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Strategies for Energy Conservation for A Large Retail Store

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
National Engineering Laboratory
Center for Building Technology
Building Equipment Division
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STRATEGIES FOR ENERGY CONSERVATION FOR A LARGE RETAIL STORE

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U.S. DEPARTMENT OF COMMERCE, Malcolm Baldrige, ***Secretary***
NATIONAL BUREAU OF STANDARDS, Ernest Ambler, ***Director***

PREFACE

This report is one of a series documenting NBS research and analysis efforts in developing energy and cost data to support the Department of Energy/National Bureau of Standards Building Energy Conservation Criteria Program. The work reported in this document was performed under the Energy Analysis of Control Strategies project, a part of the controls program element managed by Building Systems Division, Office of Building Energy Research and Development, U.S. Department of Energy. The NBS effort was supported by DoE/NBS Task Order A008-BCS under Interagency Agreement No. EA77A 01-6010.

ABSTRACT

A comparative analysis is made of the thermal performance of selected HVAC systems and control strategies commonly employed in large retail stores. The comparisons are made for six geographical locations representing wide climatic variations within the continental United States.

Hour-by-hour simulations with the BLAST computer program are used to obtain the yearly heating, cooling, and fan energy consumption of a two-story large retail store. The HVAC systems simulated are constant volume reheat, variable air volume, and with direct expansion coils. The control strategies tested are dry bulb temperature economy cycle, enthalpy economy cycle, supply air temperature resetting, lowered space heating temperature, VAV zoning variations, and the combinations of these strategies. The results of these simulations are presented and discussed. Substantial energy consumption differences are shown.

Key words: building control strategies; building energy conservation; building thermal performance; HVAC systems

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1. INTRODUCTION

This report is the second of a series of reports [1] to compare the energy savings of heating, ventilating and air conditioning (HVAC) systems and control strategies for commercial buildings. A large retail store of heavy architectural construction was used as a sample building for energy consumption comparisons in this study. According to a recent national survey [2], for buildings over 25,000 square feet of floor area, retail sales and services occupy 18 percent of the total floor area of commercial buildings. The large retail store of this study represents a department store or a variety store which is either independently situated or is attached to a large shopping complex. Buildings of this kind, as compared to small stores having unitary heating/cooling equipment with simple thermostatic on-off type temperature controls, are usually engineered by professional designers and operated by full-time trained operating personnel. Therefore, this study was intended to give comparative quantitative data and some guidelines to assist building designers and system operators with HVAC system and control strategy selections as well as with setting system operating conditions, using energy consumption of the HVAC system as a parameter. Of course, the energy consumption of a particular store is a function of many factors, such as the building geometric shape and orientation, space arrangement, building construction, climatic conditions, internal load variations, etc. It is not the intention of this study to negate the importance and necessity of having a thorough energy analysis and life cycle cost study for cost-effective HVAC system and control strategy determination. The scope of the energy analysis of this study is limited to the air handling systems. The heating/cooling plant energy consumption as well as the energy used for distributing the cooling and heating media from plant to air handling equipment are not included. Furthermore, simple assumptions of the fan and air distribution systems were made for the air handling system simulations. Consequently, readers are cautioned to use care in applying these data to their particular buildings.

The localities used for the energy consumption comparisons in this study are six cities representing various climatic conditions of the United States. They are Lake Charles, Louisiana; Madison, Wisconsin; Nashville, Tennessee; Santa Maria, California; Seattle, Washington; and Washington, D.C.

The energy program used in this study for simulation was the Building Loads Analysis and System Thermodynamics Program (BLAST) [3]. This program was used because it was recognized as one of the best building energy programs available commercially. The control loop dynamics of the control system components which may also impact significantly on the building energy consumption was not investigated in this study.

2. ENERGY SIMULATION

2.1 BUILDING MODEL

The building used for this study is a two-story, 320 feet (97.5 m) long by 240 feet (73.2m) wide, heavy masonry structure with its short axis coinciding with the true north-south orientation. There is no fenestration except glass entrance doors on all walls of the lower level. The building dimensions and pertinent construction data are as shown in figure 1. The architectural specification of this building was modified from a model building of a state energy performance standard [4]. The daily internal load profiles are shown in figure 2. It was assumed that 30 percent of the light load was removed directly by the return air stream. The 1.0 watt per square foot (10.8 W/m^2) equipment load included all appliances for sales and display, elevator and escalator loads as well as other incidental equipment loads. The infiltration load was assumed to be nil during the system-on hours and was assumed to be 0.6 air change per hour for the perimeter zones during the system-off hours.

The contents of the store, which may be important for both thermal capacitance and thermal radiation heat exchange between the surfaces for a retail store, were simulated by using 4" (.102 m) thick concrete masonry unit partitions.

The building was divided into ten thermal zones for the constant volume (CV) and variable air volume (VAV) systems using chilled water as the cooling medium. The first and second floors had separate thermal zones because of the space load differences contributed by the roof, ground and the entrance doors. The interior and perimeter spaces of different exposures had separate thermal zones. Zone multipliers were used to multiply the equally divided zone areas of 1920 square feet (178.4 m^2) to simplify the calculations. The entire building was modeled by using one air handling system. The heating and cooling energy consumptions should remain the same, should more than one system be used on account of system size limitations.

For the simulation of the direct expansion (DX) system, the building was divided into 12 thermal zones, differentiating between upper and lower floors, interior and perimeter spaces, as well as exposures, with four interior zones as contrasted to two interior zones of the chilled water system. The floor areas of the zones were made larger than those of the chilled water system and zone multipliers were not used. It was felt that, in actual system design, the 12 zone arrangement was more realistic for the DX system than having ten zones composed of 80 multipliers as was used for the chilled water system. The thermal zone arrangement of both chilled water and DX systems are shown in figure 3.

2.2 WEATHER DATA

The weather data used for the simulations were from Typical Meteorological Year (TMY) climate tapes.

2.3 HVAC SYSTEM SIMULATION AND CONTROL STRATEGIES

Three types of HVAC system were simulated.

A. One constant volume system with ten thermal zones. Eight of the ten zones had reheat controls to serve the perimeter.

B. One variable air volume reheat system with variable air volume for both eight perimeter zones only and for all ten zones.

C. Twelve direct expansion constant volume systems for twelve thermal zones. Each system had a pair of heating and cooling coils with controls preventing them from operating simultaneously.

Although the use of constant volume reheat systems is generally discouraged for energy conservation reasons, this study makes no judgment as to the overall suitability of system selection. Furthermore, reheat systems were used extensively during the past and some of these existing systems may need to be retrofitted for energy conservation. The results of these simulations may be useful in those applications.

During the zone load simulation stage, the heating and cooling capacities of all zones were sized to take into account the climatic differences among the six cities. The hourly load simulation results indicated that the maximum zone temperatures at peak cooling hours were no more than 1°F (.56°C) above the design temperature of 78°F (25.6°C). The indoor temperature during heating operations varied depending on the control strategies. The details will be described under the control strategy paragraphs later in this report. The supply air to the spaces was about 0.73 CFM per square foot (3.71×10^{-3} m^3/s per m^2) of floor area. A ventilation air quantity of 5 CFM ($2.36 \times 10^{-3} m^3/s$) per person at peak occupancy hours was assumed, which amounted to 17 percent to 22 percent of the space supply air.

The following approach was used in comparing the control strategies. A base case was first established and simulated. Then, the base case was modified by single control strategies. This means that only one control strategy was used in each simulation run, so that the effect of energy savings of this particular strategy may be compared with the base case and with other strategies. Finally, combined strategies were simulated to find the combined effect on energy consumption. For the chilled water system, the control strategies included dry-bulb temperature economy cycle, enthalpy economy cycle, reset supply air temperature by outside temperature, lower reheat temperature, applying VAV to perimeter zones and applying VAV to all zones. For the DX system, only enthalpy economy cycle was simulated to compare with the base DX strategy.

A. Base Case (Case 1).

The fan system ran continuously from 8 a.m. to 11 p.m. every day. At night, the fan ran intermittently and tried to satisfy the set-back temperatures of 52°F (11.1°C) minimum and 85 °F (29.4°C) maximum. The minimum outside damper position was fixed to give minimum ventilation air. The supply air temperature at fan discharge was set to maintain 60°F (15.6°C). The cooling coil controller throttling range was assumed to be 3°F (1.7°C) from 60°F (15.6°C) to 57°F (13.9°C). The daytime reheat controls were set to provide 72°F (22.2°C) minimum space temperature for the perimeter zones. The heating coil controller throttling range was also assumed to be 3°F (1.7°C). Except for peak cooling load hours, these cooling and heating temperature settings satisfied the recommended indoor design conditions for comfort air conditioning of the American Society for Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 90-75, Energy Conservation in New Building Design. As stated previously, only the eight perimeter zones had reheat coils.

B. Base Case with Temperature Economy Cycle (Case 2)

In this case, the mixed air temperature downstream of the return and outside air dampers was maintained by modulating the dampers to satisfy the cooling coil thermostat setting of 54.9°F (12.7°C) to 57.9°F (14.4°C) on account of 2.1°F (1.2°C) fan heat gain. When the outside air temperature rose to above the cooling coil thermostat setting, the outside air damper stayed at the minimum outside air position and the cooling was achieved by mechanical cooling.

C. Base Case with Enthalpy Economy Cycle (Case 3)

Enthalpy sensors were added to both the return air and outside air streams. When the outside air enthalpy was lower than the return air enthalpy, the outside air and return air dampers were modulated to maintain the desired cooling coil thermostat setting. When the enthalpy comparison was reversed, the system admitted only the required minimum outside air. Mechanical cooling was used to supplement the balance of the cooling load.

D. Base Case with Supply Air Temperature Reset (Case 4)

This strategy takes advantage of the space cooling load variations. It is generally achieved by two ways: to set the discharge air temperature to satisfy the highest cooling load zone, or set the discharge air temperature according to a fixed schedule using the outside air temperature as the parameter. Since this building has no window and is of heavy construction, it should not be sensitive to the hourly external load change. Therefore, it was felt that setting the cooling coil discharge air temperature by the outside air temperature would be advantageous. The discharge air temperature setting varied from 60°F (15.6°C) to 62°F (16.7°C) inversely with the outside air temperature of 75°F (23.9°C) to 32 °F (0°C). This reset schedule was determined rather arbitrarily. No attempt was made to find the optimum reset schedule for energy savings.

E. Base Case with Lowered Space Temperature (Case 5)

The minimum space temperature for the reheat was lowered from 72°F (22.2°C) of the base case to 68°F (20°C) during the store hours. During 1979 to 1981 the U.S. Department of Energy regulations required that 65°F (18.3°C) be used as the space temperature for heating in retail stores. It was evident that a space temperature lower than the 72°F (22.2°C) as recommended by ASHRAE Standard 90-75 was generally acceptable.

F. Base Case with Perimeter Zones Converting to VAV (Case 6)

In this case, the eight perimeter zones were converted to a reheat VAV system with a minimum supply air volume at 40 percent of the peak supply air. The air volume variations not only adjusted to the load shifting due to the external factors such as solar and outside air temperature changes, it also tried to match the internal load changes during off-peak hours. No detailed analysis was performed to determine the minimum supply air volume, and the 40 percent ratio was used for all six cities.

G. Base Case with All Zones Converting to VAV (Case 7)

This case is an extension of the previous case. The two large interior zones were also provided with 40 percent minimum supply air VAV controls. There was no reheat in interior zones. By adding these two zones to the VAV system, the interior zone roof load and internal load variations were also matched by reducing the supply air.

H. Base Case with Temperature Economy Cycle and Lowered Space Temperature (Case 8)

This case and the following four cases are the combination cases of the previous described cases. This case combines cases 2 and 5.

I. Base Case with Temperature Economy Cycle, Lowered Space Temperature, and Supply Air Temperature Reset (Case 9)

This case combines cases 2, 4, and 5.

J. Base Case with Enthalpy Economy Cycle, Lowered Space Temperature, and Supply Air Temperature Reset (Case 10)

This case combines cases 3, 4, and 5.

K. Base Case with Enthalpy Economy Cycle, Lowered Space Temperature, Supply Air Temperature Reset, and Perimeter VAV (Case 11)

This case combines cases 3, 4, 5, and 6.

L. Base Case with Enthalpy Economy Cycle, Lowered Space Temperature, Supply Air Temperature Reset, and All VAV (Case 12)

This case combines cases 3, 4, 5, and 7.

M. Direct Expansion Cooling Systems (Case 13)

As stated previously, 12 packaged air handling units with direct expansion refrigerant coils supplied air to each of the 12 thermal zones. The input data of the cooling coil capacities were sized to satisfy the zone temperature profiles. Since this study compares the energy consumption levels at the coil input (not at the plant input, unitary or central), no distinction should be made as to the heating media--electric, hot water or gas heat. During store hours, 66°F (18.9°C) was maintained as the minimum space temperature.

N. Direct Expansion Cooling Systems with Enthalpy Economy Cycles (Case 14)

This case had exactly the same input data as case 13 except that enthalpy economy cycle was added to compare the outside air and return air enthalpies to set the outside and return air damper positions as described in case 3 previously.

3. RESULTS AND DISCUSSION

3.1 ENERGY CONSUMPTION RESULTS*

Tables 1 through 6 list the yearly heating, cooling, and fan energy consumptions of all the 14 control strategy cases for the six cities. In comparing the strategies, it should be noted again, that not all the cases were based on the same space temperature. Case 1 is the base case for chilled water systems. Cases 2 through 7 are simple strategy cases and cases 8 through 12 are combined strategy cases. Between the hours of 8 a.m. to 11 p.m. during the heating season, cases 1 through 4 and cases 6 and 7 (all single strategy cases) tried to maintain space temperature at 72°F (22.2°C), while case 5 (single strategy), cases 8 through 12 (all combined strategy cases), and cases 13 and 14 (DX cases) tried to maintain space temperature at 68°F (20°C). The input total pressure data were 0.5, 4.0, and 2.0 inches of H_2O for the return air fans, chilled water system supply air fans, and DX system supply air fans, respectively. Since the required fan pressures were duct layout dependent and these input data were quite arbitrary, it would be difficult to have meaningful comparisons between the fan energy consumptions between the chilled water units and the DX units. Therefore, the fan energy consumption data for the DX systems are not provided in tables 1 to 6. The fan energy consumption data for the chilled water systems are listed so that comparisons may be made between constant volume and VAV systems. If more DX systems of smaller sizes are compared to larger chilled water systems, the total fan energy consumption of the DX systems should be less than that of the chilled water systems. However, this study did not try to verify it.

Tables 7 through 12 give the yearly average heating, cooling, and fan energy consumptions per square foot of gross floor area of the retail store for the six cities. These figures may provide the readers with energy budget comparisons. Of course, the readers must take into account the building construction, internal loading and other factors which affect the energy consumption of this particular building. Also listed in these tables are the comparative energy consumption ratios of the individual cases to those of the base case.

3.2 USING DEGREE-DAYS AS ENERGY CONSUMPTION PARAMETERS

The yearly energy consumption of a building with a fixed internal load pattern depends on many external factors such as solar radiation, ambient temperature

*Energy consumption results are presented in English units only, for clarity.

and humidity, wind characteristics, etc., and their frequency of occurrence during the year. It is always desirable, from the application point of view, to have some parameters to correlate with the energy consumption data of one building so that the data may be applied to similar buildings of different climatic conditions. The authors of this paper realize the difficulty of finding these parameters to apply to commercial buildings such as the large retail store of this study. However, a parameter, even a crude one, may give HVAC designers assistance in deciding system and control strategies during the early course of the building design and provide budget figures for comparisons. This is not to say that the correlated data should be used indiscriminately. Detailed energy calculations with an appropriate method must be performed for a building in order to have accurate energy consumption values.

Evidently one of the most influential parameters affecting a building energy consumption is the outdoor ambient dry-bulb temperature. The dry-bulb temperature changes the building energy consumption through ventilation air and building structure thermal conductance directly, and through ambient humidity indirectly. Some measures of the dry-bulb temperature for a climatic location are daily average temperature, monthly average temperature, degree-hour data, frequency data (BINs), etc. However, the most available data for most of the cities in the United States are the heating and cooling degree-days. Therefore, in this study the energy consumption data of the six cities were correlated to the degree-day data provided in the Local Climatological Data [5]. Figures 4 through 10 show the least square regression lines of the single strategy yearly cooling energy consumptions of the six cities using 65°F based cooling degree-days as the independent variable. Figures 11 through 15 shows these lines for various combinations of control strategies. These individual cooling consumption lines are put on the same diagram in figures 18 and 19 for easier comparisons. Figures 20 through 34 are similar diagrams, except that they show the heating energy consumption plotted against 65°F based heating degree-days.

