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UNIVERSITY OF CALIFORNIA

ENERGY & ENVIRONMENT DIVISION

To be presented at the Third Air Infiltration Centre (AIC) Conference, "Energy Efficient Domestic Ventilation Systems for Achieving Acceptable Indoor Air Quality," London, UK, September 20-23, 1982

A COMPARISON OF ALTERNATE VENTILATION STRATEGIES

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A COMPARISON OF ALTERNATE VENTILATION STRATEGIES

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SYNOPSIS

In order to decrease the energy costs associated with infiltration, many new residential structures are being built to more stringent air-tightness standards; additionally, many existing structures are being tightened. As structures are made tighter, the total amount of ventilation available for the removal of pollutants generated within the structure is reduced, thus causing a concern that indoor air quality may be suffering. Included among the many solutions to the problem of unacceptable air quality are cleaning the air within the structure (filtration) or replacing the contaminated air with fresh outdoor air (ventilation). Because filtration may not be practical for residential buildings, most of the effort in controlling indoor contaminants has been concentrated on finding the most effective means of increasing ventilation while incurring the smallest energy load. This report examines several ventilation strategies for both impact on the total ventilation and effect on the energy balance of the structure. The strategies we examine include natural systems such as ventilation stacks as well as mechanical systems such as air-to-air heat exchangers or exhaust fans with and without heat pumps.

1. INTRODUCTION

As architects, builders, and home-owners become more energy conscious, increasing attention is paid to the energy burden imposed by ventilation and air infiltration: the goal of decreasing total energy consumption has led to tighter houses and decreased air infiltration. However, as air infiltration is decreased in the name of energy conservation, the possibility of unacceptable indoor air quality grows. If indoor air quality is to be preserved at low infiltration rates, mechanical ventilation may have to be added to increase the total ventilation rate to an acceptable level at a minimum cost.

A comparison of mechanical ventilation systems requires calculating the effects such systems have on the total ventilation and ventilation (energy) load of the structure. In order to make this calculation we will use the single-zone infiltration model developed at Lawrence Berkeley Laboratory, which uses air tightness, structural parameters, and weather to calculate air infiltration; mechanical ventilation can then be combined with the natural ventilation to yield total ventilation. From the natural ventilation, the mechanical ventilation, and the properties of the mechanical system, the total ventilation load can be calculated.

The purpose of this paper is not to make a detailed calculation for a particular building at a particular site,² but rather to make a simple calculation for a typical house (defined in the

natural ventilation section below). That calculation can then be used as a basis to compare the infiltration and load effects of different systems under different conditions (of air tightness, inside/outside temperature difference, and wind speed). Because these calculations are intended to be representative but non-specific, no economic comparison will be made among systems.

We examine three alternative mechanical ventilation strategies for increasing total ventilation. We compare a balanced (mechanical) air-to-air system, an unbalanced (mechanical) exhaust system, and a ventilation stack with completely natural ventilation. The exhaust system will be subdivided into three systems that treat heat loss through the mechanical system differently.

2. NATURAL VENTILATION

Natural ventilation (or air infiltration) is caused by the interaction of the building envelope with weather-induced pressures. The expression, as derived from detailed calculation,³ depends on inside/outside temperature difference and wind speed.

$$Q_0 = L_0 \sqrt{f_s^2 \Delta T + f_w^2 v^2} \quad (1)$$

where:

Q_0 is the ventilation from air infiltration [m^3/s],
 L_0 is the total leakage area [m^2],
 ΔT is the inside/outside temperature difference [K],
 v is the wind speed [m/s],
 f_s is the stack parameter [m/s $K^{1/2}$], and
 f_w is the wind parameter.

The stack and wind parameters depend on the leakage distribution, building height, and shielding and terrain factors. To calculate the infiltration, Q_0 , we use the concept of a reference house;⁴ the reference house has typical values for the structural parameters that go into the stack and wind parameters. Specifically, the house is a single-story, typically shielded house in average terrain; half of the leakage is distributed evenly in the walls and the other half to the floor and ceiling equally. With these assumptions the stack and wind parameters are*:

$$f_s = .144 [m/s-K^{1/2}] \quad (2.1)$$

* Conforming to the conventions of the model $H = 2.5m$, shielding class = terrain class = III, $R = 0.5$, $X = 0$.

$$f_w = .123 \quad (2.2)$$

We shall use these values throughout the remainder of the report.

Even though we have used a simplified reference house, there are still three independent variables (leakage area, temperature difference, and wind speed) that make it quite difficult to represent the results. In Fig. 1a, we have plotted the natural ventilation (i.e., ventilation due to air infiltration) versus inside-outside temperature difference for three envelope leakage areas. The leakage areas were chosen to represent values for a tight house (200 cm²), a loose house (1000 cm²), and one in between (500 cm²). The curves have been calculated assuming a meteorological wind speed of 4 m/s, an average wind speed typical of many areas in North America and Western Europe. The pair of arrows that crosses each curve shows the effect of wind speed on ventilation; the base of each arrow was computed for zero wind speed, and the tip was calculated for a wind speed of 8 m/s -- twice the wind speed used in the curves.

Assuming that all infiltrating air mixes with interior air, the infiltration load can be calculated easily from the infiltration:

$$E_0 = \rho C_p \Delta T Q_0 \quad (3)$$

where:

E_0 is the energy load due to natural ventilation [W],
 ρ is the density of air [1.2 kg/m³], and
 C_p is the heat capacity of air [1023 J/kg-K].

Note that here and in the sections to follow we ignore the effect of the latent load on energy consumption and assume that the ventilation efficiency of all forms (i.e., natural and mechanical) is unity.

Fig. 1b, using the same conventions as Fig. 1a, shows the ventilation load due to natural infiltration as a function of temperature difference for the three leakage areas. Because the infiltration in Fig. 1a is a monotonically increasing function of temperature, the loads increase faster than would be shown by a straight line.

3. VENTILATION REQUIREMENTS

The purpose of mechanical ventilation is to assure that indoor pollutants are adequately diluted by fresh air; the exact amount of fresh air, however, is not generally agreed upon. Although the necessary ventilation rate will depend strongly upon the types and source strengths of the pollutants, for the purpose of this report

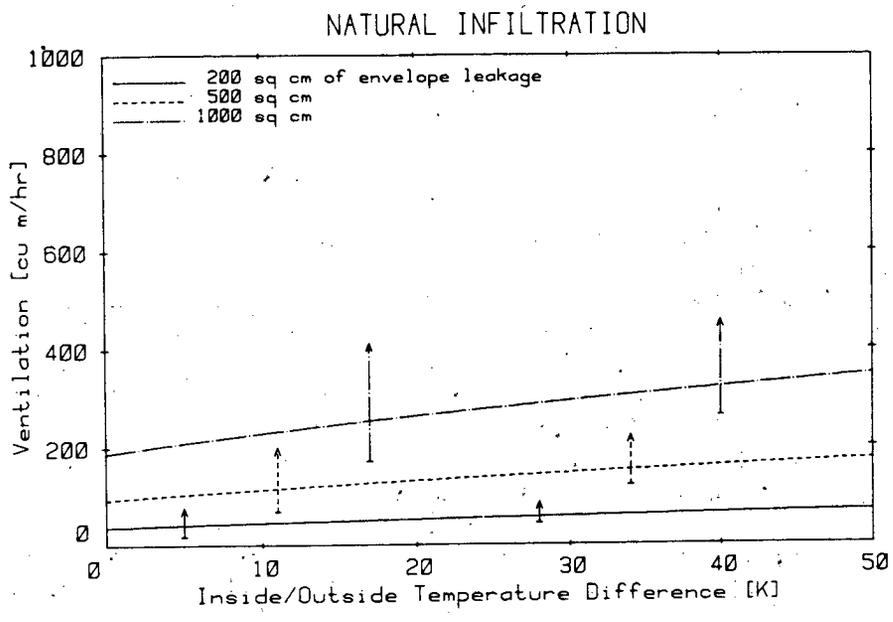


Fig. 1a. Ventilation due to air infiltration.

