance.

The effect of varying the clearance on condenser performance for Coil 5 is shown
in figs. E.23 - E.25. The results are very similar to those seen for the data just presented
for Coil 6.

5.6 Other Results

There are numerous other results that should be presented. As was discussed in
Section 4.2 - Heat Exchanger Analysis, the data were reduced using heat exchanger
methodology. The $\psi = 0$ coils are modeled as shell-and-tube heat exchangers while the
condensers with the $\psi = \pi/2$ orientation are analyzed as counterflow heat exchangers. In
the data reduction, the heat exchanger effectiveness of each condenser is calculated, and
from this the number of transfer units (NTU) is determined. These quantities are essential
in the ultimate calculation of $h_w$ and the other heat transfer parameters that are used to
compare the coils.

Fig. 5.34 is a plot of the heat exchanger effectiveness for all eight condensers.
Recall that the definition of the effectiveness is:

$$\varepsilon \equiv \frac{q}{q_{\max}} = \frac{q}{C_{\text{min}} (T_{r,\text{in}} - T_{a,\text{in}})}$$

(5.3)

The effectiveness is presented for the interested reader, but it will be explained shortly that
the value of this data is limited. In the plot, the two condensers that have proven to be the
worst performers (coils 5 and 6), show very low effectiveness values relative to the other
six condensers. This is because the total heat rate, $q$, from these coils is relatively small.
The data from the other six condensers show a large range of effectiveness values.

Likewise, the results comparing the NTU for the eight condensers should be
presented, but the value of these data are limited. NTU is defined as:

$$\text{NTU} = \frac{UA}{C_{\text{min}}}$$

(5.4)

This dimensionless quantity is a measure of the size of the heat exchanger. NTU is plotted
as a function of velocity for all eight condensers in Fig. 5.35. Again, the relatively poor
Figure 5.34 Heat exchanger effectiveness vs. air velocity for all eight coils

Figure 5.35 NTU vs. air velocity for all condensers
The performance of coils 5 and 6 can be noted in the plot. The remaining condensers show a wide range of higher NTU values.

The reason these results should not be scrutinized has to do with the heat capacity rates. The mass flow rate of the air is fairly consistent between condensers due to the predetermined test velocities, but the specific heat can vary due to changes in the ambient temperature, resulting in a range of air heat capacity rates, $C_a$. In addition, the heat capacity rate of the refrigerant is arbitrary because the mass flow rate of the refrigerant varies widely between experiments. Eqs. (5.3) and (5.4) both show the minimum heat capacity rate, $C_{\text{min}}$, in the denominator. For the low air velocities, the capacity rate of the air is $C_{\text{min}}$. However, as the air velocity increases, the capacity rate of the refrigerant, $C_r$, eventually becomes the minimum. This transition is the cause of the valleys in the data of figs. 5.34 and 5.35. The fact that $C_{\text{min}}$ differs considerably between experiments removes some value from the effectiveness and NTU results. The variation in $C_{\text{min}}$ for each air velocity tested is shown in Table 5.4. Another reason to use caution when studying the effectiveness and NTU results is the fact that the two condenser orientations are modeled as different types of heat exchangers.

The conductance was introduced in Section 4.3 - The Definition of $h_w$, but will be repeated here:

$$UA = \frac{1}{h_w \left( \frac{A_t}{\sqrt{D_t}} + \eta A_w \right)} + \frac{1}{h_r A_{ti}}$$

(5.5)
The conductance not only accounts for the surface resistance, which is the first term in the denominator, but also the internal resistance from the refrigerant inside the tubes. Fig. 5.36 is a plot comparing the conductance for all of the coils. Note that in this plot, all of the condensers are fairly well grouped with the exception of coils 5 and 6. Also, the coils with the tighter wire spacings have a slightly higher conductance than their wider wire spacing counterparts. For example, Coil 1 has a greater UA than Coil 2. This is due to the larger number of wires resulting in a higher total area.

![Figure 5.36 UA vs. air velocity for all eight condensers](image)

Before presenting the $h_w$ results for all the condensers, the fin efficiencies associated with the wires should be discussed. Recall that the fin efficiency, $\eta$, and the convection coefficient together formed a transcendental equation in which the two quantities had to be solved for iteratively. Fig. 5.37 shows the fin efficiency versus air velocity for all eight condensers tested. Note that coils 5 and 6, which have been proven to be the worst performers from a heat transfer standpoint, have the highest fin efficiencies, especially at the higher air velocities. This is due to the fact that these condensers are rejecting a relatively low amount of heat; hence, the temperature gradient in the wires of coils 5 and 6 is not as great and the fin is viewed as more 'efficient'. It should be noted that $\eta$ for all condensers is well above 0.80, as would be expected for current wire-on-tube condenser designs (Hoke, Clausing, and Swofford).
For the convenience of the reader, a plot with the convection coefficient for all eight condensers is shown in Fig. 5.38. This plot shows how all of the condensers with the exception of the smaller amplitude $\psi=\pi/2$ coils (5 and 6) actually perform quite comparably. In fact, at $V = 2$ m/s (6.6 ft/s), the $h_w$ for these six condensers are within 12% of each other. It has been shown that the performance of coils 5 and 6 can be improved by blocking the open tube bends or by reducing the clearance, but this plot compares the coils as they were manufactured and with equal clearance.

The pressure drop of the air through the coil for all eight condensers is presented in Fig. 5.39. The results show that the pressure drop is pretty close for all the condensers. Even at $V = 2$ m/s (6.6 ft/s), $\Delta p$ for all eight coils is between 15 and 20 Pa (0.060 to 0.80 in. H$_2$O). More importantly, the pressure drop is very close for all the condensers at the lower velocities: those seen in household refrigerators. Fig. 5.39 shows that even the poorly performing coils 5 and 6 have pressure drops within the ranges of the other condensers.

Although useful in that it gives the total pressure drop through the condenser, the data presented in Fig. 5.39 can be somewhat misleading. This is because the condensers each have a different number of layers due to variations in frontal area. Table 5.5 shows
Figure 5.38 $h_w$ versus air velocity for all eight condensers

Figure 5.39 $\Delta p$ vs. $V$ for all eight condensers
how many layers are associated with each condenser. Some of the condensers did not have complete layers at the ends, and if so the number of layers was rounded to the nearest integer.

**Table 5.5** The number of layers associated with each condenser

<table>
<thead>
<tr>
<th>Coil</th>
<th>Coil 1</th>
<th>Coil 2</th>
<th>Coil 3</th>
<th>Coil 4</th>
<th>Coil 5</th>
<th>Coil 6</th>
<th>Coil 7</th>
<th>Coil 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>( N_L )</td>
<td>8</td>
<td>8</td>
<td>6</td>
<td>6</td>
<td>9</td>
<td>9</td>
<td>7</td>
<td>7</td>
</tr>
</tbody>
</table>

It is useful to investigate the pressure drop per layer for each condenser. This is done simply by dividing the data from Fig. 5.39 by the number of layers in the respective condenser. The results are shown in Fig. 5.40. As can be seen, the data becomes a bit more spread out when presented this way. The condensers with the lowest pressure drop per layer are coils 5 and 6. These coils allow the air to flow through the open tube bends, reducing \( \Delta p \).

**Figure 5.40** \( \Delta p \) per layer versus air velocity for all eight condensers
The pressure drop and the pressure drop per layer are important quantities, but a more meaningful measure of the performance of the condensers is the fan power required to produce a certain level of heat transfer. The fan power not only takes into account the pressure drop through the coil, but also the area of the duct (related to the frontal area, or amplitude, of the condenser). Throughout this report when comparing the performance of condensers, the total convection heat rate, \( h_A \), has been plotted against the required fan power. It is instructive to plot these results for all the coils, and this is done in Fig. 5.41. The most evident observation that can be made from the figure is the relative lack of performance of the two smaller \( \psi=\pi/2 \) condensers (coils 5 and 6). The heat transfer performance for these two condensers at a given fan power is astonishingly inferior to that of any of the other coils.

**Figure 5.41** \( h_A \) versus fan power for all eight condensers

Because it is advisable to limit the free-stream air velocities to 1 m/s (3.3 ft/s) in domestic refrigerators, these data should be examined more closely. Fig. 5.42 zooms in on these results from Fig. 5.41. The data is fit with a curve in an effort to show the relative performance of each condenser more clearly. Fig. 5.42 indicates that the best overall performing condensers are the larger amplitude coils with the \( \psi = 0 \) orientation. However, the smaller coils with this orientation as well as the larger amplitude \( \psi=\pi/2 \) condensers display heat transfer performance that is within 10\% of Coil 3, the best overall performer,
at an equal fan power of 3 W (10 Btu/hr). Coils 5 and 6, the smaller amplitude $\psi=\pi/2$ condensers, again are seen to be the worst options. Indeed, the $h_A$ for Coil 3 is 55\% higher than that for Coil 5 given a fan power of 3 W (10 Btu/hr).

![Figure 5.42 hA versus fan power for all eight coils ($V \leq 1.0 \text{ m/s}$)](image)

As the air flow approaches the condenser, the velocity of the air increases. An important value to introduce is the maximum velocity through the condenser, $V_{\text{max}}$, which is defined as the average velocity of the air passing through the minimum flow area. $V_{\text{max}}$ should make the data collapse to some degree as opposed to the simple free-stream air velocity. Based on continuity, a ratio of the air velocity through a face area (defined as the wire spacing multiplied by the tube spacing) to the air velocity through the minimum area can be established. This ratio is equal to the minimum area divided by the face area, as seen in Eq. (5.6). Note that this analysis assumes that the tubes and the wires lie in the same plane.
The inverse of this expression is defined as $V_{\text{ratio}}$:

$$\frac{V_{\text{max}}}{V_{\text{face}}} = \frac{A_{\text{min}}}{A_{\text{face}}} = \left\{ \frac{S_w S_t - D_w S_t - D_t (S_w - D_w)}{S_w S_t} \right\}$$

$$= \left\{ 1 - \frac{D_w}{S_w} - \frac{D_t (S_w - D_w)}{S_w S_t} \right\}$$

For the eight condensers being investigated in this study, the only varied parameter that affects the maximum velocity is the wire spacing. Obviously, the coils with the tighter wire spacing have a smaller minimum flow area and therefore a higher maximum velocity. Because this is the sole factor affecting the flow area in the wire-and-tube matrix for this study, there are only two different values of $V_{\text{ratio}}$ for these coils. Table 5.6 gives the $V_{\text{ratio}}$ for the eight condensers.

