Effects of Air Infiltration on the Thermal Conductance of Internal Fiberglass Duct Insulation and on Thermal Losses From Ducts

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Abstract

Ducts that carry heated or cooled air are often internally lined with fiberglass for acoustic control and thermal insulation. If the inner face of the fiberglass liner is permeable to air, air flow in the duct may induce convection in the fiberglass and thereby increase the liner's thermal conductance (the reciprocal of its thermal resistance). In fiberglass-insulated flexible ducts with air-permeable and impermeable inner cores, the "temperature-drop" method was used to measure the variation of the thermal conductances with duct air velocity. The thermal conductance of the fiberglass blanket lining a permeable-core flexible duct increased by 36% as the duct air velocity rose from 2 to 9 m s^{-1}. The permeably-lined duct's total thermal conductance (insulation plus air films) increased with air speed at a rate approximately half that reported in a prior study. If a permeably-lined duct is located within the conditioned space of a building, the fraction of the duct air's sensible heat capacity can be saved by rendering the liner impermeable is typically 1-3%. Savings will be typically 1.5 to 2 times greater if the duct lies outside the conditioned space. Savings increase with duct length and air velocity, and decrease with duct diameter.

Introduction

Ducts that carry heated or cooled air are often internally lined with fiberglass for acoustic control and thermal insulation. If the inner face of the fiberglass is permeable to air, the air flow in the duct may infiltrate (i.e., induce convection in) the fiberglass and thereby increase its thermal conductance (reciprocal of thermal resistance). This study investigates both the effect of infiltration on the thermal conductance of fiberglass duct insulation, and the fraction of thermal energy carried by a duct that could be saved by making the inner face of a fiberglass duct liner impermeable to air.

A literature search discovered only one prior study of the effect of air infiltration on the thermal conductance of internal duct insulation (Lauvray 1978),[*] which reported that the total thermal conductance of fiberglass-insulated flexible duct (conductance of insulation plus air films) increased by 87% as the duct air speed rose from 5 to 15 m s^{-1}. The current study has two goals: first, to verify the earlier results; and second, to estimate how much thermal energy can be saved by making the interior surface of fiberglass duct insulation impermeable to air.

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[*] Results of this study are also reported in the ASHRAE Handbook of Fundamentals (ASHRAE, p. 32.15).

The ability of conditioned air to heat or cool a room is measured by its thermal capacity, equal to its flow of its room-air-referenced enthalpy. The sensible component of the thermal capacity is the product of its mass flow rate, specific heat, and difference in temperature from that of the room air. As it flows through duct system, conditioned air loses some of its sensible heat capacity through heat exchange with its surroundings across the duct wall. Heat capacity may also be lost through leaks in the duct, but only conduction losses are considered in this study.

In an airtight duct, both the sensible heat capacity lost by the conditioned air and the thermal conductance of the duct wall may be found by measuring the air flow rate and the change in air temperature between two points in the duct. This is known as the "temperature-drop" technique for measuring thermal conductance. Once the variation with air speed of the duct's thermal conductance is known, the sensible heat capacity loss can be calculated at any air speed for any length and diameter of duct. The influence of changes in duct diameter can also be estimated; however, there is some from possible differences in the effects of liner permeability for different degrees of liner curvature.

In this study, temperature-drop measurements were made to determine how the thermal conductances of permeably- and impermeably-lined ducts changed with air speed. These data were then used to estimate the sensible heat capacity lost by various size ducts due to the permeability of internal fiberglass linings.

Theory

Heat Exchange Between Duct Air and Ambient Air

<u>Components of Thermal Resistance</u>

Consider a long, circular duct of inner diameter d_i and outer diameter d_o, lined internally with fiberglass insulation (Figure 1).[*] If the ambient air temperature is T_a and the duct air temperature at axial position x is $T(x)$, the wall heat flux per unit inner-surface area A_{s_i} may be written

$$\frac{Q(x)}{A_{s_i}} = \frac{T(x) - T_a}{R} = U\left[T(x) - T_a\right].$$ (1)

Here R is the total thermal resistance between the duct air and the ambient air, and

$$U \equiv R^{-1}$$ (2)

is the total thermal conductance between the duct air and the ambient air.[†] R is the sum of the resistance of the air film inside the duct, R_i; the resistance of the air film outside the duct, R_o; and the resistance of the fiberglass insulation,[‡] R_f:

$$R = R_i + R_o + R_f.$$ (3)

The liner's resistance R_f may be obtained by subtracting estimates of the air-film resistances R_i and R_o from a measurement of the total resistance R.

The duct's total thermal conductance U can be a useful metric when examining the variation of wall flux with the duct air's mean velocity,[§] u, because a given fractional increase in thermal conductance yields the same fractional increase in wall heat flux.

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[*] The example of a circular duct is chosen for convenience, but all theory developed in this paper applies to both circular and non-circular ducts unless otherwise stated.

[†] Equation (1) defines thermal resistance and conductance, and applies to ducts of any axially-uniform cross section.

[‡] Since the thermal resistance of a standard R-4.2 hr ft^2 F Btu^{-1} (0.74 m^2 K W^{-1}) fiberglass duct liner is approximately 10^4 times that of 1-mm thick (22-gauge) steel duct wall, 10^3 that of a 0.1-mm thick plastic sheet, and 10^2 that of a 0.1 mm thick cloth sheet, the thermal resistance of the insulated wall is essentially that of its liner (White, pp. 672-675).

[§] All references to air velocity in this paper indicate the mean duct air velocity u (i.e., air flow rate divided by duct cross-sectional area) unless otherwise specified.
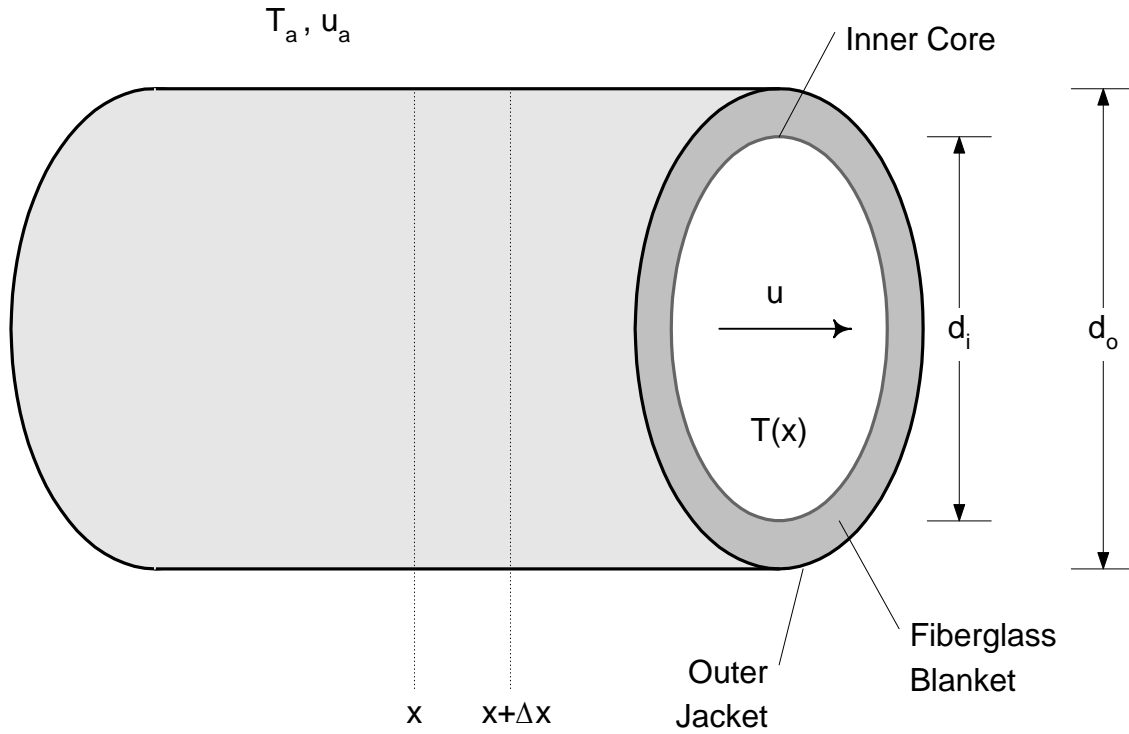
Figure 1. Axial segment of a long, circular, fiberglass-insulated duct of inner diameter d_i and outer diameter d_0. The axial dimension is x.

Resistance of the Inner Air Film

As the speed of the duct air increases, the boundary-layer thickness and the thermal resistance of the air film inside the duct will decrease. The inner-air film's forced-convection coefficient h_i may be estimated from the circular-duct Nusselt-number relation (Incropera and DeWitt 1990, p. 457)

$$Nu_{d_i} = \frac{h_i \, d_i}{k} = \frac{(f/8)(Re_{d_i} - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}. \qquad (4)$$

Here $Re_{d_i} = u \, d_i / \nu$ is the Reynolds number of the flow inside the duct; $Pr = \nu/\alpha$ is the Prandtl number of air; and k, ν, and α are the thermal conductivity, kinematic viscosity, and thermal diffusivity of air. This relation has an error of $\pm 10\%$ and is valid for $0.5 < Pr < 2000$ and $2300 < Re_{d_i} < 5 \times 10^6$. The friction factor f for air flow inside a rough-walled duct is given by (White 1988, p. 333)

$$f^{-1/2} = -1.8 \log_{10}\left[\left(\frac{\varepsilon_{\text{rough}}}{3.7 d_i}\right)^{1.11} + \frac{6.9}{Re_{d_i}}\right], \qquad (5)$$

where $\varepsilon_{\text{rough}}$ is the average wall roughness height. The roughness height of the inner surface of a lined duct is about 3 mm (ASHRAE 1997, p.32.7).

The thermal resistance of the inner air film is

$$R_i = h_i^{-1}. \tag{6}$$

Resistance of the Outer Air Film

The duct exchanges heat with the room via forced convection, free convection and radiation. The resistance of the air film outside an airtight duct does not explicitly depend on duct air speed, but its rates of free-convective and radiative heat transfer do vary with the temperature of the duct wall.

A Nusselt-number relation for forced convection over a long horizontal cylinder is (White 1988, p.344)

$$\overline{Nu}_{d_o,\text{forced}} = \frac{h_{\text{forced}}\, d_o}{k} = 0.3 + \frac{0.62\,\text{Re}_{d_o}^{1/2}\,\text{Pr}^{1/3}}{\left[1+\left(0.4/\text{Pr}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}_{d_o}}{282{,}000}\right)^{5/8}\right]^{4/5}, \tag{7}$$

where $\text{Re}_{d_o} = u_a\, d_o/\nu$ is the Reynolds number of the ambient air flow outside the duct, u_a is the ambient air velocity, and h_{forced} is the duct's coefficient of external forced convection. This relation is valid for $\text{Re}_{d_o}\,\text{Pr} \geq 0.2$ and has an uncertainty of $\pm 30\%$.