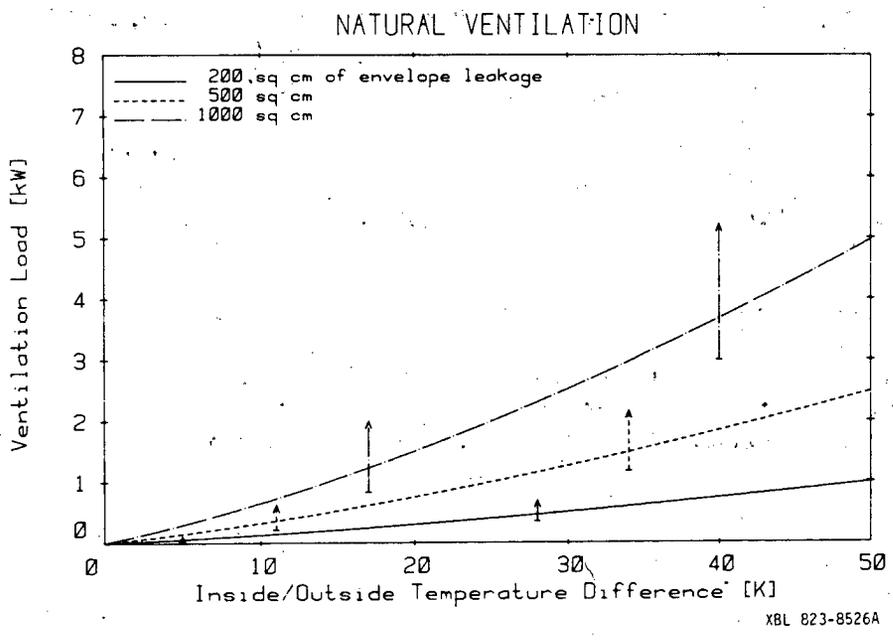


Fig. 1b. Ventilation load due to air infiltration.

we assume that $200 \text{ m}^3/\text{hr}$ is the amount of total ventilation necessary for adequate indoor air quality. This could correspond, for example, to one air change per hour in a 200 m^3 house or one-half an air change in a 400 m^3 house. Referring again to Fig. 1a, we see that under average weather conditions only the leakiest envelope meets our ventilation requirement with natural ventilation. The tighter two envelopes (and the leaky envelope under very calm conditions) require additional ventilation to meet the requirement. We must, therefore, consider mechanical ventilation systems.

4. BALANCED AIR-TO-AIR SYSTEMS

Mechanical ventilation systems employing air-to-air heat exchangers have recently received consideration as a possible systems for the control of indoor pollutant levels in residences.² In these systems there are two airstreams, one supply and one exhaust, which come into thermal contact but do not mix. Heat is transferred from one airstream to the other, reducing the increased load from additional ventilation. Because the system is assumed to be balanced (i.e., exhaust flow and supply flow are equal), the ideal heat exchanger has no effect on the internal (house) pressure. The system, however, does affect air infiltration because it contributes to the leakage area of the structure.

We can calculate the total ventilation from air infiltration, the fan flow through the exchanger, and the extra leakage area:

$$Q_1 = Q_{f1} + \left(1 + \frac{L_1}{L_0}\right) Q_0 \quad (4)$$

where:

- Q_1 is the total ventilation for the air-to-air system [m^3/s],
- Q_{f1} is the air flow from each fan [m^3/s], and
- L_1 is the additional leakage area from the exchanger [m^2].

Note that the presence of a non-zero L_1 term will effectively "unbalance" the system. This simplification ignores any non-linear interactions that may couple fan performance to external pressure. For our calculations we have sized the system as follows:

$$Q_{f1} = 150 \frac{\text{m}^3}{\text{hr}} \quad (5.1)$$

$$L_1 = 50 \text{ cm}^2 \quad (5.2)$$

The assignment of additional leakage area was based on a restricted set of calculations and, in any specific situation, could vary by a factor of three.

Figure 2a is a plot of the ventilation for this system using the same weather and leakage values as in Fig. 1a and 1b. For all temperatures and leakage areas the total ventilation rate meets the target of $200 \text{ m}^3/\text{hr}$, and may provide as much as twice that for the leaky envelope or extreme weather.

The energy load due to the combined effect of the air-to-air system and natural infiltration can be calculated from the system parameters:

$$E_1 = \rho C_p \Delta T (Q_1 - \epsilon_1 Q_{f1}) \quad (6)$$

where:

E_1 is the total ventilation load [W], and

ϵ_1 is the effective heat-recovery efficiency.

The effective heat-recovery efficiency is the fraction of the sensible heat ($\rho C_p \Delta T Q_{f1}$) that is recovered by the heat exchanger. We have assumed an efficiency of 70%.

$$\epsilon_1 = 0.70 \quad (7)$$

When outdoor temperatures are substantially below freezing, the performance of the heat exchanger may be degraded by ice forming in the exhaust stream; freezing temperatures necessitate periodic defrosting of the heat exchanger. Although the overall efficiency of this system will be reduced in very cold climates, we have not specifically included this in our analysis.

The total ventilation load will be higher with an air-to-air system than without one. Figure 2b shows the increase in ventilation load due to the heat exchanger*. Because this system is balanced, there is no effect on the internal pressure of the envelope, and the increase in ventilation load is independent of leakage area -- hence there is one curve instead of three. Furthermore, the size of the wind arrows indicates the magnitude of the effect that additional leakage area has on the load. Because the wind arrows are quite small, our assumptions about the size and interaction of

* Conventionally, the total ventilation load is compared between two systems which have the same total ventilation (i.e., a natural system and one with a heat exchanger and a much smaller amount of infiltration), implying a different envelope leakage for the two cases. We are comparing a structure without a heat exchanger to exactly the same structure with a heat exchanger.

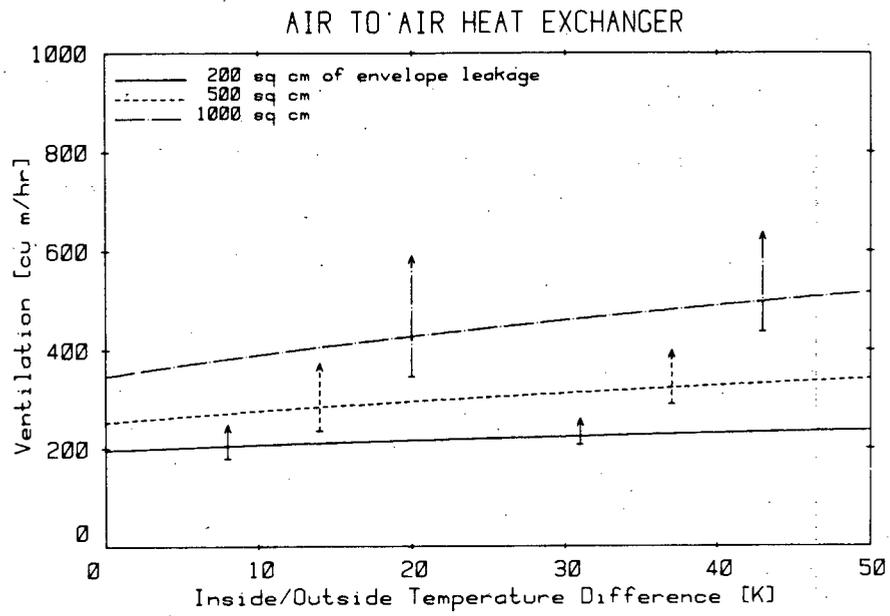


Fig. 2a. Ventilation due to a balanced heat exchanger.

