2012 Triennial Code Adoption Comments

ATTACHMENTS TO PUBLIC COMMENTS OF THE
Joint Committee on Energy and Environmental Policy

Regarding OSHPD Proposal to Amend
California Mechanical Code § 602.3.1

HVAC Flexible Duct Pressure Loss Measurements

ASHRAE RP-1333, Final Report

Submitted by
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Texas Engineering Experiment Station
College Station, TX 77843

to

American Society of Heating, Refrigerating and
Air-Conditioning Engineers, Inc.
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TC 5.2 – Duct Design

March 2011

Co-funding Organizations

Air Conditioning Contractors of America (ACCA)
Air Distribution Institute (ADI)
American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE)
Lennox Industries
Texas Utilities (Oncor Electric Delivery)

ASHRAE Handbook Chapters Affected

Fundamentals (2009), Chapter 21 – Duct Design

Manuals Affected

ACCA. 2009. Residential Duct Systems - *Manual D*. Air Conditioning Contractors of America. Washington, DC.

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ASHRAE Papers

1. Culp, C., and D. Cantrill. 2009. “Pressure Losses in 12”, 14”, and 16” Non-Metallic Flexible Ducts with Compression and Sag,” *ASHRAE Transactions*, Volume 115, Part 1.
2. Weaver, K., and C. Culp. 2007. “Static Pressure Losses in Nonmetallic Flexible Ducts,” *ASHRAE Transactions*, Volume 113, Part 2.
3. Uğursal, A., and C. Culp. 2007. “Comparative Analysis of CFD DP vs. Measured DP for Compressed Flexible Ducts,” *ASHRAE Transactions*, Volume 113, Part 1.

ASHRAE Handbook

ASHRAE Handbook (2009), Chapter 21, Figure 8, page 21.7.

Conference Presentations

Uğursal, A., and C. H. Culp. 2008. “The Effects of Geometry on Flexible Duct CFD Simulations,” *Proceedings of the Sixteenth Symposium on Improving Building Systems in Hot and Humid Climates*, Plano, TX, December 15-17.

Uğursal, A., and C. H. Culp. 2007. “CFD Calculations and Measured DP Data for 8 inch Flexible Duct,” ASHRAE Seminar, ASHRAE 2007 Summer Meeting, Long Beach, CA, June.

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Invited Presentations

Culp, C. H., and A. Uğursal. 2008. “HVAC Duct Efficiency, ASHRAE Research Project 1333,” *ASHRAE Austin Chapter Meeting*, Austin, TX, May 15.

Culp, C. H. 2008. “HVAC Duct System Study,” Air Distribution Institute, Spring Meeting, San Antonio, TX, April 14.

Culp, C. H., and K. Weaver. 2005. “HVAC Duct Efficiency Study.” Air Distribution Institute, Las Vegas, NV, April 11.

Culp, C. H. 2004. “Air Duct Measurement System for Computing Losses.” Air Distribution Institute, Chicago, IL, September 20.

Table of Contents

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We also wish to thank our sponsors, who include the Air Distribution Institute (ADI), the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), Texas Utilities, the Air Conditioning Contractors of America (ACCA) and Lennox Industries.

This research investment has had a large impact on Kevin Weaver and David Cantrill, both of whom were major contributors to the data acquisition and analysis. This project supported both of these graduate students as they studied airflow in flexible ducts for their theses. This project also included Simge Andolsun, Sangwon Lee, Ian Nelson, and Ahmet Uğursal, which helped them understand the process involved with data acquisition. In addition, this project partially supported undergraduate students and enabled their participation in funded research. These undergraduate students are: Michael Chadwell, Blake Clements, Daniel Farmer, Jon Hale, Patricia Stroud and Brian Stockard.

An effort on Computational Fluid Dynamics (CFD) was also initiated as a separate project to determine if CFD could provide data which agreed with measured data. This effort was led by Ahmet Uğursal and Charles Culp and funded separately. The results of the CFD modeling of compressed flexible duct are also included in this report.

Chapter 1 - Introduction

In 2002 the Air Distribution Institute (ADI) contracted with Texas A&M University to conduct airflow research focusing on pressure loss comparison between non-metallic flexible ducts and rigid round sheet metal ducts. In order to create broad based industry representations it was decided to invite co-funded participation by other credible industry groups. ASHRAE became a co-funding partner in 2005 with Air Conditioning Contractors of America (ACCA), Air Distribution Institute (ADI), Lennox Industries and Texas Utilities. Each co-funding partner participated in steering committee meetings to direct the research and monitor progress.

The first phase of the project was taking and analyzing (1) data for 6 in., 8 in., and 10 in. rigid sheet metal duct, and (2) data for straight board supported and joist supported configurations of 6 in., 8 in. and 10 in. flexible duct, each with maximum stretch, 4%, 15%, 30% and 45% compression. Sag was accomplished by draping the ducts over two-by-four “joists” and acquiring data with ducts that had minimum sag and maximum sag. Minimal sag involved compressing the duct on a flat surface (i.e. “board”) and then letting the duct sag with the board gently removed. Maximum sag involved setting up minimum sag and then manually extending the sag with a downward force the then releasing and letting the duct retract.

The second phase of the project was (1) to design and fabricate a 6000 cfm air flow measuring chamber, (2) repeat the 6 in., 8 in, and 10 in. board supported tests, and (3) extend the tests by adding 12 in., 14 in., and 16 in. straight board supported flexible duct, again with maximum stretch, 4%, 15%, 30% and 45% compression. Photos of the Phase 2 work follows. No rigid sheet metal and sag (joist supported configurations) were in the Phase 2 scope. Six inch, 8 in. and 10 in. rigid sheet metal and joist supported tests were done during the Phase 1 work.

During the project the Principal Investigator Dr. Culp and his students worked with ACCA’s committee to update their Manual D (“Residential Duct Systems”). Dr. Culp and his students also worked with ADI in the development of an ASHRAE/ADI flexible “Duct Size Calculator.”

Figure 1-1. 12” Rigid Sheet Metal

Figure 1-2. 12” Maximum Stretch Flexible Duct (L: Board, R: Joist)

Figure 1-3. 12” 4% Compression Flexible Duct (L: Board, R: Joist)

Figure 1-4. 12” 15% Compression Flexible Duct (L: Board, R: Joist)

Figure 1-5. 12” 30% Compression Flexible Duct (L: Board, R: Joist)

Figure 1-6. 12” 45% Compression Flexible Duct (L: Board, R: Joist)

Pressure losses through duct systems impact the total system energy efficiency, ranging from residential homes to large commercial buildings. Residential heating and cooling systems consume approximately 29% of residential energy consumption (DOE 2005). The efficiency of residential HVAC equipment has improved as higher Seasonal Energy Efficiency Ratings (SEER) become mandatory, but the efficiency of airflow through flexible duct systems has not kept pace. In 1992, the Department of Energy (DOE) required a minimum SEER value of 10 for all new AC systems. In 2006, the SEER requirements increased to 13 SEER.

Many types of air ducts exist in the market. These include rigid sheet metal, foil-faced duct board, metal spiral flexible duct, and non-metallic helical wire flexible duct. Flexible ducts can be compressed, curved, and bent for use in a variety of configurations. A common type of flexible duct utilizes a galvanized metal helix wire with a laminate adhered to each individual helix (Richards 1988). Due to its flexible nature, many contractors prefer flexible duct due to its ease of installation. Flexible duct may be easily routed around obstacles, while rigid duct requires a variety of angled fittings to make the same turns. Other benefits associated with flexible duct include low cost, sound attenuation and pre-installed duct insulation at the factory.

Flexible duct installations also exhibit several potential shortcomings. Kokayko et al. (1996) stated, “Flexible ductwork is used extensively within the residential HVAC market. These systems are generally sized and laid-out based upon “rules-of-thumb” either learned through direct experience or passed down through the trades. Poorly designed and/or installed flexible duct systems perform below the anticipated level of operation.” Flexible ducts can be installed with unneeded length, excessive compression, and can be forced to make bends. Installed compression ratios have been observed to vary from 10% compression to over 50% compression. Poor installation practices can raise the total pressure loss by as much as a factor of 10. Flexible duct total pressure loss values in ACCA’s Manual D (1995) are correlated with the measured flexible duct data in Figure 7.2-3. The 1995 ACCA data does not include compressibility factors. The 2009 ACCA Manual D discusses the effect of compression, but does not have friction charts for various degrees of flexible duct compression.

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With the popularity of flexible duct systems, an interest by the industry reflects the need in Handbooks and Manuals for configurations typically seen in residential construction, as well as the pressure drop for flexible duct compression. The data by Abushakra (2001, 2002, 2004) and this research (Culp and Cantrill 2009; Weaver and Culp 2007) extend the range of flexible duct data from fully extended to compression ratios up to 45%.

Chapter 2 - Literature Review

A literature survey was conducted at the beginning of this project. Three major publications on non-metallic helical flexible duct predict total pressure loss. The Air Conditioning Contractors of America (ACCA) published their Manual D (1995), which has flexible duct data and example calculations. The source of the data used in ACCA's pressure loss tables was not identified and efforts to find the source of this data were unsuccessful. Manual D (2009) does include the general effects of compression and specifies that the installed compression should not exceed 4%. The second major effort for flexible duct total pressure loss measurements was Kokayko et al. (1996), where the first data with compressed flexible ducts with duct compression up to 10% were reported. The third major effort for flexible duct total pressure loss measurements was Abushakra et al. (2001, 2002, 2004), where the duct compression considered was increased to 30%.

Flexible duct total pressure loss research was done at Integrated Building and Construction Solutions (IBACOS) during the Burt Hill project (Kokayko et al. 1996). IBACOS researchers measured the total pressure loss through straight run flexible duct, flexible duct elbows, and triangular duct board plenum boxes. Straight run duct lengths of 25 ft were tested in fully stretched and 10% compressed configurations. Diameters of 6 in., 8 in., 10 in. and 12 in. were evaluated and testing was conducted with the duct fully supported. Results from the testing showed an increased pressure loss of 35% to 40% for the relaxed ductwork over the fully stretched, with the sheet metal duct experiencing the lowest pressure loss. “The pressure losses associated with the relaxed flexible ductwork were much greater than the losses associated with the taut flexible ductwork” (Kokayko et al. 1996).

Kokayko et al. (1996) also measured elbow pressure loss by bending sections of flexible duct into “elbows of various radii.” The researchers used peg board forms and wooden dowels to form the duct into the various elbows. Tested radius over diameter (R/D) ratios included 0, 1, 1.5, and 2. Tested diameters were again 6 in., 8 in., 10 in., and 12 in. Results of the research indicated that the “published data for flexible ductwork elbows reasonably approximated the measured pressure losses for all ducts except the 12 in. diameter.” The IBACOS researchers did not specifically comment on why the 12 in. behaved differently, but stated that “inconsistency between test procedures may be the source of these differences.”

Triangle terminal box testing (Kokayko et al. 1996) was conducted for 8 in., 10 in., and 12 in. inlet diameters. Outlet diameters ranged between 6 in. and 10 in. For each combination of inlet/outlet three sizes of plenum were evaluated and included small, medium, and large sizes. A small plenum was defined as having the minimum area for connecting the inlet duct, or 2 in. greater than the inlet duct diameter. A medium plenum was 4 in. larger than the inlet duct, and the large plenum was 8 in. larger than the inlet duct. Results of the testing showed that “fitting pressure losses varied depending upon the inlet/outlet duct geometries and plenum box dimensions” (Kokayko et al. 1996). The large plenums exhibited the highest pressure losses while the medium plenums exhibited the lowest pressure losses.

Research was also conducted at the Lawrence Berkeley National Laboratory (Abushakra et al. 2001, 2002, 2004). Abushakra tested 6 in., 8 in., and 10 in. flexible duct on a flat floor at three compression values in draw-through negative pressure configurations. The test configurations included maximum stretch, 15% compressed and 30% compressed flexible duct. The study showed that moderate compression could increase the pressure drop by a factor of four, while further compression increased the pressure drop by a factor close to ten. Abushakra also tested duct board triangle splitter boxes in sizes of 8 in. x 6 in. x 6 in., 10 in. x 8 in. x 8 in., and 10 in. x 8 in. x 6 in., as well as 6 in., 8 in. and 10 in. flexible duct 90° elbows, and 6 in. and 8 in. supply boots.

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The Air Conditioning Contractors of America (ACCA) published *Residential Duct Systems*, Manual D (1995). This manual addresses sizing of duct systems, and introduces common types of equipment used in residential heating and cooling. Manual D includes total pressure loss charts for non-metallic flexible duct, but does not include any compression for the ducts. The pressure loss values for rigid sheet metal are taken from American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) values published in the ASHRAE Handbook - Fundamentals (Duct Design chapter). ACCA Manual D also includes a friction chart for flexible, spiral wire helix core ducts. There are no references to determine the source of the data within the chart. The same friction chart without its source is in the 2009 Manual D.

Research by Harris (1958) centered on leakage testing of very early forms of flexible duct used to transfer ventilation air within coal mines. Harris conducted several porosity and leakage tests on various types of ducts ranging from woven fabrics to soft plastics to PVC impregnated fabrics. These ducts differ from the modern type of residential flexible duct utilizing a galvanized metal helix and Mylar body, but one type of tested duct did utilize a flexible plastic outer layer wrapped around a steel helix of two turns per foot. Harris tested leakage at pressures to 20 in. H_20 in duct lengths of 100 ft with 4 joints. He established a “leakage coefficient” in order to be able to directly compare ducts of different material types. After reviewing the results, Harris established the leakage coefficient for steel duct at 10, although tested steel duct leakage coefficients were as low as 2. The resulting flexible duct leakage coefficients ranged from 2.7 up to 387, with 387 being the upper limit of measurement ability. Ducts constructed from sheet plastic or plastic impregnated fabric performed the best, being grouped into a leakage coefficient range of 2.7 to 22.5. The work by Harris provided previously unknown values for leakage and porosity in 25 types of flexible duct, but did not provide any data on the total pressure losses for these ducts.

Chapter 3 – Equations

SI equations are from ASHRAE Standard 120-1999, Section 9. I-P equations use in this project is presented below.

$T_{wb,amb}$ = Ambient wet-bulb temperature within laboratory, °F
Calculated using the "ASHRAE Psychrometric Calculator" knowing P_b, $T_{db.amb}$ and $T_{dp,amb}$.

Equation 3.1 is used to determine ambient air density given ambient dry-bulb temperature, ambient wet-bulb temperature, and barometric pressure.

P_e Saturated vapor pressure of ambient air **(3.1)**

(I-P) $= (2.96 * 10^{-4}) * (T_{wb,amb}{}^2) - (1.59 * 10^{-2}) * T_{wb,amb} + 0.41$ (in. Hg)

[SI] $= (3.25 * 10^{-3}) * (T_{wb,amb}{}^2) - (1.86 * 10^{-2}) * T_{wb,amb} + 0.692$ [kPa]

The partial vapor pressure of the water in the air is calculated by:

P_p Partial Vapor Pressure **(3.2)**

(I-P) $= P_e - (P_b * ((T_{db,amb} - T_{wb,amb}) / 2700))$ (in. HG)

[SI] $= P_e - (P_b * ((T_{db,amb} - T_{wb,amb}) / 1500))$ [Pa]

The density of the air within the laboratory is calculated by:

ρ_{amb} Ambient air density within laboratory **(3.3)**

(I-P) $= 70.73 * (P_b - 0.378 * P_p) / (53.35 / (T_{db,amb} + 459.67))$ $\left(\frac{lb_m}{ft^3}\right)$

[SI] $= (P_b - 0.378 * P_p) / (0.287 / (T_{db,amb} - 273.2))$ $\left[\frac{kg}{m^3}\right]$

From the ambient density the chamber density at Plane 5 is calculated as follows:

ρ_5 Density of air at Plane 5 (upstream of nozzles) **(3.4)**

(I-P) $= \rho_{amb} * (T_{db,amb} + 459.67) / (T_5 + 459.67) * (P_5 + 13.63 * P_b) / (13.63 * P_b)$ $\left(\frac{lb_m}{ft^3}\right)$

[SI] $= \rho_{amb} * (T_{db,amb} + 273.2) / (T_5 + 273.2) * (P_5 + 1000 * P_b) / (1000 * P_b)$ $\left[\frac{kg}{m^3}\right]$

The ratio of absolute nozzle exit pressure to absolute approach pressure is calculated using the ΔP across the flow nozzles as well as the gas constant, dry bulb temperature, and air chamber density.

α **Alpha ratio** **(3.5)**

(I-P) $= 1 - (\Delta P_n / (P_5 + 13.63 * P_b))$

[SI] $= 1 - (\Delta P_n / (P_5 + 1000 * P_b))$

The viscosity of the air flowing through the nozzle chamber and test section may be assumed as a constant for temperatures between 39 ºF and 90 ºF (4 ºC and 32 ºC).

μ_5 **Dynamic air viscosity upstream of nozzles** **(3.6)**

(I-P) $= (11 + 0.018\ T_5) * 10^{-6}$ $\left(\frac{lb_m}{ft - s}\right)$

[SI] $= (17.23 + 0.048\ T_5) * 10^{-6}$ [Pa-s]

The expansion factor of the air leaving the flow nozzles is used to quantify the interactive pressure change of the air and the flow nozzles:

Y_n **Expansion factor** **(3.7)**

(I-P) $= \left[3.5\ \alpha^{1.43}\left(\frac{1 - \alpha^{0.286}}{1 - \alpha}\right)\right]^{0.5}$

[SI] $= \left[3.5\ \alpha^{1.43}\left(\frac{1 - \alpha^{0.286}}{1 - \alpha}\right)\right]^{0.5}$

Re_d **Reynolds Number at each nozzle throat diameter** **(3.8)**

(I-P) $= 1{,}363{,}000\ (d/12)(\rho_5\ \Delta P_n)^{0.5}$

[SI] $= 70{,}900\ d\ (\rho_5\ \Delta P_n)^{0.5}$

Note: Above equations for Re_d is an approximation, thus eliminating the need for an iterative solution.

> Note: The above SI equation is from ASHRAE Standard 120. The I-P equation below (used in the RP-1333 spreadsheets) yields Re numbers very close.

(I-P) $= \frac{(0.95)(1097)\,Y_n\,d}{(60)(12)\,\mu_5}(\rho_5\ \Delta P_n)^{0.5}$

C_n **Discharge coefficient** **(3.9)**

(I-P) $= 0.9965 - .00653 * \sqrt{\frac{10^6}{Re_d}}$

[SI] $= 0.9965 - .00653 * \sqrt{\frac{10^6}{Re_d}}$

The product C_nA_n sum of each nozzle in the nozzle chamber is determined as follows:

$$\sum_{i=1}^{N} \left(C_{ni}\, A_{ni} \right) = C_{n1}\, A_{n1} + C_{n2}\, A_{n2} + C_{n3}\, A_{n3} + \bullet\bullet\bullet + C_{nN}\, A_{nN} \qquad \textbf{(3.10)}$$

Where N is the number of open (active) nozzles

Then, using the discharge coefficient, the volumetric amount of airflow passing through the nozzle bank and into the system may be determined by Equation 3.11.

Q Volumetric flow rate **(3.11)**

(I-P) $= 1097\, Y_n \sqrt{\frac{\Delta P_n}{\rho_5}} \sum_{i=1}^{N} \left(C_{ni}\, A_{ni} \right)$ $\left(\frac{ft^3}{min} \right)$

[SI] $= 1414\, Y_n \sqrt{\frac{\Delta P_n}{\rho_5}} \sum_{i=1}^{N} \left(C_{ni}\, A_{ni} \right)$ $\left[\frac{L}{s} \right]$

ΔP_{1-2} (xx%) Flexible duct total pressure drop per 100 ft (% compression) **(3.12)**

(I-P) $= \frac{\Delta P_{1-2}(100)}{L_{8-10} + L_{11-9}}$ [in. H_2O per 100 ft (% compression)]

Chapter 4 – Error Analysis

Experimental errors arise from two areas: 1) random errors (also referred to as precision errors), and 2) bias errors (also referred to as systematic errors or offset errors). Random errors mainly include effects from electrical noise, short term changes in measured variables (such as humidity, room temperature, and atmospheric pressure), and short term measurement drift. In this set of duct airflow measurements, bias errors mainly include non-uniform compression, variances in duct manufacturing, potential leakage, linearity of the airflow nozzles, and end-to-end measurement drift (which includes long term drift from the sensors and electronic circuits).

Another offset effect was observed as the airflow increased past approximately 800 fps. The duct that was not constrained would often start an accelerating expansion and would “blow up” from a straight line test duct to a zigzag configuration (saw tooth appearance). Although testing was halted when this occurred, the interior liner was very likely deformed outward before the catastrophic end of the test and would change the results of the pressure loss from the lower flow tests. When the duct was constrained to be straight, the results would shift 10% to 15% from previous runs. Finally, when a duct experienced this catastrophic event, re-straightening the duct also created an offset from prior measurements in the range of 20% to 40%. Fully stretching the duct under test and then recompressing the duct usually brought the results into a range of 5% to 10% when compared to pre-event measurements. This is important since this illustrates the variability that needs to be expected due to the internal morphology of flexible duct. Because most flexible duct has a polymer inner liner, it can exhibit a memory effect based on how the duct was packed, stored and installed. Tests were first run in the fully stretched mode and then compressed to minimize any impact from this memory effect.

Random (precision) errors were minimized by acquiring at least 5000 individual measurements over approximately one (1) minute of time for each data point. This occurs by taking 100 readings each second and calculating the average of those 100 readings. This process then repeats 50 times with each point value stored on disk. These 50 data sets are then averaged to provide one set of values for each data point. Based on the repeatability of numerous identical sets of readings done under the same conditions, random errors are estimated to be well under 0.5%.

Offset (bias) errors in flexible duct airflow measurements can be in the 10% to 15% (estimated, based on observation from different compressions) range in flexible duct measurements. Although care was taken to achieve equal spacing and uniform compression of the inner liner, distances between inner duct helical coils occur when compressing the entire duct consisting of an inner liner, a sheath of insulation (approximately 2” thick), and an outer skin of the single-helix duct liner. Removing the insulation would have solved the uniformity issue but introduced a major issue in that the insulation keeps the inner liner from expanding when airflow pressurizes the duct. Also, variances in actual product between different manufacturers also attribute to bias error. These variations should be studied as additional research.

Pressure loss values for metal duct are well documented in the ASHRAE Handbook (ASHRAE 2009).

4.1 Instrumentation (Sensor) Accuracy

To minimize sensor accuracy issues, all sensors used within the test setup had an accuracy of 1% full scale or better. Pressure sensor accuracy was 0.25% or 0.5%, depending on the sensor range. Sensors were calibrated against NIST-traceable certificates. Error from leakage was minimized by testing for total system leakage prior to data collection. The sensor specifications in Table 4.1-1 show the sensor accuracy.

Table 4.1-1. Instrument Specifications

Sensor	Mfr	Model	Range	% Accuracy	Drift	Remarks
∆Pressure	Dwyer	607-0	0 to 0.1 in. H_20	±0.5% FS	0.5% FS/yr	Note 1
∆Pressure	Dwyer	607-2	0 to 0.5 in. H_20	±0.5% FS	0.5% FS/yr	Note 1
∆Pressure	Dwyer	607-3	0 to 1.0 in. H_20	±0.25% FS	0.5% FS/yr	Note 1
∆Pressure	Dwyer	607-4	0 to 2.0 in. H_20	±0.5% FS	0.5% FS/yr	Note 1
∆Pressure	Dwyer	607-7	0 to 5.0 in. H_20	±0.5% FS	0.5% FS/yr	Note 1
Dew-point Temperature	General Eastern (GE)	Dew-10	-10ºF to 136ºF	±1.0ºF; Repeatability is ±o.1ºF	NS*	Note 2
Dry-bulb Temperature	General Eastern (GE)	Dew-10	-10ºF to 136ºF	±1.0ºF; Repeatability is ±0.1ºF	NS*	Note 2
Barometric Pressure	“Internet”					

Note 1: Differential measurements were done with pressure inputs connected to the two pressure points needed to perform the differential measurement. Non-differential measurements were done with atmospheric pressure as the reference point.

Note 2: NS* - Not specified.

4.2 System Accuracy

By applying sensor accuracy errors to each input variable, the total test apparatus percentage error can be calculated for each test. For the overall testing, the maximum instrument / sensor calculated error was ±1.6%. A second offset error is due to the use of steel ducts to connect the piezometer to the ends of the duct and the steel coupling used to connect two sections of flexible duct. This error is estimated to add less than 2% for the 0% compression test, less than 1% for the 4% compression test, under 0.6% for the 15% compression test, and under 0.5% for the 30% and 45% compression tests. The total estimated test apparatus offset error thus deceases with compression and ranges downward from 3.6% to under 2.1%. When estimated errors in the range of ±20% due to the internal duct liner variations are included, they far overshadow any anticipated offset errors. Changes of pressure loss exceeding ±20% to ±30% were observed in testing.

The total error and repeatability are influenced much more from the individual test setup and operating conditions than the accuracy of the data acquisition system and sensors. This includes variations in the morphology of the internal duct liner and manufacturing tolerances between duct manufacturers. Our experience with variations in inner duct liner morphology shows that this has the largest influence impacting test repeatability. This conclusion is consistent when comparing test configuration errors of ±2.1% to ±3.6% (depending on duct compression) to the estimated ±20% plus errors due to duct configuration and internal duct liner variations. The impact of the internal morphology of the inner liner can only be approximated and cannot be directly measured since it is inside the insulation layer and the outer Mylar sheath.

Chapter 5 - Test Protocols

5.1 ASHRAE Standard 120-1999 Requirements

The test setup (Figure 5.1-1) and calculations were based on ASHRAE Standard 120-1999.

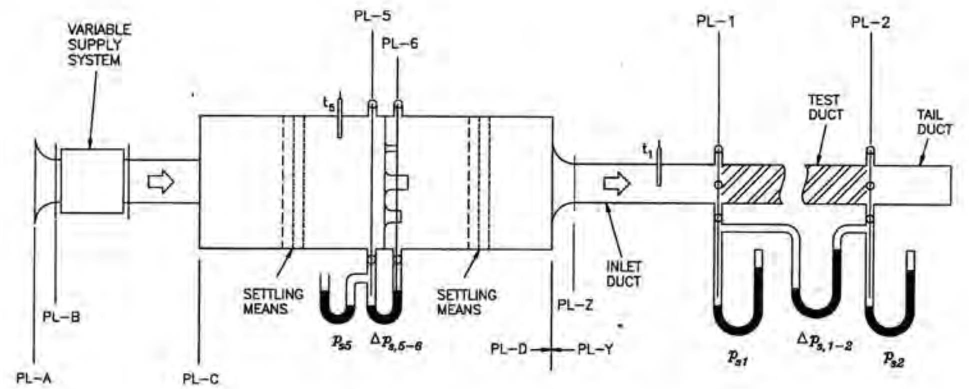

Figure 5.1-1. Straight Duct – Multiple Nozzle Chamber System
(Source: ASHRAE Standard 120, Figure F-5)

5.2 Test Setup

Several steps were taken to ensure the integrity of the test apparatus and validity of the data. First, the test apparatus was assembled and setup to the requirements of ASHRAE Standard 120. Next, all longitudinal and transverse joints were sealed with tape, and then smoothed to eliminate ripples within the tape. The transverse joints also had a second wrap of tape applied, and this wrap was also smoothed. Third, it was ensured that the duct run was straight and that the compression distributed equally as possible along the length of the test section. This was accomplished by first marking the fully stretched duct into equal 1 ft sections. Next, these sections were each compressed the desired amount. The overall test section length was checked to ensure correct compression. Note that with flexible ducts the outer liner is needed to contain the insulation, which impacts the physical morphology of the inner liner. The precision of the inner liner compression cannot be assured, so variations in measurements occurred and are considered normal.

5.3 Leak Testing

This leak test procedure included testing the nozzle board, chamber downstream of the nozzle board, transition piece, flexible duct, and inlet and outlet duct sections as one system.

1. Begin by opening the nozzle chamber access door to gain access to the flow nozzles. Cover all nozzle ends with the nozzle caps. Tape around the caps until they create an airtight seal.

2. Seal the terminal end (tail duct) of the duct system.
3. Connect the compressed air line from an air compressor to the input connection of the Victor flow meter. Connect the output side of the flow meter to the brass fitting in the nozzle chamber.
4. Determine the maximum allowable leakage rate for the current test configuration. ASHRAE Standard 120-1999, Annex D states that leakage should be less than 0.5% of the minimum flow rate to be tested. Determine the equivalent flow rate on the Victor flow meter. The flow meter lists flow in terms of cubic feet per hour (cfh). Divide this value by 60 to determine the flow rate in units of cfm. The cfm reading must be converted to air since the readings on the flow meter are for Argon. In order to convert the reading, determine the density of the selected gas at standard temperature and pressure, and find the ratio of standard air density to standard gas density. For air over Argon this ratio is 0.67. Thus the actual flow through the meter is 0.67 of that displayed.
5. Open the Sensor Monitor Output window and click "Scan" to begin reading the static pressure in the chamber, P_5.
6. Open the valve on the flow meter to introduce air into the system. Increase the fan speed until a pressure of 250 Pa, or 1.0 in. H_2O is reached. Record the flow rate required to reach this pressure and determine if the leakage rate meets the system allowable leakage requirement.
7. If the system cannot attain the required pressure a smoke bomb was used for pinpointing leaks. Leaks were sealed and the system retested.

5.4 Sensor Calibration Procedure

All pressure differential sensors were purchased with 0.25% to 0.5% accuracy. Certification to NIST was included with each Dwyer instrument. These sensors were then rechecked for end-to-end accuracy with a Dwyer Manometer as shown in Figure 5.4-1.

Figure 5.4-1. Dwyer Manometer

Calibration data is shown in Table 5.4-1. Each of these calibrations was measured by a different person. As can be seen, the deviation between the Dwyer and the liquid manometer was almost always within ~0.5% of full scale (FS). This is an end-to-end measurement, meaning that the reported value is subject to all system errors, from the sensor itself through all the electronics. The agreement is within the tolerances of the measurement requirements. In the experimental measurements, each measured data point was the average of 5000 samples, which eliminate random errors with a time constant of less than 15 to 30 seconds. Dwyer Models 607-0, 607-2 and 607-3 are not included in Table 5.4-1 because their ranges were too small to measure with a manometer.

Change terminology in Table 5.4-1 as follows so as to match terminology used by the equations in Chapter 3.