<table>
<thead>
<tr>
<th>Coils</th>
<th>Coil 1</th>
<th>Coil 2</th>
<th>Coil 3</th>
<th>Coil 4</th>
<th>Coil 5</th>
<th>Coil 6</th>
<th>Coil 7</th>
<th>Coil 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>$V_{\text{ratio}}$</td>
<td>1.85</td>
<td>1.70</td>
<td>1.85</td>
<td>1.70</td>
<td>1.85</td>
<td>1.70</td>
<td>1.85</td>
<td>1.70</td>
</tr>
</tbody>
</table>

Several quantities are plotted as a function of the maximum velocity in Appendix E - Additional Plots, including $h_w$, $h_A$, and $\Delta p$. The plots of $h_A$ and $\Delta p$ are seen in figs. E.27 and E.28, respectively. These data collapse very well compared to the same quantities plotted versus the free-stream velocity, $V$. This collapse of data is the objective of the maximum velocity, and it will be useful when the results are non-dimensionalized and correlated.

The maximum velocity is used in the definition of the Reynolds number for this report. The Reynolds number is a dimensionless number often used when correlating results; it is the ratio of the inertia forces to the viscous forces of the air flow. The definition to be used for this study is:
The dimensionless parameter that is used to quantify the heat transfer is the Nusselt number. Like the Reynolds number, the characteristic length used in this investigation is the wire diameter.

\[ \text{Re}_{\text{max}} = \frac{\rho_a V_{\text{max}} D_w}{\mu_a} \]  (5.8)

\[ \text{Nu}_w = \frac{h_w D_w}{k_a} \]  (5.9)

The Nusselt number is plotted as a function of the Reynolds number in Fig. 5.43. All eight condensers are included in these results.

The dimensionless numbers make the data collapse fairly well. As usual, coils 5 and 6 fall considerably below the trend followed by the rest of the condensers. If these two condensers are omitted and a power curve fit correlation is developed based on the remaining six coils, it takes the form \( \text{Nu}_w = C \text{Re}_{\text{max}}^n \), where \( C \) and \( n \) are constants. Eq. (5.10) defines these constants and gives the actual curve fit:

\[ \text{Nu}_w = 0.112 \text{Re}_{\text{max}}^{0.667} \]  (5.10)

**Figure 5.43** \( \text{Nu}_w \) vs. \( \text{Re}_{\text{max}} \) for all condensers (correlation excludes coils 5 and 6)
In some past investigations of wire-on-tube condensers, namely Hoke (1995), Swofford (1995) and Lum (1997), individual correlations for the two orientations ($\psi = 0$ and $\psi = \pi/2$) were developed. In this study, coils 7 and 8 have been included in the correlation with the $\psi = 0$ condensers because their results correlate with the data from coils 1-4.

It should be noted from Fig. 5.43 that the coils with the wider wire spacing tend to fall above the correlation, while those with the tighter wire spacing fall below. This type of behavior has been seen before; Hoke (1995) and Swofford (1995) observed the same trend when correlating their data from wire-on-tube condensers at various angles-of-attack. It may be advisable to include a dimensionless function of the wire spacing in the correlation, however only two wire spacings were investigated in this study. A broader range of wire spacings should be examined before this parameter is included in the correlation.

Fig. 5.44 shows the percent difference between the experimental data and the correlation for the Nusselt number. For the smaller Reynolds numbers, where there is both free and forced convection, the correlation is not as good. Even still, the average absolute difference including all the data is 8.7% with a standard deviation of 11.2%. For the data with $\text{Re}_{\text{max}} > 100$, the average absolute difference reduces to 6.4% while the standard deviation is 8.4%. One should note the way the data falls on this plot. There is a cluster

![Figure 5.44 Percent difference between Nu\textsubscript{w} data and Nu\textsubscript{w} correlation](image_url)
of data points at a certain $Re_{\text{max}}$ followed by another cluster at a slightly higher $Re_{\text{max}}$. The data at the higher Reynolds number corresponds to those condensers with a tighter wire spacing, and thus a higher maximum velocity. As was mentioned previously, the correlation over-predicts these data, whereas it under-predicts for the condensers with the wider wire spacing.

The pressure drop is a parameter that has been discussed often when comparing the performance of these saw-tooth condensers. Continuing the topic of dimensionless groups, the pressure drop can be expanded into such a quantity. The pressure coefficient, $C_p$, is similar to a coefficient of drag:

$$C_p = \frac{\Delta p}{\frac{1}{2} \rho_a V_{\text{max}}^2}$$

(5.11)

Fig. 5.45 is a plot of the pressure coefficient versus the Reynolds number for all eight condensers. The data collapse very well for these results. Coils 5 and 6 do not depart from the trend followed by the rest of the coils. This is due to the fact that the pressure coefficient is not related to heat transfer, rather it is based on the pressure drop.

**Figure 5.45** $C_p$ vs. $Re_{\text{max}}$ for all eight condensers
Since it has been shown that the pressure drop is comparable through all the condensers, it makes sense that the data collapses as it does. A correlation, using all eight condensers, has been developed for the pressure coefficient as a function of the Reynolds number. Like the previous correlations, it is a power curve fit and it takes the form

\[ C_p = C \, \text{Re}_{\text{max}}^n \]

The correlation is:

\[ C_p = 72.7 \, \text{Re}_{\text{max}}^{-0.603} \quad (5.12) \]

It is seen in Fig. 5.45 that the data associated with the condensers at \( \psi = \pi/2 \) fall along a tight line. The \( C_p \) for the smaller amplitude \( \psi = 0 \) coils (coils 1 and 2) fall above this curve; they are under-predicted by the correlation. Coils 3 and 4, the larger amplitude \( \psi = 0 \) condensers, have a \( C_p \) that is over-predicted by the correlation. The difference is due to the design of these saw-tooth condensers. The \( \psi = \pi/2 \) coils are such that the wires on one side of the tube serpentine are in tandem with the wires on the opposite side. For the \( \psi = 0 \) orientation, the way the condensers are bent means that these wires are not quite in tandem. This accounts for the variation in the pressure coefficient.

The percent difference between the experimental data and the correlation is presented in Fig. 5.46. For the reasons just discussed, and because the power curve does

![Figure 5.46](image-url)
not precisely fit the data, there is a larger average percent difference than was seen for the Nusselt-Reynolds relation. Nevertheless, the average absolute difference between the correlation and the experimental data is 11.7% with a standard deviation of 13.8%.

It should be mentioned that when Hoke (1995), Swofford (1995), and Lum (1997) correlated their data for wire-on-tube condensers subjected to forced convection, there was a large influence due to the angle-of-attack, $\alpha$. For this investigation, however, there is only one angle-of-attack ($\alpha = 60^\circ$). Thus, application of these correlations to saw-tooth condensers with other values of $\alpha$ should be done with caution.

### 5.7 Results from Coils 9 and 10

Towards the completion of this thesis, a new wire-on-tube condenser was tested. Like the eight coils that are discussed in the previous sections, this saw-tooth condenser was fabricated by Indiana Tube Corporation. This condenser was bent to form a set of five layers in parallel planes. It was tested at an angle-of-attack of $\alpha = 90^\circ$, with the air flow perpendicular to both the wires and the tubes, and at $\alpha = 0^\circ$, with the air flow perpendicular to the tubes and parallel to the wires. In order to maintain consistency with the convention used in the thesis, this condenser is termed Coil 9 for $\alpha = 90^\circ$ and Coil 10 for $\alpha = 0^\circ$. Table 5.7 gives the dimensions for coils 9 and 10, which can be compared with those of the other eight saw-tooth condensers in Appendix A - Coil Geometry. Note that since coils 9 and 10 are physically the same wire-on-tube condenser, only the amplitude and depth are different when the angle-of-attack is changed.

As can be seen in Table 5.7, the design of this saw-tooth condenser is much more compact than coils 1-8. It has five layers and a depth of just 132 mm (5.2 in.) when orientated at $\alpha = 90^\circ$ and 124 mm (4.9 in.) for $\alpha = 0^\circ$. The amplitude, or height, of coils 9 and 10 is too large to fit under the fresh food compartment in most current household refrigerators. Instead, compact condensers such as these could be placed at the rear of the refrigerator next to the compressor. This would allow for the fresh food compartment to be lowered, thereby increasing the interior volume of the refrigerator.

The experimental procedure and data reduction for coils 9 and 10 is the same as that for the other eight condensers. Coil 9 is modeled as a counterflow heat exchanger, while Coil 10 is analyzed as a shell-and-tube heat exchanger. Each of the coils was tested twice, with the data from each experiment being nearly identical. The important results for these two coils are now presented, and these data will be compared with those for coils 1-8.

Fig. 5.47 shows the convection coefficient, $h_w$, for coils 9 and 10. The heat transfer performance of Coil 9, with the air flow perpendicular to both the wires and the
Table 5.7 Coil dimensions for coils 9 and 10

<table>
<thead>
<tr>
<th>Variable</th>
<th>Units</th>
<th>Coil 9 (α = 90°)</th>
<th>Coil 10 (α = 0°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amp.</td>
<td>mm (in.)</td>
<td>124 (4.90)</td>
<td>132 (5.20)</td>
</tr>
<tr>
<td>α</td>
<td>deg</td>
<td>90</td>
<td>0</td>
</tr>
<tr>
<td>D_w</td>
<td>mm (in.)</td>
<td>1.45 (0.0570)</td>
<td>1.45 (0.0570)</td>
</tr>
<tr>
<td>S_w</td>
<td>mm</td>
<td>4.70 (0.185)</td>
<td>4.70 (0.185)</td>
</tr>
<tr>
<td>N_w</td>
<td></td>
<td>690</td>
<td>690</td>
</tr>
<tr>
<td>L_w</td>
<td>mm (in.)</td>
<td>102 (4.00)</td>
<td>102 (4.00)</td>
</tr>
<tr>
<td>D_i</td>
<td>mm (in.)</td>
<td>4.78 (0.188)</td>
<td>4.78 (0.188)</td>
</tr>
<tr>
<td>D_l,i</td>
<td>mm (in.)</td>
<td>3.30 (0.130)</td>
<td>3.30 (0.130)</td>
</tr>
<tr>
<td>S_t</td>
<td>mm</td>
<td>25.4 (1.00)</td>
<td>25.4 (1.00)</td>
</tr>
<tr>
<td>L_t</td>
<td>mm (in.)</td>
<td>437 (17.2)</td>
<td>437 (17.2)</td>
</tr>
<tr>
<td>N_t</td>
<td></td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>A_w</td>
<td>m² (in.²)</td>
<td>0.319 (494)</td>
<td>0.319 (494)</td>
</tr>
<tr>
<td>A_t</td>
<td>m² (in.²)</td>
<td>0.185 (286)</td>
<td>0.185 (286)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Dimensionless Variables</th>
</tr>
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<tr>
<td>S_t*</td>
</tr>
<tr>
<td>D_t*</td>
</tr>
<tr>
<td>S_w*</td>
</tr>
<tr>
<td>L_w*</td>
</tr>
<tr>
<td>L_t*</td>
</tr>
<tr>
<td>%(A_w/A_to)</td>
</tr>
</tbody>
</table>

tubes, is superior to that of Coil 10. For instance, at V = 1 m/s (3.3 ft/s), \( h_w \) is 75% higher for \( \alpha = 90^\circ \).