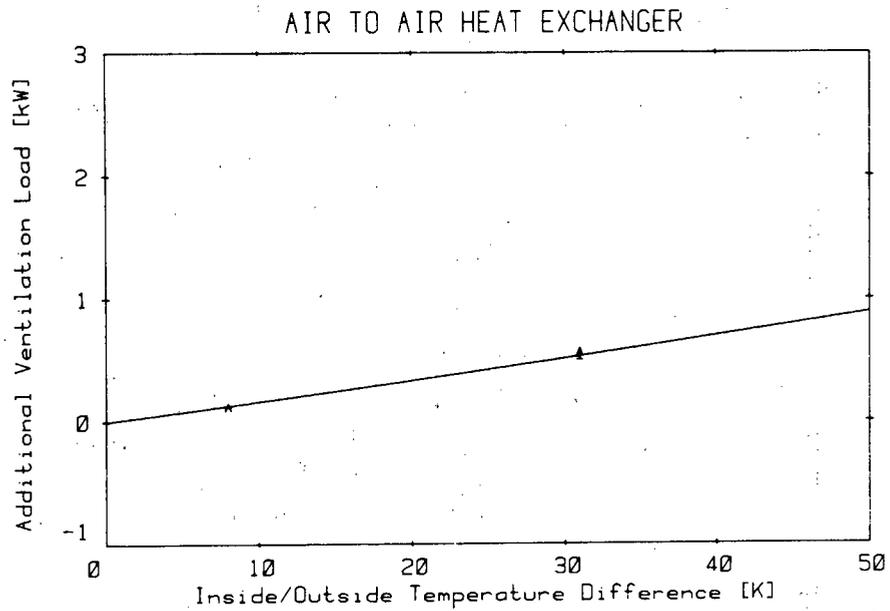


Fig. 2b. Increased load due to a balanced heat exchanger.

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the additional leakage area are not critical.

The above discussion assumes that the weather-induced pressures on the heat exchanger operate in the same direction. This would be the case if the supply and exhaust ducts were near each other as, for example, in a window unit. If the weather-induced pressures act in opposite directions (e.g., a high supply and a low exhaust on opposite sides of the building), the amount of heat recovered through the heat exchanger will be increased and the total load mitigated:

$$E'_1 = \rho C_p \Delta T \left[Q_1 - \epsilon_1 \left(Q_{f1} + \frac{L_1}{L_0} Q_0 \right) \right] \quad (8)$$

We have not, however, used this expression in our calculations.

5. EXHAUST SYSTEMS

In a balanced heat exchanger, supply and exhaust flows are matched and there is no effect on internal pressure. In an exhaust system there is no (mechanically provided) supply air; rather all air exhausted from the mechanical system must enter through the building envelope. Thus, the internal pressure will be significantly reduced as will the naturally driven part of the ventilation; the infiltration through the envelope will be increased and the exfiltration will be decreased. The overall effect is an increase in total infiltration, with a smaller absolute quantity of air crossing through the envelope. We can calculate the total infiltration rate using the rule of superposition in quadrature:

$$Q_3 = \sqrt{Q_0^2 + Q_{f3}^2} \quad (9)$$

where:

Q_3 is the total ventilation for the exhaust system [m³/s], and
 Q_{f3} is the flow through the exhaust fan [m³/s].

For our exhaust systems we have used a flow equal to our target infiltration.

$$Q_{f3} = 200 \frac{\text{m}^3}{\text{hr}} \quad (10)$$

Exhaust systems may be especially susceptible to incomplete mixing and lower ventilation efficiencies; therefore, the value of Q_{f3} should be taken as the effective flow (i.e., the product of the ventilation efficiency and the actual flow rate) for calculating both infiltration and load.

Because exhaust systems affect the internal pressure, the total ventilation will be greater than the exhaust ventilation only when the natural air infiltration is large. Figure 3a shows the total ventilation for all exhaust systems.

5.1 No Heat Recuperation

We have elected to treat the heat loss of the exhaust systems in three different ways, corresponding to three types of exhaust ventilation systems. The first of these systems is the simplest; that is, a simple system with no heat recovery whatsoever. In this case the total ventilation load is derived from the total ventilation.

$$E_{3a} = \rho C_p \Delta T Q_3 \quad (11)$$

where:

E_{3a} is the total ventilation load without recovery [W].

Figure 3b shows the additional ventilation load imposed by an exhaust system without heat recuperation. Note that the arrows indicating wind dependence point down instead of up. This indicates that the additional load imposed by the exhaust system at high wind speed is smaller than the additional load imposed by the same system at lower wind speeds. Similarly, the curves for the leaky envelope are below those for the tight envelope. This behavior is a direct consequence of the fact that exhaust systems affect the internal pressure, and is common to all the exhaust systems.

5.2 Heat Recovery

Another possible exhaust system is one in which there is some heat recovery in the exhaust stream (e.g., a regenerative heat exchanger). Systems of this type would have a single stream of air that passes through a heat storage medium in alternating directions, first storing heat in the medium during the exhaust phase and recovering it during the supply phase. It may even be possible to use a building envelope as the heat-exchange medium.⁵ In any case the total ventilation load will be reduced from the value calculated above.

$$E_{3b} = \rho C_p \Delta T (Q_3 - \epsilon_3 Q_{f3}) \quad (12)$$

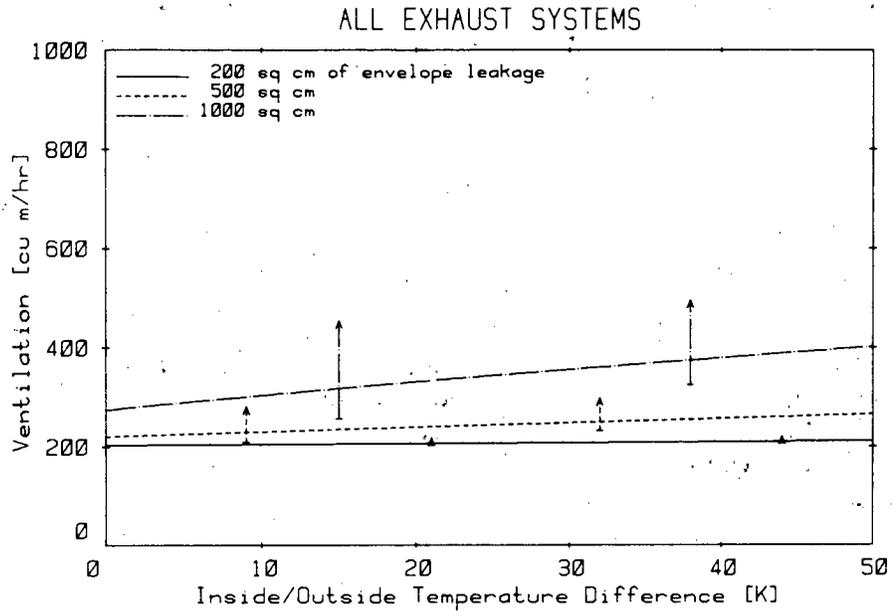


Fig. 3a. Ventilation from exhaust systems.

