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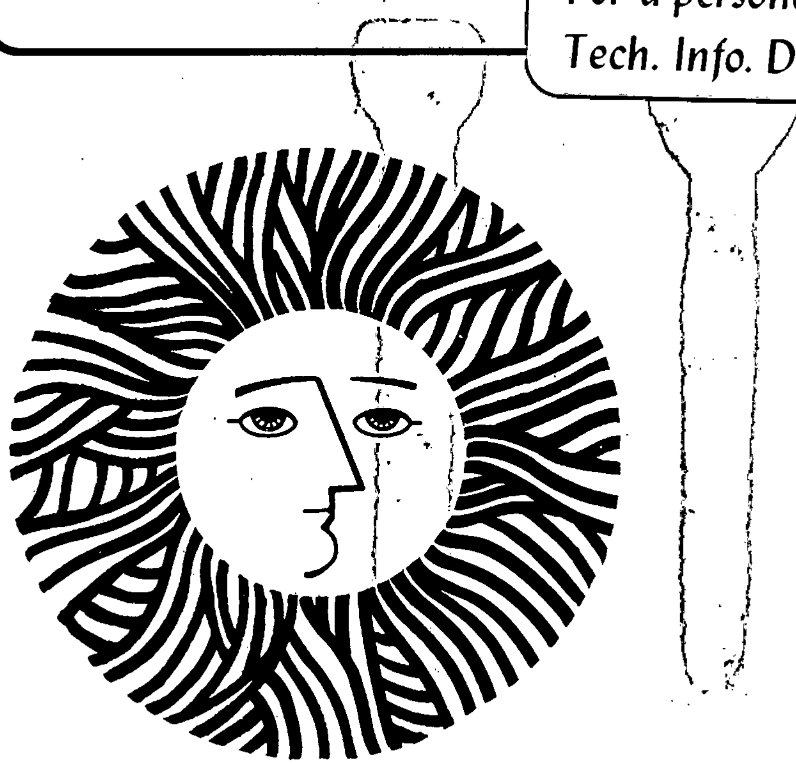

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ABSTRACT

Residential energy consumption can be decreased if air infiltration is reduced by constructing houses more tightly. In some cases, however, reduced air infiltration can lead to problems with indoor air quality (e.g., excess humidity and high levels of indoor-generated air contaminants). One solution to this problem is to install a residential air-to-air heat exchanger. The heat exchanger provides a controlled supply of ventilation which counteracts the adverse effects of reduced infiltration. In addition, the heat exchanger recovers much of the energy that would normally be lost when ventilation occurs by air infiltration. Thus, by employing heat exchangers in low-infiltration houses, it is possible to save energy without sacrificing indoor air quality.

This paper discusses the performance of residential heat exchangers and summarizes results from tests of several models. It also compares the energy consumed, during the heating season, in low-infiltration houses with heat exchangers to the energy consumed in typical houses in four cities throughout the United States. For each city, a cost-benefit analysis is performed from the point of view of a home-owner. Houses with natural gas, oil, and electrical heating systems are considered. Our analysis indicates that the energy required to heat ventilation air in homes employing heat exchangers is 5.3 to 18.0 GJ less than the energy required to heat ventilation air in typical homes. In homes with heat exchangers, the heat exchanger's fan system required 2.2 to 3.6 GJ of electrical energy during the heating season. The net present benefit for homes employing heat exchangers, when compared to typical homes, ranged from - $1350 to +$2400 and discounted payback periods ranged from five to over 30 years. The cost-effectiveness of employing heat exchangers was found to be highly affected by climate, type of heating fuel, heat exchanger performance, and ventilation rate.

INTRODUCTION

A significant amount of energy is required to heat ventilation air* in residential buildings. Estimates range from 20-40% [1, 2] of the total residential heating load or, on a national scale, 2 to 4 quads of energy yearly. Reducing infiltration rates by constructing houses more "tightly" would save a large fraction of this energy. However, some "tightly" constructed houses are subject to indoor air quality problems; e.g. excess humidity, high levels of nitrogen dioxide, formaldehyde and radon [3, 4, 5]. This problem is being addressed in Europe and Japan by the installation of mechanical ventilation systems with heat exchangers. Such systems provide a controlled supply of ventilation air to prevent increases in indoor contaminant levels and recover much of the energy that would be lost when ventilation occurs without heat recovery.

While only a small number of heat exchangers have been installed in the United States and Canada, thousands of heat exchangers are now being used in Europe and Japan. At present, little information is available on the performance of heat exchanger systems under actual operating conditions. Before such systems are employed on as large a scale in the U.S., more research is needed to investigate their performance and cost effectiveness. In addition, the relationship between indoor air quality and ventilation rate must be studied further [6] and other techniques for solving indoor air quality problems should be examined.

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*In this paper, ventilation air refers to all air entering the residence due to infiltration, natural and mechanical ventilation.

In this paper we estimate the energy savings resulting from the use of heat exchangers in new tightly constructed residential buildings, and determine their cost effectiveness in different regions of the United States. Only the winter heating season is considered. The results are very sensitive to the many assumptions that must be made to perform the analysis, thus we have presented our results in the form of two cases which span a range of likely outcomes.

DESCRIPTION OF RESIDENTIAL HEAT EXCHANGERS

The device used to provide residential mechanical ventilation with heat recovery is called a residential air-to-air heat exchanger. A residential heat exchanger generally consists of a core, two fans, and two filters all mounted in an insulated case (Figure 1). One fan brings outdoor air (supply air) through the core and into the house while the second fan causes an equal amount of house air (exhaust air) to pass through the core and out of the house. As the air passes through the core, heat is transferred from the warmer to the cooler airstream (without mixing), thus in the winter the supply air is warmed before entering the house and the exhaust air is cooled before leaving the house. Pre-warming of the supply air is the feature that renders the device energy efficient. Various core designs are available and are described in a Lawrence Berkeley Laboratory report [7]. Filters are often placed upstream of the core in each airstream. The filters remove much of the coarse dust and particulates from the airstreams, preventing deposition of these materials within the core.

Most residential heat exchangers are used with a duct system for air distribution (Figure 2). Supply ductwork carries outdoor air to the exchanger and then distributes it to various locations throughout the residence. (In many houses, the furnace duct system can be used for a portion of the supply ductwork.) Exhaust ductwork carries house air to the heat exchanger and then out of the house.

As shown in Figure 3, some units can be mounted in the wall or window (much like a window air conditioner), avoiding the need for a system of ductwork. These units are less expensive to install but they may not equally ventilate all spaces within a residence unless the indoor air is well mixed. One alternative is to install two or more small wall-mounted units at different locations within the house.