As mentioned previously, the ambient humidity level also affect the building energy consumption. It may be shown that parabolic curves may represent the cooling energy consumptions better than the straight lines because of the ambient humidity effects through the ventilation air and infiltration. For simplifying data analysis and application, only straight lines were used in the regression curve fitting.

3.3 COMPARISON OF STRATEGIES

A. Temperature economy cycle and enthalpy economy cycle

The energy consumption results and comparisons for these strategies may be seen from table 1 through 12. For single control strategies, temperature economy cycles were applied in two cases. Case 2 (figures 5 and 21) and case 8 (figures 11 and 24) were based on indoor heating design temperatures of 72°F (22.2°C) and 68 °F (20°C), respectively. The energy consumptions of these two cases should be compared to those of case 1, base case, (figures 4 and 20) and case 5 (figures 11 and 27) respectively. Case 3 (figures 6 and 22) was the result of applying enthalpy economy cycles to the base case. Figures 18 and 19 indicate that substantial cooling energy were saved and that these strategies were especially beneficial in low cooling degree-day areas. However, figures 33 and 34 show that the opposite is true: these strategies caused more heating energy to be used, especially for higher heating degree-day areas. Cases 9 and 10 were combined strategy cases where temperature and enthalpy economy cycles were applied respectively to lowered space temperature of 68°F (20°C) and varying discharge air temperature according to outside air temperature (see paragraph, 3.2.B).

The following table lists the energy comparison ratio of these cases as compared to cases 1 (base case, 72°F heating) and 5 (68°F heating).

	Lake Charles	Madison	Nashville	Santa Maria	Seattle	Washington, DC
Temperature economy (Case 2 vs Case 1, 72 °F heating)						
Heating	1.06	1.30	1.12	1.09	1.14	1.17
Cooling	.89	.57	.75	.69	.45	.67
Enthalpy economy (Case 3 vs Case 1, 72 °F heating)						
Heating	1.09	1.32	1.16	1.20	1.19	1.19
Cooling	.89	.46	.65	.35	.24	.56
Temperature economy (Case 8 vs Case 5, 68 °F heating)						
Heating	1.09	1.51	1.22	1.16	1.25	1.31
Cooling	.89	.60	.76	.69	.46	.68
Temperature economy with supply air temperature reset (Case 9 vs Case 5, 68 °F heating)						
Heating	.95	1.21	.99	.90	.93	1.06
Cooling	.86	.58	.74	.62	.42	.66
Enthalpy economy with supply air temperature reset (Case 10 vs Case 5, 68 °F heating)						
Heating	.99	1.24	1.04	1.06	1.00	1.09
Cooling	.81	.47	.66	.34	.24	.58

By applying the temperature economy cycles at different indoor heating design temperatures of 72°F (22.2°C) and 68°F (20°C) the cooling energy saving percentage were quite similar for cases 2 and 8. The heating energy increase which appeared in all single strategy economy cycles (cases 2, 3, and 8), was mainly caused by the decreased supply air temperature due to lowered cooling loads. Switching to enthalpy economy cycle from temperature economy cycle at 72°F (22.2°C) indoor design temperature increased cooling energy savings, compared to base case, from 31 percent in Santa Maria and 55 percent in Seattle to 65 percent and 76 percent, respectively. As pointed out in reference [1], Santa Maria and Seattle had many hours available for economy cycle operation. On the other hand, the cooling energy savings remained the same at 11 percent in Lake Charles.

When the control strategies were combined, heating energy were reduced substantially with further benefit of cooling energy reduction.

B. Supply Air Temperature Reset by Outside Air Temperature

Supply air temperatures were reset by the outside air temperatures in case 4 (fig. 7) to compare with case 1 (base case) and in case 9 (fig. 12) to compare with case 8 (lowered indoor heating design temperature with temperature economy cycle). In both cases, 4 and 9, the supply air temperatures varied between 60°F (15.6°C) and 62°F (16.7°C) in a linear relationship to the outside temperature of 75°F (23.9°C) and 32°F (0°C) respectively. Figures 7 and 23 show the cooling and heating consumptions of the former case and figures 12 and 28 depict those of the latter case. These comparisons are also shown in the following table.

	Lake Charles	Madison	Nashville	Santa Maria	Seattle	Washington, DC
Supply air temperature reset (Case 4 vs Case 1, 72°F heating)						
Heating	1.18	1.00	1.09	1.09	1.01	1.04
Cooling	1.01	.99	1.00	1.00	.98	.99
Supply air temperature reset (Case 9 vs Case 8, 68°F heating)						
Heating	.87	.80	.81	.99	.74	.81
Cooling	.98	.96	.97	.89	.91	.97

When the heating design temperature was 72°F (22.2°C) and no economy cycles were used, the cooling savings were minimum because of the small reset schedule 2°F (1.1°C). A larger reset schedule is difficult to achieve for this building, since the internal load is the dominant load of the building and the internal zones have larger floor areas. The heating energy consumption of this case (case 4) increased in most cities ranging from about even in Madison to 18 percent in Lake Charles. This increase was induced from higher space temperature of the reset strategy during the cooling seasons. When the heating design temperature was lowered to 68°F (20°C) and temperature economy cycles were used as for case 9, the cooling energy was reduced from 11 percent in Santa Maria to 2 percent in Lake Charles, the heating energy was reduced from 26 percent in Seattle to 13 percent in Lake Charles. Better results were obtained for case 9 than for case 4 mostly during the milder and cooler months when the economy cycle was in operation which lowered the cooling requirement and the cold deck temperature. Thus, more benefits were obtained by resetting.

C. Lower Space Heating Temperature

The following table show the ratio's of heating and cooling energy ratios of lowering space heating temperature from 72°F (22.°C) to 68°F (20°C).

	Lake Charles	Madison	Nashville	Santa Maria	Seattle	Washington, DC
Lower space heating temperature from 72°F to 68°F (Case 5 vs Case 1)						
Heating	.62	.62	.60	.55	.57	.62
Cooling	.96	.92	.95	.95	.94	.94

It is obvious that substantial heating energy was saved when the space temperature was lowered. Cooling energy was also reduced during low heating periods of the year when the cooling coil loads were decreased by having lower coil entering temperature. From tables 7 through 12, it can be seen that the air handling system fan energy was slightly increased for Madison (2 percent), Nashville (1 percent), and Washington, DC (1 percent), when the space temperatures were lowered. For these cities, because less heat was stored in daytime, longer fan operating hours at night in the heating season were needed to keep the space temperature above the night setback temperature.

D. Variable Air Volume (VAV) System

VAV systems were applied in several cases. Case 6 had VAV for perimeter zones only and case 7 applied VAV to all zones. Both cases 6 and 7 were based on 72°F (22.2°C) space reheat temperature. Therefore, they should be compared to case 1 (base case). Cases 11 and 12 were also for perimeter zone VAV and all zone VAV respectively, but they were added to case 10 which had 68 °F (20°C) space reheat temperature and used enthalpy economy cycle and supply air reset by outside temperature. These comparisons are as follows.

	Lake Charles	Madison	Nashville	Santa Maria	Seattle	Washington, DC
Perimeter zone VAV (Case 6 vs Case 1, 72°F heating)						
Heating	.56	.61	.60	.55	.57	.62
Cooling	.92	.80	.87	.95	.94	.94
All zone VAV (Case 7 vs Case 1, 72°F heating)						
Heating	.08	.23	.11	.08	.51	.57
Cooling	.78	.63	.72	.69	.80	.85
Perimeter zone VAV added to case 10 (Case 11 vs Case 10)						
Heating	1.11	1.02	1.02	1.00	.97	1.00
Cooling	.98	.93	.96	.85	.87	.95
All zone VAV added to case 10 (Case 12 vs Case 10)						
Heating	.11	.40	.20	.14	.22	.25
Cooling	.85	.78	.82	.72	.74	.80

Dramatic savings, for cases 6 and 7 were obtained by applying VAV's to the building. This was especially true when the entire building was under the VAV systems to take advantage of the internal load diversification. Less than 10 percent of the base heating energy were enough for Lake Charles and Santa Maria, if VAV's were used in all zones. Mixed results were evident for case 11. Further cooling energy reduction (case 11 vs case 10) were achieved by adding VAV's to the perimeter zones. However, some cities had more heating energy consumed, even though the increased values were small. This increased heating consumption was caused by the higher reheater discharge air temperature of the VAV zones, due to reduced air flow and heating coil loads. Among the six cities, from 18 percent to 25 percent of fan energy were saved for the all VAV case. It should be noted that these results were based on a minimum zone supply air of 40 percent of the constant volume system for all the six cities. No attempt was made to optimize the minimum air ratio or to compare the energy savings using the minimum air ratio as a parameter.

E. Direct Expansion (DX) Systems and DX Systems with Enthalpy Economy Cycle

The cooling energy consumption ratios of DX system with enthalpy economy cycles to those without economy cycles are listed below.

	Lake Charles	Madison	Nashville	Santa Maria	Seattle	Washington, DC
Enthalpy economy (Case 14 vs Case 13, DX Systems)						
Cooling	.86	.61	.75	.25	.26	.70

Similar to chilled water systems, the cooling consumption in lower cooling degree-day cities benefited more percentage-wise than higher cooling degree-day areas. The heating energy was the same in both cases. This was different from the chilled water reheat systems where heating energy was increased when economy cycles were used to reheat the lowered cooling coil discharge air temperature to the desired room temperature. A close comparison between DX system and the chilled water system may not be appropriate, since the zoning arrangement and the sizing of cooling coils were not exactly the same, although the same internal loading input data were used in both cases. The closest comparison of cooling energy between the two types of systems are case 12 (space temperature at 68°F (20°C) during the heating season with enthalpy economy cycle, supply air resetting by outside temperature, and all zones under VAV) and case 14 (DX with enthalpy economy cycle). If the cooling energy of case 12 is represented by unity, then the ratio of the two cases are as follows.

	Lake Charles	Madison	Nashville	Santa Maria	Seattle	Washington, DC
Case 12 (all zone VAV with enthalpy economy, supply air temperature reset, and 68°F heating) vs Case 14 (DX with enthalpy economy)						
	1.12	.91	.97	.63	.72	.96

In theory, the cooling energy consumption of these two cases should be close, since both cases try to match the system capacities to the space loads. The bottom two lines in figure 19 show this relationship. The larger difference in lower degree-day cities was probably caused by using the same minimum supply air ratio (.4) for all cities. Except for Santa Maria and Seattle, the difference in energy consumption (as well as savings) between the two strategies were within 3 percent as compared to the base case. Heating energy comparison between these two strategies is not appropriate, since there was a 2°F (1.1°C) difference in input data for the space temperature during heating season.

4. SUMMARY

The BLAST computer program was used to compute the yearly energy consumption of a large two-story retail store of heavy architectural construction. The mechanical systems modeled were reheat, VAV, and DX systems. The control strategies tested were temperature and enthalpy economy cycles, supply air temperature reset by outside temperature, lowering space temperature during heating season, applying VAV systems to perimeter and all thermal zones, and a few combination strategies. Cities modeled covered a variety of climatic conditions. Tables 1 through 6 list the yearly heating, cooling, and electric energy consumptions of the six cities. Tables 7 through 14 list these energy consumptions per unit gross floor area of the store as well as the normalized energy consumptions for all these strategies using the base case consumption as unity.

The cooling degree-days of these cities range from 84 in Santa Maria, CA to 2739 in Lake Charles, LA. The heating degree-days range from 1498 in Lake Charles to 7730 in Madison, WI. The cooling and heating energy consumptions of all the strategies were also plotted in figures 4 through 17 and from 20 through 32, respectively, using degree-days as parameters.

Figure 18 compares cooling energy consumption of single strategy cases 1 through 7. Ranking energy savings, from the least to the most, were base case, lowering space heating temperature to 68°F (20°C), applying VAV system (40 percent minimum) to perimeter zones, temperature economy cycle, and enthalpy economy cycle. Applying the VAV system (40 percent minimum) to all zones saved considerable amounts of energy across all cities by roughly equal amounts. Resetting of supply air temperature by outside air temperatures from 60°F (15.6°C) to 62°F (16.7°C) corresponding to 75°F (23.9°C) to 32°F (0°C) outside temperature gave about the same cooling consumption as the base case. Figure 19 compares the the combination strategy cases and the DX systems. The best cooling energy saving strategies were the combination of 68°F (20°C) space temperature for heating, resetting supply air temperature by outside air temperatures, enthalpy economy cycle, and applying VAV to all zones for the chilled water system (case 12) and enthalpy economy cycled DX system. Except for Lake Charles, DX system cooling energy was slightly lower than the chilled water system. At Lake Charles, the cooling consumption difference between the two systems was less than two percent. For the chilled water system alone, the least energy consuming strategies (case 12) could reduce the cooling energy consumption from 35 percent in Lake Charles to 83 percent in Seattle, as compared to the base case.

As to heating energy consumption (see figures 33 and 34), temperature and enthalpy economy cycles alone caused it to increase. Lowering space heating temperature and applying VAV to perimeter and all zones saved a great deal of energy. With VAV applied to all zones alone, heating savings were from 77 percent in Madison to 92 percent in Lake Charles and Santa Maria. Resetting of supply air temperature caused most cities to use more energy with the exception of Madison which had about the same consumption as the base case. The most beneficial combination of strategies for heating was case 12 which reduced the yearly heating energy, as compared to the base case, from 69 percent in Madison to 93 percent in Lake Charles. At lower degree day areas, cases 12, 13, and 14 all had very low heating energy consumptions.

The most favored strategy for reducing fan energy consumption was the all zone VAV. The savings ranged from 33 percent in Lake Charles to 41 percent in Madison. As stated in section 3.1, no comparison was made for the DX systems.

It should be stated again that the effect of energy consumption resulting from the control loop dynamics were not considered in this study and that the data presented in this report was based on a particular building with a particular set of orientation, construction, internal loading, and schedules. One should keep this in mind when using these data for preliminary design purposes or for comparative control strategy selections. The control strategy and system selections also depend on the relative costs of the energy sources, plant efficiencies, transmission losses, etc. There is no substitute for determining building energy consumptions by a detailed energy and economics analysis using final design drawings.

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Table 1. Annual energy consumption - Lake Charles, LA

CASE	STRATEGY	HEATING ENERGY Btu x 10^9	COOLING ENERGY Btu x 10^9	FAN ENERGY Btu x 10^9
1	Base (72°F heating)	1.81	13.23	2.41
2	Case 1 + temperature economy	1.91	11.71	2.41
3	Case 1 + enthalpy economy	1.98	10.68	2.41
4	Case 1 + supply air temperature reset by outside air	2.15	13.41	2.41
5	Case 1 + 68 °F heating temperature	1.12	12.76	2.41
6	Case 1 + perimeter zone VAV	1.01	12.15	1.98
7	Case 1 + all zone VAV	0.14	10.31	1.61
8	Case 5 + temperature economy	1.22	11.40	2.31
9	Case 8 + supply air temperature reset by outside air	1.06	11.17	2.41
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	1.11	10.36	2.41
11	Case 10 + perimeter zone VAV	1.23	10.16	1.98
12	Case 10 + all zone VAV	0.12	8.63	1.62
13	DX (66°F heating)	0.05	10.25	N.A.
14	DX + enthalpy economy	0.05	8.77	N.A.

Table 2. Annual energy consumption - Madison, WI

CASE	STRATEGY	HEATING ENERGY Btu x 10^9	COOLING ENERGY Btu x 10^9	FAN ENERGY Btu x 10^9
1	Base (72°F heating)	3.22	9.30	2.64
2	Case 1 + temperature economy	4.17	5.33	2.64
3	Case 1 + enthalpy economy	4.24	4.24	2.64
4	Case 1 + supply air temperature reset by outside air	3.21	8.98	2.64
5	Case 1 + 68 °F heating temperature	2.01	8.57	2.70
6	Case 1 + perimeter zone VAV	1.97	7.46	1.99
7	Case 1 + all zone VAV	0.74	5.86	1.54
8	Case 5 + temperature economy	3.04	5.14	2.70
9	Case 8 + supply air temperature reset by outside air	2.44	4.94	2.70
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	2.49	4.07	2.70
11	Case 10 + perimeter zone VAV	2.53	3.77	2.01
12	Case 10 + all zone VAV	1.00	3.16	1.57
13	DX (66°F heating)	1.79	4.72	N.A.
14	DX + enthalpy economy	1.79	2.89	N.A.