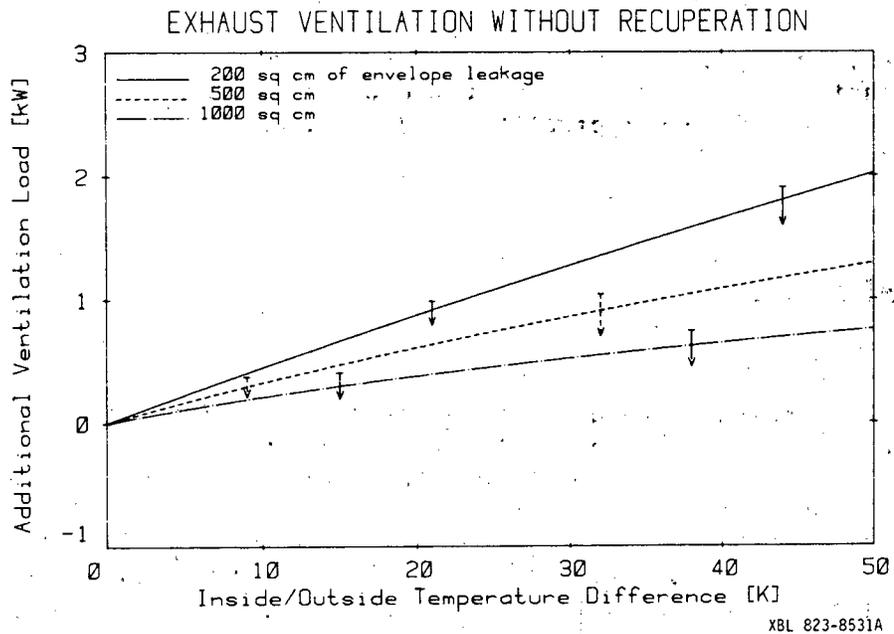


Fig. 3b. Additional ventilation load from exhaust systems--no recuperation.

where:

ϵ_3 is the effective efficiency of the heat recovery.

Because it is either storing heat (as its temperature increases) or releasing heat (as its temperature decreases), the overall efficiency of such a regenerative heat exchanger cannot be as high as for the air-to-air system in which there is a constant transfer of heat from one stream to the other. Accordingly, we have chosen a reduced value for the effective heat-transfer efficiency of this system.

$$\epsilon_3 = 0.30 \quad (13)$$

Figure 3c shows the additional load induced by an exhaust system with heat recovery. This figure demonstrates the same sort of upside-down behavior seen in the previous plot, but it is interesting to note that some parts of these curves extend below the origin; indicating that in some cases (extreme weather/leaky envelopes) an exhaust system with heat recovery can decrease the total ventilation load.

5.3 Heat Pump

Heat recovery by a heat-exchanger is limited by the difference in temperature across the heat-exchange medium; an alternate method of heat recovery is to use a heat pump to pump the heat out of the exhaust stream. In some existing systems such heat pumps are used to preheat the domestic hot water, but we will assume that the heat is pumped in the conditioned air space. Because the air in the exhaust stream will always be about the same temperature (internal), the effect of the heat pump will be to shift the ventilation load down by a fixed amount.

$$E_{3c} = E_{3a} - \rho C_p \Delta T_3 Q_{f3} \quad (14)$$

where:

ΔT_3 is the temperature drop across the exhaust stream [K].

The choice of temperature difference is essentially arbitrary and will probably depend on the systems available at the time. We have, therefore, chosen one which we believe is representative of what can be expected in consideration of latent heat and freezing issues.

$$\Delta T_3 = 20 \text{ K} \quad (15)$$

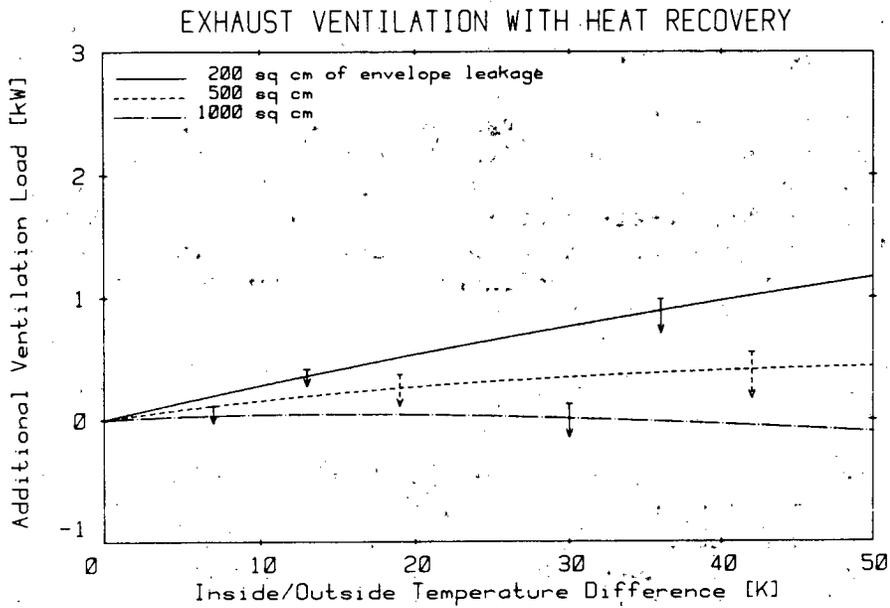


Fig. 3c. Additional ventilation load from exhaust systems--with regeneration.

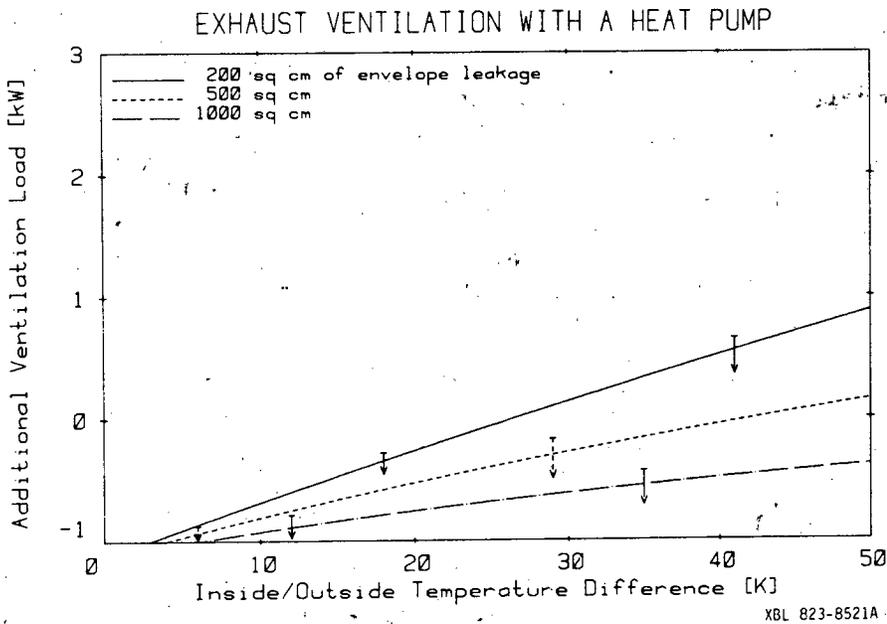


Fig. 3d. Additional ventilation load from exhaust systems--with heat pump.

Figure 3d shows the additional load induced by an exhaust system with heat-pump recuperation, which is the plot for exhaust without recuperation shifted down by 1150 watts. For most climatic conditions this system reduces the total ventilation load; and for very mild conditions the total ventilation load may become negative.