A Nusselt-number relation for free convection over a long horizontal cylinder is (White 1988, p.405)

$$\overline{\text{Nu}}_{d_o,\text{free}} = \frac{h_{\text{free}}\, d_o}{k} = \left\{0.60 + \frac{0.387\,\text{Ra}_{d_o}^{1/6}}{\left[1+\left(0.559/\text{Pr}\right)^{9/16}\right]^{8/27}}\right\}^2. \tag{8}$$

Here h_{free} is the duct's free-convection coefficient, $\text{Ra}_{d_o} = \dfrac{g\,\beta\left(T_s - T_a\right)d_o^3}{\nu\,\alpha}$ is the Rayleigh number, g is the acceleration due to gravity, $\beta = 1/T_s$ is the volume coefficient of expansion of air, T_s is the outer air film's temperature, and α is the thermal diffusivity of air. This relation is valid for $10^{-5} \leq \text{Ra}_{d_o} \leq 10^{12}$. If the flow over the duct is laminar (i.e., $\text{Re}_{d_0} < 5\times10^5$), a relation for the coefficient of mixed convection (forced plus free) is (White 1988, p.415)

$$h_{\text{mixed}} = \left(h_{\text{forced}}^3 + h_{\text{free}}^3\right)^{1/3}. \tag{9}$$

If an object of surface area A_1, emissivity ε_1, and absolute temperature T_1 exchanges radiation with only an enclosure of surface area A_2, emissivity ε_2, and absolute temperature T_2, the net radiative heat flow from object to enclosure is (White 1988, p.487)

$$Q_{\text{rad},1\to2} = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\dfrac{1-\varepsilon_1}{\varepsilon_1 A_1} + \dfrac{1}{A_1} + \dfrac{1-\varepsilon_2}{\varepsilon_2 A_2}} \, , \tag{10}$$

where $\sigma = 5.67 \times 10^{-8}$ W $^{-2}$ K^{-4} is the Stefan-Boltzmann constant. If the enclosure's area is much larger than that of the object (i.e., $A_2 \gg A_1$), the heat flow is approximately

$$Q_{\text{rad},1\to2} = A_1 \, \varepsilon_1 \sigma \left(T_1^4 - T_2^4\right). \tag{11}$$

Thus, the radiative heat flow from a straight duct to its surroundings is

$$Q_{\text{rad}} = \varepsilon \, \sigma \, A_{s_0} \left(T_w^4 - T_a^4\right) = \varepsilon \, \sigma \, A_{s_0} \left(T_w^2 + T_a^2\right)\left(T_w + T_a\right)\left(T_w - T_a\right), \tag{12}$$

where ε is the outer wall's emissivity, T_w is the outer wall's temperature, and T_a is the enclosure temperature.[*] The duct's coefficient of radiative heat exchange with its surroundings may be written

$$h_{\text{rad}} = \varepsilon \, \sigma \left(T_w^2 + T_a^2\right)\left(T_w + T_a\right), \tag{13}$$

Since radiation and convection are independent mechanisms, the outer air film's total heat transfer coefficient is

$$h_o = h_{\text{mixed}} + h_{\text{rad}} \, . \tag{14}$$

Recalling that all thermal resistances are defined with respect to heat flux per unit surface area of the duct's inner wall, the total thermal resistance of the outer air film is

$$R_o = \left(\frac{A_{s_i}}{A_{s_0}}\right) h_o^{-1} \, , \tag{15}$$

where A_{s_i} and A_{s_o} are the inner and outer surface areas of the duct, respectively.

Convection Coefficients of a Non-Circular Duct

The convection coefficients of a non-circular duct may be approximated by substituting the duct's inner and outer hydraulic diameters for d_i and d_0 in the preceding circular-duct Nusselt-number relations (Incropera and DeWitt 1990, p.461). The inner and outer hydraulic diameters are

$$d_{h_i} \equiv 4 \, A_i / P_i \tag{16}$$

and

$$d_{h_o} \equiv 4 \, A_o / P_o \, , \tag{17}$$

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[*] The enclosure temperature is assumed to equal the ambient air temperature, T_a.

where A_i and P_i are the duct's inner cross-sectional area and perimeter, and A_o and P_o are the duct's outer cross-sectional area and perimeter.[*]

Temperature-Drop Technique For Determination of Thermal Resistance

Axial Duct-Air Temperature Profile

In steady state, the enthalpy of heated air flowing into a duct segment of axial length Δx equals the sum of (a) the enthalpy flowing out of the segment and (b) the heat lost to the environment through the wall.[†] That is,

$$\rho c_p u A_i T(x) = \rho c_p u A_i T(x + \Delta x) + \frac{P_i \Delta x}{R}[T(x) - T_a], \tag{18}$$

where the product ρc_p is the volumetric heat capacity of air. As the segment's length $\Delta x \to 0$, Eq. (18) reduces to the ordinary differential equation

$$\frac{dT(x)}{dx} = -L^{-1}[T(x) - T_a], \tag{19}$$

where

$$L = \frac{\rho c_p u R d_{h_i}}{4} \tag{20}$$

is the characteristic length of the axial air temperature profile. The differential equation is solved by the exponentially-decaying temperature profile

$$\overline{T}(x) \equiv \frac{T(x) - T_a}{T_1 - T_a} = \exp\left[-\frac{(x - x_1)}{L}\right], \tag{21}$$

where $T_1 = T(x_1)$ is the air temperature at some reference position x_1. This indicates that the characteristic length L is distance over which the normalized duct-air temperature $\overline{T}(x)$ decreases by a factor of e.[‡]

Calculation of Total Thermal Resistance

Given air temperatures T_1 and T_2 measured at axial positions x_1 and x_2, Eq. (21) may be solved for the characteristic length

$$L = -\frac{\ell}{\ln\left(1 - \frac{\Delta T_{12}}{\Delta T_{1a}}\right)}. \tag{22}$$

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[*] Note that the inner and outer hydraulic diameters of a circular duct equal its actual inner and outer diameters.
[†] The example of heated air flowing through ducts is used for convenience. However, all results apply equally well to ducts carrying cooled air.
[‡] e is the base of the natural logarithms, and is approximately equal to 2.718.

Here $\ell \equiv x_2 - x_1$ is the length of the duct run, $\Delta T_{12} \equiv T_1 - T_2$ is the temperature drop across the duct run, and $\Delta T_{1a} \equiv T_1 - T_a$ is the elevation of the duct air temperature at the start of the run over that of the ambient air. The total thermal resistance between the conditioned air inside the duct and the ambient air outside the duct may then be found by solving Eqs. (20) and (22) to obtain

$$R = \frac{4\,L}{\rho\,c_p\,u\,d_{h_i}} = -\frac{4\,\ell}{\rho\,c_p\,u\,d_{h_i}\,\ln\left(1 - \dfrac{\Delta T_{12}}{\Delta T_{1a}}\right)}\,. \tag{23}$$

The liner resistance R_f can be calculated by subtracting estimated values of the inner and outer air film resistances R_i and R_o from measured values of the total resistance, R:

$$R_f = R - R_i - R_o\,. \tag{24}$$

The effective thermal conductivity[*] $k_e(T)$ of a porous matrix such as fiberglass is defined by the relation

$$Q/A = -k_e(T)\nabla T\,, \tag{25}$$

where Q/A is the heat flux per unit area normal to the matrix temperature gradient ∇T. Since the effective thermal conductivity of dry fiberglass varies with temperature,[†] a value of R_f measured when the mean liner temperature is

$$T_f \approx (T_1 + T_\infty)/2 \tag{26}$$

should be extrapolated to its value at standard room temperature $T^* \equiv 24\ {}^{\mathrm{o}}\mathrm{C}$. This separates the influence of temperature on thermal resistance from that of infiltration. The liner's temperature-corrected thermal resistance (that is, its resistance at standard room temperature) is

$$R_f^* \equiv R_f(T^*) = \frac{k_e(T_f)}{k_e(T_o)} \times R_f\,. \tag{27}$$

Correction of Liner's Nominal Thermal Resistance for Curvature

In this study, the thermal resistance of the annular duct's liner is defined with respect to the heat flow through the inner surface of the duct. Using the standard definition of the thermal resistance of an annulus (White 1988, p.55),

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[*] The conductivity is denoted "effective" because it includes heat transfer by radiation and convection in addition to conduction.

[†] The effective thermal conductivity of a flexible, fine-finer, organic-bonded fiberglass blanket of density 12 kg m^{-3} increases 4.7% per 10 ${}^{\mathrm{o}}$C rise above 24 ${}^{\mathrm{o}}$C (ASHRAE, p.24.18).

$$R_f = A_{s_i} \frac{\ln(d_o/d_i)}{2\pi \ell k_e} = \frac{d_i \ln(d_o/d_i)}{2 k_e}. \tag{28}$$

However, the nominal thermal resistance of the liner reported by the manufacturer (i.e., its "R-value") is measured with the liner in slab form. That is,

$$R_{f,\text{slab}} = \frac{d_f}{k_e}, \tag{29}$$

where $d_f \equiv \frac{1}{2}(d_0 - d_i)$ is the thickness of the liner. Thus, the ratio of the liner's annular-form resistance to its slab-form resistance is

$$\frac{R_f}{R_{f,\text{slab}}} = \left(\frac{d_i}{d_0 - d_i}\right) \ln(d_o/d_i) = \frac{d_i}{2 d_f} \ln\left(1 + \frac{2 d_f}{d_i}\right). \tag{30}$$

Measurements of the liner's annular-form resistance should be compared to its annular-form nominal resistance, rather than to its slab-form nominal resistance.

<u>Influence of Duct Air Speed on the Liner's Thermal Conductance</u>

Heat flow through the duct liner is proportional to the liner's conductance,

$$U_f = R_f^{-1}. \tag{31}$$

The liner has a non-zero conductance when the duct air is still, which may be denoted U_{f_0}:

$$U_f(u = 0) = U_{f_0}. \tag{32}$$

The effect of duct air motion on the thermal conductance of the fiberglass insulation depends on the permeability to air of the liner's inner surface. If the surface is impermeable, the liner's temperature-corrected thermal conductance should not vary with air speed.[*] If the surface is permeable, air flowing through the duct will infiltrate and induce convection within the fiberglass. This suggests that thermal conductance may be related to air speed by a function of form

$$U_f(u) = U_{f_0} + g(u), \tag{33}$$

where $g(u)$ is positive and increases monotonically.

In this study, regression is employed to fit a function of the form

$$U_f^*(u) = U_{f_0}^* + a u^n, \qquad a > 0, \quad n > 0 \tag{34}$$

to the room-temperature conductance $U_f^* = 1/R_f^*$ determined from the measurements.

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[*] It is conceivable that the performance of an impermeable but flexible inner liner could be affected by turbulence intensity due to transmission of air motion to the theoretically-still air within the insulation.

Loss of Sensible Heat Capacity From Duct Air

Measurement and Prediction of Sensible Heat Capacity Loss

Defining the duct air's sensible heat capacity with respect to room air temperature T_r as

$$C(x) \equiv \rho\, c_p\, u\, A_i\, [T(x) - T_r], \tag{35}$$

the fraction of sensible heat capacity lost between axial positions x_1 and x_2 is

$$\phi \equiv \frac{C(x_1) - C(x_2)}{C(x_1)} = \frac{T_1 - T_2}{T_1 - T_r} = \frac{\Delta T_{12}}{\Delta T_{1r}}, \tag{36}$$

where $\Delta T_{1r} \equiv T_1 - T_r$. It is convenient to express the sensible heat capacity loss ϕ as the product of two temperature ratios:

$$\phi = \gamma \times \theta, \tag{37}$$

where

$$\gamma \equiv \frac{\Delta T_{1a}}{\Delta T_{1r}} \tag{38}$$

and

$$\theta \equiv \frac{\Delta T_{12}}{\Delta T_{1a}}. \tag{39}$$

γ, the ratio of the elevation of the duct air's inlet temperature above ambient air temperature to the its elevation above room air temperature,[*] can be calculated from measurements of the duct's inlet temperature T_1, the ambient air temperature T_a, and the room air temperature T_r. It will equal one when the duct is inside a building's conditioned space, and will generally be greater than one when the duct is outside the conditioned space. If, for example, a duct carries heated air at 48 ^{o}C through an unconditioned space at 12 ^{o}C to a room at 24 ^{o}C, the ratio $\gamma = 1.5$. Or, if a duct carries cooled air at 12 ^{o}C through an unconditioned space at 36 ^{o}C to a room at 24 ^{o}C, the ratio $\gamma = 2$.