Table 3. Annual energy consumption - Nashville, TN

CASE	STRATEGY	HEATING ENERGY Btu x 10^9	COOLING ENERGY Btu x 10^9	FAN ENERGY Btu x 10^9
1	Base (72°F heating)	2.22	11.33	2.39
2	Case 1 + temperature economy	2.49	8.45	2.39
3	Case 1 + enthalpy economy	2.57	7.31	2.39
4	Case 1 + supply air temperature reset by outside air	2.41	11.30	2.39
5	Case 1 + 68 °F heating temperature	1.34	10.77	2.42
6	Case 1 + perimeter zone VAV	1.21	9.89	1.90
7	Case 1 + all zone VAV	0.25	8.19	1.53
8	Case 5 + temperature economy	1.64	8.19	2.42
9	Case 8 + supply air temperature reset by outside air	1.33	7.97	2.42
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	1.39	7.09	2.42
11	Case 10 + perimeter zone VAV	1.42	6.82	1.90
12	Case 10 + all zone VAV	0.28	5.82	1.54
13	DX (66°F heating)	0.33	7.46	N.A.
14	DX + enthalpy economy	0.33	5.62	N.A.

Table 4. Annual energy consumption - Santa Maria, CA

CASE	STRATEGY	HEATING ENERGY Btu x 10^9	COOLING ENERGY Btu x 10^9	FAN ENERGY Btu x 10^9
1	Base (72°F heating)	1.81	9.51	2.21
2	Case 1 + temperature economy	1.97	6.56	2.21
3	Case 1 + enthalpy economy	2.18	3.35	2.21
4	Case 1 + supply air temperature reset by outside air	1.98	9.51	2.21
5	Case 1 + 68 °F heating temperature	0.99	9.06	2.21
6	Case 1 + perimeter zone VAV	0.85	7.96	1.74
7	Case 1 + all zone VAV	0.14	6.58	1.41
8	Case 5 + temperature economy	1.15	6.27	2.21
9	Case 8 + supply air temperature reset by outside air	0.89	5.61	2.21
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	1.05	3.10	2.21
11	Case 10 + perimeter zone VAV	1.05	2.63	1.72
12	Case 10 + all zone VAV	0.15	2.24	1.41
13	DX (66°F heating)	0.07	5.56	N.A.
14	DX + enthalpy economy	0.07	1.41	N.A.

Table 5. Annual energy consumption - Seattle, WA

CASE	STRATEGY	HEATING ENERGY Btu x 10^9	COOLING ENERGY Btu x 10^9	FAN ENERGY Btu x 10^9
1	Base (72°F heating)	2.37	8.88	2.25
2	Case 1 + temperature economy	2.71	3.98	2.25
3	Case 1 + enthalpy economy	2.82	2.16	2.25
4	Case 1 + supply air temperature reset by outside air	2.40	8.71	2.25
5	Case 1 + 68 °F heating temperature	1.34	8.34	2.26
6	Case 1 + perimeter zone VAV	1.20	7.14	1.71
7	Case 1 + all zone VAV	0.30	5.72	1.36
8	Case 5 + temperature economy	1.68	3.82	2.26
9	Case 8 + supply air temperature reset by outside air	1.25	3.49	2.26
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	1.34	2.03	2.26
11	Case 10 + perimeter zone VAV	1.30	1.77	1.70
12	Case 10 + all zone VAV	0.30	1.51	1.38
13	DX (66°F heating)	0.38	4.15	N.A.
14	DX + enthalpy economy	0.38	1.08	N.A.

Table 6. Annual energy consumption - Washington, DC

CASE	STRATEGY	HEATING ENERGY Btu x 10^9	COOLING ENERGY Btu x 10^9	FAN ENERGY Btu x 10^9
1	Base (72°F heating)	2.65	10.67	2.47
2	Case 1 + temperature economy	3.09	7.10	2.47
3	Case 1 + enthalpy economy	3.16	6.01	2.47
4	Case 1 + supply air temperature reset by outside air	2.76	10.53	2.47
5	Case 1 + 68 °F heating temperature	1.63	10.03	2.51
6	Case 1 + perimeter zone VAV	1.50	9.02	1.91
7	Case 1 + all zone VAV	0.38	7.27	1.51
8	Case 5 + temperature economy	2.13	6.87	2.51
9	Case 8 + supply air temperature reset by outside air	1.72	6.65	2.51
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	1.77	5.79	2.51
11	Case 10 + perimeter zone VAV	1.77	5.51	1.92
12	Case 10 + all zone VAV	0.45	4.63	1.53
13	DX (66°F heating)	0.64	6.33	N.A.
14	DX + enthalpy economy	0.64	4.46	N.A.

Table 7. Comparative annual energy consumption - Lake Charles, LA

CASE	STRATEGY	HEATING ENERGY	COOLING ENERGY	FAN ENERGY
		Consumption, 10^3 Btu/ft^2 Ratio-relative to Case 1		
1	Base (72°F heating)	11.78 1.00	86.13 1.00	15.69 1.00
2	Case 1 + temperature economy	12.43 1.06	76.24 0.89	15.69 1.00
3	Case 1 + enthalpy economy	12.89 1.09	69.53 0.81	15.69 1.00
4	Case 1 + supply air temperature reset by outside air	14.00 1.19	87.30 1.01	15.69 1.00
5	Case 1 + 68 °F heating temperature	7.29 0.62	83.07 0.96	15.69 1.00
6	Case 1 + perimeter zone VAV	6.58 0.56	79.10 0.92	12.89 0.82
7	Case 1 + all zone VAV	0.91 0.08	67.12 0.78	10.48 0.67
8	Case 5 + temperature economy	7.94 0.67	74.22 0.86	15.69 1.00
9	Case 8 + supply air temperature reset by outside air	6.90 0.59	72.72 0.84	15.69 1.00
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	7.23 0.61	67.45 0.78	15.69 1.00
11	Case 10 + perimeter zone VAV	8.01 0.68	66.15 0.77	12.89 0.82
12	Case 10 + all zone VAV	0.781 0.07	56.18 0.65	10.55 0.67
13	DX (66°F heating)	0.33 0.03	66.73 0.78	N.A.
14	DX + enthalpy economy	0.33 0.03	57.10 0.66	N.A.

Table 8. Comparative annual energy consumption - Madison, Wisconsin

CASE	STRATEGY	HEATING ENERGY	COOLING ENERGY	FAN ENERGY
		Consumption, 10^3 Btu/ft^2 Ratio-relative to Case 1		
1	Base (72°F heating)	20.96 1.00	60.55 1.00	17.19 1.00
2	Case 1 + temperature economy	27.15 1.30	34.70 0.57	17.19 1.00
3	Case 1 + enthalpy economy	27.60 1.32	20.60 0.46	17.19 1.00
4	Case 1 + supply air temperature reset by outside air	20.90 1.00	58.46 0.97	17.19 1.00
5	Case 1 + 68 °F heating temperature	13.09 0.62	55.79 0.92	17.58 1.02
6	Case 1 + perimeter zone VAV	12.83 0.61	48.57 0.80	12.96 0.75
7	Case 1 + all zone VAV	4.82 0.23	38.15 0.63	10.03 0.58
8	Case 5 + temperature economy	19.79 0.94	33.46 0.55	17.19 1.00
9	Case 8 + supply air temperature reset by outside air	15.89 0.76	32.16 0.53	17.19 1.00
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	16.21 0.77	26.50 0.44	17.19 1.00
11	Case 10 + perimeter zone VAV	16.47 0.79	24.54 0.41	13.09 0.76
12	Case 10 + all zone VAV	6.51 0.31	20.57 0.34	10.22 0.59
13	DX (66°F heating)	11.65 0.56	30.73 0.51	N.A.
14	DX + enthalpy economy	11.65 0.56	18.81 0.31	N.A.

Table 9. Comparative annual energy consumption - Nashville, TN

CASE	STRATEGY	HEATING ENERGY	COOLING ENERGY	FAN ENERGY
		Consumption, 10^3 Btu/ft^2 Ratio-relative to Case 1		
1	Base (72°F heating)	14.45 1.00	73.76 1.00	15.56 1.00
2	Case 1 + temperature economy	16.21 1.12	55.01 0.75	15.56 1.00
3	Case 1 + enthalpy economy	16.73 1.16	47.59 0.65	1.00 1.00
4	Case 1 + supply air temperature reset by outside air	15.69 1.09	73.57 1.00	15.56 1.00
5	Case 1 + 68 °F heating temperature	8.72 0.60	70.12 0.95	15.76 1.01
6	Case 1 + perimeter zone VAV	7.88 0.55	64.39 0.87	12.37 0.79
7	Case 1 + all zone VAV	1.63 0.11	53.32 0.72	9.96 0.64
8	Case 5 + temperature economy	10.68 0.74	53.32 0.72	15.76 1.01
9	Case 8 + supply air temperature reset by outside air	8.66 0.60	51.89 0.70	15.76 1.01
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	9.05 0.63	46.16 0.63	15.76 1.01
11	Case 10 + perimeter zone VAV	9.24 0.64	44.40 0.60	12.37 0.79
12	Case 10 + all zone VAV	1.82 0.13	37.89 0.51	10.03 0.64
13	DX (66°F heating)	2.15 0.15	48.57 0.66	N.A
14	DX + enthalpy economy	2.15 0.15	36.59 0.50	N.A

Table 10. Comparative annual energy consumption - Santa Maria, CA

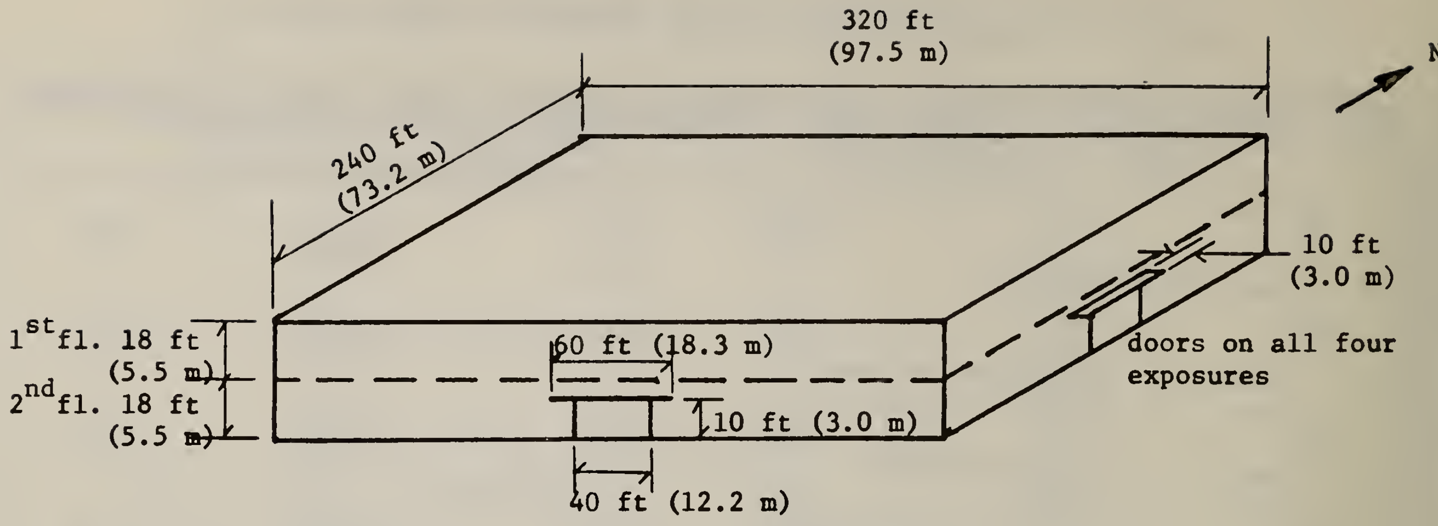
CASE	STRATEGY	HEATING ENERGY	COOLING ENERGY	FAN ENERGY
		Consumption, 10^3 Btu/ft^2 Ratio-relative to Case 1		
1	Base (72°F heating)	11.78 1.00	61.91 1.00	14.39 1.00
2	Case 1 + temperature economy	12.83 1.09	42.71 0.69	14.39 1.00
3	Case 1 + enthalpy economy	14.19 1.20	21.81 0.35	14.39 1.00
4	Case 1 + supply air temperature reset by outside air	12.89 1.09	61.91 1.00	14.39 1.00
5	Case 1 + 68 °F heating temperature	6.45 0.55	58.98 0.95	14.39 1.00
6	Case 1 + perimeter zone VAV	5.53 0.47	51.82 0.84	11.33 0.79
7	Case 1 + all zone VAV	0.91 0.08	42.84 0.69	9.18 0.64
8	Case 5 + temperature economy	7.49 0.64	40.82 0.66	14.39 0.65
9	Case 8 + supply air temperature reset by outside air	5.79 0.49	36.52 0.59	14.39 0.65
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	6.84 0.58	20.18 0.33	14.39 0.65
11	Case 10 + perimeter zone VAV	6.84 0.58	17.12 0.28	11.20 0.78
12	Case 10 + all zone VAV	0.98 0.08	14.58 0.24	9.31 0.64
13	DX (66°F heating)	0.46 0.04	36.20 0.58	N.A.
14	DX + enthalpy economy	0.46 0.04	9.18 0.15	N.A.

Table 11. Comparative annual energy consumption - Seattle, WA

CASE	STRATEGY	HEATING ENERGY	COOLING ENERGY	FAN ENERGY
		Consumption, 10^3 Btu/ft^2 Ratio-relative to Case 1		
1	Base (72°F heating)	15.43 1.00	57.81 1.00	14.65 1.00
2	Case 1 + temperature economy	17.64 1.14	25.91 0.45	14.65 1.00
3	Case 1 + enthalpy economy	18.36 1.19	14.06 0.24	14.65 1.00
4	Case 1 + supply air temperature reset by outside air	15.63 1.01	56.71 0.98	14.65 1.00
5	Case 1 + 68 °F heating temperature	8.72 0.57	54.30 0.94	14.71 1.00
6	Case 1 + perimeter zone VAV	7.81 0.51	46.48 0.80	11.13 0.76
7	Case 1 + all zone VAV	1.95 0.13	37.24 0.64	8.85 0.60
8	Case 5 + temperature economy	10.94 0.71	24.87 0.43	14.71 1.00
9	Case 8 + supply air temperature reset by outside air	8.14 0.53	22.72 0.39	14.71 1.00
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	8.72 0.57	13.22 0.23	14.71 1.00
11	Case 10 + perimeter zone VAV	8.46 0.55	11.52 0.20	11.07 0.76
12	Case 10 + all zone VAV	1.95 0.13	9.83 0.17	8.98 0.61
13	DX (66°F heating)	2.47 0.16	27.02 0.47	N.A
14	DX + enthalpy economy	2.47 0.16	7.03 0.12	N.A.

Table 12. Comparative annual energy consumption - Washington, DC

CASE	STRATEGY	HEATING ENERGY	COOLING ENERGY	FAN ENERGY
		Consumption, 10^3 Btu/ft^2 Ratio-relative to Case 1		
1	Base (72°F heating)	17.25 1.00	69.47 1.00	16.08 1.00
2	Case 1 + temperature economy	20.12 1.17	46.22 0.67	16.08 1.00
3	Case 1 + enthalpy economy	20.57 1.19	39.13 0.56	16.08 1.00
4	Case 1 + supply air temperature reset by outside air	17.97 1.04	68.55 0.99	16.08 1.00
5	Case 1 + 68 °F heating temperature	10.61 0.62	65.30 0.94	16.34 1.02
6	Case 1 + perimeter zone VAV	9.77 0.57	58.72 0.85	12.43 0.77
7	Case 1 + all zone VAV	2.47 0.14	47.33 0.68	9.83 0.61
8	Case 5 + temperature economy	2.47 0.14	47.33 0.68	9.83 0.61
9	Case 8 + supply air temperature reset by outside air	11.20 0.65	43.29 0.62	16.34 1.02
10	Case 5 + enthalpy economy + supply air temperature reset by outside air	11.52 0.67	37.70 0.54	16.34 1.02
11	Case 10 + perimeter zone VAV	11.52 0.67	35.87 0.52	12.50 0.78
12	Case 10 + all zone VAV	2.93 0.17	30.14 0.43	9.96 0.62
13	DX (66°F heating)	4.17 0.24	41.21 0.59	N.A.
14	DX + enthalpy economy	4.17 0.24	29.04 0.42	N.A.



Exterior Walls:

1st floor: 12" (.3m) concrete masonry units, 2" (.05m) insulation K = 0.27 Btu/h • ft • °F (.00389 w/m • °K), and 1/2" (.013m) gypsum board

2nd floor: same as 1st floor except 8" (.2m) concrete masonry units.

Roof:

Built-up roof, 3" (.07m) insulation K = 0.158 Btu/h • ft • °F (.00228 w/m • °K), 3 1/2" (.09m) light weight concrete, metal deck, air space, and 1/2" (.01m) ceiling panels. Ceiling height 12' (3.66m).