6. VENTILATION STACK

Once very simple approach to increasing total ventilation, which has been used in large structures but rarely on individual residences, is that of a ventilation stack. A ventilation stack is a chimney that projects into the free air above the structure and increases ventilation by its own stack and wind effect.⁶

$$Q_4 = \sqrt{Q_0^2 + Q_{f4}^2} \quad (16)$$

where:

Q_4 is the ventilation from a ventilation stack [m^3/s],
and

Q_{f4} is the flow induced by the ventilation stack [m^3/s].

The flow induced by the ventilation stack has an form analogous to that of air infiltration:

$$Q_{f4} = L_4 \sqrt{f_{s4}^2 \Delta T + f_{w4}^2 v^2} \quad (17)$$

where:

L_4 is the leakage area of the stack [m^2],

f_{s4} is the stack parameter of the ventilator [$m/s-\sqrt{K}$],
and

f_{w4} is the wind parameter of the stack.

If we assume the height of the stack to be six meters and the pressure coefficient at the top of the stack to be 0.95, we can calculate the rest of the parameters:

$$L_4 = 200 \text{ cm}^2 \quad (18.1)$$

$$f_{s4} = 0.45 \text{ m/s-K}^{1/2} \quad (18.2)$$

$$f_{w4} = 0.8 \quad (18.3)$$

Figure 4a shows the total ventilation with a stack ventilator of the above properties. For mild conditions and a tighter envelope the total ventilation is below our target level. Because this ventilation system is completely passive, there will be no ventilation if there is no wind or temperature difference. However, for most weather conditions of interest there will be an acceptable amount of ventilation from a stack ventilator. Because there is no heat recovery, the total ventilation load can be simply calculated:

$$E_4 = \rho C_p \Delta T Q_4 \quad (19)$$

where:

E_4 is the total ventilation load for a stack ventilator [W].

The additional load, as shown in Fig. 4b, shows some new features that previous plots did not. Like the exhaust plots, the curves for the tighter envelopes show larger increased ventilation loads, but like the balanced systems the arrows, which indicate the effect of increased wind speed, point upwards. Furthermore, some of the arrows do not cross their corresponding curves, indicating that the additional ventilation load may not be a monotonic function of wind speed. This occurs because the air infiltration through both the envelope and the ventilation stack are influenced by wind and temperature difference, but the relative weightings are much different.

7. COMPARISON

So far we have shown the effect that each system would have on ventilation and ventilation load for different leakage areas. We must now compare the different systems with each other. Although an economic criterion will ultimately be used to make a decision on which system to use, a comparison of effects on total ventilation and ventilation load is instructive.

Figure 5a compares the effects on total ventilation that these systems would have on a structure having 200 cm² of leakage area. The natural ventilation alone is far below the target of 200 m³/hr we are using, and would quite likely cause problems with indoor air quality. The other three systems yield adequate ventilation, with the possible exception of the stack ventilator (which does not meet the target for mild conditions). Because the stack ventilator is completely dependent on natural driving conditions, it is not surprising that it is the most weather-dependent of the mechanical systems. Conversely, because the exhaust systems have a pronounced effect on internal pressure, they are the least weather-dependent; the balanced air-to-air system is between these

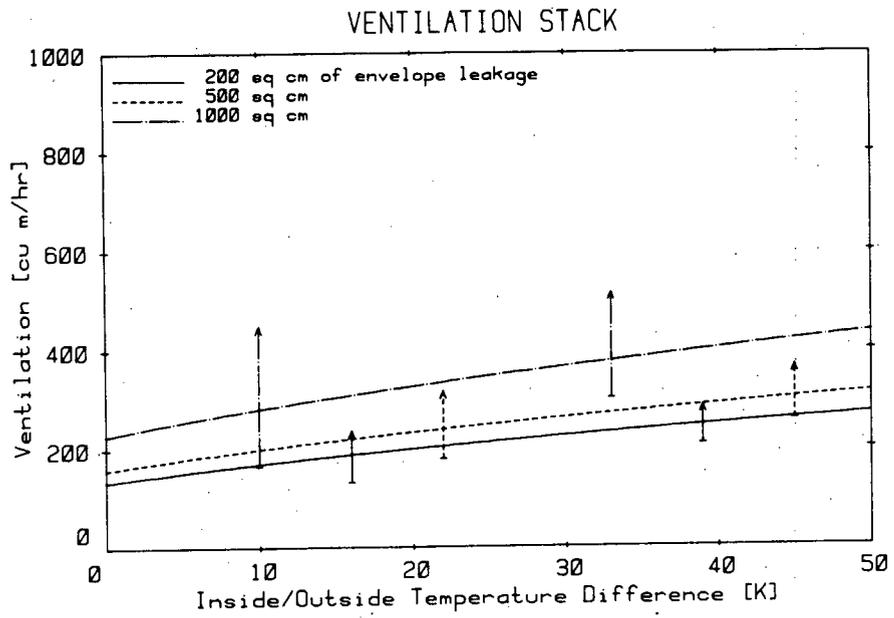


Fig. 4a. Ventilation from a ventilation stack.

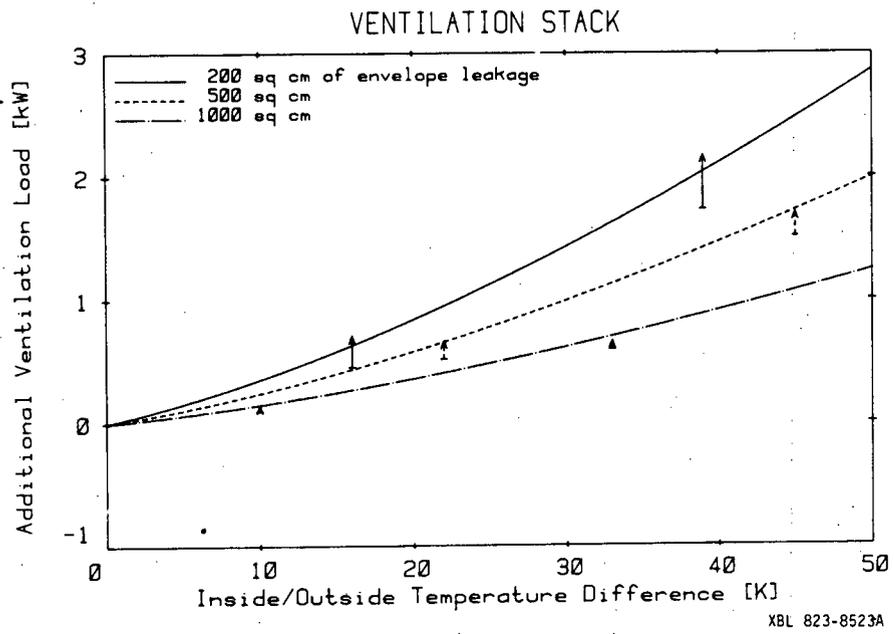


Fig. 4b. Additional load from a ventilation stack.