The magnitude of \( h_w \) for both coils 9 and 10 relative to the other eight saw-tooth condensers should be discussed. For Coil 9, \( h_w \) is at least 7% higher than for any of the other eight condensers that have a clearance of 0.25 in. (6.4 mm). The clearance for coils 9 and 10 was very close to this level. There are several reasons why Coil 9 shows improved heat transfer performance. Hoke, Clausing, and Swofford (1997) as well as Lum (1997) established that, for air flow perpendicular to the tubes, \( h_w \) increases as the angle-of-attack increases. Also, this condenser does not have the flattened out peaks that are characteristic of the other eight coils. These regions essentially act as condensers situated at \( \alpha = 0^\circ \); therefore, they reduce the overall performance of the coil.

For Coil 10, the heat transfer results are somewhat surprising considering that the air flows parallel to the wires. Whereas the air flows directly through the wire-and-tube matrix for Coil 9, Coil 10 has four large gaps (one between each layer) that are the size of the tube spacing through which the air can easily pass. However, as was pointed out
earlier, the depth of Coil 10 is just 124 mm (4.9 in.). Because the layers are so short, the boundary layer effect of the air is minimized. The boundary layer does not have time to grow appreciably, and the heat transfer is enhanced. The $h_w$ associated with Coil 10 is roughly the same as was seen for coils 5 and 6, the smaller amplitude $\psi = \pi/2$ condensers.

Fig. 5.48 is a plot of the air pressure drop through coils 9 and 10. Because Coil 10 possesses large gaps through which the air can flow with very little resistance, it comes as no surprise that the pressure drop is much lower for this condenser. Indeed, at $V = 2$ m/s (6.6 ft/s), $\Delta p$ is approximately three times higher for Coil 9. The difference in $\Delta p$ is not nearly as great at the lower air velocities, however. It should be mentioned that the pressure drop per layer associated with Coil 9 is comparable to the other $\psi = \pi/2$ saw-tooth condensers. This is due to the open tube bends and the fact that the wires are in tandem.

As was done for the other eight condensers, it is useful to look at $h_A$ and the fan power required for coils 9 and 10. In Fig. 5.49, these quantities are compared with those from coils 2 and 6, which are among the stronger and weaker performing saw-tooth condensers, respectively. Although $h_w$ for Coil 9 is greater than that for Coil 2 (and $h_w$ for Coil 10 is greater than that for Coil 6), the value of $h_A$ will be less because the total area of the condenser is considerably reduced and because a greater percentage of the total area is

Figure 5.47 $h_w$ versus air velocity for coils 9 and 10
Figure 5.48 Δp versus air velocity for coils 9 and 10

Figure 5.49 Overall performance comparison for coils 9 and 10 (Power < 3 W)
There are two main points worth noting from these results. First, the overall performance of Coil 9 is superior to that of Coil 10. This follows the trend that has been seen for the other saw-tooth condensers: the coils with the air flow perpendicular to the wires are the better overall performers. The other key finding from Fig. 5.49 is that the original saw-tooth design shows better performance than the smaller, more compact model. The plot shows that the performance of Coil 2 is markedly superior to Coil 9. Even the relatively poorly performing Coil 6 displays an hA that is similar to that of Coil 9 for a given fan power.

Further discussion is necessary to put these results into context. Coil 9 is the design that would potentially be used in household refrigerators, so it will be focused on here. Like coils 1-8, Coil 9 has a significant amount of free area due to the open tube bends and the clearance. However, the open tube bends are not as wide as for coils 5-8; they are only 0.5 in. (13 mm) wide on each side of the condenser. The percentage of free area to frontal area for Coil 9 is 15.5%, which is a smaller ratio than was seen for coils 1-8 for a 1/4 inch clearance. The performance of this coil should increase due to this smaller ratio. The reason that Coil 9 shows inferior overall performance is because of its relatively small size and because its wire area is only 63.3% of the total area. For coils 1-8 this ratio is 72.0 - 75.7%. Since the tube area relative to the total area is significantly higher for Coil 9, the overall performance suffers.

In conclusion, the more compact wire-on-tube condenser performs much better when orientated with the wires normal to the air flow. However, this design results in inferior overall performance as compared to the flatter, longer saw-tooth condensers that are the focus of this thesis. The advantage to incorporating this design is that, with proper modifications, the condenser could be situated in the machine compartment next to the compressor. This would allow for the fresh-food compartment to be lowered resulting in increased volume inside the refrigerator.

5.8 Comparison with Past Results

The only other experimental study of saw-tooth shaped wire-on-tube condensers was performed by Lum (1997). It is worthwhile to briefly present some of his main results and relate them to this investigation. Concentration will be on the data comparing condenser performance at the two orientations relative to the air flow. Lum's study involved four separate coils, each tested at $\psi = 0$ and $\psi = \pi/2$. Although experiments were
performed on coils with a range of angle-of-attacks, the data corresponding to $\alpha = 60^\circ$ will be focused on here.

To compare the heat transfer performance of the condensers, Lum used a definition of $h_w$ that is nearly identical to that presented in this thesis. Overall, the data show considerably higher values for $h_w$ than were found for the saw-tooth condensers that are the focus of this investigation. One reason for this is the fact that the coils that Lum tested were extremely confined (i.e. there was little or no clearance). The condensers were also confined on the sides such that there were no open tube bends through which the air could flow. Another explanation for the higher $h_w$ values is that the multi-layer condensers in Lum’s study were constructed from several small, flat coils. With this configuration, the poorly performing bend areas have been eliminated. Without these bends, the entire condenser is at the true angle-of-attack; hence, the heat transfer performance is not degraded.

In Fig. 5.50, the $h_w$ data from one of Lum’s saw-tooth condensers is compared with the data from coils 2 and 4. The air flow is perpendicular to the wires ($\psi = 0$) and $\alpha = 60^\circ$ for all three condensers. At $V = 2$ m/s (6.6 ft/s) the convection coefficient of

![Figure 5.50 Comparison of $h_w$ with Lum’s data ($\psi = 0$ coils)](image-url)

Figure 5.50 Comparison of $h_w$ with Lum’s data ($\psi = 0$ coils)
Lum's condenser is 30% higher than Coil 4 for the reasons discussed in the previous paragraph. Note the range of amplitudes of the three coils. Section 5.3 - Effect of Amplitude on Performance concluded that increasing the amplitude of the saw-tooth results in a higher $h_w$. At 152 mm (6.0 in.), the condenser in Lum's study is nearly twice as tall as Coil 4, and three times as tall as Coil 2. Because the free area due to clearance is eliminated, Lum's condenser can be considered the infinite amplitude limit.

The comparison of $h_w$ for coils 6 and 8 with one of Lum's $\psi = \pi/2$ saw-tooth condensers is presented in Fig. 5.51. The amplitude of the coil in Lum's study is 133 mm (5.25 in.). Although there are other factors involved, this plot illustrates the idea that as the saw-tooth amplitude is increased, so too is the heat transfer performance of the coil. Another reason that Lum's condenser shows higher values of $h_w$ is due to the fact that there is essentially no free area since the clearance and the open tube bends are eliminated. As a result, virtually the entire air flow is forced through the wire-and-tube matrix. Again, this condenser can be considered the infinite amplitude limit.

![Figure 5.51 Comparison of $h_w$ with Lum's data ($\psi = \pi/2$ coils)](image)

Of the two orientations, Lum found that the heat transfer performance of the saw-tooth condensers at $\psi = 0$ was slightly better. To show this, Fig. 5.52 is a non-
dimensional plot of $\text{Nu}_w$ versus $\text{Re}_{\text{max}}$. Because Lum’s definition of $V_{\text{max}}$ is different than the definition that was introduced in Section 5.6 - Other Results, his raw data was reduced so that the results would be consistent with those found in this thesis. From Fig. 5.52, it is clear that when the coils are positioned such that the air flow is perpendicular to the wires, there is enhanced heat transfer performance. This same trend is seen with the data from the saw-tooth condensers that are the focus of this investigation. There is a huge heat transfer advantage for the smaller amplitude condensers orientated at $\psi = 0$ and a lesser advantage for the larger amplitude coils. The relative heat transfer performance of the two orientations for the larger amplitude more closely resembles Lum’s results, as expected.

![Figure 5.52 Nu$_w$ - Re$_{\text{max}}$ plot for Lum’s data](image)

Although Lum found that the heat transfer performance is superior when the coils are orientated at $\psi = 0$, he advises against the use of these condensers because the air pressure drop is much higher than for $\psi = \pi/2$ coils. Fig. 5.53 compares the pressure drop per layer for each of the four condensers in the two orientations. When the four wire-on-tube condensers are situated at $\psi = 0$, the pressure drop per layer is between two and three times higher than if the coils were at $\psi = \pi/2$. If the air flow is perpendicular to the wires and the condenser is at an angle-of-attack, the wires are normally no longer in tandem,
causing an increase in $\Delta p$. By contrast, when the air flow is normal to the tubes, the wires are always in tandem.

Looking at Lum's pressure drop data in Fig. 5.53, the $\Delta p$ per layer for the $\psi = \pi/2$ condensers closely resembles the $\Delta p$ per layer for all eight saw-tooth condensers that are the focus of this investigation (see Fig. 5.40). The $\Delta p$ per layer associated with Lum's $\psi = 0$ coils are much higher not only because their wires are staggered but also because of their tightly confined configuration. It was shown in Section 5.4 - Effect of Clearance on Performance that when the clearance was reduced, both $h_w$ and $\Delta p$ increased markedly. A measured clearance level is used in this investigation in an effort to simulate the configuration that would be found in an actual refrigerator. By using the consistent clearance, the pressure drop per layer associated with each condenser is comparable.

![Figure 5.53 $\Delta p$ per layer for Lum's data](image)

Lum performed some tests on saw-tooth condensers with an angle-of-attack of $\alpha = 90^\circ$. Fig. 5.54 compares the $h_w$ data from one of Lum's $\alpha = 90^\circ$ coils with those from Coil 9. For the lower air velocities, the heat transfer performance is quite similar for the two condensers. The condenser tested by Lum does show superior performance for velocities greater than 0.5 m/s (1.6 ft/s). There are several reasons for the difference in $h_w$ values. First, Lum's coil was more tightly confined than Coil 9. Also, Coil 9 has small
Fig. 5.54 Comparison of $h_w$ for Coil 9 and one of Lum's $\alpha = 90^\circ$ coils

open tube bend areas whereas the condensers tested by Lum did not have these ineffective regions.