Floor:

1st floor: 5" (.13m) concrete slab on grade and carpeting
2nd floor: 9" (.23m) concrete base, 3 1/2" (.09m) light weight concrete and carpeting

Door:

Tempered glass doors

Figure 1. Large retail store model

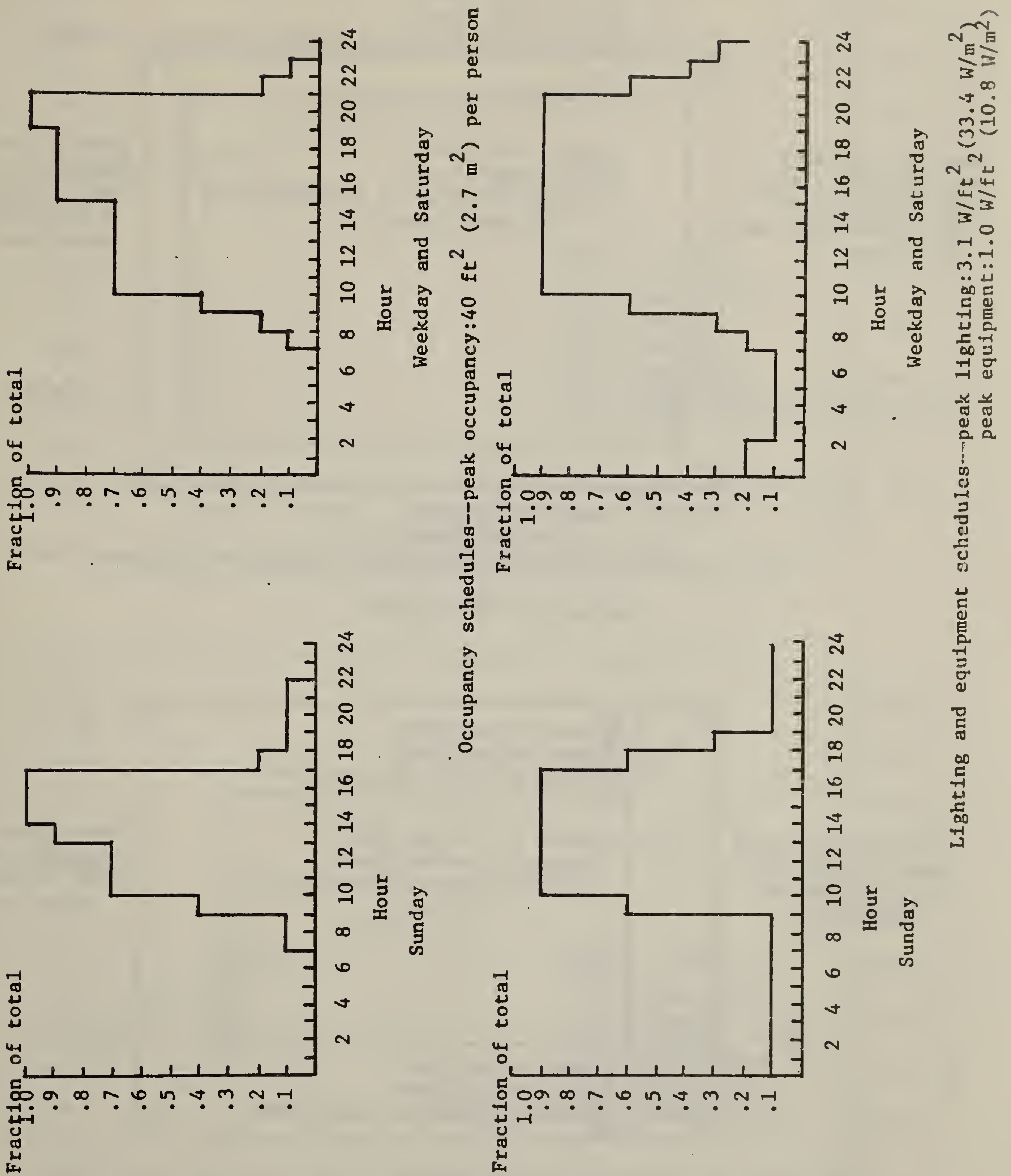
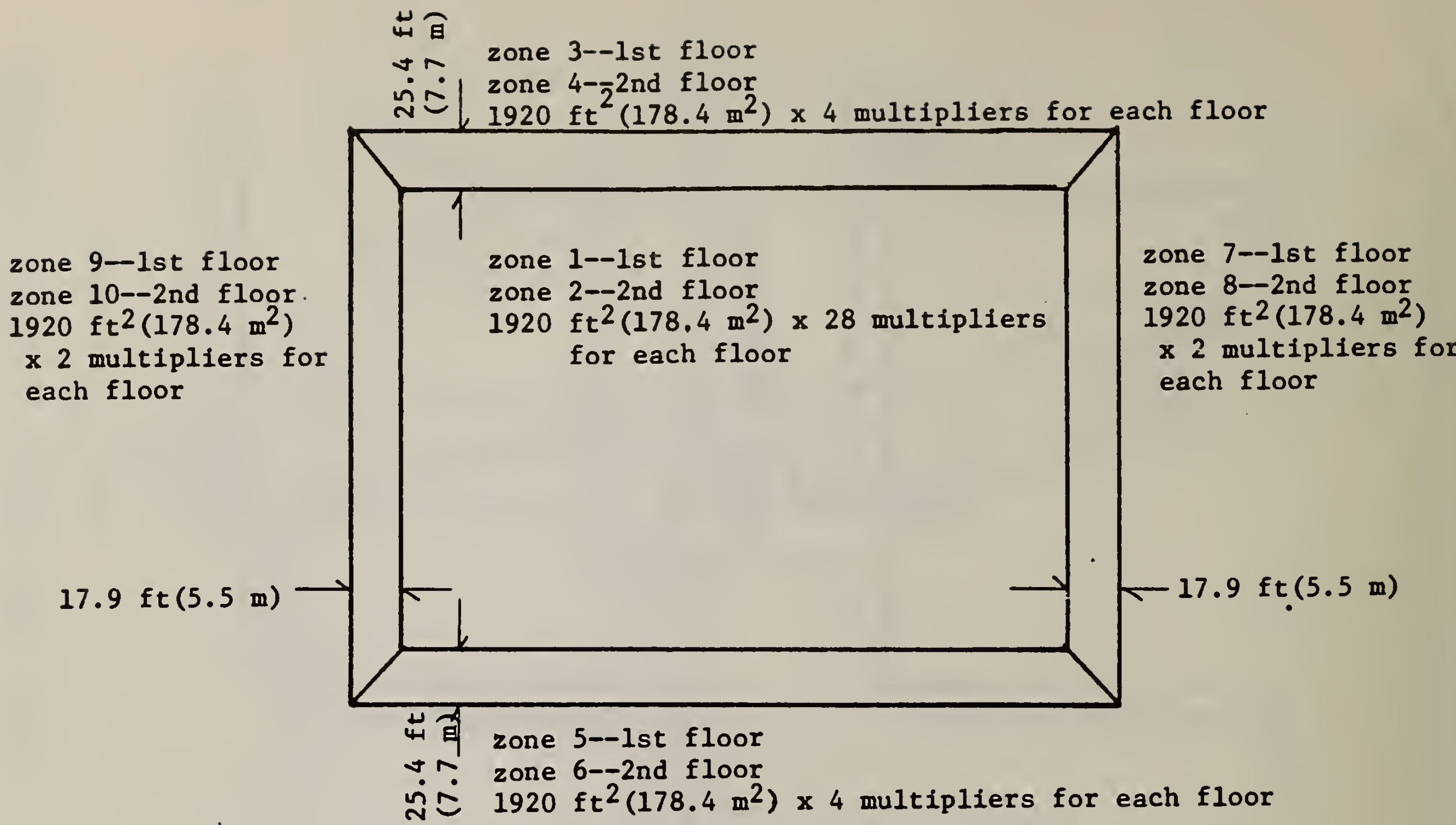
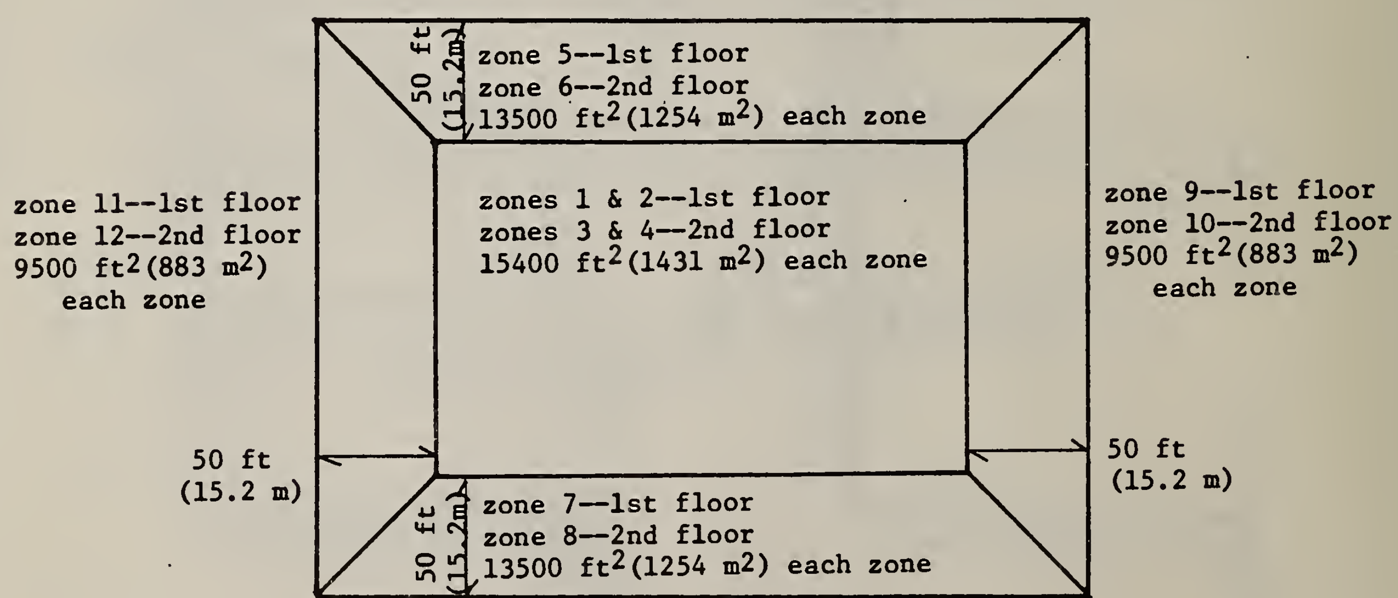


Figure 2. Internal load profiles



Reheat and VAV systems



DX systems

Figure 3. Thermal zone diagrams

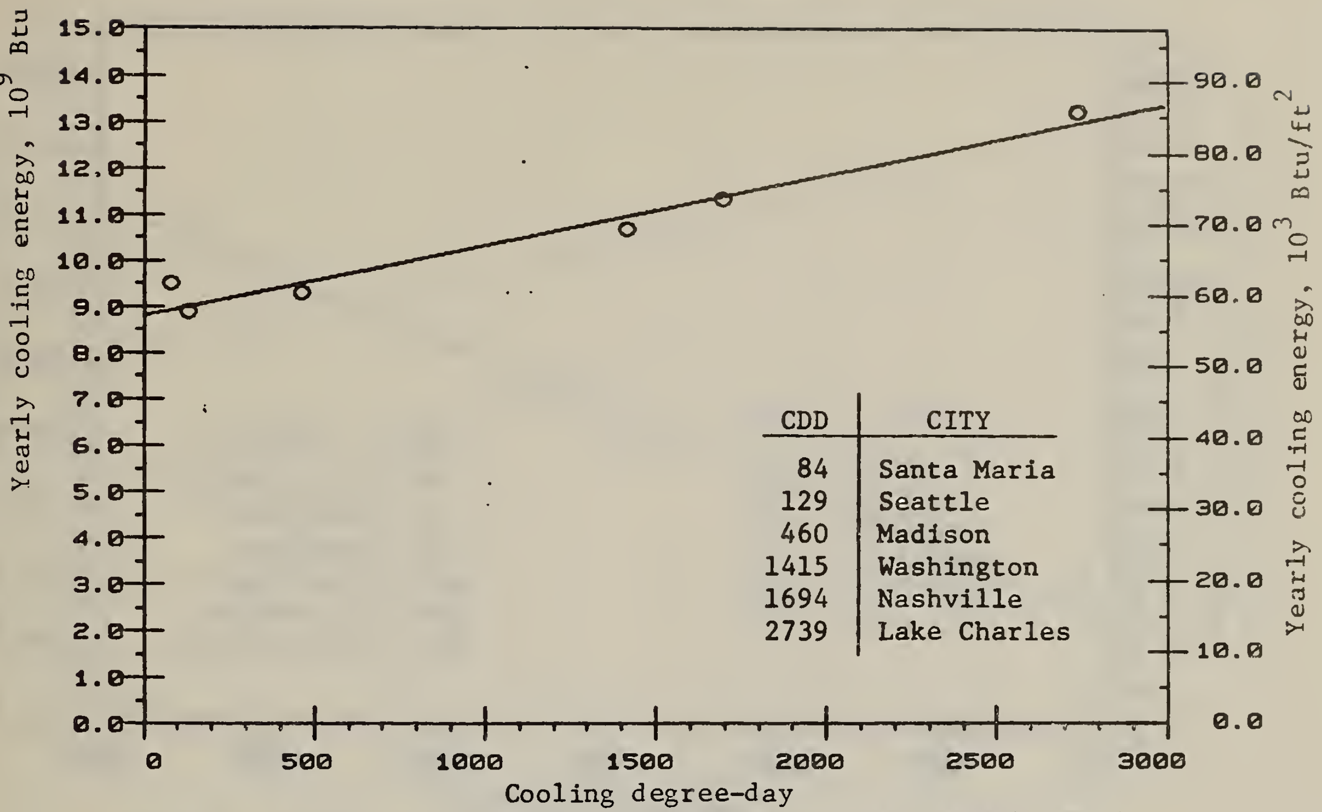


Figure 4. Cooling energy consumption of base case (Case 1)

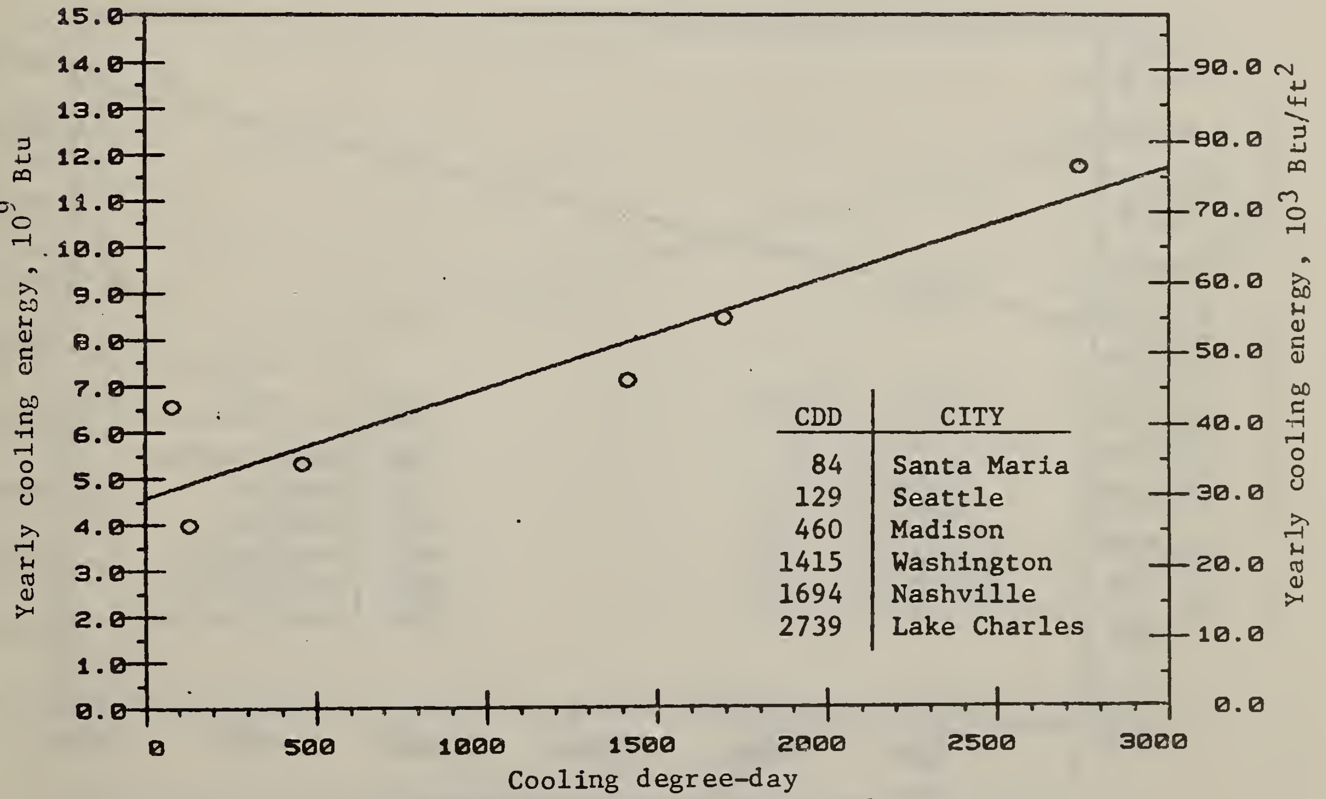


Figure 5. Cooling energy consumption of applying temperature economy cycle (Case 2)