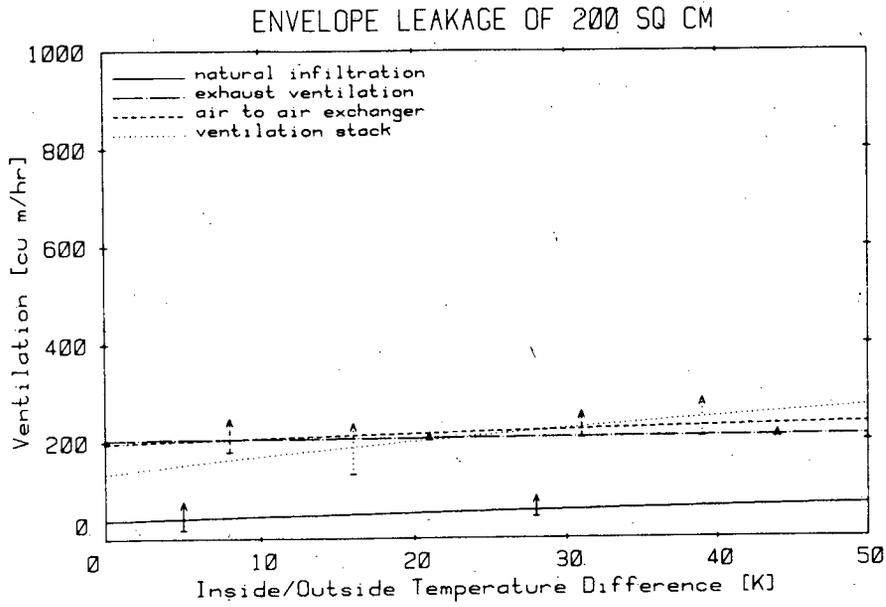


Fig. 5a. Ventilation for envelope leakage area of 200 cm².

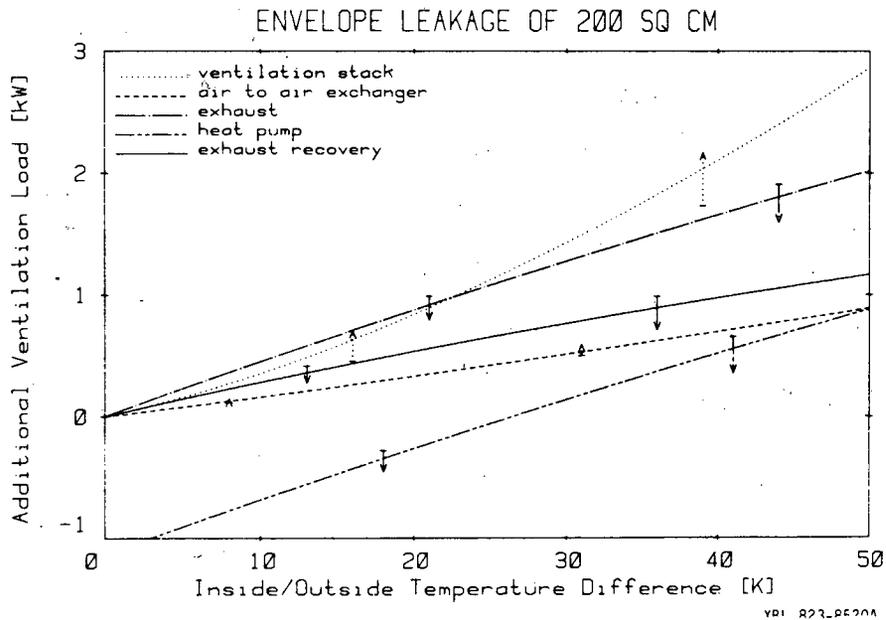


Fig. 5b. Additional ventilation load for envelope leakage area of 200 cm².

two extremes.

Figure 5b again compares the effectiveness of these systems on a structure having 200 cm^2 of leakage area, but compares additional load rather than total ventilation. Whereas in Fig. 5a the results for all systems were bunched rather closely, the differences among them are quite noticeable in Fig. 5b. The order of decreasing load is approximately: ventilation stack, exhaust without recuperation, exhaust with heat exchange, air-to-air system, and exhaust with a heat pump. Unsurprisingly, this is also the probable order of increasing cost. Because the stack ventilator is only one of the systems to show positive curvature with temperature, it will perform relatively poorly under extreme temperature differences, making it unattractive for cold climates. The stack ventilator is the most sensitive to temperature differences; the air-to-air system is the least. Under moderate weather conditions the heat-pump system is a net producer of energy; but under more extreme conditions the net load of the heat-pump system approaches that of the balanced air-to-air system.

Because envelope leakage has an important effect on the performance of each system, we will compare the systems again -- this time at 500 cm^2 of leakage area. Figure 6a shows total ventilation for all the systems under these conditions. Compared to Fig. 5a, all the curves are shifted upwards somewhat, especially the one for natural ventilation, although it is still inadequate when used alone. While the exhaust systems and ventilation stack have moved little, the balanced system has moved up as much as natural ventilation, causing the air-to-air system to deliver a bit more than the target.

The additional loads in Fig. 6b have all decreased (with the exception of the balanced system, which is unchanged). There has been some change in the order, but the general trend is the same. The air-to-air system has become relatively less attractive as the envelope leakage area has increased.

Figure 7 shows the pair of ventilation plots for a loose envelope of 1000 cm^2 of leakage area. The target ventilation is met almost all of the time (See Fig. 7a); only for the calmest conditions will natural ventilation be insufficient. Although it is unlikely that a mechanical ventilation system would be installed in such a leaky structure, it is useful to consider the effect it would have on ventilation loads.

From Fig. 7b we see that the additional ventilation load again has decreased for all the unbalanced systems. The balanced system has become about as effective as the exhaust fan without recuperation, and the ventilation stack is only marginally worse. The exhaust system with heat recovery deviates only slightly from zero -- implying that it creates no additional ventilation load; furthermore, the heat-pump system is a net energy producer under any weather conditions. These facts suggest that putting in an exhaust system with heat recuperation may be effective even in a

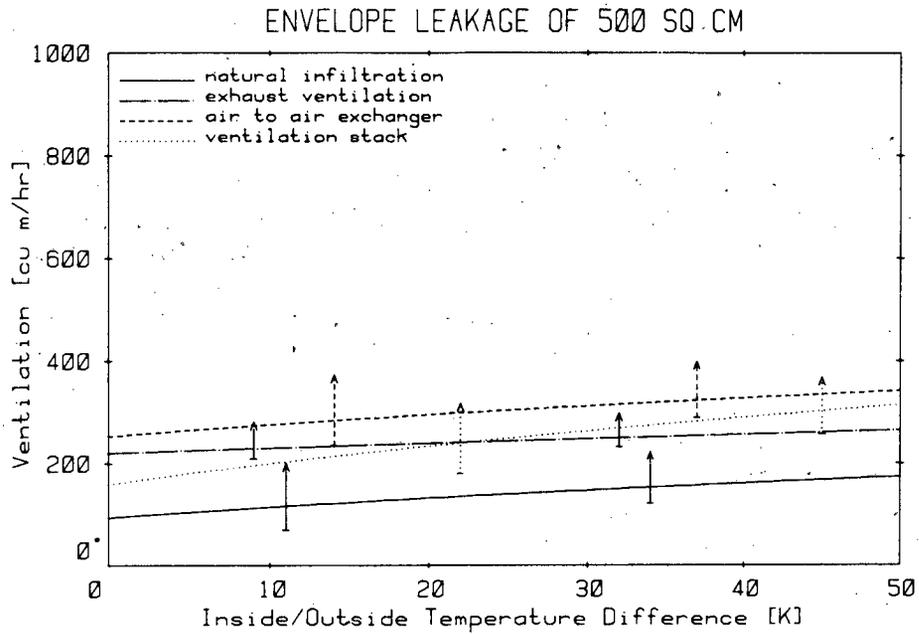
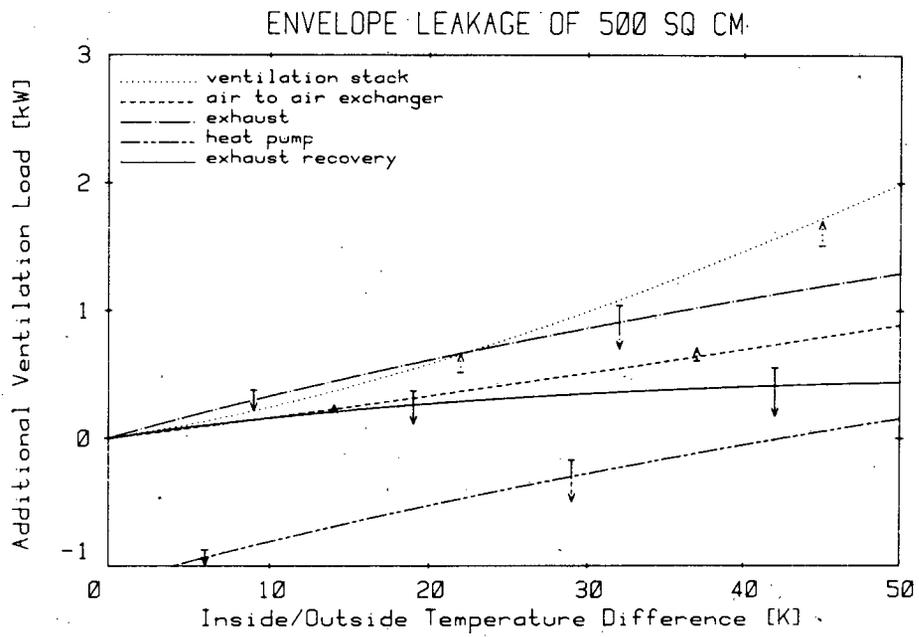