PERFORMANCE OF RESIDENTIAL HEAT EXCHANGERS

Thermal Performance

The thermal performance of a heat exchanger is usually characterized by a parameter called "effectiveness." Effectiveness is defined as the ratio of heat transfer in an actual heat exchanger to the maximum possible heat transfer one would theoretically obtain in an infinitely large counterflow exchanger operating under the same conditions. Under controlled operating conditions, a heat exchanger with an effectiveness of 70 percent will pre-heat or pre-cool the incoming air by 70 percent of the difference between indoor and outdoor temperatures. Under actual operating conditions, however, a number of factors improve or degrade heat exchanger performance and cause it to differ from that indicated by effectiveness [7]. Nevertheless, effectiveness is useful for comparing heat exchangers and as an approximate indicator for the thermal performance of heat exchangers installed in the field.

In tests performed at Lawrence Berkeley Laboratory (LBL) on ten models of residential heat exchangers the effectiveness was found to range from 45 to 84 percent [7, 8]. For our subsequent analysis, we assume that the average thermal performance of a residential heat exchanger installed in a house can be characterized by an "apparent effectiveness value" which represents the seasonal efficiency of the heat exchanger under actual operating conditions. The word "apparent" is used to distinguish this parameter from effectiveness as measured in the laboratory.

Fan Performance

The amount of ventilation provided by a residential heat exchanger and the fan power requirements are determined by the performance of the fan system and by the resistance to air flow in the heat exchanger and attached duct system. Our measurements of the fan performance of several residential heat exchanger models has shown fan power to range from 24 watts for a small window unit providing 0.018 m^3/s (38 ft^3/min) of ventilation to 185 watts for a large ducted unit providing 0.047 m^3/s (100 ft^3/min) of ventilation [8]. (Several available models of heat exchangers consume more than 200 watts of fan power, however.) Our measurements indicate that variations in fan power requirements between heat exchanger models are usually greater than the variation in fan power with air flow rate for a given model. The maximum ventilation provided by the models tested in our laboratory, ranged from 0.040 to

0.083 m^3/s (0.42 to 0.88 air changes per hour in a 340 m^3 (12000 ft^3) house).

Contaminant Control

The effectiveness of ventilation through heat exchangers in reducing contaminant levels has also been investigated by LBL in field studies [9, 10]. It appears that heat exchangers are generally effective in reducing indoor contaminant levels; however, the degree of reduction depends on the contaminant, the type of heat exchanger, and the method of heat exchanger installation.

Performance Problems

The performance of residential heat exchangers can be degraded in various ways. In cold climates, ice or frost can form inside the core and reduce both the rate of heat transfer and the flow rate of the exhaust airstream. Various freeze protection strategies are possible and some units are provided with freeze protection systems.

A second performance problem occurs when dust and particulates clog a heat exchanger's filter system and/or are deposited in the heat exchanger core. If the filters become clogged, the air flow-rates will be reduced and will become imbalanced. Imbalanced air flow causes air leakage through the building envelope, thus increasing the heat load on the furnace. Periodic cleaning or replacement of the filters is required and in some cases cleaning of the core may be required. Little data is available on the extent of this problem.

Another potential problem demanding investigation is that contaminants may be transferred from the exhaust to the supply airstream. Contaminants will be transferred if air leakage occurs from the exhaust to supply airstream. Even if there is no air leakage, some contaminants may be transferred in heat exchangers that are designed to transfer both moisture and heat between airstreams.

ENERGY SAVINGS AND FAN ENERGY CONSUMPTION

Both natural infiltration and mechanical ventilation with heat recovery impose a heat load on the home's heating system and the sum of these heat loads is called the "ventilation heat load." Ventilation with heat recovery imposes a smaller heat load than ventilation due to natural infiltration because the heat recovery system preheats the incoming air. To determine the energy savings resulting from the use of mechanical ventilation with heat recovery, we compare the energy consumption of two houses with different methods of ventilation. In the typical house, all ventilation is uncontrolled and occurs without heat recovery. In the "tight" house, a small amount of uncontrolled ventilation occurs; however, most of the ventilation is provided mechanically and passes through an air-to-air heat exchanger. Because both houses have the same total amount of ventilation during the heating season and because they are assumed identical except for the method of ventilation they should have similar levels of indoor-generated air contaminants. By subtracting the ventilation heat load in the tight house from that in the typical house, a "ventilation heat-load reduction" is determined. In addition to having different ventilation heat loads, the two houses will have different electrical demands. Operation of the fans, used in the mechanical ventilation system, causes an increase in electrical energy consumption in the tight house and the amount of increase is also determined in our calculations.

Calculation of Ventilation Heat Load in the Typical House

To calculate the ventilation heat load in the typical house, we utilized weather data from the Engineering Weather Data Manual of the U.S. Air Force [11]. This manual contains a list, by month, of the average number of hours the outdoor temperature falls within consecutive 2.8 ^{o}C (5 ^{o}F) temperature bins for cities throughout the United States. Equation 1 is utilized to calculate the ventilation heat load, "Q"

$$Q = \rho C_p V(ACH) \sum_j (T_i - T_j)\theta_j \tag{1}$$

where: ρ and Cp are the density and specific heat at constant pressure, respectively, of indoor air, V is the house volume, ACH is the air exchange rate for the house expressed in air changes per hour, and T_i is the indoor temperature. In the equation, the variable T_j is the outdoor temperature at the midpoint of a bin and θ_j, is the number of hours that the outside temperature falls within the corresponding temperature bin. The degree hour summation, in Equation 1, is computed only for hours when the outside temperature is less than the balance point of the house.* (When Equation 1. is utilized, we designate this method of calculating ventilation heat load as the bin method.)

*The balance point is the minimum outdoor temperature for which no heat is required from the home heating system and is assumed to equal 12.8 ^{o}C (55 ^{o}F).

Using this bin method, ventilation heating loads were calculated for typical homes in four U.S. cities; Minneapolis, Chicago, Washington, D.C., and Atlanta. The house volume was assumed to equal 340 m^3 (12000 ft^3) which is the average size for a new home assuming a floor-to-ceiling height of 2.5 m (8 ft.) according to the National Association of Home Builders [12]. An indoor temperature of 20 ^oC (68 ^oF) was assumed. We considered only the winter months--October through April in Minneapolis and Chicago and November through April in Washington, D.C. and Atlanta.

Calculation of Ventilation Heat Load and Fan Energy Consumption for the Tight House with Heat Exchanger

In a tight house employing mechanical ventilation with heat recovery, four factors must be accounted for when calculating the ventilation heat load: (1) uncontrolled ventilation (e.g. infiltration) in the tight house imposes a heat load, (2) operation of the heat exchanger contributes to the ventilation heat load because the heat exchanger is not 100 percent effective, (3) some fraction of the heat released by the heat exchanger's fan system is delivered to the house thus reducing the ventilation heat load, and (4) characteristics of the heat exchanger's freeze protection system affect the ventilation heat load.

For the tight house, where most of the ventilation is provided by the heat exchanger, we assume 0.2 air changes per hour (ach) due to infiltration and occupant's activities (e.g. door openings) and equation 1 is used to determine the corresponding ventilation heat load.