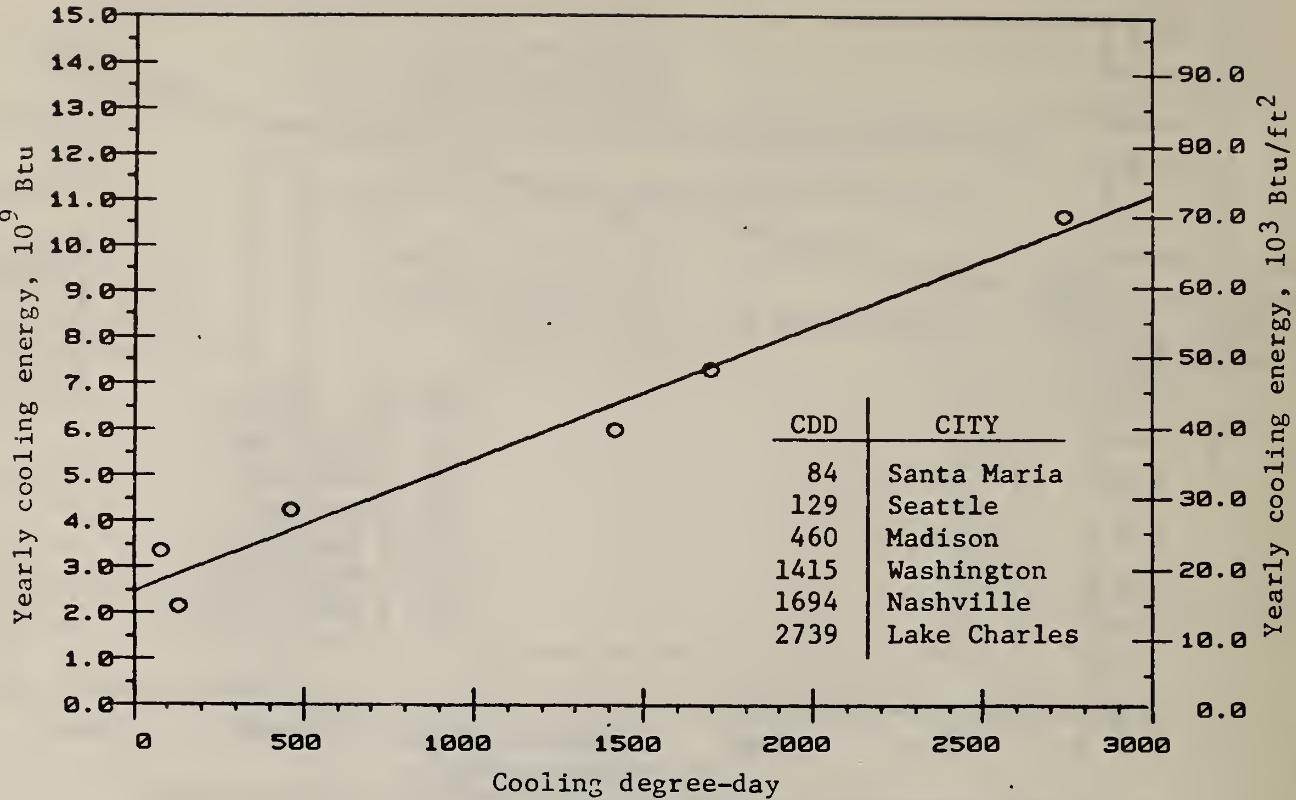


Figure 6. Cooling energy consumption of applying enthalpy economy cycle (Case 3)

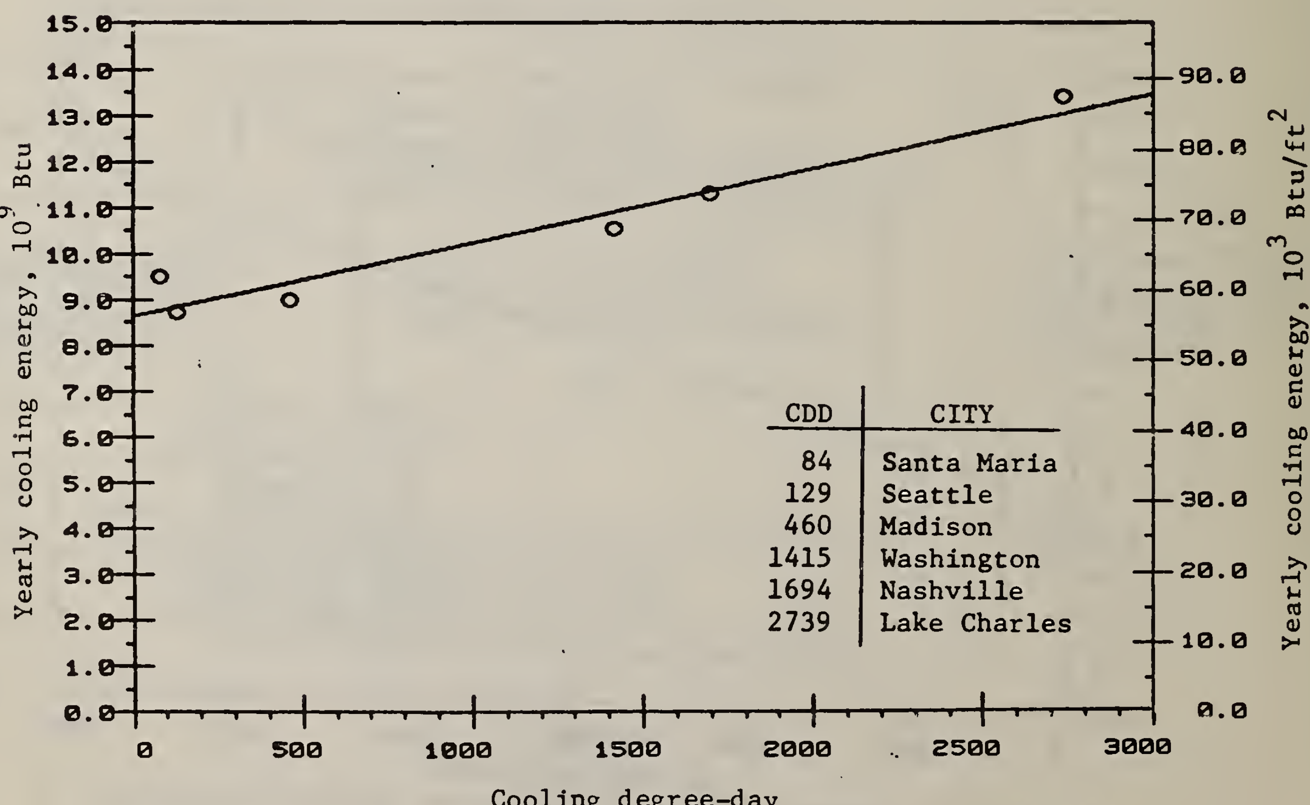


Figure 7. Cooling energy consumption of applying outside air resetting supply air temperature (Case 4)

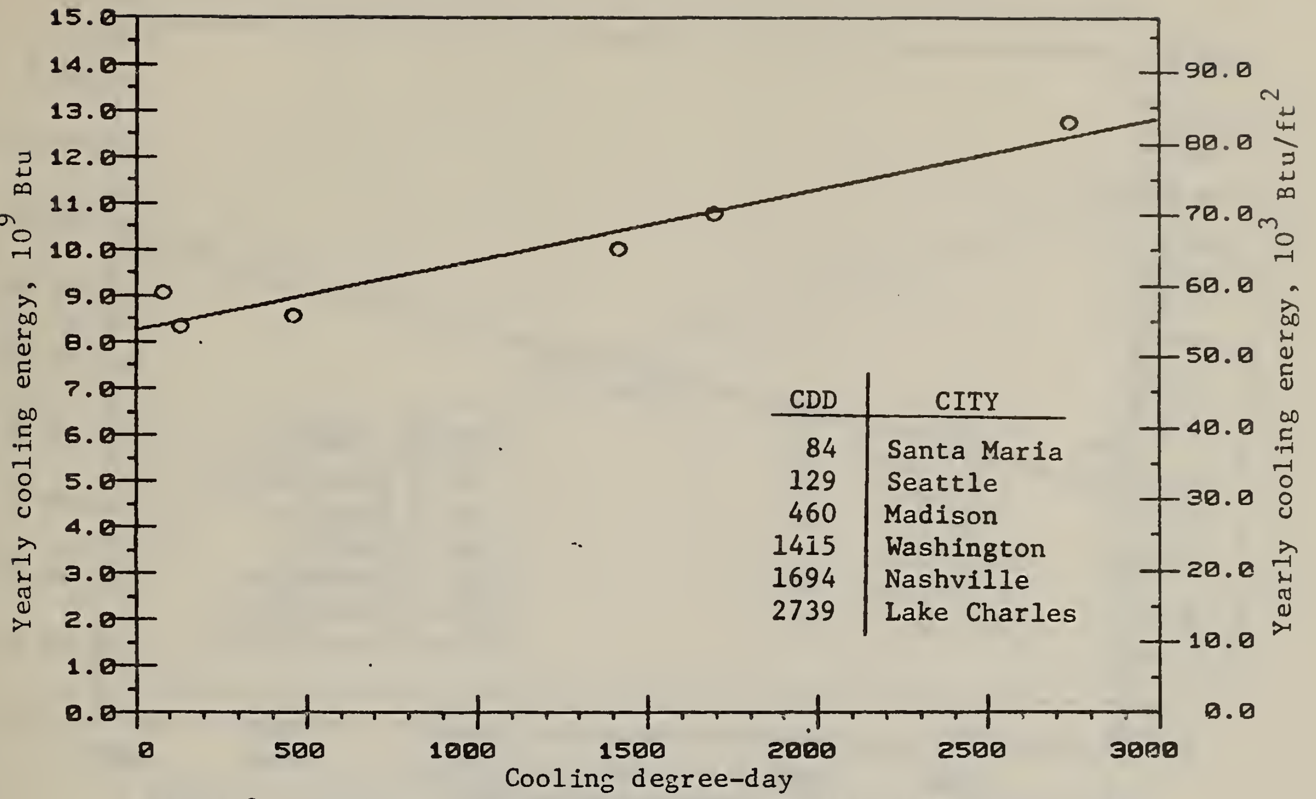


Figure 8. Cooling energy consumption of base case with lowered space heating temperature (Case 5)

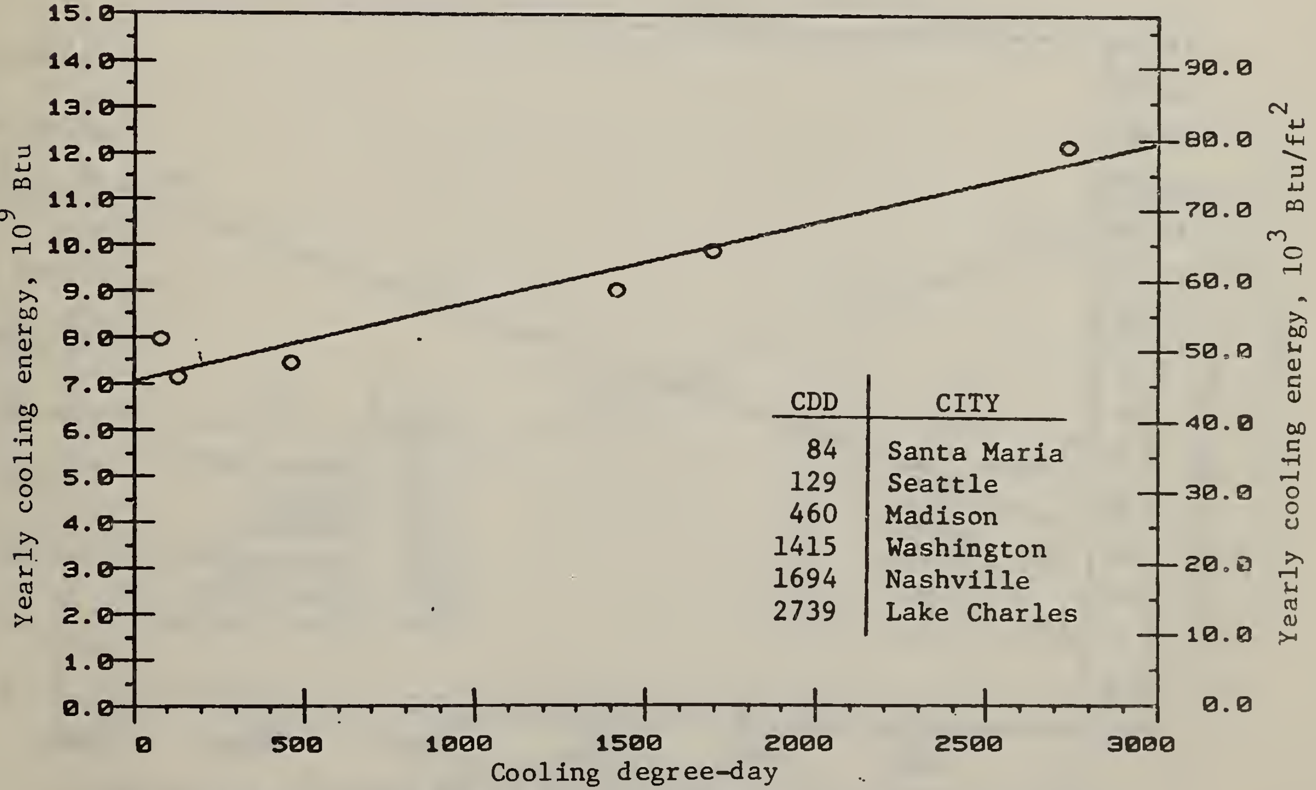


Figure 9. Cooling energy consumption of applying VAV to perimeter zones (Case 6)

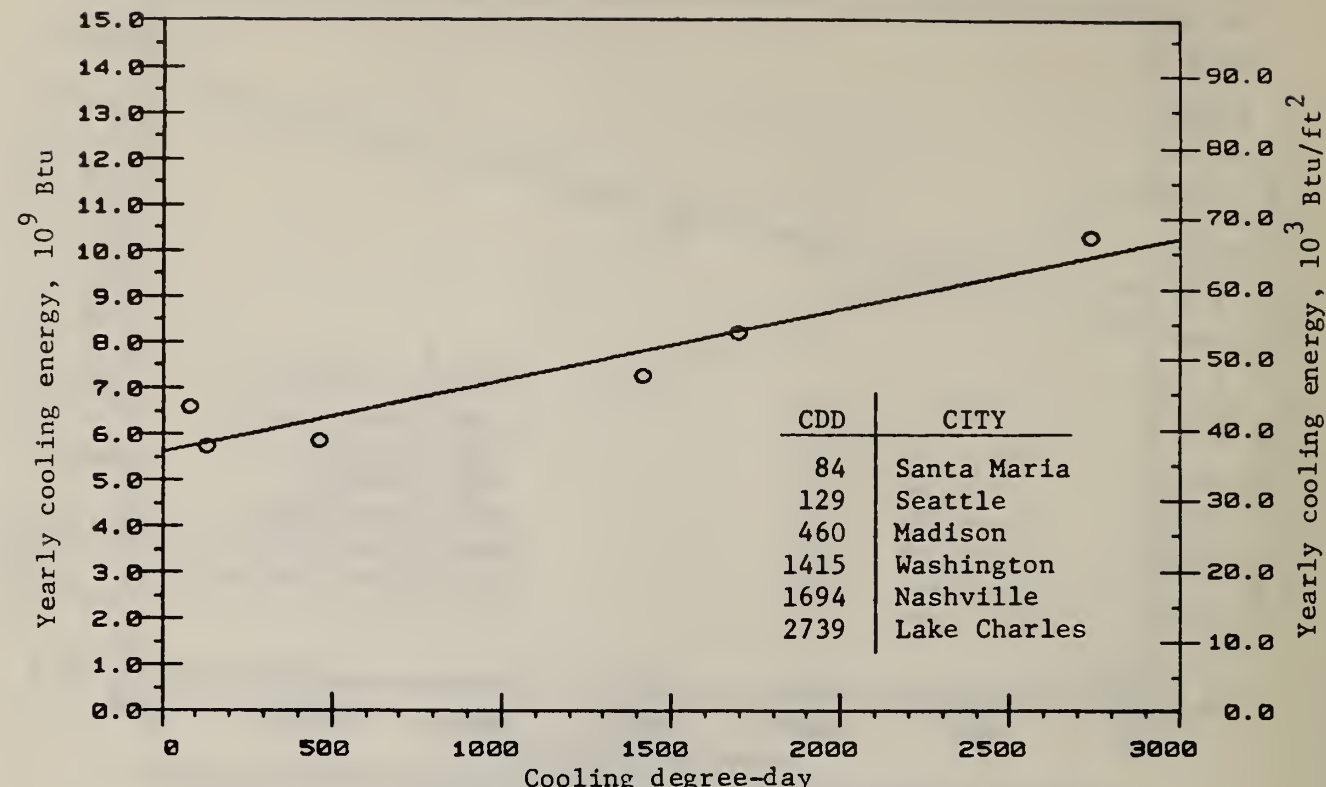


Figure 10. Cooling energy consumption of applying VAV to all zones (Case 7)

