Fall 9-2012

Optimal Reduction of Electrical Energy Consumption by Supply Air AC Motors

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Optimal Reduction of Electrical Energy Consumption by Supply Air AC Motors

By

Keyhan Rafiee

A THESIS

Presented to the Faculty of
The Graduate College at the University of Nebraska
In Partial Fulfillment of Requirements
For the Degree of Master of Science

Major: Mechanical Engineering

Under the Supervision of Professor C. W. Solomon

Lincoln, Nebraska

September, 2012
Optimal Reduction of Electrical Energy Consumption by

Supply Air AC Motors

Keyhan Rafiee, M.S.

University of Nebraska, 2012

Advisor: C. W. Solomon

The Nebraska Center for Energy Sciences Research (NCER) at the University of Nebraska-Lincoln (UNL) strives to be energy efficient through Green Energy. Of course, in meeting some of the requirements for different types of Leadership in Energy and Environmental Design (LEED) certification, certain minimum levels of energy efficiency practices must be met. One such level of energy efficiency, in particular, is reduction in the consumption of electrical energy in buildings. This thesis is a study of methods of reducing the consumption of electrical energy (measured in kW of electrical power) in the UNL Jorgensen Hall (JH) air handling unit 2 (AHU2) ‘fan wall’ of supply air AC motors/fans.

An analysis of the experimental results showed that the numerical values obtained for the total head loss (and the consequential power loss) from the equations employed in each energy method were sufficiently close to each other. This led to the conclusion that the equations employed in the energy methods were properly constructed. A comparison was made between the magnitudes of the head losses (and consequential power losses) associated with the work done by different motor combinations. Furthermore, this comparison was made while, in every combination, the AC motors were operating at a specific speed to move a mass of air through the AHU2 intake air duct system. The
Optimal performance of the AHU2 supply air AC motors was achieved through this comparison. The specific power consumption, for every different AHU2 supply air AC motor combination tested, was also taken into consideration in order to determine the combined optimum level of performance for these motors in every setting.

The effects of friction factor on major and minor head losses and consequently on total head loss were considered. A scheme was constructed for the linear optimization of the curve for the total head loss against major head loss. It was also concluded that a particular function developed to describe the relationship between friction factor and Reynolds number is not the only relationship through which friction factor can be determined. Friction factor can be determined independent of Reynolds number.

Finally, recommendations were made so as to bring about the transitioning from one motor combination to another as smoothly as practical in order to meet the specific air supply requirements in different seasonal weather conditions. This can be achieved by way of modifying the appropriate segments of an energy management system software program in constant communication with (field) controllers at all times in order to command these AC motors (operating in different combinations) to run as closely as possible to their peak efficiencies only within a rather narrow range of speed and switching them on and off as needed at all times.
This thesis is dedicated to
the memory of my Parents Assad and Cocab Rafiee, and
to the memory of my brother and mentor Siamac.
Acknowledgements

The author would like to sincerely thank his supervisor, Professor C. W. Solomon To for the considerable guidance, advice and support he has provided throughout the course of this research work. He has helped to make this work a rewarding and enjoyable learning experience.

The author would also like to extend his thanks to the members of his examining committee, Dr. David Admiraal, Dr. John Barton and Dr. Solomon To, for their assistance in reviewing this research work and for their inputs and guidance.

The author would like to express his gratitude to the faculty members of the UNL Mechanical and Materials Engineering, and Electrical Engineering Departments for their advice and support. He would like to also thank his fellow students for their comments and encouragement during the course of this research.

The author would like to express his gratitude to the UNL Building Systems Maintenance Department engineering and technical staff members, and in particular, to Jacob Olson and David Tyler for their valuable assistance and support with the instrumentation setup. He would like to thank the UNL Mechanical and Materials Engineering, and Electrical Engineering Departments staff members and in particular Ms. Nanette Rowe for logistics and support with the instrumentation orders.

Finally, the author is thankful to PCB Piezotronics, Inc., for their generous donation of the instruments and the equipment to his research and to their Customer Service, Engineering and Technical Support Departments staff members for their technical advice in utilizing the instrumentation.
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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a$</td>
<td>Slope of line $y_i = ax_i + b$ for station $i = 2, 3$ or $4$.</td>
</tr>
<tr>
<td>$b$</td>
<td>Constant of line $y_i = ax_i + b$ for station $i = 2, 3$ or $4$.</td>
</tr>
<tr>
<td>$A_i$</td>
<td>Internal cross sectional area (at station $i$, $i = 2, 3$ and $4$) of the duct.</td>
</tr>
<tr>
<td>$C$</td>
<td>Constant of line $Y_i = X_i + C$ for station $i = 2, 3$ or $4$.</td>
</tr>
<tr>
<td>$C_o$</td>
<td>Original value for $C$.</td>
</tr>
<tr>
<td>$D_H$</td>
<td>Equivalent (Hydraulic) diameter of the duct.</td>
</tr>
<tr>
<td>$f_{ij}$</td>
<td>Friction factor for stations $i$ and $j$, $i \neq j$.</td>
</tr>
<tr>
<td>$(f_{ij})_m$</td>
<td>Friction factor at midpoint between stations $i$ and $j$, $i \neq j$.</td>
</tr>
<tr>
<td>$h$</td>
<td>Internal height of the cross section for the duct.</td>
</tr>
<tr>
<td>$(L_i^h)_n$</td>
<td>Minor head loss for station $i$, $i = 2, 3$ or $4$.</td>
</tr>
<tr>
<td>$(L_{ij}^h)_m$</td>
<td>Major head loss based on $f_{ij}$.</td>
</tr>
<tr>
<td>$K_L$</td>
<td>Loss coefficient for a $90^\circ$ bend (miter type) in the duct.</td>
</tr>
<tr>
<td>$l_{ij}$</td>
<td>Horizontal length between stations $i$ and $j$.</td>
</tr>
<tr>
<td>$l_{min}$</td>
<td>Minimum savings on a larger air handling unit.</td>
</tr>
<tr>
<td>$l_{max}$</td>
<td>Maximum savings on a larger air handling unit.</td>
</tr>
<tr>
<td>$(L_h^a)_{ij}$</td>
<td>Total head loss calculated based on $Q_a$.</td>
</tr>
<tr>
<td>$(L_h)_{ij}$</td>
<td>Total head loss calculated based on $(y_{ij})_m$.</td>
</tr>
<tr>
<td>$(L_p)_{ij}$</td>
<td>Power loss based on $(L_h)_{ij}$.</td>
</tr>
<tr>
<td>$(L_p^a)_{ij}$</td>
<td>Power loss based on $(L_h^a)_{ij}$.</td>
</tr>
<tr>
<td>$L_{min}$</td>
<td>Minimum savings on all larger air handling units.</td>
</tr>
<tr>
<td>$L_{max}$</td>
<td>Maximum savings on all larger air handling units.</td>
</tr>
<tr>
<td>$p_i$</td>
<td>Static pressure at station $i = 2, 3$ and $4$.</td>
</tr>
<tr>
<td>$P_w$</td>
<td>Wetted perimeter of the duct.</td>
</tr>
<tr>
<td>$P_{bhp}$</td>
<td>Brake horse power.</td>
</tr>
<tr>
<td>$Q_i$</td>
<td>Airflow rate at station $i = 2, 3$ and $4$.</td>
</tr>
</tbody>
</table>
\( Q_a \)  
Average airflow rate at all stations \( i = 2, 3 \) and \( 4 \).

\( (Q_{ij})_m \)  
Airflow rate at midpoint (between stations \( i \) and \( j, i \neq j \)).

\( R_{ij} \)  
Reynolds number based on \( Q_a, A_i, (v_{ij})_m \) and \( D_H \).

\( s_{\text{min}} \)  
Minimum savings on smaller air handling unit.

\( s_{\text{max}} \)  
Maximum savings on smaller air handling unit.

\( S_{\text{min}} \)  
Minimum savings on all smaller air handling units.

\( S_{\text{max}} \)  
Maximum savings on all smaller air handling units.

\( U_a \)  
Average velocity of fluid.

\( U_i \)  
Grid velocity measured at station \( i, i = 2, 3 \) and \( 4 \).

\( (U_i)_a \)  
Average air velocity at station \( i, i = 2, 3 \) and \( 4 \).

\( (U_m)_i \)  
Mean velocity calculated based on energy method including energy coefficients at station \( i, i = 2, 3 \) or \( 4 \).

\( w \)  
Internal width of the cross section for the duct.

\( X_i \)  
Major head loss in line \( Y_i = X_i + C \) for station \( i = 2, 3 \) or \( 4 \).

\( (X_i)_o \)  
Original value of \( X_i \) for station \( i = 2, 3 \) or \( 4 \).

\( Y_i \)  
Total head loss in line \( Y_i = X_i + C \) for station \( i = 2, 3 \) or \( 4 \).

\( (Y_i)_o \)  
Original value of \( Y_i \) for station \( i = 2, 3 \) or \( 4 \).

\( z_{ij} \)  
Relative elevation between stations \( i \) and \( j, i \neq j \).

\( \alpha_i \)  
Energy coefficient for station \( i = 2, 3 \) or \( 4 \).

\( (y_{ij})_m \)  
Specific weight at midpoint (between stations \( i \) and \( j, i \neq j \))

average temperatures.

\( \varepsilon \)  
Relative roughness of duct.

\( (\mu_{ij})_m \)  
Dynamic viscosity at midpoint (between stations \( i \) and \( j, i \neq j \))

average temperatures.

\( (\rho_{ij})_m \)  
Density at midpoint (between stations \( i \) and \( j, i \neq j \))

average temperatures.

\( (v_{ij})_m \)  
Kinetic viscosity at midpoint (between stations \( i \) and \( j, i \neq j \))

average temperatures.

\( \Phi \)  
Minimization function.
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Chapter 1  Introduction

In this chapter a background is provided in terms of the type of equipment and the requirements which have been imposed on it will be detailed. The outlines for the scope of the theoretical approaches in this investigation are discussed. The objectives pursued in this study along with a survey of the literature will also be discussed. The organization of the remaining parts of the thesis will be described.

1.1  Background and Objectives

In the course of his employment as a Controls Engineer in the Building Systems Maintenance (BSM) Department at UNL, the author studied the design and operation of several components of different types of commercial heating, ventilation and air conditioning (HVAC) systems in the UNL office buildings. One such HVAC component was the number 2 air handling unit (AHU2) which is located in Mechanical Room 411 of the newly constructed Physical Sciences building, Jorgensen Hall (JH), in the UNL City Campus. In this study, Site A will refer to UNL/JH/HVAC system/AHU2 including its intake air duct system. The entire experimental phase of this study did not take place in a laboratory setting, but at Site A. The entire data acquisitioning, covering all of the dates in which a series of measurements have been taken of the variables such as static pressure and the flow rate of the incoming air have all been performed at Site A. The actual dates and their corresponding date numbers upon which the data acquisitioning have taken place in the process of the investigation in this study are included in Table 1.1. The UNL/BSM department has expressed their wishes to the author to conclusively determine
whether or not there is energy saving benefits to be had in the ways of operating the AHU2 ‘fan wall’ technology.

Table 1.1  Actual Dates of the Data acquisition and their Date Numbers.

<table>
<thead>
<tr>
<th>Actual Date</th>
<th>Date Number</th>
</tr>
</thead>
<tbody>
<tr>
<td>07/10/2011</td>
<td>1</td>
</tr>
<tr>
<td>10/31/2011</td>
<td>2</td>
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<tr>
<td>12/04/2011</td>
<td>3</td>
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<tr>
<td>01/01/2012</td>
<td>4</td>
</tr>
<tr>
<td>01/08/2012</td>
<td>5</td>
</tr>
<tr>
<td>02/26/2012</td>
<td>6</td>
</tr>
</tbody>
</table>

In so doing, it is also desired for the author to find an optimal control method for operating the AHU2 supply air AC motors/fans in such a way so as to meet the airflow requirements in the range of 4.25, usually for summer, to 7.08, usually for winter, \( \frac{m^3}{s} \) (9000 to 15000 cfm). Another requirement is, at the same time, to constantly maintaining a 3.81 cm (1.5 inches) height of water column of static pressure downstream of AHU2 at a distance equal to \( \frac{2}{3} \) of the length of the duct system before the last room (to be served its supply of air by AHU2) has been reached. AHU2 houses a frame of a \( 3 \times 3 \) matrix of 8 Supply Air AC motors/fans each secured in a cell in addition to an empty cell, all of which are positioned in a frame or a “wall” with the whole motor-wall assembly referred to as the ‘fan wall’ that is located in a chamber of AHU2. Such an assembly of supply air
AC motors is usually located in a particular chamber of an air handling unit of this type. In the vernacular of the manufacturing industry, AHU2 is referred to as an air handling unit which is equipped with the ‘fan wall’ technology. Figures 1.1, 1.2 and 1.3 show the location of these motors in the ‘fan wall’ of AHU2, the sequences of operations in the path of %100 outside air flowing inside AHU2, and the path of communication between controllers, computers and HVAC devices in the field, respectively.
This is the Air Handling Unit II.

This is the Fan Wall located in a chamber of the Air Handler.

This is the location of the cell with the missing AC motor.

This is a typical 7 ½ hp AC motor, a total of 8

Figure 1.1  Supply Air AC Motors Orientations in the AHU2 Fan Wall.
Figure 1.2  Air Handler Unit II and the Sequences of Operations.
(a) Air Handler II as a 100% Outside Air Type,
Figure 1.2  Air Handler Unit II and the Sequences of Operations.
(b) Sequences of Operations,

<table>
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<tr>
<th>TAG</th>
<th>DESCRIPTION</th>
<th>ACRONYM</th>
<th>PORT</th>
<th>I/O TYPE</th>
<th>BASE RANGE</th>
<th>DETAIL</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>OUTDOOR TEMP</td>
<td>JH.AH2.FA.TP</td>
<td>1</td>
<td>AI</td>
<td>-20 TO 120 DEG F</td>
<td>SX3-S01</td>
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<tr>
<td>2</td>
<td>OUTDOOR AIR HUMIDITY</td>
<td>JH.AH2.FA.RH</td>
<td>2</td>
<td>AI</td>
<td>0 TO 100% RH</td>
<td>SX3-S08</td>
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<tr>
<td>3</td>
<td>OUTDOOR AIR DAMPER</td>
<td>JH.AH2.FA.DAMPER_ISO</td>
<td>3</td>
<td>DI</td>
<td>OPEN/CLOSED</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>PREFILTER 1 PRESSURE DIFF</td>
<td>JH.AH2.FILTER.P1PR_DIFF</td>
<td>4</td>
<td>AI</td>
<td>0 TO 1 INCH WC</td>
<td>SX3-S06</td>
</tr>
<tr>
<td>5</td>
<td>HEAT RECOVERY AIR TEMP</td>
<td>JH.AH2.ERC.TP</td>
<td>5</td>
<td>AI</td>
<td>0 TO 100 DEG F</td>
<td>SX3-S02</td>
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<td>6</td>
<td>PREHEAT COIL VALVE</td>
<td>JH.AH2.PH.VALVE</td>
<td>6</td>
<td>AO</td>
<td>2 TO 10 VOLT</td>
<td>SX3-C04</td>
</tr>
<tr>
<td>7</td>
<td>PREHEAT COIL AIR TEMP</td>
<td>JH.AH2.PH.TP</td>
<td>7</td>
<td>AI</td>
<td>0 TO 100 DEG F</td>
<td>SX3-S02</td>
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<td>12</td>
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<td></td>
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<td></td>
<td></td>
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<tr>
<td>17</td>
<td>SUPPLY FAN 1 ENABLE</td>
<td>JH.AH2.SFAN1.ENABLE</td>
<td>MB-1</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C05</td>
</tr>
<tr>
<td>18</td>
<td>SUPPLY FAN 1 VFD</td>
<td>JH.AH2.SFAN1.VFD</td>
<td>MB-1</td>
<td>AO</td>
<td>0 TO 100%</td>
<td>SX3-C05</td>
</tr>
<tr>
<td>19</td>
<td>SUPPLY FAN 2 ENABLE</td>
<td>JH.AH2.SFAN2.ENABLE</td>
<td>MB-2</td>
<td>DO</td>
<td>ON / OFF</td>
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<td>SUPPLY FAN 2 VFD</td>
<td>JH.AH2.SFAN2.VFD</td>
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<td>AO</td>
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<tr>
<td>21</td>
<td>SUPPLY FAN 3 ENABLE</td>
<td>JH.AH2.SFAN3.ENABLE</td>
<td>MB-3</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C05</td>
</tr>
<tr>
<td>22</td>
<td>SUPPLY FAN 3 VFD</td>
<td>JH.AH2.SFAN3.VFD</td>
<td>MB-3</td>
<td>AO</td>
<td>2 TO 100%</td>
<td>SX3-C05</td>
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<tr>
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<td>SUPPLY FAN 4 ENABLE</td>
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<td>SX3-C05</td>
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<td>MB-5</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C05</td>
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</table>
Figure 1.2  Air Handler Unit II and the Sequences of Operations.  
(c) Sequences of Operations and Valve Assignments.

<table>
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<th>TAG</th>
<th>DESCRIPTION</th>
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<td>AO</td>
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<td>SX3-C05</td>
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<td>MB-6</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C05</td>
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<td>AO</td>
<td>1 TO 100%</td>
<td>SX3-C05</td>
<td>LOCATION: PENTHOUSE</td>
</tr>
<tr>
<td>29</td>
<td>SUPPLY FAN 7 ENABLE</td>
<td>JH.AH2.SFAN7.ENABLE</td>
<td>MB-7</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C05</td>
<td>SYSTEM LOCATION: PENTHOUSE</td>
</tr>
<tr>
<td>30</td>
<td>SUPPLY FAN 7 VFD</td>
<td>JH.AH2.SFAN7.VFD</td>
<td>MB-7</td>
<td>AO</td>
<td>2 TO 100%</td>
<td>SX3-C05</td>
<td>FIMUX LINK #: 12</td>
</tr>
<tr>
<td>31</td>
<td>SUPPLY FAN 8 ENABLE</td>
<td>JH.AH2.SFAN8.ENABLE</td>
<td>MB-8</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C05</td>
<td></td>
</tr>
<tr>
<td>32</td>
<td>SUPPLY FAN 8 VFD</td>
<td>JH.AH2.SFAN8.VFD</td>
<td>MB-8</td>
<td>AO</td>
<td>3 TO 100%</td>
<td>SX3-C05</td>
<td></td>
</tr>
<tr>
<td>33</td>
<td>NOT USED</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>34</td>
<td>NOT USED</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>35</td>
<td>HUMIDIFIER ENABLE</td>
<td>JH.AH2.HUM2.ENABLE</td>
<td>8</td>
<td>DO</td>
<td>ON / OFF</td>
<td>SX3-C09</td>
<td></td>
</tr>
<tr>
<td>36</td>
<td>HUMIDIFIER COMMAND</td>
<td>JH.AH2.HUM2.CMD</td>
<td>9</td>
<td>AO</td>
<td>0 TO 10 VOLT</td>
<td>SX3-C09</td>
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<tr>
<td>37</td>
<td>HUMIDIFIER FAULT ALARM</td>
<td>JH.AH2.HUM2.ALARM</td>
<td>10</td>
<td>DI</td>
<td>NORMAL/ALARM</td>
<td>SX3-C09</td>
<td></td>
</tr>
<tr>
<td>38</td>
<td>FREEZE STAT</td>
<td>JH.AH2.HUM2.FREEZE</td>
<td>MB-9</td>
<td>DI</td>
<td>NORMAL/ALARM</td>
<td>SX3-C08</td>
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</tr>
<tr>
<td>39</td>
<td>COOLING COIL VALVE</td>
<td>JH.AH2.CC.VALVE</td>
<td>11</td>
<td>AO</td>
<td>2 TO 10 VOLT</td>
<td>SX3-C04</td>
<td></td>
</tr>
<tr>
<td>40</td>
<td>PRE FILTER A PRESSURE DIFF</td>
<td>JH.AH2.FILTER_P2.PR_DIFF</td>
<td>12</td>
<td>AI</td>
<td>0 TO 1 INCH WC</td>
<td>SX3-S06</td>
<td>NOTES: B352+ARB24-SR</td>
</tr>
<tr>
<td>41</td>
<td>FINAL FILTER PRESSURE DIFF</td>
<td>JH.AH2.FILTER_F.PR_DIFF</td>
<td>13</td>
<td>AI</td>
<td>0 TO 2 INCH WC</td>
<td>SX3-S06</td>
<td>(2&quot; 3-WAY BALL VALVE)</td>
</tr>
<tr>
<td>42</td>
<td>SUPPLY AIR CFM</td>
<td>JH.AH2.SFAN.CFM</td>
<td>14</td>
<td>AI</td>
<td>0 TO 27000 CFM</td>
<td>SX3-S03</td>
<td></td>
</tr>
<tr>
<td>43</td>
<td>SUPPLY AIR TEMP</td>
<td>JH.AH2.CC.TP</td>
<td>15</td>
<td>AI</td>
<td>0 TO 100 DEG F</td>
<td>SX3-S01</td>
<td></td>
</tr>
<tr>
<td>44</td>
<td>SUPPLY AIR HUMIDITY</td>
<td>JH.AH2.SA.RH</td>
<td>16</td>
<td>AI</td>
<td>0 TO 100% RH</td>
<td>SX3-S08</td>
<td></td>
</tr>
<tr>
<td>45</td>
<td>HUMIDITY HIGH LIMIT</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>46</td>
<td>FIRE/SMOKE DAMPERS</td>
<td>JH.AH2.ALARM.DAMPER_SMK</td>
<td>MB-9</td>
<td>DI</td>
<td>OPEN/CLOSED</td>
<td>SX3-C08</td>
<td></td>
</tr>
<tr>
<td>47</td>
<td>SUPPLY AIR STATIC PRESSURE</td>
<td>JH.AH2.SA.PR</td>
<td>17</td>
<td>DI</td>
<td>0 TO 5 INCH WC</td>
<td>SX3-S04</td>
<td></td>
</tr>
<tr>
<td>48</td>
<td>FIRE ALARM STATUS</td>
<td>JH.AH2.ALARM.FIRE</td>
<td>MB-9</td>
<td>DO</td>
<td>NORMAL/ALARM</td>
<td>SX3-C08</td>
<td></td>
</tr>
<tr>
<td>49</td>
<td>ENERGY RCVRY COIL VALVE</td>
<td>JH.AH2.ERC.VALVE</td>
<td>18</td>
<td>AO</td>
<td>2 TO 10 VOLT</td>
<td>SX3-C04</td>
<td></td>
</tr>
<tr>
<td>50</td>
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<td></td>
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</table>
Figure 1.3  Communication Path of Controllers with Computers and Devices.
In the following, it is only the presumed superiority of the ‘fan wall’ technology which will be discussed. With this in mind, at some point in time, the author was informed that UNL has purchased a number of air handling units, all of which have utilized the ‘fan wall’ technology in their design. This technology is manufactured by HUNTAIR, Inc., which is a CES Group Company. The air handling units purchased by UNL are different sizes, e.g., larger or smaller in their physical dimensions as well as in their corresponding air handling capacities. The ‘fan wall’ technology which is employed in different air handling units are to meet specific needs of each unit. That is, each specific ‘fan wall’ is of a different matrix size holding multiple AC motors/fans. The author has knowledge of two different types of ‘fan wall’ units installed in the air handling units purchased by UNL. The smaller unit has a $3 \times 3$ matrix of a total of 8 AC motors in their ‘fan wall’, as in $8 = 3 \times 3 - 1$ supply air AC motors (each contained in a cube) and the last cube is left empty in the ‘fan wall’. The larger unit has a $4 \times 4$ matrix of a total of 16 AC motors (each contained in a cube) in the ‘fan wall’.