θ, the ratio of the temperature drop across the duct run to the elevation of the duct's air inlet temperature above the ambient air temperature, may be obtained by measuring the temperature elevation ΔT_{1a} and temperature drop ΔT_{12}. It may also be calculated from a known value of the duct's total thermal resistance by substituting Eqs. (21) and (20) into Eq. (36) to yield

$$\theta = \frac{\Delta T_{12}}{\Delta T_{1a}} = 1 - \exp[-\ell/L] = 1 - \exp[-4\ell/(\rho\, c_p\, u\, d_{h_i}\, R)], \qquad 0 < \theta < 1. \tag{40}$$

The sensible heat capacity loss may thus be written as

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[*] The ambient air temperature is that of the air outside the duct, while the room air temperature is that of the space to which the conditioned air is delivered.

$$\phi = \gamma \, \theta = \gamma \left\{ 1 - \exp\left[- 4\ell / \left(\rho \, c_p \, u \, d_{h_i} \, R\right)\right] \right\}. \tag{41}$$

Variation of Sensible Heat Capacity Loss With Duct Length

As the duct length ℓ increases, the residence time of the conditioned air inside the duct rises; therefore, the sensible heat capacity loss increases. This can be verified by observing that the derivative of the loss with respect to the duct length,

$$\frac{\partial \phi}{\partial \ell} = L^{-1} \gamma \left(1 - \theta\right), \tag{42}$$

is always positive.

Variation of Sensible Heat Capacity Loss With Duct Diameter

Since the sensible heat capacity of the flowing is proportional to the duct's inner cross-sectional area A_i, while the wall heat exchange through the wall is proportional to the wall's inner perimeter P_i, sensible heat capacity loss decreases as the duct's inner hydraulic diameter $d_{h_i} \equiv 4 \, A_i / P_i$ increases. This too can be verified by noting that the derivative of the loss with respect to the inner hydraulic diameter,

$$\frac{\partial \phi}{\partial d_{h_i}} = -\frac{\ell \, \gamma}{d_{h_i} \, L}\left(1 - \theta\right), \tag{43}$$

is always negative.

Variation of Sensible Heat Capacity Loss With Duct Air Velocity

The variation with air velocity u of the sensible heat capacity loss depends on that of the duct's total thermal resistance with air speed. As the velocity increases, the residence time of the conditioned air inside the duct decreases, lowering the sensible loss. However, the total thermal resistance $R(u)$ will also decrease,[*] increasing the sensible loss. The net effect may be found from the derivative of the loss with respect to duct air velocity,

$$\frac{\partial \phi}{\partial u} = -\left[\frac{\ell \, \gamma}{L}\right]\left[\frac{R + u\left(dR/du\right)}{u \, R}\right]\left[1 - \theta\right]. \tag{44}$$

This indicates that $\partial \phi / \partial u < 0$—that is, the influence of decreased residence time on sensible heat capacity loss will outweigh the influence of decreased thermal resistance—when $dR/du > - R/u$.

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[*] Even if the liner's resistance does not vary with air speed, the resistance of the inner air film will decrease as the duct air speed rises.

Variation of Sensible Heat Capacity Loss With Thermal Resistance

If the total thermal resistance R at a given air velocity is increased—say, by reducing the permeability to air of the duct liner—the sensible heat capacity loss will decrease because the heat exchange through the duct's wall is reduced. This can be verified by observing that the derivative of the sensible loss with respect to thermal resistance,

$$\frac{\partial \phi}{\partial R} = -\frac{\ell \gamma}{R L}(1-\theta),\tag{45}$$

is always negative.

Conservation of Sensible Heat Capacity

Sensible Heat Capacity Savings Obtained By Increasing Thermal Resistance

If the variation of total thermal resistance with duct air velocity is increased from $R = R(u)$ to $R' = R'(u)$, Eq. (40) predicts that the fraction of sensible heat capacity lost over a duct run of length ℓ will decrease by

$$\Delta\phi \equiv -(\phi' - \phi) = \gamma\,(\theta - \theta') = \gamma\,\left\{\exp\left[-4\ell/\left(\rho\,c_p\,u\,d_{h_i}\,R'\right)\right] - \exp\left[-4\ell/\left(\rho\,c_p\,u\,d_{h_i}\,R\right)\right]\right\}.\tag{46}$$

The fraction of sensible heat capacity that can be saved by this resistance change equals this decrease in sensible heat capacity loss. If the variation of a permeable liner's thermal resistance with air speed is known, Eq. (46) may be used to predict the savings in sensible heat capacity that can be achieved by making the liner impermeable, and thereby increasing its thermal resistance at a given air speed. That is, $\Delta\phi$ may be calculated with $R(u)$ equal to the total thermal resistance of a permeably-lined duct, and $R'(u) > R(u)$ equal to the total thermal resistance of the same duct with an impermeable liner.

The savings obtained for a duct located outside a building's conditioned space will be a factor of γ higher than that obtained for the same duct located within the building's conditioned space.

Note that savings are measured as a fraction of the sensible heat capacity entering the duct, not as a fraction of the sensible heat capacity lost.

Variation of Sensible Heat Capacity Savings With Duct Length

The sensible heat capacity savings $\Delta\phi$ increases with duct length until the duct's length ℓ exceeds the characteristic length of the flow, L. This may be seen by computing the derivative of the sensible heat capacity savings with respect to duct length, $\partial(\Delta\phi)/\partial\ell$.

The sensible heat capacity savings induced by an increase δR in thermal resistance may be approximated by

$$\Delta\phi \approx -\frac{\partial\phi}{\partial R} \times \delta R \, ,\tag{47}$$

where $\partial\phi/\partial R$ is given by Eq.(45). The derivative of the sensible heat capacity savings with respect to duct length is then

$$\frac{\partial\left(\Delta\phi\right)}{\partial\ell} = \frac{\partial}{\partial\ell}\left(-\frac{\partial\phi}{\partial R}\times\delta R\right) = \left(\frac{\delta R}{R}\right)\left(\frac{L-\ell}{L}\right)\gamma\left(1-\theta\right).\tag{48}$$

This is positive when $\ell < L$. Thus, the sensible heat capacity savings increases with duct length when the duct is shorter than the characteristic length of the flow, and decreases with length when the duct is longer than the characteristic length of the flow.

Variation of Sensible Heat Capacity Savings With Duct Diameter

The sensible heat capacity savings $\Delta\phi$ decreases with increasing inner hydraulic diameter d_{h_i} when the duct's length ℓ is shorter than the characteristic length of the flow, L, and increases with d_{h_i} when $\ell > L$. This may be seen by computing the derivative of the sensible heat capacity savings with respect to inner hydraulic diameter, $\partial\left(\Delta\phi\right)/\partial d_{h_i}$. Repeating the above procedure yields

$$\frac{\partial\left(\Delta\phi\right)}{\partial d_{h_i}} = \frac{\partial}{\partial d_{h_i}}\left(-\frac{\partial\phi}{\partial R}\times\delta R\right) = \left(\frac{\delta R}{R}\right)\left(\frac{\ell-L}{d_{h_i}L^2}\right)\gamma\left(1-\theta\right).\tag{49}$$

This is negative when $\ell < L$. Thus, the sensible heat capacity savings decreases with duct diameter.

Variation of Sensible Heat Capacity Savings With Duct Air Velocity

The derivative of the sensible heat capacity savings with respect to duct velocity is

$$\frac{\partial\left(\Delta\phi\right)}{\partial u} = \frac{\partial}{\partial u}\left(-\frac{\partial\phi}{\partial R}\times\delta R\right) = \left(\frac{\delta R}{R}\right)\left(\frac{R+u\left(dR/du\right)}{uR} - \frac{\ell}{uL}\right)\gamma\left(1-\theta\right).\tag{50}$$

This is positive when

$$1+\frac{\left(dR/du\right)}{\left(R/u\right)} > \frac{\ell}{L}.\tag{51}$$

Calculation Uncertainties

Uncertainties in the measurements of the temperature drop ΔT_{12} and the air velocity u can yield significant uncertainties in computed values of the thermal resistance, thermal conductance, and sensible heat capacity loss. If the uncertainties in ΔT_{12} and u are $\delta_{\Delta T_{12}}$ and δ_u, respectively, a calculated value of the total thermal resistance has uncertainty

$$\delta_R = \sqrt{\left(\frac{\partial R}{\partial \Delta T_{12}}\right)^2 \times \delta^2_{\Delta T_{12}} + \left(\frac{\partial R}{\partial u}\right)^2 \times \delta^2_u}$$

$$= \sqrt{\left(\frac{4\ell}{\rho\, c_p\, u\, d_{h_i}\left(\Delta T_{1a} - \Delta T_{12}\right)\times \ln^2\left(1 - \dfrac{\Delta T_{12}}{\Delta T_{1a}}\right)}\right)^2 \times \delta^2_{\Delta T_{12}} + \left(-\frac{R}{u}\right)^2 \times \delta^2_u} \quad . \tag{52}$$

If the uncertainties in the inner and outer air film resistances R_i and R_o are δ_{R_i} and δ_{R_o}, respectively, [*] the uncertainty in the liner's resistance R_f is

$$\delta_{R_f} = \delta_R + \delta_{R_i} + \delta_{R_o}. \tag{53}$$

The uncertainty in the liner's thermal conductance $U_f = 1/R_f$ is

$$\delta_{U_f} = \sqrt{\left(\frac{\partial U_f}{\partial R_f}\right)^2 \times \delta^2_{R_f}} = \left|\frac{\partial U_f}{\partial R_f}\right|\delta_{R_f} = \delta_{R_f}/R_f^2 \ . \tag{54}$$

The uncertainty in the fraction of sensible heat capacity lost ϕ is

$$\delta_\phi = \sqrt{\left(\frac{\partial \phi}{\partial \Delta T_{12}}\right)^2 \times \delta^2_{\Delta T_{12}}} = \left|\frac{\gamma}{\Delta T_{1a}}\right|\delta_{\Delta T_{12}}. \tag{55}$$

Uncertainty in the temperature elevation T_{1a} contributes negligibly to the uncertainties in R and ϕ.

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[*] It is assumed in this study that the fractional uncertainties in the air-film resistances are $\pm 20\%$.

Experiment

Overview

The "temperature-drop" technique for determining the thermal resistance of a duct wall works best when the duct run is long enough to yield an accurately-measured temperature drop. Fiberglass-insulated flexible duct, or "flexduct," was chosen for this experiment as the most convenient source of long ductwork made with both permeably and impermeably-faced linings. Insulated flexduct consists of a spring-wire helix frame encapsulated in an thin inner core of either plastic or fabric.[*] The inner core is surrounded by a fiberglass blanket, which is in turn encased in a plastic jacket (Figure 2). The combination of inner core and fiberglass blanket will be referred to as the fiberglass blanket's liner.

The type of internal duct insulation commonly found in rigid ducts (e.g., rectangular, sheet-metal trunk ducts) is not a fiberglass blanket, but rather a fiberglass mat. Fiberglass mat liners are denser and less permeable to air than are fiberglass blanket liners. Since the sensitivity of a liner's thermal resistance to duct air speed is expected to depend on the liner's permeability to air, the permeabilities of mat and blanket liners were measured. Their permeabilities were compared to estimate the applicability to mat liners of the thermal-resistance results obtained for blanket liners.