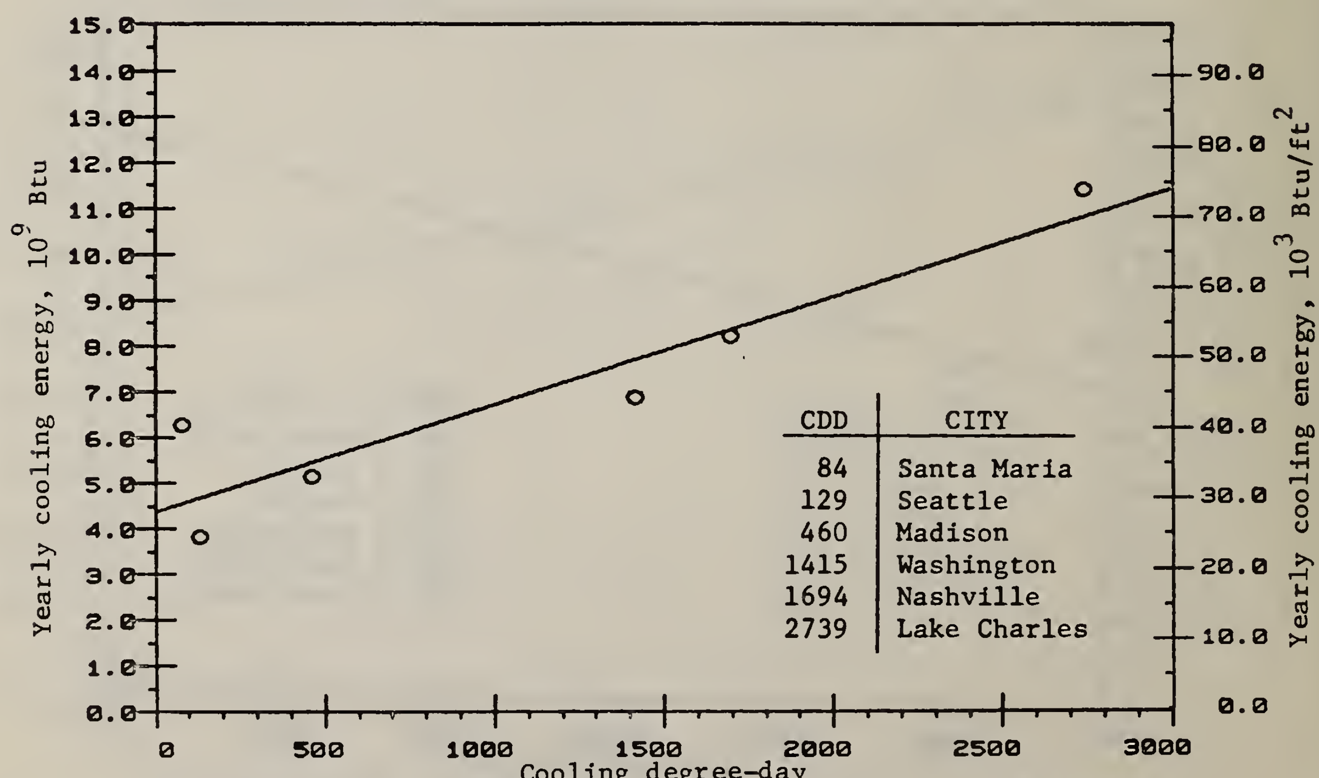


Figure 11. Cooling energy consumption of applying lowered space heating temperature and temperature economy cycle (Case 8)

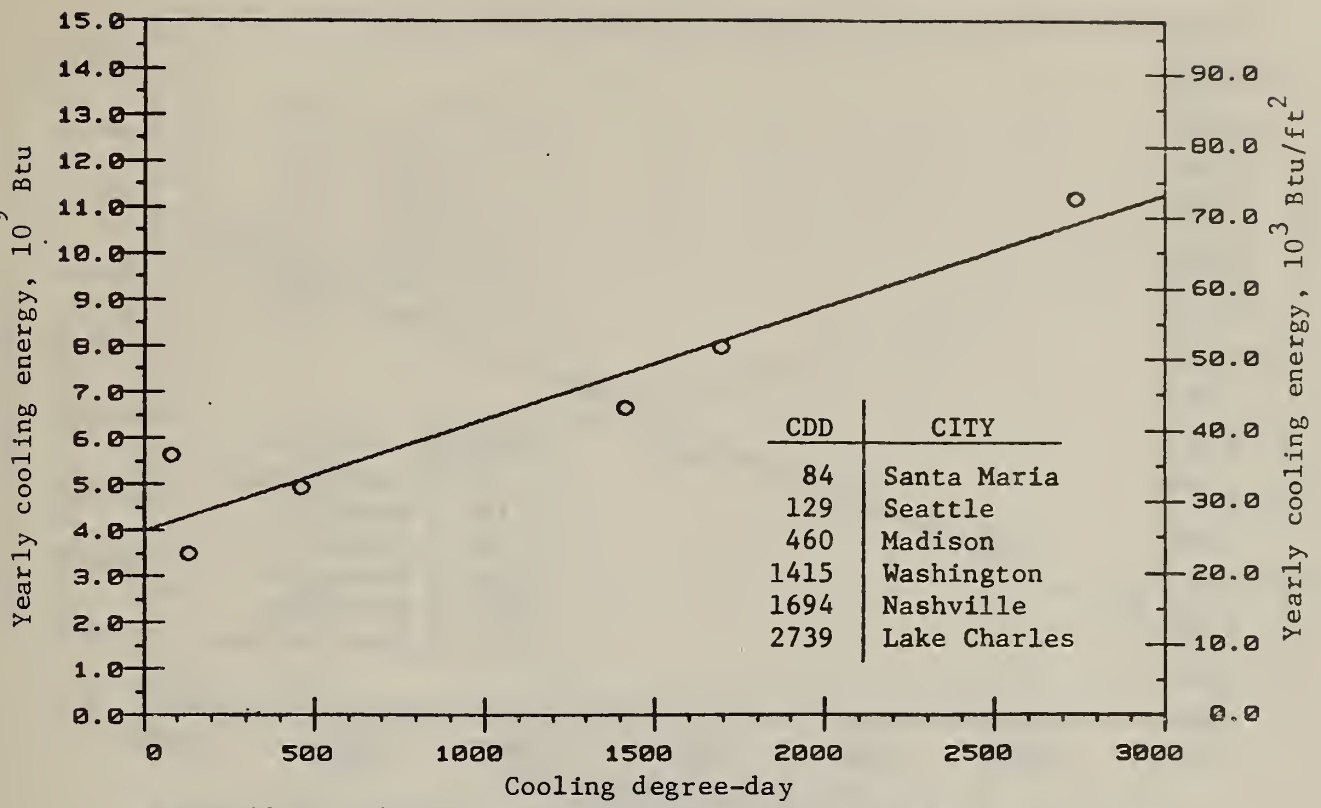


Figure 12. Cooling energy consumption of applying lowered space heating temperature, temperature economy cycle, and outside air resetting supply air temperature (Case 9)

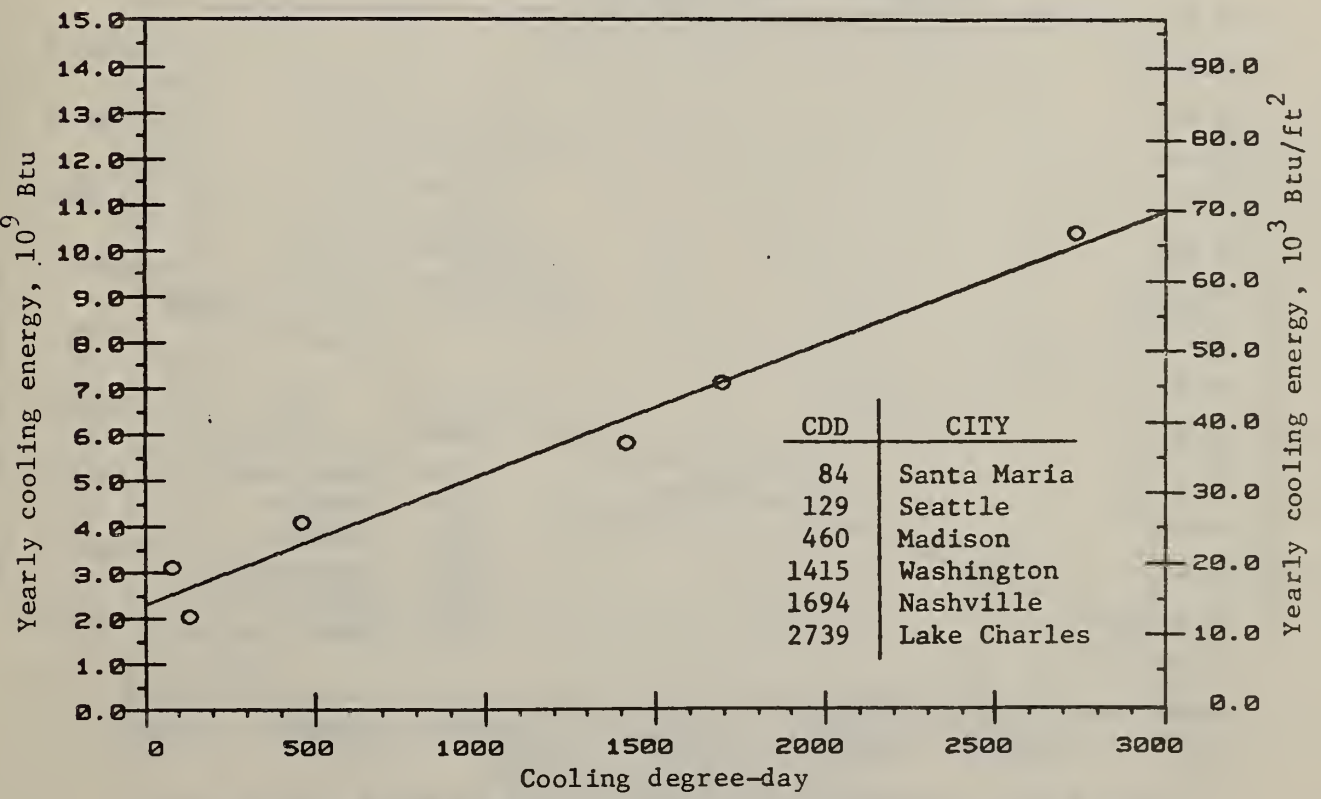


Figure 13. Cooling energy consumption of applying lowered space heating temperature, enthalpy economy cycle, and outside air resetting supply air temperature (Case 10)

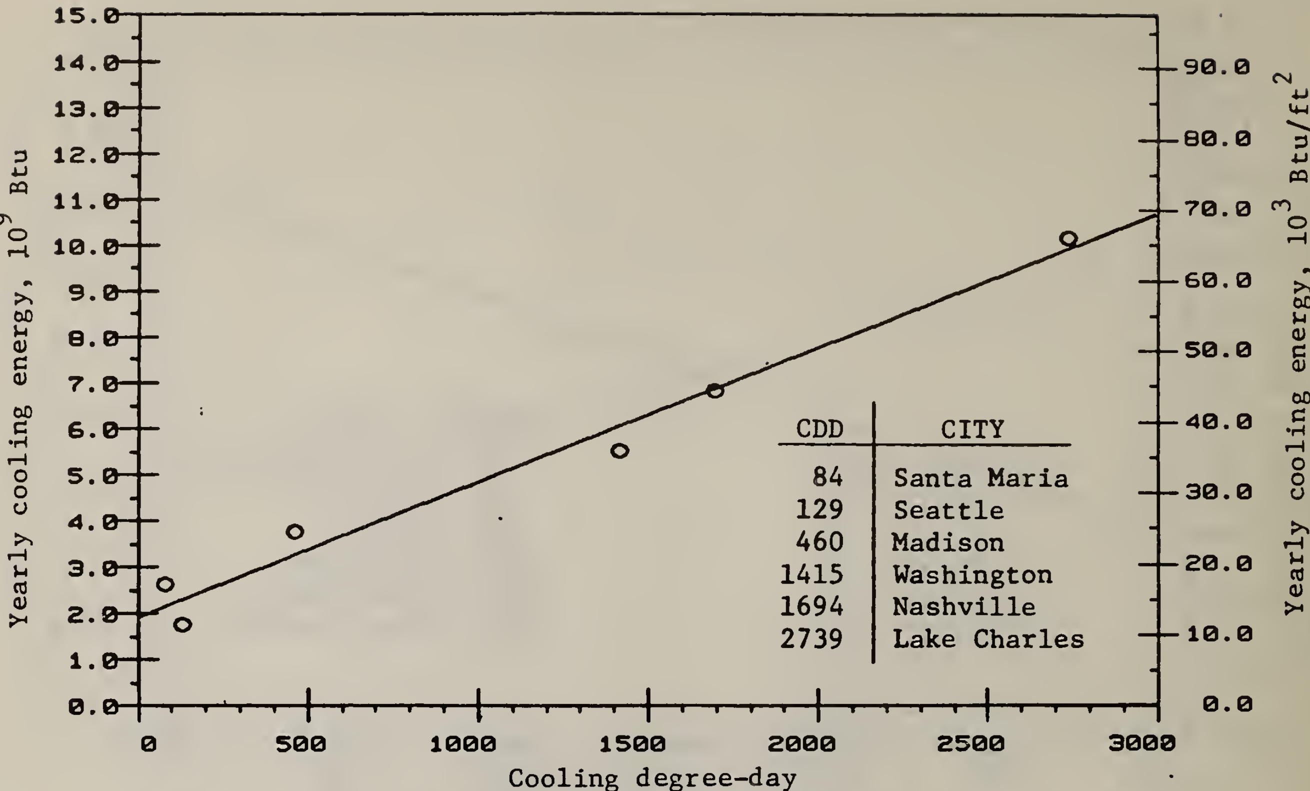


Figure 14. Cooling energy consumption of applying lowered space heating temperature, enthalpy economy cycle, outside air resetting supply air temperature, and VAV to perimeter zones (Case 11)