In a summary analysis (describing the benefits in terms of switching from a conventional type of fan system over to a ‘fan wall’ technology type) prepared by HUNTAIR, Inc., [1.1] it is claimed that, in the case of the Phoenix Convention Center (based on a tabulated form of results for savings), there was an annual savings of $122,296 and a lifetime savings of $2,445,920 in terms of energy savings alone. According to The NEWS [1.2], in the case of energy savings and at the same time exceeding the air flow and static pressure requirements, “Dave Benson, president, HUNTAIR, Inc., noted, ‘We calculated a 43 percent savings in energy use by switching to a FANWALL array, and that was just delivering the original airflow. After ramping up
to a higher airflow, and operating against a higher design static pressure, the FANWALL array still requires less BHP [Brake Horse Power] and has a lower connected load.’ ”

In a company publication prepared by CES Group [1.3] which discusses the ways to optimize the performance and energy efficiency of ‘fan wall’ systems, a comparison was made between conventional fan systems and ‘fan wall’ systems. It is claimed in this publication that, among other things, ‘fan wall’ systems can optimize air handler performance by system optimization controls allowing on/off control of individual fan cubes in such way that requirements in variable airflow applications are closely matched and at the same time the optimum number of fans and motors are in operation at their peak efficiencies. In other words, the manufacturer of the ‘fan wall’ technology, HUNTAIR, Inc., believes that when it comes to energy savings which is associated with commercial air handling units, there are a number of considerable advantages in switching from a conventional fan system to a multiple fans scheme.

The operational conditions, under which the testing of this unit took place, created certain advantages and disadvantages for this study. Some of the advantages are that the measurement process for the variables of interest in this study took place in real time, under the actual conditions, and for the specific purposes of meeting the particular needs of the UNL/JH building. Some of the disadvantages however are that in the course of experimenting on AHU2 numerous limitations were encountered in the process of acquisitioning the data that were necessary for accurately analyzing the performance of this unit. For the purposes of conducting the experimental phase of this study, the AHU2 intake air duct system has exhibited some severe physical and operational limitations. Some examples of these limitations are as follows:
(1) In terms of accessibility, only a rather short length of duct in the entire AHU2 duct system was available to take measurements of acoustic pressure, static pressure and air velocity. The AHU2 supply air AC motors must not be running below a certain frequency (or rpm) in their operating speeds while AHU2 is in operation. When these motors are operating at sufficiently low speeds, they will not be able to create sufficient power to drag an adequate volume of air to pass through the coils which are installed in a particular chamber inside AHU2. Inadequate flow of air inside AHU2 may create a condition under which the heat of the fluid (running inside the coils) will not be transferred to the air passing through the coils in time thereby causing severe damage to these internal components of this air handling unit.

(2) For the purposes of probing the AHU2 duct system downstream the supply air AC motors, with instruments to measure variables such as acoustic pressure and air velocity, the duct system of this air handling unit was virtually inaccessible.

It should also be noted that the process of investigating AHU2 in this study was delayed for the following reasons:

(1) The construction of UNL/JH building was completed in the early part of 2010.

(2) The period of time in which the commissioning of UNL/JH building took place was sometime in the spring of 2010.

(3) As early as in the summer of 2010, the process of investigating all available air handling units in the entire UNL/HVAC systems had begun in order to determine their suitability for an experimental setup in this study.
(4) In the Fall semester of 2010, it was decided that AHU2 was a suitable air handling unit for the experiment.

(5) During the Fall semester of 2010, two major theoretical approaches were considered and discussed for this study, namely, the acoustical and the hydraulic approaches.

(6) All instruments, sensors and equipment for both theoretical approaches were obtained in the early part of 2011 at which time the experimental setup had been completed and the data acquisition process had commenced.

The basis of the acoustical approach in this thesis was the original study by To and Doige [1.4]. Owing to the limitations of the AHU2 and its duct systems taking measurements for the acoustic pressure in a proper manner, however, was not possible. Among the reasons which led to the decision for abandoning the acoustical approach in this study, there was one particular reason which is divided in three parts as follows:

(1) For plane wave propagation to happen, the fluid flow must take place in a laminar regime. For a laminar regime to exist, so as to make possible taking useful measurements of acoustic pressure at certain cross sections of the AHU2 intake air duct system, the AHU2 supply air AC motors/fans must operate at sufficiently low speeds.

(2) During the experimental phase of this study, while acoustic pressure measurements were taken, the operating speeds of the AHU2 supply air AC motors/fans were reduced to as low as 24 Hz. Even at this operating speed, however, such measurements taken at the designated measuring stations 2, 3 and 4 did not yield any useful information. It was, therefore, not possible to
confirm that, in general, plane wave propagation had taken place in the duct and that the formation of standing waves at any of the measuring stations had taken place in particular.

(3) An operating speed of 24 Hz for the AHU2 supply air AC motors is at the borderline of the threshold for the lower safety limit of AHU2 operating conditions. In the course of conducting the experimental phase of this study, operating any number of these motors at any speeds lower than 24 Hz for a sufficiently long period of time may have led to inflicting severe internal damage to the components of AHU2.

Owing to the limitations that were described in the above, the existence of a laminar regime for airflow through the AHU2 intake air duct system was therefore never guaranteed. Consequently, the acoustical approach was abandoned in its entirety in the late summer 2011 in this study.

In the meantime, the process of data acquisition for the hydraulic approach had also begun sometime in April 2011. This process ended in February 2012, at which time a full cycle of the seasonal range of recordings made had been covered for measurements of the fresh air temperature to be supplied to the UNL/JH building by AHU2. During the experimental phase of this study, the author had noted that (in conjunction with the performance chart of the AHU2 supply air AC motors) the VFD digital display readings had revealed useful information in relation to the efficiency of these motors at different operating conditions. In general, the VFD settings for this type of motor are arranged in such a way that the digital display readings contain a variety of useful information in regard to the performance of each of the motors. The VFDs constantly display motor data
which consist of actual readings for the values of variables such as speed (in Hz or rpm), instant power consumed (in kW), torque, etc. Each section of Tables 3.1 and 3.2 (see Section 3.2 in Chapter 3) contains samples of such data for the respective motor combinations examined.

Furthermore, other observations which had been made during the experimental phase of this study revealed that the present operating motor combination (i.e., all 8 motors constantly running at speeds in the frequency range of (34 to 39) Hz had been operating well below the peak efficiency of this type of AC motor. The method of operation of these motors has been set up in such a way that all 8 motors will run (at speeds fluctuating in the above-mentioned range) to satisfy that portion of the airflow rate and static pressure requirements of Site A to be maintained at all times for a variety of incoming fresh air qualities.

For the purposes of this study and to better understand the reasons underlying speed fluctuations in each of the AC motors, as their fan wheels are forcing the fresh air into the building through the AHU2 intake air duct system so as to meet the air supply requirements associated with this air handler, a number of properties of the fresh air such as temperature, relative humidity, density and specific weight should be analyzed and their effects on this process (i.e., the movement of air in the AHU2 intake air duct system) be taken into account.

It is therefore helpful in this study to always keep in mind the overall picture of the UNL/JH building supply/exhaust air requirements when analyzing the performance of AHU2 supply air AC motors. A number of assumptions have also been made throughout this study in order to simplify the calculations of results based on the experimental data
which have been acquired in the investigation of AHU2. These assumptions, along with
the reasoning behind them in each case, will be discussed in detail in the appropriate
places in later chapters in this thesis.

In this connection, however, there are several factors to be considered with
respect to these AC motors at any given operational setting. These factors are:

(1) Speed of the AHU2 supply air AC motors along with the,
(2) state (temperature and quality) of the incoming air in addition to the,
(3) overall pressure balance of the building since there are a number of air handlers,
in addition to AHU2, including a number of exhaust systems in operation at the
same time which together create this pressure balance, and
(4) numerous variable air volume (VAV) and makeup air volume (MAV) terminal
boxes which are usually individually assigned to each room, office and laboratory
in the UNL/JH building that may have an effect in terms of the overall pressure
balance of the building.

Each MAV or VAV terminal box assembly is equipped with a damper in its
interior. At any time, a typical damper of this type (e.g., a rather small sized damper in
physical dimensions) may be in a fully open position or be in a variety of other positions
fluctuating in the (0 to 100) % range of open position. It is in this way that the totality of
the MAV and VAV terminal boxes in all of the air handling units in the UNL/JH building
HVAC system may significantly contribute to the overall pressure balance of the
building. However, an assumption has been made in this study that the majority, if not
all, of the MAV and VAV terminal boxes in the entire UNL/JH building HVAC systems
in general and the MAV and VAV terminal boxes in the rooms, offices and the
laboratories which are served by AHU2 in particular do not play a crucial role in the accuracy of the results of this study. It is actually assumed that the dampers in these terminal boxes will mostly remain in a certain open position at all times or otherwise do not affect the actual airflow rate measurements significantly.

Where it was deemed necessary to obtain as accurate of results as possible, it has been decided then to have all of the available and major VAV terminal boxes in the AHU2 duct system set to be in the 100% open position throughout a particular testing period during the investigation. In addition to this, the occasions chosen for the purposes of taking measurements were such that the variability and fluctuations in the AHU2 duct system was at its lowest level possible. This approach, in the measurement taking process, was adopted only in certain cases and for the purposes of making certain that the comparisons made (between the total head loss and the corresponding power loss, each associated with a certain number of selected AC motor combinations) shall be as accurate as possible.

There exists a software system which is called Java Energy Management System (JEMS) that has been developed by the UNL/BSM Department engineering staff. This software has the capability, among other things, to control the functions of as many components as there are in the UNL/HVAC systems which need to be controlled. The schematic entitled “Energy Management Control System Topology”, in Figure 1.4 (below) is a depiction of how a typical series of controllers (which are installed in the racks inside a cabinet that also houses a field computer all of which are usually located in the mechanical rooms of the UNL buildings) communicate with JEMS all the time.
Figure 1.4  Energy Management Control System Topology.
These controllers routinely send and receive signals through wire connections with sensors, actuators, instruments, valves and other similar devices in pumps, motors, machinery, equipment, chillers, heat exchangers, energy recovery systems, etc. Such controllers can also communicate with the variable frequency drive (VFD) devices associated with a typical air handling unit or with any other component in the entire HVAC system. See Figures 3.2(a) and 3.3(a) in Chapter 3, Section 3.2.

From a controls monitoring unit work station, for example, located in the energy management system center at the UNL/BSM Department or even by utilizing a personal computer (PC) equipped with a copy of the JEMS software, an authorized person can (from a remote site) communicate with any one of these controllers through an appropriate router and/or field computer. In this way for example an authorized operator, who is communicating with the controllers (in the field) which are assigned to monitor and to command the components of AHU2, can at any time utilize JEMS to monitor, increase and/or decrease the speeds of any and all AHU2 supply air AC motors.

However, upon the creation of an additional programming software (which can be developed, in JEMS, by the UNL/BSM department computer software engineering staff) there will no longer be a need for an authorized UNL/BSM personnel member to monitor and modify, from time to time, the operating conditions of a particular motor combination such as its speed settings, etc., in order to achieve an optimal overall operating conditions so as to meet the requirements assigned to AHU2 in terms of the supply of air and the static pressure. By combining this additional programming software with the existing program in JEMS a much more versatile software can be created which will then be capable of routinely communicate with the controllers in the field at all
times. An example of controllers such as these is one which have already been placed in
the field for the purposes of communicating with, and if needed through the VFDs,
commanding the AHU2 supply air AC motors, within their ‘fan wall’ structure, to deliver
the requirements of supply of air and the static pressure imposed upon AHU2 at all times.
Wasteful and inefficient operation of machinery and equipment in commercial buildings
is a major concern for a related facilities management organization. Minimizing energy
consumption in facilities and commercial buildings is a primary concern for an energy
management system. Improvements are always being made to the energy management
system in existence for maintaining the operations of the UNL buildings. The aim of this
study is to bring about an improvement in the ways of minimizing the electrical power
consumption in a particular component of the UNL/JH/HVAC systems. In a thesis
project, involving optimal reduction of electrical energy consumption as a significant part
of the operating costs of a commercial air handling unit, it is always a safe way to
implement a robust optimization method. An optimization method, as such, can employ
iterative procedures so as to optimally control a variable such as airflow rate or static
pressure in order to be capable of producing the desired results. Owing to the limitations
of the equipment that have been encountered in this study, however, the procedures
embodied in the computer programming software are the only method available which
can optimize the operation of the AHU2 supply air AC motors. In summary, the main
objectives of the present investigation are:

(1) It is imperative to provide an answer to the main question in this study, namely,
whether or not at any given time there is a significant loss of power as a mass of
fresh air is being forced to move through the AHU2 intake duct system.
(2) The answer to this question is to be sought through the results achieved by employing the hydraulic approach which has been chosen for this purpose. An assessment should be made of any and all significant losses of energy taking place in the process where a mass of fresh air is moving through the AHU2 intake air duct system.

(3) At what AHU2 supply air AC motors/fans operating speeds are the significant losses of power the highest (or lowest) value taking place. Furthermore, what particular operating motor combinations (which are housed in the ‘fan wall’ of this air handling) are associated with these losses.

(4) It is also desired to achieve as accurate of results for this power loss as it is in practice possible. Once the desired results for such power losses have been achieved, a comparison is to be made between such losses (each of which is associated with a particular combination of these motors/fans).

(5) This is in order to find arrangements (or an operating scheme) in which a number of different motor combinations are constantly switching from one to another so that the specific requirements of the airflow rate and the static pressure, as they have already been mentioned in this section, are met at all times throughout the seasonal changes in the weather condition.

(6) The particular relationship between friction factor and Reynolds number and the simplification of the effects of friction factor on head loss are to be examined.
1.2 Literature Review

Describing the impact of digital controls on the building industry in Coffin [1.5], references are made to the microprocessor-based controllers as having emerged to be an important tool for, among other things, managing human energy consumption. Like any other microprocessor device, a digital control system is serving as a tool for manipulating and processing information. As such, digital control systems provide valuable information in reducing operating costs in commercial buildings. Microprocessors, in control systems which have central communications capabilities, function to track building demand levels in order to reduce the building energy consumption. It is further recommended on page 6 in [1.5] that since the two largest expenses in operating a commercial building are electricity costs for lighting and for HVAC, then digital controls can be used in reducing energy costs, “...without creating an adverse impact on building occupants, through strategies such as load shedding, optimum start/stop, and reduction of the operating speed of electric motors.”

In a paper by M. Liu et al. [1.6], the static pressure of air in the duct, which is to be maintained by the modulations of the supply air motor/fan speed, has been treated as a typical control variable. It is stated in [1.6] that the static pressure is typically to be set at a constant set point depending on the system design information and the location of the sensor. There are cases where there is only partial load condition (of the supply fan) which is required. This is due to the fact that the required airflow being less than the design airflow would lead to much less pressure loss and the terminal box dampers to be closed more. It is argued in [1.6] that in such cases the static pressure set point should be
reset lower so as to reduce fan power and to avoid noise at terminal box dampers and box damper malfunction due to excessive pressure.

In Warren and Norford [1.7] during an eleven month experimental period in which the supply fans tested had all been running primarily using a pressure preset strategy, with the terminal boxes in the 100% open position, within a given range, the fan power increases as static pressure in the duct increases. It is concluded in [1.7] that having installed VFDs in VAV supply air systems, then the dynamic reset of static pressure, as compared to the fixed static pressure control, can have savings in fan energy consumption of

(1) 19% to 20% for two smaller fans operating only during occupied hours, and
(2) 37% to 42% for two larger fans operating 24 hours a day.

As shall be described in detail in Chapter 3, in the course of the investigation, the reliability of the collected experimental data for static pressure and airflow measurements at different measuring stations (i.e., stations 2, 3 and 4, see Figure 2.1, Section 2.1 in Chapter 2) was supported by the utilization of some very accurate instruments available which the author of this thesis used for the purposes of data acquisition in the experimental phase of this study.

However, in a study by Liu et al. [1.8], that was performed on $2 \times 93 \, kW$ supply air and $3 \times 30 \, kW$ return air conventional type fans, an in situ measurement method has been developed for measuring the fan curve without interrupting the normal operations of the air handling system. This method only requires one direct airflow measurement which can be taken at a particular airflow measuring station. It has been concluded in [1.8] that the results from the laboratory experiment show that there is agreement between the fan
curves identified by the in situ method and the full fan speed method. It has been further concluded in [1.8] that, when the airflow station method is used, the accuracy of fan airflow measurement is ensured by the in situ fan curve measurement.

In a study by Liu and Claridge on page 221 in [1.9] it is stated that: “The air static pressure is defined here as the air static pressure [produced by an air handling unit when in operation] at two-thirds of the distance down the main duct.” (parentheses added). According to the study in [1.9], one of the measures which can be implemented to either control or influence the air static pressure is by installing a VFD on the supply fan so as to control the static pressure in the cold duct at a minimum level which can be considered as retrofit (i.e., the improving of existing buildings with energy efficiency equipment).

Elsewhere in [1.9] it is mentioned that, in current engineering practices, a VFD is not used for the constant-volume system. However, it is also suggested on page 223 in [1.9] to the VFD be used in order to control the static pressure on the cold air deck to a minimum level as a consequence of which the fan power energy consumption is reduced.

In another study by Liu in [1.10], a control method for airflow in VAV systems has been developed. It has been termed, on page 318, as the variable speed drive volumetric tracking (VSDVT). In this method, the supply and return airflows are determined by using the signals of the VSDs rather than flow stations; for the simulation considered, the VSDVT maintains a constant building pressure and the required outside airflow under all load conditions and significantly reduces the supply and return air fan energy. In [1.10], the optimal reset of the static pressure is taken to be critical for minimizing supply fan energy. A reference is made on page 318 in [1.10] to a method of modulating fan speed in modern direct digital control (DDC) systems which has been
developed in Hartman [1.11] to state that: “When properly designed and maintained, this method can produce significant fan energy savings.” Another reference is made on page 318 in [1.10] for an improvement of this method that has been developed in Wei et al. [1.12], by integrating it with static pressure reset techniques to prevent it from malfunctioning under several typical building operating conditions. It is, however, mentioned on page 318 in [1.10] that this method cannot be used for a building which uses pneumatic terminal box controllers. Furthermore, for the study which has been made in [1.10], an ideal static pressure reset is assumed. It is then concluded in [1.10] that:

1. The VSDVT method that has been developed for airflow control in VAV systems can be used for typical AHUs having programmable controllers, since the volumetric tracking is implemented using VSD speed signals and the fan heads,

2. Under all load conditions, the VSDVT ensures the required outside air intake and the positive building pressure,

3. Minimal fan energy results when ideal static pressure reset is used, and

4. The VSDVT is developed based on the assumption that fan airflow can be measured accurately using fan head and fan speed.

There are two studies that are cited on page 318 in [1.10], namely, D. M. Elovitz in [1.13] and T. Cohen in [1.14], respectively, in order to state that the accuracy in the airflow measurement “has proved to be difficult if not impossible for most systems due to the lack of the appropriate length of straight ductwork.” In these two references, examples of the duct system physical characteristics which contribute to the difficulties encountered in achieving accurate measurements are discussed as follows:
(1) On page 616 in [1.13] under the heading: “DIRECT MEASUREMENT OF OUTSIDE AIR” it is mentioned that, in maintaining a constant amount of outside air ventilation, the most obvious approach is to measure the amount of outside air being brought into the system and to directly control the outside air quantity. This task cannot, however, be accomplished so easily. The outside air connection to the air handling unit is sized for 100% outside air for outside air economizer operation. The air handler is often so closely coupled to the outside air louver that the duct is short and the same size as the louver.

(2) On page 64 in [1.14], under the heading: “Outdoor air flow measurement” the method of pitot flow stations is included among the methods of continuous airflow measurement in HVAC systems. It is also mentioned that outdoor airflow measurement is particularly challenging because of the wide range of temperatures encountered in addition to convoluted ductwork and low velocities in oversized ductwork.

In a paper by L. Wu et al. [1.15], the first comprehensive Continuous Commissioning Leading Energy Project (CCLEP) case study in a luxury shopping mall has been investigated. CCLEP has been applied to this building from May 2006 to February 2007. The integration of the dual duct air handling unit, terminal boxes and central plant as a whole system with retrofit and optimal control strategies have taken place in the CCLEP process. It is concluded that, in this building, the measured hourly utility data after CCLEP has indicated that the annual HVAC electricity consumption was reduced by 56%.
In another paper by Y. H. Cho et al. [1.16], the implementation of new innovative technologies in a CCLEP has been discussed. Energy consumption is compared in detail before and after CCLEP. The case study was performed on a five story office building having two single duct VAV air handling units, two types of VAV terminal boxes (with and without reheat coil) and chilled and hot water systems with typical office hours from 8:00 a.m. to 5:00 p.m. during the weekdays. The results of this case study have shown an electricity consumption reduction of 26.8% in a one year period.

In terms of optimizing the supply fan speed control in a VAV system using a fan airflow station (FAS), there is a study which has been made by G. Liu and M. Liu in [1.17]. There are two new control methods that have been considered in [1.17]. The first method in which the supply fan speed is controlled, it is a constant system resistance which is to be maintained. This resistance is calculated based on the measured fan head and airflow which is in turn controlled by the FAS. In the second method, it is still the supply fan speed which is controlled to maintain the duct static pressure set point that is reset based on the airflow ratio measured by the FAS. Both these methods are regarded in [1.17] as innovative and it is mentioned that they can be applied to systems with either direct digital control (DDC) or pneumatic control boxes. It is mentioned in [1.17] that typically the inputs to any of the control devices such as variable-speed control of the fan motor modulating the air flow in a VAV system in order to maintain a duct static pressure set point. It is furthermore suggested in [1.17] that there are case studies which have shown that the new control method can significantly save supply fan power and its efficiency.
In a paper by L. Wu et al. [1.18], it is argued that the air duct static pressure is usually a control variable which is to be maintained by the supply fan for variable air volume (VAV) systems and that the air static pressure is set at a constant set point on design conditions. In the case of partial load conditions, however, the terminal box dampers are closed more because of the fact that the required air flow is less which results in smaller pressure loss. It is argued further in [1.18] that irrespective of whether the terminal boxes are pressure dependent or independent, the static pressure set point should be reset to a lower level so as to reduce the fan energy, the noise created in the terminal boxes and the malfunction of the terminal boxes. The static pressure set point is reset based on the airflow which is measured by the fan airflow station. It is therefore concluded in [1.18] that based on the actual airflow requirements, the reset static pressure schedule can bring about, among other things, significant improvements in controlling the supply fan speed.

1.3 Organization of Thesis

The remaining part of this thesis contains four chapters. The contents of each chapter are structured as follows:

Chapter 2 is concerned with developing the theoretical basis from which the main variables can be determined. Applications of the energy equation including the formulation of expressions for total, major and minor head losses are discussed. The relationships between head loss and friction factor on the one hand and between friction factor and parameters such as flow rate, Reynolds number on the other are also examined.
Chapter 3 focuses on the methodology and procedures in the experimental setup, measurement and data acquisition techniques in the investigation conducted.

Chapter 4 provides an analysis of the results obtained in the course of this investigation.