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We assume that the heat exchanger operates continuously during the specified heating season and that it is turned off during other times of the year. The ventilation heat load due to ventilation through the heat exchanger depends on the apparent effectiveness of the heat exchanger. This portion of the ventilation heat load is calculated using equation 1 with an air exchange rate corresponding to the rate of flow through the heat exchanger. To account for preheating of the air by the heat exchanger, the result from Equation 1 is then multiplied by the factor $(1-\Sigma)$ where Σ is the apparent effectiveness of the heat exchanger.

The third factor that affects the ventilation heat load in the tight house is the generation of heat by the heat exchanger's fans and fan motors. If both motors are located in the airstreams and downstream of the heat exchanger core, approximately 50 percent of the electrical energy consumed by the fans will be delivered to the residence in the form of heat. If the supply fan is downstream of the core and the exhaust fan is upstream; however, a larger percentage of the fan's energy consumption will be saved. Our assumption is that 75 percent of the total fan-energy consumption is saved in this case; however, the actual percentage depends on the efficiency of the fans and motors, and on the effectiveness of the heat exchanger.

As mentioned earlier, during cold weather a freeze protection system is required and its performance will affect the ventilation heat load in the tight house. We assume that the heat exchanger in the tight house employs a common freeze protection strategy, which is to periodically defrost the exchanger by shutting off the supply fan (this is usually done automatically using a thermostat and a timer). During the defrost cycle, indoor air passes through the heat exchanger, melts any frost or ice, and is exhausted from the house. Because air is being exhausted from the residence during the defrost cycle, an equivalent amount of cold outdoor air will leak into the house and contribute to the ventilation heating load. Some of this air will enter through the building envelope and some through the heat exchanger's supply ductwork if this ductwork is not blocked during defrost cycles. If air enters through the supply ductwork during defrost cycles, it will be preheated in the exchanger; however, as an approximation, we assume no preheating of incoming air during defrost cycles. To account for the corresponding portion of the ventilation heat load, we simply assume that the heat exchanger effectiveness is zero during the defrost cycles. We assume that the heat exchanger in our tight house goes into a defrost mode of operation 20 percent of the time when the outdoor temperature is below $-6.7\ ^{o}$ C (20 ^{o} F) [13]. Presently, little information is available on freeze protection requirements and the performance of freeze protection systems; however, a sensitivity analysis indicates that reasonable changes in our assumptions regarding freeze protection have a small effect on our results.

The total ventilation heat load for the tight house is calculated by summing the loads due to infiltration and mechanical ventilation (with heat exchanger inefficiency and defrost cycles accounted for) and subtracting the amount of fan energy delivered to the house. The electrical energy consumed by the fan system is determined using the time of operation for each fan and the fan power.

Sensitivity Analysis for Energy Savings

A sensitivity analysis was performed to determine how variations in our assumptions affect the projected energy savings. Because many factors have a significant effect, results for two cases will be presented. For case A, we assumed a medium to low ventilation rate (0.6 ach), a medium to low heat exchanger effectiveness, and a fairly high fan power. For case B, the ventilation rate was increased to 0.75 ach and a superior heat exchanger performance (i.e. higher effectiveness and lower fan power) was assumed. Major assumptions for each case are summarized in Table 1.

Results of Energy Analysis

Results from the energy analysis are presented in Table 2 for the four cities considered. Ventilation heat loads are tabulated for both the typical and tight homes. These loads equal the amount of energy that must be supplied by the home heating system to heat ventilation air during the specified season. Also tabulated is the "ventilation heat load reduction" which equals the ventilation heat load in the typical house minus that in the tight house. This reduction in heat load represents the savings attributable to the construction of a tight house and use of a residential heat exchanger.

The reduction in ventilation heating load ranged from 5.3 to 18.0 GJ (50 to 171 therms) with the larger reductions occurring in cities with cold climates. Percentage reduction in ventilation heat load (i.e. heat load reduction divided by heat load in the typical house) ranged from 45 to 63 percent. While absolute energy savings are much greater in cold climates, our results indicate that percentage savings are greater in cities with warm climates where less time is required to defrost the heat exchanger. However, our analysis did not consider the possibility that, in cold climates, the rate of condensation of water vapor from the warm airstream may increase, thus causing an increase in heat exchanger performance sufficient to counteract this effect.

Tabulated in the last columns of Table 2 is the total electrical energy consumed by the heat exchanger's fan system. The amount of energy consumed by the fans is less for case B because we assumed a lower fan power (i.e. more efficient fans and fan motors). Fan energy consumption was greater in Minneapolis and Chicago than in Washington D.C. or Atlanta because a longer season of operation was assumed for these cities.

The reduction in ventilation heat load can be compared to the fan energy consumption. The ratio of these two quantities (i.e. heat load reduction divided by fan energy consumption) ranges from 1.68 to 3.34 for case A, with the larger ratios occurring for the colder climates. For Case B, this ratio is much larger--ranging from 3.64 to 7.24. As shown in the subsequent economic analysis, this ratio of energy savings to energy consumption has a large impact on the cost-effectiveness of investments in residential heat exchanger systems.

To produce the electrical energy required to run the fans, approximately 3.4 times as much resource energy* is utilized at the power plant that generates the electricity [14]. Thus, for homes with natural gas

*The term "resource energy" refers to primary energy sources such as oil, natural gas, and coal.

or oil heating systems, national savings of resource energy occur only when the reduction in ventilation heating load is at least 3.4 times greater than the fan energy consumption. For case A, in many situations the amount of resource energy consumed by the tight house will be greater than that consumed by the typical house. For electrically heated homes, and for all case B homes, however, the tight house with a heat exchanger consumes less resource energy.

COMPARISON OF BIN METHOD AND DOE-2 COMPUTER PROGRAM FOR CALCULATING VENTILATION HEATING LOAD

Several approximations are made when using the bin method to calculate ventilation heat loads. For example, a constant air change rate is assumed when using the bin method (i.e. equation 1). In reality, air change rates (infiltration rates) increase with an increase in windspeed or a decrease in outdoor temperature. Because a larger fraction of the total seasonal infiltration occurs when the outdoor temperature is low, equation 1 may underpredict ventilation heating loads. To assess the accuracy of the bin method, we compared results obtained by this method to those obtained from using the DOE-2 computer program [15] described below.

DOE-2 is a public domain computer program for energy analysis of buildings. Assumptions used when DOE-2 is the computational tool are discussed by Levine and Goldstein [16]. To calculate air change rate (ACH) the DOE-2 program uses hourly weather data for a test reference year (TRY) and an equation of the form

$$ACH = a + bV + c\left|\Delta T\right| \qquad (2)$$

where V is the wind speed and ΔT is the difference between indoor and outdoor temperatures. In order to match typical field data on air exchange rates, values for constants a, b, and c are adjusted by the program on a monthly basis so that the average air change rate for the five-month winter heating season (November--March) equals 0.7 air changes per hour. The DOE-2 program calculates infiltration rates each hour and uses the results to calculate an hourly ventilation heating load. The seasonal ventilation heating load is then determined by summing the hourly loads.

