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# Comparison of Heating and Cooling Energy Consumption by HVAC System with Mixing and Displacement Air Distribution for a Restaurant Dining Area in Different Climates

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## ABSTRACT

*Different ventilation strategies to improve indoor air quality and to reduce HVAC system operating costs in a restaurant with nonsmoking and smoking areas and a bar are discussed in this paper. A generic sitting-type restaurant is used for the analysis. Prototype designs for the restaurant chain with more than 200 restaurants in different U.S. climates were analyzed to collect the information on building envelope, dining area size, heat and contaminant sources and loads, occupancy rates, and current design practices.*

*Four constant air volume HVAC systems with a constant and variable (demand-based) outdoor airflow rate, with a mixing and displacement air distribution, were compared in five representative U.S. climates: cold (Minneapolis, Minn.); maritime (Seattle, Wash.); moderate (Albuquerque, N.Mex.); hot-dry (Phoenix, Ariz.); and hot-humid (Miami, Fla.).*

*For all four compared cases and climatic conditions, heating and cooling consumption by the HVAC system throughout the year-round operation was calculated and operation costs were compared. The analysis shows:*

- *Displacement air distribution allows for better indoor air quality in the breathing zone at the same outdoor air supply airflow rate due to contaminant stratification along the room height.*
- *The increase in outdoor air supply during the peak hours in Miami and Albuquerque results in an increase of both heating and cooling energy consumption. In other climates, the increase in outdoor air supply results in reduced cooling energy consumption.*
- *For the Phoenix, Minneapolis, and Seattle locations, the HVAC system operation with a variable outdoor air supply allows for a decrease in cooling consumption up to 50% and, in some cases, eliminates the use of refrigeration machines.*
- *The effect of temperature stratification on HVAC system parameters is the same for all locations; displacement ventilation systems result in decreased cooling energy consumption but increased heating consumption.*

## INTRODUCTION

During the past twenty years, displacement ventilation has been common in industrial facilities in Scandinavia. More recently, it has been used to ventilate offices, restaurants, bars, department stores, lounges, and other commercial spaces. In 1989, it was estimated that displacement ventilation accounted for 50% of the Scandinavian market share in industrial applications and 25% in office applications. Displacement ventilation is becoming more popular in Germany, United Kingdom, Switzerland, and some other central European countries.

Due to its popularity, displacement ventilation was specified for a wide variety of applications in Europe, whether or not it had advantages, and provided better indoor air quality compared to mixing systems.

In the U.S., the consulting engineering community, which is the largest group making systems application decisions in new and remodeled buildings, has little awareness of displacement ventilation. Outside of special situations (e.g., research laboratories, clean rooms, etc.) room air heating/cooling is seen as very straightforward, using tried and true technologies conforming to design basics and guidelines that have been in place for at least 30 to 40 years. This is particularly true in the commercial building market, where low cost and price competitiveness are at the top of the list of design requirements. Dilution mixing type ventilation has long been the

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standard, and only client demand, special building needs, or clear economies will likely spur today's engineers to experiment with the displacement ventilation approach.

The misuse of displacement ventilation can be reduced by applying current knowledge about these systems and by an economic analysis of the system. Some of the general limitations of displacement ventilation are listed below.

- It is not for applications where contaminants do not have a heat source nearby to create thermal plumes with enough airflow capacity to carry the contaminants to the upper zone of the room.
- The supply air cannot be heated above the desired room air temperature, requiring a separate heating system with displacement ventilation.
- High cooling capacity cannot be achieved due to comfort criteria limiting the maximum vertical temperature gradient in the occupied zone and abnormal air velocity near the floor level. Practical experience collected by different manufacturers indicates that the cooling load through the air supply typically should not exceed 40 W/m$^2$ (12.7 Btu/[h·ft$^2$]) for commercial spaces when regular displacement ventilation air diffusers are used and can be increased to 60 W/m$^2$ (19 Btu/[h·ft$^2$]) with induction-type air diffusers. Other cooling systems (e.g., cooling ceilings) may be needed.
- It works best in rooms with a height of 3 m (10 ft) or more.

To allow for a broader application of displacement ventilation systems in the U.S. market, currently available information on displacement ventilation systems has been gathered from numerous sources and included in a comprehensive analysis (Zhivov et al. 1995). This information, together with the results of original studies conducted by an international team (Zhivov et al. 1997a), resulted in a user-friendly design procedure (Zhivov et al. 1997b). This paper presents the economic comparison of displacement and mixing type air distribution, which was a part of the above-mentioned research project.

## PROTOTYPE RESTAURANT DESIGN

A generic sitting-type restaurant with nonsmoking and smoking areas and a bar is used for the economic comparison of this study. Prototype designs for a restaurant chain with more than 200 restaurants in different U.S. climates were analyzed to collect information on building envelope, dining area size, heat and contaminant sources and loads, occupancy rates, and current design practices. This analysis resulted in the following data for the generic restaurant:

| | |
|---|---|
| Restaurant dining area: | $A_r = 334$ m$^2$ (3600 ft$^2$) |
| Room height: | $H_r = 3.5$ m (11.5 ft) |
| Number of seats: | 270 total; 170 in the nonsmoking zone, 80 in the smoking zone, 20 in the bar. |

Occupancy rate depends upon the time of the day:

- 1 a.m. through 7 a.m.—restaurant is closed.
- 7 a.m. through 11:30 a.m.—people are only in the kitchen area (system is turned on to compensate for air exhausted by kitchen hoods).
- 11:30 a.m. through 2 p.m.—65% of the smoking and nonsmoking areas are occupied, no people in the bar.
- 2 p.m. through 5 p.m.—5% of the smoking and nonsmoking areas are occupied.
- 5 p.m. through 9 p.m.—85% of the smoking and nonsmoking areas and the bar are occupied.
- 9 p.m. through 1 a.m.—10% of the smoking and nonsmoking areas and 100% of the bar area are occupied.

The minimum required outside airflow rate into the dining area is listed in Table 1.

Internal heat load:

- From 1 a.m. through 7 a.m.— no internal heat loads.
- From 7 a.m. through 11:30 a.m.—the internal load is 10.76 W/m$^2$ (3.4 Btu/[h·ft$^2$]) from lights.
- From 11:30 a.m. through 1 a.m.—the heat load from lights and miscellaneous heat sources is 64.6 W/m$^2$ (20.5 Btu/[h·ft$^2$]).