The influences of duct air speed on thermal resistance and sensible heat capacity loss were measured for insulated flexducts with two types of inner core. The first core, made of spun-bonded, non-woven fabric, was permeable to air; the second, made of plastic, was impermeable. The thermal resistance of the fabric-core liner was expected to vary significantly with duct air speed, while that of the plastic-core liner was not. Sensible capacity losses were measured by injecting hot air into each type of duct, then recording the steady-state air temperature drops along each duct. Thermal resistances were then calculated from the capacity losses. The duct air speed was varied to find the dependence of capacity loss and resistance on air velocity.

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[*] The fabric inner core is intended to provide acoustic control superior to that yielded by a plastic core.
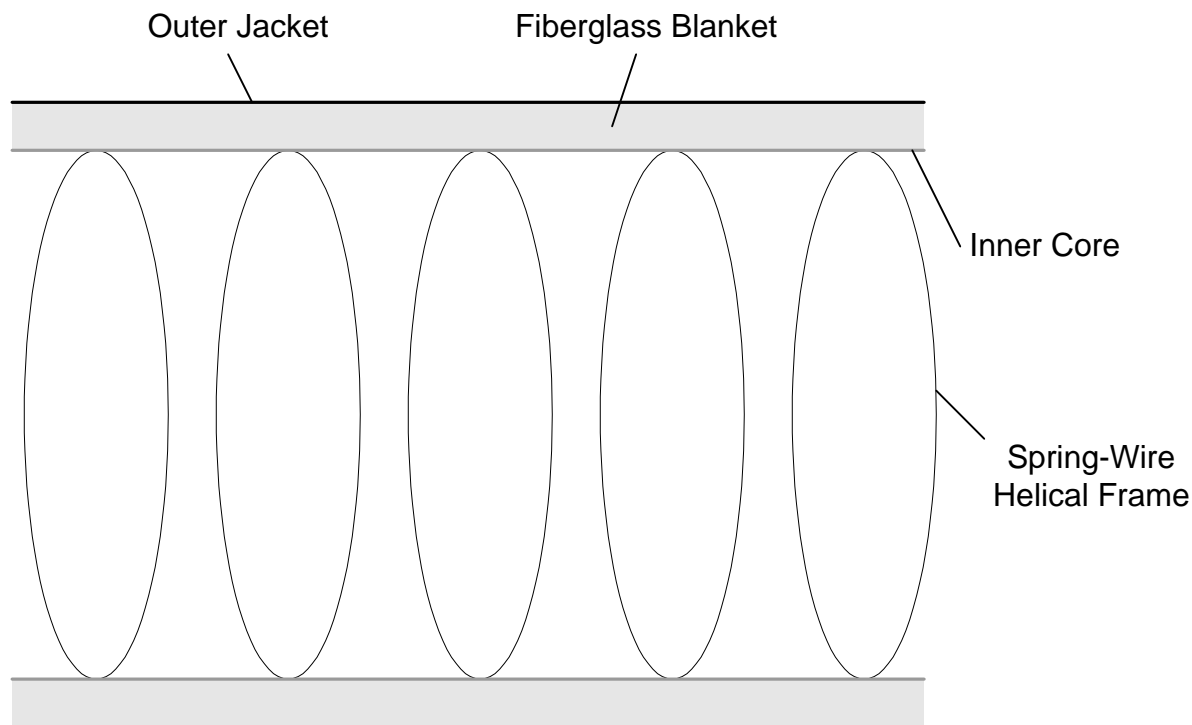
Figure 2. Side view of an insulated flexible duct. A spring-wire helical frame is encapsulated in air-carrying core. The core is surrounded by a blanket of fiberglass, which in turn is enclosed by an outer jacket.

Permeability Measurements

The permeability of a porous medium to a fluid is the ratio of the mean velocity of flow through the medium (i.e., flow per unit cross-sectional area) to the pressure drop across the medium. The permeabilities of mat and blanket liners were determined by measuring the pressure drop across each liner at various rates of air flow through the liner.

A fiberglass-mat liner for rigid ducts (Owens Corning Aeroflex PLUS Type 150)[*] was tested by clamping the mat to the outlet of a 25-cm diameter duct (Figure 3), injecting air at the duct's inlet, and then measuring the pressure in the duct versus the rate of air flow through the duct.

The permeability of the combined core and blanket of the permeable-core flexduct (JP Lamborn AMF-50)[†] was measured by sealing the outlet of the duct, injecting air at the duct's inlet, exhausting air through an exposed patch of blanket, and then measuring the pressure in the duct versus the rate of air flow through the duct. The exposed patch was created by making a 10-cm x 10-cm incision through the flexduct's outer jacket and fiberglass blanket, then removing the jacket square. An 10-cm x 10-cm plastic frame with sides 3 cm high and 3 mm wide was placed in the blanket's incision to impede air flow through the sides of the fiberglass patch. Masking tape connected the edges of the patch to the frame and to the jacket outside the frame, pressing

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[*] Thickness 2.5 cm; density 24 kg m^{-3}; thermal resistance 3.6 hr ft^2 F Btu^{-1} (0.63 m^2 K W^{-1}).
[†] Thickness 2.9 cm; density 13 kg m^{-3}; thermal resistance 4.2 hr ft^2 F Btu^{-1} (0.74 m^2 K W^{-1}).

the frame into the flexduct's inner core and reducing the area of exposed fiberglass to 8 cm x 8 cm (Figure 4).

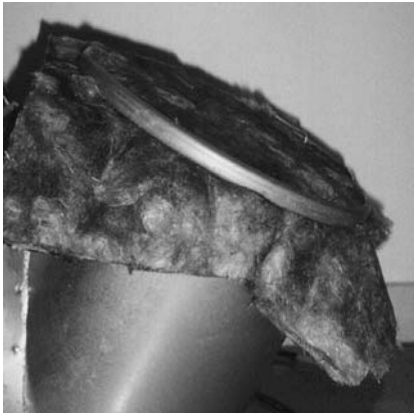

Figure 3. Fiberglass mat for rigid ducts, clamped to a duct's outlet for permeability testing.

(a)   **(b)**

Figure 4. Exposed patch of flexible duct liner's fiberglass blanket (a) surrounded on four sides with a plastic frame to prevent lateral air flow; and (b) secured to the outer jacket with masking tape, pressing the frame onto the duct's inner core.

Temperature-Drop Measurements

A variable-speed axial fan (Minneapolis Energy Conservatory Duct Blaster Series B, denoted "Fan 1") was connected in series to a constant-speed centrifugal fan (Fasco Model 1420, denoted

"Fan 2") to blow air into an electric resistance heater (Figure 5). The heater was equipped with four switchable 1.5 kW elements. The air flow rate through the system was measured with a flow meter (accuracy $\pm$ 5%) incorporated in the variable-speed fan.

A 13-m length of insulated flexduct was connected to the heater's outlet and set on a carpeted laboratory floor (Figure 6). The duct was made of a 15-cm (6") inner-diameter, spring-wire helix frame encapsulated in either an air-permeable, fabric core (JP Lamborn AMF-50) or an impermeable plastic core (JP Lamborn MF-50) (Figure 8). A 2.9-cm-thick fiberglass blanket with a nominal R-value of 4.2 hr ft^2 F Btu^{-1} surrounded the inner core, and was in turn enveloped by an airtight, metalized plastic jacket. The average long-wave radiative emissivity of the jacket's outer surface was 0.47.[*]

Resistance temperature detector (RTD) probes (Instrulab Model 4212; accuracy $\pm$ 0.03 °C) measured air temperatures at the center of the duct at two points spaced 6.1 m apart. The temperature difference was observed across a straight run of duct. However, the temperature sensors were radiatively isolated from the heating elements by a pair of 25-degree bends formed in the duct between the heater and the first temperature sensor (Figure 7). The room air temperature was measured with a thermistor (Alnor Compuflow Thermo-Anemometer Model 8500D-II, accuracy $\pm$ 0.3 °C).

The velocity of air flow over the duct induced by room ventilation was below the 0.4 m s^{-1} threshold of a vane anemometer (Sper Scientific Vane Anemometer 840032), but non-zero. It was estimated to be 0.1 m s^{-1} by measuring the time required for a smoke plume to travel a horizontal distance of 1 m. The room air temperature was maintained between 18 and 26 °C.

The fan and heater powers were adjusted to deliver duct air at speeds ranging from 2 to 9 m s^{-1} and temperatures ranging from 42 and 55 °C.[†] Each combination of heating rate and air speed was maintained for 30 minutes to achieve a steady-state air temperature profile in the duct. When the air-temperature difference measured by the RTD probes fluctuated by no more than 0.1 °C, the temperature difference, temperature at the first RTD sensor, room air temperature, and orifice meter pressure (indicating air flow rate) were recorded.

The permeable and impermeable flexducts were tested in separate trials.

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[*] The emissivity of the jacket's outer surface was measured with Lawrence Berkeley National Laboratory's FTIR Spectral Emissometer, which is based on a Bruker IFS28 FTIR spectrometer.
[†] Air speeds below 2 m s^{-1} forced a thermostatic shutoff of the heating elements, while air speeds above 9 m s^{-1} could not be achieved with the available fan power. (The long ducts employed to obtain an easily-measurable temperature drops require a great deal of fan power to achieve high duct air speeds.)
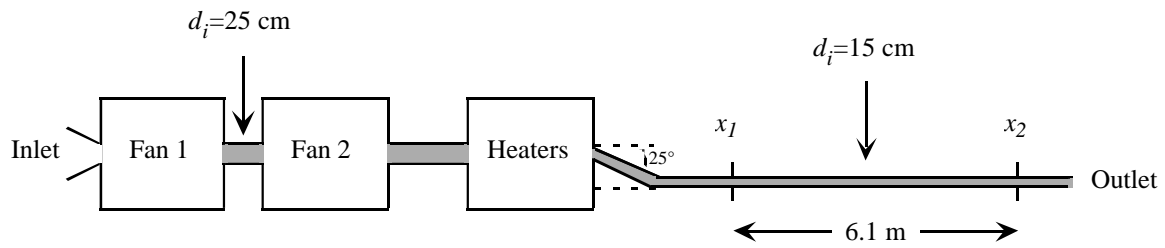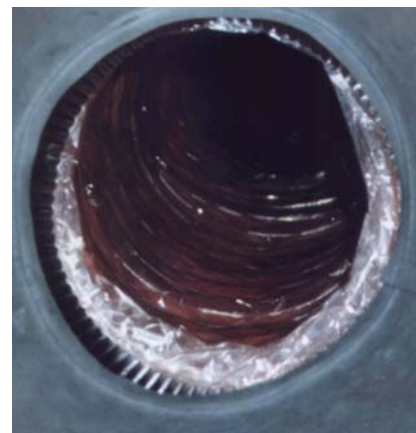
Figure 5. Schematic of temperature-drop appartus.

Figure 6. Pair of temperature probes measuring air temperature drop across a 6.1-m span of insulated flexible duct.

Figure 7. Bends in the duct radiatively isolating the duct air temperature sensors from the heating elements.

(a)

(b)

Figure 8. Insulated flexible duct cores made of (a) air-permeable, spun-bonded, non-woven fabric [JP Lamborn AMF-50]; and (b) impermeable, 2-ply plastic [JP Lamborn MF-50].

Calculations

Overview

The permeability of the permeable-core flexduct's liner[*] was compared to that of the rigid-duct fiberglass mat by measuring air flow per unit area versus pressure drop for each type of liner. The sensible heat capacity loss, thermal resistance, and thermal conductance of the permeable-core and impermeable-core flexducts were calculated from measurements of temperature drop and air velocity. A correlation was fit to the permeable-core flexduct's thermal conductance versus air speed. Finally, this conductance correlation was used to estimate the fraction of sensible heat capacity that could be saved in permeably-lined ducts of various diameters and lengths by making their linings impermeable to air.