To compare results obtained by the bin method with those derived from the DOE-2 program, we calculated a special set of ventilation heat loads using the bin method with assumptions matching those in DOE-2 (i.e. an indoor temperature of 21.1^{o}C (70^{o}F) and an air exchange rate of 0.7 ach were assumed, and heat loads were calculated only for the months of November through March).

Results of Comparison

Ventilation heat loads calculated by the bin method and by the DOE-2 program are presented in Table 3. As indicated, the DOE-2 heat loads are higher than predictions by the bin method for the cities of Minneapolis and Atlanta, 7 and 11%, respectively. For Chicago and Washington D.C., the ventilation heating loads predicted by the DOE-2 program are lower than those predicted by the bin method--6 and 3%, respectively. Thus, the agreement between the two methods is reasonable for all cities considered. Possible reasons for the discrepancies are: (1) a constant air change rate is assumed for the bin method and an hourly air change rate is calculated by the DOE-2 program, (2) a balance point temperature is assumed for the bin method whereas the DOE-2 program utilizes thermal characteristics of the house (e.g., insulation thickness) and performs an hourly energy balance on the house to determine when heat is required, and (3) the bin method is based on long term average weather data whereas the DOE-2 program utilizes weather data for a test reference year (TRY) which less accurately represents an average year. Balance point temperature can have a significant impact on ventilation heat load. For our comparison, however, the balance point temperature assumed for our bin-method calculations corresponds well to the house characteristics utilized by the DOE-2 program. Our calculations indicate that the different weather data is probably the largest cause of the observed discrepancies.

ENERGY SAVINGS DURING COOLING SEASON AND PEAK POWER SAVINGS

In the United States residential sector, air conditioning consumes only about one quad of resource energy whereas space heating consumes about ten quads. For the four cities studied, the summer ventilation load is much less than the winter ventilation load. Therefore, in these cities the magnitude of the potential energy savings from heat exchanger use in the summer is much less than during the winter season. We have not included cooling season energy savings in the calculations already described. However, in locations with significant heating and cooling loads (e.g., New York City and Washington, D.C.) additional energy savings may result if heat exchangers are utilized during the summer cooling season.

In many utility districts, air conditioning is the source of the summer peak in power demand and a driving force in the construction of expensive new power plants. The air conditioning load in houses with heat exchangers will be lower than in houses without heat exchangers thus reducing the peak power load in summer. In cases where investments in heat exchangers are not cost effective from the home owner's point of view, heat exchanger use may still be desirable because of the resulting reductions in peak power demand or because the cost of conserved energy is less than the long term incremental cost for new energy.

COST/BENEFIT ANALYSIS

Methodology

The economic desirability of employing a heat exchanger and mechanical ventilation system in a residential building can be assessed by comparing the savings in energy costs derived from the use of such a system to the incremental costs incurred from installation, maintenance, and operation of the system. There are a number of economic parameters that may be used to rank potential capital investments. These include rate of return, net present benefit or life-cycle cost, discounted payback period, and benefit to cost ratio [17]. All of these, except discounted payback period, yield the same rank ordering when potential investments of equal lifetime are compared.

In our cost-benefit analysis, we used net present benefit (NPB), benefit to cost ratio, and discounted payback period to determine cost effectiveness. NPB is calculated by subtracting capital, operating and maintenance costs from the energy cost savings. Equation (3) is the NPB equation we have used for this study.

$$\mathrm{NPB} = (\mathrm{FBS}) \sum_{i=1}^{N} \left\{ \frac{1+f}{1+d} \right\}^{i} - (\mathrm{OPC}_{F}) \sum_{i=1}^{N} \left\{ \frac{1+f_{e}}{1+d} \right\}^{i} - \mathrm{CC} - M \sum_{i=1}^{N} \frac{1}{(1+d)^{i}} \quad (3)$$

where:

FBS = fuel bill savings in year 1

OPC_f = operating cost of fans in year 1

CC = incremental capital cost of conservation measure

M = annual maintenance costs

f = real escalation rate for heating fuel price

f_e = real escalation rate for electricity price

N = lifetime of heat exchanger

d = real discount rate

The value of future cash flows is discounted by an appropriate discount factor d, that corrects for the lost opportunity to otherwise invest resources.

Table 2, discussed previously, summarizes the ventilation heat load reduction and the fan energy consumption for the two different cases, A and B, in the four cities studied. The fuel bill savings in year 1, FBS, is the product of the ventilation heat load reduction and the fuel price in year 1 divided by the furnace system efficiency (assumed to be 70% for natural gas and oil and 100% for electrical heating). The operating cost of the fans in year 1, OPC_f is the product of the fan energy consumption (Table 2) and the price of electricity in year 1.

In addition to net present benefit, two other economic parameters, benefit-to-cost ratio and discounted payback period, have been calculated. The benefit-to-cost ratio is equal to the sum of the discounted energy cost savings (i.e. the first term in equation (1)) divided by the sum of capital costs and discounted fan operating and maintenance costs (i.e. the sum of remaining terms in equation (1)).

The discounted payback period is defined as the length of time required to recover an initial investment taking into account fuel price escalation rates and the time value of money. Thus, it equals the amount of time required for the NPB to become positive.

Assumptions

To evaluate the various economic parameters which are indicators of the cost-effectiveness of using heat exchangers in low infiltration houses, we estimated the incremental capital cost of the conservation measure which consists of the costs for the added labor and materials necessary to construct a low infiltration house and the purchase price plus installation cost of the heat exchanger, fans and connecting duct-work. Based upon our communication with home builders, we have estimated the cost of constructing a tighter building envelope for a 140 m^2 (1500 ft^2) house to be approximately \$400. This procedure involves caulking around the windows and door frames, at the base of walls, around penetrations in the walls (e.g. electrical outlets, plumbing) and the installation of a polyethylene vapor barrier in the walls and ceiling. The incremental cost for installation of the polyethylene vapor barrier is small because the polyethylene barrier is a substitute for other less effective types of vapor barriers.

Well constructed heat exchangers with fans, adequate to ventilate a 340 m^3 (12000 ft^3) house, can be obtained in large quantities for \$450--\$600. From our field experience we estimate \$300 to install the heat exchanger, fans and duct work in a new house. Thus the total marginal cost of the low infiltration heat exchanger option has been chosen to be \$1150 for the more optimistic case B and \$1300 for the more pessimistic case A.

In the above cost estimation we have assumed that a heating system with duct work will be installed in each new house being considered and that the heating system's ductwork is utilized for portions of the heat exchanger's supply ductwork. In the case of homes with baseboard type electric resistance heating, there is normally no ductwork installed. If heat exchangers are installed in tightly built homes with baseboard heat, it may be possible to provide additional ventilation through window or wall mounted heat exchangers. Because the cost and performance of window-or-wall mounted heat exchangers does not match assumptions made for this analysis, our analysis is not appropriate for evaluation of these units.