The external heat-load calculation is based on the building characteristics listed in Table 2.

### Representative Climates

Based on suggestions from the National Climatic Data Center (U.S. DOC), the following cities were selected for the analysis of the prototype restaurant to represent different U.S. climates:

| | |
|---|---|
| Minneapolis, Minn. | = cold climate |
| Seattle, Wash. | = maritime climate |
| Albuquerque, N. Mex | = moderate climate |
| Phoenix, Ariz. | = hot-dry climate |
| Miami, Fla. | = hot-humid climate |

### System Load Calculation

The Building Loads Analysis and System Thermodynamics (BLAST) computer program, developed by the U.S. Construction Engineering Research Laboratory, was used to calculate the hourly loads based on the above input data and the hourly bin weather data. An example of computation results for one month and one location is presented in Table 3.

## COMPARISON OF HVAC SYSTEM TYPES

In the typical HVAC system design, the outdoor airflow rate is kept constant throughout the year at 18 m$^3$/(h m$^2$) (0.98 cfm/ft$^2$), which is approximately 33% of the total supply airflow rate. The outdoor air supply exceeds the required rate (Table 1) by 40% during the slow hours and is 12% lower than required during the peak hours. System zoning does not sepa-

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## TABLE 1
## Minimum Required Outside Airflow Rate into the Dining Area

| Outside Airflow Rate | | Time of Day | | | | | |
|---|---|---|---|---|---|---|---|
| | | 1 a.m. - 7 a.m | 7 a.m. -11:30 a.m. | 11:30 a.m. - 2 p.m. | 2 p.m. - 5 p.m | 5 p.m. - 9 p.m | 9 p.m. - 1 a.m. |
| For Ventilation | cfm | 0 | 0 | 2698 | 208 | 4038 | 1015 |
| | cfm/ft$^2$ | 0 | 0 | 0.75 | 0.06 | 1.12 | 0.28 |
| | m$^3$/h | 0 | 0 | 4583 | 353 | 6860 | 1725 |
| | m$^3$/(h·m$^2$) | 0 | 0 | 13.7 | 1.1 | 20.5 | 5.2 |
| Transfer Air Supplied into the Dining Area | cfm | 0 | 2500 | 2500 | 2500 | 2500 | 2500 |
| | cfm/ft$^2$ | 0 | 0.7 | 0.7 | 0.7 | 0.7 | 0.7 |
| | m$^3$/h | 0 | 4248 | 4248 | 4248 | 4248 | 4248 |
| | m$^3$/(h·m$^2$) | 0 | 12.7 | 12.7 | 12.7 | 12.7 | 12.7 |
| Overall | cfm | 0 | 2500 | 2698 | 2500 | 4038 | 2500 |
| | cfm/ft$^2$ | 0 | 0.7 | 0.75 | 0.7 | 1.12 | 0.7 |
| | m$^3$/h | 0 | 4248 | 4583 | 4248 | 6860 | 4248 |
| | m$^3$/(h·m$^2$) | 0 | 12.7 | 13.7 | 12.7 | 20.5 | 12.7 |

## TABLE 2
## Building Characteristics

| Building Structure | Wall Area* ft$^2$ (m$^2$) | U-Factor Btu/(h·ft$^2$·°F) [W/(m$^2$·°C)] | Glass Area† ft$^2$ (m$^2$) | U-Factor Btu/(h·ft$^2$·°F) W/(m$^2$·°C) |
|---|---|---|---|---|
| North Wall | 972 (90.3) | 0.039 (0.22) | 192 (17.8) | 0.55 (3.12) |
| East Wall | 845 (78.5) | 0.039 (0.22) | 220 (20.4) | 0.55 (3.12) |
| South Wall | 137 (12.7) | 0.039 (0.22) | 28 (2.6) | 0.55 (3.12) |
| Northwest Wall | 140 (13.0) | 0.039 (0.22) | 40 (3.7) | 0.55 (3.12) |
| West Wall | 510 (47.4) | 0.039 (0.22) | 135 (12.5) | 0.55 (3.12) |
| Roof‡ | 515 (47.8) | 0.046 (0.26) | — | — |

* Walls: color = medium, weight = medium.
† Vertical glass - double glazed, shade factor = 0.71.
‡ Roof: color = light, weight = light.

rate smoking/nonsmoking areas. Smoke is transferred from the smoking area to the nonsmoking area with return air and horizontal room air movement. Also, smoke is transferred from the dining area into the kitchen with transfer air.

For the purpose of HVAC system comparison, the following was assumed in all cases:

- Air is treated in air-handling units using one or several of the following processes: heating, cooling, and dehumidification in air-heating/cooling/dehumidification coils.
- Control systems are capable of maintaining indoor air temperature and relative humidity at levels that allow for minimal energy consumption at a given set of outdoor air parameters. Air heating in fans does not significantly affect the accuracy of the system comparison and, thus, was neglected.
- Requirements to the room air are as follows:

  Air temperature in the occupied zone:
  - in winter: $t_{o.z.}$ = 21°C - 23°C (70°F - 73°F)
  - in summer: $t_{o.z.}$ = 23°C - 26°C (73°F - 79°F)

  Relative humidity in the occupied zone:
  - in winter: $\varphi_{o.z.}$ = 20% - 40%
  - in summer: $\varphi_{o.z.}$ = 40% - 60%
- The averaged hourly cooling loads, outdoor dry-bulb temperatures (ODBT), and outdoor wet-bulb temperatures (OWBT) for all five locations are the same for all cases (see example of data presentation in Table 3).
- Due to high cooling loads, induction type displacement air diffusers are used, which allows for an increase in the maximum value of the supply air temperature difference of $\Delta t_o$ = 6°C (11°F).
- Air distribution systems in Cases 1 and 3 allow perfect