Liner Permeability

Air flows per unit area were computed for the mat liner and for the blanket liner. The slope of a zero-intercept line fit to the air flow per unit area versus pressure drop yielded each liner's permeability (Figure 9). Note that the technique employed to measure the air flow through the mat liner differed from that used to measure the permeability of the blanket liner.[†]

Sensible Heat Capacity Loss Versus Duct Air Speed

The sensible heat capacity losses in the 6.1-m runs of permeable-core and impermeable-core flexduct were computed from temperature drops using Eq. (36), then plotted versus air speed (Figure 10). Since the duct was located in the conditioned space, the ratio $\gamma \equiv (\Delta T_{1a})/(\Delta T_{1r})$ was one. The uncertainty in duct air speed was taken to be $\pm 5\%$, while the uncertainty in sensible heat capacity loss was computed with Eq. (49).

Thermal Resistance and Conductance Versus Duct Air Speed

The total thermal conductances of the permeable-core and impermeable-core flexducts were computed from Eqs. (23) and (41), then plotted versus air speed for comparison with the conductance curve reported by the prior study (Figure 11).

The thermal resistances of the fiberglass liners in the flexible ducts were calculated from Eqs. (23) and (24), extrapolated to standard room temperature (24 $^{\circ}$C) using Eq. (27), then plotted against air speed (Figure 12). The liners' temperature-corrected thermal conductances were then calculated from Eq. (31) and plotted versus air speed (Figure 13). Uncertainty in the liners' thermal resistances were computed with Eqs. (52) and (53), while uncertainty in the liners' thermal conductances were calculated with Eq. (54).

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[*] The flexduct's liner comprises its inner core and the fiberglass blanket wrapped around the core.
[†] It was not possible to clamp a section of the flexduct's core-and-blanket over the end of a duct because the spring-wire helix embedded in the core resisted straightening.

The distribution of thermal resistance in the permeable core flexduct—liner, inner air film, and outer air film—was plotted versus air speed (Figure 14).

Savings in Sensible Heat Capacity Obtained By Rendering Permeable Duct Liners Impermeable

The fraction of sensible heat capacity that could be saved by replacing a permeable duct liner with an impermeable duct liner (or by rendering such a liner impermeable with an internal sealant coating) was estimated over a range of air speeds for a long duct with various cross-sectional dimensions. The calculation was repeated at two air speeds for ducts of variable length and cross-sectional dimensions. The choice of thermal resistance relations for the permeable and impermeable liners was complicated by the fact that long trunk ducts are typically lined with fiberglass mats, rather than fiberglass blankets. The current study found that a fiberglass mat was roughly half as air-permeable as a fiberglass blanket, suggesting that its thermal resistance, and therefore its sensible heat capacity loss, could be significantly less sensitive to air speed.

Since there does not appear to be any available information on the air-speed dependence of a mat's thermal resistance, blanket-liner data from the current study was used instead. Estimates of thermal savings computed using this data should therefore be treated as upper bounds when applied to mat-lined rigid ducts.

Equation (40) was used to calculate the savings achievable at air speeds ranging from 2 to 15 m s^{-1} (Figure 15). The ducts simulated were 20-m long, with outer cross-sectional dimensions typical of round flexible ducts, rectangular rigid ducts, and vertical riser ducts.[*] The thermal resistance of the permeable liner $R_f^*(u)$ was calculated from the conductance correlation fit to the current study's permeable-core flexduct data (Figure 13). In these simulations, the thermal resistance of the impermeable liner $R_f'^*(u)$ was assigned a value equal to the resistance of the permeable-liner at 2 m s^{-1}; this ensured that the resistance of the permeable liner was never greater than that of the impermeable liner.

The simulated liners were 2.9 cm thickness, and had an outer-wall emissivity of 0.5. The duct air's inlet temperature was assumed to be 48 $^{\circ}$C, while the ambient air temperature and velocity and temperature were assumed to be 24 $^{\circ}$C and 0.1 m s^{-1}.

Savings were calculated for a duct located within a building's conditioned space ($\gamma = 1$). Savings for ducts than run outside the conditioned space will be a factor of γ higher.

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[*] The free-convection relation in the model was not modified to account for vertical orientation.

Results

Liner Permeability

The permeability to air of the permeable core flexduct's liner was found to be approximately twice that of fiberglass-mat duct liner. As mentioned above, this indicates that the variations of thermal resistance with air speed measured in this study, and in the prior study, overstate the effect to be found in rigid ducts insulated with fiberglass mats.

Sensible Heat Capacity Loss Versus Duct Air Speed

As expected, the fractions of sensible heat capacity lost in the runs of permeable and impermeable core flexduct fell with rising air speed because the conditioned air's residence time in the duct run decreased as the air speed increased. The fraction of capacity lost by air in the permeable-core flexduct decreased slower with rising air speed that did that of air in the impermeable-core flexduct, indicating that the permeable-core duct's thermal resistance decreased faster with increasing air speed than did that of the impermeable-core flexduct (Figure 10).

Thermal Resistance and Conductance Versus Duct Air Speed

Measured Versus Nominal Liner Resistances

Both the permeable and impermeable liners had nominal slab-form thermal resistances of 4.2 hr ft^2 F Btu^{-1} (0.76 m^2 K W^{-1}). In annular configurations,[*] their nominal thermal resistances were 3.6 hr ft^2 F Btu^{-1} (0.63 m^2 K W^{-1}). With the exception of one outlying data point ($R_f^* = 4.7$ hr ft^2 F Btu^{-1} at $u = 7.3 \, \text{m s}^{-1}$, discussed below), the temperature-corrected thermal resistance of the impermeable liner was 3.0-3.3 hr ft^2 F Btu^{-1} (0.53-0.58 m^2 K W^{-1}), within 20% of its annular-form nominal value. The temperature-corrected thermal resistance of the permeable liner at the minimum duct air speed[†] ($u = 2 \, \text{m s}^{-1}$) was 3.0 hr ft^2 F Btu^{-1} (0.53 m^2 K W^{-1}), also within 20% of its annular-form nominal value.

Distribution of Thermal Resistance

The fraction of the total thermal resistance of the permeable-core flexduct contributed by its fiberglass liner was approximately 65%. The outer air film contributed 25%, and the inner air film contributed the remaining 10% (Figure 14). This indicates that variations with air velocity of the liner's thermal resistance can significantly influence the duct's total thermal resistance and, in consequence, its sensible heat capacity loss.

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[*] The liner's annular-form resistance is that predicted by Eq. (30) when the 2.9 cm (1.1") thick liner is wrapped around the flexible duct's 15-cm (6") inner core. It is 15% lower than the liner's slab-form nominal resistance.
[†] The 1500-W resistance-heating unit overheated and thermostatically deactivated at duct air velocities below 2 m s^{-1}.

Conductance of Permeable Liner

A linear relation fit to the temperature-corrected thermal conductance U_f^* of the permeable-core liner,

$$U_f^*(u) = U_{f_0}^* + a\,u, \qquad U_{f_0}^* = 0.38, \quad a = 0.023, \tag{56}$$

increased by 38% as the air speed rose from 2 to 9 m s^{-1} (Figure 13). The linear fit has a correlation coefficient of only 0.39. Higher-order power-law fits yield somewhat-higher correlation coefficients ($R^2 \approx 0.5$) but, given the extent of the data scatter, do not necessarily offer meaningful physical insight into the variation of the permeable liner's conductance with air speed. Therefore, the linear relation was retained.

Conductance of Impermeable Liner

The thermal conductance of the impermeable liner was generally flat over the air-speed range of 2 to 6 m s^{-1}, but declined markedly when measured at 7.3 m s^{-1}. This temperature-drop recorded in this exceptional trial was appreciably smaller than others observed in the experiment (0.6 ^{o}C, versus 1.2-2.5 ^{o}C), and thus may be in error.

Total Conductance of Permeable-Core Flexible Duct

The total thermal conductance of the permeable-core insulated flexduct tested in this study varied less with air speed than did that of a permeable-core insulated in a prior study (Lauvray 1978). Over the range of duct air speeds (approximately 5 to 9 m s^{-1}) in which both studies found the total conductance of permeably-lined flexducts to vary with duct air velocity, total conductance increased by 20% (5% per m s^{-1}) in the current study, and by 35% (9% per m s^{-1}) in the prior study (Figure 11). Two factors make it difficult to draw strong conclusions from this discrepancy. First, the choice of correlation for the current data is highly arbitrary, given both the appreciable scatter in the data and the tall error bars on the data points. Second, the prior study presented a linear fit for the thermal conductance without showing data points or a correlation coefficient.

Savings in Sensible Heat Capacity Obtained By Making Permeable Duct Liners Impermeable

Overview

Simulations over wide range of duct sizes, lengths, and air velocities indicated that when a duct is located inside the conditioned space of a building, the sensible heat capacity savings obtainable by replacing a permeable duct liner with an impermeable duct liner ranged from 0 to 6.1%; typical values were 1 to 3%. Savings were greatest for small duct diameters, long duct lengths, and high air speeds. Savings will increase by a factor of γ (typically 1.5-2) if the duct is located outside the building's conditioned space.

N.B.: since the permeable-to-impermeable change in the resistance of the fiberglass liner was based on measurements made on flexible-duct fiberglass-blanket insulation, rather than the rigid-

duct fiberglass-mats insulations, the results in this section may be reasonably applied to flexible ducts, but should be treated as upper bounds for rigid ducts.

Variation of Sensible Heat Capacity Savings With Duct Cross-Section and Air Velocity

Savings for a fixed-length (20-m) duct in a building's conditioned space increased with duct air speed and decreased with duct diameter, to a maximum of 1.8% (Figure 15). For a small flexible duct (10" [25 cm] diameter), savings rose from 1.1% to 1.8% over the velocity range 5-15 m s^{-1}. For a rigid duct (12" x 48" [30 cm x 122 cm]), savings rose from 0.5% to 0.7%, while for a vertical riser (36" x 36" [91 cm x 91 cm]), savings increased from 0.2% to 0.3%.

Variation of Sensible Heat Capacity Savings With Duct Cross-Section and Length

Savings for a 0-100 m long duct in a building's conditioned space carrying air at 5 m s^{-1} increased with duct length and decreased with duct size, to a maximum of 3.4% (Figure 16). The maximum savings at a duct air speed of 10 m s^{-1} was 6.1% (Figure 17).

For a small flexible duct (10" [25 cm] diameter), savings at an air speed of 5 m s^{-1} rose from 0.6% to 3.4% as the length increased from 10 to 100 m.[*] For a rigid duct (12" x 48" [30 cm x 122 cm]), savings rose from 0.3% to 2.0%, while for a vertical riser (36" x 36" [91 cm x 91 cm]), savings increased from 0.1% to 1.0%. At an air speed of 10 m s^{-1}, the savings increased from 0.9% to 6.1%, 0.4% to 3.0%, and 0.2% to 1.5% for the flexible duct, rigid duct, and riser, respectively.

Conclusions

♦ The thermal conductance of permeably-faced fiberglass duct insulation was measurably increased by the flow of air through the duct. However, the rate of increase with air speed of the permeable-liner conductance was found to be only half that reported in an earlier paper.

♦ Modest savings (typically 1-3%) in sensible heat capacity may be obtained by replacing a permeable duct liner with an impermeable duct liner when the duct runs through the building's conditioned space. Savings will be typically 1.5 to 2 times greater if the duct lies outside the conditioned space. Sensible heat capacity savings increase with air speed and duct length, and decrease with duct diameter.