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Deleted: Table 5.4-1. Calibration Data

From	To
P1	P_1
DP-2	$\Delta P_{1\text{-}2}$
PL5	P_5
DPnoz	ΔP_n
PL6	P_6

Table 5.4-1. Calibration Data

	Percentage FS Deviation from Manometer Test 1				
Manometer	P1	DP-2	PL5	DPnoz	PL6
in-H_2O	%	%	%	%	%
1.00	-0.55%	-0.60%	-0.12%	-0.60%	0.05%
0.99	-0.55%	-0.60%	-0.12%	-0.60%	0.00%
0.97	-0.45%	-0.50%	-0.08%	-0.50%	0.10%
0.96	-0.30%	-0.40%	-0.04%	-0.40%	0.20%
0.95	-0.55%	-0.60%	-0.12%	-0.60%	0.00%
0.94	-0.55%	-0.60%	-0.12%	-0.60%	-0.05%
0.93	-0.55%	-0.65%	-0.14%	-0.65%	-0.05%
0.92	-0.50%	-0.60%	-0.10%	-0.60%	0.00%
0.91	-0.45%	-0.50%	-0.08%	-0.50%	0.05%
0.90	-0.50%	-0.60%	-0.10%	-0.55%	0.00%
0.89	-0.50%	-0.55%	-0.12%	-0.55%	0.00%
0.88	-0.55%	-0.60%	-0.12%	-0.55%	-0.05%
0.87	-0.60%	-0.70%	-0.16%	-0.65%	-0.15%
0.86	-0.50%	-0.60%	-0.12%	-0.55%	-0.05%
0.85	-0.50%	-0.60%	-0.14%	-0.60%	-0.05%
0.84	-0.50%	-0.55%	-0.12%	-0.55%	-0.05%
0.83	-0.55%	-0.60%	-0.14%	-0.60%	-0.10%
0.82	-0.60%	-0.70%	-0.18%	-0.65%	-0.15%
0.81	-0.45%	-0.50%	-0.10%	-0.50%	0.00%
0.80	-0.60%	-0.70%	-0.16%	-0.65%	-0.20%
0.75	-0.20%	-0.30%	-0.02%	-0.25%	0.15%
0.70	-0.40%	-0.50%	-0.08%	-0.45%	-0.05%
0.65	-0.35%	-0.45%	-0.08%	-0.35%	-0.05%
0.60	-0.35%	-0.50%	-0.10%	-0.35%	-0.10%
0.50	-0.25%	-0.40%	-0.06%	-0.20%	-0.05%
0.45	-0.10%	-0.25%	0.00%	-0.10%	0.05%
0.40	-0.20%	-0.35%	-0.04%	-0.15%	-0.10%
0.35	-0.35%	-0.55%	-0.12%	-0.35%	-0.30%
0.30	-0.15%	-0.35%	-0.04%	-0.15%	-0.15%
0.25	-0.20%	-0.40%	-0.08%	-0.20%	-0.25%
0.20	-0.20%	-0.35%	-0.06%	-0.15%	-0.25%
0.15	-0.20%	-0.35%	-0.08%	-0.15%	-0.25%
0.10	-0.20%	-0.40%	-0.08%	-0.15%	-0.30%
0.05	0.05%	-0.10%	0.04%	0.15%	0.00%
0.00	-0.15%	-0.30%	-0.04%	-0.05%	-0.25%

	Percentage FS Deviation from Manometer Test 2				
Manometer	P1	DP-2	PL5	DPnoz	PL6
in-H_2O	%	%	%	%	%
0.945	-0.05%	-0.10%	0.08%	-0.10%	0.50%
0.880	0.05%	-0.05%	0.10%	0.00%	0.50%
0.830	0.00%	-0.10%	0.06%	-0.05%	0.40%
0.776	-0.10%	-0.20%	0.04%	-0.15%	0.30%
0.725	0.00%	-0.10%	0.08%	0.00%	0.35%
0.675	-0.10%	-0.25%	0.00%	-0.15%	0.15%
0.630	-0.10%	-0.25%	0.02%	-0.10%	0.15%
0.588	-0.10%	-0.25%	0.00%	-0.15%	0.15%
0.550	-0.20%	-0.35%	-0.04%	-0.20%	0.00%
0.518	-0.25%	-0.40%	-0.04%	-0.25%	-0.10%
0.488	-0.25%	-0.40%	-0.04%	-0.20%	-0.10%
0.466	0.15%	0.00%	0.10%	0.15%	0.25%
0.432	-0.30%	-0.50%	-0.08%	-0.30%	-0.20%
0.405	-0.25%	-0.45%	-0.08%	-0.25%	-0.15%
0.376	-0.30%	-0.50%	-0.10%	-0.30%	-0.25%
0.350	-0.40%	-0.60%	-0.14%	-0.40%	-0.40%
0.330	-0.35%	-0.50%	-0.10%	-0.35%	-0.30%
0.311	-0.35%	-0.50%	-0.10%	-0.30%	-0.30%
0.295	-0.25%	-0.40%	-0.04%	-0.20%	-0.20%
0.273	-0.40%	-0.60%	-0.12%	-0.35%	-0.40%
0.257	-0.35%	-0.55%	-0.10%	-0.35%	-0.40%
0.243	-0.35%	-0.50%	-0.10%	-0.30%	-0.35%
0.230	-0.40%	-0.55%	-0.12%	-0.35%	-0.40%
0.217	-0.35%	-0.50%	-0.10%	-0.30%	-0.35%
0.203	-0.45%	-0.60%	-0.14%	-0.40%	-0.45%
0.193	-0.35%	-0.50%	-0.10%	-0.30%	-0.40%
0.180	-0.40%	-0.60%	-0.14%	-0.35%	-0.45%
0.165	-0.55%	-0.70%	-0.20%	-0.45%	-0.60%
0.157	-0.50%	-0.65%	-0.16%	-0.40%	-0.55%
0.150	-0.45%	-0.65%	-0.16%	-0.40%	-0.55%
0.143	-0.50%	-0.65%	-0.16%	-0.40%	-0.55%
0.132	-0.70%	-0.85%	-0.26%	-0.60%	-0.75%
0.130	-0.40%	-0.60%	-0.14%	-0.35%	-0.50%
0.121	-0.30%	-0.50%	-0.10%	-0.25%	-0.40%
0.112	-0.30%	-0.45%	-0.08%	-0.25%	-0.40%
0.104	-0.35%	-0.50%	-0.10%	-0.25%	-0.40%
0.096	-0.40%	-0.60%	-0.14%	-0.30%	-0.50%
0.090	-0.30%	-0.50%	-0.12%	-0.25%	-0.45%
0.086	-0.30%	-0.45%	-0.10%	-0.20%	-0.35%
0.081	-0.35%	-0.55%	-0.12%	-0.25%	-0.45%
0.070	-0.55%	-0.70%	-0.18%	-0.45%	-0.60%
0.065	-0.50%	-0.65%	-0.18%	-0.40%	-0.60%
0.055	-0.65%	-0.85%	-0.26%	-0.60%	-0.75%
0.050	-0.55%	-0.75%	-0.22%	-0.50%	-0.65%
0.044	-0.60%	-0.75%	-0.22%	-0.50%	-0.65%

Chapter 6 - Results

Flexible duct, board supported, pressure drop tests were done for 6 in., 8 in., 10 in., 12 in., 14 in. and 16 in. duct with the following compression ratios: maximum stretch (0% compression), 4%, 15%, 30%, and 45%. Also evaluated were 6 in., 8 in., and 10 in. flexible duct joist supported with both natural sag and long term sag. The natural sag setup involved compressing the flex duct while being supported on a flat surface (board supported), and then removing the boards so that the duct would sag naturally between joists on 24 in. centers. Long term sag was emulated by manually extending the duct after the natural sag data was taken and allowing it to retract. The reason to emulate long term sag is that ducts are installed and usually last for 30+ years. These ducts will naturally sag under their weight and possibly due to being moved after the ducts are installed. The intent with the long term sag was to provide an indication of what might happen in the long term. The sag data is only to be used as an indication of what might happen since numerous uncontrolled installation variables will impact the resulting pressure loss and should not be used in specifications.

6.1 General

Results for 6 in., 8 in. and 10 in. maximum stretch flex duct and rigid sheet metal duct showed essentially no difference in pressure drop (see Figure 6.1-1).

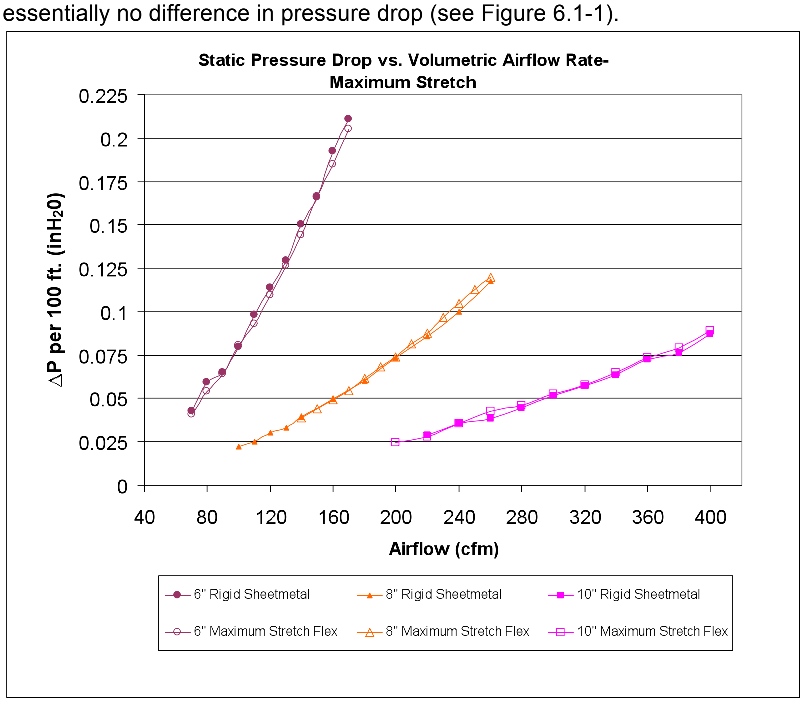

Figure 6.1-1. Static Pressure Drop in 6, 8, and 10” Maximum Stretch Configuration Non-Metallic Flexible Duct Compared with Rigid Sheet Metal Duct (Phase 1)

6.2 Results for 6 in. Duct

The results for the straight run compression done for the 2005 testing (Phase 1) on 6 in. duct are shown in Figure 6.2-1 for the range from 70 cfm (360 fpm) to 170 cfm (870 fpm). Analysis of the resulting data showed that the total pressure losses increase as compression increased. At 150 cfm (760 fpm), a typical design value, the increase in magnitude for 4%, 15%, 30% and 45% compression was roughly 3%, 8%, 16% and 21% respectively. The total pressure loss for 45% compressed long-term joist supported flexible duct at 70 cfm (360 fpm) and 150 cfm (760 fpm) was 1.9 and 8.5 in. H_2O per 100 ft. Note that the 6 in. 4% compression data shows that the board supported data and the natural sag data are essentially the same (≈3%).

The measured sag is shown in Figure 6.2-1, with the "Natural Sag" representing the minimum sag and the "Long Term Sag" emulating the maximum sag. The 0% and 4% compression exhibited no impact from the sag since these were almost fully stretched. The 15% compression data shows approximately a 60% increase in pressure loss (see solid brown arrow). The 30% compression minimum sag data shows approximately a 75% increase in pressure loss. The minimum and maximum sag are shown with the green dotted arrow, with the maximum sag being about 5% higher. The 45% compression of the minimum sag data shows approximately a 75% increase in pressure loss and the maximum sag data shows approximately a 140% increase in pressure loss (see dashed blue arrows).

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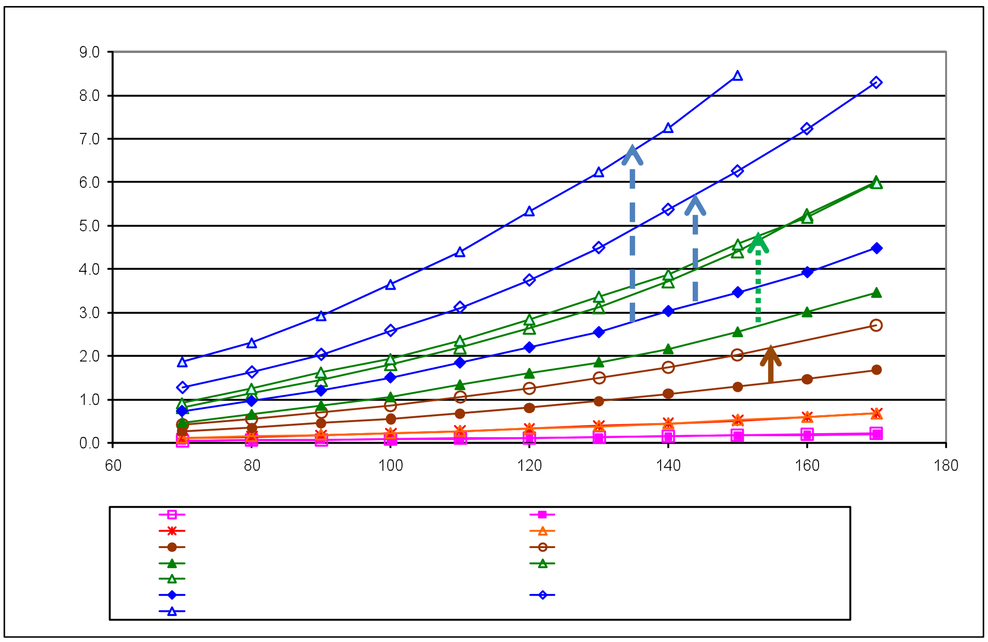

Figure 6.2-1. Total Pressure Drop for 6 in. Duct Measurements (Phase 1: Nov. 2005)

A second set of data was taken on 6/8/09 (Phase 2) to determine the variation that would occur in different ducts as shown in Figure 6.2-2. Straight and linearly compressed (board supported) ducts were measured in this set. The intent was to begin to understand differences in different flexible ducts. In the set of data shown in Figure 6.2-2 the 2005 and 2009 data is within 15% for the fully stretched condition (0% compression). At 4%, 15%, and 30%, the 2005 data is approximately twice the data measured on different ducts in 2009. An interesting effect occurred with the 45% compressed data. The test run measured airflow at 100, 120, 140, and 160 cfm. At 180 cfm (917 fpm), the duct deformed and zigzag occurred as shown by Figure 6.2-3. The duct was originally straight when the velocity of the air through the duct caused it to suddenly expand. The duct was then re-straightened by the technicians and then measurements were taken at 110, 130, 150 and 170 cfm (Figure 6.2-3). Note that the pressure loss increased about 55% for the second set of measurements. The curved lines with the arrows show the sequence for the data acquisition. The deformation also illustrates how significant variation can occur in measurements of flexible ducts. As the airflow increases, the duct is shown to change internally and measurements show that the pressure loss increases. This would cause the pressure loss to increase at a faster rate with increasing cfm that would have been observed if no internal deformation occurred. Re-stretching the duct to the fully stretched configuration for several minutes and then recompressing the duct generally resulted in matching the original data to about 5% to 10%. The attached CD (Appendix D) contains a video of the onset of the zigzag effect.

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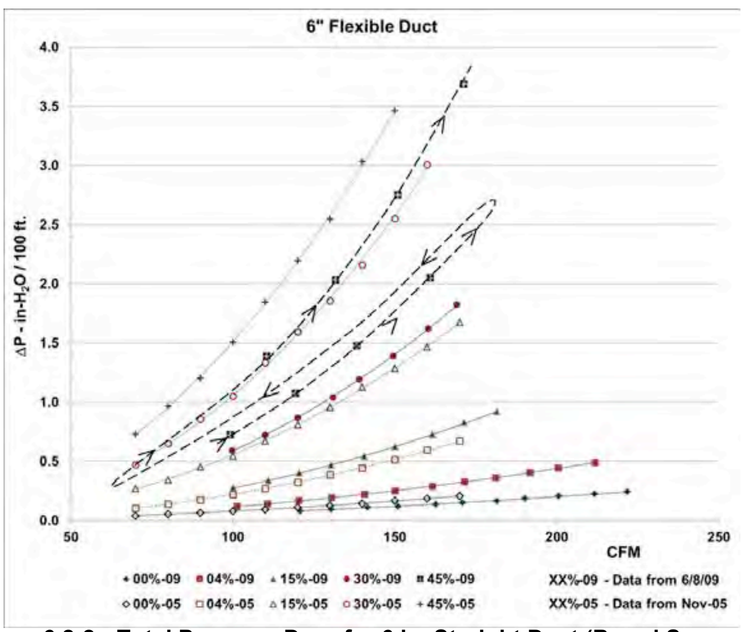

Figure 6.2-2. Total Pressure Drop for 6 in. Straight Duct (Board Supported) Compression (Phases 1 & 2)

Figure 6.2-3. Flex Duct Zigzag Effect

The equations for Figure 6.2-2 are shown in Table 6.2-1. The ΔP-30%-09 and ΔP-45%-09 data are not valid since during the 30% compression data acquisition, the duct suddenly expanded and formed a zigzag pattern, was straightened back out and then the test resumed. This created an offset in the 30% and 45% data, caused by a change in the inner lining configuration of the 6 in. duct.

Table 6.2-1. Equations for 6 in. 2005 (Phase 1) and 2009 (Phase 2) Data

ΔP-00%-05 = 1.913E-05 $(CFM)^{1.808}$	ΔP-00%-09 = 1.496E-05 $(CFM)^{1.793}$
ΔP-04%-05 = 1.466E-05 $(CFM)^{2.098}$	ΔP-04%-09 = 1.490E-05 $(CFM)^{1.942}$
ΔP-15%-05 = 3.924E-05 $(CFM)^{2.075}$	ΔP-15%-09 = 2.125E-05 $(CFM)^{2.054}$
ΔP-30%-05 =4.206E-05 $(CFM)^{2.200}$	ΔP-30%-09 Not Valid
ΔP-45%-05 = 1.232E-04 $(CFM)^{2.044}$	ΔP-45%-09 Not Valid

6.3 Results for 8 in. Duct

The results for the straight run compression done for the 2005 testing (Phase 1) on 8 in. duct are shown in Figure 6.3-1 for the range from 140 cfm (400 fpm) to 340 cfm (970 fpm). Total pressure loss for 8 in. maximum stretch flexible duct at 140 cfm (400 fpm) was 0.04 in. H_2O per 100 ft, while total pressure loss for 45% compressed long-term sag joist-supported flexible duct was approximately 1.6 in. H_2O per 100 ft. This is an increase in magnitude of 40 times. At 220 cfm (630 fpm) the same configurations showed a magnitude increase of 45 times. As in the 6 in. data, the 8 in. 4% data shows that the board supported data and the natural sag data are the same.

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The measured sag is shown in Figure 6.3-1, with the "Natural Sag" representing the minimum sag and the "Long Term Sag" emulating the maximum sag. The 0% and 04% compression again exhibited no impact from the sag since these were almost fully stretched. The 15% compression data shows approximately a 40% increase in pressure loss (see solid brown arrow). The 30% compression minimum and maximum sag data shows approximately a 45% increase in pressure loss (see dotted green arrow) with an additional 100% for long term sag. The 45% compression of the minimum sag data shows approximately a 50% increase in pressure loss and the maximum sag data shows approximately a 150% increase in pressure loss (see dashed blue arrows).

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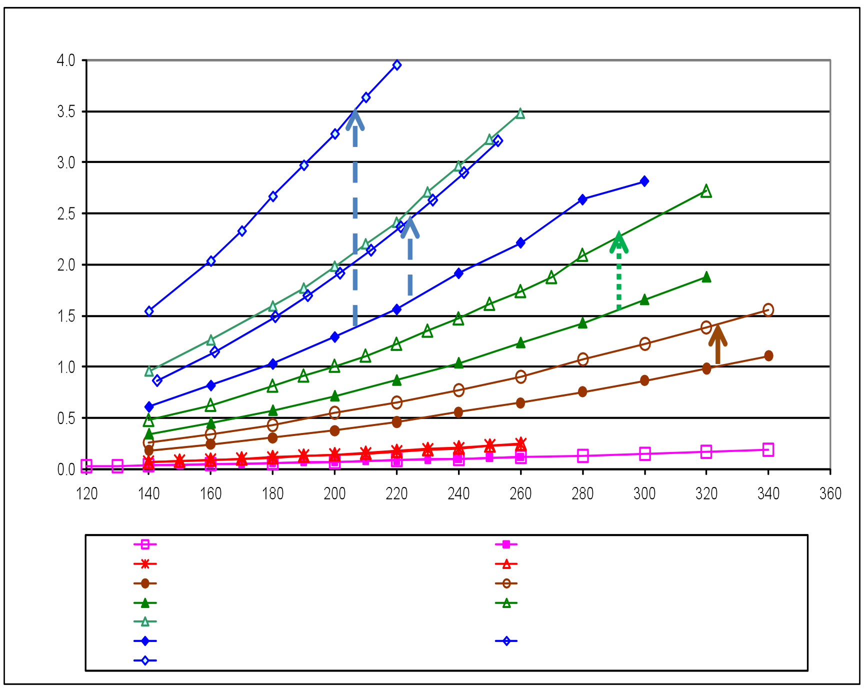

Figure 6.3-1. Total Pressure Drop for 8 in. Duct Measurements (Phase 1: Oct. 2005)

Figure 6.3-2 shows the results of three sets of measurements taken with straight flexible duct in compression. The first measurements were taken in 2005 (Phase 1). The second two sets were taken in 2009 (Phase 2) on two separate days using identical test sequences. Variations from 5% to 15%+ are observed in the 2009 data. The 2005 data is within about 20% when fully stretched, but begins to increase the difference from 4% compression and above. The 45% compression then returns to closer agreement, around 10% variation between the three sets of measurements. The equations for Figure 6.3-2 are shown in Table 6.3-1.

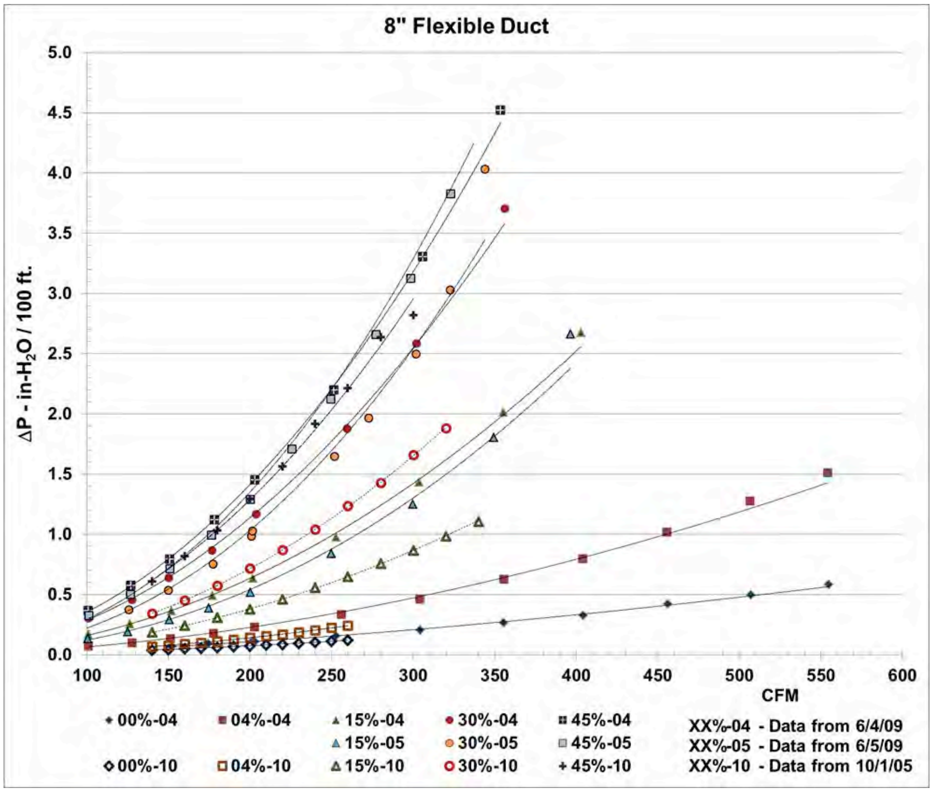

Figure 6.3-2. Total Pressure Drop for 8 in. Straight Duct (Board Supported) Compression (Phases 1 & 2)

Table 6.3-1. Equations for 8 in. 2005 (Phase 1) and 2009 (Phase 2) Data

$\Delta P\text{-}00\%\text{-}04 = 3.438E\text{-}05\ (CFM)^{1.527}$	
$\Delta P\text{-}04\%\text{-}04 = 1.950E\text{-}05\ (CFM)^{1.766}$	
$\Delta P\text{-}15\%\text{-}04 = 1.379E\text{-}05\ (CFM)^{2.028}$	$\Delta P\text{-}15\%\text{-}05 = 5.726E\text{-}06\ (CFM)^{2.162}$
$\Delta P\text{-}30\%\text{-}04 = 3.066E\text{-}05\ (CFM)^{1.986}$	$\Delta P\text{-}30\%\text{-}05 = 8.017E\text{-}06\ (CFM)^{2.221}$
$\Delta P\text{-}45\%\text{-}04 = 3.870E\text{-}05\ (CFM)^{1.982}$	$\Delta P\text{-}45\%\text{-}05 = 1.204E\text{-}05\ (CFM)^{2.195}$
$\Delta P\text{-}00\%\text{-}10 = 4.509E\text{-}06\ (CFM)^{1.833}$	
$\Delta P\text{-}04\%\text{-}10 = 2.764E\text{-}06\ (CFM)^{2.046}$	
$\Delta P\text{-}15\%\text{-}10 = 8.360E\text{-}06\ (CFM)^{2.025}$	
$\Delta P\text{-}30\%\text{-}10 = 1.292E\text{-}05\ (CFM)^{2.061}$	
$\Delta P\text{-}45\%\text{-}10 = 2.457E\text{-}05\ (CFM)^{2.051}$	

6.4 Results for 10 in. Duct

The results for the straight run compression done for the 2005 testing on 10 in. duct are shown in Figure 6.4-1 for the range from 200 cfm (370 fpm) to 400 cfm (730 fpm). Pressure loss for 10 in. maximum stretch flexible duct at 200 cfm (370 fpm) was approximately 0.02 in. H_2O per 100 ft, while total pressure loss for 45% compressed long-term sag joist-supported flexible duct was approximately 1.1 in. H_2O per 100 ft. This is an increase in magnitude of over 55 times. At 290 (530 fpm) cfm the same configurations showed a magnitude increase of over 50 times. Ten inch flexible duct again displayed an increase in total pressure drop as both compression and flow rate increased. As in the 6 in. and 8 in. data, the 10 in. 4% data shows that the board supported data and the natural sag data are the same.

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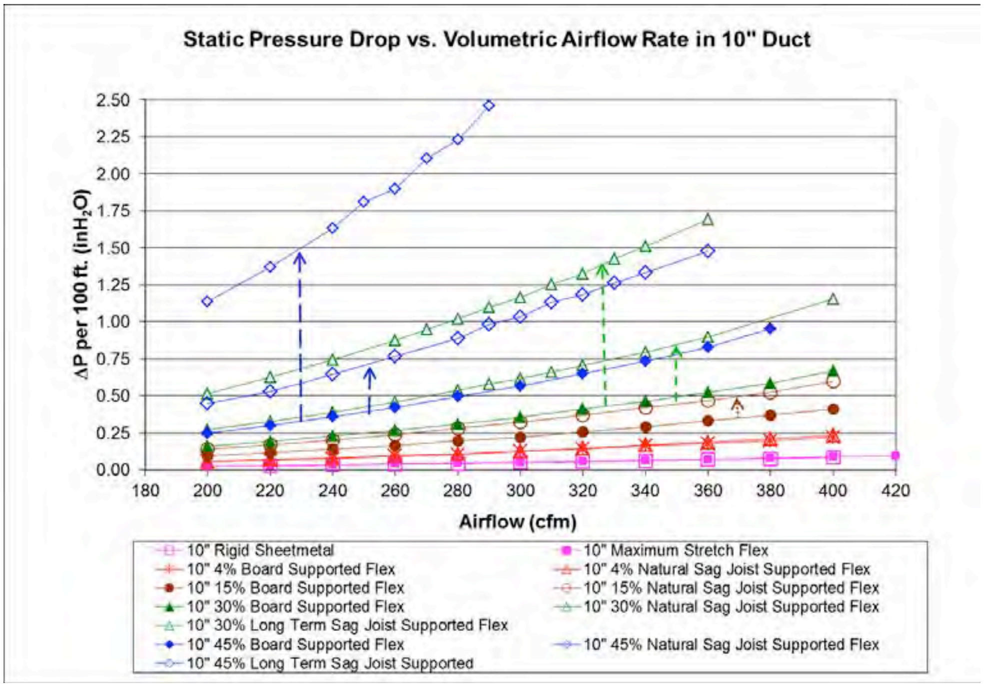

Figure 6.4-1. Total Pressure Drop for 10 in. Duct (Phase 1)

Figure 6.4-2 shows the results for 10 in. flexible duct for 2005 (Phase 1) and 2009 (Phase 2) board supported data. The values vary with a large difference (~50%) with 0% compression; results match very closely at 4% and 15% compression. Results differ by ~30% at 30% compression. Finally, the data more closely matches at 45% compression (~10% difference). The equations for Figure 6.4-2 are shown in Table 6.4-1.

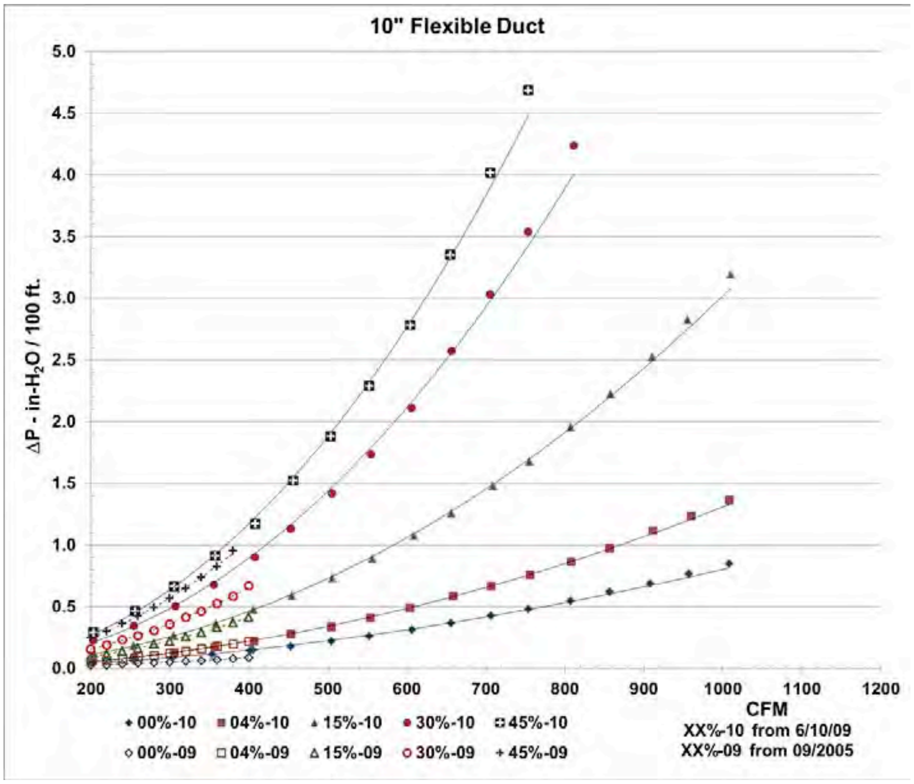

Figure 6.4-2. Total Pressure Drop for 10 in. Straight Duct (Board Supported) Compression (Phases 1 & 2)

Table 6.4-1. Equations for 10 in. 2005 (Phase 1) and 2009 (Phase 2) Data

ΔP-00%-09 = 1.357E-06 $(CFM)^{1.851}$	ΔP-00%-10 = 2.419E-06 $(CFM)^{1.841}$
ΔP-04%-09 = 1.211E-06 $(CFM)^{2.022}$	ΔP-04%-10 = 1.901E-06 $(CFM)^{1.947}$
ΔP-15%-09 = 1.282E-06 $(CFM)^{2.117}$	ΔP-15%-10 = 2.343E-06 $(CFM)^{2.035}$
ΔP-30%-09 = 2.851E-06 $(CFM)^{2.060}$	ΔP-30%-10 = 3.319E-06 $(CFM)^{2.086}$
ΔP-45%-09 = 4.076E-06 $(CFM)^{2.078}$	ΔP-45%-10 = 4.350E-06 $(CFM)^{2.086}$

6.5 Results for 12 in. Duct

The results for the straight run board supported compression testing on 12 in. duct are shown in Figure 6.5-1 at a flow range of 400 (510 fpm) to 1900 cfm (2420 fpm). Pressure loss for 12 in. maximum stretch flexible duct at 600 cfm (760 fpm) was approximately 0.1 in. H_20 per 100 ft, while total pressure loss for 45% compressed flexible duct was approximately 0.9 in. H_20 per 100 ft. This is an increase in magnitude of 9 times. At 1000 cfm (1270 fpm) the same configurations for the 2009 data showed an increase from 0.25 in. H_20 to 2.3 in. H_20 per 100 ft, or about 9 times. The 0% and 4% compressions showed closer agreement (< 10% differences) than the larger compressions. The equations for Figure 6.5-1 are shown in Table 6.5-1.

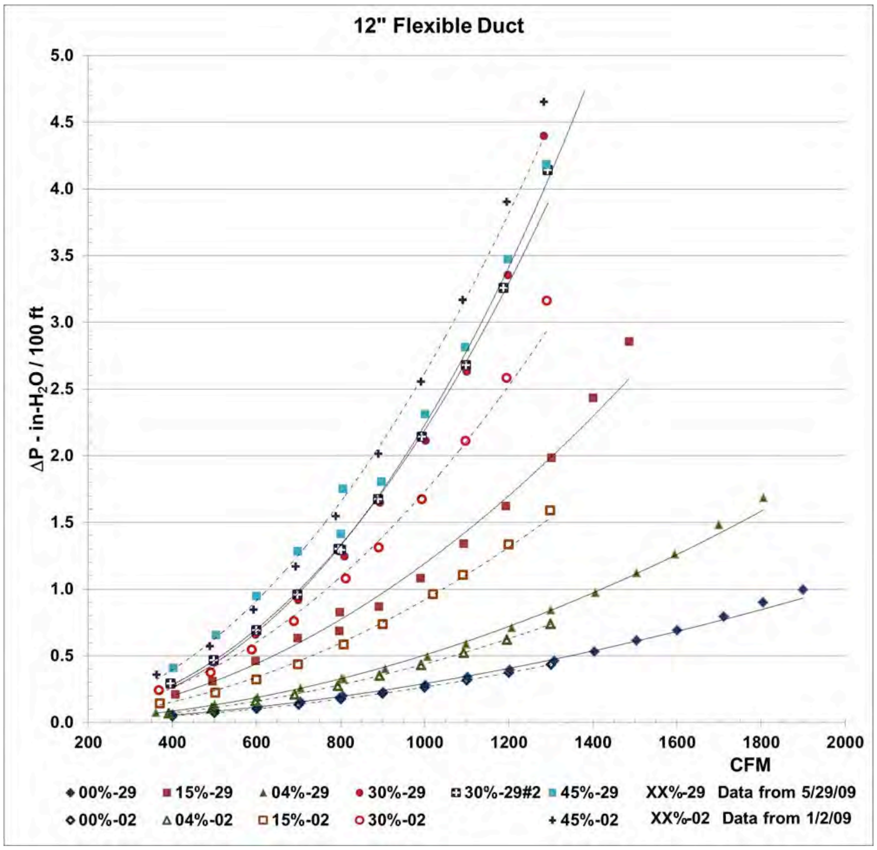

Figure 6.5-1. Total Pressure Drop for 12 in. Straight Duct (Board Supported) Compression (Phase 2)

Table 6.5-1. Equations for 12 in. Data (Phase 2)

ΔP-00%-02 = 6.332E-07 (CFM)$^{1.847}$	ΔP-00%-29 = 9.421E-07 (CFM)$^{1.830}$
ΔP-04%-02 = 5.131E-07 (CFM)$^{1.977}$	ΔP-04%-29 = 5.819E-07 (CFM)$^{1.981}$
ΔP-15%-02 = 1.283E-06 (CFM)$^{1.952}$	ΔP-15%-29 = 1.793E-06 (CFM)$^{1.942}$
ΔP-30%-02 = 1.072E-06 (CFM)$^{2.069}$	ΔP-30%-29 = 2.978E-07 (CFM)$^{2.289}$
	ΔP-30%-29#2 = 5.400E-07 (CFM)$^{2.200}$
ΔP-45%-02 = 1.676E-06 (CFM)$^{2.065}$	ΔP-45%-29 = 5.532E-06 (CFM)$^{1.878}$

6.6 Results for 14 in. Duct

The results for the straight run compression testing on 14 in. duct are shown in Figure 6.6-1 at a flow range of 400 (370 fpm) to 2100 (1960 fpm) cfm. Pressure loss for 14 in. maximum stretch flexible duct at 800 cfm (750) was 0.10 in. H_20 per 100 ft, while total pressure loss for 45% compressed flexible duct was 0.55 in. H_20 per 100 ft. This is an increase in magnitude of 5 to 6 times. At 1500 cfm (1400 fpm) the same configurations showed an increase from 0.30 in. H_20 per 100 ft to 2.0 in. H_20 per 100 ft, or about 6 to 7 times. The grouping of the data is within about 10% to 15% on these three samples. The equations for Figure 6.6-1 are shown in Table 6.6-1.