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**TABLE 3**
**January Data for the Restaurant Located in Phoenix, Ariz.**

| Time | Indoor Air T(°C) | System Load | | Internal Load (W) | ODBT (°C) | OWBT (°C) | Outdoor Airflow | | Latent Heat Load | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | W | W/m² | | | | m³/h | m³/h per m² | W | W/m² |
| 0~1 | 25.6 | −20951 | −63.0 | 25262 | 7.2 | 3.7 | 4214 | 12.7 | 13600 | 40.9 |
| 1~2 | 24.6 | 0 | 0 | 0 | 6.2 | 3.2 | 0 | 0 | 0 | 0 |
| 2~3 | 24.2 | 0 | 0 | 0 | 5.7 | 2.8 | 0 | 0 | 0 | 0 |
| 3~4 | 24.0 | 0 | 0 | 0 | 5.5 | 2.6 | 0 | 0 | 0 | 0 |
| 4~5 | 23.8 | 0 | 0 | 0 | 4.8 | 2.1 | 0 | 0 | 0 | 0 |
| 5~6 | 23.6 | 0 | 0 | 0 | 4.4 | 1.8 | 0 | 0 | 0 | 0 |
| 6~7 | 23.4 | 0 | 0 | 0 | 4.2 | 1.7 | 0 | 0 | 0 | 0 |
| 7~8 | 24.3 | 0 | 0 | 3598 | 3.8 | 1.3 | 2500 | 7.5 | 0 | 0 |
| 8~9 | 25.0 | 0 | 0 | 3598 | 5.3 | 2.4 | 2500 | 7.5 | 0 | 0 |
| 9~10 | 26.0 | 0 | 0 | 3598 | 8.4 | 4.5 | 2500 | 7.5 | 0 | 0 |
| 10~11 | 26.8 | 0 | 0 | 3598 | 11.7 | 6.4 | 2500 | 7.5 | 0 | 0 |
| 11~12 | 25.6 | −17012 | −51.1 | 19018 | 14.1 | 7.6 | 4778 | 14.4 | 6500 | 19.5 |
| 12~13 | 25.6 | −32285 | −97.0 | 34261 | 16.0 | 8.5 | 4778 | 14.4 | 13000 | 39.0 |
| 13~14 | 25.6 | −32484 | −97.6 | 34262 | 17.5 | 9.1 | 4778 | 14.4 | 13000 | 39.0 |
| 14~15 | 25.6 | −20849 | −62.7 | 22692 | 18.3 | 9.3 | 2500 | 7.5 | 1000 | 3.0 |
| 15~16 | 25.6 | −20851 | −62.7 | 22692 | 18.8 | 9.5 | 2500 | 7.5 | 1000 | 3.0 |
| 16~17 | 25.6 | −20641 | −62.0 | 22692 | 18.7 | 9.4 | 2500 | 7.5 | 1000 | 3.0 |
| 17~18 | 25.6 | −37015 | −111.3 | 39943 | 17.3 | 8.9 | 7448 | 22.4 | 18360 | 55.2 |
| 18~19 | 25.6 | −36721 | −110.4 | 39943 | 13.9 | 7.7 | 7448 | 22.4 | 18360 | 55.2 |
| 19~20 | 25.6 | −36369 | −109.3 | 39943 | 11.6 | 6.5 | 7448 | 22.4 | 18360 | 55.2 |
| 20~21 | 25.6 | −36171 | −108.7 | 39943 | 10.1 | 5.8 | 7448 | 22.4 | 18360 | 55.2 |
| 21~22 | 25.6 | −21289 | −64.0 | 25262 | 8.7 | 4.9 | 4214 | 12.7 | 13600 | 40.9 |
| 22~23 | 25.6 | −21182 | −63.7 | 25262 | 8.1 | 4.5 | 4214 | 12.7 | 13600 | 40.9 |
| 23~24 | 25.6 | −21046 | −63.3 | 25262 | 7.4 | 3.9 | 4214 | 12.7 | 13600 | 40.9 |

mixing, which results in the same return/exhaust air and occupied zone air temperatures and enthalpy, and the heat removal efficiency coefficient $K_t$ is equal to

$$K_t = \frac{I_{exh} - I_o}{I_{o.z.} - I_o} = 1 .$$

Displacement air distribution, utilized in Cases 2 and 4, creates a temperature/enthalpy gradient along room height. Calculations according to the design procedure described in Zhivov et al. (1997b) result in the following value for heat removal efficiency $K_t$:

$$K_t = \frac{I_{exh} - I_o}{I_{o.z.} - I_o} = 2 .$$

- HVAC systems in all these cases utilize the same air-handling units. The difference in the first costs and maintenance costs for the duct system in cases with mixing and displacement air distribution systems is neglected. Thus, the systems are compared only by the annual energy consumption costs.

Four cases of HVAC systems in the prototype restaurant design were compared:

Case 1. Constant outdoor airflow rate HVAC system with mixing type air distribution: outdoor airflow rate = 18 m³/h m² (0.98 cfm/ft²), $K_t = 1$, $\Delta t_o \leq 9$°C (16°F).

Case 2. Constant outdoor airflow rate HVAC system with displacement air distribution: outdoor airflow rate = 18 m³/h m² (0.98 cfm/ft²), Kt = 2, $\Delta t_o \leq 6$°C (11°F)—induction air diffusers are used.

Case 3. Variable outdoor airflow rate HVAC system with

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mixing air distribution: outdoor airflow rate varies according to the schedule in Table 3, $K_t = 1$, $\Delta t_o \leq$ 9°C (16°F).

Case 4. Variable outdoor airflow rate HVAC system with displacement air distribution: outdoor airflow rate varies according to the schedule in Table 3, $K_t = 2$, $\Delta t_o \leq 6$°C (11°F)—induction air diffusers are used.

In Cases 1 and 2, the minimum outdoor airflow rate is equal to 18 m$^3$/h m$^2$ (0.98 cfm/ft$^2$); in Cases 3 and 4—according to the schedule in Table 1. Total airflow rate supplied into the dining area is 54 m$^3$/(h/m$^2$) (2.95 cfm/ft$^2$).

## METHOD

HVAC systems in Cases 1 through 4 were analyzed by comparing such parameters as heating and cooling energy consumption throughout the year-round operating cycle. The above parameters, as well as the airflow rate and water consumption, are typically used for the evaluation of different HVAC system designs and their components. The combination of those parameters allows for the evaluation of the influence of a single input parameter on the HVAC system performance. This method is discussed in detail by Rymkevich et al. (1990, 1995).