It is very useful to compare the data that come from the experiments performed by Lum with the data from the current investigation However, the experiments dealt with idealized conditions that are improbable to be found in household refrigerators. Saw-tooth condensers will have tube bends that degrade their performance. Also, clearance is necessary to minimize vibration and ease the fabrication process.
6. CONCLUSIONS

Based on the results just presented, the following conclusions can be drawn.
They are applicable for confined, saw-tooth shaped wire-on-tube condensers with wire and tube geometries similar to those investigated. The coils were subjected to forced convection air flows with $\text{Re}_{\text{max}} \leq 370$. Any extrapolation from these limits may invalidate the conclusions.

1) For the condensers with the $\psi = 0$ orientation, amplitude has little effect on the overall performance. The $h_w$ associated with the larger amplitude coils is slightly higher, but the required fan power is also higher.

2) Amplitude has a significant effect on performance for the saw-tooth condensers orientated at $\psi = \pi/2$ radians. The larger amplitude tested has at least a 50% greater $h_w$. The pressure drop through the two amplitudes at this orientation is comparable.

3) Of the two wire spacings tested, the $h_w$ associated with the wider wire spacing is slightly higher. However, because the condensers with the tighter wire spacing have a higher number of wires (and hence a greater area), their total heat transfer performance is slightly better. In addition, there are trends based on the wire spacing with regard to the correlations developed for these coils. It is advised that more extensive research be performed to study the effects of wire spacing and perhaps include a function of the wire spacing in the correlations.

4) The clearance, or degree to which a condenser is confined, has been shown to have a significant effect on performance. Reducing the clearance from 1/4 inch (6.4 mm) to 1/8 inch (3.2 mm) results in much improved heat transfer but also increased pressure drop. However, the increase in pressure drop for the low air velocities, which are of interest for household refrigerators, is not appreciable. Therefore, a clearance of more than 1/8 inch is not advised.

5) Eq. (5.10) correlates well the $\text{Nu}_w$ data obtained from all of the coils with the exception of the smaller amplitude $\psi = \pi/2$ condensers (coils 5 and 6). The average absolute difference between the experimental data and the correlation is 8.7%. When looking only at the data with $\text{Re}_{\text{max}} > 100$, the average absolute difference is reduced to 6.4%.

6) The pressure coefficient associated with all eight condensers can be adequately correlated using Eq. (5.12). The average absolute difference between the experimental $C_p$ and those predicted by the correlation is 11.7%.
REFERENCES


92
REFERENCES (CONT.)


APPENDIX A. COIL GEOMETRY

### Table A.1a Metric Coil Dimensions, Coils 1-4 (ψ = 0)

<table>
<thead>
<tr>
<th>Variable</th>
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<th>Coil 3</th>
<th>Coil 4</th>
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<td>Amp.</td>
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<td>56.4</td>
<td>81.3</td>
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<td>α</td>
<td>deg</td>
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<td>60</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>D_w</td>
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<td>1.59</td>
<td>1.59</td>
<td>1.59</td>
</tr>
<tr>
<td>S_w</td>
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<td>5.77</td>
<td>4.78</td>
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<td>711</td>
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<tr>
<td>D_t</td>
<td>mm</td>
<td>4.78</td>
<td>4.78</td>
<td>4.78</td>
<td>4.78</td>
</tr>
<tr>
<td>D_t,i</td>
<td>mm</td>
<td>3.30</td>
<td>3.30</td>
<td>3.30</td>
<td>3.30</td>
</tr>
<tr>
<td>S_t</td>
<td>mm</td>
<td>25.4</td>
<td>25.4</td>
<td>25.4</td>
<td>25.4</td>
</tr>
<tr>
<td>L_t</td>
<td>mm</td>
<td>594</td>
<td>594</td>
<td>594</td>
<td>594</td>
</tr>
<tr>
<td>N_t</td>
<td>-</td>
<td>28</td>
<td>28</td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>A_w</td>
<td>m²</td>
<td>0.781</td>
<td>0.646</td>
<td>0.781</td>
<td>0.646</td>
</tr>
<tr>
<td>A_t</td>
<td>m²</td>
<td>0.251</td>
<td>0.251</td>
<td>0.251</td>
<td>0.251</td>
</tr>
</tbody>
</table>

### Dimensionless Variables

<table>
<thead>
<tr>
<th>Variable</th>
<th>Units</th>
<th>Coil 1</th>
<th>Coil 2</th>
<th>Coil 3</th>
<th>Coil 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>S_t*</td>
<td>-</td>
<td>16.0</td>
<td>16.0</td>
<td>16.0</td>
<td>16.0</td>
</tr>
<tr>
<td>D_t*</td>
<td>-</td>
<td>3.01</td>
<td>3.01</td>
<td>3.01</td>
<td>3.01</td>
</tr>
<tr>
<td>S_w*</td>
<td>-</td>
<td>3.01</td>
<td>3.63</td>
<td>3.01</td>
<td>3.63</td>
</tr>
<tr>
<td>L_w*</td>
<td>-</td>
<td>447</td>
<td>447</td>
<td>447</td>
<td>447</td>
</tr>
<tr>
<td>L_t*</td>
<td>-</td>
<td>374</td>
<td>374</td>
<td>374</td>
<td>374</td>
</tr>
<tr>
<td>%(A_w/A_tot)</td>
<td>-</td>
<td>75.7</td>
<td>72.0</td>
<td>75.7</td>
<td>72.0</td>
</tr>
</tbody>
</table>

### Table A.1b English Coil Dimensions, Coils 1-4 (ψ = 0)

<table>
<thead>
<tr>
<th>Variable</th>
<th>Units</th>
<th>Coil 1</th>
<th>Coil 2</th>
<th>Coil 3</th>
<th>Coil 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amp.</td>
<td>in.</td>
<td>2.22</td>
<td>2.22</td>
<td>3.20</td>
<td>3.10</td>
</tr>
<tr>
<td>α</td>
<td>deg</td>
<td>60</td>
<td>60</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>D_w</td>
<td>in.</td>
<td>0.0626</td>
<td>0.0626</td>
<td>0.0626</td>
<td>0.0626</td>
</tr>
<tr>
<td>S_w</td>
<td>in.</td>
<td>0.188</td>
<td>0.227</td>
<td>0.188</td>
<td>0.227</td>
</tr>
<tr>
<td>N_w</td>
<td>-</td>
<td>220</td>
<td>182</td>
<td>220</td>
<td>182</td>
</tr>
<tr>
<td>L_w</td>
<td>in.</td>
<td>28.0</td>
<td>28.0</td>
<td>28.0</td>
<td>28.0</td>
</tr>
<tr>
<td>D_t</td>
<td>in.</td>
<td>0.188</td>
<td>0.188</td>
<td>0.188</td>
<td>0.188</td>
</tr>
<tr>
<td>D_t,i</td>
<td>in.</td>
<td>0.130</td>
<td>0.130</td>
<td>0.130</td>
<td>0.130</td>
</tr>
<tr>
<td>S_t</td>
<td>in.</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>L_t</td>
<td>in.</td>
<td>23.4</td>
<td>23.4</td>
<td>23.4</td>
<td>23.4</td>
</tr>
<tr>
<td>N_t</td>
<td>-</td>
<td>28</td>
<td>28</td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>A_w</td>
<td>in.²</td>
<td>1210</td>
<td>1210</td>
<td>1210</td>
<td>1210</td>
</tr>
<tr>
<td>A_t</td>
<td>in.²</td>
<td>389</td>
<td>389</td>
<td>389</td>
<td>389</td>
</tr>
</tbody>
</table>
### Table A.2a Metric Coil Dimensions, Coils 5-8 ($\psi = \pi/2$)

<table>
<thead>
<tr>
<th>Variable</th>
<th>Units</th>
<th>Coil 5</th>
<th>Coil 6</th>
<th>Coil 7</th>
<th>Coil 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amp.</td>
<td>mm</td>
<td>65.3</td>
<td>64.3</td>
<td>85.6</td>
<td>85.9</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>deg</td>
<td>60</td>
<td>60</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>$D_w$</td>
<td>mm</td>
<td>1.59</td>
<td>1.59</td>
<td>1.59</td>
<td>1.59</td>
</tr>
<tr>
<td>$S_w$</td>
<td>mm</td>
<td>4.78</td>
<td>5.77</td>
<td>4.78</td>
<td>5.77</td>
</tr>
<tr>
<td>$N_w$</td>
<td>-</td>
<td>220</td>
<td>182</td>
<td>220</td>
<td>182</td>
</tr>
<tr>
<td>$L_w$</td>
<td>mm</td>
<td>711</td>
<td>711</td>
<td>711</td>
<td>711</td>
</tr>
<tr>
<td>$D_t$</td>
<td>mm</td>
<td>4.78</td>
<td>4.78</td>
<td>4.78</td>
<td>4.78</td>
</tr>
<tr>
<td>$D_{tt}$</td>
<td>mm</td>
<td>3.30</td>
<td>3.30</td>
<td>3.30</td>
<td>3.30</td>
</tr>
<tr>
<td>$S_t$</td>
<td>mm</td>
<td>25.4</td>
<td>25.4</td>
<td>25.4</td>
<td>25.4</td>
</tr>
<tr>
<td>$L_t$</td>
<td>mm</td>
<td>594</td>
<td>594</td>
<td>594</td>
<td>594</td>
</tr>
<tr>
<td>$N_t$</td>
<td>-</td>
<td>28</td>
<td>28</td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>$A_w$</td>
<td>m$^2$</td>
<td>0.781</td>
<td>0.646</td>
<td>0.781</td>
<td>0.646</td>
</tr>
<tr>
<td>$A_t$</td>
<td>m$^2$</td>
<td>0.251</td>
<td>0.251</td>
<td>0.251</td>
<td>0.251</td>
</tr>
</tbody>
</table>

**Dimensionless Variables**

| $S_t^*$  | -     | 16.0  | 16.0  | 16.0  | 16.0  |
| $D_t^*$  | -     | 3.01  | 3.01  | 3.01  | 3.01  |
| $S_w^*$  | -     | 3.01  | 3.63  | 3.01  | 3.63  |
| $L_t^*$  | -     | 447   | 447   | 447   | 447   |
| $L_q^*$  | -     | 374   | 374   | 374   | 374   |
| $%(A_w/A_{tot})$ | -   | 75.7  | 72.0  | 75.7  | 72.0  |