Chapter 5 provides conclusions and recommendations for future studies. Prospective work is also debated.
Chapter 2  Theoretical Development

The governing equations for the hydraulic approach will be developed in this chapter. Two energy methods for the determination of the losses will be treated in the first two sections. In the next two sections, the effects of friction factor on head loss are first considered and major and minor head losses will be discussed later. In the section before the last, an optimization technique is introduced for the simplification of effects of friction factor on head loss. The last section will be concerned with the relationship between power and head loss.

2.1 Energy Method by Bernoulli Equation

It should be kept in mind that all variations in the ways of operating the AHU2 supply air AC motors are in accordance with the assumptions made earlier, as has been outlined in the introductory section of this thesis. It should be emphasized that some properties of the incoming fresh air should be taken into account in order to achieve as accurate results as possible in this study. Properties such as temperature, relative humidity, density, specific weight, dynamic viscosity, etc., will affect the magnitude of head loss (and consequently power loss) associated with each of the overall performances for different motor combinations. The effects of these properties on head loss will be addressed in the latter parts of this study. It will be shown that a comparison to be made between performance characteristics of the AHU2 supply air AC motors, when grouped together in different motor combinations, will be directly related to a comparison made between their respective head losses.
It has been stated previously in this thesis that, as one of the requirements to be met, AHU2 should constantly produce an airflow rate in the range of 4.25, usually for summer, to 7.08, usually for winter, $\frac{m^3}{s}$ (9000 to 15000 cfm). Therefore, using the upper limit in this range and performing some simple calculations, the highest value for the corresponding average air speed in the AHU2 intake air duct and thereby the associated Mach number can be calculated as follows. Referencing Figure 2.1 (below), these calculations can be made as follows. The average speed of the air moving through the AHU2 intake air duct system and the associated Mach number are

$$U_a = \frac{Q_a}{A}, \quad M = \frac{U_a}{c}$$ \hspace{1cm} (2.1, 2.2)

where

$U_a =$ average speed of air flowing through the intake air duct, $\frac{m}{s}$

$Q_a =$ average air flow rate, $\frac{m^3}{s}$

$A =$ cross sectional area of the intake air duct, $m^2$

$c =$ speed of sound at 0 °C and standard pressure, 331.5 $\frac{m}{s}$

$M =$ Mach number associated with the average air speed

So, for example, having a flow rate of $Q = 7.08 \frac{m^3}{s}$ (15000 cfm), for a duct having a height of $h = 0.71 \text{ m}$ (28 inches), and a length of $w = 1.98 \text{ m}$ (78 inches) with a cross sectional area $A = w \times h = 1.41 m^2$ with an average air velocity of $V = 5 \frac{m}{s}$ (990ft/min), the corresponding Mach number is

$$M = \frac{5}{331.5} = 0.015 << 0.3.$$
As a “rule of thumb” in Young et al. [2.1], a flow of perfect gas may be considered as incompressible provided that the Mach number is less than 0.3. Elsewhere, according to the assumptions made on page 98 in [2.1], and on page 11 in Schlichting [2.2], respectively, this rather low value of $M$ will guarantees that air flow in the AHU2 intake air duct system can be considered as incompressible. However, it can be seen that this type of flow does not occur in a laminar regime. The relatively high value for the average Reynolds number, $R_a$, which results irrespective of the relatively low value of air speed in this case, indicates that the flow is indeed turbulent. Reynolds number can be calculated based on the hydraulic diameter which is associated with the rectangular cross section of AHU2 intake air duct system as follows.

According to the formulation on page 575 in [2.2], the hydraulic diameter can be defined as

$$D_H = \frac{4A}{P_w} \quad (2.3)$$

where

- $D_H =$ hydraulic diameter of duct, m
- $A =$ cross sectional area of duct, $m^2$
- $P_w =$ wetted perimeter of duct, m.

Reynolds number is defined on page 14 in [2.2] as the ratio of the product of the cross sectional area diameter of pipe by the fluid velocity to the kinetic viscosity

$$R_a = \frac{D_H U_a}{\nu_a} \quad (2.4)$$

where

- $R_a =$ average Reynolds number
- $U_a =$ average air velocity, $\frac{m}{s}$
\[ \nu_a = \text{average kinetic viscosity, } \frac{m^2}{s} \]

Elsewhere in Claiborne [2.3], equivalent diameter (4 times hydraulic radius) is given the same definition as that of given for the hydraulic diameter in [2.2]. Leutheusser [2.4] confirms that the extensive literature review compiled in [2.3] (of the experimental studies on turbulent fluid flow in smooth noncircular conduits) shows that the equivalent diameter is indeed the characteristic length parameter of the cross section. It is argued in [2.4] that this is the case even though Claiborne’s diagrams in [2.3] reveal that the function which establishes the relationship between the friction factor and Reynolds number is not unique.

Referencing Figure 2.1 (following the next two pages) and Table 3.2 in Chapter 3, the airflow rates (taken from all of the data measured on Date 6 in a test which was performed on the AHU2 intake air duct system for a 6 motor combination) are as follows:

\[ Q_2 = 8.01 \frac{m^3}{s}, \quad Q_3 = 7.96 \frac{m^3}{s} \quad \text{and} \quad Q_4 = 7.99 \frac{m^3}{s} \]

These values for the flow rates were measured at stations 2, 3 and 4, respectively. Having an AHU2 intake air duct cross sectional area of \( A = 1.4089 \, m^2 \), the average air speed in the duct is simply calculated then to be the ratio of the average flow rate to the cross sectional area, as in the following:

\[ U_a = \frac{Q_2 + Q_3 + Q_4}{3A} \]

\[ = 5.7 \, \frac{m}{s}, \quad (2.5) \]

such that the average temperature at stations 2, 3 and 4 is about \(-2\) °C with an average kinetic viscosity which is about

\[ \nu_a = 1.35 \times 10^{-5} \, \frac{m^2}{s}. \]

From equation (2.3), with the value of the wetted perimeter as
\[ P_w = 2(w + h) = 2(1.98 + 0.71) = 5.38 \, m \] and
\[ D_H = 4 \left( \frac{1.4089}{5.38} \right) = 1.0444 \, m. \]
Reynolds number is then calculated to be
\[ R_a = 1.0444 \times \frac{5.7}{135 \times 10^{-5}} = 4.384 \times 10^5. \]

This rather high value for the average Reynolds number clearly indicates that the flow in the AHU2 intake air duct system is turbulent. An attempt shall now be made to provide an answer to the main question, namely, whether or not at any given time there is a significant loss of power as a mass of fresh air is being forced by the AHU2 supply air AC motors/fans (in operation and at what speed) to move through the intake duct system depending on a particular motor combination in the ‘fan wall’. If so, then it is desired to achieve the accurate results for this power loss. A rather simple comparison between such power losses (each of which is associated with a particular motor combination) may reveal some useful information. This can lead to finding an arrangement or operating scheme in such a way that a number of different motor combinations (at times constantly alternating from one to another and at the same time by switching a number of motors on and off) will satisfy requirements for the airflow rate and the static pressure at all times.
Figure 2.1  Air Handling Unit II Intake Air Duct Systems.
In studying fluid flow through a pipe or a duct, employing useful equations and formulas can greatly assist in the calculations of head loss. It is in [2.1] where such equations and formulas are discussed in detail. The Bernoulli equation is discussed in Chapter 3 while its equivalent forms and a number of examples of its applications are treated in Chapters 5 and 8 of [2.1]. The expression for the extended Bernoulli equation which is sometimes called the mechanical energy equation is also mentioned in Chapter 5 on page 178 [2.1].

In this form, the energy equation for steady incompressible flow between two cross sections of a pipe is given an expression for on page 322 of [2.1]. In order to serve the purposes of this study, the equation chosen to serve as the governing equation in this chapter will be introduced in the following manner. For \( i, j = 2, 3 \) and 4, \( i \neq j \), then

\[
\zeta_i + \frac{p_i}{\gamma} + \frac{(U_j)_a^2}{2g} = \zeta_j + \frac{p_j}{\gamma} + \frac{(U_j)_a^2}{2g} + (L_h)_{ij}
\]

(2.6)

where

\((U_i)_a\) or \((U_j)_a\) = average air velocity at any station of duct, \( i \) or \( j, i \neq j, \frac{m}{s} \)

\(p_i\) or \(p_j\) = average absolute pressure, at any station \( i \) or \( j, i \neq j, \text{Pa} \) \( (= \frac{N}{m^2}) \)

\(\rho\) = density, \( \frac{kg}{m^3} \)

\(\gamma\) = acceleration due to gravity, \( \frac{m}{s^2} \)

\(\gamma\) = specific weight of fluid, \( \frac{N}{m^2} \) \( (= \rho g) \)

\(\zeta_i\) or \(\zeta_j\) = elevation, \( m \)

\((L_h)_{ij}\) = total head loss between stations \( i \) and \( j, i \neq j \) of duct.

This is the same equation which is discussed in both chapters 5 and 8 in [2.1], except that it has been slightly modified here to suit the notational needs for the measurement scheme of this study. In all actual fluid flows, however, since all real fluids
have finite viscosity some energy will be lost in overcoming friction. The loss of energy due to frictional losses can be referred to as the head loss. This is the same as considering if the fluid were to rise in a vertical pipe (or duct in this case), then it will rise to a lower height than predicted by Bernoulli’s equation. The head loss will cause the total pressure to decrease in the direction of the flow. The energy equation for a real fluid flow, however, should take into account some other factors as well. For analysis of fluid flow in real conditions, the effects of viscosity, frictional losses and sudden changes in geometry of flow path should all be taken into consideration. It is with these assumptions and considerations that all of the components in equation (2.6) have been calculated in this study.

Another useful equation to be employed in this study which plays a significant role in determining the magnitude of the value for the total head loss in equation (2.6) is the following equation. The expression for this type of equation is discussed on page 326 in [2.1]. For the notational purposes here again, a slightly modified form of this equation will be considered here. For \( i = 2, 3 \) or \( 4 \), then

\[
(L^h_i)_n = K_L \frac{U_i^2}{2g}
\]  \hspace{1cm} (2.7)

where
\( U_i \) = air velocity at station \( i \) of duct, \( \frac{m}{s} \)
\( K_L \) = loss coefficient
\( (L^h_i)_n \) = minor head loss for a bend (elbow) associated with station \( i \) of duct, \( m \).

The application of this equation is for the determination of minor head loss such as bends in a duct. Having constructed such formulations, as in equations (2.6) and (2.7),
a relatively straight forward Matlab code can now be utilized to calculate the head loss and power loss at any section of the duct between stations 2, 3 or 4 and for any motor combination chosen to operate in AHU2. A sample of such a code can be found in Appendix B as Matlab code I.

2.2 Energy Method by Energy Coefficients

In a problem for the flow of a real fluid from one location to another through a pipe (or a duct, as is the case in this study), the energy equation should also include the following factors:

(1) Frictional resistances to fluid,

(2) Flow losses due to sudden changes in flow path, and

(3) Real velocity distribution.

For this energy method, equation (2.6) will now be considered in a different way. That is to say, the energy equation shall be considered with the average velocity terms having multipliers which are distinctly determined for each of the measuring stations 2, 3 and 4 in the duct. This is based on the assumption, among other things, that the velocity profiles for a flow in a turbulent region may not be the same at different sections of the AHU2 intake air duct system. This is especially true for a region of the AHU2 intake air duct system where there are bends such as in the section confined between stations 2, 3 (see Figure 2.1 in the previous section).

The horizontal distance between the louver where the fresh air first enters the building through the duct and station 4 (which is farthest from the louver as compared with stations 2 and 3) is no longer than 15 meters. It has been suggested that, in a fully
turbulent fluid flow, it may take about 10 to 50 times the (hydraulic) diameter length of a straight (duct) pipe for a complete velocity profile to be developed through the (duct) pipe. The reason for this is that, in an ideal situation, in order to get the most accurate results from an experiment of this kind, it is beneficial to take the appropriate measurements at a measuring station located in a straight stretch of the length of the duct (pipe) and at a distance equivalent to 50 times the (hydraulic) diameter of the duct (pipe) from the entrance of the duct (pipe). Owing to the limitations of the AHU2 duct system, however, the last suitable station (i.e., station 4) is situated as far away from the entrance of the AHU2 intake air duct system which is at a horizontal distance of about 14.5 meters (see Figure 2.1 in Section 2.1 in this chapter). In the problem at hand, however, the hydraulic diameter is determined to be about 1 meter. In this horizontal distance (of less than 15 meters) between the louver and station 4 in the AHU2 intake air duct system, there are two 90 degree bends which interrupt the straight length requirement of a minimum of 10 meters for an air velocity profile of a fully developed fluid flow to form in the duct. All of these expressions will be discussed in Chapter 2 and to be used later to calculate the head losses associated with different motor combinations in this study.

A particular form of this relationship will also be discussed in Chapter 2 and applied later when a comparison between the magnitudes of power loss associated with different motor combinations becomes necessary in order to analyze the optimal performance of the AHU2 supply air AC motors/fans.

For information on the minimum requirements of straight duct length for achieving accuracy in proper measurement techniques in HVAC systems see section 9 in [2.5].
It is therefore on the basis of this type of reasoning that the hydraulic diameter is adopted in this study to serve as the equivalent diameter for the rectangular cross section of the AHU2 intake air duct system. It is for this reason that it may be beneficial to introduce energy coefficients into the energy equation in order to compensate for the lack of a fully developed air velocity profile. In applying the energy equation for a steady turbulent flow, the Coriolis velocity distribution coefficient $\alpha$ is introduced into this equation. This coefficient is referred to, in this thesis, as the energy coefficient. A definition for this coefficient is given in Montes [2.6]. The same definition for this coefficient will be adopted in this section in the form of equations (2.8a) and (2.8b). Since the air velocity at each spatial node in a cross sectional area of the AHU2 intake air duct system may have a significantly different value than that of an adjacent node, it may be more beneficial to calculate these energy coefficients at all of the measuring stations 2, 3 and 4 in the duct. In this way, each energy coefficient can then be multiplied by the second power of the respective mean velocity term which will also be calculated at each of the corresponding measuring stations 2, 3 and 4. Thus a modified version of equation (2.6), for which all of the assumptions and considerations already made in the first method will also apply, can be introduced here such that it includes these energy coefficients in the following manner. For $i, j = 2, 3$ and 4, $i \neq j$, then

$$z_i + \frac{p_i}{\gamma} + \alpha_i \frac{(U_m)_i^2}{2g} = z_j + \frac{p_j}{\gamma} + \alpha_j \frac{(U_m)_j^2}{2g} + (L_h)_{ij}$$

where all the terms in this equation remain the same as in equation (2.6) and $\alpha_i$ or $\alpha_j =$ energy correction factor at any station $i$ or $j$, $i \neq j$,

such that $\alpha_i$ and $\alpha_j$ are defined as follows
\[ \alpha_i = \frac{\int u_i^3 dA}{U_m^3 (A_t)_i} = \frac{\sum_k u_i^3 (k) A_k}{U_m^3 (A_t)_i}, \quad i = 2, 3 \text{ and } 4, \; k = 1, 2, \ldots, N \] (2.8a)

where

\[ (A_t)_i = \sum_i A_i, \quad i = 1, 2, \ldots, N \]

and

\[ \alpha_j = \frac{\int u_j^3 dA}{U_m^3 (A_t)_j} = \frac{\sum_k u_j^3 (k) A_k}{U_m^3 (A_t)_j}, \quad j = 2, 3 \text{ and } 4, \; k = 1, 2, \ldots, N \] (2.8b)

where

\[ (A_t)_j = \sum_j A_j, \quad j = 1, 2, \ldots, N \]

The total head (energy) loss, \((L_h)_{ij}\), between stations \(i\) and \(j\), based on the mean velocity \(U_m\) can therefore be calculated as follows. For \(i, j = 2, 3\) and \(4, i \neq j\), then

\[ (L_h)_{ij} = \left[ z_i + \frac{p_i}{(\gamma_{ij})_m} + \frac{\alpha_i (U_m)_i^2}{2g} \right] - \left[ z_j + \frac{p_j}{(\gamma_{ij})_m} + \frac{\alpha_j (U_m)_j^2}{2g} \right] \] (2.9)

where

\( p_i = \) average centerline static pressure at station \(i\),

\( p_j = \) average centerline static pressure at station \(j\),

\( z_i = \) elevation at station \(i\),

\( z_j = \) elevation at station \(j\),

\( U_i = \) grid velocity at station \(i\),

\( U_j = \) grid velocity at station \(j\),

\((U_m)_i = \) mean velocity at station \(i\),

\((U_m)_j = \) mean velocity at station \(j\),

\((\gamma_{ij})_m = \) specific weight of fluid averaged at midpoint between stations \(i\) and \(j\),

\((L_h)_{ij} = \) total head loss between stations \(i\) and \(j\), \(i \neq j\), of duct.
Expressions (2.8a) and (2.8b) for the energy coefficient $\alpha_i$ or $\alpha_j$ developed in the above are based on a relation that is defined on page 26 in [2.6]. The values for $\alpha_i$ or $\alpha_j$ and the mean velocity, $(U_m)_i$ or $(U_m)_j$, can easily be determined by making use of the sample Matlab code II in Appendix C. This sample code is a specific instance of a general Matlab code devised for any suitable motor combination including its associated data which were collected in the course of the investigation in this study.

The following numerical values for the energy coefficients have been obtained by setting up the instruments for experimentation at each measuring stations 2, 3 and 4 for two different motor combinations selected from all of the AHU2 supply air AC motors.

1. In the case of a 6 motor combination where these motors were set to operate at the speed of 45 Hz on Date 6, the following energy coefficients were obtained by applying the appropriate measurement data in the sample Matlab code II:

$$\alpha_2 = 1.008, \; \alpha_3 = 1.025, \; \text{and} \; \alpha_4 = 1.022.$$  

2. While in the case of an 8 motor combination, all of these motors were set to operate at 40.8 Hz on Date 5, using the appropriate measurement data in a Matlab code (similar to the sample Matlab code II), resulted in the following coefficients:

$$\alpha_2 = 1.005, \; \alpha_3 = 1.027, \; \text{and} \; \alpha_4 = 1.023.$$  

2.3 Friction Factor on Head Loss

Assuming that the (hydraulic) diameter of the duct is constant, implying that $D_i = D_j$ so that $U_i = U_j$ for a section of a duct between stations $i$ and $j$, $i \neq j$, having a horizontal length of $(l_{ij})$ with $(z_i = z_j)$, then equation (2.6) will be reduced to the following equation. For $i, j = 2, 3$ and 4, $i \neq j$, then
\begin{equation}
(L_{ij}^h)_m = \frac{f_{ij}l_{ij}}{2D_H g} U_a^2
\end{equation}

This equation (with the exception of having the hydraulic diameter $D_H$ substituted for a pipe diameter $D$) is the so called the Darcy-Weisbach equation. It is developed on page 322 in [2.1] for the calculation of major head. An expression for the Colebrook formula can be found on page 324 in [2.1] in the following form. For $i, j = 2, 3$ and $4$, $i \neq j$, then

\begin{equation}
\frac{1}{\sqrt{f_{ij}}} = -2 \log \left( \frac{\varepsilon}{3.7 D_H} + \frac{2.51}{R_{ij} \sqrt{f_{ij}}} \right)
\end{equation}

where

$f_{ij} =$ friction factor for a section of a duct between stations $i$ and $j$, $i \neq j$

$l_{ij} =$ length of a section of a duct between stations $i$ and $j$, $i \neq j$, $m$

$U_a =$ average velocity of fluid, $\frac{m}{s}$

$D_H =$ hydraulic diameter of duct, $m$

$g =$ acceleration due to gravity, $\frac{m}{s^2}$

$\varepsilon =$ equivalent roughness of duct, $m$

$R_{ij} =$ Reynolds number for a section of a duct between stations $i$ and $j$, $i \neq j$

$\nu =$ kinetic viscosity, $\frac{m^2}{s}$

$(L_{ij}^h)_m =$ major head loss for frictional resistances between stations $i$ and $j$, $m$

In this section, a particular formula which relates the Reynolds number to the friction factor has been employed to determine the friction factor. As a consequence of this determination, major head loss can be obtained by considering a unique form of the energy equation, that is, the so called Darcy-Weisbach equation. Similar derivations for
the Bernoulli equation, Darcy-Weisbach equation and Colebrook formulas have been treated in Chapter 35 Duct Design in [2.7]. It is safe to assume that the material used to construct the AHU2 intake air duct system is made of commercial steel. Table 8.1 on page 324 in [2.1] gives the value of the relative equivalent roughness for this material as \( \varepsilon = 0.045 \times 10^{-3} \) m. Referencing Figure 2.1, in Section 2.1 of this chapter, it is assumed that the duct has two 90\(^\circ\) miter type bends with guide vanes, in the length which is confined between stations 2 and 3. Therefore, value for the loss coefficient for this type of bend is found to be \( K_L = 0.2 \) which is mentioned at the bottom of Figure 8.17 (b) on page 330 in [2.1]. There are particular sections in Matlab codes I or II, in Appendices B and C, respectively, which can be used to perform the appropriate calculations for each of the energy methods discussed here. These sections are given in the following steps:

1. The values of all parameters that are involved in equations (2.1), (2.3) and (2.4) are all given or have been measured during the experimental phase (of the investigation in this study) at each of the measuring stations 2, 3 and 4 in the duct. The values for all of these parameters are inserted in the initial section of either of these two Matlab codes.

2. In the next section of these Matlab codes, calculation for the unknowns in equations (2.1), (2.3) and (2.4), namely, air velocity, hydraulic diameter and Reynolds number can easily be made in the same manner as was discussed in Section 2 of this chapter. There is an additional section, in Matlab code II, which is made exclusively for calculating the energy coefficients using the appropriate data acquired at each of the measuring stations 2, 3 and 4.
(3) Using equation (2.5), then the value of average velocity $U_a$ along with the value for the average Reynolds number can be calculated using the average values for the velocity and kinetic viscosity in the section following step (2).

(4) Since both relative roughness $\varepsilon$ and hydraulic diameter $D_H$ are constant, then according to the Colebrook formula (i.e., equation (2.11)) friction factor $f$ depends only on Reynolds number $R$. Assuming that the effects of the kinetic viscosity are negligible, Reynolds number $R$ itself will depend only on the variations of the average air velocity $U_a$. So, it can be said that, in a duct with a constant cross section, friction factor is dependent on the variations of average air velocity or on the average airflow rate $Q_a$. Finally, in equation (2.11) all parameters have already been determined except for the value of the friction factor $f$, an initial assumption can be made for $f$ and then an improved value for it can be determined through a trial and error procedure. By using an iterative process (using the equation that is provided in the special Matlab code provided in Appendix D for determining the value of friction factor without resorting to the trial and error procedure) a unique value for $f$ can be determined at once. Substituting this newly found value for the friction factor in equation (2.10), the value for major head loss can then be determined. Additionally, using equation (2.7), minor head loss can be determined in the section following step (3).

(5) This process can be repeated by utilizing the measurements data in this study to determine the corresponding friction factor and major head loss for any motor combination. Total head loss and power loss are all determined using equations (2.6) or (2.9) or (2.12) and (2.22) or (2.23), respectively, in the last section.
2.4 Major and Minor Head Losses

Another way of giving expression to total head loss is to consider its relationship to major and minor head losses. As such, total head loss can be regarded as the sum of major and minor head losses. An expression of this kind is discussed in section 8.4 entitled “Dimensional Analysis of Pipe Flow” on page 321 in [2.1]. In this study for notational purposes, however, a modified version of this expression will be considered in the following form. For $i, j = 2, 3$ and $4, i \neq j$, then

$$\left( L_h \right)_{ij} = (L_{ij}^h)^m + (L_i^h)^n + (L_j^h)^n \tag{2.12}$$

where

$(L_{ij}^h)^m = $ major head loss for frictional resistances between stations $i$ and $j$, $m$

$(L_i^h)^n = $ minor head loss for a bend (elbow) associated with station $i$ of duct, $m$

$(L_j^h)^n = $ minor head loss for a bend (elbow) associated with station $j$ of duct, $m$

$(L_h)_{ij} = $ total head loss between stations $i$ and $j$, $i \neq j$ of duct, $m$.