This approach is used to evaluate the influence of (a) the outdoor airflow rate and (b) the method of air supply, mixing or displacement, on energy consumption by the HVAC system in restaurants located in different climates.

## RESULTS AND DISCUSSION

Heat and cold consumption by HVAC systems in different cases throughout the year-round operating cycle is presented for different locations in Figures 1 through 10. The total annual heating and cooling energy consumption by the system in each case and in each location is compared on bar graphs in Figures 11 and 12 and in Table 4.

The analyses show that the restaurant location (climate) has insignificant influence on the total heating/cooling loads from internal heat sources, and heat transfer through the building envelope is due to a low U-factor of the building envelope. However, the location of the restaurant has a significant effect on the heating/cooling energy consumption by the HVAC system for the heating and cooling of outside air.

As can be seen from Figures 1, 2, 5, and 6, the increase in outdoor air supply in Miami and Albuquerque results in an increase of both heating and cooling energy consumption. In other climates, the increase in outdoor air supply results in reduced cooling energy consumption. The analysis shows that for the Phoenix, Minneapolis, and Seattle locations, HVAC system operation with a variable outdoor air supply allows a decrease in cooling consumption up to 50% and in some cases the elimination of the use of refrigeration machines.

In operation modes with heat consumption, it is always important to reduce the outdoor airflow rate. Such operation modes have long duration in locations such as Seattle and Minneapolis.

**TABLE 4**
**Annual Energy Cost Comparison for HVAC Systems with Mixing and Displacement Air Distribution**

| Location | Annual Energy Usage, kWh/m² (1000 Btu/ft²): Constant Outdoor Airflow, Mixing Air Distribution | | Constant Outdoor Airflow, Displacement Air Distribution | | Variable Outdoor Airflow, Mixing Air Distribution | | Variable Outdoor Airflow, Displacement Air Distribution | | Energy Prices (1996): Power $/kWh | Gas | | Annual Energy Cost, $/m² ($/ft²): Constant Outdoor Airflow, Mixing Air Distribution | Constant Outdoor Airflow, Displacement Air Distribution | Variable Outdoor Airflow, Mixing Air Distribution | Variable Outdoor Airflow, Displacement Air Distribution |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Cool | Heat | Cool | Heat | Cool | Heat | Cool | Heat | | $/Therm | $/kW* | | | | |
| Miami, Fla. | 804.6 (255.0) | 8.4 (2.7) | 698.4 (221.4) | 9.3 (2.9) | 769.2 (243.8) | 4.4 (1.4) | 674.9 (213.9) | 5.0 (1.6) | 0.085 | † | - | 69.1 (6.4) | 60.2 (5.6) | 65.8 (6.1) | 57.8 (5.4) |
| Phoenix, Ariz. | 572.2 (181.4) | 33.9 (10.7) | 469.6 (148.8) | 44.2 (14.0) | 582.4 (184.6) | 27.8 (8.8) | 485.2 (153.8) | 29.9 (9.5) | 0.1 | 0.33 | 0.011 | 57.6 (5.4) | 47.4 (4.4) | 58.5 (5.4) | 48.8 (4.5) |
| Albuquerque, N. Mex. | 315.9 (100.1) | 150.9 (47.8) | 246.3 (78.1) | 183.1 (58.0) | 294.7 (93.4) | 91.7 (29.1) | 234.1 (74.2) | 121.3 (38.4) | 0.095 | 0.41 | 0.014 | 32.1 (3.0) | 26.0 (2.4) | 29.3 (2.7) | 23.9 (2.2) |
| Seattle, Wash. | 96.3 (30.5) | 188.1 (59.6) | 53.0 (16.8) | 243.1 (77.1) | 112.2 (35.6) | 112.7 (35.7) | 67.2 (21.3) | 155.2 (49.2) | 0.038 | 0.45 | 0.015 | 6.5 (0.6) | 5.7 (0.5) | 6.0 (0.6) | 4.9 (0.5) |
| Minneapolis, Minn. | 163.3 (51.8) | 447.1 (141.7) | 116.3 (36.9) | 495.4 (157.0) | 179.9 (57.0) | 261.6 (82.9) | 134.2 (42.4) | 300.7 (95.3) | 0.066 | 0.35 | 0.012 | 16.1 (1.5) | 13.6 (1.3) | 15.0 (1.4) | 12.5 (1.2) |

* 1 therm = 29.3 kWh
† Electric power is used for heating.

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The effect of temperature stratification on HVAC system parameters is the same for all locations: displacement ventilation systems result in decreased cooling energy consumption, but increased heating consumption. The data presented in Figure 12 show that at a fixed outdoor airflow rate, HVAC systems with displacement air distribution have from 12% to 18% lower energy costs (depending on the climatic region), compared to the currently used systems with mixing air distribution. In the cases with demand-based outdoor airflow rate, the HVAC system with displacement air distribution allows for a reduction in energy consumption from 16% to 26%. In addition to reduced energy consumption, displacement ventilation systems allow for a better indoor air quality in the breathing zone due to contaminant stratification along the room height.

The analysis described in this paper is based on the given set of operation modes performed by typically used air-handling units. Using exhaust air heat recovery, second recirculation (outdoor air is mixed with return air after its heating/cooling and dehumidification) vs. first recirculation (outdoor air is mixed with return air prior to its treatment in air-handling unit), and other modifications of air treatment and control strategies in combination with displacement air distribution will result in improved HVAC system first and operating costs.

Other means of improving indoor air quality in restaurants with smoking and nonsmoking areas may include system zoning based on smoking/nonsmoking areas; air exhaust from the bar and the smoking area—not returning it back to smoking or nonsmoking areas; and location of nonsmoking area close to the kitchen with a transfer air supply into this area.