Fig. 6a. Ventilation for envelope leakage area of 500 cm².



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Fig. 6b. Additional ventilation load for envelope leakage area of 500 cm².

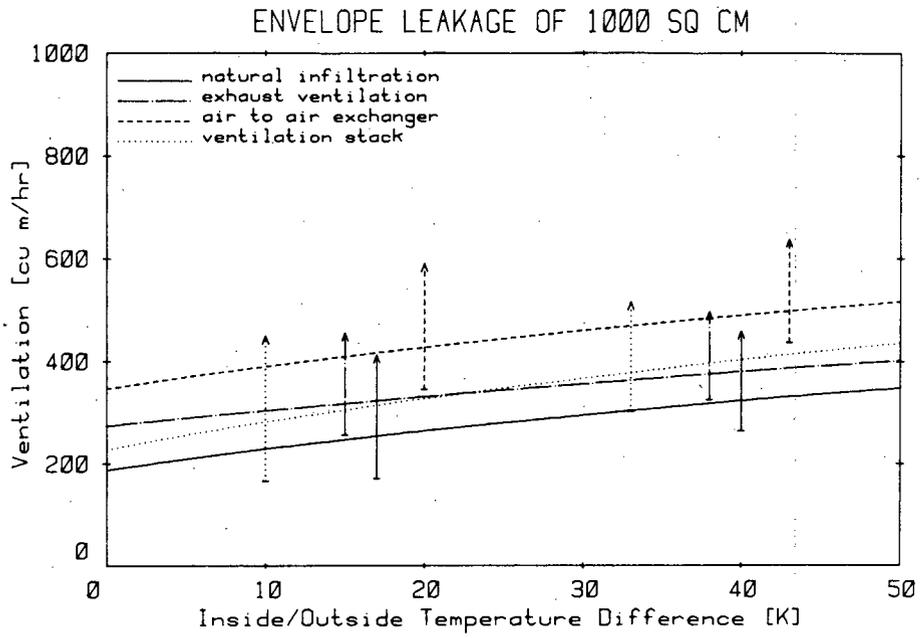


Fig. 7a. Ventilation for envelope leakage area of 1000 cm².

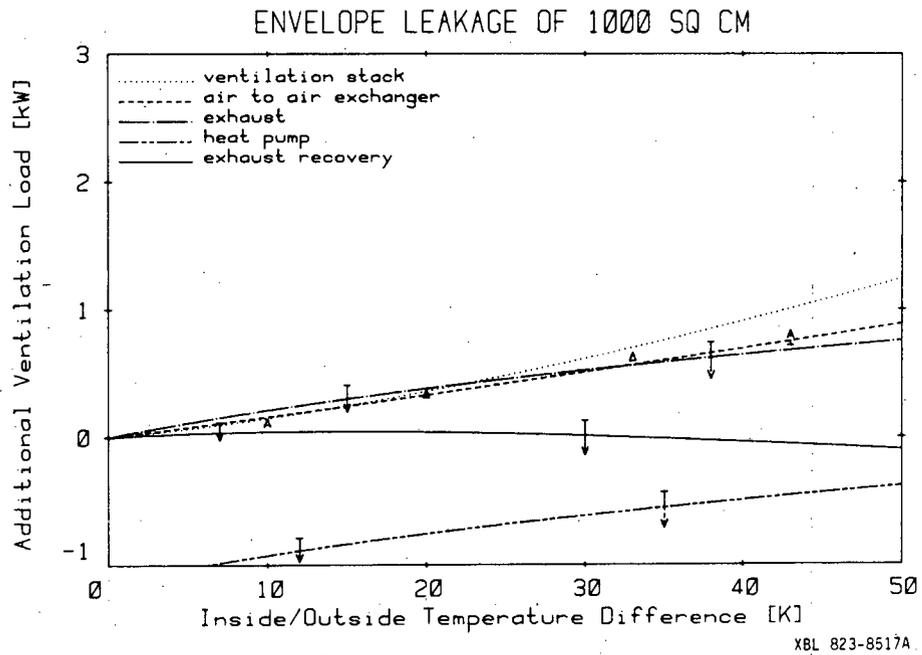


Fig. 7b. Additional ventilation load for envelope leakage area of 1000 cm².

leaky structure.

Because all of these systems have different construction costs, maintenance costs, and operation characteristics it is difficult to compare them in general. For example, we can look at the power requirements of each system. Natural ventilation and a stack ventilator are completely passive and require no power; the other systems all require air-handling fans that will consume between 50 and 200 watts of power; additionally, the heat pump will use an extra 400 to 600 watts to pump the heat back into the house. The disposition of this energy will be different also. In the balanced system and the regenerative system about half the power will be recovered; in the pure exhaust system none of it will be recovered; and in the heat-pump system almost all of it will be recovered. The effect of this recovered heat on the energy costs of the house will depend on such factors as climate and type of fuel and heating system used.

8. SUMMARY

The effect of leakage area on the total ventilation provided by various systems can be quite different. The natural ventilation will be proportional to the leakage area, and the balanced air-to-air system will increase the ventilation by about the same amount; the unbalanced systems, however, are far less sensitive to air infiltration than is the balanced one. Thus, as can be seen in Fig. 8, the natural ventilation and air-to-air system are strongly influenced by an increase in leakage area, but the exhaust systems are not.

The influence of leakage area on different systems becomes more apparent when one looks at the additional loads, as shown in Fig. 9. The additional load for the balanced system is independent of leakage area, but the additional load for any of the unbalanced systems decreases with increasing leakage area, thus favoring the balanced system in tighter structures and the unbalanced systems in looser structures.

In this report we have presented a comparison of the ventilation and load aspects of a few mechanical ventilation systems as a function of weather and leakage area for a typical detached structure. The results do not purport to indicate the best ventilation system for all cases; rather, we have shown that which system is best, and of course whether one needs a ventilation system at all, is critically dependent on the envelope leakage. Furthermore, the choice of an economical optimum will depend strongly on climate, system performance, existing HVAC plant, fuel type, and costs. Previous studies for much more restricted conditions have borne this out.