Equation (2.6) can be rearranged to solve for the total head loss, $(L_h)_{ij}$, as follows:

$$\left( L_h \right)_{ij} = (z_i + \frac{p_i}{\gamma_{ij}} + \frac{U_i^2}{2g}) - (z_j + \frac{p_j}{\gamma_{ij}} + \frac{U_j^2}{2g}) \tag{2.13}$$

In this study a number of assumptions and considerations have been made so that the results achieved by employing equations (2.6) or (2.13) and (2.7) can lead to meaningful conclusions. These assumptions are as follows:

1. Since AHU2 is not located at a high elevation, then the effects of the atmospheric pressure on both sides of equations (2.6) or (2.9) and (2.13) can be ignored.
(2) The effects of the relative humidity on the calculations for head loss and the consequential power loss for each motor combination are also ignored.

(3) Furthermore, the variations of air density from one station to another (among stations 2, 3 and 4) are assumed to be negligible as well.

(4) Aside the measurements which were made for determining the airflow rate and the static pressure at each of the measuring stations 2, 3 and 4 in the AHU2 intake air duct system, the only other measurements made were of the air temperature and the relative height of these stations (with respect to each other).

(5) When calculating values for head losses (and the corresponding power losses) at each of the respective measuring stations, the data collected from all of the measurements have all been collectively taken into account at the same time.

(6) Either of equations (2.6) and (2.9) can be modified in such a way so as to solve for the total head loss by keeping it on the left side of the equation while collecting all other terms on the right hand side of the equation, as in the form of equation (2.13).
2.5 Simplification of Effects on Head Loss

The effects of the friction factor on the minor loss can be assumed to be negligible. This means that the variations of air velocity $U_i$, at any section between stations $i = 2, 3$ and $4$, in the duct and their effects on the value of the corresponding Reynolds number and consequently on the friction factor itself which is calculated by using equation (2.11), will all be assumed to be negligible. In this way, the last 2 terms in equation (2.13) can then be set to equal to a constant in the following way. For $i, j = 2, 3$ and $4$, $i \neq j$,

$$(L_i^h)_n + (L_j^h)_n = C \quad (2.14)$$

where $C$ is a constant. The left hand side of equation (12) can be called $Y$ and by letting

$$(L_{ij}^h)_m = X \text{, for } i \text{ and } j = 2, 3 \text{ and } 4, i \neq j \quad (2.15)$$

Equation (2.12) can then be expressed in the following way

$$Y = X + C \quad (2.16)$$

It is now desired to linearly approximate the curve which is generated (by plotting the different values of $Y$ against the corresponding values of $X$) from equation (2.16) by a line with an equation which is described as follows

$$y = ax + b \quad (2.17)$$

such that $a$ and $b$ are both assumed to be arbitrary constants. The aim of this section is to see if equation (2.16) can be approximated by equation (2.17) by employing the method of “least mean square”.

In order to accomplish this task, an attempt shall now be made to perform a minimization procedure as follows. For every value of the friction factor, there is a point
on the curve for the plot of $Y$ against $X$ (it is assumed that this plot will generate a curve) which can indicate how $Y$ varies with respect to $X$. Thus, for $i, j = 1, 2$ and $3$, $i \neq j$, let the following expression hold

$$\Phi = \text{minimize} \left[ \sum_i (\bar{y}_i)^2 \right], \quad i = 1, 2 \text{ and } 3 \quad (2.18)$$

such that

$$\bar{y}_i = y_i - Y_i = ax_i + b - Y_i, \quad x_i = X_i, \quad i = 1, 2 \text{ and } 3.$$ 

Then, the coefficients $a$ and $b$ can then be determined, only if both of the following conditions hold

$$\frac{\partial \Phi}{\partial a} = 0 \quad \text{and} \quad \frac{\partial \Phi}{\partial b} = 0 \quad (2.19)$$

$$\frac{\partial^2 \Phi}{\partial a^2} > 0 \quad \text{and} \quad \frac{\partial^2 \Phi}{\partial b^2} > 0 \quad (2.20)$$

Conditions in (2.20) hold since $2(x_2^2 + x_3^2 + x_4^2) > 0$ and $2 > 0$ are both positive numbers. Thus, a system of equations for the solution of $a$ and $b$, in matrix form, is specifically as follows

$$\begin{bmatrix} 2(x_2^2 + x_3^2 + x_4^2) & 2(X_2 + X_3 + X_4) \\ 2(X_2 + X_3 + X_4) & 6 \end{bmatrix} \begin{bmatrix} a \\ b \end{bmatrix} = \begin{bmatrix} 2(X_2Y_2 + X_3Y_3 + X_4Y_4) \\ 2(Y_2 + Y_3 + Y_4) \end{bmatrix} \quad (2.21)$$

A new value for the friction factor can be found using the following steps.

1. Using the coefficients $a$ and $b$ in conjunction with the original value for $X = x$ in equation (2.17) will then produce a new value for $y$.

2. Equation (2.17) is constructed in such a way so that it can describe a line approximating the curve which is generated by plotting $X$ against $Y$ in equation (2.16). The new value of $y$ obtained in step 1 shall now (serve as) or represent a new value for $Y$ in equation (2.16).
(3) Substituting this newly found value of $Y$ in equation (2.16) will produce a new value for $X$. The new value of $X$, being equivalent to $(hLij)_m$ in equation (2.15), will in turn produce a new value of the friction factor using equation (2.10).

The procedure for the determination of a new set of parameters such as major head loss and total head loss including $a$ and $b$ can be repeated as many times as needed as in the following manner.

(1) The only unknowns in equation (2.11) are the friction factor and the Reynolds number, as the relative roughness of the duct and the hydraulic diameter both are constants. Using the new value of the friction factor, obtained from the 3 step scheme in the above in equation (2.11), a new value for the Reynolds number will now be determined.

(2) Kinetic viscosity is assumed to have an average value at the midpoint between two stations forming a section of the duct. This average value can be interpolated from its value at a reference temperature such as 0 °C. Using equation (2.11), a new value for Reynolds number can then be calculated. From the new value of Reynolds number, new values for the (average) air velocity and flow rate can then be determined by using equations (2.4) and (2.1), respectively. The new values of all the necessary parameters can be plugged into equation (2.10) to determine the newest value for major head loss $X$.

In this way, after a certain number of iterations, the linear approximation of the curve representing the plot of total head loss against major head loss will be achieved. Some numerical results for this linearization procedure will be discussed in the last part of chapter 4 in this paper.
2.6 Relationship between Power Head and Flow

In the case of a pipeline (or a duct system, as in this study) transmitting power, relatively simple expressions of Power available at the outlet and Power supplied at inlet each of which are based on the specific weight of the fluid, discharge (flow rate), the head at the outlet and the gross head, respectively, are given found in Douglas [2.8]. This useful relationship will be employed here as well. In order to equate the loss in power with the head loss that is associated with the fluid flow through a pipe (duct), For the notational purposes in this paper and for the sole consideration of the power lost in the transmission of the fluid, a modified version of either expression to fit the expression in mind will be considered as follows. For \( i, j = 2, 3 \) and 4, \( i \neq j \), then

\[
(L_p^a)_{ij} = (Y_{ij})_m \times Q_a \times (L_h^a)_{ij} \quad (2.22)
\]

\[
(L_p)_{ij} = (Y_{ij})_m \times (Q_{ij})_m \times (L_h)_{ij} \quad (2.23)
\]

where

\[
(Y_{ij})_m = \text{specific weight of fluid calculated at the middle of stations } i \text{ and } j, \frac{N}{m^2}
\]

\[
Q_a = \text{average airflow rate calculated for all stations } i \text{ and } j, \frac{m^3}{s}
\]

\[
(Q_{ij})_m = \text{airflow rate calculated at the middle of stations } i \text{ and } j, \frac{m^3}{s}
\]

\[
(L_h^a)_{ij} = \text{total head loss, based on } Q_a \text{ for any pair of stations } i \text{ and } j, i \neq j \text{ of duct, } m
\]

\[
(L_h)_{ij} = \text{total head loss between stations } i \text{ and } j, i \neq j \text{ of duct, } m
\]

\[
(L_p^a)_{ij} = \text{total power based on } (L_h^a)_{ij} \text{ for any pair of stations } i \text{ and } j, i \neq j \text{ of duct, } kW
\]

\[
(L_p)_{ij} = \text{total power based on } (L_h)_{ij} \text{ for any pair of stations } i \text{ and } j, i \neq j \text{ of duct, } kW.
\]
Chapter 3  Experimental Equipment and Measurements

In this chapter, the procedural aspects of measurement taking, which are all based on the hydraulic approach, for the variables of interest namely static pressure and air velocity will be discussed. Different techniques in data acquisition for each of these two variables will be described in detail. Samples of the recorded data collected in the course of the investigation in the form of Excel sheets and graphs that reflect the details of the experimental conditions under which such data were collected will also be discussed.

3.1  Static Pressure Experimental Setup and Measurements

The complete cross section of the intake air duct system of AHU2 is shown in Figure 3.1 in the next page. It is subdivided in such a way so as to create a grid as indicated in this figure. A suitable measuring instrument for collecting experimental data such as the static pressure and the speed of the air flowing in this duct consists of a Shortridge instrument with a digital display (see Figure 3.3 (b) and a pitot tube. As it is clearly shown on Figure 3.1, the center for each of the subareas a1 through a65 is where the tip of the pitot tube must be properly positioned opposite the direction of airflow in the duct when such measurements are taken at each of the measuring stations 2, 3 and 4. The tip of the pitot tube should be facing (i.e., positioned in the opposite direction of) the flow in the AHU2 intake air duct system in such a way that the most accurate measurement of air flow can be achieved in as a short amount of time as possible.
Figure 3.1  Air Handler Unit II Intake Air Duct System Cross Section.

21 Gauge thickness = 0.0366 (TYP.)

ho = 28

wo = 78

NOT IN SCALE
UNLESS OTHERWISE STATED
ALL DIMENSIONS ARE IN INCHES
By selecting proper settings on the digital display of the Shortridge instrument, readings made for both static pressure and speed of air flowing in the duct by this instrument can be recorded and saved in an Excel sheet as the measured and the collected data to be analyzed, at a later time, in the calculation of useful hydraulic parameters such as head loss. Figure 2.1 shows that the AHU2 intake air duct system is situated in such a way that the fresh air entering the duct from the west side of the building through the louver (i.e., the entrance structure for the fresh air from outside to get into the duct inside the building) is moving in the easterly direction.

Referencing Figure 3.1, at the measuring stations 2, 3 and 4 (each of which is also a cross section of the duct) the measurements for static pressure can be performed as follows:

1. Having adjusted the instrument settings to measure differential pressure (to measure static pressure) and properly attaching the connecting hose to the instrument and the pitot tube, starting at the number 7 hole which has been already drilled out at the bottom side of the duct, but may still be covered by a red plug as shown in Picture 9 in this chapter. Then, unplug this hole, if necessary, to send the pitot tube up vertically to align the tip of the pitot tube at the center of each of the subareas a31 through a35. In this way, the operator of this instrument will be positioning the tip of the pitot tube to be parallel to the air flow streamlines and facing east which is to be opposite of the direction of the flow. See also Figure 3.1 to take notice of the fact that in the grid column 7 includes the subareas a31 through a35.
(2) Then, starting with the subarea a31, the pitot tube which has been already sent through number 7 hole at the bottom of the duct so that it is moving upwards (i.e., vertically) in the duct until its tip can be properly positioned, as closely as possible, at the center of the designed subarea a31 in the grid. At this point in time a measurement of the static pressure can be taken by the instrument.

(3) The tip of the pitot tube will again be moved upward and positioned in the same manner as in (2) to be at the center of each of the remaining subareas from a32 to a35, for the measurements be taken again one at a time, respectively, until all of the 5 static pressure measurements at are recorded, averaged and saved by the Shortridge instrument.

Finally, the average value of all 5 static pressure readings (i.e., the average value of all static pressure readings taken at subareas a31, a32, a33, a34 and a35) along with the value of the middle subarea (i.e., subarea a33, if needed) will be recorded and saved on an Excel sheet for analysis at a later time. Once the first set of 5 measurements at the first station (i.e., station 2) have been taken, recorded and stored, similar measurements can then be taken for the second time or as many times as needed in order to ascertain that the most accurate measurements at all stations 2, 3 and 4 have been obtained in the experimental setup. All such measurements should be taken, recorded and saved in the same manner for all stations 2, 3 and 4 in as shortest amount of time as possible.

3.2 Air Velocity Experimental Setup and Measurements

Applying the same measurement techniques described in Section 3.1, measuring the air speed can be performed in a similar manner as for the static pressure. However,
each measurement of air velocity in the duct, at each of the measuring stations 2, 3 and 4, is to be taken in the same manner as in the static pressure measurements were taken except that in this case the starting point for the air velocity measurements is the hole which is at the furthest left drilled out hole at the bottom of the duct which is associated with the subarea a1. Thus, the air velocity measurements consist of the following steps.

(1) Having changed the instrument settings to pitot tube (to measure air velocity) and properly attaching the connecting hoses to the instrument and the pitot tube, the pitot tube is then sent up into the duct through the number 1 hole with its tip moving upward (i.e., vertically) to be positioned as closely as possible at the center of subarea a1 so as to take the most accurate first air velocity measurement (the tip of pitot tube is then to move from the center of subarea a1 to the center of the next subarea (i.e., subarea a2) to take the second measurement and so on and so forth until measurements at the center of subareas a1 through a5 are all recorded.

(2) The first set of measurements consisting of 5 measurements should be taken at the centers of a1 through a5, respectively, which are all in column 1 of the grid in Figure 3.1. Then the pitot tube and the instrument are moved to the adjacent column (i.e., column 2) and the proper measurements are taken for this column in the same manner as in column 1. Other sets of measurement will then be completed for columns 3, 4, through 13 of the grid.

(3) Taking the average value of all 65 (= 5 × 13) measurement readings (i.e., the average value of each measurement reading is taken at subareas a1, a2, a3,
. . ., a63, a64 and a65), all measurement values will be recorded and saved on an Excel sheet for analysis at a later time.

These measurements can be performed continuously at each station until all necessary measurements for both static pressure and air velocity have been completely finished. All such measurements will then be recorded and saved in the same manner for each of the steps (1) through (3) in Sections 3.1 and 3.2 in this Chapter and for all of the measuring stations 2, 3 and 4 at the duct one after another in the same sequence, but always starting from station 2 then moving to station 3 and ending at station 4 to complete a full set of measurements before starting the next set of measurements.

All of the figures immediately following this page illustrate some images of parts of the equipment upon which the investigation was conducted in this study, some of the instruments which were utilized in the measurements of the variables and some of the instrumentation setup for the experimental procedure as described in the above. They are Figures 3.2 through 3.4.
(a) An image of a total of 8 variable frequency drive devices assigned to control the Air Handling Unit 2 supply air AC motor.

(b) An image a strobe light instrument digital display pointing toward the fan wheel rivets (see also Figure 3.4 (b)) to verify the supply air AC motor/fan speed.

Figure 3.2  (a) Variable Frequency Drives, (b) Strobe Light Instrument.
(a) An image of the digital display of a variable frequency drive device showing 2115 rpm as the speed for an AHU2 supply air AC motor.

(b) An image of the Shortridge measuring instrument utilized in this investigation.

Figure 3.3  (a) Variable Frequency Drive Digital Display, (b) Shortridge Instrument.
(a) An image of a temperature sensor probing the AHU2 intake air duct system at a typical measuring station through a hole (drilled for a column of spatial nodes).

(b) An image of 3 typical AHU2 supply air AC motor/fan wheels.

Figure 3.4  (a) Temperature Sensor Instrument Digital Display, (b) Air Handling Unit II Supply Air AC Motor/Fan Wheels.
In this section, two samples of the measurements data collected at station 2 at Site A on Date 5 and Date 6 will be introduced. These samples contain a variety of VFD digital display readings and measurements taken by all of the instruments utilized in the course of the investigation in this study. These sample data cover only the cases for the 8 and the 6 AC motor combinations, respectively. Each of these sample measurements data have been compiled in the form of Excel sheets. They include the corresponding air velocity profile graphs detailing the experimental conditions in addition to a record of the collected data for measuring certain variables of interest in this study. Such variables are the static pressure, air velocity/flow, air temperature at each of the measuring stations 2, 3 and 4 and the average speed, torque and the electrical energy consumption of the AHU2 supply air AC motors while in operation for each of the motor combinations examined.

Figure 3.5 (below) is a copy of an image of a particular section of an existing file in JEMS. This image is introduced here to show a snapshot of a ‘session’ (an image of which has been generated on a PC screen) of this software. From this snapshot, one can decipher exactly the same readings of the motor data as can be provided by selecting the appropriate settings from the VFD digital display (see Figure 3.3 (a)). The example of the motor data being referenced here includes, among other things, the instantaneous electrical energy consumption in kW for each of the AHU2 supply air AC motors as they were being tested on Date 5. This image should also serve as an example of a picture which depicts only a particular part of a rather elaborate procedure in which for instance the testing of an 8 motor combination operating took place while all of the measurement data corresponding to this 8 motor combination were being collected on that occasion. Located below Figure 3.5, are the aforementioned two samples of the measurements data.
Figure 3.5 Image of a Session in JEMS.
Table 3.1 Data Collected at Station 2 on Site A at Date 5.

<table>
<thead>
<tr>
<th>Measurement Data collected at station 2 on Site A at Date 5</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Notes</strong></td>
</tr>
<tr>
<td><strong>Station</strong></td>
</tr>
<tr>
<td><strong>Date</strong></td>
</tr>
<tr>
<td><strong>PM</strong></td>
</tr>
<tr>
<td><strong>SR</strong></td>
</tr>
<tr>
<td><strong>Temperature at 20°C</strong></td>
</tr>
<tr>
<td><strong>Temperature at the station</strong></td>
</tr>
<tr>
<td><strong>Airflow rate</strong></td>
</tr>
<tr>
<td><strong>value of airflow rate is measured by the Vortek instrument @ an airflow station in the intake duct designated for this purpose.</strong>*</td>
</tr>
</tbody>
</table>

**Notes**:
- Station 4 is located approx. 500 m downstream of station 2. Station 3 is located approx. 200 m upstream of station 4. Notes were all turned on.
- Airflow rate measurements taken at station 2. 3 and 4 one after another in this sequence.
Figure 3.6  Air Velocity Profiles at Station 2 on Site A at Date 5.
Table 3.2  Data Collected at Station 2 on Site A at Date 6.

Measurement data collected at station 2
*** on Site A at Date 6

<table>
<thead>
<tr>
<th>Duct Dimensions</th>
<th>Measured Data</th>
<th>*</th>
<th>VFD supply fan data</th>
<th>phase</th>
<th>amps</th>
<th>volts</th>
<th>hp</th>
</tr>
</thead>
<tbody>
<tr>
<td>Width</td>
<td>1.98</td>
<td>6.5 (ft)</td>
<td>Air velocity, m/s</td>
<td>5.75</td>
<td>1132 (fpm)</td>
<td>487.0</td>
<td>17,200 (cfm)</td>
</tr>
<tr>
<td>Height</td>
<td>0.71</td>
<td>2.3 (ft)</td>
<td>Flow rate, m³/s</td>
<td>8.1</td>
<td>17169 (cfm)</td>
<td><strong>SP</strong></td>
<td>0.05 m</td>
</tr>
<tr>
<td>Area</td>
<td>1.41</td>
<td>15.2 (ft²)</td>
<td>Mass rate</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Total air velocity (fpm) 73.593 Notes Motor load = 75%, Torque = 7.5 Nm
Total number of nodes 65 Center line -0.0028 m Dist. Bet. adj. inner nodes, horizontal, m 0.15 6 (inches)
Average air velocity (fpm) 1132 static pressure -0.1098 (inw) Dist. Bet. adj. inner nodes, vertical, m 0.15 6 (inches)

Notes
Fresh Air Relative Humidity 38%, @ Station 29.0%
Station 4 is located approx. 8.00 m downstream of station 2.
Station 3 is located approx. 2.05 m upstream of station 4.
Supply air humidity 100%
Airflow rate measurements taken at stations 2, 3 and 4 one after another in this sequence.

<table>
<thead>
<tr>
<th>Motors</th>
<th>#</th>
<th>2 and 8</th>
<th>Ti</th>
<th>Tf</th>
<th>Quality</th>
<th>Cold</th>
<th>Clear</th>
<th>Calm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Date</td>
<td>02 26 2012</td>
<td>Operator Density at 20 °C</td>
<td>1.21 kg/m³</td>
<td>0.075 (lb/ft³)</td>
<td>Outside Air Temp., °C</td>
<td>8.3</td>
<td>47.0</td>
<td>7.6</td>
</tr>
<tr>
<td>Time</td>
<td>18:30 PM</td>
<td>KR</td>
<td>Temp, at the station, °C</td>
<td>9.3</td>
<td>48.7</td>
<td>8.8</td>
<td>47.9 (deg. F)</td>
<td>number</td>
</tr>
</tbody>
</table>

ALL AVAILABLE VAV BOXES ARE IN 100% OPEN POSITION.

* VALUE OF AIRFLOW RATE IS MEASURED BY THE VORTEK INSTRUMENT @ AN AIR FLOW STATION IN THE INTAKE AIR DUCT DESIGNATED FOR THIS PURPOSE.

** THIS VALUE OF STATIC PRESSURE IS CONSTANTLY MAINTAINED.

*** SI UNITS APPLY, UNLESS OTHERWISE STATED.
Figure 3.7  Air Velocity Profiles at Station 2 on Site A at Date 6.
Each of the two samples of the measurement data collected in a tabulated form that are designated here as Tables 3.1 and 3.2, respectively are typical of the method used for the data collection during the investigation in this study. They are entitled: “Data Collected at Station 2” on Site A. Tables 3.1 and 3.2 contain the measurements data collected on Dates 5 and 6, respectively. The corresponding air velocity profiles immediately following each of these two tables are designated as Figures 3.6 and 3.7, respectively. Some useful observations can immediately be made about these data.

(1) Among the data recorded in Table 3.1, which contains all of the measurements data collected for the 8 motor combination tested on Date 5, are as follows. The flow rate values measured at station 2 in the AHU2 intake air duct was about \(7.8 \frac{m^3}{s}\) (16520 cfm) with the average fresh air temperature reading of about 0.1 °C at this station such that the VFD reading for the number 4 motor of the instantaneous electrical energy consumption was 1.72 kW with a motor loading of 68% percent and a torque rating of 6.3 N.m. This value of electrical energy consumption can be assumed to represent the value of all readings for the 8 motors operating at fixed speed of 40.8 Hz (2448 rpm).

(2) Table 3.2 indicates that, on Date 6, when 6 motors operated at 45Hz (2700 rpm), the flow rate measured at station 2 in the AHU2 intake air duct was about \(8.11 \frac{m^3}{s}\) (17172 cfm) with an average fresh air temperature reading of about 9 °C at this station such that the VFD reading for the number 4 motor of the instantaneous electrical energy consumption was 2.25 kW with a motor loading of 75% percent and a torque rating of 7.5 N.m. This value of
electrical energy consumption can be assumed to represent the value of all readings for the 6 motors operating at fixed speed of 45 Hz (2700 rpm).