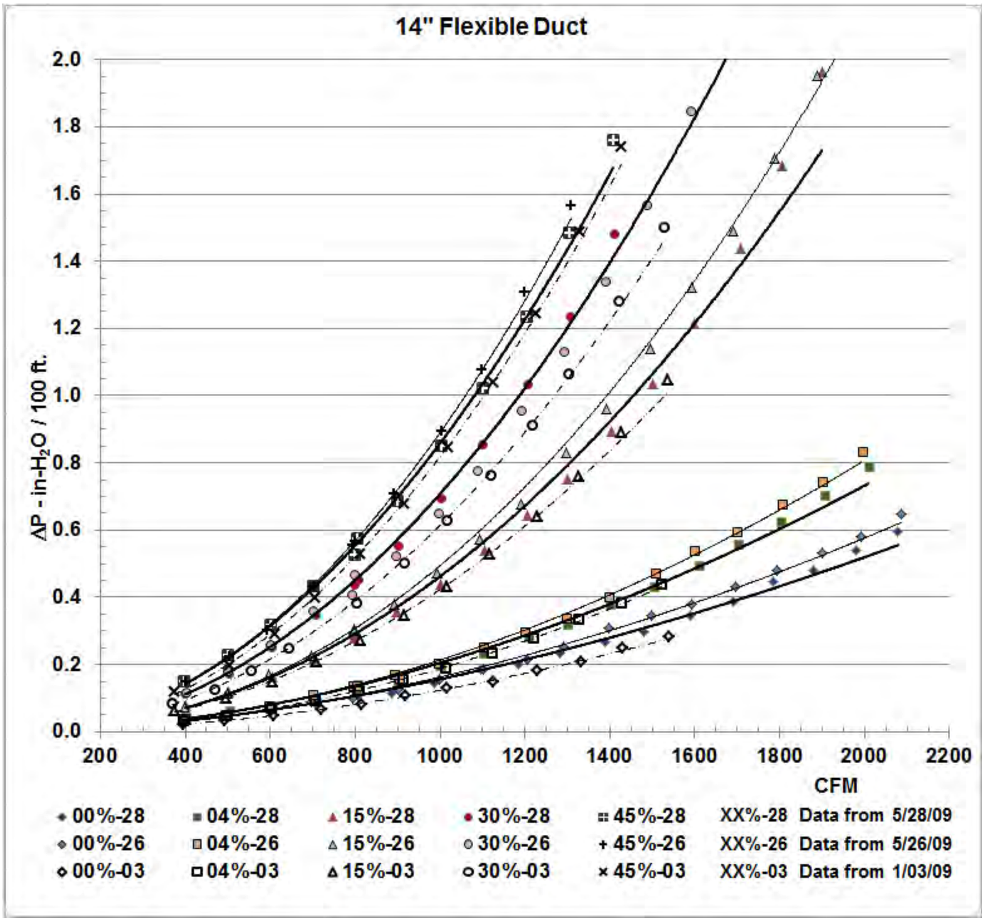

Figure 6.6-1. Total Pressure Drop for 14 in. Straight Duct (Board Supported) Compression (Phase 2)

Table 6.6-1. Equations for 14 in. Data (Phase 2)

$\Delta P\text{-}00\%\text{-}03 = 2.810E\text{-}07\ (CFM)^{1.881}$	$\Delta P\text{-}00\%\text{-}26 = 5.457E\text{-}07\ (CFM)^{1.825}$
$\Delta P\text{-}04\%\text{-}03 = 2.170E\text{-}07\ (CFM)^{1.979}$	$\Delta P\text{-}04\%\text{-}26 = 3.680E\text{-}07\ (CFM)^{1.921}$
$\Delta P\text{-}15\%\text{-}03 = 3.541E\text{-}07\ (CFM)^{2.026}$	$\Delta P\text{-}15\%\text{-}26 = 2.054E\text{-}07\ (CFM)^{2.127}$
$\Delta P\text{-}30\%\text{-}03 = 4.481E\text{-}07\ (CFM)^{2.046}$	$\Delta P\text{-}30\%\text{-}26 = 6.233E\text{-}07\ (CFM)^{2.018}$
$\Delta P\text{-}45\%\text{-}03 = 6.400E\text{-}07\ (CFM)^{2.036}$	$\Delta P\text{-}45\%\text{-}26 = 8.442E\text{-}07\ (CFM)^{2.008}$
$\Delta P\text{-}00\%\text{-}28 = 9.051E\text{-}07\ (CFM)^{1.746}$	
$\Delta P\text{-}04\%\text{-}28 = 6.653E\text{-}07\ (CFM)^{1.833}$	
$\Delta P\text{-}15\%\text{-}28 = 1.310E\text{-}07\ (CFM)^{2.177}$	
$\Delta P\text{-}30\%\text{-}28 = 2.726E\text{-}07\ (CFM)^{2.145}$	
$\Delta P\text{-}45\%\text{-}28 = 5.806E\text{-}06\ (CFM)^{2.062}$	

A zigzag effect was observed on the 14 in ducts as can be seen by Figure 6.6-2. In this example the duct went from a straight line duct to the one in the picture in a few seconds. The gap in the data is likely to be caused by inner liner changes which began in the 30% compressed data. A movie of this is also included in the attachments (Appendix D).

Figure 6.6-2. Zigzag Effect.

6.7 Results for 16 in. Duct

The results for the straight run compression testing on 16 in. duct are shown in Figure 6.7-1 at a flow range of 400 (290 fpm) to 2200 cfm (1580 fpm). Pressure loss for 16 in. maximum stretch flexible duct at 1000 cfm (715 fpm) was approximately 0.1 in. H_20 per 100 ft, while total pressure loss for 45% compressed flexible duct was approximately 0.4 in. H_20 per 100 ft. This is an increase in magnitude of about 4 times. At 1500 cfm (1075 fpm) the same configurations showed an increase from approximately 0.2 in. H_20 per 100 ft to approximately 0.9 in. H_20 per 100 ft, or just over 4 times. The equations for Figure 6.7-1 are shown in Table 6.7-1.

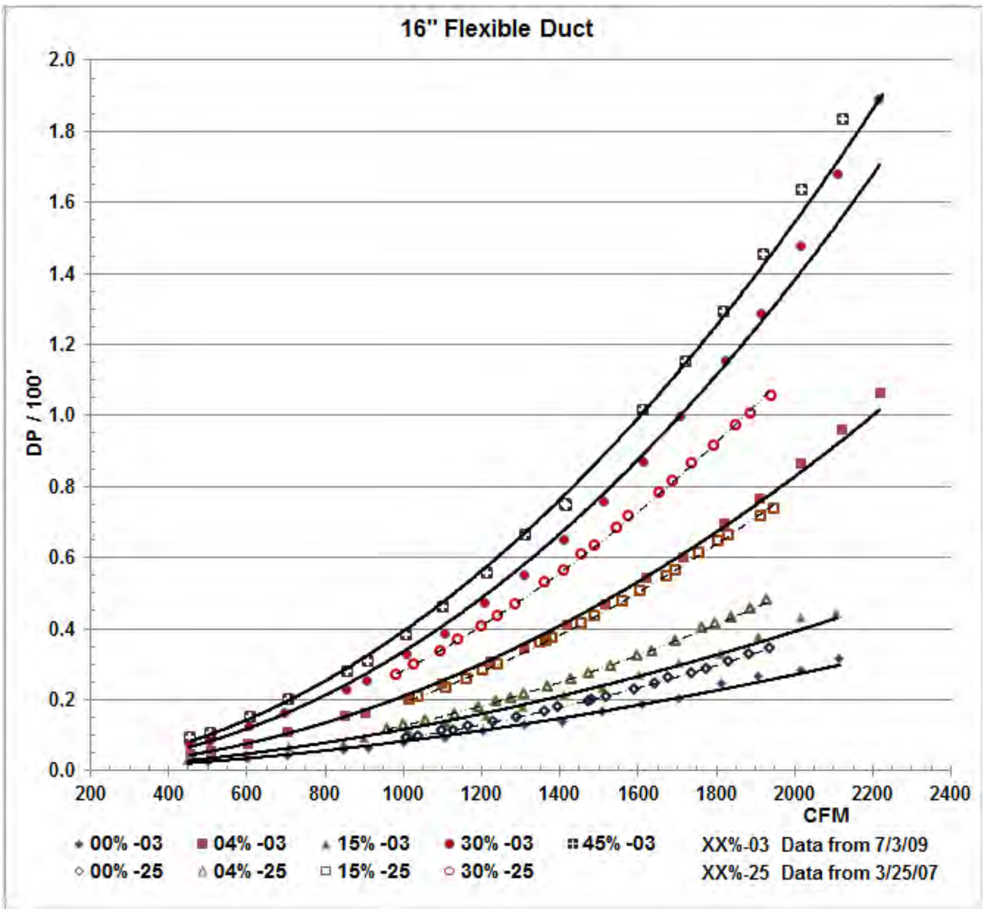

Figure 6.7-1. Total Pressure Drop for 16 in. Straight Duct (Board Supported) Compression (Phase 2)

Table 6.7-1. Equations for 16 in. Data (Phase 2)

$\Delta P\text{-}00\%\text{-}25 = 6.866E\text{-}08\ (CFM)^{2.038}$	$\Delta P\text{-}00\%\text{-}03 = 6.379E\text{-}07\ (CFM)^{1.705}$
$\Delta P\text{-}04\%\text{-}25 = 1.177E\text{-}07\ (CFM)^{2.010}$	$\Delta P\text{-}04\%\text{-}03 = 6.582E\text{-}07\ (CFM)^{1.750}$
$\Delta P\text{-}15\%\text{-}25 = 1.462E\text{-}07\ (CFM)^{2.040}$	$\Delta P\text{-}15\%\text{-}03 = 2.386E\text{-}07\ (CFM)^{1.981}$
$\Delta P\text{-}30\%\text{-}25 = 2.509E\text{-}07\ (CFM)^{2.017}$	$\Delta P\text{-}30\%\text{-}03 = 2.574E\text{-}07\ (CFM)^{2.039}$
	$\Delta P\text{-}45\%\text{-}03 = 4.828E\text{-}07\ (CFM)^{1.971}$

Chapter 7 - Discussion

The following summarizes the major findings and recommendations.

7.1 Major Findings

1. Results for 6 in., 8 in. and 10 in. maximum stretch flex duct and rigid sheet metal duct showed essentially no difference in pressure drop (see Figure 6.1-1). Twelve inch, 14 in. and 16 in. metal ducts were not in the scope of work.

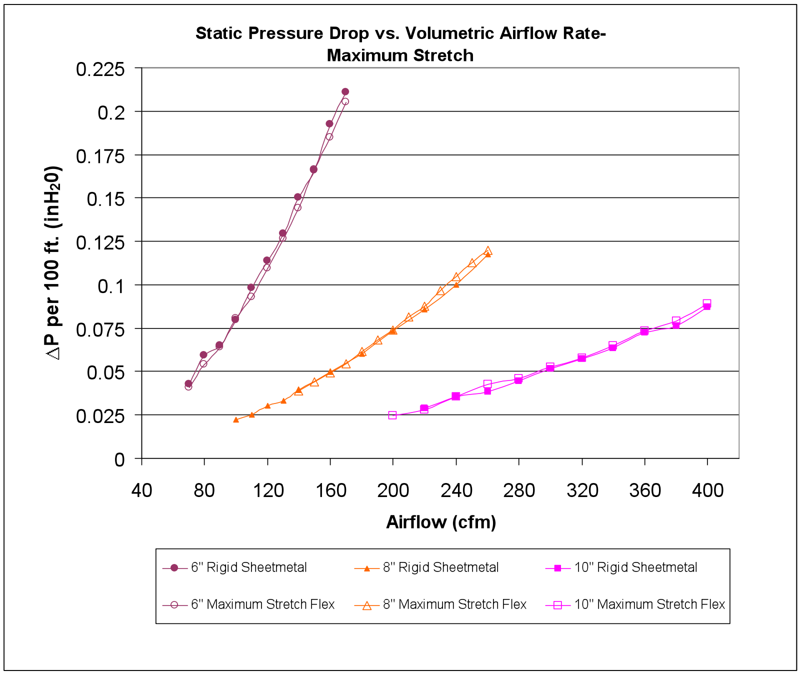

Figure 6.1-1. (Repeated) Static Pressure Drop in 6, 8, and 10" Maximum Stretch Configuration Non-Metallic Flexible Duct Compared with Rigid Sheet Metal Duct (Phase 1)

2. Analysis of the data showed that the pressure losses increased as compression increased as shown by Table 7-1. The ***bold italics red*** is the work of Weaver (Phase 1) and the <u>underlined black</u> numbers are the work by Cantrill (Phase 2). The **bold black** on the bottom of each box below are the previous correlations by LBNL and Texas A & M (Abushakra et al. 2002, 2004; Culp and Cantrill 2009) appearing in the ASHRAE Handbook (2009: Figure 8, page 21.7). <u>On this same page is where ASHRAE states not more then 5 feet of flex should be used.</u>

Table 7.1-1. Summary of Pressure Loss Multipliers

Airflow (Velocity)	Diameter, in.	Flex Duct Compression, Percent			
		4%	15%	30%	45%
150 cfm (760 fpm)	6 in.	<u>2.1</u> to ***3.3*** **2.1**	<u>5.2</u> to ***7.8*** **5.1**	***15.7*** **9.2**	***21.0*** **13.3**
260 cfm (745 fpm)	8 in.	***1.7*** to <u>2.6</u> **1.9**	***4.6*** to <u>7.6</u> **4.2**	***8.7*** to <u>13.6</u> **7.4**	***15.6*** to <u>16.9</u> **10.5**
400 cfm (730 fpm)	10 in.	<u>1.5</u> to ***2.5*** **1.8**	<u>3.1</u> to ***4.7*** **3.5**	<u>6.0</u> to ***7.3*** **5.9**	<u>7.9</u> to ***11.7*** **8.4**
600 cfm (765 fpm)	12 in.	<u>1.6</u> to <u>1.9</u> **1.5**	<u>3.8</u> to <u>4.0</u> **2.9**	<u>5.9</u> to <u>7.0</u> **4.8**	<u>8.0</u> to <u>10.7</u> **6.8**
800 cfm (750 fpm)	14 in.	<u>1.3</u> **1.4**	<u>2.7</u> to <u>2.8</u> **2.5**	<u>4.1</u> to <u>4.2</u> **4.0**	<u>5.2</u> to <u>5.3</u> **5.5**
1000 cfm (715 fpm)	16 in.	<u>1.4</u> **1.3**	<u>2.2</u> to <u>2.5</u> **2.2**	<u>3.2</u> to <u>4.1</u> **3.3**	<u>4.7</u> **4.5**

Italic Bold Red: Phase 1 (Weaver)
<u>Underlined Black</u>: Phase 2 (Cantrill)
Bold Back: Handbook Equation

The amount of variance between measurements was estimated to be around ±20% to ±30%. Variations in pressure loss can be expected for the following reasons:

- **Duct diameters** are not the same. A variation has been reported by manufacturers that variations of up to 0.25 in. or larger occur with different manufacturers. Although a variety of reasons for this have been stated, the net effect is a reduction in pressure loss for the slightly larger sizes.
- **Non-uniform compression.** Establishing uniform compression is difficult in the best of circumstances with ducts having the outer insulation and duct sleeve installed. This

insulation acts as a force to reduce the inflation of the inner liner.

- **Different materials.** Manufacturers use a variety of inner liner materials and coil attachment processes.
- **High flow deformation.** Although this was not viewed directly by inspecting the inner liner, when the duct airflow is taken to the level where the duct deforms to a zigzag type of configuration and then is re-straightened, the duct pressure loss has been seen to increase ~20%..

3. The 0% and 4% compression exhibited no impact from the sag since these were almost fully stretched (Figures 6.2-1, 6.3-1, and 6.4-1). The 15% compression data shows approximately a 60% increase in pressure loss. The 30% compression minimum sag data shows approximately a 75% increase in pressure loss with the maximum sag being anywhere from 5% to 100 % higher. The 45% compression of the minimum sag data shows approximately a 75% increase in pressure loss and the maximum sag data shows approximately a 140% increase in pressure loss.
4. At roughly 900 fpm the flex duct becomes unstable (Figure 6.2-1). As velocity increases the flex duct changes internally with zigzag occurring, and measurements show that the pressure loss increases. See Appendix D for a video of zigzag.

7.2 Recommendations

7.2.1 ASHRAE Handbook

1. The absolute roughness of galvanized steel rigid sheet metal and flex duct fully extended in the ASHRAE Handbook are 0.0003 ft (Medium Smooth Roughness Category) and 0.003 ft (Medium Rough) (ASHRAE 2009; Table 1, page 21.6). It is recommended the flex duct fully extended be listed as “Medium Smooth.”
2. It is recommended that the “Pressure Loss Correction Factor for Flexible Duct Not Fully Extended” figure in the Duct Design chapter (ASHRAE 2009: Fig. 8, page 21.7) be retained with the flex duct range expanded from 6 in. to 16 in. to 4 in. to 24 in. (Figure 7-1). Dash lines are those extrapolated.

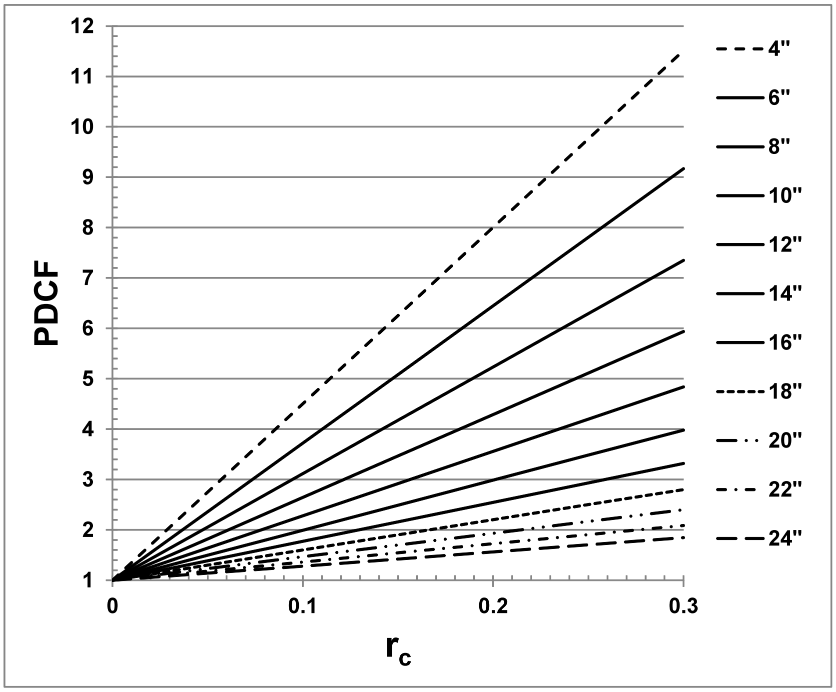

Figure 7.2-1. Pressure Drop Correction Factor for Flexible Duct Not Fully Extended

7.2.2 ACCA Residential Duct Systems - Manual D

1. The ACCA Manual D should have a figure (such as Figure 7.2-1) or a table (Table 7.2-2) that shows the increase in pressure drop as flex duct is compressed. Table 7.2-2 is based on Equation 7-1. Figure 7.2-1 is a graphical representation of the effect of flexible duct compression on pressure drop at 500 fpm from fully extended to 30% compression.

$$\mathrm{PDCF} = 1 + 58\, r_c\, e^{-0.126\, D} \tag{7-1}$$

with $$r_c = 1 - (L / L_{FE}) \tag{7-2}$$

where

PDCF = pressure drop correction factor

r_c = compression ratio, dimensionless

D = flexible duct diameter, in.

L = installed duct length, ft

L_{FE} = duct length fully extended, ft

Table 7.2-2. Pressure Drop Correction Factor

Duct Diameter	Flex Duct Compression				
	0%	4%	15%	30%	45%
4 in.	1.0	2.4	6.3	11.5	16.8
6 in.	1.0	2.1	5.1	9.2	13.3
8 in.	1.0	1.8	4.2	7.4	10.5
10 in.	1.0	1.7	3.5	5.9	8.4
12 in.	1.0	1.5	2.9	4.8	6.8
14 in.	1.0	1.4	2.5	4.0	5.5
16 in.	1.0	1.3	2.2	3.3	4.5
18 in.	1.0	1.2	1.9	2.8	3.7
20 in.	1.0	1.2	1.7	2.4	3.1
22 in.	1.0	1.1	1.5	2.1	2.6
24 in.	1.0	1.1	1.4	1.8	2.3

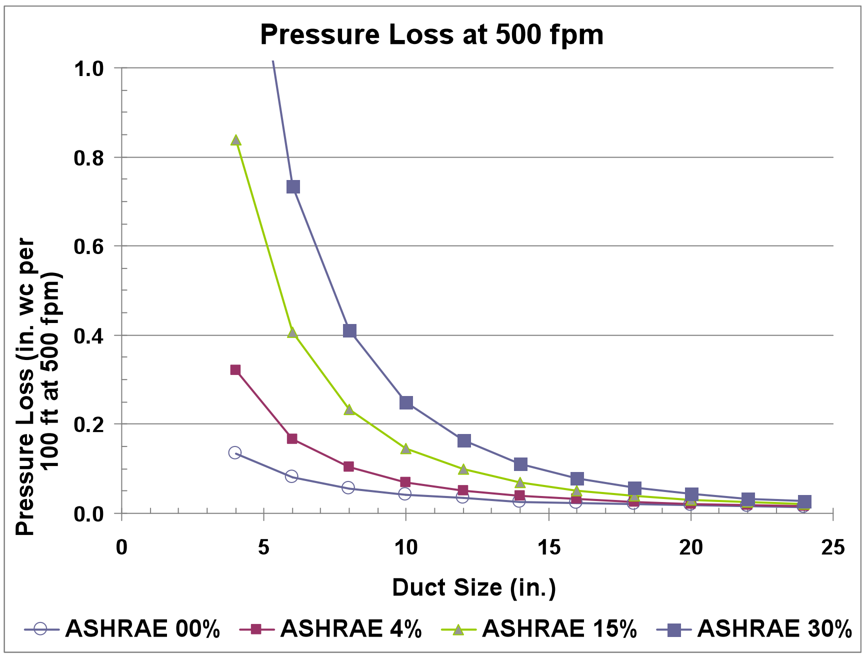

Figure 7.2-2. Effect of Flexible Duct Compression

2. The pressure drop difference at 500 fpm between ACCA's Chart 7 and ASHRAE's base Friction Chart [absolute roughness (0.0003 ft) and standard air density (0.075 lb_m/ft^3)] is shown by Figure 7.2-3. The chart proportions would be the same for other velocities. Values for ACCA were read from their Chart 7. The ASHRAE base Friction Chart values were calculated by an algorithm in ASHRAE's Duct Fitting Database, which are Equations 18 and 19 on page 21.6 of the 2009 ASHRAE Handbook.

   The difference between Chart 7 in ACCA Manual D (Flexible, Spiral Wire Helix Core Ducts) when compared to a base Friction Chart, and the base Friction Chart multiplied by Equation 7-1 for 4% and 15% compression is as follows:

   a. For duct sizes less than 9 in., the ACCA Manual D pressure loss values represent less than 4% compression.
   b. For the 9 in. duct size, the ACCA Manual D pressure loss values represent about a 4% compression.
   c. For duct sizes 10 in. and greater, the ACCA Manual D pressure loss values represent more than 4% compression and less than 15% compression.
   d. For the 20 in. duct size, the ACCA Manual D pressure loss values represent a 15% compression.

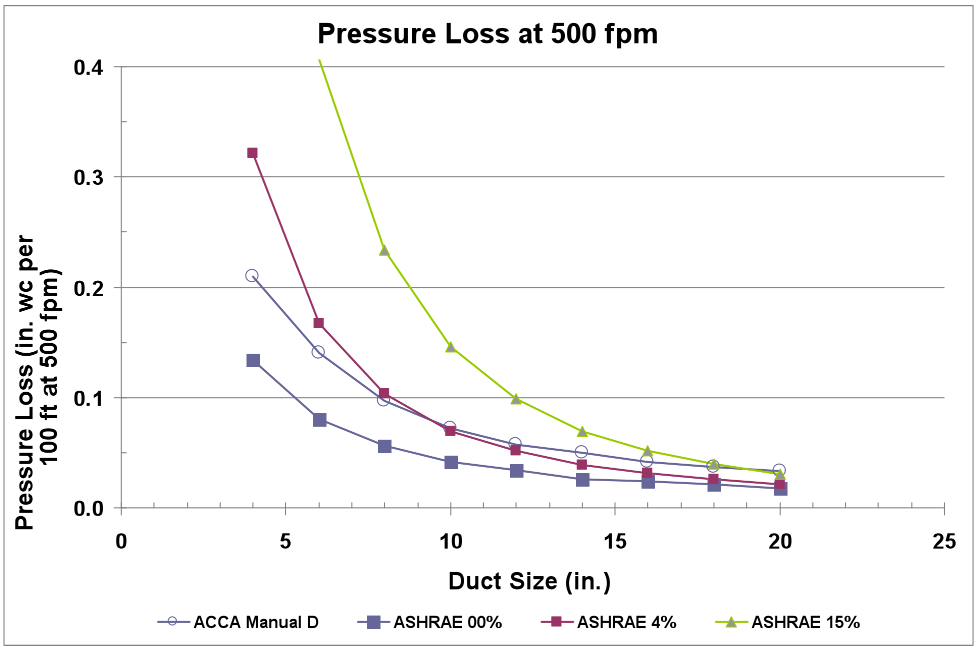

Figure 7.2-3. ASHRAE / ACCA Pressure Loss Comparison

3. Add charts which show the pressure loss per 100 feet for each of the major compression values. This includes 0%, 4%, 15%, 30% and 45%. This would be very useful for graphically showing the pressure drop increase that occurs with increased compression.

7.2.3 Air Distribution Institute (ADI)

1. The ADI "Duct Size Calculator" should be expanded to include 4 in., 18 in., 20 in., and 22 in.
2. It is recommended the title of the calculator be changed from "Duct Size Calculator" to "Flexible Duct Size Calculator."

7.2.4 General

1. Flexible duct should be installed in a fully stretched or as close as reasonable, but not more than 4% compression. One challenge that the industry faces is that improperly installed flexible duct is difficult to detect, since compression is difficult to see or measure when installed.
2. Flexible ducts should not be compressed or forced through constrained building features since this will also increase the pressure loss by choking the airflow.
3. Abushakra et al. (2002) show that loss coefficients for bends in flexible ductwork vary widely from condition to condition, with no uniform or consistent trends. Loss coefficients vary from a low of 0.87 to a high of 3.27. Flexible duct elbows should not be used in lieu of rigid elbows. For comparison purposes an 8 in. die stamped 90º elbow with a centerline r/D ratio of 1.5 the loss coefficient is 0.11.
4. Flexible ducts should be sealed properly. Tape and mastic used to close and seal flexible air ducts and flexible air connectors shall be listed and labeled to UL 181B, Part 1 or Part 2 and shall be marked "181B-FX" for pressure-sensitive tape or "181B-M" for mastic. Mechanical fasteners for use with nonmetallic flexible air ducts shall be either stainless-steel worm-drive hose clamps or non-metallic straps listed and labeled to UL 181B, Part 3, and marked "181B-C." Non-metallic mechanical fasteners shall have a minimum tensile strength rating of 150 lb. force. When non-metallic mechanical fasteners are used, beaded fittings are required, and the maximum duct positive operating pressure shall be limited to 6 in. w.c.
5. As a subject for future work, it is recommended that ASHRAE study actual installations. As part of the preparation for this project, numerous housing and industrial installations were reviewed. In this limited survey, it was found that every installation was not in compliance with Manual D and ADC (2003) requirements.

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Symbols and Subscripts

A	=	Area, ft^2 (m^2)
A_n	=	Nozzle throat area, ft^2 (m^2)
C_n	=	Nozzle discharge coefficient, dimensionless
D	=	Diameter, in. (mm)
d	=	Nozzle throat diameter, in. (m)
$L_{x-x'}$	=	Length of duct between planes, ft (m)
P_b	=	Barometric pressure, in. Hg (kPa)
P_e	=	Saturated vapor pressure of ambient air, in. Hg (kPa)
P_p	=	Partial vapor pressure of ambient air, in. Hg (kPa)
P	=	Static pressure, in. H_2O (Pa)
P_5	=	Static pressure recorded before nozzle bank, in. H_2O (Pa)
Q	=	Volume flow rate, cfm (L/s)
Re_d	=	Reynolds number at nozzle throat diameter, dimensionless
T	=	Temperature, ºF (ºC)
T_{wb}	=	Wet-bulb temperature of air within laboratory, °F (°C)
Y_n	=	Nozzle expansion factor, dimensionless
ΔP_n	=	Static pressure difference across nozzle, in. H_20 (Pa)
α	=	Ratio of absolute nozzle pressure to absolute approach pressure, dimensionless
μ	=	Dynamic pressure of air, lb_m/ft-s (Pa-s)
ρ	=	Air density, lb_m/ft^3 (kg/m^3)

Subscripts

amb	ambient air
d	duct
db	dry-bulb
dp	dew-point
n	nozzle
wb	wet-bulb
1,2,5,6	planes

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Appendix A – Test Apparatus

A.1 General

The test apparatus was designed to allow testing to range from approximately 50 cfm to 5000 cfm. To accomplish this, two chambers were constructed to perform the needed airflow measurements. The small chamber was designed to provide up to 800 cfm and was used for duct diameters of 6 in., 8 in. and 10 in. The large chamber was designed to allow up to 5000 cfm and was used for duct diameters of 12 in., 14 in. and 16 in.

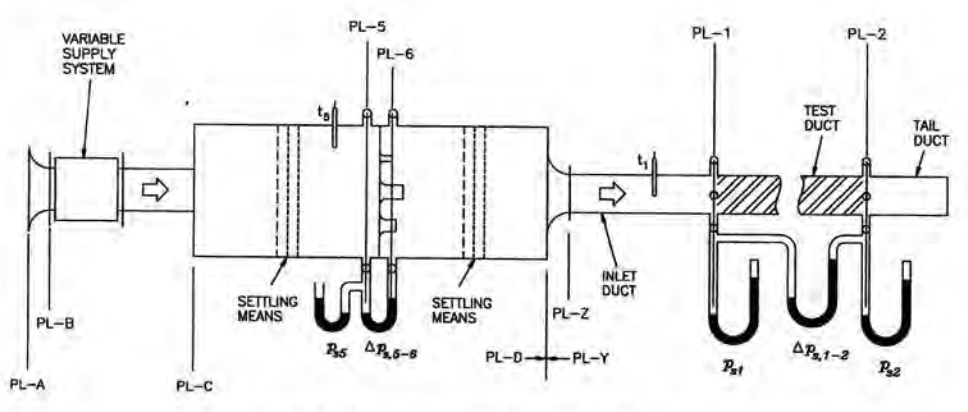

Figure A.1-1. Test setup (Figure 5.2-1 repeated)

The test apparatus is a blow-through system as shown by Figure A.1-1. This system was constructed following the requirements of ASHRAE Standard 120. Computer control using a National Instruments input/output control board in a personal computer was used to process all measurements and to control the VFD to supply the required cfm of air through the system.

A.2 Small Chamber

The small nozzle chamber (2 ft diameter x 5 ft length) was constructed to specifications set forth in Section 7 of ASHRAE Standard 120. The chamber (Figure A.2-1) utilizes two flow nozzles, a 2.5 in. and a 5 in. The nozzles are ANSI-24c compliant, using a low beta ratio design as specified in Section 6 of ASHRAE Standard 120. Sixteen (16) gauge galvanized steel was used for the shell of the chamber, and the nozzle plate was ¼ in. steel plate. The chamber has six flow straightness made from perforated steel, three upstream and three downstream of the nozzle board. The perforations have ½ in. diameter holes equally spaced. The straighteners were welded perpendicularly into the shell of the chamber, each equally spaced 2 in. from one another. A 14 in. x 18 in. access door was cut into the chamber to allow nozzle access. The seams around the door used a rubber gasket in conjunction with silicone sealant to make the door air tight. A rubberized coating was applied to the chamber after completion to fill any surface leaks, and improve the aesthetics of the unit. Twenty (20) gauge galvanized steel transitions were designed to transition the air from the blower cabinet to the nozzle chamber. The chamber uses three interchangeable nose cones to transition the air into the inlet duct. These nose cones allow the chamber to be adapted to 6 in., 8 in., and 10 in. ducts.