## ACKNOWLEDGMENTS

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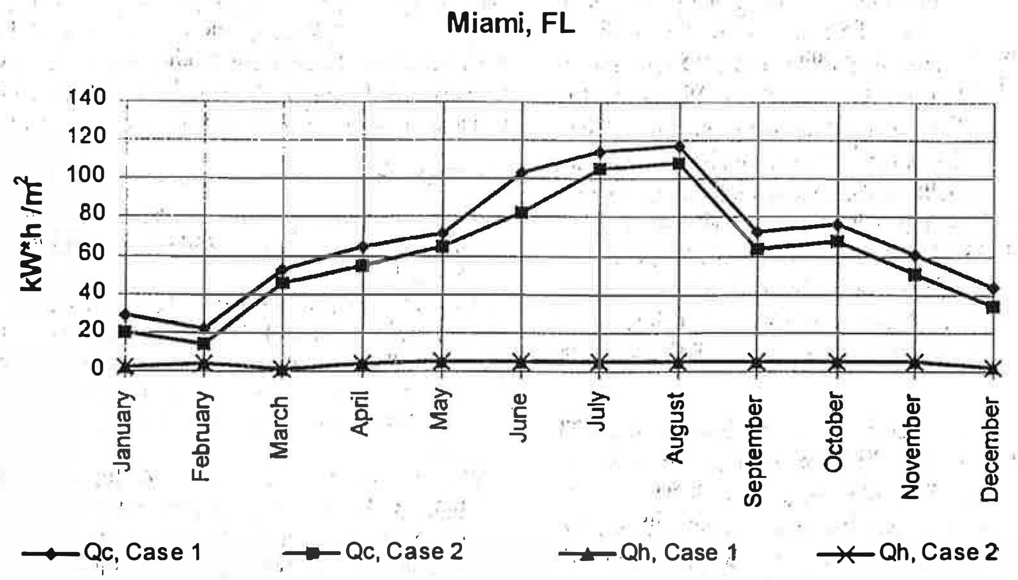


***Figure 1*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by constant outdoor airflow HVAC system with mixing (Case 1) and displacement (Case 2) air distribution in Miami, Fla.*

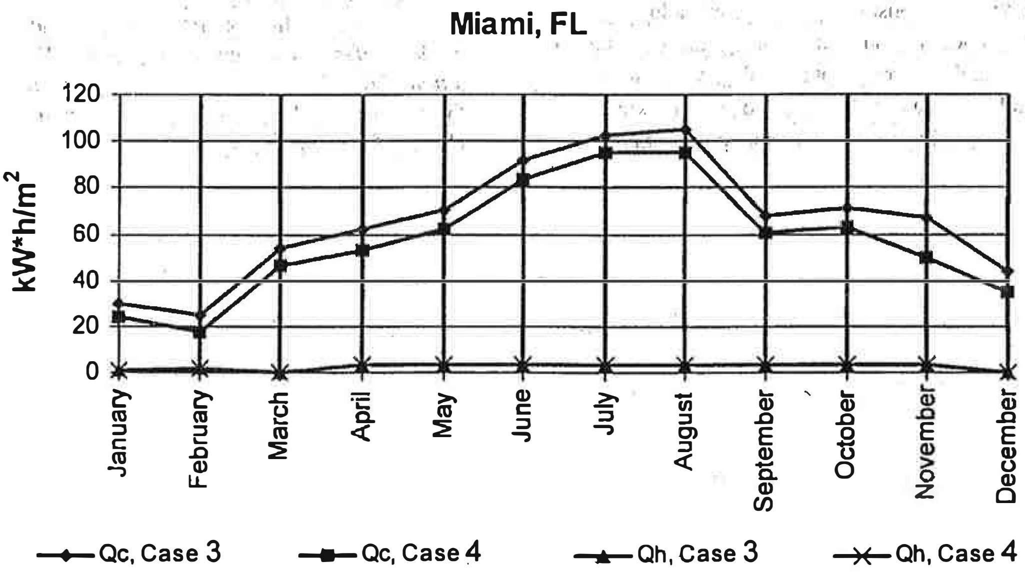


***Figure 2*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by variable outdoor airflow HVAC system with mixing (Case 3) and displacement (Case 4) air distribution in Miami, Fla.*

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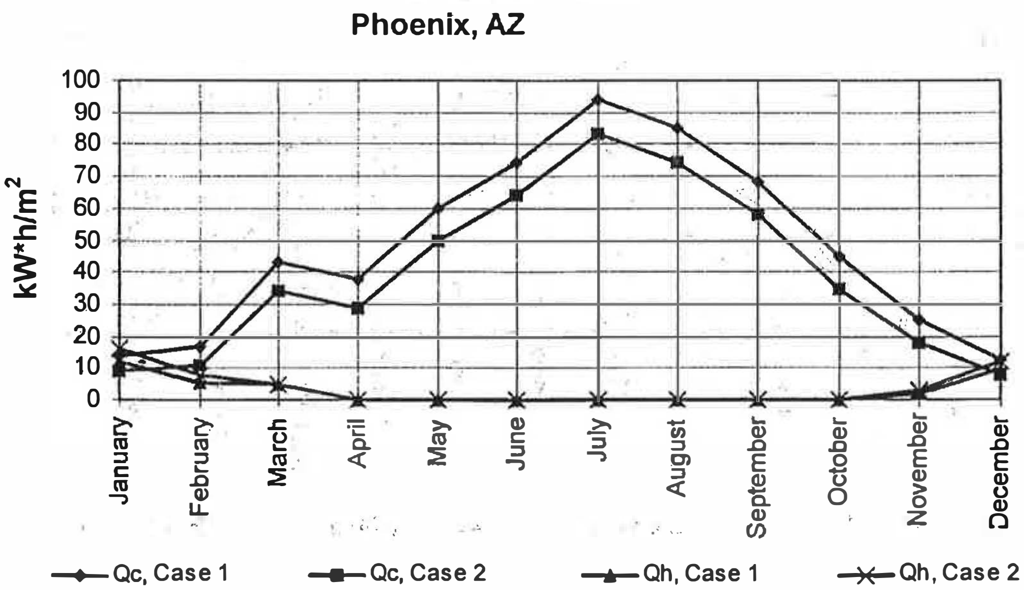


***Figure 3*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by constant outdoor airflow HVAC system with mixing (Case 1) and displacement (Case 2) air distribution in Phoenix, Ariz.*