There is presently little experience with maintenance costs for heat exchangers. We have assumed that filters will need periodic cleaning or replacement and that the core may also need periodic cleaning. Case A & B include a maintenance cost of $30 and $10 per year respectively. This cost is assumed to grow at the rate of inflation, thus the escalation rate is zero in real terms.

Table 4 shows the initial fuel prices and Table 5 shows the fuel price escalation rates (in real terms) for the various heating fuels considered in our calculations. We have used 1981 fuel prices gathered from the U.S. Department of Labor, Bureau of Labor Statistics. The fuel price escalation rates were obtained from the mid range estimates used for obtaining energy consumption projections displayed in the National Energy Policy Plan (NEPP). The discount rate was chosen to be 5% real and the lifetime of the heat exchanger system was assumed to be

20 years.

Results and Discussion

Table 6 summarizes the results of our economic analysis for four cities and three heating fuels. The net present benefit, benefit to cost ratio, and the discounted payback period are shown for case A and case B. For case A the net present benefit is positive in only two situations, for electric heating in Minneapolis and Chicago. Thus, under case A conditions, heat exchangers installed in low-infiltration houses would appear to be cost-effective only in electrically heated houses in Minneapolis and Chicago. For all other combinations of fuel and location, except oil heating in Minneapolis, the discounted payback period is greater than 30 years.

Under case B conditions, the net present benefit is positive for all three heating fuels in Minneapolis, Chicago and Washington D.C. In Atlanta the NPB is negative for both heating fuels (i.e. natural gas and electricity). The payback period for houses with oil or electric heating is 11 years or less in all cities except Atlanta. For all four cities, the payback period for natural gas heating is always greater than for oil or electricity. If the conditions shown in Table 1 for case B are satisfied, then the use of heat exchangers in low-infiltration houses should be cost-effective in climates with 2200 or more heating degree days (^{o}C) for all heating fuels.

The differences in the assumptions used for cases A and B can be ranked in order of significance. We assumed a superior heat exchanger performance for case B as compared to case A (i.e., the heat exchanger has a higher effectiveness and a lower fan power, and a larger percentage of the energy consumed by the fans is delivered to the residence). This set of assumptions causes the largest difference in economic results for the two cases. For case B we also assumed a higher ventilation rate, and this assumption has a very significant but slightly smaller impact on the economic results. Finally, we assumed lower initial and maintenance costs for case B. These cost related assumptions had a smaller impact than the performance and ventilation assumptions described above, especially in the cities with cold climates.

Sensitivity Studies

The results of the economic analysis depend on estimates for several factors. These include discount rate, fuel price escalation rate, appliance lifetime, initial fuel price, and initial capital cost. There is almost always some uncertainty in projecting values for these parameters. Thus, a sensitivity analysis was performed to determine the importance of several key assumptions. Table 5 shows the low, medium, and high predictions for escalation rates in the price of heating oil, for which the NEPP indicates the greatest future uncertainty. We performed a cost-benefit analysis using both the high and low price forecasts. For case A, a change in NPB from negative to positive occurs in Minneapolis and Chicago for the high price escalation rate. For case B, no change in the sign of the NPB occurs for either the high or low fuel price escalation rates.

The rate at which future benefits and costs are discounted to the present can have a significant effect on the results of our cost-benefit analysis. Our original assumption of a 5% real discount rate implies that new home buyers, as an alternative to investing in a more energy-efficient house, could choose to invest differently and earn 5% more than the inflation rate. Thus, if the inflation rate were 11%, consumers could obtain 16% interest with their money. We assessed the sensitivity of our results to both higher and lower discount rates of 10% and 3% in two cities, Minneapolis and Washington D.C. For case A, the NPB remains negative for all fuels in Washington D.C. for discount rates of 3 to 10%. In Minneapolis, the NPB can change sign for oil and electric heating, producing a positive NPB for oil heating with d = 3% and a negative NPB for electric heating with d = 10%. For case B, there is only one change in sign (NPB becomes negative); it occurs in Washington D.C. for gas heating at a discount rate of 10%.

The effect of a change in the initial capital cost on NPB is easy to determine and reasonable changes do not have a large effect on our results. For example, if initial capital costs are $200 higher than assumed for our analysis, the NPB is reduced by $200. If initial capital costs are $200 less than assumed, the NPB is increased by $200.

Assuming a 30 year life for the heat exchanger instead of 20 years generally produces a small increase in the NPB but only in one case does a negative NPB become positive (Case A, Minneapolis, oil heating, where the NPB changes from--$105 to $428). Decreasing the heat exchanger lifetime to 10 years causes the NPB for case B to become negative in four situations where it was previously positive. These are natural gas heating in Minneapolis, Chicago, and Washington, D.C. and oil heating in Washington, D.C. For case A, in the the two situations where NPB was positive, electric heating in Minneapolis and Chicago, the NPB becomes negative. In summary, the net present benefit is not as sensitive to the expected range of values for fuel price escalation rate and discount rate as to the expected range of heat exchanger performance parameters, ventilation rate, and appliance lifetime.

Sensitivity to Method of Freeze Protection

Various techniques can be used to prevent freezing in heat exchangers and, as mentioned earlier, freeze protection requirements are not well understood. A brief sensitivity analysis was performed to evaluate the economic impact of variations in our assumptions regarding freeze protection. For our original analysis, we assumed that freezing initiated when the outdoor temperature was below -6.7 °C (20 °F). We also assumed that the heat exchanger had a defrost freeze protection system which defrosted the heat exchanger 20 percent of the time when the outdoor temperature was below the onset of freezing (20 °F). In a sensitivity analysis, the amount of defrost time was decreased and increased by 50 percent of the original value. In addition, the temperature at which freezing was assumed to initiate was reduced to -12.2 °C (10 °F) and increased to 0 °C (32 °F). We also compared the defrost technique of freeze protection to the use of electrical resistance preheat for the prevention of freezing. In systems with electrical resistance preheat, an electric resistance heater preheats the incoming outdoor air (upstream of the heat exchanger core) sufficiently to prevent freezing within the core. For our analysis, we assumed that the incoming air was preheated to the temperature corresponding to the onset of freezing.

The sensitivity study of freeze protection techniques was performed only for Minneapolis--the city with the coldest winter climate where changes in freeze protection techniques should have the greatest impact. We considered only case B. As shown in Table 7, a 50% increase in defrost time causes less than a $300 decrease in net present benefit and a 50 percent decrease in defrost time has a similarly small effect on NPB. Decreasing the temperature corresponding to the onset of freezing to $-12.2\ ^{o}C$ ($10\ ^{o}F$) causes only small increases in NPB. Increasing this temperature to $0\ ^{o}C$ ($32\ ^{o}F$), however, can cause a $520 decrease in NPB for a heat exchanger with electric preheat in a home heated with natural gas. The NPB for systems with electric preheat is usually close to that for systems with defrost freeze protection. When compared to defrost systems, preheat systems are least attractive in homes with inexpensive gas heat. Summarizing, the effect of different freeze protection techniques on NPB is generally less than the effect of heat exchanger performance, ventilation rate, or type of heating fuel.