### Table A.2b English Coil Dimensions, Coils 5-8 ($\psi = \pi/2$)

<table>
<thead>
<tr>
<th>Variable</th>
<th>Units</th>
<th>Coil 5</th>
<th>Coil 6</th>
<th>Coil 7</th>
<th>Coil 8</th>
</tr>
</thead>
<tbody>
<tr>
<td>Amp.</td>
<td>in.</td>
<td>2.57</td>
<td>2.53</td>
<td>3.37</td>
<td>3.38</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>deg</td>
<td>60</td>
<td>60</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>$D_w$</td>
<td>in.</td>
<td>0.0626</td>
<td>0.0626</td>
<td>0.0626</td>
<td>0.0626</td>
</tr>
<tr>
<td>$S_w$</td>
<td>in.</td>
<td>0.188</td>
<td>0.227</td>
<td>0.188</td>
<td>0.227</td>
</tr>
<tr>
<td>$N_w$</td>
<td>-</td>
<td>220</td>
<td>182</td>
<td>220</td>
<td>182</td>
</tr>
<tr>
<td>$L_w$</td>
<td>in.</td>
<td>28.0</td>
<td>28.0</td>
<td>28.0</td>
<td>28.0</td>
</tr>
<tr>
<td>$D_t$</td>
<td>in.</td>
<td>0.188</td>
<td>0.188</td>
<td>0.188</td>
<td>0.188</td>
</tr>
<tr>
<td>$D_{tt}$</td>
<td>in.</td>
<td>0.130</td>
<td>0.130</td>
<td>0.130</td>
<td>0.130</td>
</tr>
<tr>
<td>$S_t$</td>
<td>in.</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
<td>1.00</td>
</tr>
<tr>
<td>$L_t$</td>
<td>in.</td>
<td>23.4</td>
<td>23.4</td>
<td>23.4</td>
<td>23.4</td>
</tr>
<tr>
<td>$N_t$</td>
<td>-</td>
<td>28</td>
<td>28</td>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>$A_w$</td>
<td>in.$^2$</td>
<td>1210</td>
<td>1210</td>
<td>1210</td>
<td>1210</td>
</tr>
<tr>
<td>$A_t$</td>
<td>in.$^2$</td>
<td>389</td>
<td>389</td>
<td>389</td>
<td>389</td>
</tr>
</tbody>
</table>
APPENDIX B. DATA REDUCTION PROGRAM

C**PROGRAM DataReduction--Data reduction program for wire-on-tube condensers
C Programmed by A.M.Clausing; Version: 19 March 1998
C
C Sw= wire spacing, m
C St= tube spacing, m
C Dwbare= bare wire diameter, m
C Dw= effective wire diameter = (wire dia. + 2*paint thichness), m
C Dbare= bare tube diameter, m
C Dt= effective tube outer diameter = (tube dia. + 2*paint thichness), m
C depth= coil depth (center of top wire to center of bottom wire)
C Nt= number of tube passes
C Nw= number of wires
C XLw= length of wire, m
C XF= length of tube (bend centerline to bend centerline, i.e., 1 tube pass), m
C XKp= thermal conductivity of paint, W/m-K
C DELT Ap= thickness of paint, m
C IHXtype=Type of HX: 1 -- Counterflow, 2 --Parallelflow; 3 Shell & Tube
C with an even number of tube passes
C No. of condenser layers (longitudinal layers -- complete air flows thru all layers)
Implicit none
INTEGER II(0:30),ICoil(499),Nw,Nt,NL,IX,IN,K,NITER,Ncase,IER,
INTEGER IUout,Npts(30)
REAL XMdot(499),V(499),Tai(499),Tri(499),Tro(499),DP(499),Cr(499),Ca(499),
Dtot(499),Q(499),Tao(499),HXe(499),XLMTD(499),Qrad(499),hr(499),
Eta(499),hw(499),XNTU(499),Rcap(499),Re(499),XNu(499),Vmax(499),
Cd(499),HA(499),P(499)
REAL Dw,Sw,XLw,Aw,Dt,Dti,St,XLt,At,Alpha,DELT Ap,XKp,EMISS,Tau,XKs,
Ad,FanEff,XMbase,Xm,Ate,Vratio,VFws,VFts,Ad,ROH,XMU,XK,CP,GPB,PR,
Cmin,Tam,Qmax,E1,E2,DTa,DTb,Tm,Rint,UA,RSurf,Hold,DeltaH,DeltaQrad,
Tf,QradOld,AwP,AtP,hr,Cae
LOGICAL SI,DEGC
Character CPsi*5
C**Definition of NAMELIST variables called NAM
NAMELIST /NAM/Dw,Sw,XLw,Aw,Dt,Dti,St,XLt,At,Alpha,DELT Ap,XKp,EMISS,Tau,XKs,
Ad,FanEff,XMbase,Xm,Ate,Vratio,VFws,VFts,Ad,ROH,XMU,XK,CP,GPB,PR,
Cmin,Tam,Qmax,E1,E2,DTa,DTb,Tm,Rint,UA,RSurf,Hold,DeltaH,DeltaQrad,
Tf,QradOld,AwP,AtP,hr,Cae
Character CPsi*5
C**Determine direction of output; read test points; print date and program information
CALL DATAIN(Ncase,Npts,IUout,ICoil,XMdot,V,Tai,Tro,DP,II)
C
C** Beginning of Case (Outer) Loop
DO 3 J=1,Ncase
C**Read coil parameters.
READ(7,NAM)
**Calculate various constants -- Constants during testing of respective coil.**

\[ \text{XMbase} = \text{St} \times \sqrt{\text{I.1} \times (\text{XKs} \times \text{Dw})} \]
\[ \text{Ati} = \text{At} \times \text{Dti} / \text{Dt} \]
\[ \text{At} = \text{At} \times (\text{Dw} / \text{Dt})^{0.5} \]
\[ \text{Vratio} = 1.1 \times (1 - \text{Dw} / \text{Sw} + (\text{Dt} \times (\text{Sw} - \text{Dw}) / \text{Sw} / \text{St})) \]

**Determine View Factors**

Call ViewFactor(NL, Alpha, Tau, Dw, Sw, Dt, St, VFws, VFts)

Write(*, *)VFws, VFts

**Write coil parameters and coil constants**

IF(Aw.EQ.0.)THEN

\[ \text{Aw} = \text{Nw} \times \pi \times \text{Dw} \times \text{XLw} \]

WRITE(IUout,102) ICoil(II(J)), Nw, XLw

102 FORMAT(// 'Coil Number:',I3//' No. of Wires=',I4,' Wire Length [m]=',FS.31)

ELSE

WRITE(IUout,103) ICoil(II(J))

103 FORMAT(//'Coil Number:',131)

END IF

WRITE(IUout,104) Dw*1000, Sw*1000, Aw, Dt*1000, St*1000, At, EMISS, Tau,
2 Ad, Dt*1000, XKs, IHXtype, Aw/(Aw+At), (Sw/Dw-1), FanEff*100,
3 Alpha, Cpsi, Vratio, NL

104 FORMAT(// 'Wire Dia.[mm]=',F5.2,T29,'Wire Pitch [mm]=',F5.2,T58,
2 'Wire Area [m^2]=',F6.3//' Tube Dia.[mm]=',F5.2,T29,'Tube Pitch [mm]=',
3 F5.1,T58,'Tube Area [m^2]=',F6.3//' Coe Emissivity=',F5.2,T29,
4 'Tau=',F4.1,T29,'Duct Area [m^2]=',F7.4//' Tube ID [mm]=',F5.2,
5 T29,'k(steel) [W/m-K]=',F5.1,T58,'HX Type=',I2//' Aw/Atot=',F6.3,
6 T29,'(Sw/Dw)-1=',F6.3,T58,'Fan/Duct % Eff.=',F5.1,
7 'Alpha [deg.]=',F4.0,T29,'Flow perpend. to ',A,T29,'Vmax/V=',FS.2,
8 'Number of Coil Layers =',I3//)

**If IHXType.GT.10, calculate effective duct area, Ade**

IF(IHXType.GT.10) THEN

IHXtype=IHXType-10

Ade=(NL*Ad)/(NL-1)

Write(IUout,105) Ade

105 FORMAT(//' Ade [m^2]=','F7.4/)

ELSE

Ade=Ad

ENDIF

**Calculate view factor weighted wire and tube areas**

AwP=Aw*VFws

AtP=At*VFts

C

**Beginning of inner loop to process test points for respective coil**

DO 5 I=II(J-1)+1,II(J)

CALL GASPT(1,Tai(I),RHO,XMU,XK,CP,GPB,PR,IER)

Ca(I)=RHO*CP*V(I)*Ad

Cae=Ca(I)*Ade/Ad

Cr(I)=XMdot(I)*4180

IF(Cae.EQ.0=")THEN

Rcap(I)=Cae/Cr(I)
Cmin = Ca

ELSE

Rcap(I) = Cr(I)/Cae
Cmin = Cr(I)

ENDIF

C** Calculate q(total), qrad, Ta, out, and Ta, mean
Qtot(I) = Cr(I)*(Tri(l) - Tro(l))

C** Beginning of 4 iterations to calculate q(rad) which is dependent on eta
Qrad(I) = 0

DO 6 NITER = 1, 4
QradOLD = Qrad(I)
Call Radiation(Qtot(I), Tai(I), Tri(l), Tro(I), Aw, AwP, AtP, Emiss, Ca(I),
2 Qrad(I), NITER, Eta(I), hwR)
DeltaQrad = ABS(QradOLD - Qrad(I))

Q(I) = Qtot(I) - Qrad(I)
Tao(I) = Tai(I) + Q(I)/Cae
Tam = (Tao(I) + Tai(I))/2.

C** Calculate heat exchanger effectiveness, HXe, NTU, and LMTD
Qmax = Cmin*(Tri(l) - Tai(I))
HXe(I) = Q(I)/Qmax

C** Calculate the size of the heat exchanger: NTU
IF (IHXtype.EQ.1) THEN

C** Countercflow

XNTU(_I) = LOG((HXe(I) - 1.)/(HXe(I)*Rcap(I) - 1.))/(Rcap(I) - 1.)