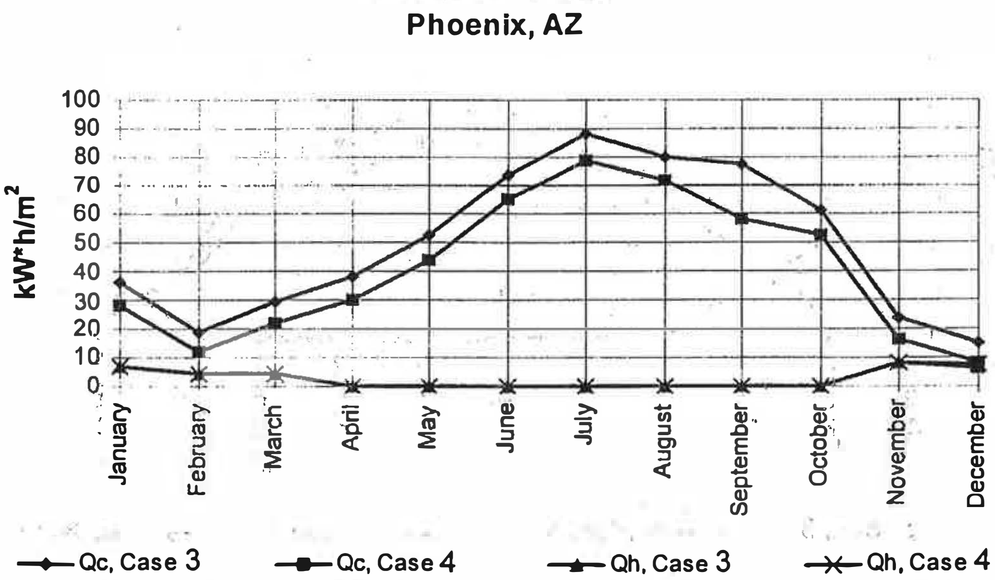


***Figure 4*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by variable outdoor airflow HVAC system with mixing (Case 3) and displacement (Case 4) air distribution in Phoenix, Ariz.*

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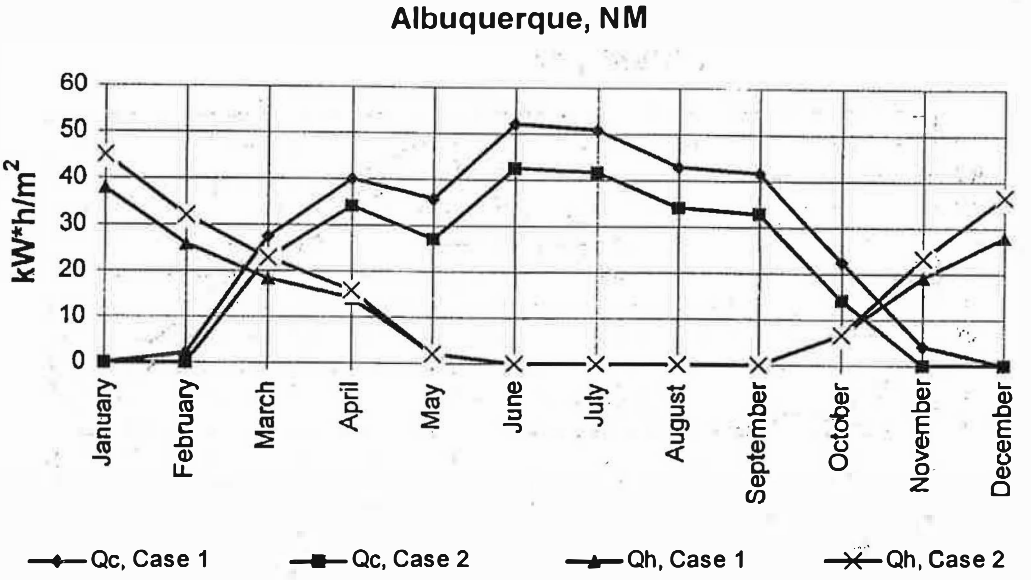


***Figure 5*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by constant outdoor airflow HVAC system with mixing (Case 1) and displacement (Case 2) air distribution in Albuquerque, N. Mex.*

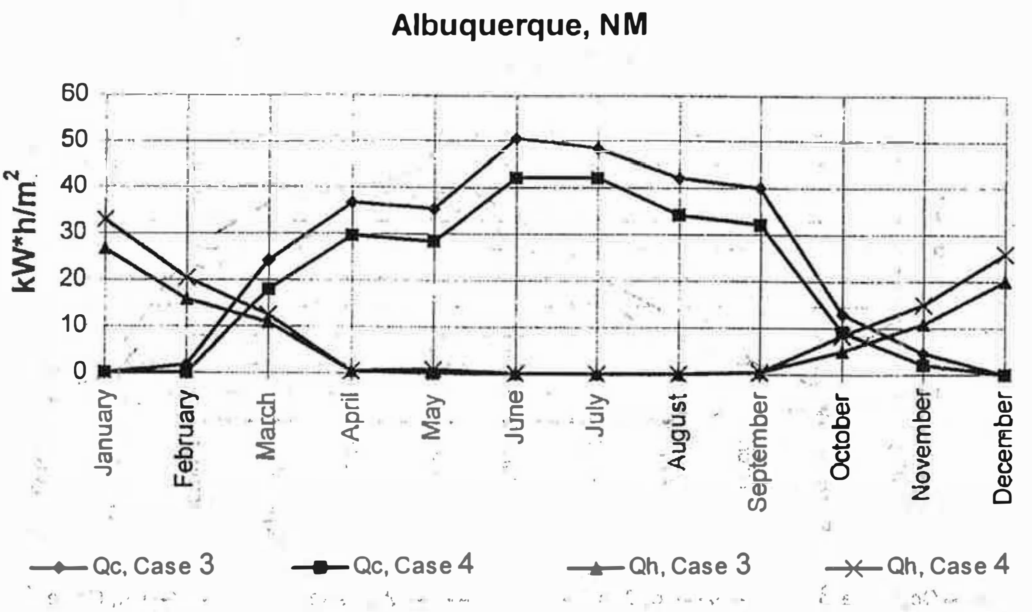


***Figure 6*** *Monthly cold ($Q_c$) and heat ($Q_h$) consumption by variable outdoor airflow HVAC system with mixing (Case 3) and displacement (Case 4) air distribution in Albuquerque, N. Mex.*