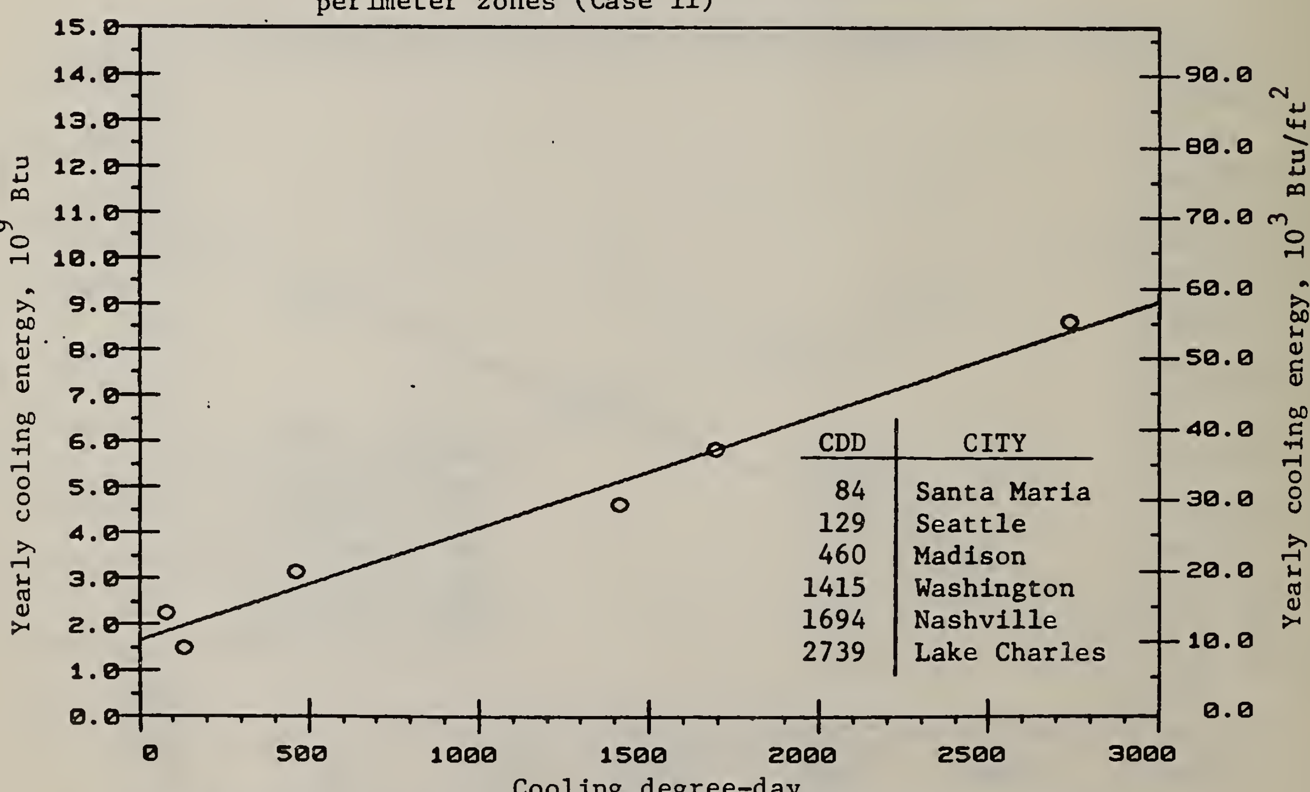


Figure 15. Cooling energy consumption of applying lowered space heating temperature, enthalpy economy cycle, outside air resetting supply air temperature, and VAV to all zones (Case 11)

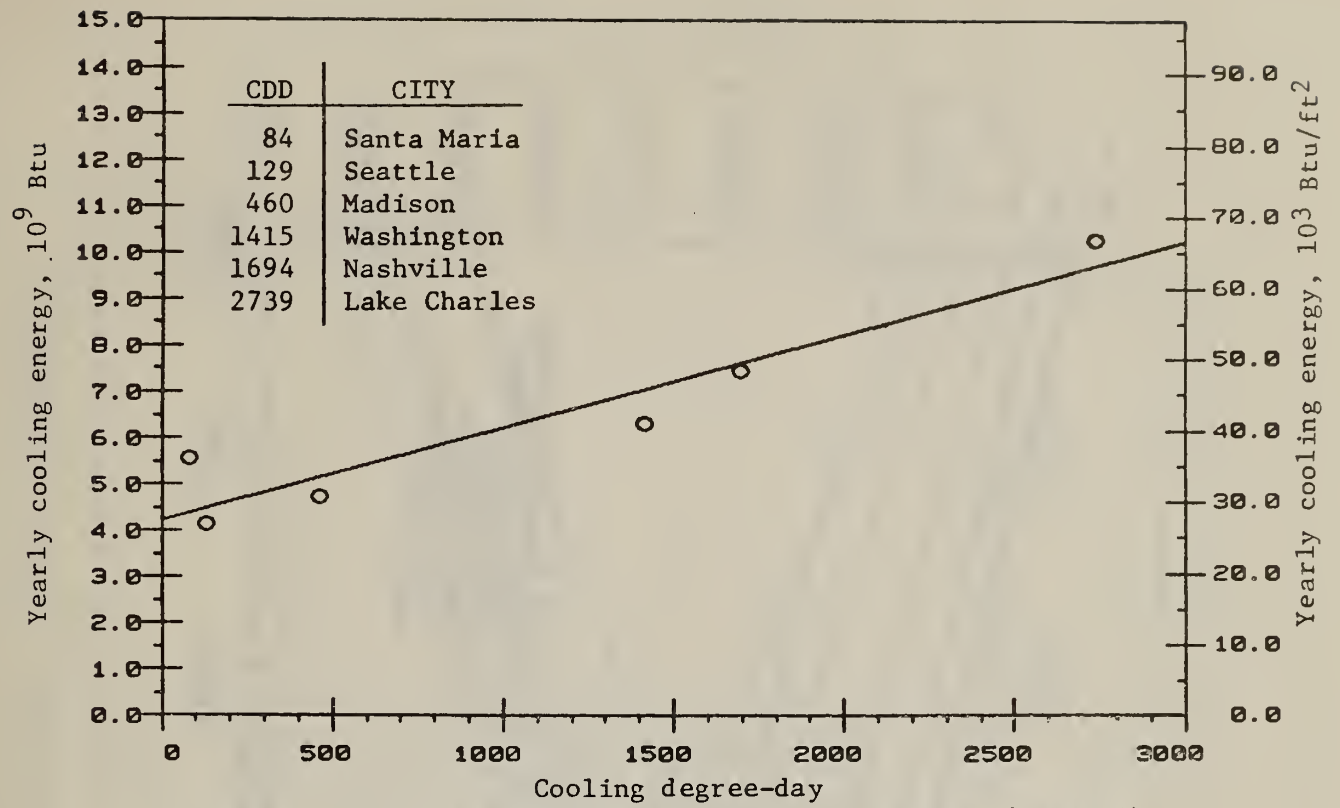


Figure 16. Cooling energy consumption of DX system (Case 13)

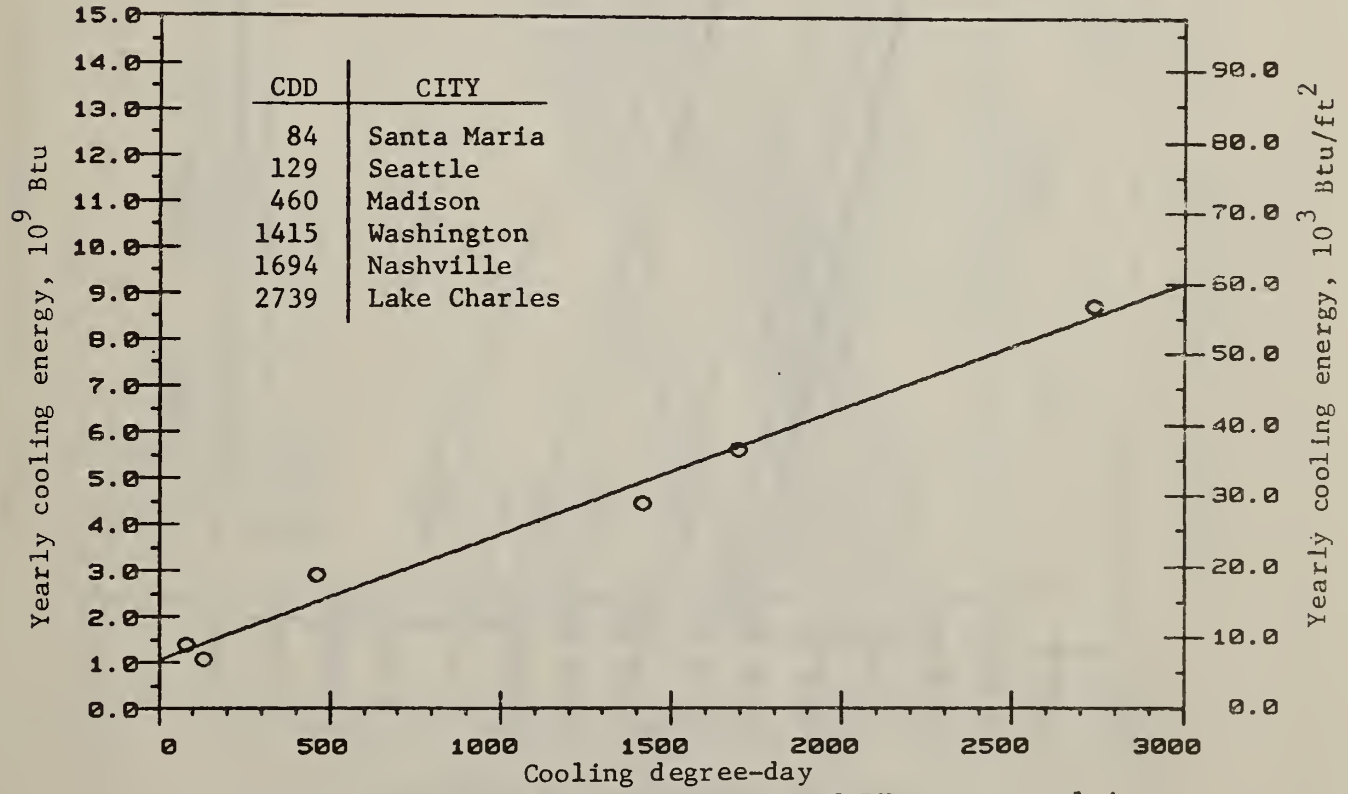


Figure 17. Cooling energy consumption of DX system applying enthalpy economy cycle (Case 14)

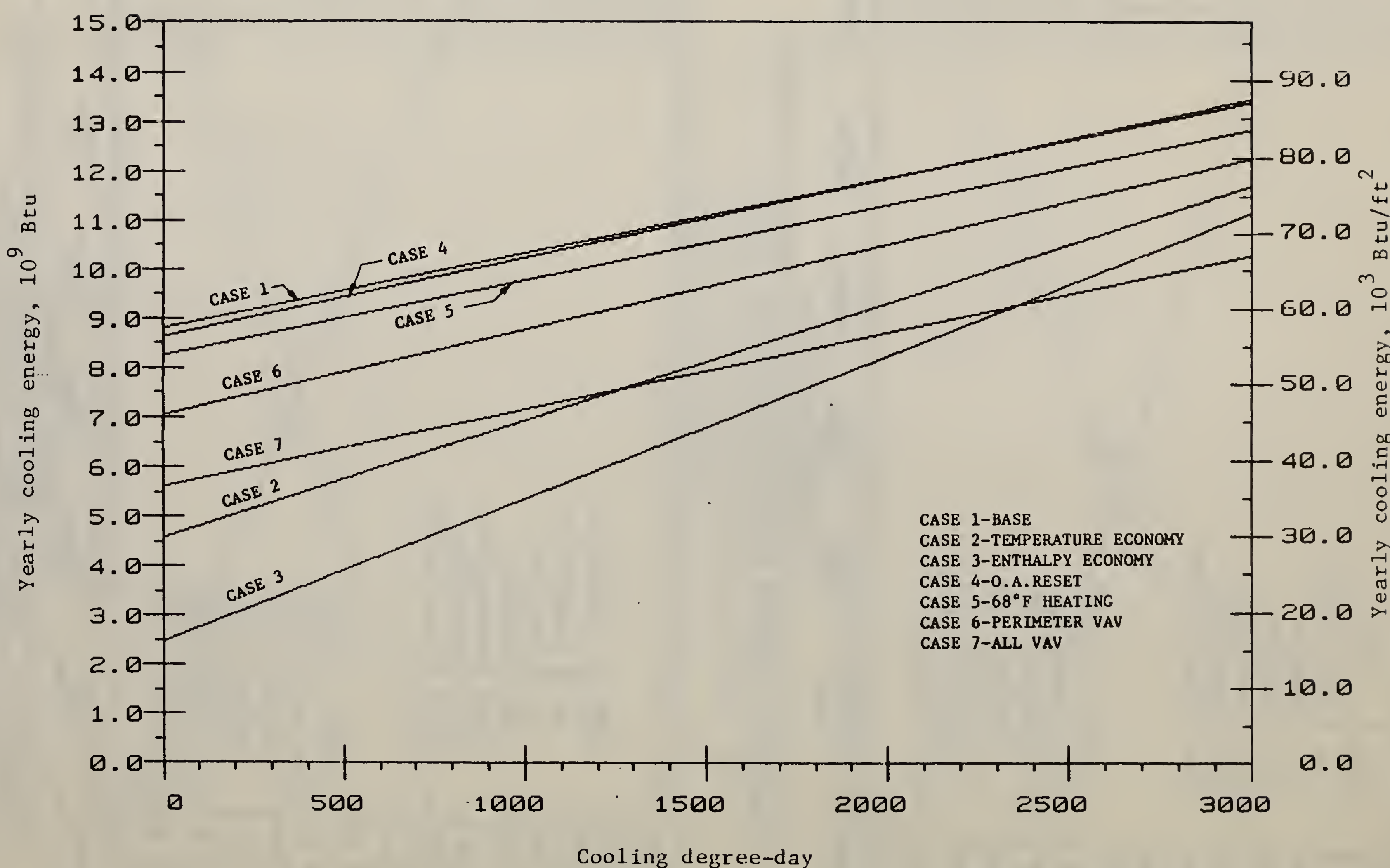


Figure 18. Cooling energy consumption of all single control strategy cases

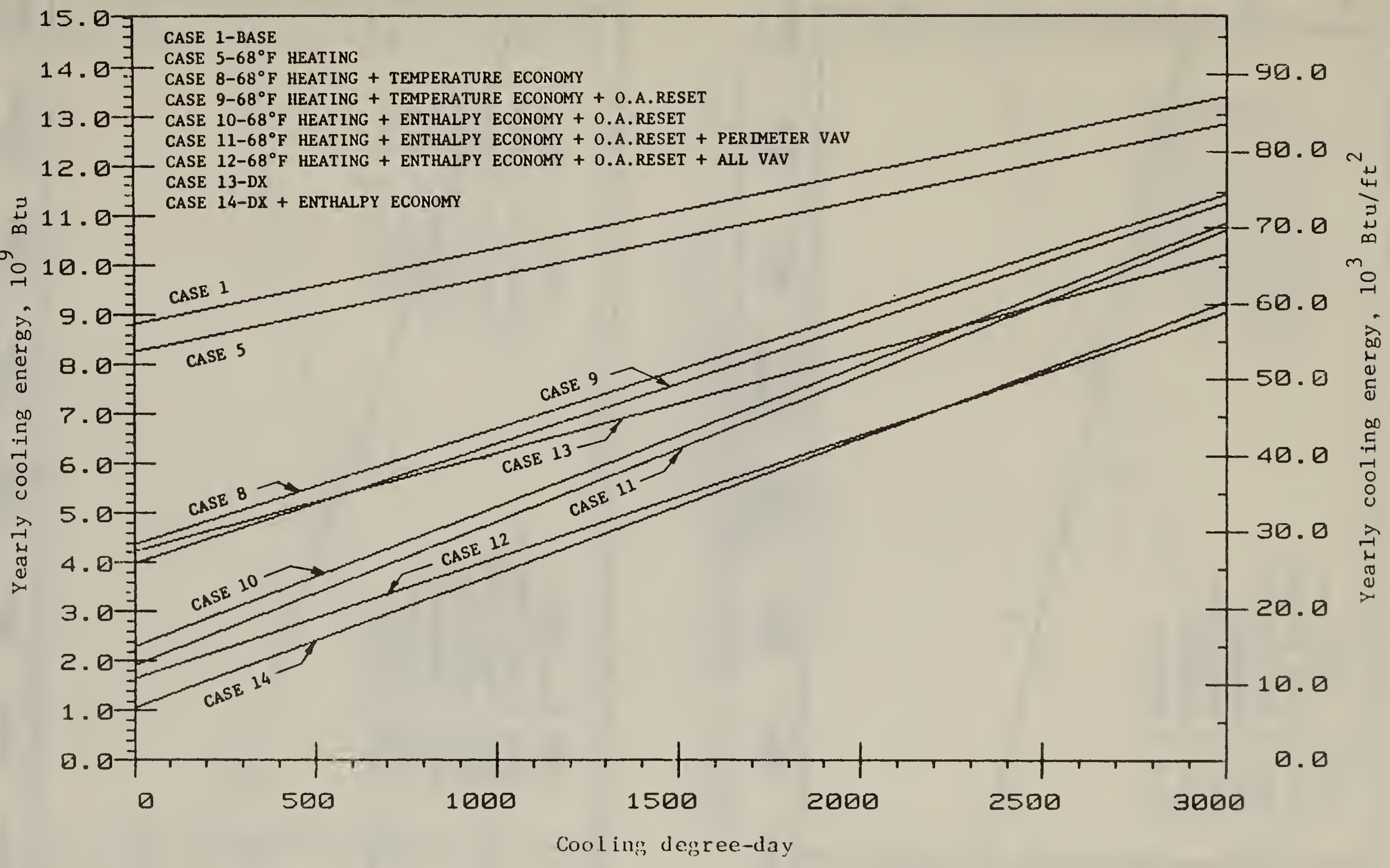


Figure 19. Cooling energy consumption of all combined control strategy cases and DX cases