END IF
IF (IHXtype.EQ.2) THEN

C** Parallel Flow

XNTU(_I) = -LOG(1. - HXe(I)*(1. + Rcap(I)))/(1. + Rcap(I))

END IF
IF (IHXtype.EQ.3) THEN

C** Shell and Tube (2, 4, ... tube Passes)

E2 = (1. + Rcap(I)**)0.5
E1 = (2./HXe(I)*(1. + Rcap(I)))/E2

XNTU(_I) = -LOG((E1 - 1.)/(E1 + 1.))/E2

END IF
DTa = Tri(l) - Tao(I)
DTb = Tro(I) - Tai(I)

XLMTD(I) = (DTa - DTb)/LOG(DTa/DTb)

C** Calculate internal, pipe, and paint resistances
Tm = (Tri(I) + Tro(I))/2.
CALL INTERNAL (Tm, Dit, St, Ati, XMdot(I), hr(I), Rint)

C** Solve for hw using Successive Substitutions
UA = Cmin*XNTU(I)
Rsurf = 1./UA - Rint
Eta(I) = 1.
Hold = .0

DO 7 K = 1, 10

hw(I) = 1./(Rsurf*(AtE(I)*Aw))
XM=XmBase*SQRT(hw(I)+hwr)
Eta(I)=TANH(XM)/XM
DeltaH=ABS(Hold-hw(I))
Hold=hw(I)

7 CONTINUE
IF(DeltaH.GT.0.01) WRITE(6,106) DeltaH,1
106 FORMAT(//** Delta hw (\textasciitilde F5.3, \textasciitilde ) is > 0.01 for I \textasciitilde =\textasciitilde ,13, ***)
C**End of loop to determine q(rad) iteratively since eta is unknown
6 Continue
IF(DeltaQrad.GT.0.1) WRITE(6,107) DeltaQrad,1
107 FORMAT(//** Delta Qrad (\textasciitilde F5.1, \textasciitilde ) is > 0.1 for I \textasciitilde =\textasciitilde ,13, ***)
C**Calculation of hA, Vmax, Nu, Re, pumping power, P, and Pressure coeff., Cd
HA(I)=hw(I)*(At*(Dw/Dk)**.5+Eta(I)*Aw)
TF=(Tai(I)+Tao(I)+Tri(I)+Tro(I))/4.
CALL GASPT(1,TF,RHO,XMU,XK,CP,GPB,PR,IER)
XNu(I)=hw(I)*Dw/XK
Vmax(I)=Vratio*V(I)*Tamffai(I)
Re(I)=RHO*Vmax(I)*Dw/XMu
Cd(I)=(2*DP(I))/(RHO*(Tam/Hf)*Vmax(I)**2)
P(I)=DP(I)*Ad*V(I)/FanEff
5 CONTINUE
C
C**End of Loop for respective coil; Write key results for this coil.
WRITE(IUout,108)
108 FORMAT(5x,'V',5x,'Cr',5x,'Ca',5x,'q',5x,'NTU',3x,'HX e',
2 3x,'Eta',5x,'h',5x,'HA',6x,'Nu',5x,'Re',4x,'Cd')
Do 9 I=II(J-1)+1, II(J)
WRITE(IUout,110) V(I),Cr(I),Ca(I),Q(I),XNTU(I),HXe(I),Eta(I),
2 hw(I),HA(I),XNu(I),Re(I),Cd(I)
110 FORMAT(F7.2,2F7.1,F7.0,F6.2,2F7.3,F7.1,F7.2,F7.2,F7.0,F6.2)
9 CONTINUE
3 CONTINUE
C
C**End of the processing of all test points. Write result files
CALL DataOut(Ncase,Npts,Ncoil,XMdot,V,Tai,Trid,Tro,Dp,Dr,Ca,
2 Q,Tao,Hxe,Rcap,XNTU,Eta,hw,HA,XNu,Re,Cd,P,XLMTD,hr,Qrad,NL)
C
Rwall=LOG(DT/DTi)/(2.*PI*XLT*XKS)
C
Rpaint=DELTAP/(PI*(DT+DELTAP)*XLT*XKP)
C
Rwallt=Rwall+Rpaint
WRITE(IUout,199)
199 FORMAT(// ALL DATA HA BEEN PROCESSED)
IF(IUout.EQ.8) CLOSE(8)
STOP
END

C***********************************************************************
C
Subroutine DataIn(Ncase,Npts,IUout,I Coil,XMdot,V,Tai,Trid,Tro,Dp,II)
imPLICIT NONE
INTEGER Ncase,Npts(30),II(0:30),I Coil(499),IUOUT,II,NptsT
REAL XMdot(499),V(499),Tai(499),Trid(499),Tro(499),Dp(499)
LOGICAL SI,DEGC
CHARACTER FNAME*60,CDATE*9,CTIME*8,LlNE1*5,LlNE2*5
C**Open Files; Write Program Description and Date; Read Test Data
WRITE(6,100)
100 FORMAT(/' TYPE NAME OF TEST DATA FILE/'?')
READ(*,'(A)') FNAME
C**Open input and output files
OPEN(7,FILE=FNAME)
REWIND 7
OPEN(10,FILE='Results-'/IFNAME)
OPEN(11,FILE='ResultsW-'/IFNAME,CARRIAGECONTROL='FORTRAN')
WRITE(6,104) 'Results-'/IFNAME,'ResultsW-'/IFNAME
104 FORMAT(I' COMMA DELIMITED RESULTS FILES ARE:'1T38,A,1T30,'AND'1T38,A/)
WRITE(6,106)
106 FORMAT(I'T10,'DIRECT OUTPUT TO:'/1T20,'SCREEN:',T36,'Type 6'
2 '/T20,'OUTPUT FILE:',T36,'Type 8'/T20,'PRINTER:',T36,'Type 9'/'?')
READ(*,*) IUout
IF(IUout.EQ.8) THEN
OPEN(8,FILE='Answers-'/IFNAME)
WRITE(6,108) 'Answers-'/IFNAME
108 FORMAT(I' OUTPUT WILL BE WRITTEN IN FILE: ',A)
ENDIF
C**Write Program Description, Date and Time
CALL DATE(CDATE)
CALL TIME(CTIME)
WRITE(IUout,110)CDATE,CTIME
110 FORMAT(7X,'Program: Data Reduction',T63,'Date: ',A1017X,
2 'Version: 19 March 1998',T63,'Time: 'A91'
Programmed by: A. M. Clausing')
C**Read all test points; the first two lines in the data file are discarded
READ(7,111)Line1,Line2
111 FORMAT(A/A)
DO 3 1=1,499
   READ(7,*,END=99) ICoil(I),XMdot(I),V(I),Tai(I),Tri(I),Tro(I),DP(I)
   NptsT=I
3 CONTINUE
99 CLOSE(7)
WRITE(6,112)
112 FORMAT(/' TYPE NAME OF COIL PARAMETER FILE/'?')
READ(*,'(A)') FNAME
C**Open input and output files
OPEN(7,FILE=FNAME)
REWIND 7
C**Determine number of cases, Ncase, the index of last record in each case,
C II(Ncase), and the number of points in each case, Npts(Ncase).
   Ncase=1
   II(0)=0
   DO 5 l=1,NptsT-1
      IF(ICoil(l).NE.ICoil(l+1)) THEN
         II(Ncase)=I
5 CONTINUE
ENDIF

CONTINUE

II(Ncase)=NptsT

Npts(Ncase)=II(Ncase)-II(Ncase-1)

RETURN

END

C**********************************************************************
C
Subroutine DataOut(Ncase,Npts,II,ICoil,XMdot,V,Vmax,Tai,Tri,Tro,DP,Cr,CA,
2Q,Tao,HXe,Rcap,XNTU,Eta,hw,HA,XNu,Re,Cd,P,XLMTD,hr,Qrad,NL)
implicit none
INTEGER Ncase,Npts(30),II(0:30),ICoil(499),Ncol,NWcol,JS,J,I,NL
REAL XMdot(499),V(499),Tai(499),Tri(499),Tro(499),XLMTD(499),
2DP(499),Cr(499),Ca(499),Q(499),Tao(499),HXe(499),hr(499),
3Eta(499),hw(499),XNTU(499),Rcap(499),Re(499),XNu(499),
4Vmax(499),Cd(499),HA(499),P(499),A(30,270),Qrad(499)
CHARACTER Title(270)*5,NCoil*3
Parameter(NWcol=10)
C**Create Titles and Write wide plotting file (one case per column)
Ncol=Ncase*NWcol
Do 3 J=1,Ncase
JS=NWcol*(J-1)+1
DO 5 I=1,Npts(J)
   A(IJS)=V(II(J-1)+I)
   A(IJS+1)=Vmax(II(J-1)+I)
   A(IJS+2)=hw(II(J-1)+I)
   A(IJS+3)=DP(II(J-1)+I)
   A(IJS+4)=Re(II(J-1)+I)
   A(IJS+5)=Xnu(II(J-1)+I)
   A(IJS+6)=Cd(II(J-1)+I)
   A(IJS+7)=HA(II(J-1)+I)
   A(IJS+8)=P(II(J-1)+I)
   A(IJS+9)=DP(II(J-1)+I)/NL
5 CONTINUE
C**Convert coil number,II(J)), to a Character variable, NCoil*3
Call CharAMC(II(J)),NCoil)
Title(JS)=V '/NCoil
Title(JS+1)=Vm'/NCoil
Title(JS+2)=hw'/NCoil
Title(JS+3)=DP'/NCoil
Title(JS+4)=Re'/NCoil
Title(JS+5)=Nu'/NCoil
Title(JS+6)=Cp'/NCoil
Title(JS+7)=HA'/NCoil
Title(JS+8)=P '/NCoil
Title(JS+9)=dp'/NCoil
3 CONTINUE
Write(11,101) (Title(J), J=1,Ncol)
101 FORMAT(270(A5,','))
DO 9 I=1,Npts(I)
   Write(11,104) (A(I,J), J=1,Ncol)
   FORMAT(270(A5,','))
9 CONTINUE
C**Create Titles and Write long plotting file
Write(10,112)
112 FORMAT(270(A5,','))
DO 11 I=1,II(NCASE)
   Write(10,114) ICoil(l), V(I), Tai(I), Tri(I), Tro(I), Cr(I), Ca(I), Q(I),
      Tau(I), HXe(I), Rcap(I), XNTU(I), Eta(I), hw(I), HA(I), Nu(I), Re(I),
      DP(I), Cd(I), P(I), XLMTD(I), hr(I), Qrad(I), DP(I)/NL)
   CONTINUE
11 CLOSE(10)
CLOSE(11)
RETURN
END

C**********************************************************
subroutine viewfactor(NL,Alpha,Tau,Dw,Sw,Dt,St,VFws,VFs)
c
This subroutine calculates the view factors between elements of
c individual multi-layer condenser layers. The wires and tube passes are
c each assumed to be uniform in temperature along each layer.
c
implicit none
INTEGER NL
REAL Alpha, Tau, Dw, Sw, Dt, St
REAL fww, fws1, fwt, fws, ft, fts1, ftw, fts, fs1w, fs1t,VFws,VFs
c
c fww -- view factor between consecutive wires within the same layer
c fws1 -- view factor from the condenser wires to an imaginary surface
c fwt -- view factor from the condenser wires to the tube passes of
c the same layer
c fws -- view factor from the condenser wires to the surroundings
c ft -- view factor between consecutive tube passes within the same
c layer
c fts1 -- view factor from the condenser tube passes to an imaginary
c surface
c ftw -- view factor from the condenser tube passes to the wires of
c the same layer
c fts -- view factor from the condenser tube passes to the
-- view factor from an imaginary surface to the condenser wires
-- view factor from an imaginary surface to the condenser tube passes
-- effective transmissivity through a condenser layer