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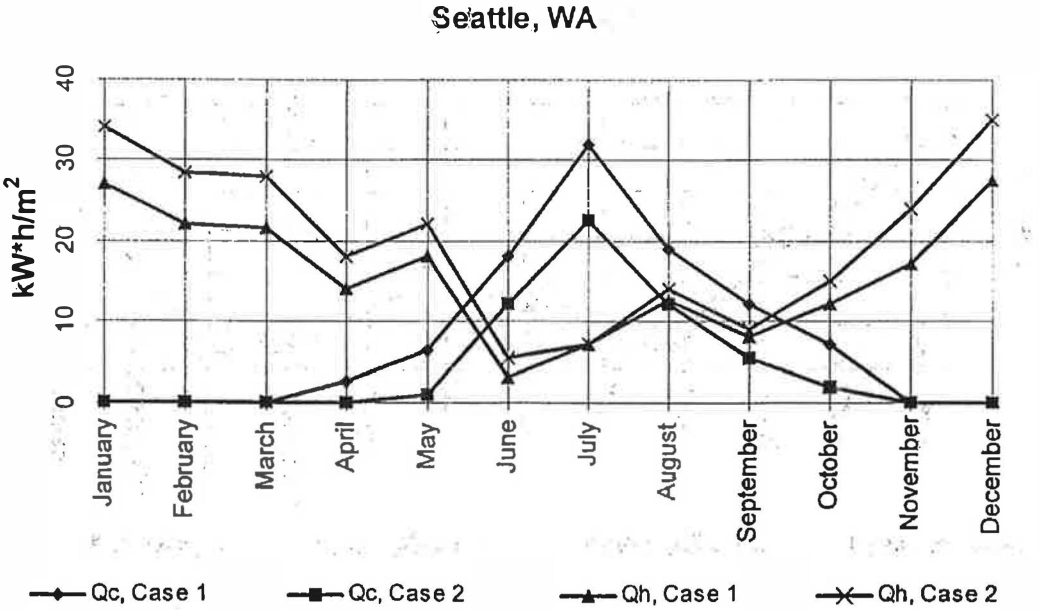


*Figure 7* Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by constant outdoor airflow HVAC system with mixing (Case 1) and displacement (Case 2) air distribution in Seattle, Wash.

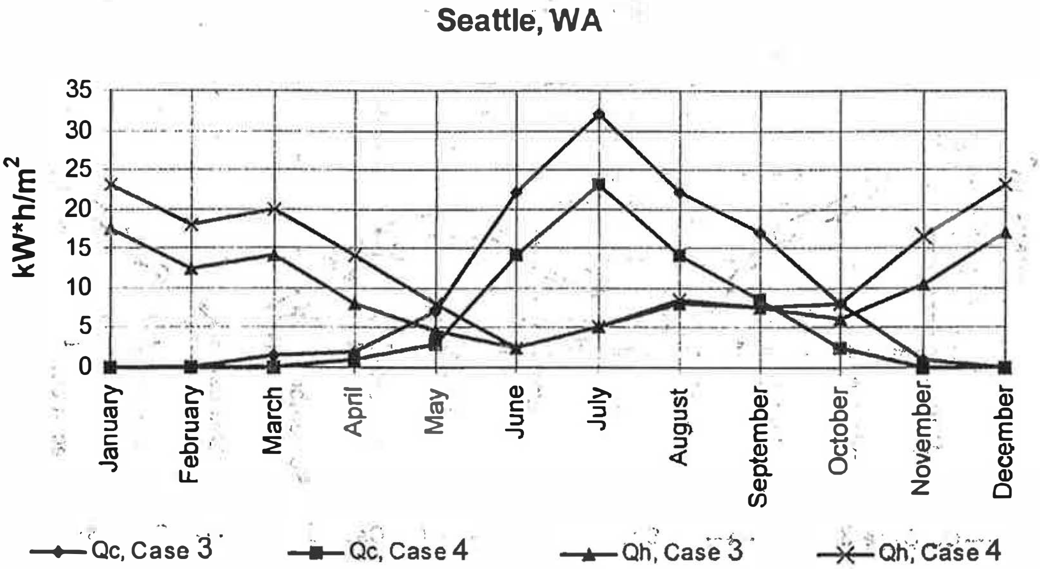


*Figure 8* Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by variable outdoor airflow HVAC system with mixing (Case 3) and displacement (Case 4) air distribution in Seattle, Wash.

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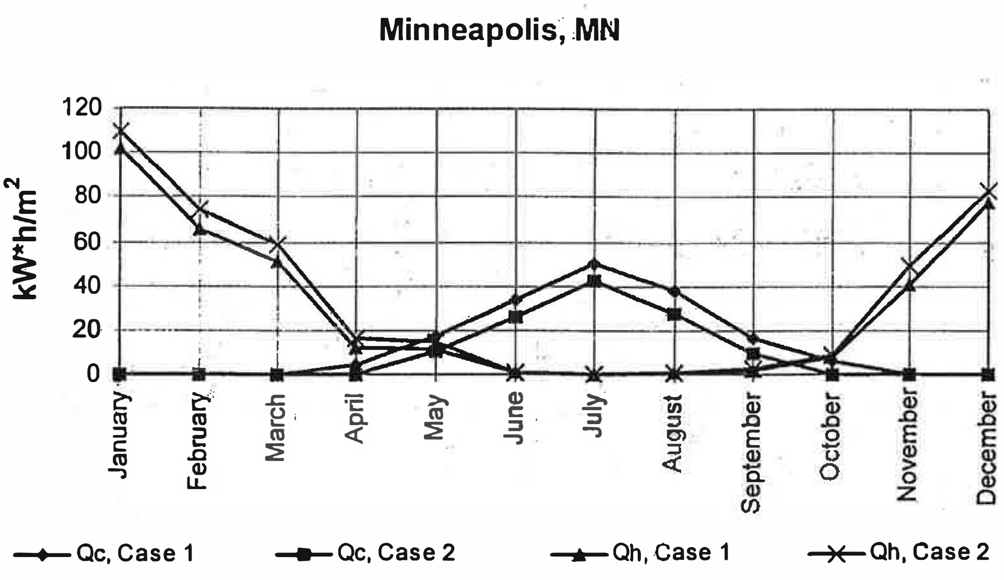