Figure A.2-1. Small Nozzle Chamber and Associated Transition Pieces

The VFD Controller and the Pressure Transducer Panel are shown in Figure A.2-2. A computer controlled variable frequency drive (VFD) adjusts the airflow. The VFD provides for varying the fan RPM to provide a range of 50 to 800 cfm. A voltage signal produced from one of the analog output (AO) channels of the DAQ card controls the VFD. It then delivers an operating RPM proportional to that voltage. Figure A.2-3 shows a picture of the data acquisition system (DAQ) with the airflow chamber, computer monitor and sensor cabinets.

Figure A.2-2. Variable Frequency Drive

Figure A.2-3. Data Acquisition System

A.3 Large Chamber

The large chamber is approximately 12 ft long, 60 in. diameter and weighs over 1200 pounds. This chamber was designed to provide over 6000 cfm of air. Figure A.3-1 shows the construction of the chamber. The chamber is 16 gauge galvanized steel built in eight sections. The cylindrical sections have a two inch angle ring welded to the chamber. A urethane gasket is between sections, and the angle rings are connected by ¼ in. bolts. The interior and exterior surfaces are painted. The nozzle board consists of seven nozzles: one 7 in., two 6 in., two 4 in. and two 3” bore. Three flow straightening screens are installed on each side of the nozzle board.

Figure A.3-1. Large Chamber

The identical control system was used on the small chamber setup as the large chamber. The VFD was a Toshiba 30kW unit with a 480 VAC input. A 15 HP motor blower assembly supplied air to the large chamber as shown by Figure A.3-2.

a. Toshiba VFD

b. Blower Assembly

Figure A.3-2. Large Chamber VSD and Blower

The layout of the nozzles is shown in Figure A.3-3. With two 6 in. nozzles, airflow of over 2000 cfm was obtained. With one 3 in. nozzle the chamber provides a low flow of 100 cfm. Although the chamber was only operated up to 3000 cfm with all nozzles open the chamber is designed to deliver 6000 cfm. These nozzles were purchased from the Helander Metal Spinning Company.

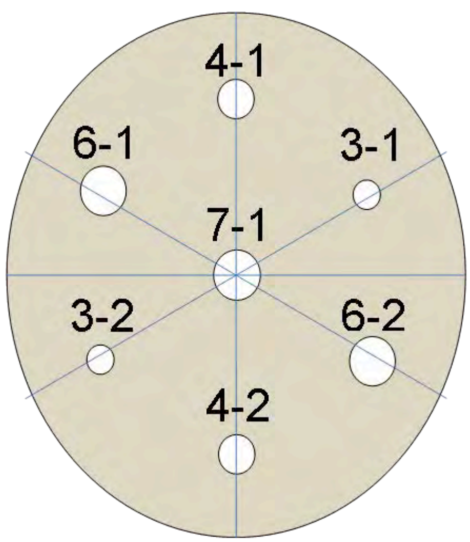

Figure A.3-3. Large chamber nozzle plate

A.4 Sensors

An array of Dwyer Series 607 pressure transmitters were mounted in a panel as shown by Figure A.4-1. These 4 – 20 mA transmitters produce a current proportional to the amount of applied pressure. A 249.0 Ω resistor (±0.25%) converts the current loop outputs from the sensors to voltage inputs to the Data Acquisition System (DAQ). The DAQ processes the voltages and the program in the computer performs the requisite calculations and display functions.

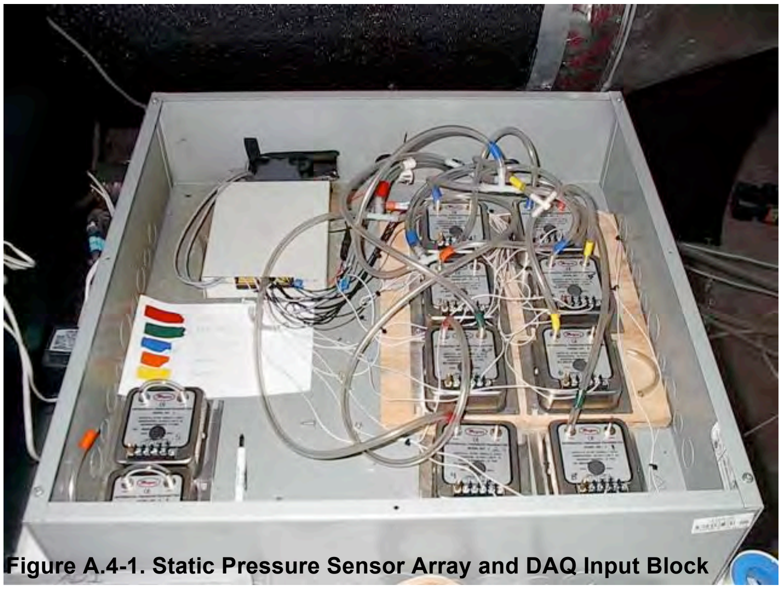

Figure A.4-1. Static Pressure Sensor Array and DAQ Input Block

The static pressure measurement in the system occurs through pressure taps set up in piezometer rings. Each piezometer ring utilizes four equally spaced static pressure taps around the circumference of the test duct. Four taps are set at 90° from each other and connected to the same instrument air line to the pressure sensor. In this manner any small variation between pressure taps averages out. ASHRAE Standard 120 requires that each tap within the ring record a static pressure value ±2% in reference to one another, be spaced 90° around the circumference of the duct from one another, and be located on the same cross-sectional plane. The piezometer rings used in this experiment satisfy the requirements set forth in ASHRAE Standard 120.

The rings were constructed from 24-gauge copper plate and 1/4 in. outside diameter copper tubing. The copper plate provides excellent malleability along with the ability to shape to various duct curvature. The copper tubing provides a tight seal when covered with 1/8 in. silicone tubing. Fabrication of the piezometers began by cutting the copper sheet into rectangles measuring approximately 3 in. x 3 in. This size provides sufficient area to shape to duct curvature while still fitting all four taps around 6 in. to 16 in. ducts. The copper tubing was cut into several 1 in. lengths using a tubing cutter. The drill hole in the Cu plate was deburred and squared with the surface. The tubing was then soldered to the Cu plate centered on the hole. The basic parts are shown below in Figure A.4-2. A complete and installed piezometer ring can be viewed in Figure A.4-3.

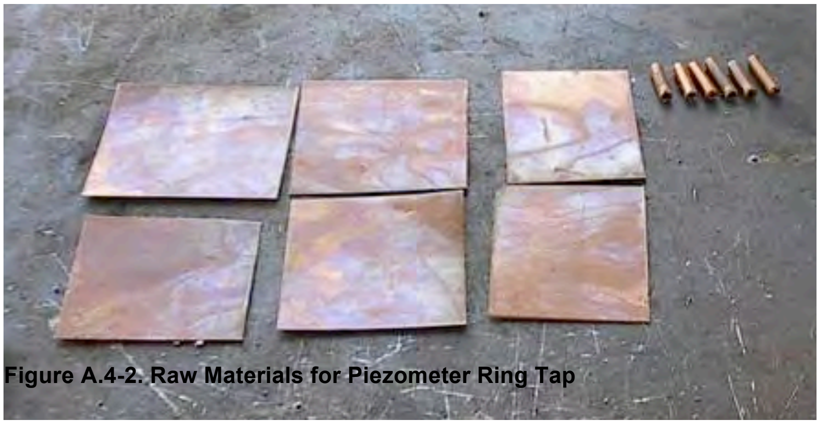

Figure A.4-2. Raw Materials for Piezometer Ring Tap

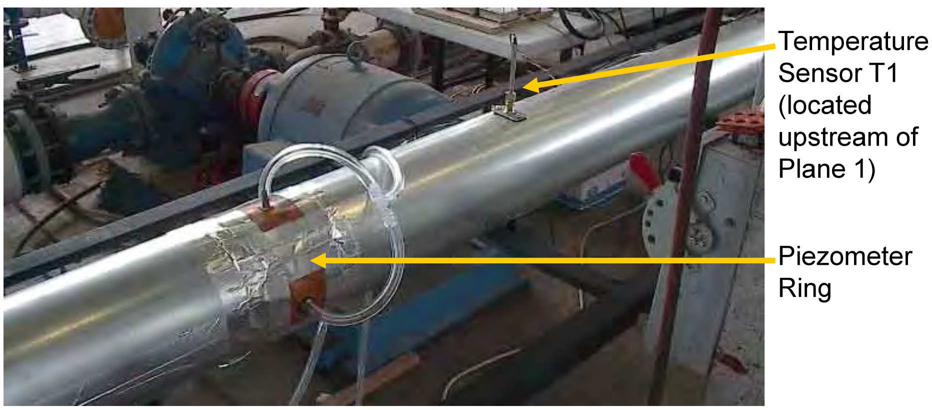

Figure A.4-3. Piezometer Ring and Temperature Sensor

The piezometer rings were checked to ensure that they satisfy the requirements of ASHRAE Standard 120-1999, section 6.2.6.1, which states that each tap must measure within ±2% of the others in the ring. To check the pressure of each tap, three of the four taps were sealed off using short lengths of tubing with one end plugged. The pressure line was connected to the remaining tap and then a set of ten data points were measured for that tap. Each tap was tested in sequence with the same setting for airflow. Ten (10) data points were recorded for each tap. The average for each tap was determined and then the four averaged values were compared. All taps were within ±2% of one another.

The temperature measurement throughout the test run uses two silicon-junction transistor type devices located at: the beginning and end of the test section. The sensor located in the nozzle chamber at Plane 5 was a General Eastern combination dry-bulb/dew-point instrument

Appendix B – Computation Fluid Dynamics

B.1 Background

Flexible ducts were modeled and simulated under computational fluid dynamics (CFD) with the project starting in August 2005. The objective of the study was to determine the optimum flexible duct modeling geometry and CFD simulation method. A validated and verified CFD model has the capacity to eliminate the time and cost issues associated with laboratory tests. The verified model can then be used for design purposes and to understand the behavior of airflow inside ducts. Six in., 8 in. and 10 in. diameter flexible ducts were simulated under various compression factors and volumetric airflow scenarios (Table B.1-1). Level of agreement between the simulated and measured data varied depending on the duct diameter and the compression factor. Parametric tests of duct geometry and turbulence model were conducted to explain the discrepancies between the simulated and measured data.

CFD methodology was applied to different duct types and fittings by several researchers (Rechia et al. 2007; Koskela 2004; Gan & Riffat 1995; Shao & Riffat 1995). Taghavi et al. (2007) simulated flexible ducts including 90º and 180º bends. Previous studies on duct CFD simulations used the k-ε turbulence model family which is a Reynolds Averaged Navier-Stokes equation (RANS). Our study included the k-ε models as well as other turbulence models to check their accuracy on flexible ducts. Model details and simulation results of 6 in. duct are presented in two publications (Uğursal and Culp 2006, 2007).

Table B.1-1. Volumetric Airflows Used in the Study

6 in.	**cfm**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**	**160**	
	ft/s	6.0	6.8	7.7	8.5	9.4	10.2	11.1	11.9	12.8	13.6	
8 in.	**cfm**	**140**	**160**	**180**	**200**	**220**	**240**	**260**	**280**	**300**	**320**	
	ft/s	6.7	7.7	8.6	9.6	10.5	11.5	12.5	13.4	14.4	15.3	
10 in.	**cfm**	**200**	**220**	**240**	**260**	**280**	**300**	**320**	**340**	**360**	**380**	**400**
	ft/s	6.1	6.7	7.3	8.0	8.6	9.2	9.8	10.4	11.0	11.6	12.2

B.2 Methodology

CFD simulations used duct diameters, compression factors and volumetric airflows which corresponded with the measured data. The measured data compared to the CFD calculated values were taken first and then compared to the CFD calculations. CFD models were considered to be well-performing when the simulation results fell within 3% of the measured data, which is the acceptability margin for measurement errors. Table B.2-1 presents the measured data which is used for the comparisons.

The super computer at Texas A&M University was used for the computations of this study. The K2 machine, which was used in the study, has 64 processors of 1000 MFLOPS floating speed with 64 GB of shared memory. Four processors in parallel with up to 2 GB memory were used which took approximately three hours for each volumetric airflow simulation to converge.

Table B.2-1. Measured Total Pressure Loss Data (in. H_2O) This will be the data used on the Air Duct Calculator flex duct equivalent scale

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	cfm	70	80	90	100	110	120	130	140	150	160	
6 in.	ΔP for Max Stretch	0.041	0.054	0.064	0.081	0.093	0.110	0.127	0.144	0.166	0.185	
	ΔP for 4%	0.105	0.140	0.176	0.220	0.270	0.324	0.385	0.443	0.513	0.596	
	ΔP for 15%	0.269	0.343	0.454	0.547	0.672	0.809	0.958	1.127	1.287	1.468	
	ΔP for 30%	0.470	0.650	0.856	1.051	1.332	1.596	1.855	2.159	2.552	3.011	
8 in.	cfm	140	160	180	200	220	240	260	280	300	320	
	ΔP for Max Stretch	0.039	0.049	0.062	0.074	0.088	0.105	0.120	-	-	-	
	ΔP for 4%	0.069	0.090	0.113	0.141	0.171	0.204	0.241	-	-	-	
	ΔP for 15%	0.185	0.244	0.307	0.379	0.462	0.559	0.650	0.758	0.866	0.985	
	ΔP for 30%	0.345	0.451	0.574	0.716	0.869	1.039	1.237	1.430	1.659	1.880	
10 in.	cfm	200	220	240	260	280	300	320	340	360	380	400
	ΔP for 15%	0.095	0.116	0.140	0.166	0.197	0.221	0.259	0.291	0.331	0.371	0.413

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B.3 CFD 3-D Flexible Duct Models

Figure B.3-1 shows the simulated 3-D model, which consisted of a 5 ft long flexible section and 2 ft long sheet metal end sections. Pressure differentials from different sections of the 5 ft long portion were converted to in. H_2O/100 ft values and then compared to the measured data. The 3 ft long section is a well-developed flow region which is also free of end effects of the duct profile change. Comparison of the simulation results from three regions showed that the 3 ft long section presented closest results to the measured data. Data presented in this report are based on the 3 ft long section.

In the first phase of the study, the 3-D geometry, which represented the fully inflated blow-through flexible duct condition, was modeled for 6 in. duct. The 30% compressed and maximum stretched ducts showed the closest proximity to the measured data. However, the 4% and 15% compressed duct simulations presented deviations from the measured data. The same methodology was applied to 8 in. duct and similar results were obtained. Fifteen percent (15%) compressed 10 in. duct also presented similar discrepancies. In the second phase, 3-D duct geometry and turbulence models were further investigated as potential sources of error. Those geometries involve helical and periodic (non-helical) circular and triangular wall sections. Duct geometries are explained in detail as follows. The inner core of a flexible duct was modeled. This model is composed of a two-ply, polyester inner wall with a helix shaped steel wire for the structural integrity of the duct. Fiberglass insulation and the metalized vapor barrier layers covered the inner core. The exterior two layers were not included in the model because the polyester inner wall is adequate to represent the boundary layer geometry. In the fully stretched condition, the helix core extends a distance of 1.5 in. along the longitudinal axis at each full rotation (360°). The compression factors (4%, 15%, and 30%) were determined based on the percentage of contraction from the fully stretched condition.

The modeled geometries of 6 in. and 8 in. ducts are 9 ft long with a 5 ft long flexible section in the middle and 2 ft long sheet metal end sections (Figure B.3-1). The 10 in. model is composed of 3 ft long flexible section instead of a 5 ft long. A 2 ft long section was added to the model at the inlet and at the outlet to emulate the laboratory setup.

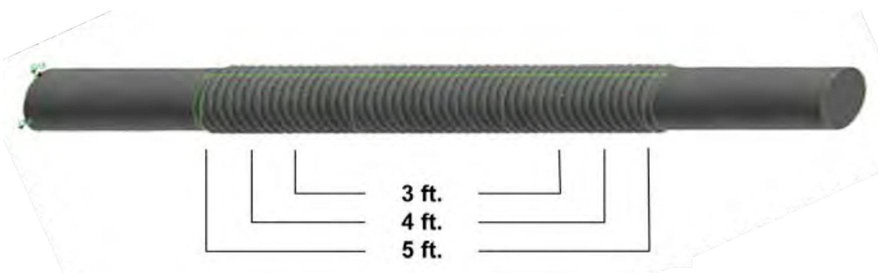

Figure B.3-1. 5 ft Flexible Duct Model with 2 ft End Sections

Six different wall geometries were used at different stages of this study (Figure B.3-2). Gambit 2.2.30 software was used to generate the 3-D geometries and the mesh. The generated mesh was then exported to Fluent 6.2.16 for CFD simulations. The central core consists of structural mesh with linear hexahedral elements, whereas, the near-wall region consists of unstructured mesh with tetrahedral, pyramidal, and wedge elements to ensure the geometric representation of the wall region. The number of cells in a model varied between 1,300,000 to 1,600,000 depending on the duct diameter and the compression factor.

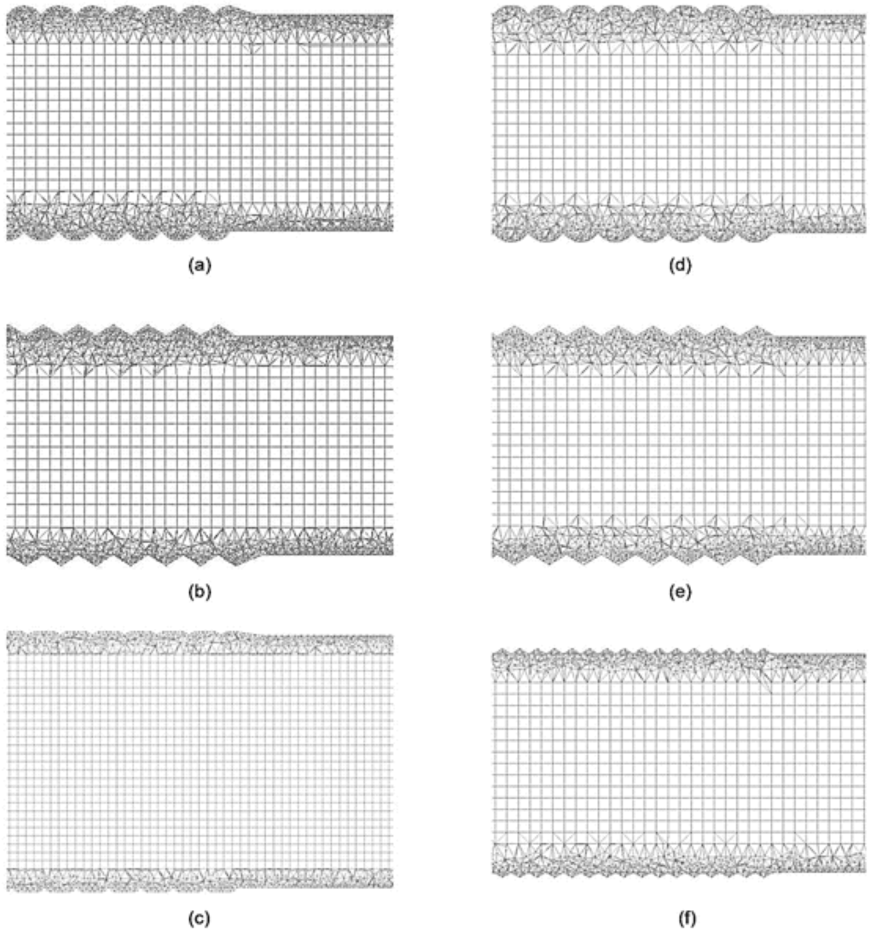

Figure B.3-2. Parametric Geometry Test for 15% Compressed Duct (a: Helical-circular, b: Helical-triangular, c: Helical-circular-alternative, d: Periodic-circular, e: Periodic-triangular, f: Periodic-triangular-alternative)

B.4 Comparison to Laboratory Data

The key in the CFD analysis was to ensure the solution convergence. Uğursal and Culp (2007) showed the effect of numerous iterations on the accuracy of the CFD results. Figure B.4-1 illustrates how the residuals of the kinetic energy (k) and turbulence dissipation rate (ε) changes as a function of the number of numeric iterations. On average, the solutions converged after 130 iterations to within a preset error band for the region of interest.

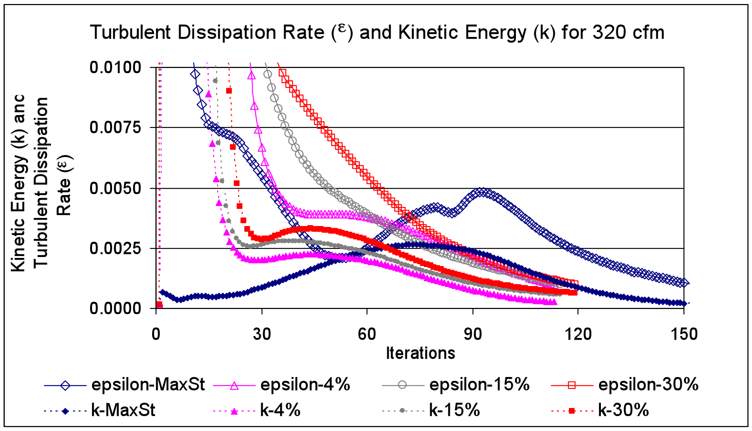

Figure B.4-1. Residuals of k and ε based on the Number of Iterations

The standard k-ε model was used for the simulations presented in this section. The RNG k-ε model was used in the 10 in. duct as a part of the parametric study of turbulence models. Total pressure values of the simulation domain were taken from the data points which are on the central longitudinal axis. Data points were located based on the travel distance of the helix core at each full turn. Figure B.4-2 shows the total pressure values at each data point inside the flexible as well as the straight end sections. Figure B.4-3 shows the change in total pressure between the two preceding data points. This figure shows that total pressure change stabilizes after 2 ft into the flexible section which indicates a well-developed flow region.

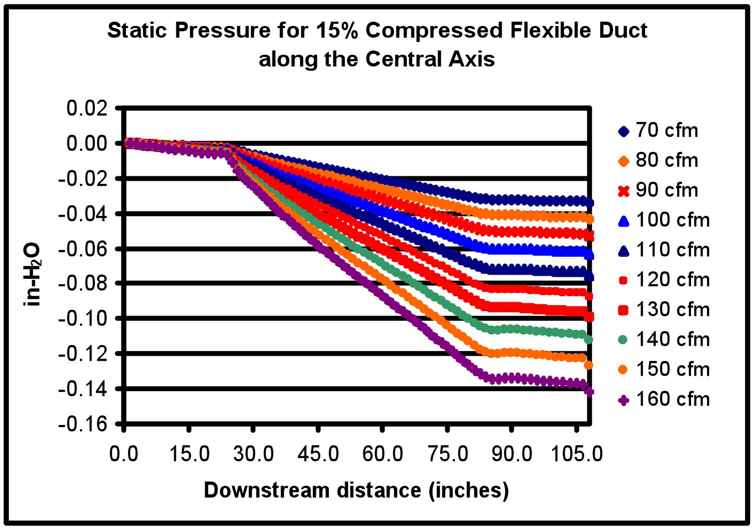

Figure B.4-2. Total Pressure Drop for 15% Compressed 6 in. Duct along Central Axis

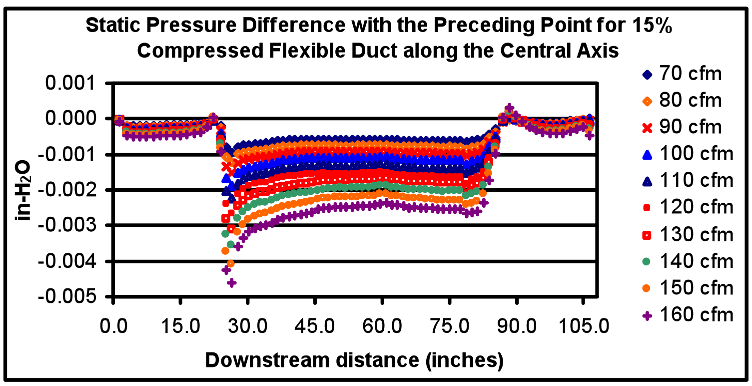

Figure B.4-3. Total Pressure Difference between Two Preceding Points

Simulation results of 6 in. duct pressure differentials are shown in Figure B.4-4 in comparison with the measured data. The 30% compressed and maximum stretched ducts showed close agreement with the measured data. However, discrepancy exists for 4% and 15% compressed ducts. The 8 in. duct showed similar behavior for the same compression factors (Figure B.4-5). The 10 in. duct was modeled for 15% compression and it also showed a similar discrepancy (Figure B.4-6). The total pressure values and differences between simulated and measured data are presented in Tables B.4-1 and B.4-2.

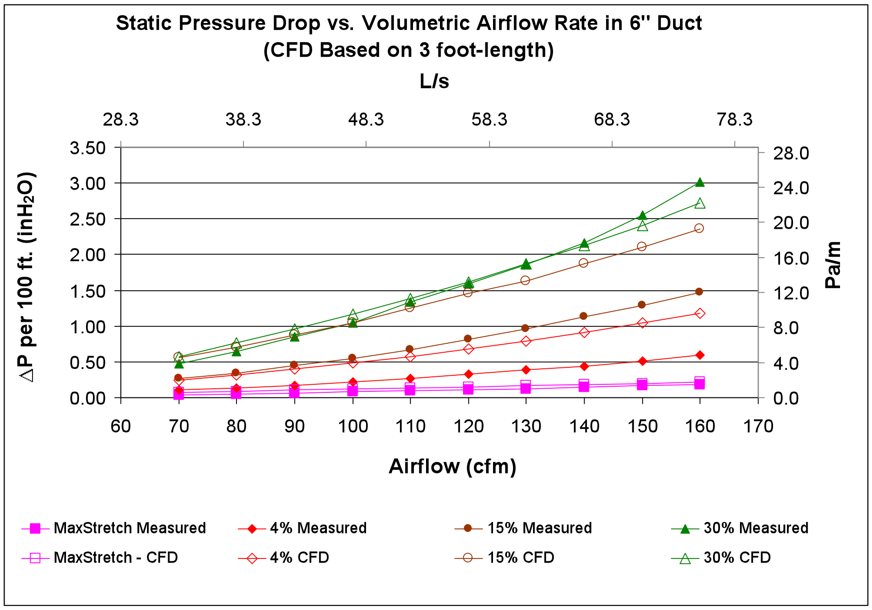

Figure B.4-4. Total Pressure Drop in 6 in. Duct based on Volumetric Airflow

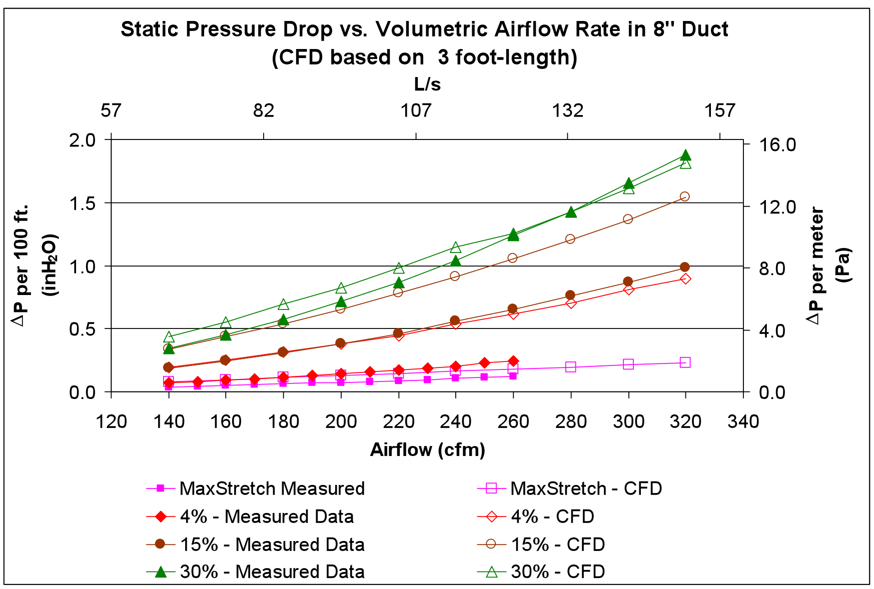

Figure B.4-5. Total Pressure Drop in 8 in. Duct Based on Volumetric Airflow

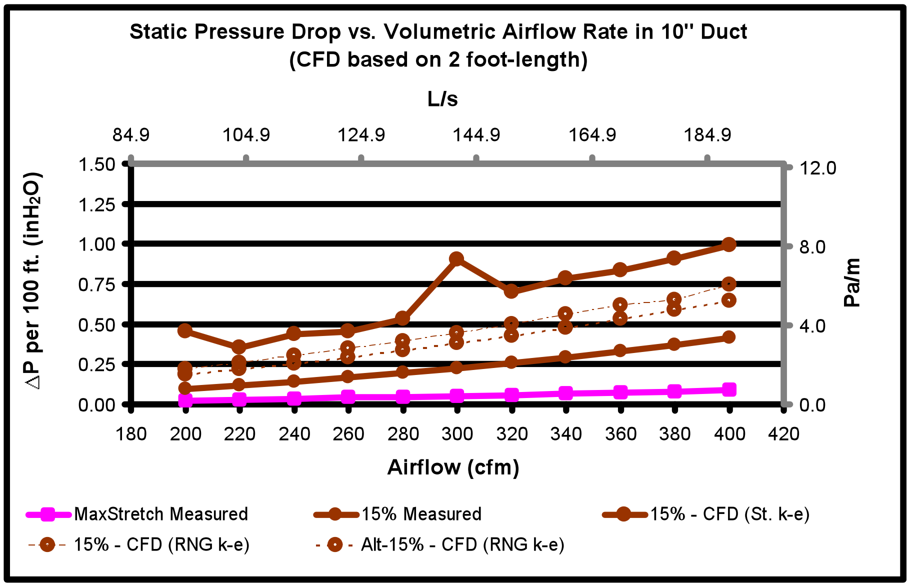

Figure B.4-6. Total Pressure Drop in 10 in. Duct Based on Volumetric Airflow

Table B.4-1. Simulated ΔP (normalized to in. H_2O/100 ft) for Three CFD Duct Types

6 in.	**cfm**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**	**160**	
	ΔP for Max Str. in 3 ft	0.073	0.087	0.103	0.120	0.133	0.150	0.167	0.183	0.200	0.213	
	ΔP for 4% in 3 ft	0.247	0.317	0.397	0.483	0.577	0.680	0.793	0.913	1.043	1.177	
	ΔP for 15% in 3 ft	0.557	0.707	0.873	1.047	1.253	1.463	1.633	1.867	2.103	2.353	
	ΔP for 30% in 3 ft	0.573	0.770	0.960	1.163	1.380	1.613	1.870	2.123	2.406	2.716	
8 in.	**cfm**	**140**	**160**	**180**	**200**	**220**	**240**	**260**	**280**	**300**	**320**	
	ΔP for MaxStr. in 3 ft	0.080	0.096	0.113	0.128	0.144	0.161	0.179	0.195	0.212	0.230	
	ΔP for 4% in 3 ft	0.195	0.251	0.314	0.382	0.448	0.535	0.616	0.701	0.812	0.900	
	ΔP for 15% in 3 ft	0.338	0.438	0.540	0.652	0.778	0.912	1.052	1.206	1.364	1.541	
	ΔP for 30% in 3 ft	0.437	0.550	0.693	0.827	0.983	1.150	1.253	1.427	1.613	1.817	
10 in.	**cfm**	**200**	**220**	**240**	**260**	**280**	**300**	**320**	**340**	**360**	**380**	**400**
	ΔP for 15% in 2 ft	0.454	0.352	0.434	0.452	0.531	0.900	0.698	0.785	0.833	0.909	0.992

Table B.4-2. ΔP Difference (Normalized to in. H_2O/100 ft) between CFD Simulations and Laboratory Experiments based on 3 ft and 2 ft duct lengths (ΔP_{CFD} - ΔP_{Lab})

6 in.	**cfm**	**70**	**80**	**90**	**100**	**110**	**120**	**130**	**140**	**150**	**160**	
	Max Stretch	0.032	0.033	0.039	0.040	0.040	0.040	0.040	0.039	0.034	0.028	
	4%	0.141	0.177	0.221	0.263	0.307	0.356	0.408	0.470	0.531	0.581	
	15%	0.288	0.364	0.420	0.500	0.582	0.654	0.675	0.739	0.816	0.885	
	30%	0.103	0.120	0.104	0.112	0.048	0.018	0.015	-0.036	-0.145	-0.293	
8 in.	**cfm**	**140**	**160**	**180**	**200**	**220**	**240**	**260**	**280**	**300**	**320**	
	Max Stretch	0.041	0.047	0.051	0.0540	0.057	0.057	0.059	-	-	-	
	4%	0.126	0.162	0.201	0.240	0.277	0.331	0.375	-	-	-	
	15%	0.154	0.194	0.233	0.274	0.316	0.353	0.402	0.449	0.498	0.556	
	30%	0.092	0.099	0.119	0.111	0.114	0.111	0.016	-.003	-.046	0.064	
10 in.	**cfm**	**200**	**220**	**240**	**260**	**280**	**300**	**320**	**340**	**360**	**380**	**400**
	15%	0.359	0.236	0.294	0.286	0.334	0.678	0.439	0.494	0.502	0.539	0.579

The initial 3-D modeling efforts of this study showed that wall geometry has a dominating effect on the total pressure losses inside the simulation domain. The first set of simulations (Figures B.4-4 through B.4-6) was conducted using the idealized fully rounded wall geometry which corresponds with the fully inflated blow through configuration. Alternative wall geometry of 10 in. duct which has an extra cusp in each segment was created (Figure B.3-2c). This geometry is closer to the real-life condition of flexible ducts.