A simple calculation, however, shows that according to the data recorded in these tables the following considerations can be made.

(1) In the case of 8 motors operating at 40.8 Hz, the total electrical energy consumed is about \[8 \times 1.72 = 13.56 \text{ kW},\] and

(2) In the case of 6 motors operating at 45.0 Hz, the total electrical energy consumed is about \[6 \times 2.25 = 13.50 \text{ kW}.\]

It is therefore safe to assume that, under similar testing conditions, it has been revealed that in the 6 motor combination operating at a speed higher than 40.8 Hz have altogether consumed less electrical energy. Furthermore, in the case of the 6 motor combination, the 6 motors have collectively moved a larger volume of fresh air into the building than that of the 8 motor combination. Additionally, it should be noted that under the current operating conditions, all 8 supply air AC motors in the AHU2 are constantly running at speeds within the range of (34 to 39) Hz 24 hours a day.

Observations made over an extended period of time have also revealed that the operating speeds of all of these 8 motors have always stayed within this frequency range. This has always been the case even though the speed of these motors are constantly fluctuating and are from time to time increasing or decreasing in order to accommodate different demands put upon AHU2 for providing its share of the supply of fresh air in the UNL/JH building. From time to time, the operating conditions of these motors have also been verified to be consistently the same by choosing the appropriate settings to check
the speed, energy consumption and other parameters from each of the digital display for the VFD devices that have been individually assigned to each of these 8 AC motors.

Figure 3.8  Velocity Distributions of a Fully Developed Flow in a Pipe (Ghosh, et al. [3.1]), (a) Laminar and (b) Turbulent.

Figures 3.6 and 3.7 indicate that, in each case, the air velocity profiles generated at station 2 exhibit patterns resembling a mixture of peaks and valleys in a turbulent flow. A copy of “Figure 5” from page 2 of a paper by S. Ghosh, et al. in [3.1] is reprinted here as Figure 3.8. This figure is a representation a fully developed, axisymmetric pipe flow at a radial distance $r$ from the pipe centerline. It is argued in [3.1] that the axial velocity $u = u(r)$ is independent of the direction in which $r$ is measured. This figure, however, shows that the shape of the velocity profile is different for laminar and turbulent flows.

It is, therefore, safe to assume that a simple comparison made between the velocity profile for the turbulent flow in Figure 3.8 and the respective velocity profile in Figures 3.6 and 3.7 will show that the airflow in the AHU2 intake air duct system, though
steady, is not uniform. That is to say that unlike the case of an ideal fluid in a fully developed flow, under the testing conditions in this investigation, a nearly perfect velocity profile cannot be expected to develop in the AHU2 intake air duct system. The safeness of this assumption is reinforced by considering the fact that the AHU2 intake air duct system is constructed in such way (see Figure 2.1) that its straight length is interrupted by the two bends between stations 2 and 3 and one bend beyond station 4 before the duct merges with the entrance of AHU2 itself.

As has been previously mentioned in Chapter 2, Section 2.2, in either case of the 8 or the 6 motor combinations tested on Date 5 and Date 6, respectively, the calculated values for the energy coefficients at the measuring station 2 were found to be nearly unity. Such is not the case, however, at the measuring stations 3 and 4. All of these results point to the fact that at the measuring station 2, in either case of these two motor combinations tested, the air velocity profile, even though not uniform, is nevertheless not considered to be drastically different from a velocity profile which is associated with a fully developed flow. The values for energy coefficients at measuring stations 3 and 4, however, are not sufficiently close to unity. Therefore, it may very well be the case that at these two stations the airflow is both non uniform and far from being fully developed.

Referencing Figures 3.6 and 3.7, it can be noted that as fresh air is being transmitted through the AHU2 intake air duct system, once numerical values of the energy coefficients are determined at each of the designated measuring stations 2, 3 and 4, the numerical results for head loss and consequently the loss of power can then be calculated by considering any permissible combination of the AC motors operating in the AHU2 ‘fan wall’. It should also be noted that the calculations made for head loss and the
associated power loss can be based on either equations (2.6) or (2.9) depending on which energy method is to be considered for obtaining the desired results based on the data which have been gathered and recorded in this investigation.

According to the supply air AC motor/fan specifications and performance data sheets for AHU2 which is supplied by the manufacturer of the ‘fan wall’ technology, HUNTAIR, Inc., a total air flow rate of 27,000 cfm and a total of 20.32 cm (8.0 inches) height of water column of static pressure can be produced when all of these 8 motors are running at a speed of 57.5 Hz (3450 rpm) for a combined 60 hp (= 8 × 7.5 hp per motor). The numerical values obtained from these specifications and performance data sheets indicate that these AC motors can achieve their peak efficiencies only at higher speeds which are well above the speeds (in the range of 34 to 39 Hz) at which the AHU2 supply air AC motors are currently operating.

The information obtained from these specifications and performance data sheets coupled with the considerations discussed above, about the two 8 and 6 motor combinations in the two forgoing cases (1) and (2), have revealed an important piece of information about the current operational settings of these motors. That is, these motors are not currently operating in an efficient manner.
Chapter 4  Analysis and Discussion on Experimental Results

The results that have been obtained in the course of this investigation for the electrical energy consumption and the savings based on the reduction of the energy consumption associated with every motor combination considered which are based on the calculations made for determining the head loss and the corresponding power loss are all analyzed and discussed in this chapter. The results of the simplification of the effects of friction factor on total and major head losses are also discussed. In the section before the last, a comparison is made between a single fan as an equivalent fan to that of all of the supply air AC motors combined in the AHU2 ‘fan wall’ in terms their associated energy savings. Inconsistent results that have been encountered will also be treated in the last section.

4.1  Calculations for Head Loss and Power Loss

In this investigation, under a variety of fresh air conditions, a collection of motor combinations has been chosen from a total of 8 motors to operate on a number of occasions. Early on however the results that were compiled, based on the measurements collected for the 6, 5 and 4 motor combinations that had been taken on the same occasion, led to some inconsistent results. These results were produced during the compilation of the collected data for each motor combination to run the Matlab code for the calculation of the total head loss and the associated power loss in a length of the duct between any pair of measuring stations 2, 3 and 4 data. These results have been retained and are graphically represented in two figures which will be discussed in the last section of this chapter.
The tabulated data in Tables 4.1 were extracted from JEMS on Date 4. It should be noted here that during this testing procedure for determining the performance of these motors (operating in whatever motor combinations by turning a number of motors on and off) the speed of the motors had not been locked into only one particular operating frequency by the VFDs settings. Rather, at different times while the data were being collected on Date 4, the speed of the motors was locked into different speeds. On this occasion, the aim of the testing procedure was to determine the individual motor power consumption in any particular motor combination chosen at different speeds and the corresponding static pressure produced and measured by a sensor at a specific location length of the duct system downstream of these motors.
Figure 4.1  Supply Air AC Motors Data Extracted from JEMS on Date 4.
It can readily be noted in Figure 4.1 that the electrical energy consumption for a 5 motor combination is substantially less than that for each of the 8 motor combinations tested on Date 4. By studying the data contained in Figure 4.1, it can also be noted that operating the AHU2 supply air AC motors at higher frequencies, while not exceeding their peak efficiency, can result in substantially reducing the electrical energy consumption of these motors. By applying the data in Figure 4.1, a simple calculation will show that

(1) For the 5 motor combination running at 47 Hz with the fresh air temperature at about 4.4 °C, electrical energy consumption \( W_5 \) is

\[
W_5 = 2.02 \ kW \times 5 \times \frac{3.81}{3.73} = 10.31 \ kW
\]

(2) For the 8 motor combination running at 38 Hz with the fresh air temperature at about 4.4 °C, electrical energy consumption \( W_8 \) is

\[
W_8 = 1.4 \ kW \times 8 \times \frac{3.81}{3.91} = 10.92 \ kW
\]

For the sake of simplicity, it is assumed that the electrical energy consumption increases linearly as the static pressure produced by these motors increases. This assumption is made here to show that the correction factors \( \frac{3.81}{3.73} \) and \( \frac{3.81}{3.91} \) can be used in order to accurately calculate the corresponding electrical energy consumptions \( W_5 \) and \( W_8 \), respectively, in (1) and (2) above.

A comparison between these two electrical energy consumptions will show that the following energy savings can then be achieved by choosing the 5 motor combination over that of the 8 motor combination

\[
\Delta W_{min} = W_8 - W_5 = 0.61 \ kW
\]
As was discussed in the introductory section of Chapter 1, the design of AHU2 supply air motors ‘fan wall’ is based on a total of $8 (= 3 \times 3 - 1)$ AC motors in a $3 \times 3$ matrix of AC motors (see Figure 1.1) with one cell (or cube) of the ‘fan wall’ left empty. In terms of the flow of air through AHU2, it is not clear as to whether the selection of the cell (in the ‘fan wall’) for the missing AC motor in this matrix can be a significant factor in designing a ‘fan wall’ of this type for AHU2. Moreover, in this study, the process of measuring the variables of interest (i.e., air velocity and static pressure) at the designated sections of the AHU2 intake air duct system (see Figure 2.1) was purely arbitrary. Furthermore, for the purposes of experimenting with motor combinations totaling less than 8, the selection for which motor(s) in this matrix to be turned off was made completely arbitrary.

For each of the energy methods which have been discussed in Chapter 2, there exists a corresponding Matlab code that has been devised to perform all the necessary calculations for determining any variable of interest. These Matlab codes have been designed in order to facilitate the calculations for all of the parameters that are necessary in determining the final results, namely, the head loss and the associated power loss. In order to determine which motor combination(s) has produced the least amount of head loss and the associated power loss, a comparison can then be made between each of these final results.

These two Matlab codes are universally applicable for any motor combination chosen for their respective energy method in order to calculate the head loss and its corresponding power loss associated with the movement of air through each section of the duct confined between the designated measuring stations 2 and 3, 2 and 4, and 3 and
4, respectively. In order to avoid any confusion, each code has been given a name and a description in the following manner.

(1) Matlab code I is to be applied for performing all necessary calculations based on the first energy method in Chapter 2, and

(2) Matlab code II is to be applied for performing all necessary calculations based on the second energy method in chapter 2.

Each of the aforementioned Matlab codes is a piece of computer programming software which contains multiple of sections. Each section has been written in such a way so as to be used for the calculation of one (or more) parameter(s) involved in determining the final results. Each of the sample Matlab codes I and II (both of which are included in the Appendices B and C) is for a specific motor combination. The details of these two specific motor combinations are as follows:

(1) This is an 8 motor combination chosen to perform the calculations using the sample Matlab code I. All 8 motors are operating at 40.8 Hz with the fresh air temperature at about 0.1 °C and the specific weight \( \gamma = 12.67 \frac{N}{m^3} \) at 0 °C.

(2) This is a 6 motor combination chosen to perform the calculations using the sample Matlab code II. All 6 motors are operating at 45.0 Hz with the fresh air temperature at about 9.0 °C and the specific weight \( \gamma = 12.67 \frac{N}{m^3} \) at 0 °C.

In the case of (2), the sample Matlab code II (in addition to having the capability of performing the same calculations as in the case of (1) also contains a special section that has been written for the computations of the energy coefficients based on the input data for this motor combination in which the results obtained are based on the mean air velocity \((U_m)_i\) (i.e., the average value of all 65 grid air velocities) which is calculated at
each of the designated stations $i = 2, 3$ and 4. Care has been taken to make certain that all of the designated measuring stations 2, 3 and 4 were chosen to be as far away from the louver (entrance, see Figure 2.1) of the AHU2 intake air duct system as possible\textsuperscript{1}.

Figures 4.2 and 4.3 show the head loss and the corresponding power loss, respectively, for each motor combination selected based on the first energy method.

Figures 4.4 and 4.5 show the head loss and the corresponding power loss, respectively, for each motor combination selected based on the second energy method.

There are observations that can immediately be made by studying Figures 4.2 through 4.5 to obtain some useful information.

(1) Referencing Figure 4.2 for the 6 motor combination that was tested on Date 3, it can easily be seen that the amount of head losses occurred in the lengths confined between stations 3 and 4, stations 2 and 3 and stations 2 and 4 (i.e., elements 1, 2 and 3), respectively, is the least when compared to the respective head losses for all other motor combinations.

(2) Similarly, referencing Figure 4.3 for the 6 motor combination that was tested on Date 3, it can easily be seen that the amount of power losses occurred in the lengths confined between stations 3 and 4, stations 2 and 3 and stations 2 and 4 (i.e., elements 1, 2 and 3), respectively, is the least as compared to the respective power losses for all other motor combinations.

\textsuperscript{1} See footnote 1 in Chapter 2, Section 2, on page 38.
Figure 4.2  Seasonal Head Losses for Motor Combinations Method I.

Description of Axis:  
Element 1 = Length of intake air duct between stations 3 and 4
Element 2 = Length of intake air duct between stations 2 and 3
Element 3 = Length of intake air duct between stations 2 and 4

* Measurements taken with all available VAV boxes in 100% open position.
Figure 4.3  Seasonal Power Losses for Motor Combinations Method I.

Description of Axis: Element 1 = Length of intake air duct between stations 3 and 4
Element 2 = Length of intake air duct between stations 2 and 3
Element 3 = Length of intake air duct between stations 2 and 4

* Measurements taken with all available VAV boxes in 100% open position.
Figure 4.4  Seasonal Head Losses for Motor Combinations Method II.

Description of Axis:  Element 1 = Section of duct between stations 3 and 4
Element 2 = Section of duct between stations 2 and 3
Element 3 = Section of duct between stations 2 and 4

* Measurements taken with all available VAV boxes in 100% open position.
Figure 4.5  Seasonal Power Losses for Motor Combinations Method II.
(3) Neither one of the observations (1) and (2) can be considered to be an example of an inconsistent result which may have been encountered in this study. On the contrary, these two examples show once again that, under similar conditions, a 6 motor combination operating close to (or at) the peak efficiency of these motors can perform more efficiently than that of an 8 motor combination when operating in the 34 to 39 Hz range which is well below the peak efficiency of these motors.

(4) For each of the motor combinations 6, 8 and 6 tested on Dates 6, 5 and 3, respectively, the respective magnitude of their head losses shown in Figure 4.2 can be considered sufficiently close to the respective magnitude of their counterpart head losses in Figure 4.4.

(5) Similarly, for each of the motor combinations 6, 8 and 6 tested on Dates 6, 5 and 3, respectively, the respective magnitude of their power losses shown in Figure 4.3 can be considered sufficiently close to the respective magnitude of their counterpart power losses in Figure 4.5.

The requirements to be met by AHU2 on all occasions covering Dates 3, 6 and 5, are at the highest level such that a $7.08 \frac{m^3}{s}$ (15000 cfm) uniform airflow through the AHU2 intake air duct system in addition to maintaining a total $3.81 \text{ cm}$ (1.5 inches) of water column of static pressure, at a distance equivalent to $\frac{2}{3}$ of the length of the duct system downstream before the last room to be served its supply of air, should be maintained.

A comparison between the following cases should make the point clear as to which motor combination(s) can be considered to be the most efficient while meeting (or even exceeding) these requirements.
(1) In the case of the 8 motor combination tested on Date 5, between stations 2 and 4, where all of the 8 motors were operating at 40.8 Hz at an average fresh air temperature of about 0.3 °C for the specific weight $\gamma = 12.67 \frac{N}{m^3}$ at 0 °C, an average airflow of $7.75 \frac{m^3}{s}$ (16520 cfm) is created with a 5.44 cm (2.14 inches) of water column of static pressure maintained at the designated point downstream in the duct system. The average instantaneous electrical energy consumption per motor recorded in this case was 1.72 kW. Thus in this case, in order to meet the requirements above, the total electrical energy consumption in this case can be calculated as follows:

$$ (W_8)_1 = 1.72 \text{ kW} \times 8 \times \frac{1.5}{2.14} \times \frac{15000}{16520} = 8.76 \text{ kW} . $$

(2) In the case of the 6 motor combination tested on Date 3, between stations 2 and 4, where the 6 motors are operating at 45 Hz at an average fresh air temperature at about -3.2 °C for the specific weight $\gamma = 12.67 \frac{N}{m^3}$ at 0 °C, an average airflow of $7.2 \frac{m^3}{s}$ (15241 cfm), approximately, is created with a 6.50 cm (2.56 inches) height of water column of static pressure maintained at the same designated point downstream in the duct system. The total electrical energy consumption by 6 motors in this case is

$$ (W_6)_2 = 2.3 \text{ kW} \times 6 \times \frac{1.5}{2.56} \times \frac{15000}{15241} = 7.96 \text{ kW} . $$

(3) In the case of the 6 motor combination tested on Date 6, between stations 2 and 4, in which the 6 motors are operating at 45 Hz with an average fresh air temperature at about 8.5 °C for the specific weight $\gamma = 12.67 \frac{N}{m^3}$ at 0 °C, an
average airflow of $8.05 \frac{m^3}{s}$ (17169 cfm), approximately, is created with a 5.46 cm (2.15 inches) height of water column of static pressure maintained at the designated point downstream in the duct system. The total electrical energy consumption by 6 motors in this case is

$$(W_6)_3 = 2.25 \text{ kW} \times 6 \times \frac{1.5}{2.15} \times \frac{15000}{17169} = 8.23 \text{ kW} .$$

Assuming that electrical energy consumption increases linearly with the magnitudes of the static pressure and the flow rate created, then a simple comparison, shows that

$$(\Delta W)_{1-2} = (W_8)_1 - (W_6)_2 = 0.80 \text{ kW} ,$$

$$(\Delta W)_{1-3} = (W_8)_1 - (W_6)_3 = 0.53 \text{ kW} .$$

These results are further indication that in either case the 8 motor combination consumes a larger amount of electrical energy. As the specifications and the performance data sheets for this type of AC motor/fan show, these motors will run more efficiently at higher frequencies such as 45 Hz than at frequencies in the range of 34 to 39 Hz, as they are currently operating at Site A. There is, of course, a limit as to how efficient an AC motor/fan of this type can operate.

For the purposes of this study, however, it is sufficient to show that for this type of AC motor/fan, as compared with an 8 motor combination operating in a frequency range of 34 to 39 Hz, when a 6 motor combination is operating in a frequency range of 40 to 45 Hz can run more efficiently and consume less electrical energy. This fact has been clearly illustrated in the comparison that was made between the cases (1), (2) and (3) above. In addition to this, in a 6 motor combination the amount of head loss (as well as the corresponding power loss) associated with the movement of air in the lengths of the AHU2 intake air duct between stations 2, 3 and 4, respectively, is less than that of an 8
motor combination when a comparison is made under the same conditions (or under sufficiently similar conditions in which the differences in the average fresh air temperature in each case are no more than only a few degrees Celsius). This point has also been clearly made in a comparison made between the results illustrated in Figures 4.2 through 4.5.

In the course of conducting the investigation, through a series of trial and error testing, it was also revealed that under the same conditions (i.e., having a constant fresh air temperature with the entire available VAV terminal boxes fixed at 100% open position) the following motor combinations did not produce any desired results to be recorded and discussed here.

(1) In the case of only 3 motors, 2 motors or 1 motor operating (even at frequencies in excess of 55 Hz) a sufficient amount of power could not be produced in order to drag a sufficient volume (or mass) of air to flow through the AHU2 intake air duct system. Nor was it possible to maintain the required amount of a total of 3.81 cm (1.5 inches) height of water of static pressure at a distance of about \( \frac{2}{3} \) of the length of the duct downstream before the last room to be supplied with its share of fresh air. The results obtained from running these motor combinations were therefore discarded.

(2) In the case of the 7 motor combination running at a frequency higher than the range of 34 to 39 Hz in which all 8 motors are normally operated, it did break even with the 8 motor combination in terms of consumption of electrical energy. But, as its electrical energy consumption in this case was clearly not as low as that of the 6 motor combination, it was decided early on during the investigation that
this motor combination is not a viable alternative to the 8 motor combination. The results obtained from running this motor combination were also discarded.

Having discarded all of the unacceptable data, it can be said that even in the worst case scenario in which the electrical energy consumed by a 6 motor combination operating at 45 Hz happens to break even with that of an 8 motor combination which may be operating at as high a frequency as 40.8 Hz, it can still be argued in favor of the 6 motor combination operating at 45 Hz in the following sense.

The 6 motor combination should clearly be chosen over the 8 motor combination for the simple reason that, for as long as the two requirements of the supply of air by AHU2 (in terms of the required amount of airflow rate flowing and at the same time maintaining the required amount of static pressure at the designated point downstream in the duct) are met, it would still make more sense to operate as low a number of AC motors among all available 8 motors in the AHU2 ‘fan wall’ at any given time. A rather simple cost-benefit analysis will clearly show that maintaining a higher number of motors constantly in operation is always more costly than maintaining a lower number of the same motor combinations at all times.

With this in mind, the entire procedure which has been developed in this study is to be implemented in order to optimize the electrical energy consumption of this type of AC motors in the sense of making a choice from a particular selection to include a specific number of AC motors to be operating in order to supply the air for the operational purposes of AHU2. The selection of a specific number of AC motors is to be made in the range of 6 to 4 motors running the frequency range of (40 to 47) Hz, respectively. This procedure is to take effect provided that, in the case of the 6 motor
combination, the speed of the motors will be confined to the frequency range of (40 to 47) Hz without triggering an emergency situation of any kind in the UNL/JH building or causing any type of damage to the equipment or their surroundings.

The whole purpose being discussed here is that, by way of utilizing the aforementioned computer program in JEMS, a number of controllers (if not in their entirety) that are, for the most part, currently in place and operating shall function in concert to command a specific number (i.e., in the range of 6 to 4) of the AHU2 supply air AC motors to operate in the desired speed range (i.e., in the frequency range of (40 to 47) Hz) at any given time and for all seasonal weather conditions. That is, it is possible in practice to shift from one motor combination to another while at the same time reaching the maximum (or the minimum) level of output in a specified range in terms of the flow rate and, at the same time, meet a particular static pressure requirement somewhere in the downstream duct system.

The aforementioned program in JEMS is currently capable of commanding any motor combination to span a safe frequency range of, for example, (25 up to 55) Hz while the motors will be running only in a fixed arrangement of the number of motors in operation. There can be as many as up to 8 motors in operation permanently staying in the turned on position, provided the speed of the 8 motors will not exceed 41 Hz so that the is not drastically increased beyond the maximum airflow rate requirement of 7.08 m$^3$/s (15000 cfm) so as to trigger the emergency hatch doors to open in order to let the excess air escape from the duct system downstream of AHU2. In the case of any motor combinations with less than 8 motors, permanently operating to meet requirements 1 and 2 in the above, the remaining number of motors will permanently remain in the off
position. The same precautionary measures described for the 8 motor combination should be taken into consideration for all other motor combinations with less than 8 motors running.

It is therefore believed that this software program may require some modifications or some further development so that its capabilities will be increased such a way so as to enable it to command the AC motors to operate in a much more efficient manner in order to perform the tasks which are going to be recommended in the next section.