SUMMARY OF RESULTS AND RECOMMENDATIONS

A summary of our results follows:

1. The reduction in ventilation heating load ranged from 5.3 to 18 GJ (50 to 170 therms) per heating season for tight houses with heat exchangers relative to loose houses of equivalent ventilation rate.

2. Ventilation heating loads calculated by the bin method and by the DOE-2 computer program agreed to within ±11% for the four cities studied.

3. For case A, heat exchangers are cost-effective only in electrically heated houses in climates as severe as Chicago.

4. For case B, heat exchangers are cost-effective for homes heated with natural gas, oil, or electricity in Minneapolis, Chicago and Washington D.C. The payback period for houses with oil or electric heating is 11 years or less in all three cities.

5. The results of our economic analysis are not highly sensitive to reasonable variations in assumptions concerning freeze protection, fuel price escalation rate and discount rate.

6. The results of our economic analysis are highly sensitive to assumptions made concerning heat exchanger effectiveness and fan power.

7. The assumptions of ventilation rate strongly affect the results of our cost-benefit analysis. If low ventilation rates (<.5 ach) are adequate to provide suitable indoor air quality, then heat exchangers in low infiltration houses will not be cost-effective except in the coldest climates and only when high performance heat exchangers are used.

8. A decrease in heat exchanger lifetime (from the assumed 20 yrs to 10 yrs) has a significant adverse effect on the cost effectiveness of heat exchangers in low infiltration houses.

Based upon the results of our study we make the following recommendations:

1. Consumers should consider heat exchanger performance carefully and be willing to pay a significantly higher first cost for a unit with

a superior performance.

2. Accurate data on heat exchanger performance should be made available to consumers.

3. Further research is needed to investigate factors that degrade heat exchanger performance such as freezing of moisture within the core, transfer of contaminants between airstreams, poor distribution of ventilation air, and fouling of heat transfer surfaces. Further study is also needed to evaluate maintenance requirements and costs for residential heat exchangers.

4. From the homeowner's point of view, investing in a tightly constructed house and installing a heat exchanger is often not economical, even though, in a number of cases, considerable energy savings occur when such a strategy is employed. Because, some of the benefits of the energy savings are not accrued by the homeowner, the potential for using government or utility subsidies, low-interest loans or tax incentives to stimulate the use of heat exchangers should be evaluated.

5. Mechanical ventilation with heat recovery is only one of many possible strategies for conserving energy and controlling indoor contaminant levels. Further research is needed to evaluate other strategies that may be superior to the use of heat exchangers.

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Table 1. Major Assumptions for Energy and Economic Analysis--Cases A and B.

	Case A typical house	Case A tight house	Case B typical house	Case B tight house
Infiltration rate (ach)	0.60	0.20	0.75	0.20
Ventilation through HX* (ach)	--	0.40	--	0.55
House volume (m^3 ($ft^3 \times 10^{-3}$))	339.6 (12)	339.6 (12)	339.6 (12)	339.6 (12)
Balance point (^oC(^oF))	12.8 (55)	12.8 (55)	12.8 (55)	12.8 (55)
Indoor temperature (^oC (^oF))	20 (68)	20 (68)	20 (68)	20 (68)
Apparent HX effectiveness	--	0.65	--	0.75
Total HX fan power (W)	--	200	--	140
Percent of total fan power that is delivered to residence in the form of heat (%)	--	50	--	75
Outdoor temperature at onset of freezing in HX (^oC (^oF))	--	-6.7 (20)	--	-6.7 (20)
Defrost time fraction (%)	--	20	--	20
Initial capital cost ($)	--	1300	--	1150
Yearly maintenance cost ($)	--	30	--	10

*HX = Heat Exchanger

Table 2. Results of Energy Analysis for Houses in Four U.S. Cities.

City	Case	*Vent. heat load in typical house		†Vent. heat load in tight house with heat exchanger		‡Vent. heat load reduction		§Fan energy consumption	
		GJ	Therms	GJ	Therms	GJ	Therms	GJ	Therms
Minneapolis	A	26.3	249	14.3	136	11.9	113	3.6	34
	B	32.8	311	14.8	140	18.0	171	2.5	24
Chicago	A	21.9	208	11.4	108	10.6	100	3.6	34
	B	27.5	261	11.5	109	16.0	152	2.5	24
Washington D.C.	A	15.2	144	7.5	71	7.8	74	3.1	30
	B	19.1	181	7.4	70	11.7	111	2.2	21
Atlanta	A	9.9	94	4.6	44	5.3	50	3.1	30
	B	12.3	117	4.5	43	7.8	74	2.2	21

*Heat load imposed on heating system during the specified heating season due to uncontrolled ventilation.

†Heat load imposed on heating system due to uncontrolled ventilation, heat exchanger inefficiency, and heat exchanger defrost cycles minus the amount of energy that is released by fan system and delivered to the residence.

‡Ventilation heat load in typical house minus that in tight house.

§Total electrical energy consumption by both fans.

Table 3. Comparison of Ventilation Heat Loads Calculated by Bin Method and DOE-2 Computer Program.

	Ventilation Heat Load* GJ (Therms)				
	Bin Method		DOE-2		% Difference
Minneapolis	27.4	(260)	29.4	(279)	7
Chicago	23.1	(219)	21.7	(206)	6
Washington, D.C.	17.5	(166)	16.9	(160)	3
Atlanta	11.8	(112)	13.1	(124)	11

*For the specified heating season.

Table 4. 1981 Fuel Prices*

City	Natural Gas	Oil $/GJ ($$10^6$ Btu)	Electricity
Minnesota	3.78 (4.01)	7.32 (7.76)	14.34 (15.20)
Chicago	3.52 (3.74)	7.64 (8.10)	19.06 (20.20)
Washington D.C.	4.77 (5.06)	7.88 (8.35)	16.04 (17.00)
Atlanta	4.95 (5.25)	--	13.02 (13.80)

*Source: U.S. Department of Labor, Bureau of Labor Statistics. January 1981 prices. 1.4 therms/gallon of oil assumed.

Table 5. Annual Fuel Price Escalation Rates*

Time Period	Natural Gas	Oil Low	Oil Mid	Oil High	Electricity
1981-85	13.1	2.9	5.7	8.1	0.6
1986-90	2.5	1.6	2.9	5.6	2.2
1991-2010	1.8	1.7	2.7	6.3	1.4

*Real escalation rates obtained from National Energy Policy Plan, 1981. Mid-range forecasts are given for natural gas and electricity and mid, low, and high range forecases are given for heating oil.

Table 6. Results of Cost Benefit Analysis for Cases A and B.