The objective of the wall-geometry study is to determine the shape of the duct wall which gives accurate results without increasing the geometric and computational complexity. Four different versions of 8 in. 15% compressed duct were created (Figures B.3-1a, B.3-1b, B.3-1d, and B.3-1e). The first comparison is between the helical and periodic (non-helical segments) geometries. Periodic geometry is considerably easier to model and takes less computational time to solve. In addition, it allows two-dimensional CFD models. On the other hand, helical geometry can be solved only in three-dimensional models. The second comparison is between circular and triangular geometries. Figure B.4-9 shows that triangular wall geometry is considerably more accurate than the circular wall. This is because sharp edges are more typical to a compressed flexible duct wall. In order to further test this effect, an extra cusp was introduced to the triangular wall (Figure B.3-2f). This CFD model is able to simulate airflow in 8 in. ducts within 5-10% of the measured data.

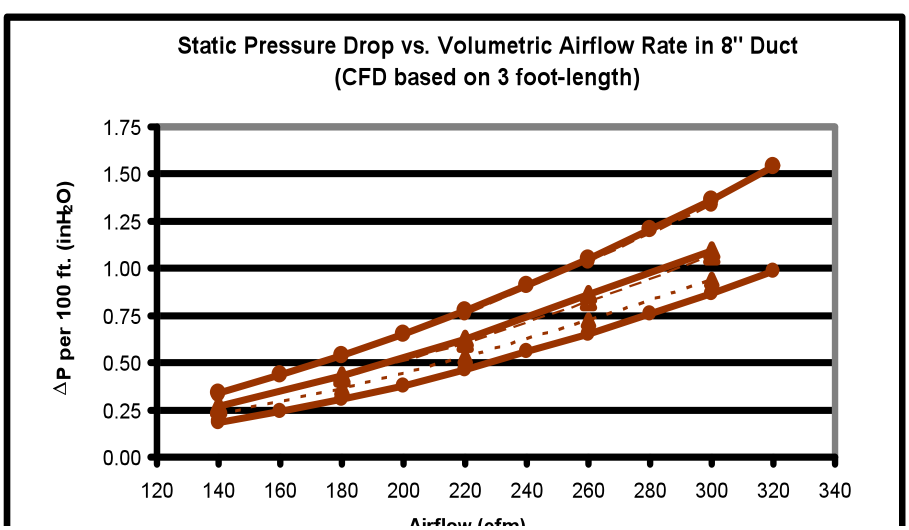

Figure B.4-7. Parametric Study of Pressure Loss in 15% 8 in. Duct

B.5 CFD Summary

Standard k-ε turbulence model was used to simulate maximum stretched, 4%, 15% and 30% compressed 6 in. and 8 in. flexible ducts. Standard and RNG k-ε models were used to simulate 10 in. 15% compressed flexible duct. Model domains (3 ft long for 6 in. and 8 in. ducts and 2 ft long for 10 in. duct) which are free of end effects were used in the pressure loss calculations. Simulations showed agreement for the maximum stretched and 30% compressed 6 in. and 8 in. ducts. However, considerable discrepancy existed for 4% and 15% compressed ducts. Similar discrepancy existed for the 15% compressed 10 in. duct. The RNG k-ε model which was also used by Taghavi et al. (2007) presented closer agreement with the measured data.

The second part of the study focused on explaining the discrepancy by creating more realistic wall geometries. Parametric studies on the 15% compressed 8 in. duct wall geometry showed that modeling helical geometries do not improve simulation results. Periodic geometry with triangular wall generated the closest agreement with the measured data. The reason is that triangular wall geometry with extra cusp better represents the irregular real-world character of the flexible ducts.

Finally, CFD method has the potential to accurately simulate airflow in complex flexible duct conditions once the discrepancies are explained. Determining pressure losses in various flexible duct installations are critical to improving comfort, conserving energy and increasing equipment life.

Appendix C – File Illustrating Zigzag Effect

You will need either quick time or VLC media player software to see the movie.

Figure C.1-1. Zigzag Effect (Duplicate of Figure 6.2-2)

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MECHANICAL ENGINEERING
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Record of Changes (changes are indicated by \1\1…/1/1)

Change No.	Date:	Location

FOREWORD

The Unified Facilities Criteria (UFC) system is prescribed by MIL-STD 3007 and provides planning, design, construction, sustainment, restoration, and modernization criteria, and applies to the Military Departments, the Defense Agencies, and the DoD Field Activities in accordance with USD(AT&L) Memorandum dated 29 May 2002. UFC will be used for all DoD projects and work for other customers where appropriate. All construction outside of the United States is also governed by Status of forces Agreements (SOFA), Host Nation Funded Construction Agreements (HNFA), and in some instances, Bilateral Infrastructure Agreements (BIA.) Therefore, the acquisition team must ensure compliance with the more stringent of the UFC, the SOFA, the HNFA, and the BIA, as applicable.

UFC are living documents and will be periodically reviewed, updated, and made available to users as part of the Services' responsibility for providing technical criteria for military construction. Headquarters, U.S. Army Corps of Engineers (HQUSACE), Naval Facilities Engineering Command (NAVFAC), and Air Force Civil Engineer Support Agency (AFCESA) are responsible for administration of the UFC system. Defense agencies should contact the preparing service for document interpretation and improvements. Technical content of UFC is the responsibility of the cognizant DoD working group. Recommended changes with supporting rationale should be sent to the respective service proponent office by the following electronic form: Criteria Change Request (CCR). The form is also accessible from the Internet sites listed below.

UFC are effective upon issuance and are distributed only in electronic media from the following source:

- Whole Building Design Guide web site http://dod.wbdg.org/.

Hard copies of UFC printed from electronic media should be checked against the current electronic version prior to use to ensure that they are current.

AUTHORIZED BY:

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UNIFIED FACILITIES CRITERIA (UFC)
NEW DOCUMENT SUMMARY SHEET

Document: UFC 3 400-10N, *Mechanical Engineering*
Superseding: None.

Description: This UFC 3-400-10N provides mechanical engineering design and analysis criteria for design-build and design-bid-build projects.

Reasons for Document:

- Provide technical requirements for mechanical systems design criteria.
- The Design-Build processes as defined herein reflect current contract requirements.
- Define minimum requirements for contract documents in terms of drawing types and content, and specification information.

Impact: There are negligible cost impacts. However, the following benefits should be realized.

- Promotes the use of and moves the DoD toward more efficient commercial model codes and standards.
- Standardized guidance has been prepared to assist environmental engineers in the development of the plans, specifications, design analyses, and Design/Build Request for Proposals (RFP).
- This guidance has been prepared along with updates to the associated Performance Technical Specifications and Engineering Systems Requirements documents. The three types of documents have been aligned to allow improved consistency in the preparation of project requirements.

CONTENTS

TABLES

CHAPTER 1 INTRODUCTION

1-1 PURPOSE AND SCOPE.

The purpose of this document is to provide technical guidance and outline technical requirements for the more typical aspects of the mechanical engineering portion of Architect/Engineer (A/E) contracts for the Naval Facilities Engineering Command (NAVFAC). The information provided in this guide shall be utilized by mechanical designers in the development of the plans, specifications, calculations, and Design/Build Request for Proposals (RFP) and shall serve as the minimum mechanical design requirements. Project conditions may dictate the need for design that exceeds these minimum requirements.

1-2 APPLICABILITY.

This document is applicable to the traditional mechanical services customary for Design-Bid-Build construction contracts and for Design-Build construction contracts.

1-3 REFERENCES.

Appendix A contains the list of references used in this document. Furthermore, this document references UFC 1-200-01, *General Building Requirements*, except as modified herein.

1-4 COMMUNICATIONS.

Direct communication with the government's project manager and mechanical reviewer is encouraged. This may avoid unnecessary re-submittal of plans and specifications due to a misunderstood comment. The reviewer's name, phone number and email address can be found on the comment sheets.

1-5 ADDITIONAL REQUIREMENTS.

Local and regional requirements may differ from those included herein. Contact the Government Project Manager for guidance.

1-6 PERMITS: CONSTRUCTION, ENVIRONMENTAL & OTHER.

Obtain the permits necessary for environmental, construction and operation of facilities. Pay any fees associated with each permit. Refer to UFC 3-200-10N, *Civil Engineering*, for more information.

CHAPTER 2 REQUIREMENTS

2-1 ENERGY SUPPLY

2-1.1 Fuel Source and HVAC System Selection.

New facilities and facilities undergoing major and minor renovation as defined in UFC 3-400-01, *Energy Conservation*, are required to be analyzed to determine the most cost effective and practical fuel source(s) and heating and cooling system types. Provide energy analysis in accordance with UFC 3-400-01.

2-1.2 Energy Conservation.

Provide mechanical system based on lowest life cycle cost. Provide completed compliance forms provided in ASHRAE 90.1 User's Manual, any additional documentation to support compliance with this Standard, and applicable state government required forms.

2-1.2.1 **Facility Energy Conservation.** The Energy Policy Act of 2005 (EPACT05) has increased the energy conservation requirements over those listed in UFC 3-400-01. Per EPACT05, new facilities shall be designed to achieve energy conservation levels that are at least 30% below the levels established by ASHRAE Standard 90.1-2004 or the International Energy Code (for residential buildings).

2-1.3 Energy Star.

All HVAC equipment, appliances, related electrical equipment, and water saving fixtures shall meet or exceed the minimum efficiencies listed by Energy Star and Federal Energy Management Program (FEMP). The FEMP website lists all Energy Star and FEMP rated products and provides recommended efficiencies and life cycle data. The FEMP website is at http://www.eren.doe.gov/femp/. For product groups where Energy Star labels are not yet available, select products that are in the upper 25% of energy efficiency as designated by FEMP.

2-1.4 Sustainable Design.

Integrate sustainable development principles into the mechanical system selection and design. Refer to NAVFAC Instruction 9830.1, Sustainable Development Policy and utilize the U.S. Green Building Council's LEED Green building Rating System as a tool to apply sustainable development principles and as a metric to measure the sustainability achieved through the planning, design, and construction processes.

2-1.5 Indoor Environmental Quality and Mold.

Provide outside air ventilation as prescribed by the latest edition of ASHRAE Standard 62. Consider the factors of "Multiple Spaces", "Ventilation Effectiveness", and "Intermittent or Variable Occupancy" as specified in ASHRAE Standard 62. The building and mechanical system must be designed and constructed to prevent the growth of mold. Refer to UFC 3-800-10N, *Environmental Engineering for Facility*

Construction, for additional information and requirements. Air Force facilities must comply with AF ETL 04-3.

2-1.6 **Humid Areas.**

All heating, ventilating and air conditioning designs in humid areas must comply with the requirements of the Interim Technical Guidance FY05-02, *NAVFAC Humid Area HVAC Design Criteria*. Humid areas are defined in paragraph 5-1.4 of the Interim Technical Guidance.

2-1.7 **Economizer Cycles.**

Economizer cycles are not allowed in areas defined as humid in Interim Technical Guide FY05-02. Contact the Government Project Manager for other areas.

2-1.8 **Building Pressurization.**

Maintain the building under positive pressure in order to negate infiltration.

2-2 **HEAT GENERATING SYSTEMS**

2-2.1 **Boilers.**

Install boiler(s) and associated hot water pumps in a mechanical room inside the facility unless otherwise noted in the Project Program. Passageways around all sides of boilers shall have an unobstructed width of 1 meter (3 feet), or the clearances recommended by the boiler manufacturer, whichever is greater.

2-2.1.1 **Multiple Boilers.** In multiple boiler installations, the lead boiler should operate up to full capacity prior to starting the next boiler. During heating season, multiple boilers should be kept warm and ready should the lead boiler fail to operate.

2-2.1.2 **Boiler Procurement.** DoD policy requires that boilers procured be ASME certified.

2-2.1.3 **Boiler Emissions.** Boilers must comply with regulatory requirements under the Clean Air Act regarding Title V and New Source Review permits programs as well as requirements under New Source Performance Standards and National Emissions Standards for Hazardous Air Pollutants (NESHAP). Contact the local or regional Public Works Department or Base Civil Engineering Branch for specific requirements.

2-2.1.4 **Draft Hoods.** Provide for each gas-fired piece of equipment, except power vented and condensing type equipment.

2-2.1.5 **Barometric Dampers.** Provide barometric dampers for all boilers requiring negative draft.

2-2.2 Steam Boilers.

On boiler start-up, the condensate in a gravity system may not return quick enough to maintain the boiler water level. Contact the boiler manufacturer for boiler feed system tank size and location.

2-2.3 Condensing Boiler Systems.

Provide hydronic systems with condensing gas-fired boilers with a water volume equal to five (5) minutes of water flow through the system pump (minimum), or as required by the boiler manufacturer. This insures there is sufficient water volume to prevent short cycling of the burner. If there is insufficient water volume, an inertia tank must be installed to attain the minimum system volume required. Non-condensing boilers do not require this minimum.

2-2.4 Combustion Air.

Provide combustion air for gas and oil-fired equipment in accordance with International Mechanical Code (IMC) and NFPA requirements.

2-2.5 Steam Heating.

Steam heat should not be used except on rehabilitation projects where budget constraints preclude conversion of an existing steam heating system to hot water.

2-2.6 Infra-Red Gas Radiant Heaters.

When using non-condensing gas infrared heaters, the length of the exhaust flue should be minimized. To minimize condensation, run the flue horizontally with a slight pitch down from the heater to a sidewall exit. Heaters should be properly braced where excessive movement, such as by wind through an open hangar bay door, may cause separation of radiant pipe sections and rupture of gas connections. Consider condensing type IR heaters for larger applications. Provide sufficient overhead ventilation for condensing type IR heaters to carry water vapor out of the building.

2-3 COOLING GENERATING SYSTEMS

2-3.1 Condensing Temperatures.

The design condensing temperature for air-cooled condensers, chillers, etc must be ambient design temperature plus 2.8 degrees C (5 degrees F) dry bulb.

2-3.2 Chilled Water Systems.

Chiller manufacturers recommend minimum system volumes to prevent short-cycling of the chiller(s) to promote long chiller life and good chilled water temperature control, especially in smaller chilled water systems. In small systems it may be necessary to install an inertia tank in the chilled water loop to achieve the required minimum system volume. Check the requirements of the chiller manufacturer and provide an insulated, inertia tank of sufficient volume when required. Install the chilled water storage tank downstream of the chiller and upstream of the cooling coils. Provide calculations to demonstrate compliance with this requirement. Volumes for components may be estimated where manufacturer's data is not available.

2-3.3 **Chillers.**

When multiple chillers serve a common central chilled water system, install a flow control balancing valve on the leaving side of the chilled water and condenser water (where applicable) of each chiller. Flow orifices with butterfly valves should be provided for larger pipe sizes. On multiple chiller systems, design pumping and piping systems to prevent water flow through chillers that are not in operation. Avoid the use of reciprocating compressors when possible. Utilize roof mounted chillers only as a last resort. If located on the roof, mount the chiller on a steel skid with isolators supported from the structural roof framing.

2-3.4 **Cooling Towers.**

Provide a butterfly or 3-way diverting valve in the by-pass line for all cooling towers that are specified to have a condenser water by-pass for regulating the condenser water supply temperature. Provide automatic isolation valves on inlet and outlet of each cell for multi-cell or multi-tower applications. Size condenser water flow to chiller for the design flow rate, not the oversized tower flow rate. Cooling tower piping shall by-pass to the cooling tower's sump.

2-3.5 **Ground-Coupled Heat Pump (GCHP) System Design Guidance**

The guidelines that follow are intended to complement the guidance and requirements of ASHRAE and recognized consortiums, such as the International Ground Source Heat Pump Association (IGSHPA).Nonresidential, commercial scale ground source heat pump systems require the utilization of computer design software. Such software should consider the interaction with adjacent loops and long-term buildup of rejected heat in the soil.

2-3.5.1 Provide a bypass line around the condenser of each heat pump unit to facilitate flushing and purging the condenser loop without subjecting the condenser coil to residual construction debris.

2-3.5.2 Provide isolation valves and valved tee connections for flushing and purging of the well field independently from the building condenser water system.

2-3.5.3 Do not provide automatic water makeup in residential GCHP systems. Reserve the added complexity and cost to larger, non-residential systems of 10 tons or larger. Utilize cupronickel refrigerant-to-water heat exchangers in open condenser loops only.

2-3.5.4 Provide test ports (sometimes referred to as “Pete’s plugs”) on the inlet and outlet to each heat pump unit, circulating pump and desuperheater, if incorporated.

2-3.5.5 Utilize reverse return headers in large well fields. For heat pumps with reduced flow requirements of 2 GPM/ton or less, consider series return in order to maintain fluid velocities necessary to foster good heat transfer. Base the decision to commit to reverse return on installed cost, pumping costs and the system flow requirements. Consult ASHRAE and IGSHPA Design documentation for additional information.

2-3.5.6 Regulatory requirements for vertical wells vary widely among States. Some regulations require partial or full grouting of the borehole. The State of Virginia, for example, requires bentonite or cement grout seals in the top 6.1 meters (20 feet) of a borehole; while North Carolina requires a full depth seal. A full depth bentonite seal, however, is not necessary; the wells in N.C. may be grouted with cement mixed with soil from the bore drilling. Confirm with the Government Project Manager and consult current state and federal regulations, as well as relevant building codes.

2-3.5.7 The thermal conductivity of grouting materials is typically low when compared to the conductivity of native soils. Grout acts as an insulator and will, thus, hinder heat transfer to the well field. When governing regulations permit, consider the following alternatives:

a. Reduce the quantity of grout to an absolute minimum. Fine sand may be used as backfill where permitted, but caution must be exercised to ensure the interstitial space between pipe and borehole is filled to enhance conductivity.

b. Utilize thermally enhanced grout. Consult ASHRAE, *Commercial/Institutional Ground-Source Heat Pump Engineering Manual*. Reduce the borehole diameter as much as possible to reduce the insulating effects of grout or backfill.

2-3.5.8 In geographic areas with heating dominated climates, an antifreeze solution may be required if condenser loop temperatures are expected to drop below 5 degrees C (41 degrees F). Avoid use of antifreeze, but if necessary, keep concentrations to a minimum. Utilize condenser water circulating pumps with high efficiency motors. Design them to operate near their peak of maximum efficiency.

2-3.6 **Refrigerants.**

The use of Ozone Depleting Substances (ODS) as well as the qualifications and credentials of personnel servicing equipment that contains ODS is restricted. Refrigerants shall have an Ozone Depletion Factor (ODF) of 0.055 or less. The ODF shall be in accordance with the "Montreal Protocol on Substances That Deplete the Ozone Layer", September 1987, sponsored by the United Nations Environment Program.

2-3.7 **Refrigerant Piping.**

Avoid refrigerant piping runs longer than 15 meters (50 feet) unless specifically allowed by the equipment manufacturer. Size refrigerant piping in accordance with the manufacturer's recommendations.

2-4 DISTRIBUTION SYSTEMS

2-4.1 Air Distribution.

2-4.1.1 **Air Change.** The quantity of supply air shall be sufficient to provide a minimum of four air changes per hour within the conditioned space. If the supply air quantity determined from the sensible cooling load does not provide four air changes, room air may be mixed with conditioned air in a fan-powered terminal to increase the quantity of supply air. Use a maximum ceiling height of 3.1 meters (10 feet) when calculating air changes per hour.

2-4.1.2 **Locker Room HVAC Systems.** Draw conditioned air into locker rooms from adjacent spaces, and provide additional supply air as required. This uses the outside air required for human occupancy in the adjacent spaces for secondary air conditioning of the locker space and maintains locker spaces at a negative pressure with respect to adjacent spaces. No air shall be returned from the locker space to the building HVAC system.

2-4.1.3 **Closets in Bachelor Housing (BH).** BH facilities shall be designed in accordance with UFC 4 721-10, Navy and Marine Corps Bachelor Housing. In Humid Areas, provide a Dedicated Makeup Air System 100% outside air supply register in each clothing closet in new BH modules, sized to provide approximately 7.5 L/s (15 CFM) for humidity control.

2-4.1.4 **Outside Air Ducts.** Size outside air ducts for velocities a minimum of 4.1 m/s (800 fpm) for accurate flow measurement. Provide a straight duct of suitable length to facilitate an airflow measurement traverse. Provide an air flow measuring station for VAV systems to verify proper outside air quantities. Equipment layout shall allow for the straight duct length requirements of the air flow measuring station in accordance with the manufacturer's recommendations.

2-4.1.5 **Variable Speed Drives.** Select system equipment to deliver design flows so that maximum operational flexibility is maintained. Verify fan performance at minimum and maximum operating points.

2-4.1.6 **Vibration and Noise Isolation.** Where vibration and/or noise isolation is required, provide a vibration isolator schedule on the drawings indicating type of isolator, application, and deflection in mm (inches).

2-4.1.7 **Access Panels.** Provide access panels in floors, walls, and ceilings (except in lay-in tile applications) as required to access valves, smoke dampers, fire dampers, balancing dampers, balancing valves, air vents, drains, duct coils, filters, equipment, etc. Indicate location and size on drawings. Verify that the dimensions will yield reasonable accessibility.

2-4.1.8 **Equipment Supports.** Provide for vibration isolation where required and schedule the vibration isolation components on the drawings. Coordinate with and provide hardware required to meet Anti-terrorism requirements in UFC 4-010-01 and seismic requirements in accordance with UFC 1-200-01. All equipment mounted on a roof must be fastened to the building as recommended by the structural engineer.

2-4.1.9 **Space Noise Levels.** Design and install systems to maintain noise levels below those recommended in the ASHRAE Handbooks.

2-4.1.10 **Variable Air Volume (VAV) HVAC System Design Guidance**

The guidelines that follow are intended to complement the guidance and requirements of ASHRAE.

2-4.1.10.1 Do not oversize the system. Do not add safety factors in the load calculations. Safety factors not only have the ramification of added cost, but also penalize the system during frequent part load conditions commonly experienced in humid, coastal locations.

2-4.1.10.2 Utilize computerized load calculations based on the ASHRAE Transfer Function Method. Select all central air handling equipment and central plant equipment for “block” loads. Spread diversity through the supply ducts, taking full diversity at the air handling unit, and lessening diversity when moving away from the air handling unit toward the VAV terminal units, until no diversity is taken at the distant VAV terminal run outs.

2-4.1.10.3 Design for both peak and part load conditions (minimal wall transmission load, low occupancy, etc.). VAV Systems shall provide acceptable air circulation and proper outside air for all conditioned spaces regardless of the loading conditions.

2-4.1.10.4 Address the psychrometric performance of the cooling coils, with full consideration of the method of capacity control and its limitations, during part load conditions when the sensible heat ratio can be significantly reduced. Submit part load design calculations. Check the fan operating characteristics throughout the range from the minimum to the maximum flow conditions that will be experienced. Evaluate the off-peak turndown requirements for the main air handler VAV fan. Do not utilize discharge dampers or inlet vanes for air flow modulation. Provide variable frequency drives for air volume modulation.

2-4.1.10.5 Design a positive means of maintaining ventilation rates during part load conditions. Select the minimum primary air requirements of the VAV terminal units to maintain at least the minimum outside air ventilation requirements. The Direct Digital Control (DDC) system shall comply with the requirements of ASHRAE 62 for polling of boxes to maintain proper ventilation levels. Provide an air flow monitoring station in the outside air duct controlling the outside and return air dampers or a constant volume outside air fan to maintain the minimum outside air requirements. Constant volume outside air fans are the most reliable method of maintaining outside air rates and are preferred. When using airflow measuring stations (AFMS) for monitoring and maintaining constant outside air ventilation rates, avoid placement of the AFMS in the outside air duct unless a minimum of 12 duct diameters of straight duct downstream of the outside air louver can be provided. Turbulence generated by the outside air intake louver will generate high turbulence and a highly unstable control loop. For large systems using a constant air volume (CAV) fan use a pressure independent velocity controller in the outdoor air intake to keep outdoor airflow constant as the VAV air handler fan modulates. Provide a low velocity filter module upstream of the air injection fan to prevent dust/dirt build up that may clog the pitot tubes associated with the volume regulator. Provide a duct access door at the inlet to the CAV terminal box for periodic inspection and cleaning.

2-4.1.10.6 Utilize the static regain method in design of the supply ductwork. Design return ductwork using the equal friction method.

2-4.1.10.7 Provide control for a constant cooling supply air temperature. Resetting the supply air temperature upwards increases the coil sensible heat ratio and results in elevated space relative humidity.

2-4.1.10.8 Provide electronic controls; pneumatic controls present problems with repeatability and maintenance.

2-4.1.10.9 Locate the static pressure sensor for modulating fan capacity two-thirds to three-quarters the distance from the supply fan to the end of the main trunk duct. Locate in straight run of ductwork. Provide static pressure reset in accordance with ASHRAE Standard 90.1. Provide protection against over pressurization of the supply duct system. Utilize pressure independent (PI) terminal units. Do not use light troffer return units. Light troffers reduce room sensible loads with undesirable effects on room air changes and outdoor ventilation distribution. Control the cooling coil capacity, especially in the more humid climates. VAV is inherently one of the best of the chilled water systems for air conditioning in tropical climates.

2-4.1.10.10 Do not utilize DX VAV systems without prior approval of the Government Project Manager. Direct expansion equipment shall be specifically designed and manufactured for VAV applications. The same manufacturer shall provide central air handling units, VAV boxes/zone dampers and zone controls. Airflow through the evaporator coils shall not be modulated. Provide duct mounted zone control damper units with integral control box, designed for use with DX VAV packaged systems. Self-modulating air diffusers will not be allowed.

2-4.1.10.11 Provide round or flat oval duct systems for primary air on all VAV supply systems. Utilize round ducts wherever space availability permits.

2-4.1.10.12 Proper VAV box primary air entry conditions are critical for achieving stable, accurate airflow delivery. Every effort must be made to avoid high turbulence in the proximity of the VAV terminal flow sensor. Design the primary air duct branches to the VAV terminals with a straight duct section of at least 6 to 8 duct diameters (more if required by specific manufacturers). Reducer and increaser duct fittings installed immediately upstream of the VAV terminal connection collars are prohibited. If the branch duct size is other than the VAV terminal connection collar size, install the reducer or increaser fitting upstream of the aforementioned straight duct section.

2-4.1.10.13 Primary air connections to VAV terminals should always be with a rigid duct. If a section of flexible duct, or a flexible connection, is required for vibration control, limit the length to no more than 305 mm (12 inches), and ensure that it is placed at least 6 to 8 duct diameters upstream of the VAV box collar connection/flow sensor.

2-4.1.10.14 VAV terminal boxes have minimum primary air velocity limitations imposed by the volume regulators utilized. Though many manufacturers claim their VAV boxes can deliver minimum primary air at flow rates resulting in inlet velocities of 189 L/s (400 fpm) and a velocity pressure of 2.48 Pa (0.01 inch w.g.), the lack of a certifying agency to test the manufacturer's claims support a more conservative approach. Minimum primary airflow rates shall be established to attain minimum velocity pressures of no less than 7.45 Pa (0.03-inch w.g.). Do not utilize system-powered (also called "pressure dependent") terminal units.

2-4.1.10.15 Special consideration must be given when fan-powered VAV boxes are specified and when it is necessary to specify a VAV box fan CFM in excess of the specified maximum primary air CFM. When used with a dropped ceiling return plenum, the excess VAV box CFM will introduce secondary air into the conditioned space. This has the effect of transferring return side coil cooling loads to room-side sensible loads. Always make sure the transferred sensible heat is taken into account in the calculated room-side sensible heat. Failure to do so may result in inadequate primary airflow rates to satisfy the room sensible heat loads.

2-4.1.10.16 Discharge dampers shall be installed on all series fan-powered VAV boxes (SFPVAV), regardless of the type of fan speed control utilized (3-speed fan switch or solid state speed control).

2-4.1.10.17 When it is necessary to install VAV terminals at high elevations above finished floors, service and maintenance accessibility must be carefully analyzed. Where mounting heights are in excess of 3.6 m (12 feet) above finished floors, special accommodations are necessary:

a. Do not use fan-powered VAV boxes in such locations, since there are many serviceable components involved. Instead, consider using non fan-powered terminal boxes for use in high mounting height locations to eliminate the need for fan servicing and filter change access.

b. When DDC controls are installed, specify the installation of the DDC digital controller to facilitate ease of access.

c. If scaffolding, scissor lifts, ladders or other means is required to access VAV units, special considerations must be made. Be sure clear floor area is available below the VAV boxes to facilitate the means of access (i.e.

scaffolding, etc) and in an area that will be likely to remain clear of permanent or semi-permanent equipment or furnishings.

d. When DDC controls are provided for VAV boxes, specify the ability to monitor VAV box hot water control valve position (if provided with hot water coils), control damper position, primary airflow, flow sensor pressure differential, and box leaving supply air temperature. The means to monitor VAV box function will maximize the means to troubleshoot remotely, thus reducing the frequency for above ceiling access by maintenance personnel.

e. Utilize electronic controls for VAV boxes mounted in high areas.

f. Specify the integral mounting of communication ports for the VAV box digital controllers to the room zone temperature sensor. When occupied/unoccupied modes of control are required of the VAV system, specify remote momentary override switch mounted integral to the room zone temperature sensors to permit non-standard schedule operation during unoccupied modes.

2-4.1.10.18 Fan-powered VAV terminal boxes can be noisy. Perform an acoustic analysis to ensure designs are within acceptable NC criteria noise levels. Pay particular attention to noise attenuation in locations where the boxes are installed in spaces without dropped ceilings. Analyze potential for sound breakout from main supply air ducts. Provide attenuation as required. Do not provide acoustical duct liner for attenuation.

2-4.1.11 **Duct Lining.** Duct lining shall only be used for room to room transfer applications. Increase the duct dimensions as required. Acoustical duct lining shall not serve as thermal insulation for duct.

2-4.1.12 **Fire Dampers.** Provide fire dampers and access panels in ductwork penetrating fire rated walls and floors in accordance with NFPA 90A.

2-4.1.13 **Flexible Connections.** Provide flexible connections in ductwork at equipment. Support duct at flexible connections to ensure proper alignment.

2-4.1.14 **Flexible Duct.** Flexible duct lengths shall not exceed 1829 mm (6 feet).

2-4.1.15 **Louvers.** Provide rain or storm proof louvers at wall intakes and exhausts. Indicate dimensions, airflow rate, and air pressure drop. Consider the potential for carry-over of wind driven rain.

2-4.1.16 **Screens.** Provide insect or bird screens, as applicable, at all building intakes and exhausts.

2-4.1.17 **Door Louvers.** Size for minimal pressure drop.

2-4.2 **Water Distribution.**

2-4.2.1 **Variable Speed Drives.** Select system equipment to deliver design flows so that maximum operational flexibility is maintained. Verify pump performance at minimum and maximum operating points.

2-4.2.2 **Chilled Water Pumps.** Provide a dedicated primary pump and condenser water pump for each chiller. Provide piping and valve configuration that allows each chiller to operate with any primary pump and with any condenser water pump. Provide back-up or standby pumps so that the total system capacity is available with any one pump out of service.

2-4.2.3 **Hot Water Pumps.** Provide back-up or standby pumps so that the total system capacity is available with any one pump out of service.

2-4.2.4 **Piping systems.** When terminal equipment loads are relatively equal in percentage of total load, design closed system piping using reverse return method.

2-4.2.5 **Pressure and Temperature Taps.** Provide pressure and temperature taps ("Pete's Plugs") on the inlets and outlets of all coils, pumps, chillers, heat exchangers, and other equipment.

2-4.2.6 **Expansion and Compression Tanks.** Utilize diaphragm type expansion tanks. Size the expansion tank according to the latest edition of the ASHRAE Systems Handbook. Indicate the acceptance volume, nominal dimensions, configuration (i.e. horizontal or vertical) and pre-charge air pressure.

2-4.2.7 **Expansion Loops and Devices.** Provide expansion loops and/or devices as required for proper piping protection. Detail and dimension loops and schedule joints indicating minimum total traverse and installed expansion traverse. Indicate guide spacing. Avoid the use of expansion joints where possible due to the high resultant thrust. Instead utilize geometry and ball joints where possible.

2-4.2.8 **Cold Water Make-up.** Provide for make-up to each water system. Provide pressure gauges up and downstream of the PRV. Provide bypass line with a globe valve for each PRV. Provide hose bibbs in the make-up water line to cooling towers and evaporator condensers for washdown of equipment.

2-4.2.9 **Drain Lines.** Provide drain lines from air handling units, fan coil units, etc. Provide a water seal on drains as required. Terminate condensate drain lines in accordance with the IMC.

2-4.2.10 **Backflow Preventers.** Backflow preventers are required at all connections to the potable water system in accordance with the IPC.

2-4.2.11 **Chemical Feeders.** Fill openings should be no higher than 1.2 meters (4 feet) above the finish floor for ease of filling.

2-4.2.12 **Air Vents.** Provide in locations as required in the IMC. Provide manual type vents where possible. Use of automatic air vents is discouraged and should be minimized. Pipe the drains from automatic vents away from concealed areas for visual inspection and to prevent damage to ceilings, etc. Provide manual shut-off valves or stop-cocks for automatic air vents.

2-4.2.13 **Drain Valves.** Provide drain valves at all low points in piping systems. Pipe drain valves to floor drains where possible. Where not possible, provide hose connection.

2-4.2.14 **Check Valves.** Provide check valves to prevent backflow and at the discharge of most pumps. When used in drain lines, verify sufficient head to open flap to regain flow. Provide non-slam type on high head applications. Provide damping type on air compressor discharges.

2-4.2.15 **Freeze Protection.** Design pipe temperature maintenance systems (i.e. heat trace) to the lowest recorded temperature in UFC 3-400-02, *Engineering Weather Data*.