References

American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc (ASHRAE). 1997. *Handbook of Fundamentals, SI Edition*. Atlanta, GA.

Incropera, F.P., and DeWitt, D.P. 1990. *Introduction to Heat Transfer*, second edition, John Wiley & Sons, New York.

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[*] A typical length for a flexible duct would be far closer to 10 m than to 100 m.

Lauvray, T.L. 1978. "Experimental heat transmission coefficients for operating air duct systems," ASHRAE Journal, 69-73 (June).

White, F.M. 1988. *Heat and Mass Transfer*. Addison-Wesley, New York.

Acknowledgements

Permeability to Air of Duct Liners

J.P. Lamborn AMF-07 Flexible Duct Liner w/Acoustic Core
[thickness 2.9 cm; density 13 kg m^{-3}; R-value 4.2 hr ft^2 F Btu^{-1}]

$v = 1.3 \, \Delta P \quad [R^2 = 0.99]$

Owens Corning Aeroflex Plus Type 150 Duct Liner
[thickness 2.5 cm; density 24 kg m^{-3}; R-value 3.6 hr ft^2 F Btu^{-1}]

$v = 0.71 \, \Delta P \quad [R^2 = 0.99]$

Flow Per Unit Area v [cm s^{-1}]

Pressure Drop Across Liner ΔP [Pa]
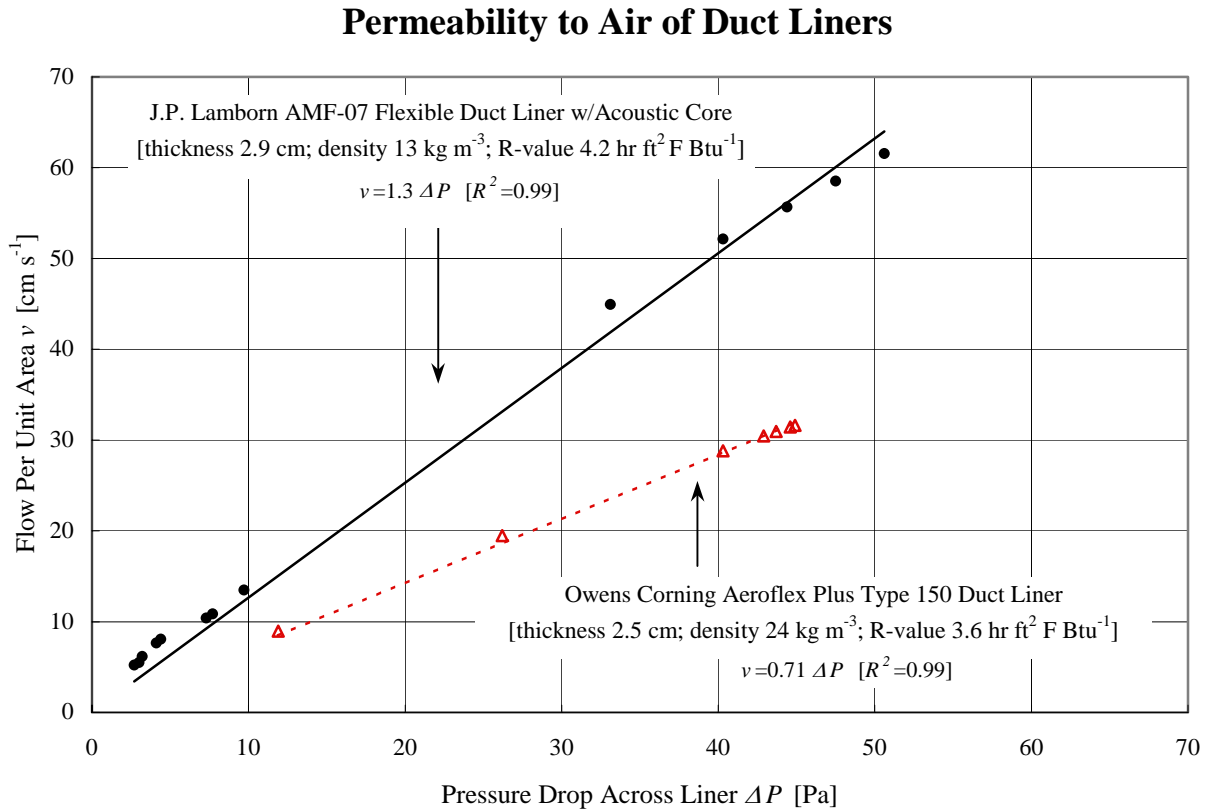
Figure 9. Permeability to air of a fiberglass-blanket duct liner (JP Lamborn AMF-07) and a fiberglass-mat duct liner (Owens Corning Aeroflex Plus Type 150).
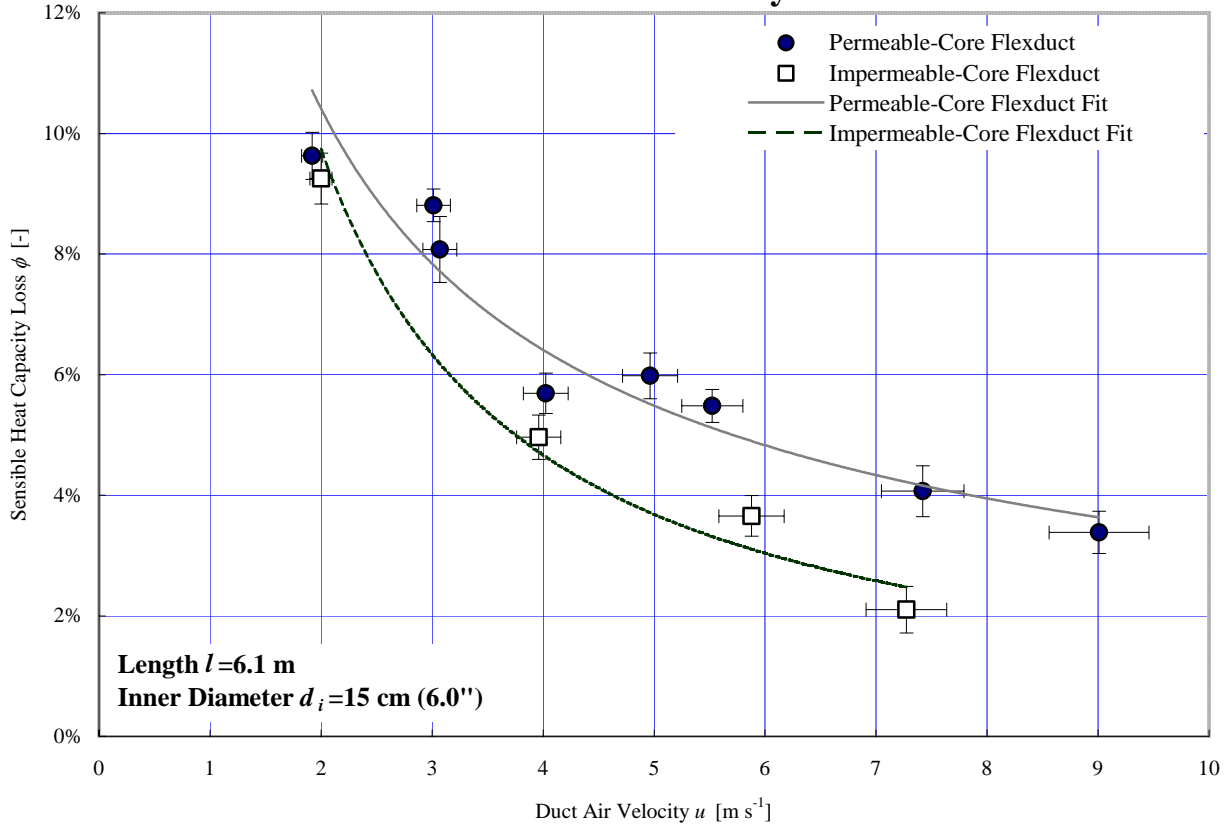
Figure 10. Fraction of sensible heat capacity lost versus duct air velocity in permeable-core and impermeable-core insulated flexible ducts. The losses were measured across 6.1-m lengths of 15-cm (6") inner-diameter ducts with nominal thermal resistances of 4.2 hr ft^2 F Btu^{-1} (0.76 m^2 K W^{-1}). The ducts were located within the conditioned space.

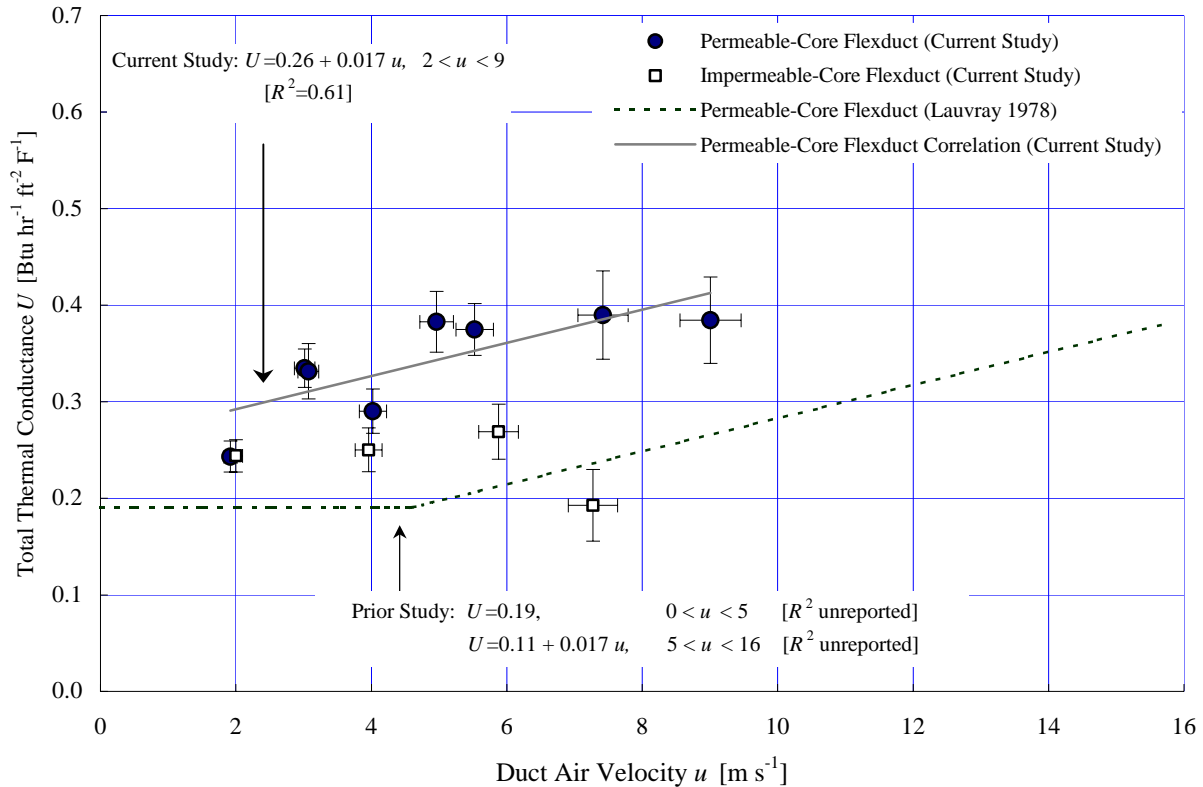## Total Thermal Conductance of Fiberglass-Insulated Flexible Duct (Liner + Air Films) Vs. Duct Air Velocity

Figure 11. Total thermal conductance versus duct air velocity of the insulated permeable-core flexible duct [JP Lamborn AMF-50, inner diameter 15 cm (6"), insulation thickness 3.2 cm] and impermeable-core flexible duct [JP Lamborn MF-50, same dimensions]. Duct air temperatures were 42-55 $^{\circ}$C , while ambient air temperatures were 18-26 $^{\circ}$C. The ambient air velocity was approximately 0.1 m s^{-1}. Shown for comparison is the total thermal conductance versus duct air velocity of an insulated permeable-core flexible duct [dimensions and test conditions unknown] reported in a prior study (Lauvray 1978). Thermal conductances include air films on the inner and outer duct surfaces and are based on each duct's inner surface area.