City (Heating °C Days*)	Heating fuel	Net Present† Benefit $		Discounted Benefit-Cost† Ratio		Payback Period (years)	
		Case A	Case B	Case A	B	A	B
Minneapolis (4657)	Gas	-844	+670	0.65	1.40	>30	12
	Oil	-105	+1790	0.96	2.07	22	7
	Electricity	+9	+1963	1.00	2.17	20	6
Chicago (3688)	Gas	-1358	+83	0.49	1.05	>30	19
	Oil	-520	+1348	0.80	1.73	>30	9
	Electricity	+194	+2425	1.07	2.31	17	5
Washington, D.C. (2347)	Gas	-1099	+246	0.54	1.15	>30	16
	Oil	-767	+743	0.68	1.45	>30	11
	Electricity	-620	+963	0.74	1.58	>30	9
Atlanta (1645)	Gas	-1352	-247	0.40	0.84	>30	29
	Electricity	-1284	-147	0.43	0.91	>30	25

*yearly total degree days, 18.3 °C base.

†based on a 20 year life for heat exchanger and zero salvage value.

Table 7. Effect of Freeze Protection System on Net Present Benefit for Minneapolis, case B.

Onset of Freezing °C (°F)	Heating Fuel	Net Present Benefit* Defrost F†=0.1	F=0.2	F=0.3	Preheat‡
	Gas	819	760	701	727
-12.2 (10.0)	Oil	2015	1926	1837	1967
	Elec.	2200	2106	2013	2159
	Gas	774	670	566	499
-6.7 (20.0)	Oil	1947	1790	1633	1760
	Elec.	2128	1963	1798	1955
	Gas	690	501	313	-21
0.0 (32.0)	Oil	1818	1533	1247	1288
	Elec.	1993	1692	1392	1490

*based on a 20 year life for heat exchanger

†hours of defrost divided by total hours when outdoor temperature is below onset of freezing.

‡electric resistance heater preheats outdoor air to temperature at onset of freezing

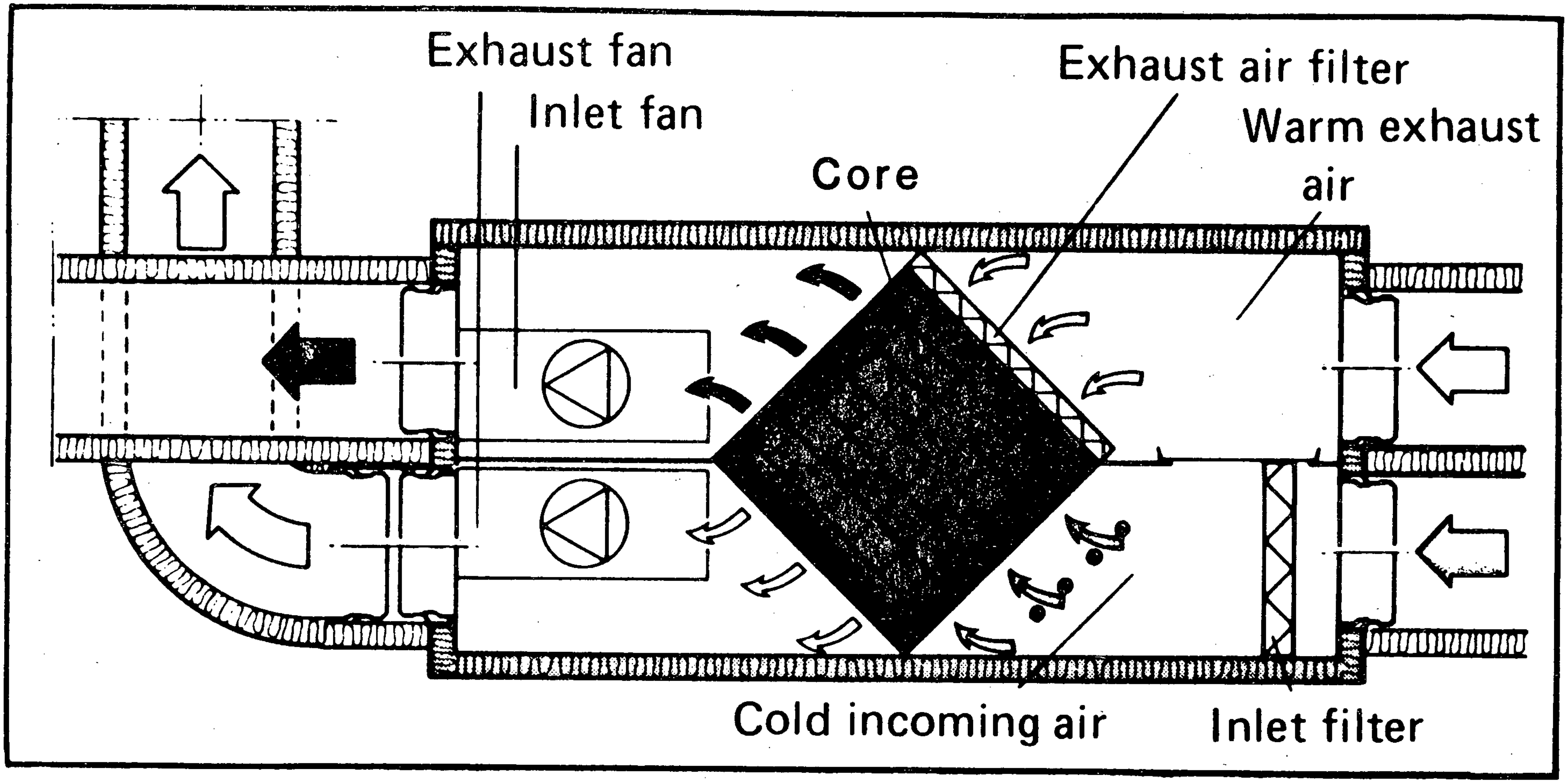
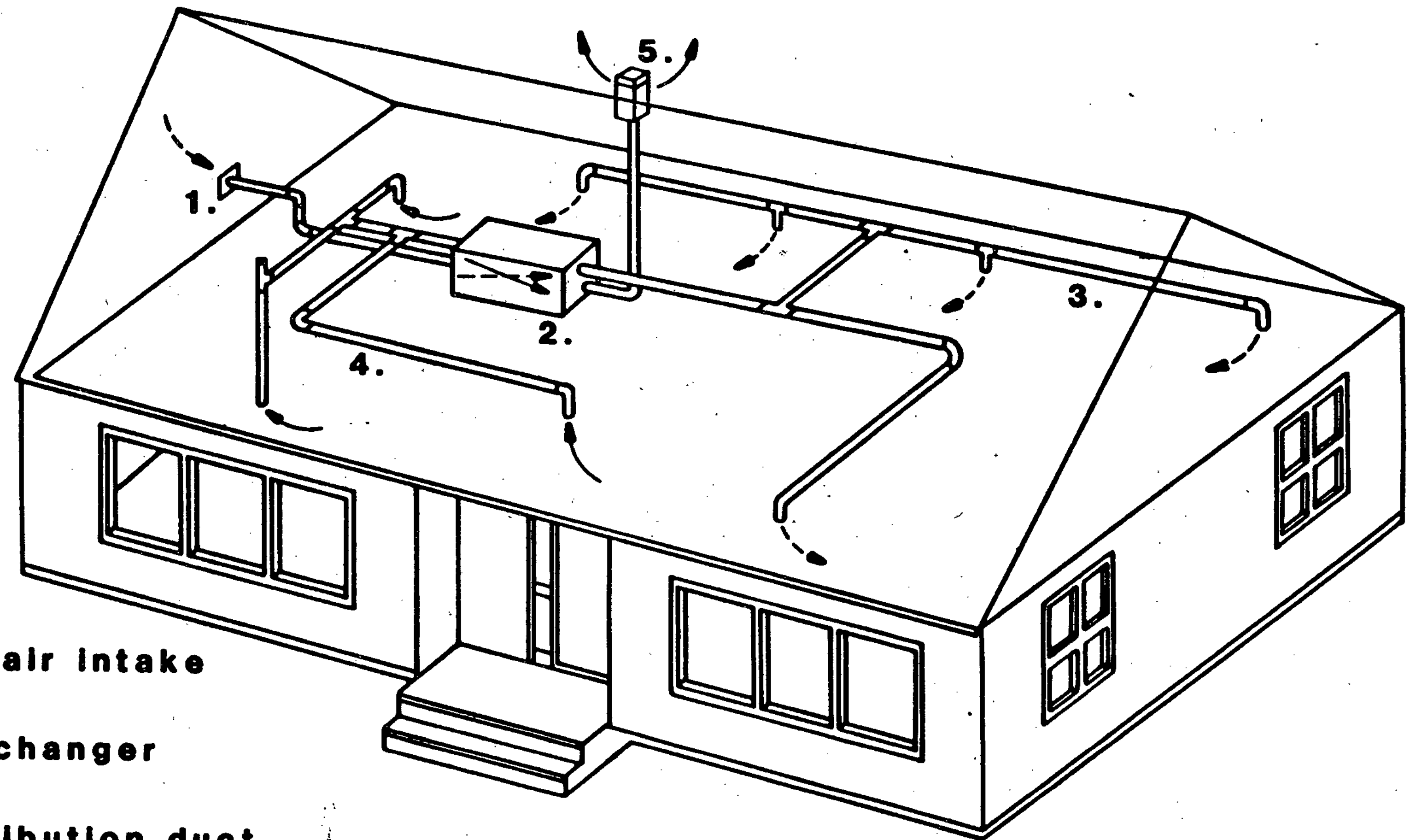