2-4.2.16 **Underground Piping Systems.** Underground piping systems for steam, condensate and chilled and hot water must be factory-prefabricated, pre-insulated, and direct bury type. The Underground Heat Distribution System manufacturer is the company responsible for the design and manufacture of the pre-engineered system. The manufacturer directs the installation of their system, and provides a representative on the job site.

2-4.2.17 **Legionella Disease.** Design waterside systems to avoid potential exposure to Legionella Disease.

2-4.3 **Building Exhaust System.**

Provide exhaust system for removal of heat, fumes, dust, and vapors in various spaces in accordance with ASHRAE. If natural ventilation is proposed, provide calculations to support its use as a reliable means of ventilation.

2-4.3.1 **Equipment Room Ventilation.** Provide mechanical and electrical equipment rooms with 10 air changes per hour or an exhaust rate to limit room temperature rise to 5.6 degrees C (10 degrees F) above the outdoor summer design dry bulb, whichever is greater. Ventilate equipment rooms with a thermostatically controlled exhaust fan, and a weather tight inlet air louver or hood. To ensure that equipment rooms containing combustion burners for boilers, water heaters, or furnaces do not operate as negative pressure areas, utilize supply fans rather than exhaust fans for ventilation. For design heating temperatures less than 4.4 degrees C (40 degrees F), provide motor operated, normally closed dampers at air inlet and exhaust openings. Equipment rooms containing refrigeration equipment shall be ventilated in accordance with IMC and ASHRAE Standard 15.

2-4.3.2 **Exhaust/Intake Locations.** Provide adequate separation between outside air intakes and exhaust outlets, waste vents and boiler stacks. Consider prevailing winds and force protection requirements. Outside air intakes must be 3.0 m (10 ft) minimum above ground elevation to satisfy Anti-Terrorism (AT) requirements.

2-4.3.3 **Roof Fans.** Roof exhaust fans should be avoided due to maintenance access restrictions and roof leak potential. If provided and where feasible, utilize direct drive fan motors with speed controllers to reduce maintenance requirements.

2-4.4 Fire Station Diesel Exhaust.

Provide an engineered fire apparatus exhaust removal system. Refer to Interim Technical Guide (ITG) # FY00-06 for additional information. The system should include an overhead sliding track mechanism to permit a flexible exhaust hose to travel with the fire apparatus into and out of the apparatus bays. The fire apparatus exhaust hose shall automatically disconnect from the vehicle as it exits the bay.

2-4.5 Maintenance Bay Vehicle Exhaust.

Provide an engineered vehicle exhaust removal system. The system shall include an overhead or under floor system. Overhead ductwork system shall be provided with a retractable flexible exhaust hose to travel from the vehicle exhaust into and out of the ductwork. The exhaust fan for all systems shall be specifically designed and manufactured for vehicle exhaust.

2-4.6 Kitchen (Galley) HVAC Systems.

Check project documentation to determine if air conditioning of kitchens is allowed. No air shall be returned from the kitchen to the HVAC system. Design dining facilities in accordance with UFC 4-722-01 and so that air flows from dining areas to kitchen areas to provide make-up air for kitchen exhausts. Maximize the use of dining area make-up air to the kitchen as this will provide secondary cooling for the kitchen staff. If additional make-up air is required for kitchen exhausts, provide push-pull kitchen hoods with built-in heated make-up air supply. Design kitchen hoods in accordance with UFC 4-722-01. Kitchen hoods with built-in make-up air shall be of the horizontal face discharge type. "Short circuit" hoods are prohibited. Provide control interlocks for supply and exhaust fans to ensure that the HVAC system balance is maintained and that the proper direction of airflow is maintained during normal operations. Do not utilize evaporative coolers on kitchen supply air in humid areas. The increased humidity of the ventilation air will negate any small cooling affect. Provide fire suppression system for hoods in accordance with UFC 3-600-10N, *Fire Protection Engineering*.

2-4.7 Industrial Ventilation Systems

2-4.7.1 **General.** Design industrial ventilation systems in accordance with the latest edition of *Industrial Ventilation, A Manual of Recommended Practice*, published by American Conference of Government Industrial Hygienists (ACGIH). For Navy projects, also comply with UFC 3-410-04N. Air Force projects shall comply with AFOSH Standard 161-2.

2-4.7.2 **Design Guidelines.** Research the process or operation before design starts (i.e. find out contaminants, toxicity, process temperature, etc.).

2-4.7.3 Provide hoods designed for effective capture of contaminants while minimizing air flow for energy conservation. Do not specify or provide a canopy hood unless process is nontoxic.

2-4.7.4 Specify the appropriate fan for the application. When selecting a fan, consider noise generation, material handled through the fan (e.g., corrosives, flammables, etc.), and future expansion or flexibility of the system.

2-4.7.5 Provide tempered make-up air for all ventilation systems. Ensure that make-up air does not cause turbulence at the exhaust hood. Interlock make-up air fan to exhaust fan. Do not recirculate exhaust air.

2-4.7.6 Provide an offset discharge stack, with drain, for exhaust systems. Do not utilize a "conical cap" exhaust stack. Provide at least 7.5 m (25 feet) between exhaust outlets and outside air inlets to prevent circulating contaminated exhaust air back into the building.

2-4.7.7 Provide an air cleaning device when required by state and federal regulations. Obtain clear guidance and direction from the Government Project Manager. Select air cleaning devices that will maximize contaminant removal and ease of maintenance while minimizing cost.

2-4.7.8 Provide air flow and static pressure calculations with each design following the methods in the latest edition of the ACGIH Ventilation Manual.

2-5 TERMINAL & PACKAGE UNITS

2-5.1 System Selection Criteria.

Do not utilize room fan coil units or packaged terminal units, such as individual through-wall heat pumps, for facilities such as office buildings and Bachelor Quarters or for any facility larger than 465 square meters (5000 square feet), unless conditioned make-up air is provided to each space through a central, continuously operating, dedicated make-up air system. Conditioned make-up air shall be ducted to each room or to the return side of each fan coil or terminal unit.

2-6 CONTROLS & INSTRUMENTATION

2-6.1 General Controls.

Provide the simplest HVAC controls that will accomplish the intended function.

2-6.2 Control Dampers.

Provide parallel blade dampers for two-position, on/off control. Provide opposed blade dampers for modulating applications, but for best performance, their pressure drop should be between 5% and 20% of the total system pressure drop. They are effective for two-position, on/off applications as well, but are more expensive than parallel dampers.

2-6.3 **Carbon Monoxide Detectors.**

In Family Housing containing fuel-fired appliances, provide UL 2034 listed, line voltage operated, residential carbon monoxide (CO) detectors. Detectors shall feature digital readout and shall be located and installed in accordance with the manufacturer's instructions.

2-6.4 **DDC Systems.**

Prior to designing the DDC system, check with the Government Project Manager to see if an existing energy management network has been established on the Base. Provide DDC equipment which is compatible with existing systems to the maximum extent practicable. Where use of a specific DDC system is mandatory, a Justification and Authorization (J&A) for the utilization of proprietary DDC equipment shall be provided by the Government.

2-6.4.1 For new buildings, DDC systems shall utilize all electric or electronic actuators. On rehab projects, eliminate pneumatics whenever possible. Actuator positions or responses shall be based on position sensors or position feedback indicators and not on controller output signals.

2-6.4.2 For new installations, provide DDC equipment which is user-friendly and which maximizes compatibility with other manufacturers' equipment. ASHRAE's BACnet standard (ANSI/ASHRAE 135) provides guidance on DDC system compatibility. Provide an operator programmable DDC system as a distributed control system.

2-6.4.3 Unless combined with a larger control system, DDC controls for small HVAC systems (i.e.- DX systems of less than ~10 tons) are not cost-effective. Utilize programmable electronic thermostats for these smaller facilities

2-6.4.4 Require the DDC installer to provide training for government facility personnel on all new DDC equipment. Provide training as required by the RFP for Design/build projects, and request guidance from the base Public Works or USAF Base Civil Engineering Office where the project is located for number and type of personnel to attend.

2-6.4.5 Provide Silicon Control Rectifiers (SCRs) when precise control is required.

2-6.4.6 Provide hardware equipment utilizing the latest technology which will accomplish the desired control with a processor speed of not less than 1.1 GHz.

2-6.4.7 Where direct drive fan packages are not available for rooftop fan applications, provide current sensing devices on fan motors to alarm for broken drive belts.

2-6.5 Multiple Chillers.

When multiple chillers are provided, control the chillers by a single central chiller control panel provided by the chiller manufacturer. This is to ensure that the chillers are loaded and unloaded optimally for best performance, reliability, and energy efficiency. Provide connection and communication between the chiller panel and the DDC system.

2-7 SYSTEMS TESTING & BALANCING

2-7.1 Balancing Valves and Cocks.

Provide calibrated balancing valves for hydronic balance. The designer shall specify the size of the balancing valves required in each application, cognizant of the required differential pressure requirements in the pipe systems; do not assume line size valves as appropriate for the application. A balancing device is required in coil bypasses only when coil drops are in excess of 6 kPa (2 feet w.g.).

2-7.2 Flow Control Balancing Valves.

Provide flow control balancing valves in the discharges of all closed circuit pumps and at all hydronic terminals. For pipe sizes larger than 80 mm (3 inches), a flow orifice combined with a butterfly valve shall be specified. Install all flow control balancing valves in accordance with the manufacturer's recommendations regarding the minimum straight lengths of pipe up and downstream of the device. Designers shall select the proper size flow control-balancing valve for each application to ensure the devices are not oversized; valves shall be selected using the median flow rating indicated in the manufacturer's published performance data. Oversized flow control balancing valves yield inaccurate flow readings.

2-7.3 Balancing Dampers.

Provide manual volume dampers for all main and branch ducts; these should include all supply, return, and exhaust ducts. Do not use splitter dampers or air extractors for air balancing; neither are endorsed by SMACNA for balancing applications. Provide opposed blade manual balancing damper for outside air. Indicate opposed blade manual balancing dampers for both the main supply and return duct and the main relief duct on all return air fans; dampers shall be in close proximity to the automatic return and relief dampers.

2-7.4 Duct Leakage and Testing.

All new duct systems, except ducts under 1 inch static pressure, shall be leak tested, unless the requirement is waived by the Government. Refer to Appendix D for duct pressure table.

2-7.5 Variable Speed Drives.

Variable speed drives on pumps or fans shall not be manually adjusted to achieve system balance. Balance systems to deliver design flows with variable speed drives operating at between 55 and 60 Hz so that maximum operational flexibility is maintained. Replace or adjust fan drive sheaves and throttle pump discharges to achieve system balance. Consider trimming pump impellers on larger systems.

2-8 OTHER HVAC SYSTEMS AND EQUIPMENT

2-8.1 Antiterrorism.

Design all inhabited buildings to meet the requirements of UFC 4-010-01, *DoD Minimum Antiterrorism Standards for Buildings*, and/or Combatant Commander Anti-terrorism/Force Protection construction Standards.

2-8.2 Conflicts.

Avoid conflicts with other disciplines and building features. Most common are: electric lights and diffusers; electric duplex outlets and fin radiation; rain leaders or soil stacks and ductwork; bond beams or joists and ducts, etc.

2-8.3 Clearances and Equipment Service Space.

Ensure that all equipment will fit allotted space with manufacturers' recommendations for service and maintenance space adopted. Indicate on drawings filter and tube or coil pull areas for all major equipment, including chillers, boilers, converters, etc. Verify adequate door dimensions to permit passage of equipment into mechanical spaces.

2-8.3.1 **Electrical Rooms.** No pipes (pressure or gravity) shall be installed within, or pass through, electrical or communication rooms.

2-8.4 Seismic.

Pipe and duct supports must comply with the requirements of SMACNA *Seismic Restraint Manual*. Provide details to structural engineer for support verification and sizing.

CHAPTER 3 DESIGN ANALYSIS AND DOCUMENTATION

3-1 GENERAL

3-1.1 Field Investigation.

Conduct detailed field investigation and interview the appropriate field personnel. Do not rely solely on the as-built drawings.

3-1.2 Energy Studies.

The design A&E shall satisfy the energy conservation requirements in accordance with UFC 3-400-01.

3-1.3 Energy Standard.

All new facilities and major renovation projects shall conform to ASHRAE/IESNA Standard 90.1. Note that compliance with this Standard imposes Architectural, Mechanical, and Electrical requirements on the design of the facility. Provide solar analysis for Design/Build projects when required in Project Program. Provide the following for Design/Bid/Build projects:

3-1.3.1 **Energy Analysis Form.** The number and type of alternatives to be analyzed shall be based on project information provided in the scope of work. The Energy Analysis Form (Form E-1) shall be submitted with the proposed alternatives and zones and shall be accompanied with the best available floor plan clearly depicting the zones. Upon submission to the Government by the design A&E at the project concept stage, the Government will review the recommendations and return the form to the A&E: “Approved”, “Approved as Noted”, or “Disapproved”. Contact the Government Project Manager prior to submitting Form E-1 if you have any questions. A copy of the form is included in Appendix D.

3-1.3.2 **Solar Analysis.** When required by the Scope of Work, the economic feasibility of incorporating an active solar domestic water preheating system will be evaluated by the Government with building information provided by the A&E via submission of the Solar Analysis Form (Form S-1) at the project concept stage. A copy of the form is included in Appendix D.

3-1.4 Computerized Energy Analysis.

After receiving the approved forms from the Government, the A&E shall perform a computerized energy analysis and a life cycle cost analysis in accordance with the Scope of Work and UFC 3-400-01.

3-2 DESIGN CONDITIONS

3-2.1 **Outside Design Temperatures.**

Utilize the Unified Facility Criteria document, UFC 3-400-2, *Design Engineering Weather Data*, and utilize the Design Criteria Data available from the referenced Air Force Combat Climatology Center website. For Design/Build projects, the data may be defined in the RFP documents.

3-2.1.1 **Cooling Systems**:

3-2.1.1.1 **Mission-Critical Facilities, where equipment failure due to high heat would be unacceptable:** For design utilize the "0.4% Occurrence" value for outside air "Dry Bulb Temperature (T)" "Design Value (°F)" and the "Mean Coincident (Average) Values" "Wet Bulb Temperature (°F)" for the Design Cooling Day.

3-2.1.1.2 **Humid Area Facilities, Specialized De-humidification Systems, and 100% Outside Air Systems:** For design, utilize the "1% Occurrence" value of outside air "Dry Bulb Temperature (T)" "Design Value (°F)" and the "Mean Coincident (Average) Values" "Wet Bulb Temperature (°F)" for the Design Cooling Day. Also, design for Maximum Humidity conditions, using the "1.0% Occurrence" value of outside air "Humidity Ratio (HR)" "Design Value (gr/lb)" and the "Mean Coincident (Average) Values" "Dry Bulb Temperature (°F)."

3-2.1.1.3 **Other Typical Facilities and Systems:** For design, utilize the "1% Occurrence" value of outside air "Dry Bulb Temperature (T)" "Design Value (°F)" and the "Mean Coincident (Average) Values" "Wet Bulb Temperature (°F)" for the Design Cooling Day.

3-2.1.1.4 **Cooling Towers or Evaporative Cooling Equipment:** For sizing, utilize the "Median of Extreme Highs" value for outside air "Wet Bulb Temperature (T)" "Design Value (°F)" and the "Mean Coincident (Average) Values" "Dry Bulb Temperature (T)" for the Design Cooling Day.

3-2.1.2 **Heating Indoor Design Conditions.** Space Design conditions shall be 21.1 Cdb (70 Fdb) & a minimum of 30% RH, during the Design Heating Day outside air conditions. At all other than design day, occupied times, maintain the space within the "Winter" conditions shown in ASHRAE Handbook of Fundamentals – 2001, Chapter 8, Figure 5, but not more than 21.1 Cdb (70 Fdb).

3-2.1.2.1 **Heating Equipment:** For design, utilize the "99% Occurrence" value for outside air "Dry Bulb Temperature (T)" "Design Value (°F)."

Process heating conditions are determined by the respective process requirements.

> Note: Spaces requiring comfort heating shall be maintained at temperatures no higher than 21.1 Cdb (70 Fdb). During unoccupied hours, temperatures shall be set no higher than12.8 Cdb (55 Fdb).

3-2.1.2.2 **Heating Inside Design Conditions for Laboratories, Shops, Warehouses, etc:** Space Design conditions shall be 18.3 Cdb (65 Fdb) during the Design Heating Day outside air conditions for areas with moderate activity employment, 15.5 Cdb (60 Fdb) for areas with heavy activity employment, and 10 Cdb (50 Fdb) for storage areas.

3-2.1.3 **Cooling Indoor Design Conditions.** Space Design conditions shall be 24.4 Cdb (76 Fdb) & 50% RH, during the Design Cooling Day outside air conditions. At all other than design day, occupied times, maintain the space within the “Summer” conditions shown in the latest edition of ASHRAE Handbook of Fundamentals, but not less than 24.4 Cdb (76 Fdb). 100% Outside Air systems shall operate continuously in Humid Areas, to prevent mold growth.

Process cooling conditions are determined by the respective process requirements.

> Note: Spaces authorized comfort cooling shall be designed for inside temperatures no lower than 24.4 Cdb (76 Fdb). During unoccupied hours, cooling systems shall be secured where appropriate.

3-3 BASIS OF DESIGN

3-3.1 Plumbing Basis of Design.

Address the following:

3-3.1.1 **Design Criteria.** Identify the governing codes and criteria, including federal and military handbooks, utilized for the design. Include the titles and the date of the latest edition or publication. References to codes and criteria should be made in the narratives of the Basis of Design.

3-3.1.2 **Estimated Water Demand.** Estimate the water demand in L/s (gpm) based on the type and number of fixtures required for each building.

3-3.1.3 **Water Pressure.** Indicate the minimum and maximum water pressure in kPa (psi) at each building. Indicate if booster pumping will be required.

3-3.1.4 **Domestic Hot Water.** Indicate the type, size and design water temperature of the domestic water heater and the distribution system. Indicate the extent of domestic hot water recirculation within the building. If shown economically feasible by life cycle cost analysis, state whether heat recovery will be utilized.

3-3.1.5 **Special Mechanical Systems.** Provide a description of special mechanical systems such as compressed air, hydraulic, nitrogen, lubrication oil, etc.

3-3.1.6 **Backflow Prevention.** Identify the systems and fixtures requiring backflow preventers.

3-3.2 **Mechanical Basis of Design.**

Address the following:

3-3.2.1 **Design Criteria.** Identify the governing codes and criteria, including federal and military handbooks, being utilized for the design. Include the titles and the date of the latest edition or publication. References to codes and criteria should be made in the narratives of the "Basis of Design".

3-3.2.2 **Design Conditions.** Provide a tabulation of the design indoor and outdoor heating and cooling conditions for all occupied and unoccupied areas.

3-3.2.3 **Base Utilities.** Describe the source of thermal energy that will be used (i.e. extension of central high pressure steam, hot water, natural gas, or stand alone heat source with the type of fuel utilized). Where more than one source of thermal energy is considered economically feasible, or where a facility is deemed appropriate for study as defined under the heading entitled "Energy Computations", include a computerized Life Cycle Cost Analysis to justify the selection. Metric and English conversion factors are shown in Table D-1 in Appendix D.

3-3.2.4 **Heating System.** Provide a description of the heating system proposed, including an explanation of why this system is preferred over others. Indicate locations of major components of the system. Resistance electricity and L.P. gas are not allowed for space comfort heating, except with approval of the mechanical branch head.

3-3.2.5 **Ventilation System.** State whether a gravity or mechanical system is to be used and provide a brief description of the ventilation system proposed. Indicate the outside air ventilation rates in cfm/person (L/s/person) for various room types. The prescribed rates must be in compliance with the latest edition of ASHRAE 62. Describe the operation of the ventilation system in summer and winter modes. Indicate the number of outside air changes per hour in various areas, the type of infiltration, and whether OSHA requirements are applicable.

3-3.2.6 **Cooling System.** Provide a description of the cooling system proposed including an explanation of why this system is preferred over others. Indicate locations of major components of the system. Identify special humidification or dehumidification requirements. Indicate ASHRAE Standard filter efficiencies and any other special filtration requirements.

3-3.2.7 **HVAC Control System.** Briefly describe the HVAC control system type and its functions. If applicable, indicate a requirement to tie into an existing Base-wide EMCS.

3-3.2.8 **Sustainable Design.** Briefly describe all energy and water conservation features, systems, and components used in the project and the expected energy savings. Describe all features being utilized for lead credits and include the completed LEED forms.

3-3.2.9 **Energy Conservation.** Provide mechanical system based on lowest life cycle cost. Provide completed compliance forms provided in ASHRAE 90.1 User's Manual and any additional documentation to support compliance with this Standard, including a narrative describing the method of compliance, descriptions of building systems and components to be incorporated, and computer analysis discussion, input and output. Provide a signed statement by a registered mechanical engineer indicating compliance with ASHRAE Standard 90.1.

3-4 CALCULATIONS

3-4.1 Plumbing Calculations.

Plumbing system design shall comply with the requirements of UFC 3-420-01.

The following calculations are required:

3-4.1.1 **Domestic Hot Water Heating.** Calculate the hot water storage and demand requirements of the facility. Indicate the basis for the calculations including the incoming and storage water temperatures, the facility type, fixture types, fixture quantities, and the demand and storage factors.

3-4.1.2 **Domestic Water Pressure Calculations.** Determine the sufficiency of the water pressure available at the building to meet the required minimum fixture outlet pressure. Provide detailed pressure loss calculations including losses attributed to meters, fittings, pipe, backflow preventers, and pipe risers.

3-4.1.3 **Domestic Hot Water Recirculation.** Reference the plumbing code by which the domestic hot water recirculation rate is calculated. Calculate the recirculation rate and the recirculation pump head.

3-4.2 Mechanical Calculations.

The following calculations are required:

3-4.2.1 **"U" Factor Calculations.** Utilize the latest edition of ASHRAE Standard 90.1 to determine the minimum "U" factors. Calculate "U" factors for all composite wall and roof systems using the latest edition of ASHRAE Fundamentals. Include cross sections drawings of all wall and roof systems to supplement the calculations.

3-4.2.2 **Building Exhaust Calculations.** Calculate exhaust requirements for removal of heat, fumes, dust, and vapors in various spaces in accordance with ASHRAE. Provide a building exhaust summary.

3-4.2.3 **Outside Air Requirements/Calculations.** Calculate the outside air ventilation requirements as prescribed by the latest edition of ASHRAE Standard 62. Calculations must consider the factors of "Multiple Spaces", "Ventilation Effectiveness" and "Intermittent or Variable Occupancy" as specified in ASHRAE Standard 62. Provide a summary showing compliance with the ventilation requirements.

3-4.2.4 **Building Air Balance Calculations.** Provide air balance calculations addressing the relationship between supply, return, outside air, and exhaust air quantities and indicating pressurization. Special requirements for space pressurization shall be reflected and referenced in the air balance calculations.

3-4.2.5 **Heating and Cooling Load Calculations.** Use of professionally recognized, nationally used computerized load calculating program is required. Load calculations are required for each room or zone by the ASHRAE method indicated in the latest edition of the Fundamentals Handbook. Copies of input and output data are required. Psychrometric calculations shall be illustrated on psychrometric charts and submitted with the 100% submittal. Computer disks may also be requested (see 100% submittal requirements).

3-4.2.6 **Duct Pressure Drop Calculations.** Provide pressure drop calculations for all supply, return, outside and exhaust air systems. All Variable Air Volume (VAV) supply duct systems shall be sized by the static regain method. Equal friction method shall be used for VAV return ducts. The static regain, equal velocity or equal friction methods may be performed on non-VAV supply duct systems.

3-4.2.7 **Hydronic System Pressure Drop Calculations.** Provide pressure drop calculations for all supply and return piping systems.

3-4.2.8 **Pipe Expansion Calculations.** Provide pipe stress calculations for all low-pressure 103 kPa (15 psi) steam, condensate and hot water piping systems where pipe diameters exceed 100 mm (4 inches) and/or the length exceeds 30 m (100 linear feet) without a directional change. Provide pipe stress calculations for all medium and high-pressure steam and high temperature hot water systems.

3-4.2.9 **Equipment Sizing Calculations.** Provide equipment sizing calculations and psychrometric calculations and charts, if applicable, to justify the selection of equipment, including the following:

a. Terminal equipment including VAV boxes, fan coil units, etc.
b. Pumps.
c. Control valves and dampers.
d. Meters and metering devices.
e. Fans.
f. Air Handling Units.
g. Chillers.
h. Boilers.
i. Closed Circuit Coolers and Cooling Towers.

3-4.2.10 **Heat Gain Calculations.** Perform heat gain calculations for duct systems using 90% insulation efficiency. Include heat gain from chilled water pumps on the chilled water system. Size terminal cooling coils with the effects of pump heat gain considered.

3-4.2.11 **Duct Leakage Calculations.** Provide for high pressure systems 746 Pascals or greater (3 inches of water column or greater). Calculate the expectant duct leakage based on the designer's requirements for the duct, seal, and leakage classes for each duct system using the latest edition of the SMACNA *HVAC Air Duct Leakage Test Manual*.

3-5 DRAWINGS

Drawings shall be sufficiently complete to indicate all aspects of installation. Where alternate methods or systems are intended, drawings must detail both alternatives. Judgement should be exercised to avoid overly congested drawings.

3-5.1 Drawing units

Unless otherwise authorized, the SI system of measurement shall be utilized on the drawings in accordance with UFC 1-300-09N, *Design Procedures*. Metric and English pipe sizes are listed in Table D-2 in Appendix D.

3-5.2 Seismic.

Show all pertinent seismic detailing on the contract drawings.

3-5.3 Plumbing Drawings

3-5.3.1 **Demolition.** "Demolition" plans should be separate and distinct from "new work" plans.

3-5.3.2 **Orientation.** Provide north arrows an all building and site plans. The orientation of plumbing drawings shall be arranged with the north arrow toward the top of the plotted sheets, unless overriding circumstances dictate otherwise. The orientation of all partial building or site plans shall be identical to that of the larger plan from which it is derived or referenced. Consistency in drawing orientation shall be maintained with all disciplines.

3-5.3.3 **Legend.** Provide legends to clarify all symbols and abbreviations used on the plumbing drawings.

3-5.3.4 **Enlarged Plans.** Enlarged plans shall be drawn at no less than 1:50 (¼" = 1'-0").

3-5.3.5 **Riser Diagrams.** Provide separate waste and water riser diagrams for all fixture groupings. All riser diagrams shall be drawn 3-dimensional (flat, 2-dimensional risers are unacceptable) and shall account for all pipe directional changes indicated on the floor plans.

3-5.3.6 **Plumbing Fixture Schedule.** Provide a fixture schedule utilizing fixture designations coordinated with the contract specifications.

3-5.4 Mechanical Drawings

3-5.4.1 **Demolition.** "Demolition" plans should be separate and distinct from "new work" plans.

3-5.4.2 **Orientation.** Provide north arrows an all building and site plans. The orientation of mechanical drawings shall be arranged with the north arrow toward the top of the plotted sheets, unless overriding circumstances dictate otherwise. The orientation of all partial building or site plans shall be identical to that of the larger plan from which it is derived or referenced. Consistency in drawing orientation shall be maintained with all disciplines.

3-5.4.3 **Legend.** Provide legends to clarify all symbols and abbreviations used on the mechanical drawings.

3-5.4.4 **Design Conditions.** Provide a schedule indicating indoor and outdoor design temperatures for each room type.

3-5.4.5 **Floor Plans.** Exercise judgment to avoid overly congested drawings. When drawing congestion is likely, ductwork and piping should not be shown on the same plan.

3-5.4.6 **Sections and Elevations.** Provide as required to supplement plan views.

3-5.4.7 **Enlarged Plans.** Mechanical rooms should be drawn at no less than 1:50 (¼" = 1'-0"). Congested mechanical rooms shall be drawn at no less than 1:20 (½" = 1'-0"). Mechanical room plans should be supplemented by at least one section; at least two sections for more complex, congested applications.

3-5.4.8 **Schematic Diagrams.** Provide a 3-dimensional isometric diagram representing the mechanical room piping or a 2-dimensional diagram indicating the entire system.

3-5.4.9 **Kitchen Hood Diagram.** Provide a detailed air balance diagram on the drawings for every kitchen/dining facility design to show compliance with the ventilation requirements. Indicate required capture velocities and capture distances for all hoods on the drawings. Provide notes and contractor instructions on plans indicating that fan airflows shown for hoods are approximate and requiring the contractor to balance the system to achieve the capture velocities indicated. The scheduled fan and motor size shall allow for adjustment of the airflow.

3-5.4.10 **Details.** Details shall be edited to reflect the configurations and construction materials shown on the plans.

3-5.4.11 **Flow and Slope Arrows.** Indicate the flow direction of pipe on the drawings. Show slope direction and rate of slope on all piping systems.

3-5.4.12 **Duct Construction Classifications.** Indicate duct static pressure, seal and leakage classifications on the drawings in accordance with SMACNA-*HVAC Air Duct Leakage Test Manual*.

3-5.4.13 **Guides for Piping.** Show pipe guide locations on all aboveground anchored piping.

3-5.4.14 **Pipe Anchors.** Show anchor locations on plans. Provide anchor detail(s).

3-5.4.15 **Duct Lining.** Indicate acoustical duct lining where required on the drawings. Drawings shall indicate the inside clear dimensions of ducts with acoustical duct lining.

3-5.4.16 **Door Louvers.** Show location or coordinate with architectural drawings.

3-5.4.17 **Roof Fans.** Details of roof exhaust fans shall include a requirement for airtight seals between the fan frame and the wood nailer of the roof curb. The details shall require the duct of ducted exhaust fans to extend up through the fan curb to a flanged and sealed termination at the top of the curb.

3-5.4.18 **Equipment Supports.** Show hanger rods and structural supports for all ceiling or roof-mounted air handling units, heating/ventilating units, fan coil units, exhaust or supply fans, expansion tanks, etc in drawing details.

3-5.4.19 **Pressure Gauges.** Indicate pressure gauge ranges; system operating pressures should be midrange on the graduated scale.

3-5.4.20 **Cold Water Make-up.** Detail all accessories, to include pressure reducing valves (PRV), relief valves, and backflow preventers. Show pressure reducing and relief valve pressure settings.

3-5.4.21 **Air Vents.** Show location of automatic and manual air vents required in piping systems.

3-5.4.22 **Drain Lines.** Show drain lines from air handling units, fan coil units, etc.

3-5.4.23 **Fouling Factors.** Indicate fouling factors for all water-to-air and water-to-water heat exchangers (i.e. coils, converters, chillers, etc). Indicate in the appropriate equipment schedule. Fouling factors shall be accompanied with their appropriate English or SI units.

3-5.4.24 **Equipment Schedules.** The HVAC equipment actually installed on a project may be different from that used as your basis of design. Therefore, mechanical equipment schedules shall reflect actual required equipment capacities as calculated, not capacities provided by manufacturers' catalog data. This helps ensure that the installed equipment is optimally sized for the application.

3-5.4.25 **Motor Starters.** Indicate motor starter NEMA sizes in the mechanical equipment schedules.

3-5.4.26 **Control Valves.** Indicate flow rates, minimum Cv or maximum pressure drop, nominal valve size, service (i.e. steam, hot water, etc), configuration (i.e. 2-way or 3-way), and action (i.e. modulating or 2-position). Use a “Control Valve Schedule”.

3-5.4.27 **Metric Valve Coefficient.** The metric version of the valve coefficient, Kv, is calculated in cubic meters per second at 1 kPa pressure drop. Do not use Cv, the English version, on a metric project.

3-5.4.28 **Balance Valves.** Contract drawings shall specify the valve size and flow for each application. When an existing system is modified, provide all information required for re-balancing in the construction documents. Detail installation of all flow control balancing valves.

3-5.4.29 **Balance Dampers.** All dampers and their intended locations shall be clearly delineated on the floor plans.

3-5.4.30 **Control Diagrams.** Provide for all HVAC systems. Show controller functions, such as normally open (NO), normally closed (NC), common (C), etc. Indicate all set points.

3-5.4.31 **Thermostats.** Show thermostat locations on the plans. Identify heating, cooling, heating/cooling and ventilation thermostats. Indicate thermometer temperature ranges; system operating temperature should be midrange on the graduated scale.

3-5.4.32 **Humidistats.** Show locations on drawings, when required.

3-5.4.33 **Controls.** Show system control schematics and a detailed written sequence of controls on the drawings for each mechanical system. Describe all controlled equipment operating modes, sequence of events, set points, and alarms. For Direct Digital Control (DDC) systems, include an input/output points list and a system architecture schematic. Table 3-1 indicates a minimum points list per system (to be used as applicable).

3-5.4.34 **Ductwork Testing.** Indicate those HVAC duct systems to be leak tested on the contract drawings. Specify the test type and test pressure for each duct system (supply air, return air, exhaust air, and outside air ductwork) subject to testing. See “Duct Construction Classifications”.

3-5.4.35 **Site Work.** Show the type and routing of the heat source conveyance system on the drawings. Exterior above and below grade steam and condensate distribution and below grade chilled and hot water distribution plans shall be accompanied by profile drawings. Profile drawings shall clearly depict all other utilities in the proximity of the new work.