Referencing Figure 4.1, a comparison was made between the electrical energy consumption data obtained from JEMS for the motor combinations 5 and 8. The correction factors $\frac{3.81}{3.73}$ and $\frac{3.81}{3.91}$ were used for the 5 and 8 motor combinations, respectively, in order to determine their respective electrical energy consumption on Date 4 thereby achieving a minimum savings of $\Delta W_{\text{min}} = 0.61 \text{ kW}$. For calculation of the maximum savings in electrical energy consumed, however, these correction factors can be disregarded such that

$$\Delta W_{\text{max}} = (1.4 \times 8 - 2.02 \times 5) \text{ kW} = 1.1 \text{ kW}.$$ 

In the case that the AC motors are operating under partial load, for example, in two out of a total of four seasonal weather conditions, then only a fraction of (e.g., 9000/15000) savings should be considered for half of the year. Assuming now that the average cost of electrical energy consumption for operating a piece of equipment such as the AHU2 supply air AC motor, in a commercial building such as UNL/JH, is about $0.11$ per kWh, then total amount of annual savings associated with the optimal reduction of the AC motors throughout the year under full load can be calculated as follows:
(1) For a minimum energy saving of $\Delta W_{min} = 0.61 \, kW$, the minimum annual savings is

$$s_{min} = 0.61 \times \$0.11 \times 24 \times 365 \times 0.5 \times (1 + \frac{9}{15}) = \$470.2 \ , \text{and}$$

(2) For a maximum energy saving of $\Delta W_{max} = 1.1 \, kW$, the maximum annual savings is

$$s_{max} = 1.1 \times \$0.11 \times 24 \times 365 \times 0.5 \times (1 + \frac{9}{15}) = \$848 \ .$$

It is believed that a total of 10 small-sized air handling units of the AHU2 type have already been purchased to operate in the UNL buildings while a total of 40 large-sized air handling units have already been purchased to operate in the UNL buildings. The larger air handling unit is of the type that contains a total of 16 AC motors in a $4 \times 4$ matrix (bank) formation which are housed in their ‘fan wall’. The total annual savings for the smaller units can therefore be calculated as follows:

$$S_{min} = 10 \times s_{min} = \$4702 \ ,$$

$$S_{max} = 10 \times s_{max} = \$8480 \ .$$

Assuming that there can be twice as much savings associated with the larger units, then the total annual savings for the larger units will be:

$$L_{min} = 2 \times 40 \times s_{min} = \$37,616 \ ,$$

$$L_{max} = 2 \times 40 \times s_{max} = \$67,840 \ .$$

Assuming that in ten years’ time, for operating this type of equipment, the cost of electrical energy will remain at the same rate of $0.11/kWh$. It is therefore estimated that, in the first decade after the conversion has been made from the current settings of the controls for all of these AC motors operating in the UNL/HVAC Systems over to the improved settings in accordance to the recommendations made in this study, there may be
a total savings in the range of $(376,160 \text{ to } 678,400)$ to be had. It may of course be the case that, in terms of the actual rate for the cost of electrical energy consumption, UNL is utilizing the benefits of a two-tier system offered to it by the local provider(s) of electric power. In this case then, may be currently paying less than half of $0.11/kWh$. As has been mentioned on page 222 in [4.1], the cost of electrical energy in $kWh$ may still be about $0.052/kWh$. Accordingly, the total savings per decade for UNL can be adjusted to be in the range above or about $(177,820 \text{ to } 320,698)$.

The recommendations that are going to be made in the next section are such that all of the supply air AC motors which are housed in their respective ‘fan walls’ in all of the aforementioned 50 air handling units will be operating at their peak efficiencies in accordance with a seasonal schedule (or a variation thereof). For all of the 50 air handling units purchased by UNL, there is also an estimated total savings of $270,000 for the total cost of the left over (extra) AC motors in the UNL/HVAC systems after the conversion to the new seasonal schedule for the operation of the supply air AC motor combination and the associated VFD devices. These estimated total savings are as follows:

\[
\text{Cost of a } 7\frac{1}{2}\text{hp AC motor} = $1000
\]

\[
\text{Cost of the associated VFD device} = $500
\]

\[
\text{Number of Extra AC motors} = 2 \times 10 + 4 \times 40 = 180
\]

\[
\text{Number of associated VFDs} = 2 \times 10 + 4 \times 40 = 180
\]

Therefore, the estimated total savings for the leftover equipment is about

\[
$(1000 + 500) \times 180 = $270,000.
\]
4.2 Results for Effects of Friction Factor on Head Losses

In this section, a comparison shall be made between different values of the head loss resulting at the measuring stations chosen in this study. In so doing, any and all significant aspects of such head losses shall also be discussed. The results for the plot of total head loss against major head loss at stations 2, 3 and 4 in the AHU2 intake air duct system are shown in Figure 4.6(a). This figure shows the curve of the total head loss against major head loss. It also shows a straight line which is chosen to represent a hypothetical linear approximation for this curve. In order for this line to approximate the curve, it is cutting through it at two distinct points. It cuts the curve at a point between the elements 1 and 2 and at another point between the elements 2 and 3, respectively.

As it is shown in Figure 4.6(a), the numerical value for element 1, representing the length of the duct between stations 3 and 4, is 0.0092 m for both the total and the major head losses. This value is insignificant when compared to the value of the corresponding parameters for elements 2 and 3 in either case of total or major head loss. The reason for this is that there are no bends or any elevation in the length of the duct confined between stations 3 and 4. Therefore, the head losses produced as a result of air flowing through this length of the duct are substantially less than the head losses for elements 2 and 3, i.e., the lengths confined between stations 2 and 3 and between stations 2 and 4, respectively. The actual line representing the linear approximation passes through two points on the curve depicted in Figures 4.6(b). These two points are positioned between elements 1 and 2 and between elements 2 and 3, respectively. A special Matlab code has been devised in such a way that this approximation can be performed for elements 1, 2 and 3 independent of each other.
Figure 4.6  Effects of Friction Factor on Total against Major Head Loss Curve.  
(a) Original and the Hypothetical Linear Approximation of Head Loss.
Figure 4.6  Effects of Friction Factor on Total against Major Head Loss Curve. (b) Original and the Actual Linear Approximation of Head Loss.
A sample of this special Matlab code (which has been devised for the purposes of linearization of the curve in the Figure 4.6(a)) is provided in Appendix D. The calculated values for the slope $a$ and the constant $b$ of the line in equation (2.17) using this special Matlab code are given below. As the values assumed by $x$, $y$, $X$ and $Y$ must always be a positive real number (for these are all physical quantities for the magnitudes corresponding to head losses) then the following conditions must always hold.

\[ x = X > 0 \text{ and } y - C > 0 \]

In this special Matlab code, using any number of iterations (e.g., 1, 10, 100, etc.) for each element $i = 1, 2$ and 3 the value of the first entry for the slope $a_i$ and the constant $b_i$ of the line approximating the curve as described by equation (2.16), depicted in Figure 4.6(b), were calculated as follows:

1. For element 1, on the 1st iteration step,
   \[ a_1 = 20.277499295186118 \text{,} \]
   \[ b_1 = -0.151187040143357 \text{,} \]

2. For element 2, on the 1st iteration step,
   \[ a_2 = 20.277499295186317 \text{,} \]
   \[ b_2 = -0.151187040143367 \text{,} \]

3. For element 3, on the 1st iteration step,
   \[ a_3 = 20.277499295186352 \text{,} \]
   \[ b_3 = -0.151187040143366 \text{.} \]

As it can be seen from these results, the values obtained for the slope and the constant of the line in equation (2.17) for elements 1, 2 and 3 are all virtually identical. The average value of these results, namely $a_a$ and $b_a$, have been used in order to
construct the line approximating the curve as illustrated in Figure 4.6(b) in the following manner:

\[ y = a_a x + b_a \]

\[ = 20.2775x - 0.1512 \] (4.1)

The results for the values of total, major and minor head losses for elements 1, 2 and 3 which are illustrated in the plot of the curve in Figure 4.6(b) show that this curve has been approximated by the line, as described by equation (4.1), passes through two points on the curve which happen to closely approximate the \( Y \) coordinates of elements 1 and 3. According to the approximation method which has been described in Chapter 2, for each point having coordinate pairs \( X_i \) and \( Y_i, \ i = 1, 2 \) and 3 on the curve in Figure 4.6(a), it is only the \( Y_i \) coordinate that needs to be approximated by each point having coordinate pairs \( x_i \) and \( y_i, \ i = 1, 2 \) and 3 on the line.

This task has been accomplished by minimizing the sum of squares of each the differences for each value of the \( Y_i \) coordinate on the curve and the \( y_i \) coordinate on the line representing the linear approximation of this curve. It may be accidental that the linear approximation line which has been constructed in this way is approximating the \( Y \) coordinate of element 2 not as closely as it is the \( Y \) coordinates of elements 1 and 3. It may also be argued that this is because, at each iterative step in the special Matlab code (in Appendix D), the value of the coordinate pairs \( X_i \) and \( Y_i \) are updated independently and for each particular element \( i, \ i = 1, 2 \) and 3. Next, the slope and the constant \( a_i \) and \( b_i, \ i = 1, 2 \) and 3, respectively, of the linear approximation line are then updated independently and for each particular element \( i, \ i = 1, 2 \) and 3.
In order to ensure that $X$ will always assume a positive value in equation (2.16), the condition $Y > C$ must hold. Therefore, for this linear approximation to work, it may take a few iteration steps (e.g., updates or improvements) to be implemented prior to any acceptable values for $X$ can be generated. It may very well be the case that this line may actually turn out to be very similar to the linear approximation shown in Figure 4.6(b). In this figure, however, the linear approximation line is passing through the two points on the curve. One point is between elements 1 and 2 on the one hand and the other point is between elements 2 and 3. It should be noted that these two points may turn out to be coinciding elements 2 and 3, as these are the dominant elements on the curve of the plot of the total against major head losses having, as their respective nontrivial values, 0.0347 m for $(X_2)_o$ coordinate and 0.6535 m for $(Y_2)_o$ coordinate as well 0.04393 m for $(X_3)_o$ coordinate and 0.6653 m for $(Y_3)_o$ coordinate on the horizontal and vertical axes in Figure 4.6(a), respectively. These values are much greater in magnitude than 0.0092 m which is the same value for $(X_1)_o$ and $(Y_1)_o$ coordinates of both total and major head losses in element 1.

It should be noted that by making some modifications to this iterative procedure, it may be possible to calculate the universal values for $a$ and $b$ by simultaneously achieving the approximation points having coordinate pairs such as $X_1, Y_1$ and $X_3, Y_3$ for elements 1 and 3, respectively, as well as the approximation point having the coordinate pair $X_2, Y_2$ for element 2. To construct such an iterative method, that is capable of successfully performing the iteration steps for this total approximation scheme, it may be required to devise a Matlab code that can turn out to be much more complicated than the special Matlab code in Appendix D.
4.3 Comparing Single Fan with ‘Fan wall’

For an accurate comparison to be made between an equivalent single fan and a motor combination in the AHU2 ‘fan wall’, the single fan should be equivalent in horse power rating and (airflow and total static pressure) capacity to the combined horse power rating and (airflow and total static pressure) capacity of the 8 AC motors in the AHU2 ‘fan wall’ supply air AC motors. According to the design specifications and the performance curves in [4.2] for the AHU2 ‘fan wall’ AC motors/fans, these motors

1. have a combined power rating of $7.5 \text{ hp} \times 8 = 60 \text{ hp}$ and operating at a speed of 60.4 Hz are

2. capable of creating a total of $12.75 \frac{m^3}{s}$ (27000 cfm) uniform airflow rate,

3. can maintain a total static pressure of 20.32 cm (8.0 inches) corresponding to

4. a total brake horse power of 53.25 bhp based on a 6.66 bhp per fan.

For the purposes of making a comparison in energy efficiency and savings, the selection of a single fan to serve as an equivalent of the 8 AC motors in the AHU2 ‘fan wall’ should be made in such a way that the above considerations, (1) through (4), are met as closely as possible. According to the fan data provided in an online publication by Greenheck Fan Corporation [4.3], the Double-Width Centrifugal Fan, model number 24-AFDW-41 (Maximum rpm Class III = 2762), can be considered to be a suitable fan for this purpose. This single fan, tested at an elevation of 0.0 m with the airstream temperature at about 21.1 °C, has the following performance characteristics.

1. It has a 60 hp motor and, operating at a speed of 40.85 Hz, is
(2) capable of creating a total of $12.75 \frac{m^3}{s} (27000 \text{ cfm})$ uniform airflow rate while
(3) maintaining a total of $20.32 \text{ cm} (8.0 \text{ inches})$ of static pressure with an
(4) operating brake horse power of $53.93 \text{ bhp}.$

In order to meet the requirements imposed on the AHU2 ‘fan wall’ 8 AC motors, this
single fan should perform under the following operating conditions:

Operating at a speed of about $27.0 \text{ Hz}$, this fan should be capable of creating a
$7.08 \frac{m^3}{s} (15000 \text{ cfm})$ airflow rate and also maintain a total static pressure of about
$11.29 \text{ cm} (4.44 \text{ inches})$.

Based on the data provided on page 56 in [4.3], the estimated value for the brake horse
power to meet the above air supply requirements can then be calculated to be

$$P_{bhp} = 14.63 \text{ bhp}.$$  

The results obtained in the previous section of this chapter showed that in the case
of all 8 AC motors in the AHU2 ‘fan wall’ running at $40.8 \text{ Hz}$ to create a $7.08 \frac{m^3}{s}$
($15000 \text{ cfm}$) airflow rate and to maintain a total static pressure of about
$11.29 \text{ cm}$ (or $8 \times \frac{15000}{27000} = 4.44 \text{ inches}$), the electrical energy consumption was
calculated to be $W_8 = 8.76 \text{ kW}.$

This corresponds to an estimated brake horse power of

$$(P_{bhp})_8 = 8.76 \text{ kW} \times \left(\frac{1.341 \text{ bhp}}{1 \text{ kW}}\right) = 11.75 \text{ bhp}.$$  

A comparison between the numerical results for the estimated brake horse power in
the case of the single fan and that of the 8 motor combination (in the AHU2 ‘fan wall’)
shows that the AHU2 ‘fan wall’ can operate more efficiently than the single fan with an estimated energy savings \[\frac{(14.63 - 11.75)}{14.63} \times 100 = 19.69\%\].

4.4 Remarks

A comparison shall now be made between different values of the head losses which have already been discussed and the head losses that were obtained from some inconsistent results based on the data acquired on Date 2 in the course of this investigation. These results are shown in Figures 4.7 and 4.8. Referencing Figure 4.7 it can be noted that, on this occasion, the value of the head loss at element 1 is greater than that of element 2. This happens to be the case even though for element 2 the duct contains two 90 degree bends in addition to an elevation of 1.75 m and a considerably larger horizontal length compared to element 1 with no bends contained in it. Referencing Figure 4.8, a similar observation can be made for the same motor combinations 6, 5 and 4 tested on Date 2 in that the value of power loss at element 2 is less than that of element 1.
Description of Axis:  
Element 1 = Length of intake air duct between stations 3 and 4  
Element 2 = Length of intake air duct between stations 2 and 3  
Element 3 = Length of intake air duct between stations 2 and 4

Figure 4.7 Other Seasonal Head Losses for Motor Combinations Tested.
Figure 4.8 Other Seasonal Power Losses for Motor Combinations Tested.

Description of Axis: Element 1 = Length of intake air duct between stations 3 and 4
Element 2 = Length of intake air duct between stations 2 and 3
Element 3 = Length of intake air duct between stations 2 and 4
This is an example where the experimental data obtained during the investigation may have led to obtaining inconsistent results. Such results, however, need not be regarded to be extreme for the head loss and the consequential power loss in the sense that the conditions imposed on AHU2 in the occupied hours when variability in the position of the individual VAV terminal boxes is the highest and the overall requirements of the building for the supply of air may vary from hour to hour. This fact coupled with the quality of the fresh air moving through the AHU2 intake air duct system can account for such an inconsistency.
Chapter 5  Conclusions and Recommendations for Future Work

In this chapter, there will first be general conclusions to be drawn. There will then be a discussion about the savings to be had in terms of the particular range of percentages of electrical energy reductions associated with the motor combination considered to be optimal while meeting the supply of air requirements. There are recommendations which will be made in terms of selection of motor combinations and their associated operating speeds. Conclusions in regard to the effects of friction factor on head losses are discussed later. Future work on improving the energy saving process will then be discussed. Possibility of finding a global solution to determine the slope and the constant of the line to approximate the curve of the total against major head loss will also be discussed.

5.1  Conclusions

At the outset, a significant conclusion can be drawn here from the results shown in the graphs in Figures 4.2 through 4.5. It can be seen from these figures that the numerical results obtained for the total head loss (and the consequential power loss) by applying the equations that have been employed in the energy methods in Sections 2.1 and 2.2 have confirmed the fact that when the results produced by employing either energy methods are sufficiently close to each other, then it can safely be concluded that all of the equations employed in each method have been properly constructed.

Referencing Figure 2.1, in each of the two methods applied in this study, the consistency and the accuracy of the results are also confirmed by taking note of the following:
(1) That the magnitudes of the head loss (and the consequential power loss) are at a minimum, intermediate and at a maximum levels for elements 1, 2 and 3, respectively.

(2) The results obtained for the total head loss in elements 1, 2 and 3, respectively, show that they are calculated to be as should be expected. This is can be seen by considering a comparison between the following cases:

(2a) In element 1, there is only a straight and the shortest horizontal length of the duct which is confined between the measuring stations 3 and 4. Therefore it is expected that the total (energy) losses associated with this element should be the smallest in magnitude.

(2b) In element 3, there is a length of the duct, confined between the measuring stations 2 and 4, in which there is an elevation of about 1.75 m in addition to two bends and a horizontal length between these two stations that is the largest. Therefore it is expected that the losses associated with this element should be the largest in magnitude.

(2c) In element 2, there is a length of the duct, confined between the measuring stations 2 and 3, in which there is an elevation of about 1.75 m in addition to two bends and a horizontal length between these two stations that is larger than the one in element 1 but shorter than the one in element 3. Therefore it is expected that the losses in this element should be larger in magnitude than the losses in element 1 but smaller in magnitude than the losses in element 3.

The results in Figures 4.1 and 4.2 show that even when the extreme cases, in which the temperature fresh air entering the building is below at 0 °C (32 °F) or
above at 38 °C (about 100 °F), are included among the current settings for the operational conditions of these motors, these motors are still operating inefficiently at the present time. Moreover, the author believes that the existing computer program which has already been created in JEMS (as has been referred to in this thesis numerously) can be modified or be further developed in such a way so as to bring about a smooth transition from one motor combination to another when ramping up (or reducing) the speed of these motors and at the same time switching on (or switching off) as many motors as needed to meet any air supply requirements at any time by AHU2.

It is also concluded that for a fluid flow through a noncircular duct in a turbulent regime, the function describing the relationship between the friction factor and Reynolds number is not the only relationship through which friction factor can be determined even though a unique value can always be calculated for the friction factor for each of the corresponding Reynolds number in this function.

5.1.1 Energy Savings

The calculations which were made in Section 4.1, Chapter 4 clearly showed that only selections made from a specific number of AHU2 supply air AC motors in the range of 6 to 4 motors and their corresponding operating speed in the frequency range of (40 to 47) Hz, respectively, will guarantee conditions under which the optimal energy consumption reductions can be achieved. This is provided that proper safety precautions as discussed in the same section have already been implemented so as to avoid the
creation of any kind of emergency situation and damage to the UNL/JH building and its surroundings.

In the case of smaller air handling units such as AHU2, for the supply air AC motors operating in the aforementioned ranges (i.e., in terms of the number motors selected and the speed at which these motors will be operating) an estimated minimum and maximum energy consumption savings of 5.5% and 10%, respectively, can be expected.

Each of the larger air handling units, in operation in the UNL buildings that is equipped with the ‘fan wall’ technology, employs twice as many supply air AC motors that are housed in the ‘fan wall’ of AHU2. It is therefore reasonable to assume that the estimated range of savings in energy consumption associated with the larger air handling units will be twice as much as that of the smaller units. This is an estimated savings of a minimum of 11% and a maximum of 20% in electrical energy consumption.

5.1.2 Recommendations on Selection of Motor Combinations

It is already possible, for example, by way of manipulating the VFDs settings to arrange an efficiently running motor combination (having less than 8 as the total number of motors) to operate permanently at a relatively high frequency to meet the AHU2 supply air requirements for a specific seasonal weather condition. More specifically, these motors can be arranged in such a way to have the capability of performing the following two tasks in order to meet the AHU2 share of the air supply requirements in the UNL/JH building:
(1) Produce a flow rate of fresh air in the AHU2 intake duct system in the range of $4.25 \ \frac{m^3}{s} (9000 \text{ cfm})$, in the summer to $7.08 \ \frac{m^3}{s} (15000 \text{ cfm})$, in the winter.

(2) Maintain, at all times, a total of $3.81 \ \text{cm} (1.5 \text{ inches})$ height of water column of static pressure at a distance equivalent to the $\frac{2}{3}$ length of the duct system downstream before the last room is to be served its supply of air.

As was suggested previously, it is possible in practice to shift from one AC motor combination to another while at the same time reaching the maximum (or the minimum) level of output in terms of the flow rate in a specified range while at the same time maintaining a particular requirement for the static pressure somewhere referred to in this study as the designated point in the duct system downstream the AHU2 ‘fan wall’. It is also easily possible to set, by way of manipulating the VFDs settings, an efficiently running motor combination which employs less than 8 (i.e., the total number of motors available for operation) and keep them permanently running at a relatively high frequency in order to meet the following air supply requirements in the UNL/JH building, in the case of AHU2 supply air AC motors shall at all times have the capability of accomplishing the two tasks in (1) and (2) above.

As it has been suggested in the previous section, the aforementioned program is currently capable of commanding any motor combination to run in a fixed (permanent) arrangement. In an arrangement as such, some or all motors will stay in the turned on position (and the remaining available motors will stay in the turned off position) while spanning the range of motor speed frequencies from 25 up to 55 Hz in order to fulfill (1) and (2) above in order to meet the AHU2 air supply requirements without triggering an
emergency situation in the UNL/JH building or causing any type of damage to the
equipment (AHU2 and its duct system) or to their surroundings.

The author believes that the JEMS software can be modified (or be further
developed by augmenting and combining different segments from different programs
with one another) so as to increase its capabilities. This will be done in order to enable it
to command (by way of communicating with specific controllers in the field) the motors
to run at higher or lower speeds while switching on and off as many number of the
motors in order to perform the tasks that are discussed below.

These tasks will be performed in conjunction with the functions of the VFDs (see
Figure 3.5). The modified software program in question can simultaneously monitor and
control the speed and the operating conditions (i.e., ON/OFF position) of each of the AC
motors in the ‘fan wall’ of the UNL/JH AHU2. The modifications (or the further
development) to be made to this program will specifically allow it to command the
controllers constantly communicating with the VFDs to turn as many motors on and off
as needed while maintaining the speed of the active AC motors in a rather narrow
frequency range (e.g., 40 to 47 Hz).

This procedure may become necessary at times when rather abrupt transitions
from one motor combination to another should altogether be avoided. The required
transitions would be made from A to B, B to C, C to D and D to A, as in the following
schedule.

To fulfill all of these tasks and to also meet the air supply requirements mentioned
in (1) and (2) above, it is recommended that the AC motors should operate, at all times,
according to the following annual operational schedule with the selected motor
combinations and the corresponding operating speeds based on a full climate cycle to cover all four seasonal weather conditions:

(A) In Fall to Winter 5 motors to run in (40 to 47) Hz,
(B) In Winter to Spring 6 motors to run in (43 to 40) Hz,
(C) In Spring to Summer 5 motors to run in (47 to 40) Hz,
(D) In Summer to Fall 4 motors to run in (40 to 43) Hz.