Figure 1. Schematic Diagram of an Air-to-Air Heat Exchanger.



1. Outside air intake

2. Heat exchanger

3. Air distribution duct

4. Exhaust air duct

5. Exhaust air vent

Figure 2. Fully Ducted Installation of Heat Exchanger.

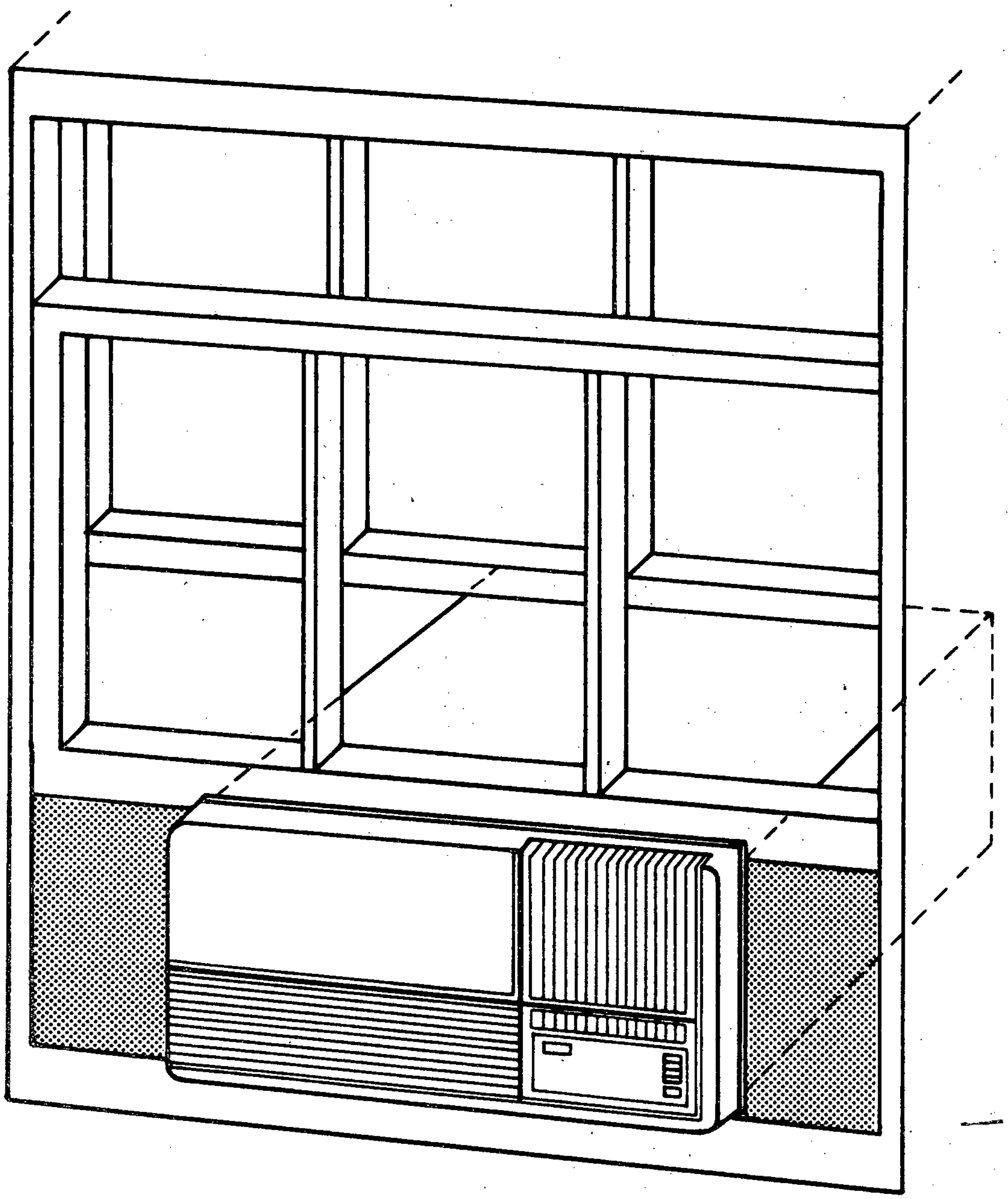


Figure 3. Window Installation of Heat Exchanger.

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