Table 3-1. DDC Minimum Points List

Hot Water Heating System	VAV System
a) Hot water pump status	a) VAV box inlet velocity pressure
b) Hot water supply temperature	b) Airflow rate of each VAV box
c) Hot water return temperature	c) Fan control start/stop
d) Hot water flow rate	d) Air valve actuator
e) Hot water mixing valve position	e) VAV box damper position
f) Differential pressure across pump	f) Discharge air temperature at each VAV box
g) Boiler status	g) VAV box hot water valve position
h) Alarms	h) Electric reheat (on/off and number of stages)
i) Heat exchanger inlet temperatures	i) Space temperature for each zone with set point adjustment
j) Heat exchanger leaving temperatures	j) Space humidity for each zone with set point adjustment
k) Building steam meter	
l) Variable speed pump drive frequency	

Chilled Water System	Air Distribution System
a) Chiller enable/disable	a) Supply air temperature
b) Chiller status	b) Supply air static pressure
c) Entering and leaving water temperatures at each chiller	c) Supply airflow rate
d) Chilled water flow rates for each chiller	d) Outside air temperature
e) Secondary loop chilled water flow rate	e) Return air temperature
f) Chilled water supply and return temperatures for the central plant	f) Mixed air temperature
g) Water temperature in the common piping of the primary/secondary loop	g) Discharge temperature from each heating or cooling coil
h) Chilled water system differential pressure at central chilled water plant	h) Filter status
i) Chilled water system differential pressured used for control of secondary pumps	i) Supply/return damper positions
j) Primary chilled water pump start/stop	j) Outside air damper positions
k) Primary chilled water pump status	k) Chilled water valve positions
l) Outside air temperature	l) Hot water valve positions
m) Outside air relative humidity	m) Electric heater status (on/off and number of stages energized or % power)
n) Building electrical meter	n) Freezestat
o) Building water meter	o) Smoke detector
p) Cooling tower fan status (high-low-off)	p) Supply fan start/stop
q) Cooling tower fans - Adjustable frequency drive functions and alarms	q) Supply fan speed control
r) Condenser water supply and return temperature	r) Supply fan run status
s) Cooling tower bypass valve position	s) Supply fan fault status
t) Variable speed pump drive frequency	t) Exhaust fan run status
	u) Outside air fan run status
	v) Heat recovery wheel rotation status

APPENDIX A REFERENCES

Utilize the latest Code or standard edition applicable, including any amendments, at the time of award of contract. Where there is a conflict between Naval Criteria and National Codes follow Naval Criteria. Refer to CCB for other applicable criteria. Comply with the required and advisory portions. All work shall comply with the latest edition of all applicable criteria, standards, and codes including, but not limited to, the following:

GOVERNMENT PUBLICATIONS:

Federal Energy Management Program (FEMP)

Energy Star Program

Military Handbooks/Standards

DM 3.05, *Design Manual for Compressed Air and Vacuum Systems*

MIL-HDBK 1013/1A, *Design Guidelines for Physical Security of Facilities*

MIL-HDBK-1012/3, *Telecommunication Premises Distribution Planning, Design, and Estimating*

AF ETL 04-3, *Design Criteria for Prevention of Mold in Air Force Facilities*

Unified Facilities Criteria (UFC)

UFC 1-300-09, *Design Procedures*

UFC 1-200-01, *General Building Requirements*

UFC 3-100-10N, *Architecture*

UFC 3-400-01, *Design: Energy Conservation*

UFC 3-410-02N, *Design: Heating, Ventilating, Air Conditioning and Dehumidifying Systems*

UFC 3-410-04N, *Design: Industrial Ventilation Systems*

UFC 3-400-02, *Engineering Weather Data*

UFC 3-420-01, *Design: Plumbing Systems*

UFC 3-430-08N, *Design: Central Heating Plants*

UFC 3-430-09N, *Design: Exterior Mechanical Utility Distribution*

UFC 3-460-01, *Design: Petroleum Fuel Facilities*

Draft UFC 3-580-10, *Draft Design: Navy and Marine Corps Intranet (NMCI) Standard Construction Practices*

UFC 3-600-01, *Design: Fire Protection Engineering for Facilities*

UFC 3-600-10N, *Fire Protection Engineering*

UFC 4-010-01, *Design: DoD Minimum Antiterrorism Standards for Buildings*

UFC 4-510-01, *Design: Medical Military Facilities*

UFC 4-721-10, *Design: Navy And Marine Corps Bachelor Housing*

Standard Government publications are available at www.hnd.usace.army.mil/techinfo.

UFC are available at http://65.204.17.188//report/doc_ufc.html.

NON-GOVERNMENT PUBLICATIONS:

American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE)

ASHRAE Std. 90.1, *Energy Standards for Buildings Except Low Rise Residential Buildings*

ASHRAE Std. 15, *Refrigeration Equipment Rooms*

ASHRAE Std. 62, *Ventilation Requirements*

ASHRAE Handbooks

International Ground Source Heat Pump Association (IGSHPA)

Ground Coupled Heat Pump System Design Guide

Building Codes

The International Building Code (IBC)

The International Mechanical Code (IMC)

The International Plumbing Code (IPC)

National Fire Protection Association (NFPA)

NFPA 30, *Flammable and Combustible Liquids Code*

NFPA 54, *National Fuel Gas Code*

NFPA 58, *Liquefied Petroleum Gas Code*

NFPA 70, *National Electrical Code*

NFPA 90A, *Air Conditioning and Ventilating Systems*

NFPA 90B, *Warm Air Heating and Air Conditioning Systems*

Sheet Metal and Air Conditioning Contractors' National Association (SMACNA)

HVAC Air Duct Construction Standards

HVAC Air Duct Leakage Test Manual

Seismic Restraint Manual

U.S Green Building Council

LEED Rating System

APPENDIX B BEST PRACTICES

B-1 HVAC SYSTEMS

B-1.1 Consider heat recovery for all air systems or buildings providing 2500 cubic feet per minute or greater of outdoor ventilation air.

B-1.2 Consider variable speed pumping on all distribution systems with pumps two horsepower or greater.

B-1.3 Maximize distribution air and water temperature differences to reduce flow rates.

B-1.4 Consider radiant heating systems for hangars and other large, open areas.

B-2 CHILLERS

B-2.1 Specify efficiency based on IPLV part load efficiency unless chiller is expected to mostly fully loaded.

B-3 COOLING TOWERS

B-3.1 Use induced draft fans instead of forced draft fans.

B-3.2 Use variable speed fan motors for capacity control for fans of two horsepower and over.

B-3.3 Design tower based on ASHRAE 0.4% design wet bulb temperature.

APPENDIX C ENERGY FORMS

1-1 ENERGY AND SOLAR ANALYSIS FORMS.

Contact the Mechanical Engineering Branch prior to submitting any forms, for applicability or if you have any questions. The Energy Analysis Form (Form E-1) and Solar Analysis Form (Form S-1) shall be submitted to the Mechanical Engineering Branch for review and recommendations.

Energy Analysis Form (E-1)

Constr Contr No.: ______________________________

Project No.: P-____________ FY: ___________

Project Title: ___________________________________

Location: ____________________________________

A&E Firm:

Building Information

Estimated Cooling: __________ **(tons) (kW)**
Number of Zones: __________
Building Floor Area: __________ (sf) (sm)
Zone Descriptions: **(attach annotated floor plan)**

Energy Analysis Program (check one):

- ☐ BLAST
- ☐ Carrier EC 20-II HAP
- ☐ DOE 2.1
- ☐ Trane Trace-Ultra
- ☐ Other. Indicate: ___________________
  (provide documentation)

Study Alternatives

Alternatives (Describe heating, ventilation and cooling systems; list primary & terminal equipment, energy source (steam, electric, mech etc), & air or water cooled heat rejection)	FOR **GOVERNMENT** USE ONLY			
	*A	*AN	*D	**Remarks**
#1				
#2				
#3				
#4				

*** "A" – Approved; "AN" – Approved as Noted; "D" – Disapproved & Resubmit**

This completed and signed form must be included in the "Basis of Design."

EFD/EFA Approved: **Name:** ______________________________ **Date:** __________

Solar Analysis Form (S-1)

Constr Contr No.: ____________________

Project No.: P-__________ FY: __________

Project Title: ______________________________

Location: ______________________________

A&E Firm:

Building Information

Escalation Dates:
Contract Award: Month: _____ Year: _____
B.O.D.: Month: _____ Year: _____
Contingency: _____ %
Solar System Analyzed (check one):
☐ DHW
☐ Space Heating
Domestic Hot Water:
Recovery Rate Per Hour: _______ (gph) (Lph)
Back-up Energy Source (check one):
☐ Oil ☐ Steam/Oil ☐ Electric
☐ Gas ☐ Steam/Coal ☐ Other
Building Floor Area: ________ (sf) (sm)
Number of Occupants: ________
Compass Direction of Collector (normal): ________

Study Results

☐ In view of recent past studies for active solar utilization at this location, which clearly indicated solar energy not to be feasible, solar was not studied for this specific project.

☐ A life cycle cost solar analysis was performed by ______________________. Results indicate a solar system, as described above, is not feasible for this project.

☐ A life cycle cost solar analysis was performed by ______________________. Results indicate a solar system, as described above, is cost effective. The following solar study results apply:

Solar Design Cost ($1000)**:** $______________
Solar Collector Area: ______________ (sf) (sm)
Solar System Cost ($1000)**:** $______________
Energy Contribution from Solar System: ______________ (%)
Savings to Investment Ratio (SIR)**:** ______________
Discount Payback (years)**:** ______________
Annual Energy Saved: ______________ (MBtu/yr) (MJ/yr)

This completed and signed form must be included in the "Basis of Design".

EFD/EFA Approved: **Name:** ______________________________ **Date:** ____________

APPENDIX D STANDARD CONVERSIONS AND TABLES

Table D-1. Fuel Conversion Factors

Type of Fuel	Conversion Factors (See note (a))	Notes
Anthracite Coal	28.4 Million Btu/Short Ton	
	29.9 kJ/kg	
Bituminous Coal	24.6 Million Btu/Short Ton	
	25.9 kJ/kg	
Electricity	3413 Btu/KWH	See note (b)
	12.3 MJ	
No. 2 Distillate Fuel Oil	138,700 Btu/Gallon	
	38.7 MJ/L	
Residual Fuel Oil	149,700 Btu/Gallon	
	41.8 MJ/L	
Kerosene	135,000 Btu/Gallon	
	37.7 MJ/L	
LP Gas	95,500 Btu/Gallon	
	26.6 MJ/L	
Natural Gas	1,031 Btu/Cubic Foot	
	38.5 MJ/L	
Purchased or Steam from Central Plant	1,000 Btu/Pound	See note (c)
	2.3 MJ/kg	

Notes:

(a) At specific installations where the energy source Btu content is known to vary consistently by 10% or more from the values given below, the local value may be used provided there is adequate data on file for two years or more to justify the revision and that this value is expected to hold true for at least five years following building occupancy.

(b) When 10% of more of a building's annual heating consumption will be derived from electric resistance heating, the electric resistance portion shall be multiplied by a factor of 2.2 to reflect additional conversion losses.

(c) High temperature, medium temperature, or chilled water from a central plant shall use the heat value of fluid based on the actual temperature and pressure delivered to the 1.5 m (5 ft) line.

Table D-2. Metric Pipe Size Equivalence

NPS (Inches)	DN (mm)	NPS (Inches)	DN (mm)
1/8	6	2-1/2	65
3/16	7	3	80
1/4	8	3-1/2	90
3/8	10	4	100
1/2	15	4-1/2	115
5/8	18	5	125
3/4	20	6	150
1	25	8	200
1-1/4	32	10	250
1-1/2	40	12	300
2	50		

Notes:
1. NPS is the inch-pound designation for "nominal pipe size".
2. DN is the metric designation for "diameter nominal".
3. For pipe sizes over 12 inches, use the conversion factor of 25 mm per inch.

Table D-3. Metric Ductwork Dimensions

Inches	mm
3	80
4	100
5	130
6	150
7	180
8	200
10	250
12	300

Notes:
1. For dimensions over 12 inches, increase mm size in increments of 50.

Table D-4. Ductwork Construction and Leakage Testing

System	Duct Pressure Class Inches of Water Column				Supply				Return/Outside Air		Duct Test Pressure: Inches of Water Column	Notes
					Round/oval		Rectangular					
	Supply Duct	Return Duct	Exhaust Duct	Outside Air Duct	Duct Seal Class	Duct Leak Class	Duct Seal Class	Duct Leak Class	Duct Seal Class	Duct Leak Class		
Packaged Rooftop - VAV	4	-	-	-	A	3	A	6	-	-	4.0	1
	-	-2	-	-	-	-	-	-	A	24	2.0	1
Packaged Rooftop -CV	2	-	-	-	-	-	A	24	-	-	2.0	1
	-	-1	-	-	-	-	-	-	A	24	1.0	1
Air Handling Unit with Economizer-Constant Volume	2	-	-	-	A	12	A	24	-	-	2.0	1
	-	-1	-	-	-	-	-	-	A	24	1.0	1
	-	-	-.5	-	-	-	A	24	-	-	0.5	1
	-	-	-	-1	-	-	-	-	A	24	1.0	1
Series VAV Units	2	-	-	-	-	-	A	24	-	-	2.0	1
	-	-.5	-	-	-	-	-	-	A	24	0.5	1,2
Exhaust Duct	-		-1	-	-	-	A	24	-	-	1.0	2

Notes:
1. Test in accordance with Specification Section 15950, HVAC Testing, Adjusting, and Balancing and the procedures in SMACNA HVAC Air Duct Leakage Test Manual..
2. No test required..

APPENDIX E ABBREVIATIONS AND ACRONYMS

ACGIH	American Conference of Government Industrial Hygienists
AFMS	Airflow Measuring Stations
AFOSH	Air Force Occupational Safety and Health
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigeration and Air Conditioning Engineers
ASME	American Society of Mechanical Engineers
AT	Anti-Terrorism
BH	Bachelor Housing
C	Celsius
CAV	Constant Air Volume
Cdb	Celsius Dry Bulb
CFM	Cubic Feet per Minute
CO	Carbon Monoxide
DDC	Direct Digital Controls
DoD	Department of Defense
EMCS	Energy Monitoring Control System
EPACT05	Energy Policy Act of 2005
F	Fahrenheit
Fdb	Fahrenheit Dry Bulb
FEMP	Federal Energy Management Program
GCHP	Ground Coupled Heat Pump
GHz	Gigahertz
GPM	Gallon Per Minute
HVAC	Heating, Ventilating and Air Conditioning
Hz	Hertz
IBC	International Building Code
IESNA	Illuminating Engineering Society of North America
IGSHPA	International Ground Source Heat Pump Association
IMC	International Mechanical Code
IPC	International Plumbing Code
ITG	Interim Technical Guide
J&A	Justification & Authorization
kPA	Kilopascals
LEED	Leadership in Energy and Environmental Design
m	Meters
mm	Millimeters
NESHAP	National Emissions Standards for Hazardous Air Pollutants
NFPA	National Fire Protection Association
ODS	Ozone Depleting Substances
OSHA	Occupational Safety and Health Administration
Pa	Pascal
PI	Pressure Independent
PRV	Pressure Reducing Valve
psi	Pounds per Square Inch
RFP	Request for Proposal
RH	Relative Humidity

SCR	Silicon Control Rectifier
SFPVAV	Series Fan-powered Variable Air Volume
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
T	Temperature
UFC	Unified Facilities Criteria
UL	Underwriters Laboratories
VAV	Variable Air Volume
w.g.	Water Gauge

Fittings
Any round duct fitting can have an equivalent fitting made in flat oval duct. All seams and joints in the fittings should be continuously welded. If the zinc coating is burned off the steel during the welding, the joints should be painted to prevent corrosion.

Elbows
With elbows, a hard bend elbow denotes the bend in the plane of the duct width, whereas an easy bend elbow denotes the bend in the plane of the duct height. Flat oval elbows may be of the radius type, with the centerline radius equal to 1.5 times the duct width in the plane of the bend. Flat oval elbows may also be mitered, either hard bend or easy bend, and either with or without turning vanes.

Branch Fittings
Any branch fitting in round duct can have an equivalent fitting made in flat oval duct with the branch tap being either round or flat oval. The tap on the flat oval fitting can be located anywhere on the circumference of the fitting body, to minimize the need for elbows to change direction of airflow. The branch tap may be either straight or conical.

If the diameter of a round tap is greater than the height of the flat oval body, a transition can be made from flat oval to round, providing an equivalent area at the base of the transition.

As with round duct fittings, the dimensions may vary with each manufacturer, but the typical dimensions will be similar to those for the round duct fittings.

Transitions and Reducers
Transitions can be made from one size of flat oval to a different size of flat oval, or from flat oval to round. Likewise, reducers can be made from flat oval to flat oval, or flat oval to round. Reducers can be eccentric or concentric.

Notice: Flat oval duct is for positive pressure application only unless special designs are developed for negative pressures. Consult the producer for information on pressure rating.

c. Rectangular Ductwork
HVAC Duct Construction Standards—Metal and Flexible (SMACNA 1995) lists construction requirements for rectangular steel ducts and includes combinations of duct thicknesses, reinforcement, and maximum distance between reinforcements. Transverse joints (e.g., standing drive slips, pocket locks, and companion angles) and, when necessary, intermediate structural members are designed to reinforce the duct system. Ducts 85 inches (2160 mm) and larger at a pressure of 6 in.w.g. (1500 Pa) and greater require internal tie rods to maintain their structural integrity. *Rectangular Industrial Duct Construction Standards* (SMACNA 1980) gives construction details for ducts up to 168 inches (4267 mm) wide at a pressure up to ± 30 in.w.g. (7500 Pa).

d. Flexible Ducts
Flexible ducts connect mixing boxes, light troffers, diffusers, and other terminals to the air distribution system. SMACNA (1995) has an installation standard and a specification for joining, attaching, and supporting flexible duct. Because unnecessary length, offsetting, and compression of these ducts significantly increase airflow resistance, they should be kept as short as possible and fully extended.

UL *Standard* 181 covers testing of materials used to fabricate flexible ducts that are categorized as air ducts and connectors. NFPA *Standard* 90A defines the acceptable use of these products. The flexible duct connector has less resistance to flame penetration, has lower puncture and impact resistance, and is subject to many restrictions listed in NFPA *Standard* 90A. Only flexible ducts that are air duct rated should be specified.

e. Plenums and Apparatus Casings
Carefully analyze plenums and apparatus casings on the discharge size of a fan for maximum operating pressure in relation to the construction detail being specified. On the suction side of a fan, plenums and apparatus casings are normally constructed to withstand negative air pressure at least equal to the total upstream static pressure loss. The accidental stoppage of intake airflow can apply a negative

Cont. Table 21-3 Circular Equivalents of Rectangular Ducts for Equal Friction and Capacity (Metric Units)
Dimensions in mm

Side Rectangular Duct	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	Side Rectangular Duct
1000	1093																				1000
1100	1146	1202																			1100
1200	1196	1256	1312																		1200
1300	1244	1306	1365	1421																	1300
1400	1289	1354	1416	1475	1530																1400
1500	1332	1400	1464	1526	1584	1640															1500
1600	1373	1444	1511	1574	1635	1693	1749														1600
1700	1413	1486	1555	1621	1684	1745	1803	1858													1700
1800	1451	1527	1598	1667	1732	1794	1854	1912	1968												1800
1900	1488	1566	1640	1710	1778	1842	1904	1964	2021	2077											1900
2000	1523	1604	1680	1753	1822	1889	1952	2014	2073	2131	2186										2000
2100	1558	1640	1719	1793	1865	1933	1999	2063	2124	2183	2240	2296									2100
2200	1591	1676	1756	1833	1906	1977	2044	2110	2173	2233	2292	2350	2405								2200
2300	1623	1710	1793	1871	1947	2019	2088	2155	2220	2283	2343	2402	2459	2514							2300
2400	1655	1744	1828	1909	1986	2060	2131	2200	2266	2330	2393	2453	2511	2568	2624						2400
2500	1685	1776	1862	1945	2024	2100	2173	2243	2311	2377	2441	2502	2562	2621	2678	2733					2500
2600	1715	1808	1896	1980	2061	2139	2213	2285	2355	2422	2487	2551	2612	2672	2730	2787	2842				2600
2700	1744	1839	1929	2015	2097	2177	2253	2327	2398	2466	2533	2598	2661	2722	2782	2840	2896	2952			2700
2800	1772	1869	1961	2048	2133	2214	2292	2367	2439	2510	2578	2644	2708	2771	2832	2891	2949	3006	3061		2800
2900	1800	1898	1992	2081	2167	2250	2329	2406	2480	2552	2621	2689	2755	2819	2881	2941	3001	3058	3115	3170	2900
Side Rectangular Duct	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	Side Rectangular Duct

Equation for Circular Equivalent of a Rectangular Duct

$$D_e = 1.30\,[(ab)^{0.625}/(a+b)^{0.250}]$$

where

a = length of one side of rectangular duct, mm.
b = length of adjacent side of rectangular duct, mm.
D_e = circular equivalent of rectangular duct for equal friction and capacity, mm

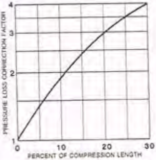


FIGURE 21-4 Correction Factor for Unextended Flexible Duct

TABLE 21-1 Duct Material Roughness Factors

Duct Material	Roughness Category	Absolute Roughness ε, ft	mm
•Uncoated carbon steel, clean (0.00015 ft) (0.05 mm) •PVC plastic pipe (0.0003 to 0.00015 ft) (0.01 to 0.05 mm) •Aluminum (0.00015 to 0.0002 ft) (0.04 to 0.06 mm)	Smooth	0.0001	0.03
•Galvanized steel, longitudinal seams, 4 ft (1200 mm) (0.00016 to 0.00032 ft) (0.05 to 0.1 mm)	Medium Smooth	0.0003	0.09
•Galvanized steel, spiral seams, 12 ft (3600 mm) joints (0.00018 to 0.00038 ft) (0.05 to 0.12 mm)	(New Duct Friction Loss Chart)		
•Hot-dipped galvanized steel, longitudinal seams, 2.5 ft (760 mm) joints (0.0005 ft) (0.15 mm)	Old Average	0.0005	0.15
•Fibrous glass duct, rigid •Fibrous glass duct liner, air side with facing material (0.005 ft) (1.5 mm)	Medium Rough	0.003	0.9
•Fibrous glass duct liner, air side spray coated (0.015 ft) (4.5 mm) •Flexible duct, metallic (0.004 to 0.007 ft) (1.2 to 2.1 mm) when fully extended •Flexible duct, fabric and wire (0.0035 to 0.015 ft) (1.0 to 4.6 mm) when fully extended •Concrete (0.001 to 0.01 ft) (0.3 to 3.0 mm)	Rough	0.01	3.0

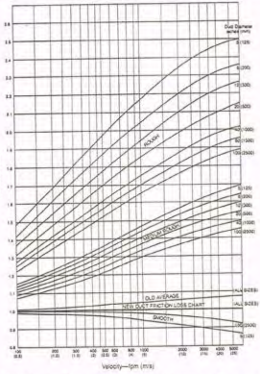


FIGURE 21-3 Duct Friction Loss Correction Factors

energy design resources

design brief

INTEGRATED DESIGN FOR SMALL COMMERCIAL HVAC

By using recommended design methods for rooftop heating, ventilation, and air conditioning (HVAC) systems, significant improvements in operational savings, energy efficiency, and indoor comfort can be achieved.

Summary

Small HVAC systems are the workhorses of the light commercial building market, which represents more than half of the annual commercial new construction floor area in California. Design, installation, and operations issues can prevent these systems from performing up to their full potential. This design brief focuses on actions that the architects, engineers, and design/build contractors can take to improve the energy efficiency of small HVAC systems, reduce operating costs, and improve indoor comfort and environmental quality. These actions include:

- Practice energy-efficient design strategies such as reduced lighting power, high-performance glass and skylights, cool roofs, and improved roof insulation techniques in the overall building design.
- Size units appropriately using American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc. (ASHRAE) approved methods that account for the energy efficiency strategies implemented in the design, and use reasonable assumptions on plug load power and ventilation air quantities when sizing equipment.
- Select unit size and airflow based on calculated sensible loads without oversizing. Consider increasing unit flow rate to improve sensible capacity in dry climates.
- Specify units that meet the Consortium for Energy Efficiency Tier 2 efficiency standards; and incorporate premium efficiency fan motors, thermostatic expansion valves, and factory-installed and run-tested economizers.
- Design distribution systems with lower velocities to reduce pressure drop and noise. Seal and insulate duct systems located outside the building thermal envelope.

CONTENTS

- Operate ventilation systems continuously to provide adequate ventilation air. Incorporate demand-controlled ventilation to reduce heating and cooling loads.

- Specify commercial grade thermostats with the capability to schedule fan operation and heating and cooling setpoints independently.

- Commission the systems prior to occupancy through a combination of checklists and functional testing of equipment control, economizer operation, airflow rate, and fan power.

- Develop clear expectations of the services provided by HVAC maintenance personnel.

Introduction

This design brief incorporates findings from a recent study of small HVAC systems in commercial buildings conducted for the California Energy Commission (CEC).[1] A total of 75 buildings and 215 roof top units were studied. The project identified a number of issues with HVAC systems that are installed and operated in the field. The problems included broken economizers, improper refrigerant charge, fans running during unoccupied periods, fans that cycle on and off with a call for heating and cooling rather than providing continuous ventilation air, low airflow, inadequate ventilation air, and simultaneous heating and cooling. Correcting these problems represents a major opportunity for improvements in energy efficiency, operations, and indoor comfort.

Why Small HVAC?

Packaged direct expansion (DX) air conditioners and heat pumps cool more than half of the total commercial new construction floor space in California.[2] Of these, single package rooftop air conditioners dominate the market, representing approximately three-quarters of the total DX system capacity.

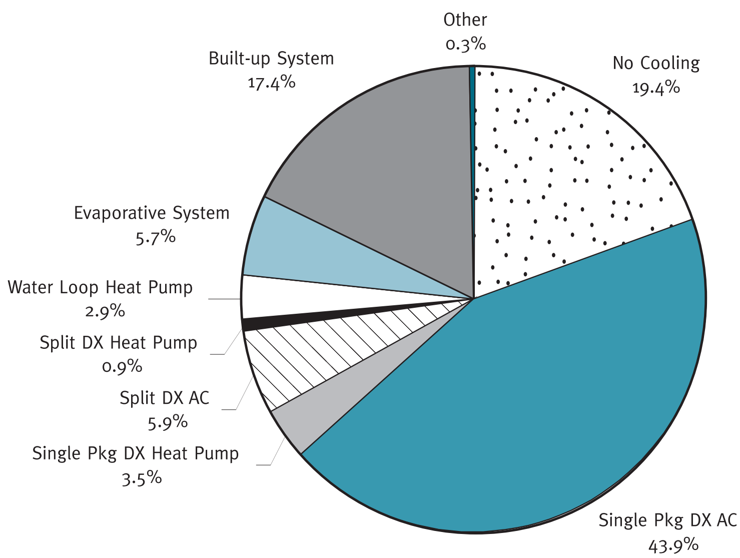
The rooftop air conditioner market is dominated by small systems, defined here as systems 10 tons and smaller, representing almost 60 percent of the total installed DX cooling capacity. The most popular unit size (in terms of units sold) is five tons (Figures 1 and 2).

These small rooftop units are the workhorses of the commercial building industry, yet many systems fail to reach their full potential due to problems with design, installation, and operation.

Figure 1: Floor space distribution of HVAC systems in new commercial buildings in California

Single package DX air conditioners are the most popular HVAC system type in new construction in the state, cooling about 44 percent of the total floorspace. Built-up systems are the second most popular, conditioning about 17 percent of the total floorspace. The combined total of single package and split DX air conditioners and heat pumps represents slightly more than half of the total floorspace in California. Note that a significant portion (about 19 percent) of the total floorspace is not cooled.

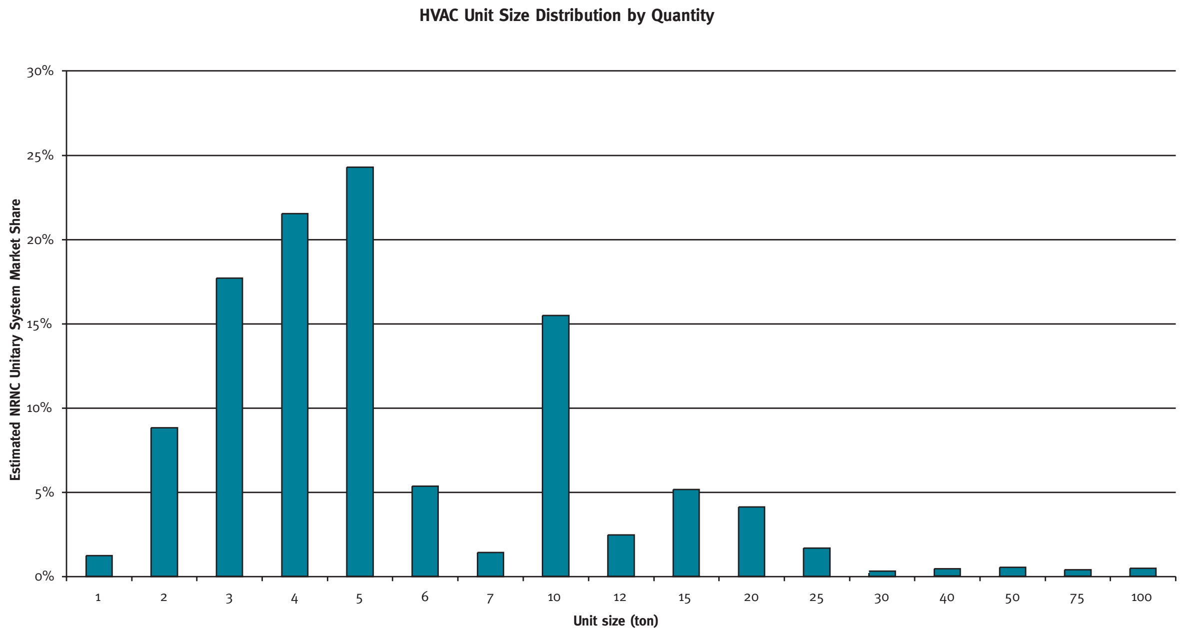
Cooling System Type Distribution by Floorspace



Source: Architectural Energy Corporation

Figure 2: Distribution of packaged DX system size

In terms of number of systems installed, the most popular packaged DX system size is five tons. Units between one and 10 tons represent close to 90 percent of the total unit sales in new buildings in California.



Source: Architectural Energy Corporation

Building Design

HVAC systems, like all systems in the building, do not function in isolation, but are part of an interactive system of components. Before addressing the design of the HVAC system, it is important to address several aspects of building design that influence the loads imposed on the HVAC system. By including these energy efficiency strategies in the building design, the size and energy consumption of the HVAC system can be reduced.

Reduce Lighting Power

Lighting represents a major opportunity for energy savings in small buildings. Although Title 24 is one of the most stringent energy codes in the country, there is ample opportunity to reduce lighting power below Title 24 allowances. New generation T-5 and T-8 lamps, fluorescent high-bay fixtures, task/ambient lighting design, lighting controls, and daylighting represent opportunities to reduce lighting energy and the size of the HVAC system required to remove heat generated by lighting systems. Lighting amounts to approximately 30 percent of the energy consumed in typical office buildings. The lighting designer should try to design to lighting power density levels that are 10 percent less than Title 24 allowances.

Use High-Performance Glazing and Skylights

High-performance glass also represents a major opportunity for energy efficiency in commercial buildings. Tinted, low-e glazing systems that reduce solar heat gain and conduction losses are available from most glass suppliers, thereby reducing the size of the air conditioning system. High-performance glass also improves occupant thermal comfort and reduces glare. Similarly, high-performance skylights are available that reduce solar heat gains and heat conductance, while maintaining sufficient visible light transmission for daylighting applications.

Title 24 requirements exclude single pane glass from most applications, and require double pane, low-e glass in many climate zones. However, glazing systems with higher performance are available in virtually all applications.

Use Cool Roofing Materials

Roofing materials with low solar absorptance and high thermal emittance ("cool" roofs) can reduce peak HVAC loads and energy consumption. Cool roofs reflect solar radiation while enhancing radiant heat transfer to the sky, thus reducing the "roof" load of the building. Reductions in heat gains through the roof have an effect on the temperature of the plenum space located between the drop ceiling and the roof, which contains the majority of the ductwork in small commercial buildings. Duct heat gains and air leakage losses (especially on the return side) can increase HVAC loads on the order of 30 percent, so a cool plenum can reduce energy consumption and improve occupant comfort, especially in commercial buildings where systems run continuously during occupied hours. Cool roofs can also reduce the outdoor air temperature at the roof level.

Figure 3: Lay-in insulation applied to a warehouse-to-office conversion. Note the poor insulation coverage and ductwork located in an unconditioned space.

Source: Architectural Energy Corporation

Avoid Lay-in Insulation

The roof or ceiling insulation location can also have a major effect on HVAC system performance. Roof insulation can be installed directly on the roof deck, while ceiling insulation is sometimes applied on top of the drop ceiling (called "lay-in" insulation).[3] When the insulation is applied to the roof, the plenum is located within the thermal envelope of the building, and the impacts of duct conductive losses and duct leakage on HVAC system efficiency is substantially less. Although the surface area of the thermal boundary of the building expands due to the inclusion of the plenum walls, overall conductance losses decrease due to improved insulation coverage.

Lay-in insulation generally has incomplete coverage due to lighting fixtures, HVAC diffusers, fire sprinklers, and other devices installed into the dropped ceiling grid that interfere with insulation installation. Insulation installed on ceiling tiles inevitably gets displaced as ceiling tiles are moved to gain access to the plenum space for data and telecom wiring, reconfiguring the HVAC diffuser layout, and other maintenance activities. The use of lay-in insulation increases the likelihood of increased duct losses and lower HVAC system efficiency.

These seemingly unrelated aspects of building design can have a profound effect on the size and cost of the HVAC system. Architects and design/build contractors should consider including the above-listed aspects into their designs to achieve superior performance. The incremental costs of these energy-efficiency strategies can be offset by reduced HVAC system size and cost.