5.2 Simplification of Effects of Friction Factor on Head Loss

In Chapter 2, it was noted that in [5.1] the literature was critically examined for pressure drop data on isothermal fluid flow through non-circular ducts in a turbulent regime. It was, however, concluded in [5.1] that in the case of fluid flow in non-circular ducts the function for friction factor, in distinction to a turbulent flow in pipes, can be determined independent of the Reynolds number.

In [5.2], referencing the work of [5.1] approvingly, it has also been noted that the function which establishes the relationship between the friction factor and the Reynolds number should not be considered to be the only relationship through which friction factor can be determined. In this study, the author is also inclined to believe, in conjunction with the results obtained by employing the iteration method which was introduced in the last section of Chapter 2 (in addition to the results which were discussed in last part of Chapter 4), that equation (2.11) may only be one expression among many which determines friction factor. That is to say, there may be other expressions (independent of Reynolds number and as yet to be developed) which can determine friction factor in a fully developed turbulent fluid flow through non-circular ducts.
Graphical results in Figure 4.6(b) show that the linear approximation method which has been discussed in section 2.4 is a useful method to employ in simplifying the curve of the total head loss against the major head loss within the structure of the experimental measuring scheme chosen in this study.

5.3 Future work Improving on Energy Saving Process

It should be mentioned that relative humidity and the effects of its variations on other variables such as acoustic pressure, flow rate and the static pressure were all absent during the entire experiment which was conducted in this study. Therefore, a future study which considers a two phase flow, although requiring a much more detailed analysis by taking into account the effects of relative humidity, may lead to more accurate results. In the work reported in this thesis, calculations were based on the average velocity and pressure at each station, further study can be carried out in a three-dimensional flow model. The fact that AHU2 was tested as it is in place and in operation, in and of itself, is both advantageous and disadvantageous.

(1) The advantage is that in this investigation the data, which were gathered from all of the measurements taken at the AHU2 intake air duct system, were obtained in real time (i.e., when the AHU2 ‘fan wall’ and the air handling unit itself were in operation, as intended). Therefore, this procedure resulted in measuring the actual performance of the AHU2 supply air AC motors while in operation. Therefore, the minimization of electrical energy consumption by the AC motors took place actually as it is intended to be.
(2) The disadvantage is that, regardless of which (acoustic or hydraulic) approach is chosen in the investigation, it may be more beneficial to test AHU2 (with the inclusion of an appropriate length of its intake and/or outgoing duct system) in a laboratory setting. In this way, the conditions imposed on AHU2 by the requirements of the UNL/JH building can be simulated in isolation from the rest of UNL/JH/HVAC system of which AHU2 is a part. AHU2 can then be tested under controlled conditions so as to reduce to a minimum the degree of uncertainty that exists when the whole system is in operation in real time. As such, the time consuming delays as a result of, for example the completion of the task of commissioning HVAC system of a newly constructed building like the UNL/JH building, is also avoided. Moreover, to test and experiment with an air handling unit like AHU2 under the most favorable conditions, an appropriate range of fresh air temperature can be chosen so as to avoid spending time to wait for change of seasons. Fresh air with a reasonably high quality can then be supplied for the purpose of simulating the movement of air through the entrance of the AHU2 intake air duct system into the duct and then into the air handling unit itself.

The versatility of this special Matlab code, devised in this study for the purposes of approximating the curve for the total head loss against the major head loss by a straight line, can be tested in such a way so as to ascertain whether it is possible to find a global solution for determining the parameters such as the slope and the constant of the line. This task can be performed by taking into account all of the elements 1, 2 and 3 on the curve in Figure 4.6 (b) at the same time. One way to accomplish this task can be to
simultaneously update all of the constituents which are involved in the calculation of the head loss for elements 1, 2 and 3. It may be possible to create a much more elaborate iterative method, to be designated as a particular Matlab code, to be employed within this special Matlab code in order to update all different values for total and major head losses at the same time. This will be done in conjunction with calculating, by way of updating, all values of the intermediary parameters such as the average airflow rate, Reynolds number and consequently the friction factor, for all three elements 1, 2 and 3. By applying a method such as this, a set of global values for such linear approximation line parameters such as the slope, \( a \), and the constant, \( b \), can be found by taking into consideration a rather large number of measuring stations in the AHU2 air intake duct system. The linear approximation obtained in this way can then be considered a generalized linear approximation of the curve in question.
References


Appendices

A List of Symbols Used in Matlab Codes

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>a</td>
<td>Slope of line $y_i = ax_i + b$ for station $i$, $i = 2, 3$ and $4$.</td>
</tr>
<tr>
<td>b</td>
<td>Constant of line $y_i = ax_i + b$ for station $i$, $i = 2, 3$ and $4$.</td>
</tr>
<tr>
<td>a_avg</td>
<td>Average value of $a$.</td>
</tr>
<tr>
<td>b_avg</td>
<td>Average value of $b$.</td>
</tr>
<tr>
<td>ak</td>
<td>Grid subarea ($k = 1, 2, 3, \ldots, 65$) at a duct cross section.</td>
</tr>
<tr>
<td>aa(i,j)</td>
<td>Grid subarea (of column $i$ and row $j$, $i \neq j$).</td>
</tr>
<tr>
<td>aa_tot</td>
<td>Total grid area.</td>
</tr>
<tr>
<td>A</td>
<td>External duct cross sectional area.</td>
</tr>
<tr>
<td>Ai</td>
<td>Internal duct cross sectional area at station $i$, $i = 2, 3$ and $4$.</td>
</tr>
<tr>
<td>C</td>
<td>Constant of line $Y_i = X_i + C$ for station $i$, $i = 2, 3$ and $4$.</td>
</tr>
<tr>
<td>D_H</td>
<td>Hydraulic (Equivalent) diameter of the duct.</td>
</tr>
<tr>
<td>e</td>
<td>Relative roughness of the duct.</td>
</tr>
<tr>
<td>fij</td>
<td>Assumed friction factor for stations $i$ and $j$, $i \neq j$.</td>
</tr>
<tr>
<td>fijmid,</td>
<td>Friction factor at midpoint between stations $i$ and $j$, $i \neq j$.</td>
</tr>
<tr>
<td>h0</td>
<td>External height of the cross section for the duct.</td>
</tr>
<tr>
<td>h</td>
<td>Internal height of the cross section for the duct.</td>
</tr>
<tr>
<td>hL_i_mnr</td>
<td>Minor head loss for station $i = 2, 3$ or $4$.</td>
</tr>
<tr>
<td>hL_ij_mjr</td>
<td>Major head loss based on $fijmid$.</td>
</tr>
<tr>
<td>hLij_tot</td>
<td>Total head loss calculated based on $Q_{avg}$.</td>
</tr>
<tr>
<td>hLijmean_tot,</td>
<td>Total head loss calculated based on $\Gamma_{avg,mid}$.</td>
</tr>
<tr>
<td>KL</td>
<td>Corner loss coefficient for a $90^\circ$ bend (miter type) in the duct.</td>
</tr>
<tr>
<td>Lij_Hor</td>
<td>Horizontal length between stations $i$ and $j$, $i \neq j$.</td>
</tr>
<tr>
<td>Lij_Ver</td>
<td>Vertical length between stations $i$ and $j$, $i \neq j$.</td>
</tr>
<tr>
<td>Nij_RH_FOR</td>
<td>Reynolds number based on $Q_{avg}$, $A$, $Nuijmid$ and $D_H$.</td>
</tr>
<tr>
<td>Pi</td>
<td>Static pressure at station $i$, $i = 2, 3$ and $4$.</td>
</tr>
<tr>
<td>PowerLossij_tot,</td>
<td>Power loss based on $hLij_{tot}$.</td>
</tr>
</tbody>
</table>
PowerLossijmean_tot  Power loss based on hLijmean_tot.
Qi  Airflow rate at station i, i = 2, 3 and 4.
Qavg  Average airflow rate for all stations i = 2, 3 and 4.
Qijmid  Airflow rate midpoint (between stations i and j, i ≠ j) average temperatures.
Qviv_tot  Total airflow at station i, i = 2, 3 or 4, based on v_i_v and aa_tot.
Q_T_i  Total airflow rate at station i, i = 2, 3 or 4 using energy coefficient.
Q_T_ijmid  Average total airflow rate at midpoint of stations i and j, i ≠ j.
tii  Initial temperature at station i, i = 2, 3 and 4.
tfi  Final temperature at station i, i = 2, 3 and 4.
Tiavg  Average temperature at station i, i = 2, 3 and 4.
Tijmid  Midpoint (between stations i and j, i ≠ j) of average temperature.
vivk  Air velocity (for grid subarea ak, k= 1, 2, 3, . . ., 65) measured at station i, i = 2, 3 and 4.
v_i_v  Average of (all grid subareas) air velocities measured at each station i, i = 2, 3 or 4.
Vmeani  Mean velocity calculated based on the energy method with energy coefficients at station i, i = 2, 3 or 4.
w0  External width of the cross section for the duct.
w  Internal width of the cross section for the duct.
Xi  Major head loss for station i = 2, 3 or 4.
Yi  Total head loss for station i = 2, 3 or 4.
alphai  Energy coefficient for station i = 2, 3 or 4.
Gammaijmid  Specific weight at the midpoint average temperatures between stations i and j, i ≠ j.
Muijmid  Average dynamic viscosity at the midpoint (between stations i and j, i ≠ j) of average temperatures.
Nuoijmid  Kinetic viscosity at the midpoint average temperatures between stations i and j, i ≠ j.
Rhoijmid  Density at the midpoint (between stations i and j, i ≠ j) of average temperatures.
Matlab code I for Energy Method using the Bernoulli Equation

This is a Matlab code for the calculation of major, minor & total head losses, power loss including a test for the determination of the friction factor.

```matlab
format long
Frequency = 40.8; \% Frequency, all 8 Ac motors running, Hertz
g = 9.80664993437008; \% Gravitational acceleration, m/s^2
ci = 2.54/100.0; \% Convert inches to meters, m
cf = ci*12.0/60.0; \% Convert feet/min to m/s
ccf = 1000.0*g*2.54/100.0; \% Convert one inch of water column to Pascals

w0=78.0*2.54/100.0; \% Outer width of cross sectional area of duct
h0=28.0*2.54/100.0; \% Outer height of cross sectional area of duct
deltax_21Gage=0.0366*ci; \% 21 Gauge Galvanized steel material
w = (w0-2.0*deltax_21Gage); \% Outer width of cross sectional area of duct
h = (h0-2.0*deltax_21Gage); \% Outer height of cross sectional area of duct
A = b*c; \% Cross sectional area of the duct
A2 = A;A3 = A2;A4 = A3; \% Uniform cross sectional area of duct
D_H = 4.0*w*h/(2.0*(w+h)); \% Hydraulic diameter of a duct with a rectangular cross section pipe:

\% Temperature:
ti2 = -0.1;tf2 = 0.3;
T2avg = (ti2 + tf2)/2.0;
ti3 = 0.3;tf3 = 0.9;
T3avg = (ti3 + tf3)/2.0;
ti4 = 0.2;tf4 = 0.8;
T4avg = (ti4 + tf4)/2.0;
T23mid = (T2avg + T3avg)/2.0;
T34mid = (T3avg + T4avg)/2.0;
T24mid = (T2avg + T4avg)/2.0;

\% Static Pressure:
P2 = -0.0897*ccf;
P3 = -0.1672*ccf;
P4 = -0.1882*ccf;

\% Flow rate:
Q2 = 468.0/60.0;Q3 = 460.0/60.0;Q4 = 462.0/60.0;
Qavg =(Q2+Q3+Q4)/3.0;

L23_Hor = 5.95; \% Horizontal length between stations 2 & 3
L23_Ver = 1.75; \% Vertical Length between stations 2 & 3
L22_Ver = 0.0;
L34_Hor = 2.05; \% Horizontal length between stations 3 & 4
L34_Ver = 0.0; \% Vertical Length between stations 3 & 4
```
L33_Ver = 0.0;
L24_Hor = L23_Hor+L34_Hor;  % Horizontal length between stations 2 & 4
L24_Ver = L23_Ver+L34_Ver;  % Vertical length between stations 2 & 4

% Absolute or Dynamic Viscosity of Air in Pa*sec, 1.0 Pa*s = Kg/(m*s)
% = 0.102 Kgf*s/m^2.0:
Mu_0C = 1.72*(10.0^(-5.0));  % @ 0 degrees C
Mu_20C = 1.85*(10.0^(-5.0));  % @ 20 degrees C in Pa*s, 1.0 Pa*s
% = Kg/(m*s)= 0.102 Kgf*s/m^2.0!
Mu23mid = Mu_0C*(T23mid+272.0)/(0.0+272.0);
Mu34mid = Mu_0C*(T34mid+272.0)/(0.0+272.0);
Mu24mid = Mu_0C*(T24mid+272.0)/(0.0+272.0);

% Air density @ 0 deg C in Kg/m^3
Rho_0C = 1.293;
Rho_20C = 1.21;
Rho23mid = Rho_0C*(0.0+272.0)/(T23mid+272.0);
Rho34mid = Rho_0C*(0.0+272.0)/(T34mid+272.0);
Rho24mid = Rho_0C*(0.0+272.0)/(T24mid+272.0);

% To calculate Reynolds number based on average flow between stations:
N23_RH_FOR = Qavg*D_H/(Nu23mid*A);
N24_RH_FOR = Qavg*D_H/(Nu24mid*A);
N34_RH_FOR = Qavg*D_H/(Nu34mid*A);

% Specific Weight of Air @ 0 degrees C in N/m^3.0
Gamma_0C = 12.67;
Gamma23mid = Gamma_0C*(0.0+272.0)/(T23mid+272.0);
Gamma24mid = Gamma_0C*(0.0+272.0)/(T24mid+272.0);
Gamma34mid = Gamma_0C*(0.0+272.0)/(T34mid+272.0);

% Relative Roughness
\( e = 0.045/1000.0; \)  % in meters

% Assumed friction Factor f:
\( f_{23} = 0.0030510; \)
\( f_{24} = 0.0030508; \)
\( f_{34} = 0.0030518; \)
\( cc_{H} = e/(D_{H}*3.7); \)
\( dd_{23\_H} = 2.51/(N23\_RH\_FOR*(f_{23}\*0.5)); \)
\( dd_{24\_H} = 2.51/(N24\_RH\_FOR*(f_{24}\*0.5)); \)
\( dd_{34\_H} = 2.51/(N34\_RH\_FOR*(f_{34}\*0.5)); \)
\( f_{23\_mjr} = (1.0/(2.0*log(cc_{H}+dd_{23\_H})))^2.0; \)
\( f_{24\_mjr} = (1.0/(2.0*log(cc_{H}+dd_{24\_H})))^2.0; \)
\( f_{34\_mjr} = (1.0/(2.0*log(cc_{H}+dd_{34\_H})))^2.0; \)
\( hL_{23\_mjr} = (f_{23\_mjr}*(L23\_Hor+L23\_Ver)/D_{H})*((Qavg/A)^2.0)/(2.0*g); \)
\( hL_{24\_mjr} = (f_{24\_mjr}*(L24\_Hor+L24\_Ver)/D_{H})*((Qavg/A)^2.0)/(2.0*g); \)
\( hL_{34\_mjr} = (f_{34\_mjr}*(L34\_Hor/D_{H}))*((Qavg/A)^2.0)/(2.0*g); \)

KL = 0.2;  % Corner loss coefficient in duct
hL2mnr = KL *((Q2/A)^2.0)/(2.0*g); % 1st corner minor loss for 2
hL3mnr = KL *((Q3/A)^2.0)/(2.0*g); % 2nd corner minor loss for 3
hL4mnr = KL *((Q4/A)^2.0)/(2.0*g); % 2nd corner minor loss for 4

hL23mean_tot = (P2-P3)/Gamma23mid+(((Q2/A)^2.0)-((Q3/A)^2.0))/(2.0*g)+...
L22_Ver-L23_Ver+hL2mnr+hL3mnr;

hL24mean_tot = (P2-P4)/Gamma24mid+(((Q2/A)^2.0)-((Q4/A)^2.0))/(2.0*g)+...
L22_Ver-L24_Ver+hL2mnr+hL4mnr;

hL34mean_tot = (P3-P4)/Gamma34mid+(((Q3/A)^2.0)-((Q4/A)^2.0))/(2.0*g)+...
L33_Ver-L34_Ver;

hL23_t = hL_23_mjr+hL2mnr+hL3mnr;

hL24_t = hL_24_mjr+hL2mnr+hL4mnr;

hL34_t = hL_34_mjr;

PowerLoss23mean_tot = Gamma23mid*Qavg*hL23mean_tot; % in kW
PowerLoss24mean_tot = Gamma24mid*Qavg*hL24mean_tot; % in kW
PowerLoss34mean_tot = Gamma34mid*Qavg*hL34mean_tot; % in kW

hpl23mean_tot = PowerLoss23mean_tot/0.746; % in horse power
hpl24mean_tot = PowerLoss24mean_tot/0.746; % in horse power
hpl34mean_tot = PowerLoss34mean_tot/0.746; % in horse power
Matlab code II for Energy Method using Energy Coefficients

%% This is a Matlab code for the calculation of major, minor & total head losses, power loss including a test for the determination of the friction factor.

\textbf{C} 

\begin{verbatim}
format long
Frequency = 45.0; % Frequency, only 6 Ac motors running, Hertz
g = 9.80664993437008; % Gravitational acceleration, m/s^2
ci = 2.54/100.0; % Convert inches to meters, m
cf = ci*12.0/60.0; % Convert feet/min to m/s
ccf = 1000.0*g*2.54/100.0; % Convert one inch of water column to Pascals
w0 = 78.0*2.54/100.0; % Outer width of cross sectional area of duct
h0 = 28.0*2.54/100.0; % Outer height of cross sectional area of duct
deltax_21Gage = 0.0366*ci; % 21 Gauge Galvanized steel material
w = (w0 - 2.0*deltax_21Gage);
h = (h0 - 2.0*deltax_21Gage);
A = w*h; % Cross sectional area of the duct
A2 = A; A3 = A2; A4 = A3; % Uniform cross sectional area of duct

D_H = 4.0*w*h/(2.0*(w+h));

Temperature:
ti2 = 9.3; tf2 = 8.8;
T2avg = (ti2 + tf2)/2.0;
ti3 = 8.4; tf3 = 8.2;
T3avg = (ti3 + tf3)/2.0;
ti4 = 7.7; tf4 = 7.9;
T4avg = (ti4 + tf4)/2.0;

T23mid = (T2avg + T3avg)/2.0;
T34mid = (T3avg + T4avg)/2.0;
T24mid = (T2avg + T4avg)/2.0;

Static Pressure:
P2 = -0.1098*ccf;
P3 = -0.1845*ccf;
P4 = -0.1928*ccf;

Flow rate:
Q2 = 486.2/60.0; Q3 = 477.4/60.0; Q4 = 479.1/60.0;
Qavg = (Q2+Q3+Q4)/3.0;

L23_Hor = 5.95; % Horizontal length between stations 2 & 3
L23_Ver = 1.75; % Vertical Length between stations 2 & 3
L22_Ver = 0.0;
L34_Hor = 2.05; % Horizontal length between stations 3 & 4
L34_Ver = 0.0; % Vertical Length between stations 3 & 4
\end{verbatim}
L33_Ver = 0.0;
L24_Hor = L23_Hor+L34_Hor;  % Horizontal length between stations 2 & 4
L24_Ver = L23_Ver+L34_Ver;  % Vertical Length between stations 2 & 4
% Absolute or Dynamic Viscosity of Air in Pa*sec, 1.0 Pa*s = Kg/(m*s)
% = 0.102 Kgf*s/m^2.0:
Mu_0C = 1.72*(10.0^(-5.0));  % @ 0 degrees C
Mu_20C = 1.85*(10.0^(-5.0));  % @ 20 degrees C in Pa*s, 1.0 Pa*s
% = Kg/(m*s)= 0.102 Kgf*s/m^2.0!
Mu23mid = Mu_0C*(T23mid+272.0)/(0.0+272.0);
Mu34mid = Mu_0C*(T34mid+272.0)/(0.0+272.0);
Mu24mid = Mu_0C*(T24mid+272.0)/(0.0+272.0);
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
Rho_0C = 1.293;  % Air density @ 0 deg C in Kg/m^3
Rho_20C = 1.21;  % Air density @ 20 deg C in Kg/m^3
Rho23mid = Rho_0C*(0.0+272.0)/(T23mid+272.0);
Rho34mid = Rho_0C*(0.0+272.0)/(T34mid+272.0);
Rho24mid = Rho_0C*(0.0+272.0)/(T24mid+272.0);
% Specific Weight of Air @ 0 degrees C in N/m^3.0
Gamma_0C = 12.67;  Gamma23mid = Gamma_0C*(0.0+272.0)/(T23mid+272.0);
Gamma24mid = Gamma_0C*(0.0+272.0)/(T24mid+272.0);
Gamma34mid = Gamma_0C*(0.0+272.0)/(T34mid+272.0);
% 21Gage
a1 = (5.0-deltax_21Gage)*(6.0-deltax_21Gage)*(ci^2.0);
a2 = (5.0-deltax_21Gage)*6.0*(ci^2.0);
a3_dx = 6.0*(6.0-deltax_21Gage)*(ci^2.0);
a3 = 6.0*6.0*(ci^2.0);
aa(1,1) = a1;
aa(1,2) = a3_dx;
aa(1,3) = aa(1,2);
aa(1,4) = aa(1,3);
aa(1,5) = a1;
aa(1,61) = aa(1,5);
aa(1,62) = a3_dx;
aa(1,63) = aa(1,62);
aa(1,64) = aa(1,63);
aa(1,65) = aa(1,61);
aa = [a1 a3_dx a3_dx a3_dx a3_dx a3_dx a1 a2 a3 a3 a2 a2 a3 a3 a3 a2 a2 a3 a3 a3 a2 a3 a3 a2 a2 a3 a3 a3 a2 a3 a3 a3 a2 a3 a3 a2 a2 a3 a3 a3 a2 a1 a3_dx a3_dx a3_dx a3_dx a1];
aa_tot = sum(aa);
v2v26 = 1128.0; v2v27 = 1152.0; v2v28 = 1152.0; v2v29 = 1136.0; v2v30 = 1064.0; v2v31 = 1181.0; v2v32 = 1131.0; v2v33 = 1128.0; v2v34 = 1163.0; v2v35 = 1185.0; v2v36 = 1032.0; v2v37 = 1183.0; v2v38 = 1101.0; v2v39 = 1171.0; v2v40 = 1088.0; v2v41 = 1108.0; v2v42 = 1143.0; v2v43 = 1178.0; v2v44 = 1092.0; v2v45 = 1143.0; v2v46 = 1029.0; v2v47 = 1136.0; v2v48 = 1102.0; v2v49 = 1150.0; v2v50 = 1010.0; v2v51 = 1086.0; v2v52 = 1197.0; v2v53 = 1142.0; v2v54 = 1187.0; v2v55 = 1113.0; v2v56 = 1094.0; v2v57 = 1162.0; v2v58 = 1220.0; v2v59 = 1032.0; v2v60 = 1049.0; v2v61 = 1181.0; v2v62 = 1196.0; v2v63 = 1171.0; v2v64 = 1052.0; v2v65 = 1025.0;