Thermal Resistance of Fiberglass Duct Liner at 24 ^{o}C
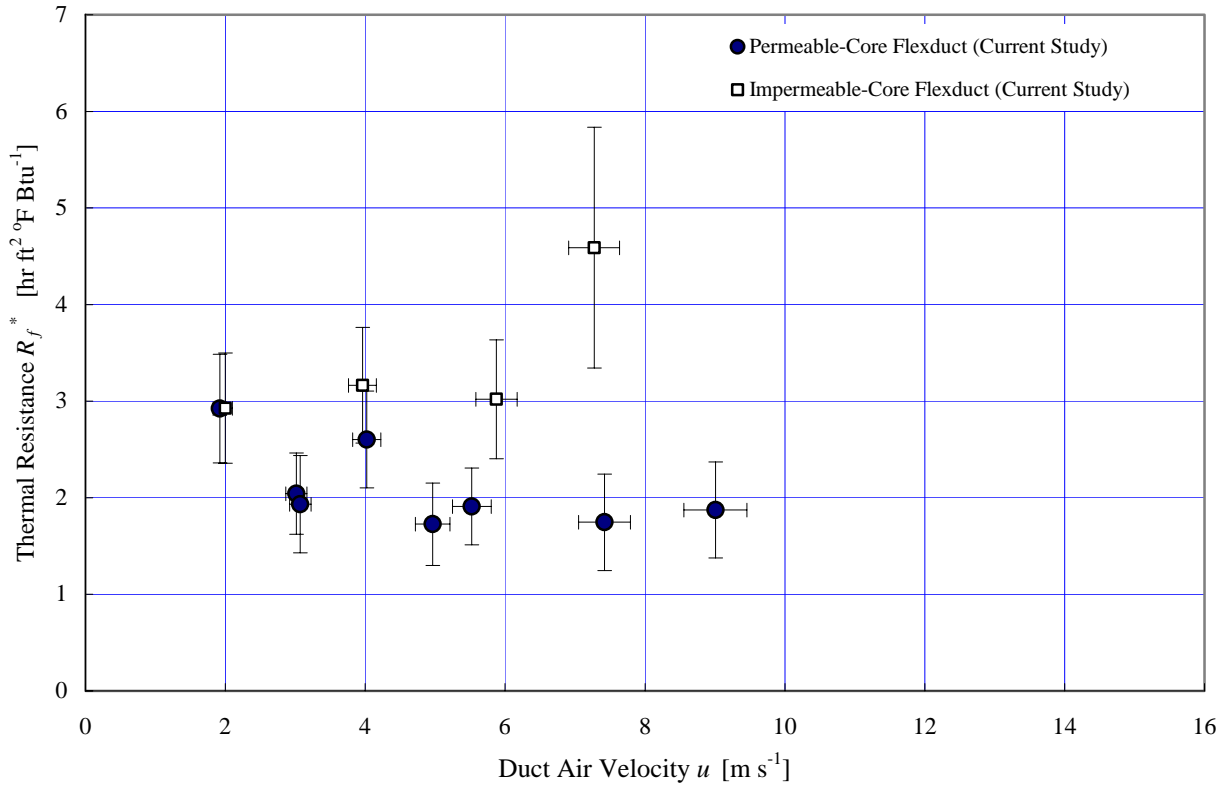## Vs. Duct Air Velocity

Figure 12. Thermal resistance at 24 ^{o}C versus duct air velocity of fiberglass liners in permeable-core flexible duct [JP Lamborn AMF-50, inner diameter 15 cm (6"), insulation thickness 3.2 cm, nominal thermal resistance 4.2 hr ft^2 F Btu^{-1} (0.76 m^2 K W^{-1})] and impermeable-core flexible duct [JP Lamborn MF-50, same dimensions and nominal thermal resistance]. Duct air temperatures were 42-55 ^{o}C , while ambient air temperatures were 18-26 ^{o}C. The ambient air velocity was approximately 0.1 m s^{-1}.

Thermal Conductance of Fiberglass Duct Liner at 24 ^{o}C Vs. Duct Air Velocity

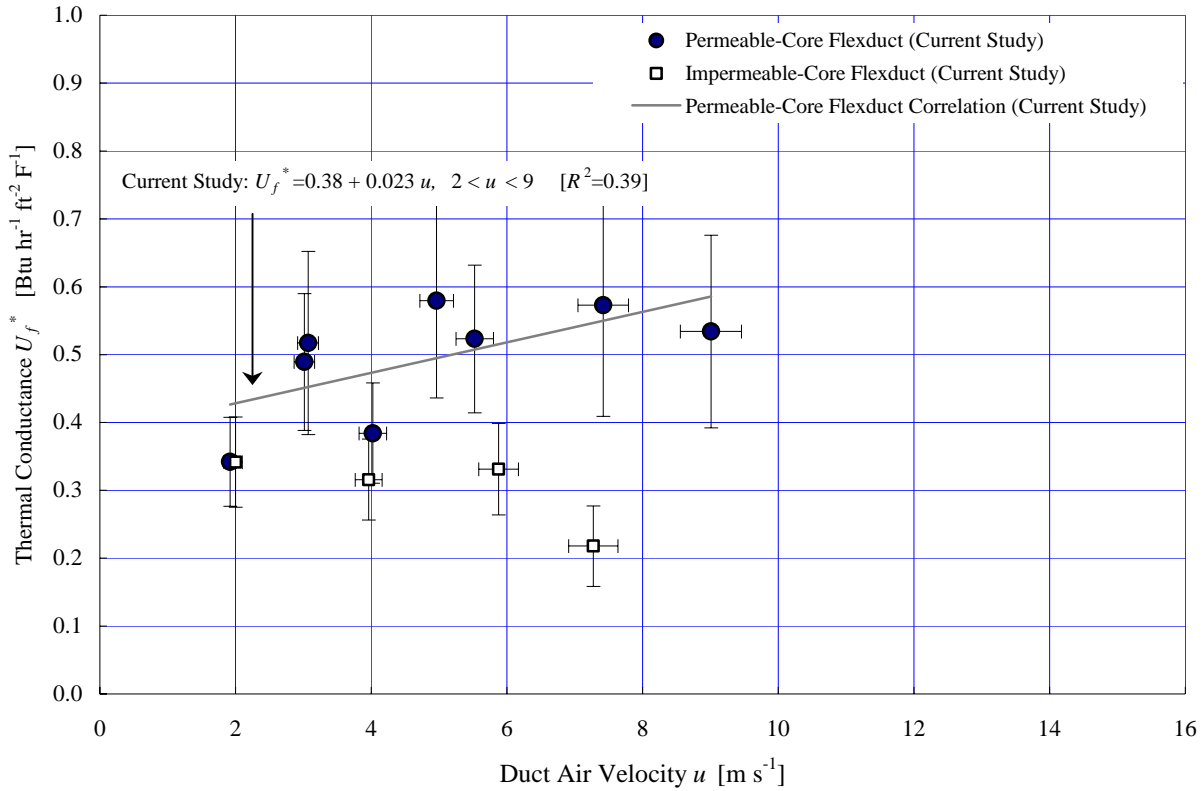Current Study: $U_f^* = 0.38 + 0.023\,u,\quad 2 < u < 9\quad [R^2 = 0.39]$

Figure 13. Thermal conductance at 24 ^{o}C versus duct air velocity of fiberglass liners in permeable-core flexible duct [JP Lamborn AMF-50, inner diameter 15 cm (6"), insulation thickness 3.2 cm, nominal thermal conductance 0.23 Btu hr^{-1} ft^{-2} F^{-1} (1.3 W m^{-2} K^{-1})] and impermeable-core flexible duct [JP Lamborn MF-50, same dimensions and nominal thermal conductance].

Thermal Resistance Distribution in Permeable-Core Insulated Flexible Duct vs. Duct Air Speed
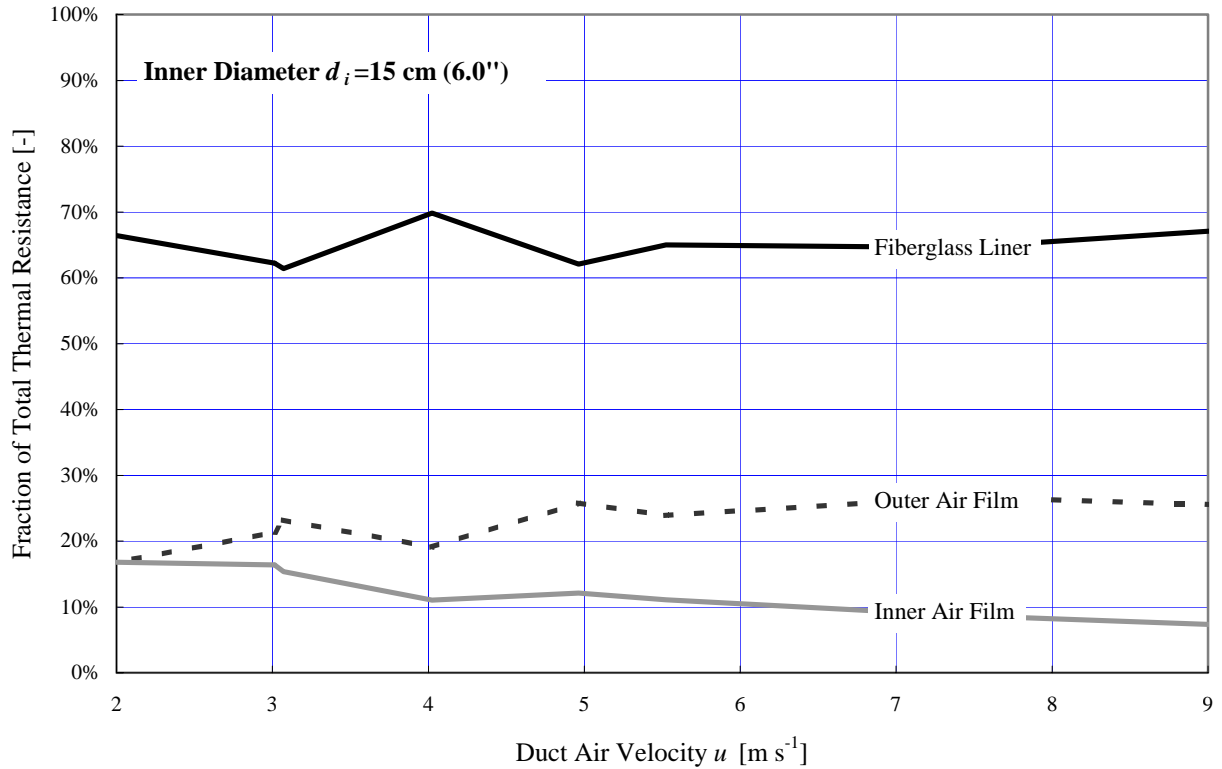
Figure 14. Fractional contributions to the total thermal resistance of the permeable-core duct liner made by the fiberglass liner, the outer air film, and the inner air film. The flexible duct had an inner diameter of 15 cm (6"), and was insulated with an R-4.2 hr ft^2 F Btu^{-1}, 2.9-cm (1.1") thick fiberglass blanket. Duct air temperatures were 42-55 ^{o}C , while ambient air temperatures were 18-26 ^{o}C. The ambient air velocity was approximately 0.1 m s^{-1}.