***Figure 9*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by constant outdoor airflow HVAC system with mixing (Case 1) and displacement (Case 2) air distribution in Minneapolis, Minn.*

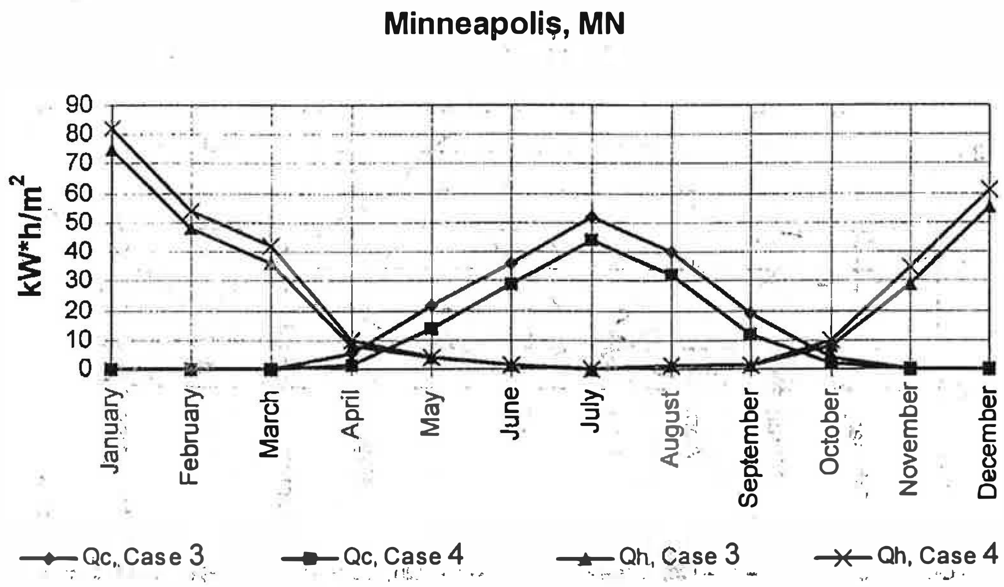


***Figure 10*** *Monthly cooling ($Q_c$) and heating ($Q_h$) energy consumption by variable outdoor airflow HVAC system with mixing (Case 3) and displacement (Case 4) air distribution in Minneapolis, Minn.*

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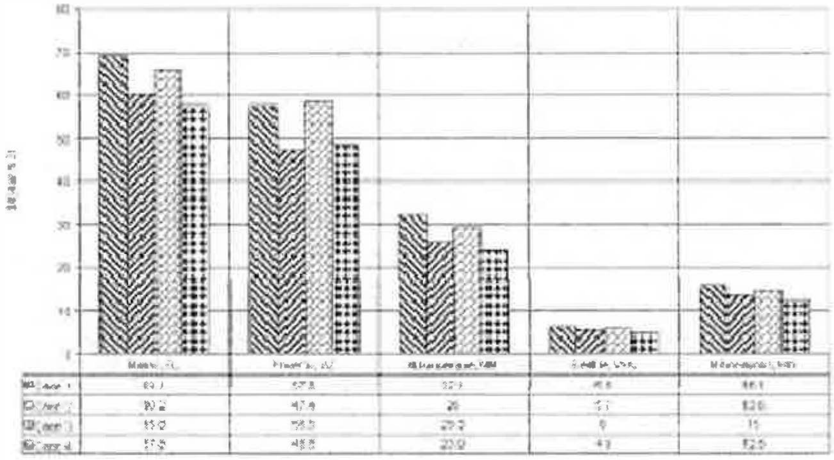

*Figure 11* *Annual energy cost comparison for HVAC systems: Case 1: constant outdoor airflow, mixing air distribution; Case 2: constant outdoor airflow, displacement air distribution; Case 3: variable outdoor airflow, mixing air distribution; Case 4: variable outdoor airflow, displacement air distribution.*

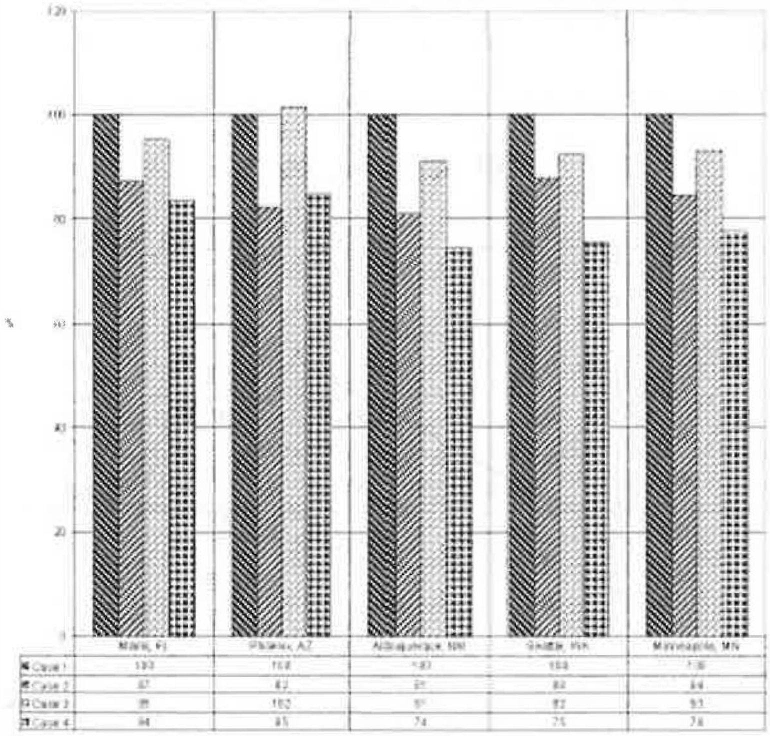


*Figure 12* *Relative annual energy cost comparison for HVAC systems: Case 1: constant outdoor airflow, mixing air distribution; Case 2: constant outdoor airflow, displacement air distribution; Case 3: variable outdoor airflow, mixing air distribution; Case 4: variable outdoor airflow, displacement air distribution.*