v2v = cf*[v2v1 v2v2 v2v3 v2v4 v2v5 v2v6 v2v7 v2v8 v2v9 v2v10 v2v11 v2v12
... v2v13 v2v14 v2v15 v2v16 v2v17 v2v18 v2v19 v2v20 v2v21 v2v22 v2v23 v2v24...
 v2v25 v2v26 v2v27 v2v28 v2v29 v2v30 v2v31 v2v32 v2v33 v2v34 v2v35 v2v36...
 v2v37 v2v38 v2v39 v2v40 v2v41 v2v42 v2v43 v2v44 v2v45 v2v46 v2v47 v2v48...
 v2v49 v2v50 v2v51 v2v52 v2v53 v2v54 v2v55 v2v56 v2v57 v2v58 v2v59 v2v60...
 v2v61 v2v62 v2v63 v2v64 v2v65];

v_2_v = sum(v2v)/65.0;
Qv2v_tot = v_2_v*aa_tot;
% in m^3.0/s
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

v3v1 = 912.0; v3v2 = 1058.0; v3v3 = 1177.0; v3v4 = 1187.0; v3v5 = 944.0;
v3v6 = 1086.0; v3v7 = 1139.0; v3v8 = 1255.0; v3v9 = 1189.0; v3v10 = 1015.0;
v3v11 = 1050.0; v3v12 = 1167.0; v3v13 = 1170.0; v3v14 = 1183.0; v3v15 = 1028.0;
v3v16 = 1034.0; v3v17 = 1254.0; v3v18 = 1247.0; v3v19 = 1192.0; v3v20 = 814.0;
v3v21 = 1049.0; v3v22 = 1201.0; v3v23 = 1237.0; v3v24 = 1214.0; v3v25 = 993.0;
v3v26 = 980.0; v3v27 = 1158.0; v3v28 = 1073.0; v3v29 = 1080.0; v3v30 = 915.0;
v3v31 = 1037.0; v3v32 = 1148.0; v3v33 = 1110.0; v3v34 = 1141.0; v3v35 = 985.0;
v3v36 = 1076.0; v3v37 = 1122.0; v3v38 = 1224.0; v3v39 = 1230.0; v3v40 = 1061.0;
v3v41 = 1059.0; v3v42 = 1104.0; v3v43 = 1155.0; v3v44 = 1217.0; v3v45 = 941.0;
v3v46 = 970.0; v3v47 = 1095.0; v3v48 = 1267.0; v3v49 = 1256.0; v3v50 = 1077.0;
v3v51 = 1030.0; v3v52 = 1132.0; v3v53 = 1262.0; v3v54 = 1222.0; v3v55 = 1010.0;
v3v56 = 1054.0; v3v57 = 1168.0; v3v58 = 1251.0; v3v59 = 1237.0; v3v60 = 1046.0;
v3v61 = 957.0; v3v62 = 1088.0; v3v63 = 1248.0; v3v64 = 1216.0; v3v65 = 1054.0;

v3v = cf*[v3v1 v3v2 v3v3 v3v4 v3v5 v3v6 v3v7 v3v8 v3v9 v3v10 v3v11 v3v12...
 v3v13 v3v14 v3v15 v3v16 v3v17 v3v18 v3v19 v3v20 v3v21 v3v22 v3v23 v3v24...
 v3v25 v3v26 v3v27 v3v28 v3v29 v3v30 v3v31 v3v32 v3v33 v3v34 v3v35 v3v36...
 v3v37 v3v38 v3v39 v3v40 v3v41 v3v42 v3v43 v3v44 v3v45 v3v46 v3v47 v3v48...
 v3v49 v3v50 v3v51 v3v52 v3v53 v3v54 v3v55 v3v56 v3v57 v3v58 v3v59 v3v60...
 v3v61 v3v62 v3v63 v3v64 v3v65];

v_3_v = sum(v3v)/65.0;
Qv3v_tot = v_3_v*aa_tot;
% in m^3.0/s
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

v4v1 = 925.0; v4v2 = 1071.0; v4v3 = 1121.0; v4v4 = 1181.0; v4v5 = 1013.0;
v4v6 = 1140.0; v4v7 = 1223.0; v4v8 = 1245.0; v4v9 = 1286.0; v4v10 = 1160.0;
v4v11 = 1062.0; v4v12 = 1049.0; v4v13 = 1095.0; v4v14 = 1070.0; v4v15 = 1049.0;
v4v16 = 1166.0; v4v17 = 1210.0; v4v18 = 1261.0; v4v19 = 1173.0; v4v20 = 933.0;
v4v21 = 1006.0; v4v22 = 1181.0; v4v23 = 1118.0; v4v24 = 1119.0; v4v25 = 1037.0;
v4v26 = 1077.0; v4v27 = 1185.0; v4v28 = 1064.0; v4v29 = 1061.0; v4v30 = 886.0;
v4v31 = 1140.0; v4v32 = 1187.0; v4v33 = 1226.0; v4v34 = 1177.0; v4v35 = 1002.0;
v4v36 = 1118.0; v4v37 = 1119.0; v4v38 = 1121.0; v4v39 = 1091.0; v4v40 = 972.0;
v4v41 = 1016.0; v4v42 = 1029.0; v4v43 = 1237.0; v4v44 = 1213.0; v4v45 = 944.0;

v4v46 = 1056.0; v4v47 = 1082.0; v4v48 = 1229.0; v4v49 = 1297.0; v4v50 = 1103.0;

v4v51 = 991.0; v4v52 = 1165.0; v4v53 = 1267.0; v4v54 = 1233.0; v4v55 = 1089.0;

v4v56 = 1062.0; v4v57 = 1215.0; v4v58 = 1259.0; v4v59 = 1229.0; v4v60 = 1155.0;

v4v61 = 961.0; v4v62 = 1082.0; v4v63 = 1198.0; v4v64 = 1216.0; v4v65 = 1042.0;

v4v = cf*[v4v1 v4v2 v4v3 v4v4 v4v5 v4v6 v4v7 v4v8 v4v9 v4v10 v4v11 v4v12 ...

v4v13 v4v14 v4v15 v4v16 v4v17 v4v18 v4v19 v4v20 v4v21 v4v22 v4v23 v4v24 ...

v4v25 v4v26 v4v27 v4v28 v4v29 v4v30 v4v31 v4v32 v4v33 v4v34 v4v35 v4v36 ...

v4v37 v4v38 v4v39 v4v40 v4v41 v4v42 v4v43 v4v44 v4v45 v4v46 v4v47 v4v48 ...

v4v49 v4v50 v4v51 v4v52 v4v53 v4v54 v4v55 v4v56 v4v57 v4v58 v4v59 v4v60 ...

v4v61 v4v62 v4v63 v4v64 v4v65];

v_4_v = sum(v4v)/65.0;
Qv4v_tot = v_4_v*aa_tot; % in m^3.0/s
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

for i=1:65

AA_2(i) = aa(1,i);
Q_2(i) = v2v(1,i)*AA_2(i);
vv3aa_2(i) = (v2v(1,i)^3.0)*aa(1,i);
AA_3(i) = aa(1,i);
Q_3(i) = v3v(1,i)*AA_3(i);
vv3aa_3(i) = (v3v(1,i)^3.0)*aa(1,i);
AA_4(i) = aa(1,i);

Q_4(i) = v4v(1,i)*AA_4(i);
vv3aa_4(i) = (v4v(1,i)^3.0)*aa(1,i);

end

A_T_2 = sum(AA_2);
Q_T_2 = sum(Q_2);
vv3aa_T_2 = sum(vv3aa_2);
Vmean2 = Q_T_2/A_T_2;
alpha2 = vv3aa_T_2/((Vmean2^3.0)*A_T_2);

A_T_3 = sum(AA_3);
Q_T_3 = sum(Q_3);
vv3aa_T_3 = sum(vv3aa_3);
Vmean3 = Q_T_3/A_T_3;
alpha3 = vv3aa_T_3/((Vmean3^3.0)*A_T_3);

A_T_4 = sum(AA_4);
Q_T_4 = sum(Q_4);
vv3aa_T_4 = sum(vv3aa_4);
Vmean4 = Q_T_4/A_T_4;
alpha4 = vv3aa_T_4/((Vmean4^3.0)*A_T_4);

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% $\alpha_2 = (\alpha_2)^1.0;$
% $\alpha_3 = (\alpha_3)^1.0;$
% $\alpha_4 = (\alpha_4)^1.0;$
% $\alpha_2 = (\alpha_2)^1.5;$
% $\alpha_3 = (\alpha_3)^1.5;$
% $\alpha_4 = (\alpha_4)^1.5;$

% Absolute or Dynamic Viscosity of Air in Pa*sec, 1.0 Pa*s = Kg/(m*s)

% $\mu_0C = 1.72*(10.0^{(-5.0)});$ @ 0 degrees C
% $\mu_{20C} = 1.85*(10.0^{(-5.0)});$ @ 20 degrees C in Pa*s, 1.0 Pa*s = Kg/(m*s)= 0.102 Kgf*s/m^2.0!

$\mu_{23mid} = \mu_{0C}*(T_{23mid}+272.0)/(0.0+272.0);$ $\mu_{34mid} = \mu_{0C}*(T_{34mid}+272.0)/(0.0+272.0);$ $\mu_{24mid} = \mu_{0C}*(T_{24mid}+272.0)/(0.0+272.0);$

% Absolute or Dynamic Viscosity of Air in Pa*sec, 1.0 Pa*s = Kg/(m*s)

% $\rho_{0C} = 1.293;$ Air density @ 0 deg C in Kg/m^3
% $\rho_{20C} = 1.21;$ Air density @ 20 deg C in Kg/m^3

$\rho_{23mid} = \rho_{0C}*(0.0+272.0)/(T_{23mid}+272.0);$ $\rho_{34mid} = \rho_{0C}*(0.0+272.0)/(T_{34mid}+272.0);$ $\rho_{24mid} = \rho_{0C}*(0.0+272.0)/(T_{24mid}+272.0);$

$\nu_{23mid} = \mu_{23mid}/\rho_{23mid};$ $\nu_{34mid} = \mu_{34mid}/\rho_{34mid};$ $\nu_{24mid} = \mu_{24mid}/\rho_{24mid};$

% To calculate Reynolds number based on flow between stations:

$N_{23\_RH\_FOR} = Q_{T\_23mid}*D_H/(\nu_{23mid}*A);$ $N_{24\_RH\_FOR} = Q_{T\_24mid}*D_H/(\nu_{24mid}*A);$ $N_{34\_RH\_FOR} = Q_{T\_34mid}*D_H/(\nu_{34mid}*A);$ $\gamma_{0C} = 12.67;$ Specific Weight of Air @ 0 degrees C in N/m^3.0

Gamma23mid = Gamma_0C*(0.0+272.0)/(T23mid+272.0);
Gamma24mid = Gamma_0C*(0.0+272.0)/(T24mid+272.0);
Gamma34mid = Gamma_0C*(0.0+272.0)/(T34mid+272.0);

% Relative Roughness
e = 0.045/1000.0; % in meters
% Assumed friction Factor f:
f23 = 0.00305946;
f24 = 0.00305721;
f34 = 0.00306042;

$cc_H = e/(D_H*3.7);$ $dd_{23\_H} = 2.51/(N_{23\_RH\_FOR}*(f23^{0.5}));$ $dd_{24\_H} = 2.51/(N_{24\_RH\_FOR}*(f24^{0.5}));$ $dd_{34\_H} = 2.51/(N_{34\_RH\_FOR}*(f34^{0.5}));$

% Vmean_avg = (Vmean2+Vmean3+Vmean4)/3.0;

Vmean_avg = (Vmean2+Vmean3+Vmean4)/3.0;

f23mjr = (1.0/(-2.0*log(cc_H+dd_{23\_H})))^2.0;
f24mjr = (1.0/(-2.0*log(cc_H+dd_{24\_H})))^2.0;
f34mjr = (1.0/(-2.0*log(cc_H+dd_{34\_H})))^2.0;
hL_{23\ mjr} = f_{23\ mjr}\*/((L_{23\ Hor}+L_{23\ Ver})/D_H)^*(((Q_{avg}/A)^2.0)/(2.0*g));

hL_{24\ mjr} = f_{24\ mjr}\*/((L_{24\ Hor}+L_{24\ Ver})/D_H)^*(((Q_{avg}/A)^2.0)/(2.0*g));

hL_{34\ mjr} = f_{34\ mjr}*((L_{34\ Hor}/D_H)^*(((Q_{avg}/A)^2.0)/(2.0*g));

KL = \text{0.2}; \quad \% \text{Corner loss coefficient in duct}

hL_{2mnr} = KL*(((Q/2/A)^2.0)/(2.0*g)); \quad \% \text{1st corner minor loss for 2}

hL_{3mnr} = KL*(((Q/3/A)^2.0)/(2.0*g)); \quad \% \text{2nd corner minor loss for 3}

hL_{4mnr} = KL*(((Q/4/A)^2.0)/(2.0*g)); \quad \% \text{2nd corner minor loss for 4}

hL_{23\ mean\ tot} = (P_2^{}-P_3^{})/\Gamma_{23\ mid}^{}+(((Q/2/A)^2.0)-((Q/3/A)^2.0))/(2.0*g)+...
L_{22\ Ver}^{}-L_{23\ Ver}^{}+hL_{2mnr}^{}+hL_{3mnr}^{};

hL_{24\ mean\ tot} = (P_2^{}-P_4^{})/\Gamma_{24\ mid}^{}+(((Q/2/A)^2.0)-((Q/4/A)^2.0))/(2.0*g)+...
L_{22\ Ver}^{}-L_{24\ Ver}^{}+hL_{2mnr}^{}+hL_{4mnr}^{};

hL_{34\ mean\ tot} = (P_3^{}-P_4^{})/\Gamma_{34\ mid}^{}+(((Q/3/A)^2.0)-((Q/4/A)^2.0))/(2.0*g)+...
L_{33\ Ver}^{}-L_{34\ Ver}^{};

hL_{23\ t} = hL_{\ 23\ mjr}^{}+hL_{2mnr}^{}+hL_{3mnr}^{};

hL_{24\ t} = hL_{\ 24\ mjr}^{}+hL_{2mnr}^{}+hL_{4mnr}^{};

hL_{34\ t} = hL_{\ 34\ mjr}^{};

\text{PowerLoss23mean\ tot} = \Gamma_{23\ mid}^{}*Q_{T\ 23\ mid}^{}*hL_{23\ mean\ tot}; \quad \% \text{in kW}

\text{PowerLoss24mean\ tot} = \Gamma_{24\ mid}^{}*Q_{T\ 24\ mid}^{}*hL_{24\ mean\ tot}; \quad \% \text{in kW}

\text{PowerLoss34mean\ tot} = \Gamma_{34\ mid}^{}*Q_{T\ 34\ mid}^{}*hL_{34\ mean\ tot}; \quad \% \text{in kW}

hpl_{23\ mean\ tot} = \text{PowerLoss23mean\ tot}/0.746; \quad \% \text{in horse power}

hpl_{24\ mean\ tot} = \text{PowerLoss24mean\ tot}/0.746; \quad \% \text{in horse power}

hpl_{34\ mean\ tot} = \text{PowerLoss34mean\ tot}/0.746; \quad \% \text{in horse power}
D     Special Matlab code

The following is the special Matlab code for the linearization of the curve for the plot of hLtot against hLmjr including an m-file for the determination of the friction factor:

```
C34 = 0.0;
C23 = 0.314715136588819+0.304047606208425;
C24 = 0.314715136588819+0.306697246028124;
Xx1 = 0.009239123448373;
Xx2 = 0.034694104429489;
Xx3 = 0.043927558415473;
X1 = hL_34_mjr;
X2 = hL_23_mjr;
X3 = hL_24_mjr;
Yy1 = hL34_t;
Yy2 = hL23_t;
Yy3 = hL24_t;
Y1 = X1+C34;
Y2 = X2+C23;
Y3 = X3+C24;
s = 1;S1 = 10;
matrix1 = zeros(S1,3);
matrix2 = zeros(S1,3);
matrix3 = zeros(S1,3);
w1 = 1;w2 = 1;w3 = 1;
while s<=S1
    gg1 = 2.0*(X1^2.0+Xx2^2.0+Xx3^2.0);
    gg2 = 2.0*(Xx1^2.0+X2^2.0+Xx3^2.0);
    gg3 = 2.0*(Xx1^2.0+Xx2^2.0+X3^2.0);
    hh1 = 2.0*(X1+Xx2+Xx3);ii1 = hh1;jj1 = 6.0;
    hh2 = 2.0*(Xx1+X2+Xx3);ii2 = hh2;jj2 = 6.0;
    hh3 = 2.0*(Xx1+Xx2+X3);ii3 = hh3;jj3 = 6.0;
    Aa1 = [gg1,hh1;ii1,jj1];
    aA1 = [jj1,-hh1;-ii1,gg1];
    invAa1 = (1.0/detAa1)*aA1;
    Aa2 = [gg2,hh2;ii2,jj2];
    aA2 = [jj2,-hh2;-ii2,gg2];
    invAa2 = (1.0/detAa2)*aA2;
    Aa3 = [gg3,hh3;ii3,jj3];
    aA3 = [jj3,-hh3;-ii3,gg3];
    invAa3 = (1.0/detAa3)*aA3;
```

\[ k_1 = 2.0(X_1Y_1 + Xx_2Yy_2 + Xx_3Yy_3); \]
\[ k_2 = 2.0(Xx_1Yy_1 + X_2Y_2 + Xx_3Yy_3); \]
\[ k_3 = 2.0(Xx_1Yy_1 + Xx_2Yy_2 + X_3Y_3); \]
\[ l_1 = 2.0(Y_1 + Yy_2 + Yy_3); \]
\[ l_2 = 2.0(Yy_1 + Y_2 + Yy_3); \]
\[ l_3 = 2.0(Yy_1 + Yy_2 + Y_3); \]
\[ k_{l1} = [k_1; l_1]; k_{l2} = [k_2; l_2]; k_{l3} = [k_3; l_3]; \]
\[ m_{n1} = \text{invAa1} * k_{l1}; m_{n2} = \text{invAa2} * k_{l2}; m_{n3} = \text{invAa3} * k_{l3}; \]
\[ a_2 = m_{n2}(1,1); a_1 = m_{n1}(1,1); a_3 = m_{n3}(1,1); \]
\[ b_2 = m_{n2}(2,1); b_1 = m_{n1}(2,1); b_3 = m_{n3}(2,1); \]
\[ y_1 = a_1 * X_1 + b_1; XX_1 = y_1 - C_{34}; \]
\[ y_2 = a_2 * X_2 + b_2; XX_2 = y_2 - C_{23}; \]
\[ y_3 = a_3 * X_3 + b_3; XX_3 = y_3 - C_{24}; \]

```matlab
if \ y_3 > C_{34} \\
newf_{34mjr} = \text{fzero}(@(z)\text{funct1}(z,N_{34\_RH\_FOR}),0.01); \\
QQavg_{34} = ((2.0*\text{g} * \text{D}_H * \text{A}^2.0/(L_{34\_Hor}+L_{34\_Ver})... \\
*(XX_1/newf_{34mjr}))^0.5; \\
newN_{34\_RH\_FOR} = QQavg_{34} * \text{D}_H/(\text{Nu}_{34mid} * \text{A}); \\
newh_{34\_mjr} = newf_{34mjr}*((L_{34\_Hor}+L_{34\_Ver})... \\
/\text{D}_H)^((QQavg_{34}/\text{A})^2.0)/(2.0*\text{g}); \\
newX_1 = newh_{34\_mjr}; \\
newY_1 = newX_1+C_{34}; \\
N_{34\_RH\_FOR} = newN_{34\_RH\_FOR}; \\
F_{34mjr} = newf_{34mjr}; \\
X_1 = newX_1; \\
Y_1 = newY_1; \\
matrix1(w_{1,:}) = [X_1 Y_1 F_{34mjr}]; \\
w_{1} = w_{1}+1; \]
``` 
```matlab
else \\
X_1 = XX_1; \\
end 
```
```
if \ y_2 > C_{34} \\
newf_{23mjr} = \text{fzero}(@(z)\text{funct2}(z,N_{23\_RH\_FOR}),0.01); \\
QQavg_{23} = ((2.0*\text{g} * \text{D}_H * \text{A}^2.0/(L_{23\_Hor}+L_{23\_Ver})... \\
*(XX_2/newf_{23mjr}))^0.5; \\
newN_{23\_RH\_FOR} = QQavg_{23} * \text{D}_H/(\text{Nu}_{23mid} * \text{A}); \\
newh_{23\_mjr} = newf_{23mjr}*((L_{23\_Hor}+L_{23\_Ver})... \\
/\text{D}_H)^((QQavg_{23}/\text{A})^2.0)/(2.0*\text{g}); \\
newX_2 = newh_{23\_mjr}; \\
newY_2 = newX_2+C_{23}; \\
N_{23\_RH\_FOR} = newN_{23\_RH\_FOR}; \\
F_{23mjr} = newf_{23mjr}; \\
X_2 = newX_2; \\
Y_2 = newY_2; \\
matrix2(w_{2,:}) = [X_2 Y_2 F_{23mjr}]; \\
w_{2} = w_{2}+1; \]
``` 
```matlab
else \\
X_2 = XX_2; \\
end 
```
if y3>C24;
newf24mjr = fzero(@(z)funct3(z,N24_RH_FOR),0.01);
QQavg24 = ((2.0*g*D_H*A^2.0/(L24_Hor+L24_Ver))... *
(XX3/newf24mjr))^0.5;
newN24_RH_FOR = QQavg24*D_H/(Nu24mid*A);
newhL_24_mjr = newf24mjr*((L24_Hor+L24_Ver)... /
D_H)^((QQavg24/A)*2.0)/(2.0*g);
newX3 = newhL_24_mjr;
newY3 = newX3+C24;
N24_RH_FOR = newN24_RH_FOR;
F24mjr = newf24mjr;
X3 = newX3;
Y3 = newY3;
matrix3(w3,:) = [X3 Y3 F24mjr];
matrixab1(s,:) = [a1 b1];
matrixab2(s,:) = [a2 b2];
matrixab3(s,:) = [a3 b3];
w3 = w3+1;
else;
X3 = XX3;
end;
s = s+1;
end;
matrixab1;
matrixab2;
matrixab3;
a_avg = (matrixab2(1,1)+matrixab1(1,1)+matrixab3(1,1))/3.0;
b_avg = (matrixab2(1,2)+matrixab1(1,2)+matrixab3(1,2))/3.0;

% m file:
function fn23 = funct2(f,N_R)
function fn24 = funct3(f,N_R)
function fn34 = funct4(f,N_R)
ci = 2.54/100.0; % Convert inches to meters, m
w0=78.0*2.54/100.0; % Outer width of cross sectional area of duct
h0=28.0*2.54/100.0; % Outer height of cross sectional area of duct
deltax_21Gage=0.0366*ci; % 21 Gauge Galvanized steel material
w = (w0-2.0*deltax_21Gage);
h = (h0-2.0*deltax_21Gage);

% Hydraulic diameter of a duct with a rectangular cross section pipe:
D_H = 4.0*w*h/(2.0*(w+h));
e = 0.045/1000.0; % Relative roughness, m
cc_H = e/(D_H*3.7);

dd23_H = 2.51/(N_R*(f.*0.5));
fn23 = 1.0/(f.^0.5)+2.0*log(cc_H+dd23_H);
dd24_H = 2.51/(N_R*(f.*0.5));
fn24 = 1.0/(f.^0.5)+2.0*log(cc_H+dd24_H);
dd34_H = 2.51/(N_R*(f.^0.5));
fn34 = 1.0/(f.^0.5)+2.0*log(cc_H+dd34_H);