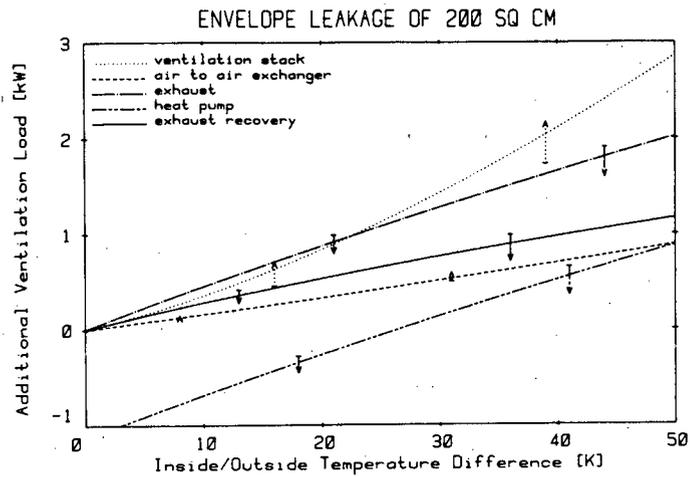


Fig. 8a. Summary of total ventilation for 200 cm² of leakage area.

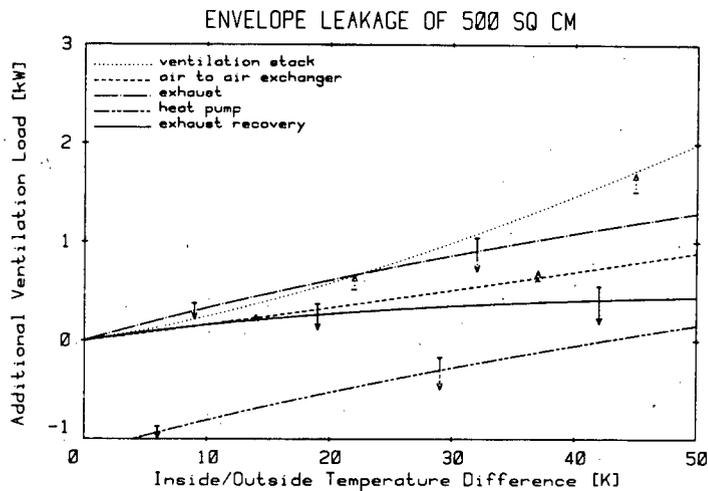


Fig. 8b. Summary of total ventilation for 500 cm² of leakage area.

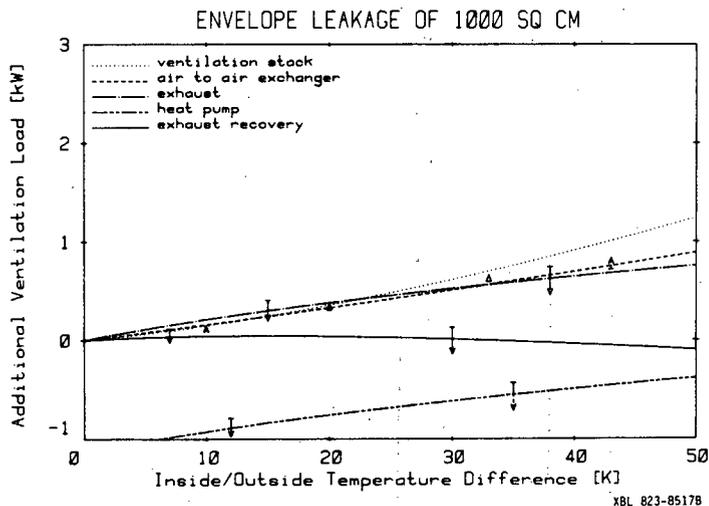


Fig. 8c. Summary of total ventilation for 1000 cm² leakage area.

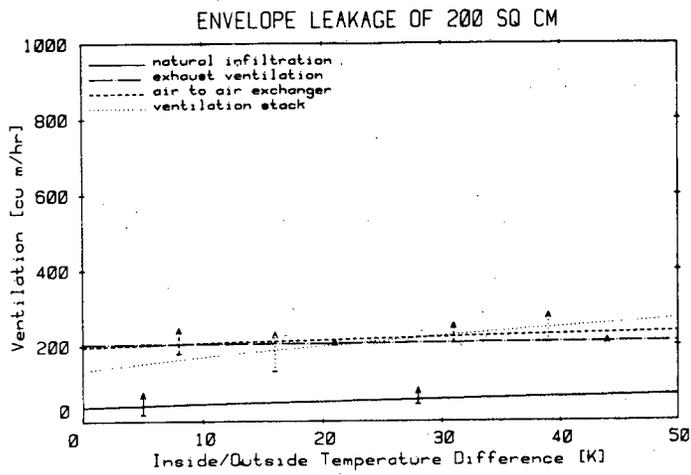


Fig. 9a. Summary of additional load for 200 cm² of leakage area.

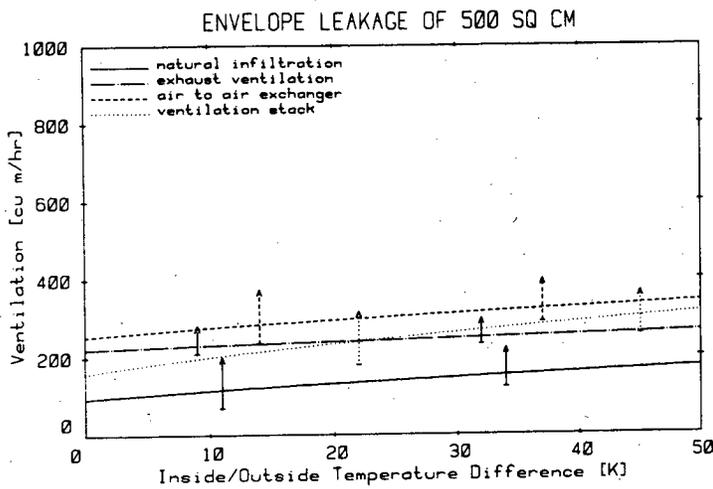


Fig. 9b. Summary of additional load for 500 cm² of leakage area.

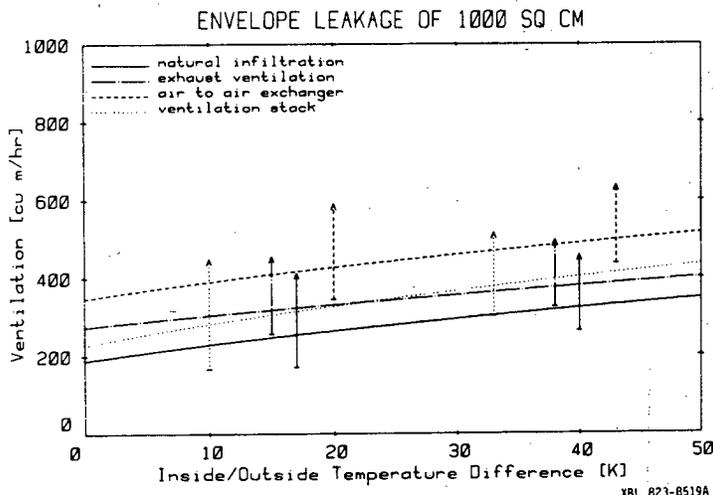


Fig. 9c. Summary of additional load for 1000 cm² of leakage area.

10. ACKNOWLEDGEMENT

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