Unit Sizing

Many small HVAC systems are significantly oversized, resulting in inefficient operation, reduced reliability due to frequent cycling of compressors, and poor humidity control. Oversized systems also result in wasted capital investment in both the HVAC unit and distribution system. System oversizing affects the ability of the system to provide simultaneous economizer and compressor operation, and exacerbates problems with distribution system fan power, since larger units are supplied with larger fans.

Use Sizing Methods Responsive to Efficiency Strategies

A variety of sizing methodologies are used to determine HVAC system size, including "rule of thumb" sizing based on square foot per ton (sf/ton), manual methods (e.g. ACCA Manual N), and computerized load calculations. A recent survey of design practices in the small commercial building market indicated that although computerized load calculations are used more often than manual methods, the assumptions used in the load calculations are based on conservative assumptions about the building shell, lighting design, and occupant densities.[4] To reap the advantages of lower first costs, energy efficiency strategies that reduce peak loads should be included in the load calculations.

Use Reasonable Assumptions for Plug Loads and Ventilation Air

Engineers often base HVAC sizing decisions on the full nameplate or "connected" load of computers, copiers, printers, and so on; and assume simultaneous operation of such equipment. In fact, most of this equipment operates at a fraction of the nameplate value, and rarely operates simultaneously.[5] Many HVAC designs are based on plug load assumptions on the order of five W/sf in office spaces. According to an ASHRAE study (see sidebar), one W/sf is a reasonable upper bound when equipment diversity and reasonable estimates of the true running load are included.

The peak occupant load and the corresponding ventilation load can contribute substantially to equipment capacity in certain spaces such as lobbies and public assembly areas. Often actual occupant loads are substantially less than peak

ASHRAE Study on Plug Loads in Offices

An ASHRAE study on plug loads measured equipment load densities in 44 commercial office buildings. The measured equipment power ranged between 0.4 and 1.2 W/sf. Values above 1.0 W/sf occurred in only five percent of the square footage studied.

Source: ASHRAE Journal, December 1997.

egress loads to which building codes often defer. While code changes may be in order, it also makes sense for designers to be knowledgeable about the applicable code and balance good air quality with energy efficiency. Many building codes reference ASHRAE Standard 62, which allows the designer to base the design on the actual anticipated occupant density, so long as justification is provided.

Avoid Oversizing

California Title 24 limits cooling capacity to 121 percent of the calculated peak cooling load. Since most sizing methods are based on conservative assumptions, it is recommended that designers use the calculated load and round up to the next available unit size only to avoid excessive oversizing.

Unit Selection

Efficiency

Energy codes are generally set to correspond to the basic “standard efficiency” HVAC unit. High efficiency units are available in most size ranges that are up to 30 percent more efficient than code. These units generally incorporate larger condenser and evaporator coils, efficient compressors, improved cabinet insulation, and higher efficiency fans and motors. Designers should consider specifying units that meet the Consortium for Energy Efficiency (CEE) Tier 2 efficiency standards. It is also important to consider both the rated full load energy efficiency ratio (EER), and the seasonal energy efficiency ratio (SEER) when selecting a unit. However, if the unit design is optimized for efficient part-load rather than peak load operation, multi-compressor units with high SEERs may not perform much better than a standard unit at peak cooling conditions, since the SEER includes part-load efficiency in the overall calculation.

Table 1. Title 20 (2003), Title 24 (2001) and CEE Tier 2 Efficiency Standards

Size	Title 20/24	Tier 2
<5.4 ton	9.7 SEER	13 SEER/ 11.2 EER
5.4–11.2 ton.	10.3 EER	11 EER

Source: CEC and CEE

Select Capacity Based on Design Conditions

Designers should consider the unit capacity under actual design conditions, not nominal values. The peak cooling capacity is reduced as outdoor temperatures increase. This can be especially important in desert climates where peak cooling conditions on the roof can exceed the data in manufacturers' standard catalogs. The unit should be sized to meet the calculated sensible load, and the latent cooling capacity should be reviewed. High-efficiency equipment generally has less latent cooling capacity than standard equipment. Also, energy-efficient buildings have reduced sensible loads but comparable outdoor air requirements compared to standard buildings; thus the sensible heat ratio of an energy-efficient building may be reduced.

Select Airflow Rate to Meet Sensible Loads

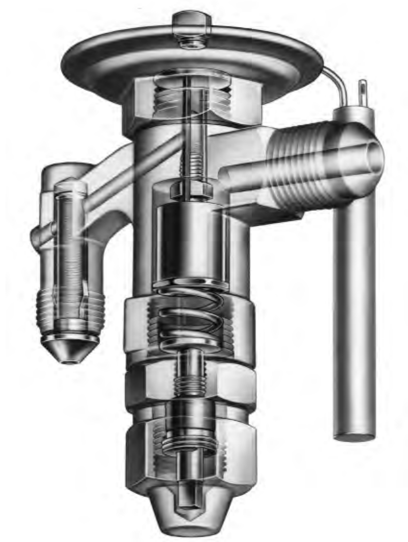
The cooling capacity of most packaged air conditioners is based on a nominal flow rate of 400 cfm (cubic feet/minute) per ton of cooling capacity. Nominal flow rates in packaged equipment are selected to provide adequate dehumidification in climates that are more humid than California. Increasing the flow rate can extract extra sensible cooling capacity out of the unit, allowing the selection of a smaller "nominal" unit. However, the designer should assess the fan energy, which may increase dramatically with higher flow rates, if the unit capacity is near the maximum offered in a particular case size.

Specify Premium Efficiency Fan Motors

Premium efficiency fan motors are important in commercial applications, since fans in general run continuously during occupied periods. In systems equipped with economizers in mild climates such as coastal California, fan energy can be a significant portion of the total HVAC energy consumption. Selection of a premium efficiency motor on the supply fan is cost effective in all climates.

- When selecting a unit, designers should consider peak rooftop temperature and sensible heat ratio under design conditions.
- Designers should also evaluate the trade-off between additional sensible cooling capacity and fan power when selecting air flow rate.

Figure 4: Thermostatic expansion valves (above) and direct drive economizer actuators (below) can improve unit reliability.



Source: Sporlan Valve Company
www.sporlan.com

Source: Belimo Aircontrols
www.belimo.com

Specify Thermostatic Expansion Valves

Refrigerant charge in units degrades over time, due to refrigerant leaks and/or poor maintenance practices. Specifying units with thermostatic expansion valves makes the units more tolerant of refrigerant charge variations by maintaining unit efficiency over a wide range of under- or over-charged conditions. Thermostatic expansion valves are available as a factory option in most units.

Specify Reliable Economizers

Economizers are required by code in units exceeding 6.25 tons and are used in many smaller units. Energy savings from functioning economizers can exceed 50 percent in certain climates and building types. Although most manufacturers offer a factory-installed economizer, the majority of economizers are installed by the distributor or in the field. Specifying a factory installed and fully run-tested economizer can improve reliability.

Distribution Systems

After sizing and selection, the distribution system (ductwork and diffusers) is the next important part of the HVAC system. Installed costs for duct systems can approach the cost of the HVAC unit itself. Often, there is intense pressure to reduce duct system costs. However, the quality of the duct system can have a profound effect on the efficiency and comfort delivered by the HVAC system. Fan energy in small commercial buildings can approach the cooling energy consumption. Duct losses through leakage and conduction can affect the efficiency of the system and the amount of cooling delivered to the space. A poorly balanced distribution system is one of the leading causes of poor indoor comfort in small systems.

Reduce Duct System Pressure Drop

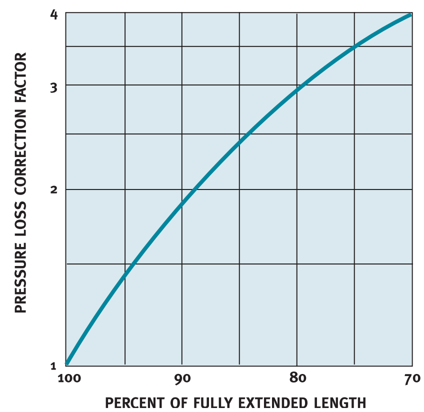
Poor ductwork design can lead to inadequate HVAC unit airflow and excessive fan power. Tested airflow rates in buildings averaged about 325 cfm/ton, rather than the nominal 400 cfm/ton used in system efficiency ratings. Reduced airflow can contribute to coil icing, comfort problems, and a reduction in cooling efficiency of approximately 10 percent.

Design values. Duct systems in small buildings are generally sized using the equal friction or modified equal friction method. Principle design variables are the design velocity (chosen for noise control) or the design friction loss (in Water Columns per 100 ft.). Typical design friction rates are 0.1 inch WC per 100 ft. in commercial buildings. Reducing the design friction rate to 0.05 inch WC per 100 ft. increases the duct size and costs by 15 percent, but cuts the portion of the total pressure drop attributable to the ductwork by 50 percent, and the overall distribution system pressure drop by approximately 40 percent when diffuser losses are included. Upsizing the duct system can provide fan energy savings on the order of 15 to 20 percent.

Use of flex duct. Flex duct, which is used extensively in light commercial construction, has more than a 60 percent higher pressure drop than galvanized metal duct of the same diameter. Flex duct runs should be limited to six feet or less. Flex duct should also be fully extended and well supported at five-foot intervals to minimize pressure losses. The bend radius should be greater than one times the duct diameter to avoid kinking.

Duct layout and fittings. The duct system should be laid out to minimize duct length, turns, and fittings. Radius or section elbows are suggested for all turns greater than 45 degrees. Other recommendations include:

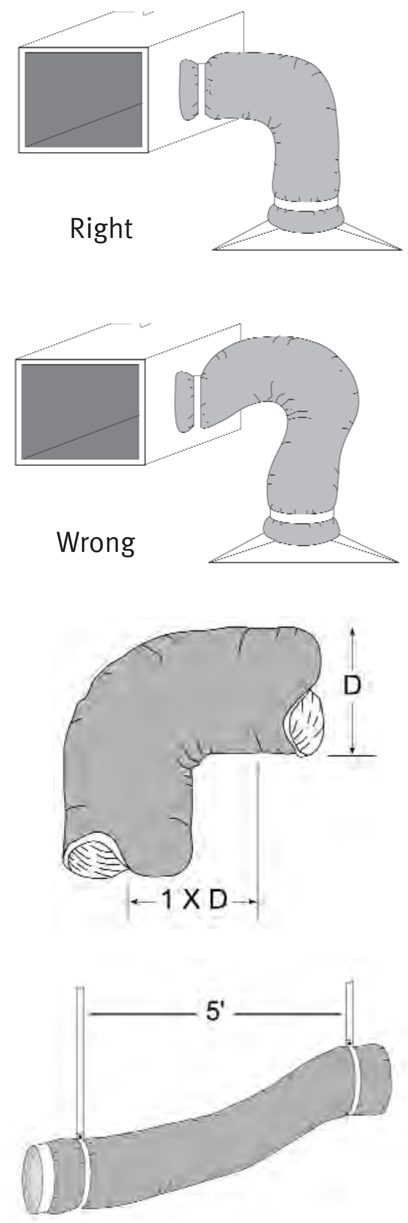
Figure 5: Flex duct should be fully extended to minimize pressure drop. A 30 percent reduction in flex duct extension causes a four-fold increase in pressure drop.



Source: ASHRAE Handbook of Fundamentals (2001)

Figure 6: Flex duct installation guidelines

These installation guidelines should be followed to insure adequate airflow is maintained through distribution systems containing flex duct. See www.flexibleduct.org for more information.



Source: Architectural Energy Corporation

- Use smooth wye branch fittings instead of right angle fittings for branch takeoffs, and avoid turns immediately before a supply or return air register.
- Avoid duct connection details at the unit that degrade fan performance (called the "system effect").
- Provide at least two feet of straight duct before the first turn to minimize noise and loss of fan capacity.
- Install turning vanes in supply ducts at the first turn after entering the building.

Seal Duct Leakage

Leaky ductwork is a common problem plaguing small commercial systems. A recent study of 350 small commercial HVAC systems in Southern California found that 85 percent of the systems tested had excessive duct leakage.[6] The average combined supply and return leakage in these systems exceeded 35 percent of the total air volume, causing energy waste and poor thermal comfort. Cooling energy savings from sealing leakages in duct systems approaches 20 percent. Peak cooling loads are reduced even more when ducts are sealed since attic or outdoor air is extremely warm under peak conditions. Duct leakage testing and sealing should be done prior to installation of a dropped ceiling while access to the duct system is uncomplicated. Contractors should use sealing materials that meet UL Standard 181 such as mesh tape and mastic. Duct tape should not be used to seal duct leaks, since it tends to degrade over time. The duct systems should be sealed to allow a maximum of six percent combined supply and return leakage rate at 25 Pa test pressure.

Aeroseal is a new technique that combines duct leakage testing and sealing into one operation. A calibrated duct pressurization fan is attached to the duct system, and the leakage flow is measured at a preset duct system pressure. An elastomeric aerosol-sealing compound is injected into the duct system until the leakage level is reduced to an acceptable level.

Increase Duct Insulation Levels to R-8

Most duct systems are insulated with one inch of fiberglass insulation (R-4.2). Duct wrap and duct liner two inches thick are commonly available, and improve the insulation level to R-8. Increased insulation is cost effective in duct systems located outside the conditioned space, such as attics or plenum spaces with lay-in insulation, or outdoors.

Reduce Duct System Noise

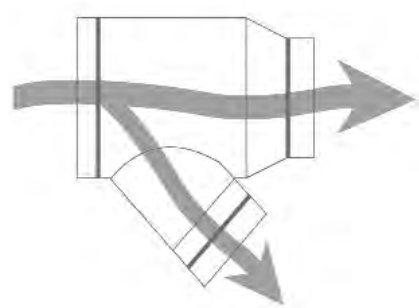
Poorly designed duct systems produce and/or convey noise. Excessive noise can degrade indoor environmental quality (IEQ) and productivity in certain spaces, especially classrooms. Research conducted by the Heshong-Mahone Group for the CEC (see listing under "For More Information" section) listed noise as a leading problem in school HVAC systems.

Reducing the design friction rates also reduces duct velocity, which reduces duct noise. The use of lined ducts should be avoided for noise control, since the duct lining increases pressure drop. A common problem is to solve a noise problem related to high duct velocity with duct liners or silencers, which further increases pressure drop. Increasing duct size and following good design practices at diffuser connections can address noise and pressure drop problems simultaneously.

Ventilation

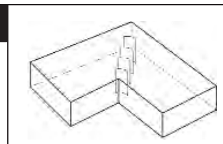
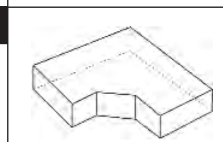
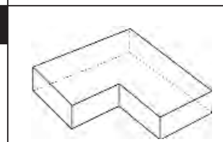
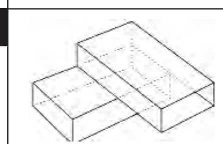
Providing adequate ventilation is a key component of indoor air quality. Strategies to provide adequate ventilation are often at odds with energy efficiency; however, meeting ventilation code requirements should be the first priority of designers and operators of buildings, with the goal of meeting these requirements in the most energy efficient manner possible.

Figure 7: A wye-branch takeoff is recommended instead of a right-angle takeoff since wye-branch creates less pressure loss.



Source: Architectural Energy Corporation

Figure 8: Techniques for reduced pressure loss in 90 degrees turns with rectangle ductwork.

Relative Pressure Loss	
BEST 1.0	
GOOD X 1.3	
FAIR X 4.7	
POOR X 13.0	

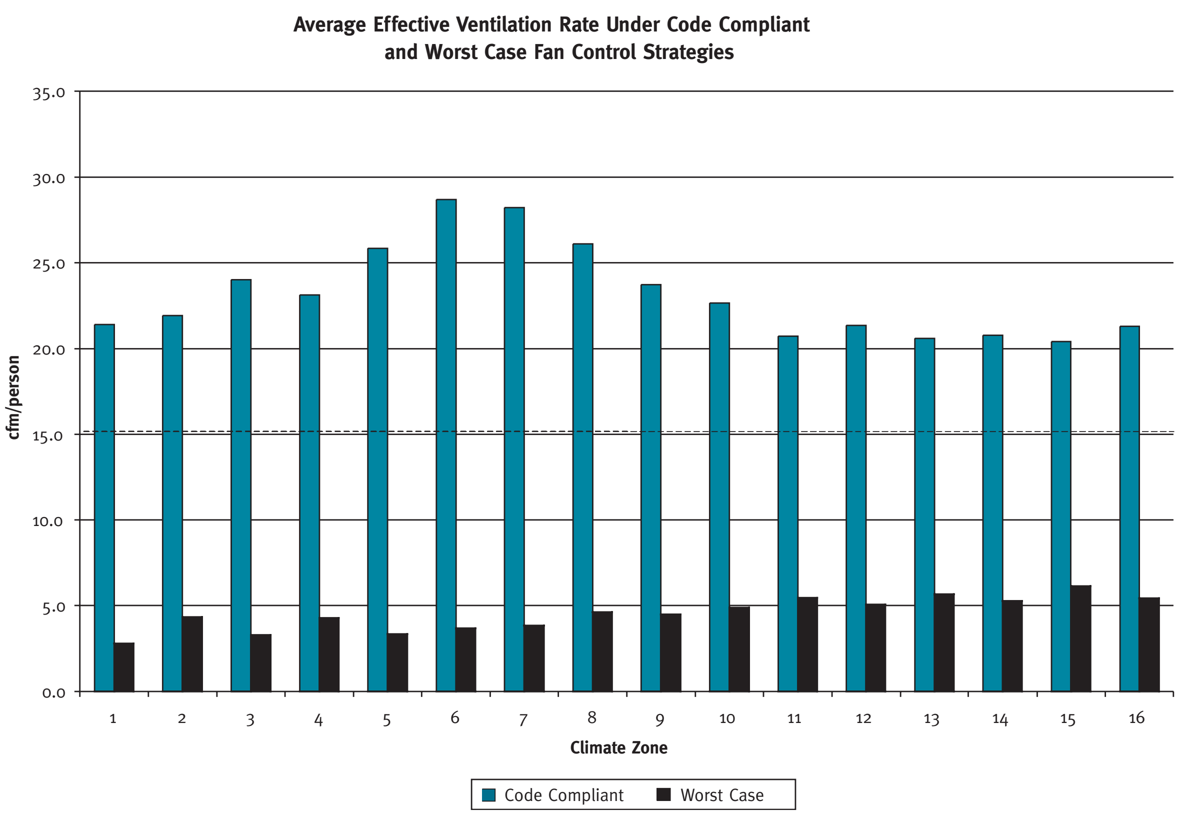
Source: Architectural Energy Corporation

Operate Unit Fans Continuously

Building codes generally require continuous ventilation during occupied hours. This is generally accomplished by operating the HVAC unit fan continuously and introducing fresh air at the unit. When HVAC unit fans are cycled on and off with a call for heating or cooling, the ventilation rates drop dramatically. The effect of cycling fans on effective ventilation rates is shown in Figure 9. It should be noted that the effective ventilation rate for units with cycling fans is on the order of five cfm/person, or about one third the minimum rate mandated by the Title 24 Standards. Continuous fan operation also reduces stuffiness and localized temperature variations that are among the most common complaints in buildings served by small rooftop units.

Figure 9: Effective ventilation rate for HVAC units with continuous and cycling fans

In both cases, the minimum outdoor air damper is set to provide 15 cfm/person of outside air. The code compliant case used continuous ventilation and an air-side economizer. Economizer operation increased the effective ventilation rate above the nominal 15 cfm/person rate. A unit not equipped with an economizer and operated with cycling fans provided an effective ventilation rate of less than five cfm/person in most climate zones.



Source: Architectural Energy Corporation

Use Demand-Controlled Ventilation

Demand-controlled ventilation systems modulate outdoor air quantities based on measured indoor air quality. Indoor CO^2 concentration is commonly used as an indicator of indoor air quality. Many economizer controllers have the built-in capability to implement demand-controlled ventilation with the simple addition of a CO^2 sensor. This strategy can reduce outside air requirements during periods of partial occupancy, and provide energy savings and reduced humidity.

Demand-controlled ventilation is commonly used in systems serving spaces with highly variable occupancies, such as auditoriums, meeting rooms, and so on. These systems can also save energy in other space types with high design occupant densities to prevent over-ventilating the spaces.

Alternative Ventilation Strategies

The HVAC unit supply flow rate is generally four times larger than the required outdoor air ventilation rate, requiring excessive fan power during ventilation-only operation. Alternative design strategies for providing ventilation air, such as two-speed or variable-speed fan systems interlocked with the OA (outdoor air) damper and/or a CO^2 sensor, can be used to reduce fan power during ventilation-only mode. Another strategy is to use a dedicated ventilation fan that brings in a constant supply of fresh air rather than relying on the HVAC unit fan. In this case, the ventilation fan would run continuously during occupied hours, and the HVAC unit fan would cycle on a call for heating or cooling.

Natural ventilation using operable windows can also be used to supply ventilation in lieu of mechanical ventilation. This strategy can be effective in serving perimeter zones in mild climates. Proximity switches installed on operable windows should be used to lock out the HVAC systems when windows are open to prevent energy waste.

Figure 10: CO^2 Sensors

CO^2 sensors attached to a standard economizer controller add demand-controlled ventilation to many rooftop units.

Reprinted with permission of Honeywell International, Inc. www.honeywell.com

Figure 11: Thermostat location

Thermostats controlling three different units serving three different computer labs at a community college are located in the corridor, where they are unable to effectively sense the temperature of the rooms they are controlling.

Source: Architectural Energy Corporation

Thermostats and Controls

Controls used in small HVAC systems come from a variety of sources and may not provide the full range of control options required for optimal system performance. A simple room thermostat is used to control most systems, though energy management systems (EMS) are making inroads into the small commercial building market.

Use Two-Stage, Commercial Grade Thermostats

The primary function of the thermostat is to control the heating and cooling output of the unit, but most thermostats also control the operation of the supply fan. Fans are required to run continuously during operating hours, and cycle on and off with a call for heating or cooling during unoccupied hours. Most of the systems studied have the capability to implement this strategy, yet were not set up correctly. Commercial (not residential) thermostats should be used to provide continuous fan operation/ventilation during occupancy. The thermostat should be programmed for intermittent fan operation during unoccupied hours, and provide a one hour "purge" of the building prior to occupancy. Designers should specify controls with default settings that are appropriate for commercial applications. Systems with economizers should use thermostats with two-stage cooling to allow integrated operation of the economizer and mechanical cooling system.[7]

The location of the thermostat can dramatically affect system loads and occupant comfort. Since the system responds to the air temperature at the thermostat, proper location is key to comfort and energy efficiency. Locating several thermostats in the same general area with conflicting heating and cooling setpoints can invite problems with simultaneous heating and cooling, where adjacent units "fight" each other to maintain selected setpoints.

Controller Options and Interfaces

Modern HVAC units can be configured with a variety of controller options, including standard electromechanical controls, microprocessor controls, and controllers with EMS interface capability. Standard controls allow the use of thermostats from a variety of vendors. In some units with microprocessor control, the thermostat control logic is contained within the unit controller and the zone thermostat is merely a temperature sensor. Interfaces to energy management systems allow the units to be controlled by one of several energy management systems, including both manufacturer-supplied systems and third party systems. These interfaces allow the energy management system to take over most of the unit control function, including calls for heating and cooling, fan operation and scheduling, and economizer control. Additional digital I/O channels are included to provide alarm capability for fan failure, dirty filters, compressor high- or low-pressure lockout, and economizer status. Supply and return air temperature information can also be transmitted to the EMS console. These systems are very popular in chain retail and foodservice environments, allowing central control over HVAC system operation and limited unit diagnostic capability. The systems work best in buildings that are occupied on a regular schedule; applications in schools have been problematic.

Commissioning

Commissioning is a quality-assurance process that increases the likelihood that a new building will meet the intent of the design team and, ultimately, the client's expectations. In large projects, the commissioning process can encompass the entire design and construction process:

- During the design phase, commissioning begins with the selection of a commissioning agent—who helps ensure that the project documentation reflects the intentions of both the designer and owner.

Figure 12: Flow grid measures airflow

A flow grid is used to measure as-installed airflow rate. A series of flow grids (see below) are installed in place of the filters; the airflow rate through each flow grid is displayed on a digital manometer.

Source: Architectural Energy Corporation

- Next, the designer incorporates commissioning requirements into the design specifications.
- During construction, the commissioning agent is responsible for inspecting the building to identify construction defects that are difficult to correct after the building is finished.
- When the project is near completion, the commissioning agent and contractors conduct performance tests of the systems to be commissioned.
- At the end of the commissioning process, the designer and vendors train the building operators on how to properly operate and maintain the building.

Commissioning of small HVAC systems generally focuses on providing documentation of the design intent, including commissioning testing in the building plans and specifications, testing the system, correcting deficiencies, and providing operation and maintenance training to the building occupants. Incorporating the commissioning requirements into the specifications is very important, since the contractor will base the bid on the plans and specifications. Also, setting the expectation up front that commissioning will be done will save a lot of trouble during the construction process. The commissioning plan should also include a sample maintenance contract to assist the building owner or operator in obtaining ongoing maintenance services.

Perform Pre-Functional Inspections

Prior to conducting any commissioning tests, the units are inspected according to a checklist called a pre-functional checklist. Items on the checklist generally include:

- Document submittal (spec sheets, operations and maintenance instructions).
- Verification of correct make and model number.

- Installation checks, such as tight curb connections, operable cabinet door with gaskets in place, shipping materials and hold-downs removed, and adequate maintenance access.
- Duct insulation installed and in good condition.
- Filters installed properly.
- Fan motor aligned and belt tension correct.
- Economizer linkages tight, with smooth operation.
- Safety disconnect properly installed.

Perform Functional Performance Tests

The heart of the commissioning process is a series of tests called functional performance tests. For small packaged units, functional performance testing usually includes:

- Cycling unit through its various operating modes and observing unit response according to the control sequence of operations. For example, does the outdoor air damper close when the unit is turned off? Does the second compressor come on during a second stage call for cooling?
- Performing economizer tests—Does the economizer actuator work? Do the dampers move freely over their full range? Are the sensors calibrated? Does the unit respond correctly when subjected to conditions where the economizer should operate?
- Checking sensor calibration—Are the room temperature, outdoor air temperature, and/or supply air temperature sensors installed in a reasonable location and properly calibrated?
- Verifying correct rotation of supply and condenser fan motors.
- Checking for correct thermostat programming, including fan controls—Are the set points and operating schedule correct according to the design documents? Does the fan run continuously during occupied hours?

Additional functional tests may also be included. These tests can detect less obvious, but important problems with HVAC installations:

- Verify airflow through unit is correct. This generally requires the use of a flow grid to measure unit airflow.
- Verify duct leakage is within acceptable limits. This generally requires the use of a duct pressurization device to measure duct leakage rate.
- Verify correct refrigerant charge. Refrigerant pressure measurements combined with refrigerant line temperatures should be checked to verify correct superheat (for fixed throttling devices) or correct sub cooling (for thermostatic expansion valve units).
- Verify adequate outdoor airflow. A flow grid can be used to make this measurement.

Operations and Maintenance

Packaged rooftop units are generally designed for a shorter service life than other HVAC equipment. The units are also exposed to various weather elements that can be stressful to the equipment operation. Both factors can contribute to more frequent maintenance needs. Problems tend to occur during periods of system stress caused by extremely hot or cold weather. This discourages timely inspection and repair. If the problems occur during wet or icy weather, maintenance and repair can actually be hazardous.

Keeping these issues in mind will help building owners better plan maintenance of units. A little preventive maintenance during nice weather should help optimize operation, energy use, and comfort while minimizing “surprises” during inclement weather.

Provide Reasonable Access to Rooftop

Maintenance of packaged rooftop units is often ignored because the units are on the roof. Typical access is by a vertical ladder and roof hatch. Stored items can block access to the ladder, which does not encourage frequent inspections. Building owners should be sure the roof access is kept free of obstructions, and maintenance personnel have access to the key to the roof hatch padlock.

Routine Maintenance

Regular maintenance is an important component of energy efficiency, comfort, and the prevention of premature equipment failure. Simple routine checks can avoid costly contractor calls to diagnose or fix simple maintenance problems. A few routine maintenance items include:

- Check fan belts—tension/wear
- Check filters
- Verify economizer damper linkage/movement
- Check refrigerant—check site glass and test refrigerant charge
- Lubricate moving parts (including dampers and linkage)
- Check access panels for tight fit
- Inspect electrical wiring/connections
- Check coils for debris and clean as necessary

Annual maintenance contracts are common. If considering one, ensure the staff has good experience. Maintenance staff in buildings with rooftop units are often under skilled with limited training and experience. Routine maintenance tasks should be placed on easy-to-use reference sheets and lists posted in locations that encourage pro-active maintenance. Maintenance logs and manufacturer service instructions for all units should be kept in a readily accessible binder. Maintenance contracts should require a log that remains on site.

Figure 13: Maintenance Hall of Shame

The following photos were taken at a newly constructed restaurant soon after a visit by the HVAC service contractor. Note the roof was littered with old, filthy filters and bent and discarded "bird screens" intended to protect the unit's outdoor air opening (top). A closer inspection revealed several instances of missing filters and filthy cooling coils (bottom).

Source: Architectural Energy Corporation

A less obvious problem can occur when well-meaning but improper maintenance procedures are employed. A recent study conducted in California indicated that over half of the units tested were either over- or under-charged, with an average energy penalty on the order of 10 percent of the annual cooling costs.[6] Adding refrigerant until the suction line is "beer can cold" rather than following more rigorous procedures can impact comfort and energy efficiency. This particular problem is likely due to inadequate staff training, experience, or time allocated for the procedure.

Conclusion

In this design brief, a number of topics have been discussed relating to the design, installation, operation, commissioning, and maintenance of small HVAC systems. Most problems documented in the field have roots traceable to one or more of these areas. How can the industry avoid these problems in the future? Design teams and contractors should ensure rooftop HVAC systems are properly sized and the appropriate components selected and properly placed. The distribution, ventilation, thermostat, and control systems should be integrated. Also, the entire HVAC system should be commissioned to ensure it performs as designed, and regular maintenance checkups should be scheduled. By emphasizing these areas, building owners can improve the level of indoor comfort and lower operating costs associated with small HVAC systems.

For More Information

New Buildings Institute

The New Buildings Institute hosts a website that contains additional information about this project and other elements of their PIER research program. For more information, consult:

www.newbuildings.org/pier

California Energy Commission

The California Energy Commission is responsible for conducting Public Interest Energy Research on a number of topics. For more information on this and other PIER Research, consult:

www.energy.ca.gov/pier/buildings

Consortium for Energy Efficiency

The Consortium for Energy Efficiency (CEE) is a non-profit, public benefit corporation that actively promotes the use of energy-efficient products and services through its members, including electric and gas utilities, public benefit administrators (such as state energy offices, non-profit organizations, and regional energy groups), and research and development laboratories. They have established efficiency guidelines for commercial rooftop units, and have published a small commercial HVAC design guideline. For more information, consult:

www.cee1.org

Air Conditioning and Refrigeration Technology Institute

The Air Conditioning and Refrigeration Technology Institute (ARTI) conducts the Twenty-First Century Research (21-CR) initiative, which is a private-public sector research collaboration of the heating, ventilation, air-conditioning and refrigeration (HVAC/R) industry. ARTI has conducted research into design practices for small commercial HVAC systems. For more information, consult:

www.arti-21cr.org

Northwest Energy Efficiency Alliance

The Northwest Energy Efficiency Alliance (NEEA), along with Portland Energy Conservation Inc. (PECI), is conducting a pilot program to assess the market opportunities for enhanced operation and maintenance services for packaged heating and cooling systems in small commercial buildings. The pilot project is developing and testing an array of diagnostic tools and procedures, training selected contractors, developing marketing materials, and documenting the market acceptance of the service in selected markets around the Northwest. For more information, consult:

www.nwalliance.org

Air Conditioning Contractors Association

The Air Conditioning Contractors Association (ACCA) publishes several manuals on design practices for small commercial HVAC systems. For more information, consult:

www.acca.org

Air Diffusion Council

The Air Diffusion Council publishes an installation guideline for flexible duct systems. For more information, consult:

www.flexibleduct.org

Sheet Metal and Air Conditioning Contractors' National Association

The Sheet Metal and Air Conditioning Contractors' National Association (SMACNA) publishes technical manuals and construction standards relating to the construction and installation of air distribution systems. For more information, consult:

www.smacna.org

Notes

1 For more information about this project, see www.newbuildings.org/pier. Follow the links to Element 4—Integrated Design of HVAC Systems for Small Commercial Buildings.

2 See the results of the market research conducted for this project at www.newbuildings.org/pier.

3 Applications of lay-in insulation were not allowed in earlier versions of Title 24, and the practice, while not widespread, is permissible under the current (2001) Standards.

4 See State-of-the-Art Review, Whole Buildings and Building Envelope Simulation and Design Tools, Air Conditioning and Refrigeration Technology Institute (ARTI), www.arti-21cr.org.

5 Modera, M. and Proctor, J. "Combining Duct Sealing and Refrigerant Charge Testing to Reduce Peak Electricity Demand in Southern California," Final Project Report for Southern California Edison, July 2002.

6 Proctor, et al., Small commercial HVAC system inspections in Sacramento.

7 Heat pumps may require three dedicated cooling stages. The additional stage is for the reversing valve.

Pacific Gas and Electric Company

SDGE

A Sempra Energy utility

SOUTHERN CALIFORNIA EDISON

An EDISON INTERNATIONAL Company

Southern California Gas Company

A Sempra Energy utility

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