*** Calculate the view factors between consecutive wires and consecutive tube passes within the same layer

\[ f_{ww} = \left(1 - 2 \cdot f_{ww}\right) / 2. \]
\[ f_{tt} = \left(1 - 2 \cdot f_{tt}\right) / 2. \]

*** Calculate the view factors between the wires & an imaginary surface and the tube passes & the same surface. Take the surface to be a plane which separates the wires from the tube passes.

\[ f_{wsl} = \left(1 - 2 \cdot f_{wsl}\right) / 2. \]
\[ f_{slw} = f_{wsl} \cdot \left(\pi \cdot DW\right) / SW \]

*** Calculate the view factors between the wires and the tube passes of the same layer

\[ f_{wlt} = f_{wsl} \cdot f_{slt} \]
\[ f_{tsw} = 2 \cdot f_{ Isl} \cdot f_{slw} \]

*** Calculate the view factors between the wires and the tube passes of the same layer

\[ f_{ws} = 1 - 2 \cdot \left(f_{wsw} - f_{wsl} \cdot \left(1 - f_{slt}\right) \cdot f_{slw} - f_{wlt}\right) \]
\[ f_{is} = 1 - 2 \cdot f_{ Isl} - f_{tsw} \]

*** Correct these configuration factors for that absorbed by other layers

\[ V_{Fws} = (1 + (NL-1) \cdot \cos(\alpha) \cdot (1 - \text{Tau}) + (NL-1) \cdot \text{Tau}) \cdot f_{wsl} / \text{REAL}(NL) \]
\[ V_{Fts} = (1 + (NL-1) \cdot \cos(\alpha) \cdot (1 - \text{Tau}) + (NL-1) \cdot \text{Tau}) \cdot f_{ts} / \text{REAL}(NL) \]
Return
End

C**********************************************************
Subroutine Radiation(Qtot,Tai,Tri,Tro,Aw,AwP,AtP,Emiss,Ca,Qrad, 2 NITER,Eta,hwr)
implicit none
INTEGER NITER
REAL Qtot,Tai,Tri,Tro,Aw,AwP,AtP,Emiss,Ca,Qrad, 2 Eta,hwr
Data Sigma/5.67E-8/
Tcoil=(Tri+Tro)/2.
Tao=Tai+Qtot/Ca
Ts=(Tai+Tao)/2.
If(NITER.eq.1) then
  Qrad=(AwP+AtP)*Sigma*Emiss*(Tcoil**4-Ts**4)
  Qwr=Qrad*AwP/(AtP+AwP)
Else
  Qrad=(AtP+Eta*AwP)*Sigma*Emiss*(Tcoil**4-Ts**4)
  Qwr=Qrad*Eta*AwP/(AtP+Eta*AwP)
End if
hwr=Qrad/Aw/(Tcoil-Ts)
Return
End

C**********************************************************
Subroutine CharAMC(I,C)
C**Converts a three digit integer, I, to a Character variable, C*3
implicit none
INTEGER I
CHARACTER C*3,CI*I,C2*I,C3*1
IF(I.LT.IO) then
  C=Char(I+48)
  return
END IF
IF(I.LT.100) THEN
  CI=Char(1I10+48)
  C2=Char(I-(IIIO* 10)+48)
  C=ClIIC2
ELSE
  C1=Char(III00+48)
  C2=Char(1I10-(111 00* 10)+48)
  C3=Char(I-(1I10*10)+48)
  C=(C1/C2)/C3
END IF
Return
End

C**********************************************************
Subroutine internal(Tf,Dti,St,Ati,XMdot,HR,R_int)
c
This subroutine determines the thermal resistance corresponding to the
c convection heat transfer between the refrigerant flow and the inner
c surface of the condenser tube wall. The Gnielinski Correlation is used
c to determine the Nusselt number associated with refrigerant flow.
c
C
C RHOR -- density [kg/m^3] of the refrigerant
XMUR -- viscosity [kg/m-s] of the refrigerant
XKR -- thermal conductivity [W/m-K] of the refrigerant
PRR -- Prandtl number of the refrigerant
RER -- Reynolds number associated with the refrigerant flow in the
condenser tube

-- dimensionless friction factor associated with the refrigerant flow in the condenser tube

-- Nusselt number associated with the refrigerant flow in the condenser tube

-- internal pressure [Pa] difference between the inlet and the outlet of a particular condenser layer

implicit none
real Dti, St, Ati, XMdot, Tf
real RHOR, XMUR, XKR, PRR, RER, F, XNUR, HR, R_int, Pdrop

RHOR = 989.

*** Calculate the thermophysical properties of the refrigerant flowing through each condenser layer

XMUR = (8.7128*10.**-2.) - (7.4592*10.**-4.) * Tf +
2 * (2.1577*10.**-6.) * Tf**2. - (2.0997*10.**-9.) * Tf**3.

XKR = - (2.8155) + (2.6844*10.**-2.) * Tf - (7.0477*10.**-5.) * Tf**2.
2 + (6.3449*10.**-8.) * Tf**3.

PRR = (731.79) - (6.3177) * Tf + (1.8383*10.**-2.) * Tf**2.
2 - (1.7969*10.**-5.) * Tf**3.

*** Determine the internal thermal resistance using the Gnielinski Correlation

RER = 4. * XMdot / (PI*Dti*XMUR)
IF (RER.lt.2300) write(*,*) 'Error - Flow is laminar'
IF (RER.gt.5000000.) write(*,*) 'Error - Correlation limits violated'
F = (0.79*log(RER)-1.64)**(-2.)

XNUR = (F/8.) * (RER-1000.) * PRR
2 / (1. + 12.7 * (F/8.)**(1/2.) * (PRR**(2./3.))**(-1.))

HR = XNUR * XKR / Dti

R_int = 1. / (HR*Ati)

Pdrop = 8. * (PI*Dti*real(NT)*10.25+real(NT)*PI**2.*Dti*(ST/2.))

RETURN

END

!!! Gaspt.f
APPENDIX C. EXPERIMENTAL UNCERTAINTY

Table C.1 Absolute uncertainty associated with experimental measurements

<table>
<thead>
<tr>
<th>Variable</th>
<th>Range</th>
<th>P   (precision limit)</th>
<th>B   (bias limit)</th>
<th>U   (uncertainty)</th>
<th>% U</th>
</tr>
</thead>
<tbody>
<tr>
<td>V</td>
<td>0.2 - 2.0 m/s</td>
<td>0.03 m/s</td>
<td>0.003 m/s</td>
<td>0.03 m/s</td>
<td>0.15 - 1.5 %</td>
</tr>
<tr>
<td>T_a,in</td>
<td>&gt; 293 K</td>
<td>0.026 K</td>
<td>0.051 K</td>
<td>0.057 K</td>
<td>&lt; 0.02 %</td>
</tr>
<tr>
<td>T_r,in</td>
<td>&gt; 295 K</td>
<td>0.045 K</td>
<td>0.052 K</td>
<td>0.069 K</td>
<td>&lt; 0.02 %</td>
</tr>
<tr>
<td>T_r,out</td>
<td>&gt; 295 K</td>
<td>0.045 K</td>
<td>0.052 K</td>
<td>0.069 K</td>
<td>&lt; 0.02 %</td>
</tr>
<tr>
<td>Δm</td>
<td>&gt; 3000 g</td>
<td>8.0 g</td>
<td>0.0050 g</td>
<td>8.0 g</td>
<td>&lt; 0.3 %</td>
</tr>
<tr>
<td>Δt</td>
<td>&gt; 180 s</td>
<td>0.30 s</td>
<td>0.0065 s</td>
<td>0.30 s</td>
<td>&lt; 0.2 %</td>
</tr>
<tr>
<td>m_i</td>
<td>9.67 - 15.3 g/s</td>
<td>0.10 g/s</td>
<td>0.25 g/s</td>
<td>0.27 g/s</td>
<td>1.8 - 2.8 %</td>
</tr>
<tr>
<td>Δp</td>
<td>0.679 - 23.320 Pa</td>
<td>0.020 Pa</td>
<td>0.062 Pa</td>
<td>0.065 Pa</td>
<td>0.28 - 9.6 %</td>
</tr>
</tbody>
</table>

Table C.2 Absolute uncertainty associated with calculated quantities

<table>
<thead>
<tr>
<th>Variable</th>
<th>Range</th>
<th>P   (precision limit)</th>
<th>B   (bias limit)</th>
<th>U   (uncertainty)</th>
<th>% U</th>
</tr>
</thead>
<tbody>
<tr>
<td>c_p,r</td>
<td>4.180 kJ/kg-K</td>
<td>0.001 kJ/kg-K</td>
<td>0.001 kJ/kg-K</td>
<td>0.001 kJ/kg-K</td>
<td>0.03 %</td>
</tr>
<tr>
<td>q_tot</td>
<td>142 - 900 W</td>
<td>0.4 - 0.7 W</td>
<td>0.9 - 1.6 W</td>
<td>1.0 - 1.7 W</td>
<td>&lt; 1.2 %</td>
</tr>
<tr>
<td>ε</td>
<td>0.8</td>
<td>0.05</td>
<td>-</td>
<td>0.05</td>
<td>6.3 %</td>
</tr>
<tr>
<td>F_ij</td>
<td>0 - 1.0</td>
<td>0.1</td>
<td>-</td>
<td>0.1</td>
<td>≥ 10 %</td>
</tr>
<tr>
<td>q_rad</td>
<td>23 - 36 W</td>
<td>0.7 - 3.6 W</td>
<td>0.02 - 0.07 W</td>
<td>0.7 - 3.6 W</td>
<td>&lt; 10 %</td>
</tr>
<tr>
<td>q_conv</td>
<td>114 - 875 W</td>
<td>-</td>
<td>-</td>
<td>1.2 - 4.0 W</td>
<td>&lt; 4.0 %</td>
</tr>
<tr>
<td>h_w</td>
<td>11.5 - 107.6 W/m²-K</td>
<td>-</td>
<td>-</td>
<td>1.2 - 4.0 W</td>
<td>3.7 - 10.4 %</td>
</tr>
</tbody>
</table>
APPENDIX D. TABULAR DATA