Sensible Heat Capacity Savings Obtained
By Rendering Liner Impermeable vs. Duct Air Velocity
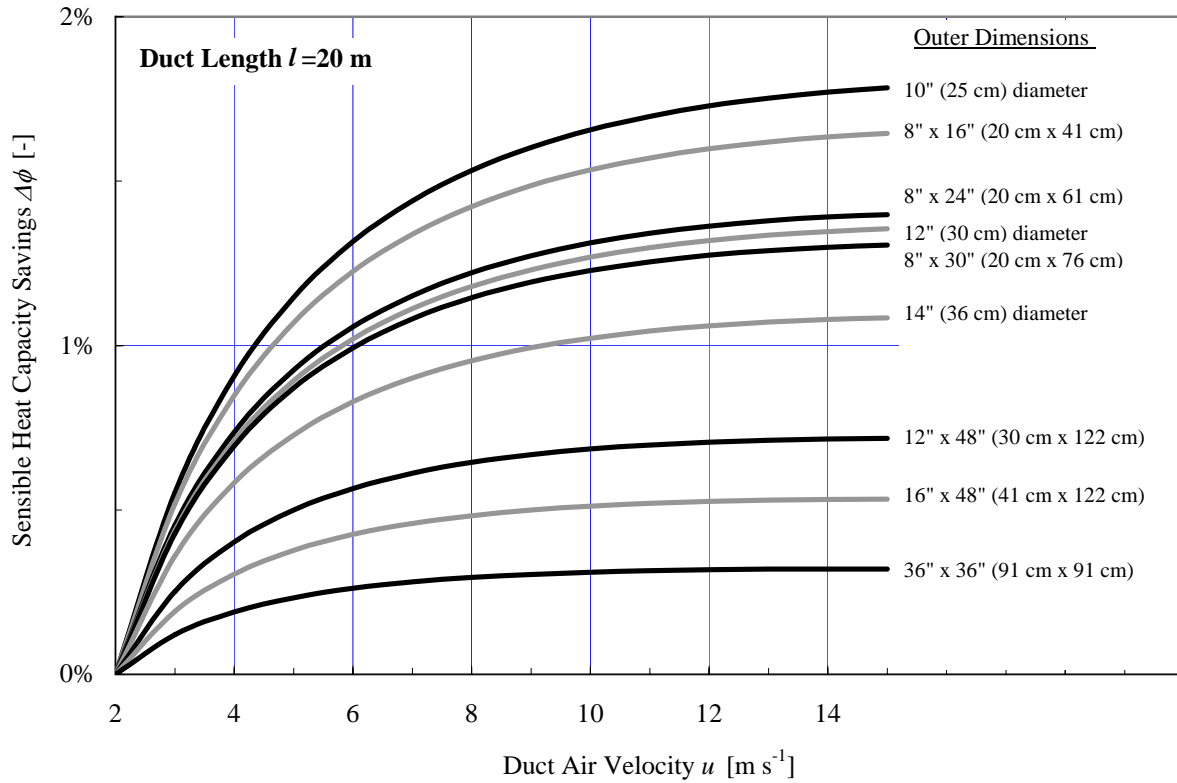

Figure 15. Fraction of sensible heat capacity that can be saved in ducts of various cross-sections by rendering impermeable a permeably-faced fiberglass duct liner, plotted versus duct air velocity. Savings are simulated for a duct length of 20 m. Duct inlet air temperature is 48 °C, ambient air temperature is 24 °C, ambient air velocity is 0.1 m s⁻¹, and the duct's exterior emissivity is 0.5. Results apply to a duct located within the building's conditioned space; savings will be a factor of γ (typically 1.5 to 2) times greater when the duct lies outside the conditioned space.

Sensible Heat Capacity Savings Obtained
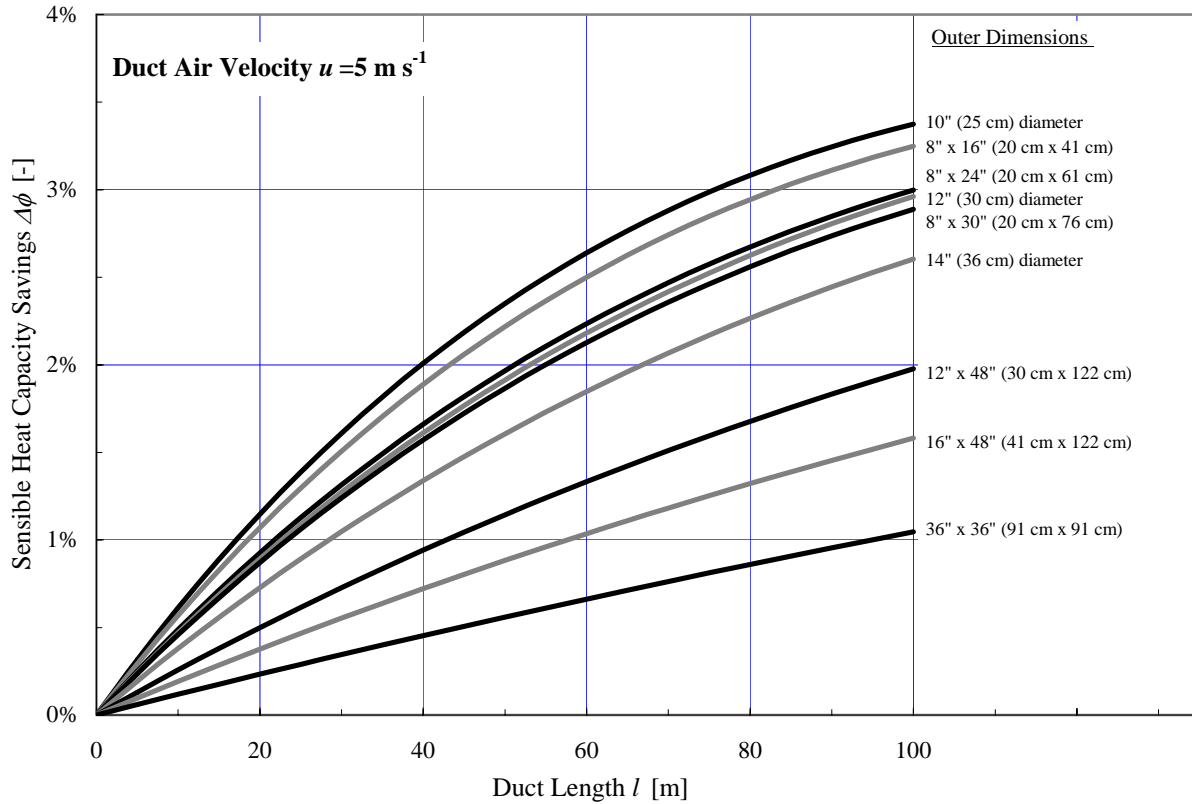## By Rendering Liner Impermeable vs. Duct Length

Figure 16. Fraction of sensible heat capacity that can be saved in ducts of various cross-sections by rendering impermeable a permeably-faced fiberglass duct liner, plotted versus duct length. Savings are simulated for a air velocity of 5 m s^{-1}. Duct inlet air temperature is 48 $^{\circ}$C, ambient air temperature is 24 $^{\circ}$C, ambient air velocity is 0.1 m s^{-1}, and the duct's exterior emissivity is 0.5. Results apply to a duct located within the building's conditioned space; savings will be a factor of γ (typically 1.5 to 2) times greater when the duct lies outside the conditioned space.

Sensible Heat Capacity Savings Obtained
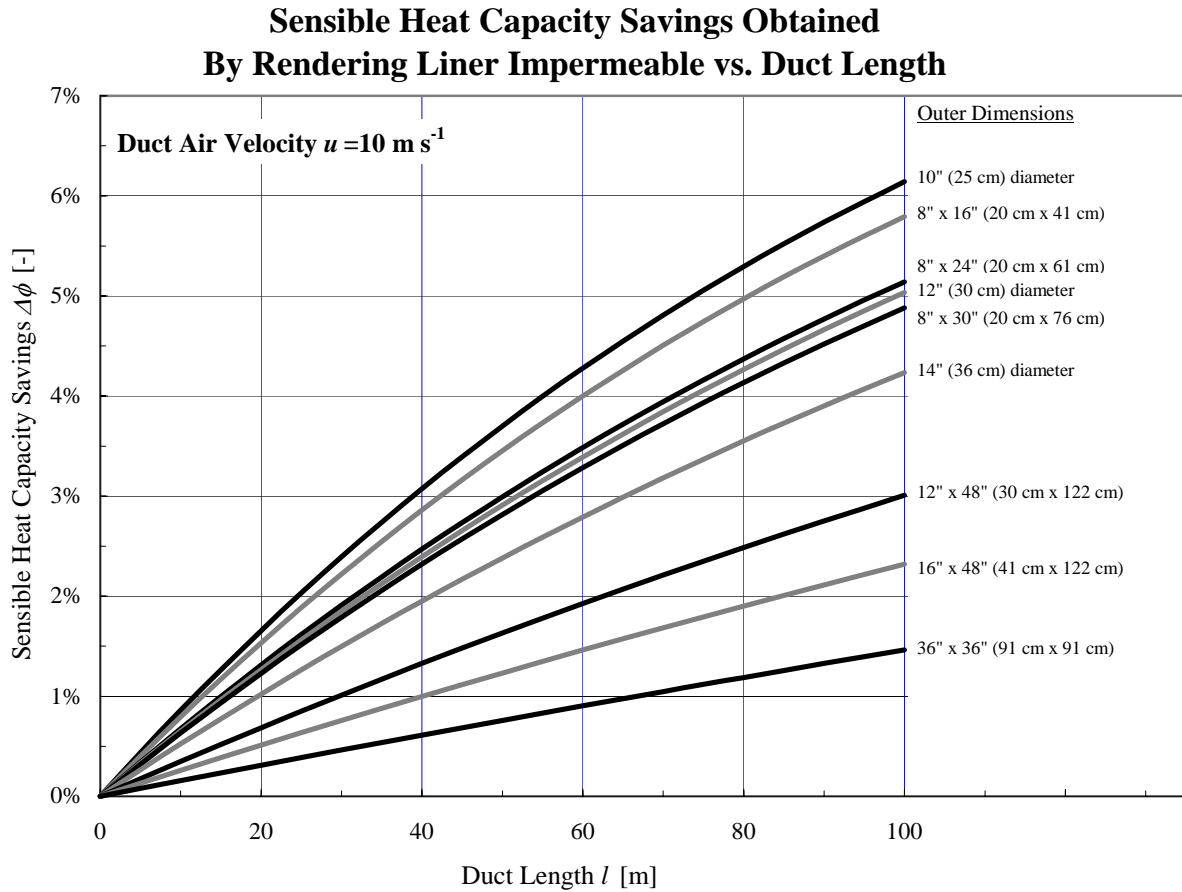## By Rendering Liner Impermeable vs. Duct Length

Figure 17. Fraction of sensible heat capacity that can be saved in ducts of various cross-sections by rendering impermeable a permeably-faced fiberglass duct liner, plotted versus duct length. Savings are simulated for a duct air velocity of 10 m s^{-1}. Duct inlet air temperature is 48 ^{o}C, ambient air temperature is 24 ^{o}C, ambient air velocity is 0.1 m s^{-1}, and the duct's exterior emissivity is 0.5. Results apply to a duct located within the building's conditioned space; savings will be a factor of γ (typically 1.5 to 2) times greater when the duct lies outside the conditioned space.