This appendix presents the raw data associated with the condensers involved in the study. Only the raw data that pertains to the graphical results that are presented in Chapter 5 - Results and Discussion and Appendix E - Additional Plots have been shown here. Tables are listed in order of increasing coil number, and if more than one data set is shown for a particular condenser, the set that represents the majority of the presented results will be shown first.
### Table D.1 Raw data for Coil 1 with 1/4 inch clearance

<table>
<thead>
<tr>
<th>$V$ (m/s)</th>
<th>$\dot{m}_r$ (kg/s)</th>
<th>$T_{a,in}$ (K)</th>
<th>$T_{r,in}$ (K)</th>
<th>$T_{r,out}$ (K)</th>
<th>$\Delta p$ (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.20</td>
<td>0.01266</td>
<td>295.66</td>
<td>319.56</td>
<td>315.54</td>
<td>0.791</td>
</tr>
<tr>
<td>0.25</td>
<td>0.01267</td>
<td>295.75</td>
<td>320.15</td>
<td>315.38</td>
<td>0.997</td>
</tr>
<tr>
<td>0.35</td>
<td>0.01258</td>
<td>295.82</td>
<td>320.40</td>
<td>314.52</td>
<td>1.370</td>
</tr>
<tr>
<td>0.50</td>
<td>0.01256</td>
<td>295.89</td>
<td>320.34</td>
<td>312.73</td>
<td>2.124</td>
</tr>
<tr>
<td>0.75</td>
<td>0.01255</td>
<td>296.10</td>
<td>320.27</td>
<td>310.42</td>
<td>3.780</td>
</tr>
<tr>
<td>1.01</td>
<td>0.01248</td>
<td>296.32</td>
<td>320.53</td>
<td>309.01</td>
<td>6.545</td>
</tr>
<tr>
<td>1.25</td>
<td>0.01249</td>
<td>296.50</td>
<td>320.40</td>
<td>307.75</td>
<td>8.612</td>
</tr>
<tr>
<td>1.50</td>
<td>0.01248</td>
<td>296.65</td>
<td>320.39</td>
<td>306.74</td>
<td>12.193</td>
</tr>
<tr>
<td>1.75</td>
<td>0.01247</td>
<td>296.87</td>
<td>320.18</td>
<td>305.87</td>
<td>15.910</td>
</tr>
<tr>
<td>2.00</td>
<td>0.01243</td>
<td>297.01</td>
<td>320.48</td>
<td>305.35</td>
<td>20.375</td>
</tr>
</tbody>
</table>

### Table D.2 Raw data for Coil 2 with 1/4 inch clearance (Set #1)

<table>
<thead>
<tr>
<th>$V$ (m/s)</th>
<th>$\dot{m}_r$ (kg/s)</th>
<th>$T_{a,in}$ (K)</th>
<th>$T_{r,in}$ (K)</th>
<th>$T_{r,out}$ (K)</th>
<th>$\Delta p$ (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.21</td>
<td>0.01175</td>
<td>296.30</td>
<td>318.90</td>
<td>314.96</td>
<td>0.760</td>
</tr>
<tr>
<td>0.25</td>
<td>0.01177</td>
<td>296.16</td>
<td>319.30</td>
<td>314.63</td>
<td>0.903</td>
</tr>
<tr>
<td>0.35</td>
<td>0.01174</td>
<td>296.11</td>
<td>319.37</td>
<td>313.71</td>
<td>1.221</td>
</tr>
<tr>
<td>0.49</td>
<td>0.01171</td>
<td>296.06</td>
<td>319.56</td>
<td>312.17</td>
<td>1.856</td>
</tr>
<tr>
<td>0.75</td>
<td>0.01167</td>
<td>296.16</td>
<td>319.74</td>
<td>309.93</td>
<td>3.425</td>
</tr>
<tr>
<td>1.00</td>
<td>0.01163</td>
<td>296.28</td>
<td>319.82</td>
<td>308.37</td>
<td>5.504</td>
</tr>
<tr>
<td>1.25</td>
<td>0.01162</td>
<td>296.34</td>
<td>319.67</td>
<td>307.13</td>
<td>7.976</td>
</tr>
<tr>
<td>1.50</td>
<td>0.01166</td>
<td>296.44</td>
<td>319.74</td>
<td>306.13</td>
<td>11.201</td>
</tr>
<tr>
<td>1.75</td>
<td>0.01157</td>
<td>296.53</td>
<td>319.80</td>
<td>305.41</td>
<td>14.956</td>
</tr>
<tr>
<td>2.00</td>
<td>0.01158</td>
<td>296.74</td>
<td>320.04</td>
<td>304.93</td>
<td>19.009</td>
</tr>
</tbody>
</table>
### Table D.3 Raw data for Coil 2 with 1/4 inch clearance (Set #2)

<table>
<thead>
<tr>
<th>V (m/s)</th>
<th>( \dot{m}_r ) (kg/s)</th>
<th>( T_{a,\text{in}} ) (K)</th>
<th>( T_{r,\text{in}} ) (K)</th>
<th>( T_{r,\text{out}} ) (K)</th>
<th>( \Delta p ) (Pa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.20</td>
<td>0.01244</td>
<td>296.15</td>
<td>318.08</td>
<td>314.44</td>
<td>0.685</td>
</tr>
<tr>
<td>0.25</td>
<td>0.01243</td>
<td>296.15</td>
<td>318.42</td>
<td>314.19</td>
<td>0.822</td>
</tr>
<tr>
<td>0.35</td>
<td>0.01242</td>
<td>296.17</td>
<td>318.77</td>
<td>313.42</td>
<td>1.183</td>
</tr>
<tr>
<td>0.50</td>
<td>0.01242</td>
<td>296.26</td>
<td>318.98</td>
<td>312.13</td>
<td>1.806</td>
</tr>
<tr>
<td>0.75</td>
<td>0.01236</td>
<td>296.31</td>
<td>319.14</td>
<td>310.11</td>
<td>3.413</td>
</tr>
<tr>
<td>0.99</td>
<td>0.01234</td>
<td>296.35</td>
<td>319.29</td>
<td>308.75</td>
<td>5.430</td>
</tr>
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### Table D.4 Raw data for Coil 3 with 1/4 inch clearance

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<th>( T_{r,\text{in}} ) (K)</th>
<th>( T_{r,\text{out}} ) (K)</th>
<th>( \Delta p ) (Pa)</th>
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Table D.5 Raw data for Coil 3 with 1/8 inch clearance

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<th>( \Delta p ) (Pa)</th>
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Table D.6 Raw data for Coil 4 with 1/4 inch clearance

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<th>( T_{r,\text{in}} ) (K)</th>
<th>( T_{r,\text{out}} ) (K)</th>
<th>( \Delta p ) (Pa)</th>
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Table D.7 Raw data for Coil 4 with 1/8 inch clearance

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<th>( \Delta p ) (Pa)</th>
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Table D.8 Raw data for Coil 5 with 1/4 inch clearance

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### Table D.9 Raw data for Coil 5 with 1/8 inch clearance

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<th>( T_{r,\text{in}} ) (K)</th>
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<th>( \Delta p ) (Pa)</th>
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### Table D.10 Raw data for Coil 6 with 1/4 inch clearance

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<th>( T_{r,\text{in}} ) (K)</th>
<th>( T_{r,\text{out}} ) (K)</th>
<th>( \Delta p ) (Pa)</th>
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### Table D.11 Raw data for Coil 6 with 1/8 inch clearance

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<th>$T_{r,in}$ (K)</th>
<th>$T_{r,out}$ (K)</th>
<th>$\Delta p$ (Pa)</th>
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### Table D.12 Raw data for Coil 6 with 1/4 inch clearance and open tube passes blocked

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Table D.13 Raw data for Coil 7 with 1/4 inch clearance

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<th>( T_{r,in} ) (K)</th>
<th>( T_{r,out} ) (K)</th>
<th>( \Delta p ) (Pa)</th>
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Table D.14 Raw data for Coil 8 with 1/4 inch clearance (Set #1)

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Table D.15 Raw data for Coil 8 with 1/4 inch clearance (Set #2)

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<th>( T_{r,\text{in}} ) (K)</th>
<th>( T_{r,\text{out}} ) (K)</th>
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APPENDIX E. ADDITIONAL PLOTS

Figure E.1 Effect of orientation on $h_w$ for smaller amplitude

Figure E.2 Effect of orientation on $\Delta p$ for smaller amplitude
Figure E.3 Overall performance for the two orientations at the smaller amplitude

Figure E.4 Overall performance for the two orientations at the smaller amplitude
\( (V \leq 1.0 \text{ m/s}) \)
Figure E.5 Effect of orientation on $h_w$ for larger amplitude

Figure E.6 Effect of orientation on $\Delta p$ for larger amplitude
Figure E.7 Overall performance for the two orientations at the larger amplitude

Figure E.8 Overall performance for the two orientations at the larger amplitude
$(V \leq 1.0 \text{ m/s})$
Figure E.9 Effect of amplitude on $h_w$ for $\psi = 0$

Figure E.10 Effect of amplitude on $\Delta p$ for $\psi = 0$
Figure E.11 Overall performance comparison for the different amplitudes at $\psi = 0$

Figure E.12 Overall performance comparison for the different amplitudes at $\psi = 0$

$(V \leq 1.0 \text{ m/s})$
Figure E.13 Effect of amplitude on $h_w$ for the $\psi = \pi/2$ orientation

Figure E.14 Effect of amplitude on $\Delta p$ for the $\psi = \pi/2$ orientation
Figure E.15 Overall performance comparison for the different amplitudes at $\psi = \pi/2$

Figure E.16 Overall performance comparison for the different amplitudes at $\psi = \pi/2$

$(V \leq 1.0 \text{ m/s})$
Figure E.17 Effect of wire spacing on $h_w$

Figure E.18 Effect of wire spacing on $h_A$
Figure E.19 Overall performance comparison based on the wire spacing 
(V ≤ 1.0 m/s)

Figure E.20 Effect of varying the clearance on $h_w$ for Coil 4
Figure E.21 Effect of varying the clearance on Δp for Coil 4

Figure E.22 Overall performance comparison for Coil 4 with varying clearance (V ≤ 1.0 m/s)
Figure E.23 Effect of varying the clearance on $h_w$ for Coil 5

Figure E.24 Effect of varying the clearance on $\Delta p$ for Coil 5
**Figure E.25** Overall performance comparison for Coil 5 with varying clearance (\( V \leq 1.0 \) m/s)

**Figure E.26** \( h_w \) vs. maximum velocity for all eight condensers
Figure E.27 $hA$ vs. maximum velocity for all coils

Figure E.28 $\Delta p$ vs. maximum velocity for all eight condenser