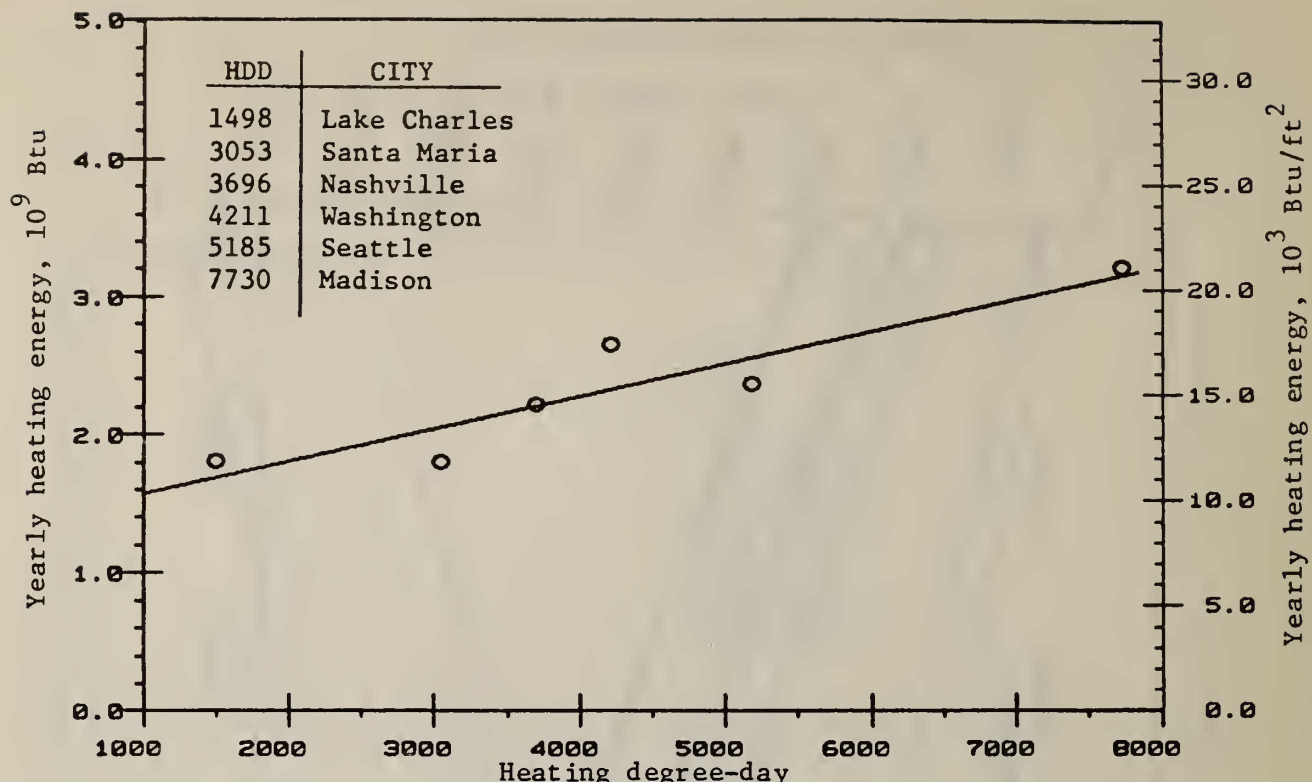


Figure 20. Heating energy consumption of base case (Case 1)

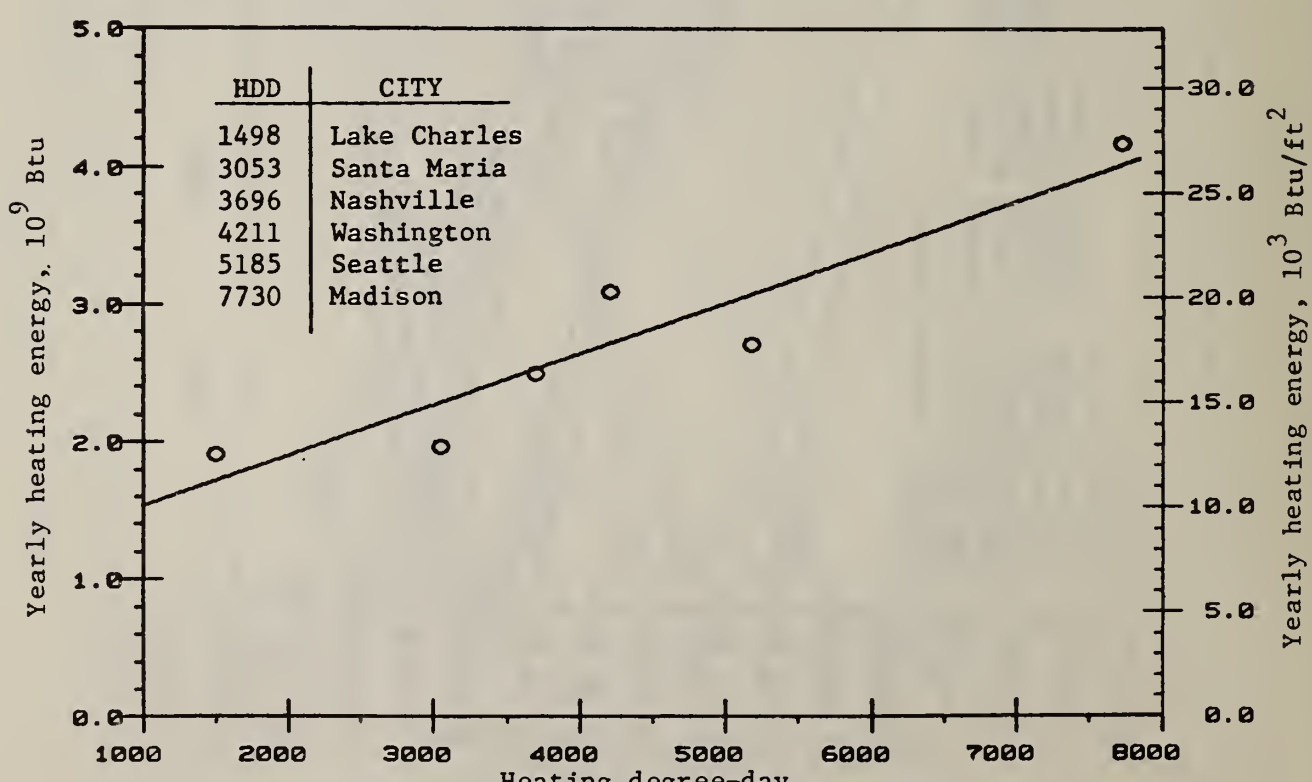


Figure 21. Heating energy consumption of applying temperature economy cycle (Case 2)

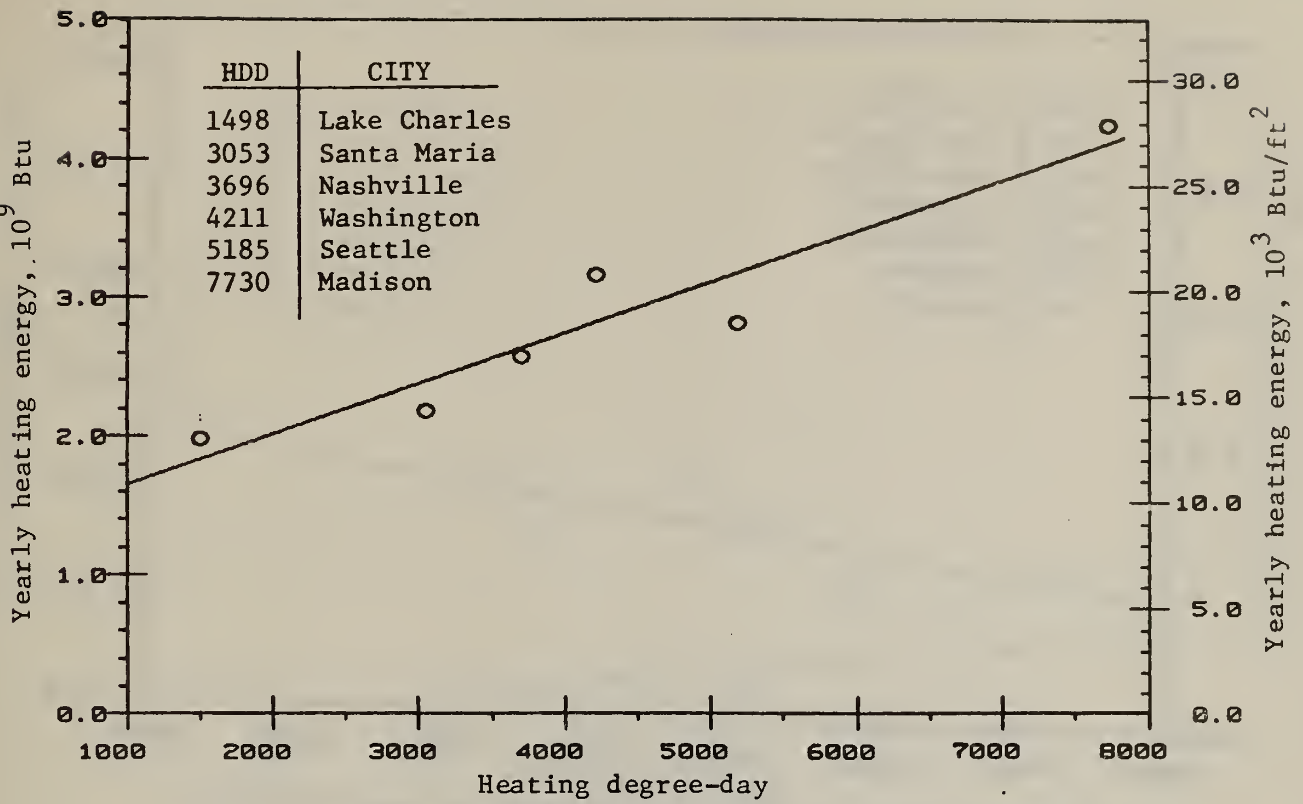


Figure 22. Heating energy consumption of applying enthalpy economy cycle (Case 3)

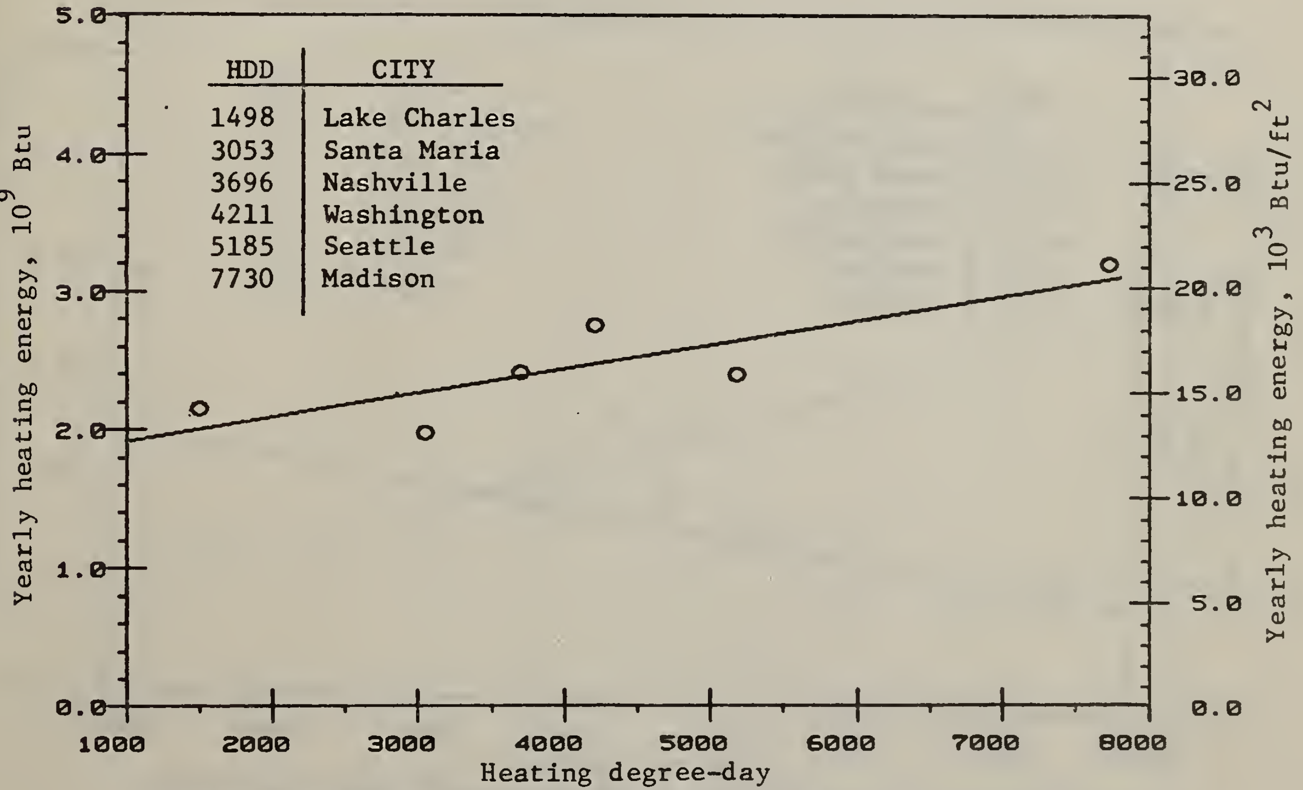


Figure 23. Heating energy consumption of applying outside air resetting supply air temperature (Case 4)

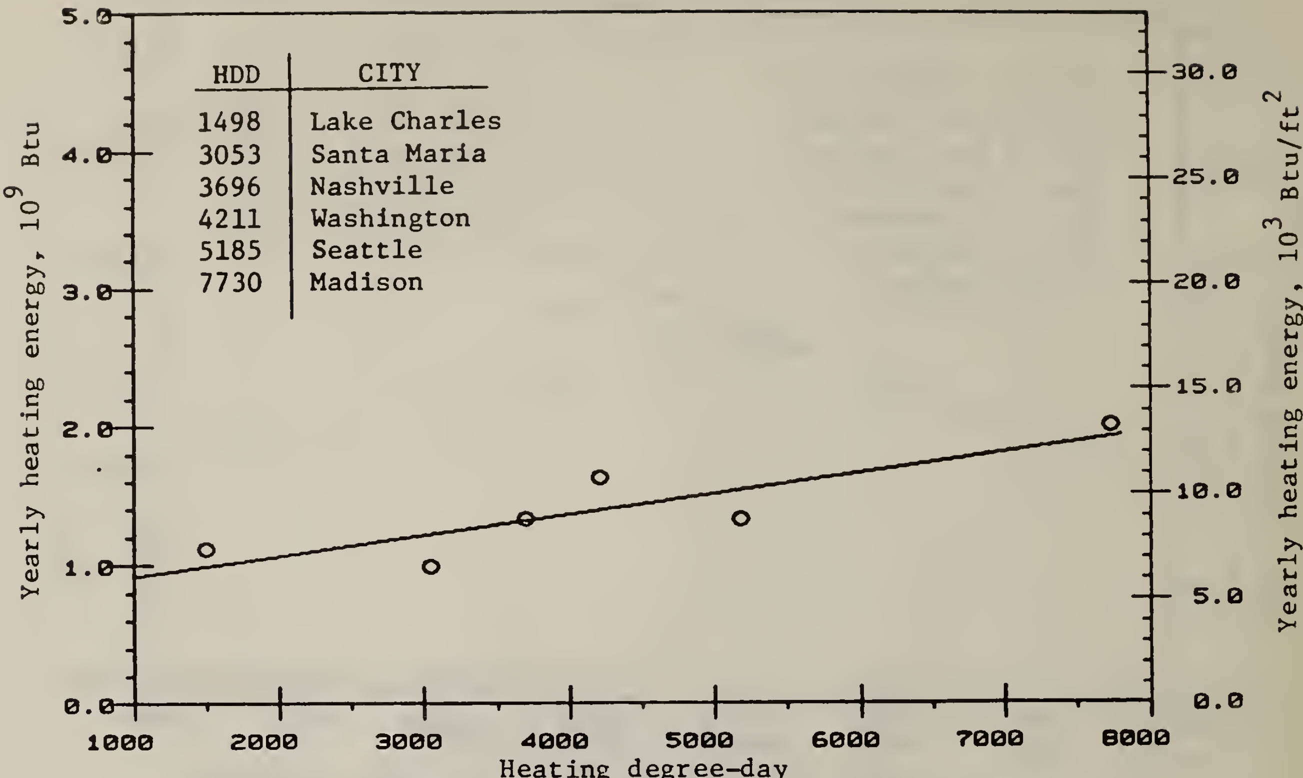


Figure 24. Heating energy consumption of base case with lowered space heating temperature (Case 5)

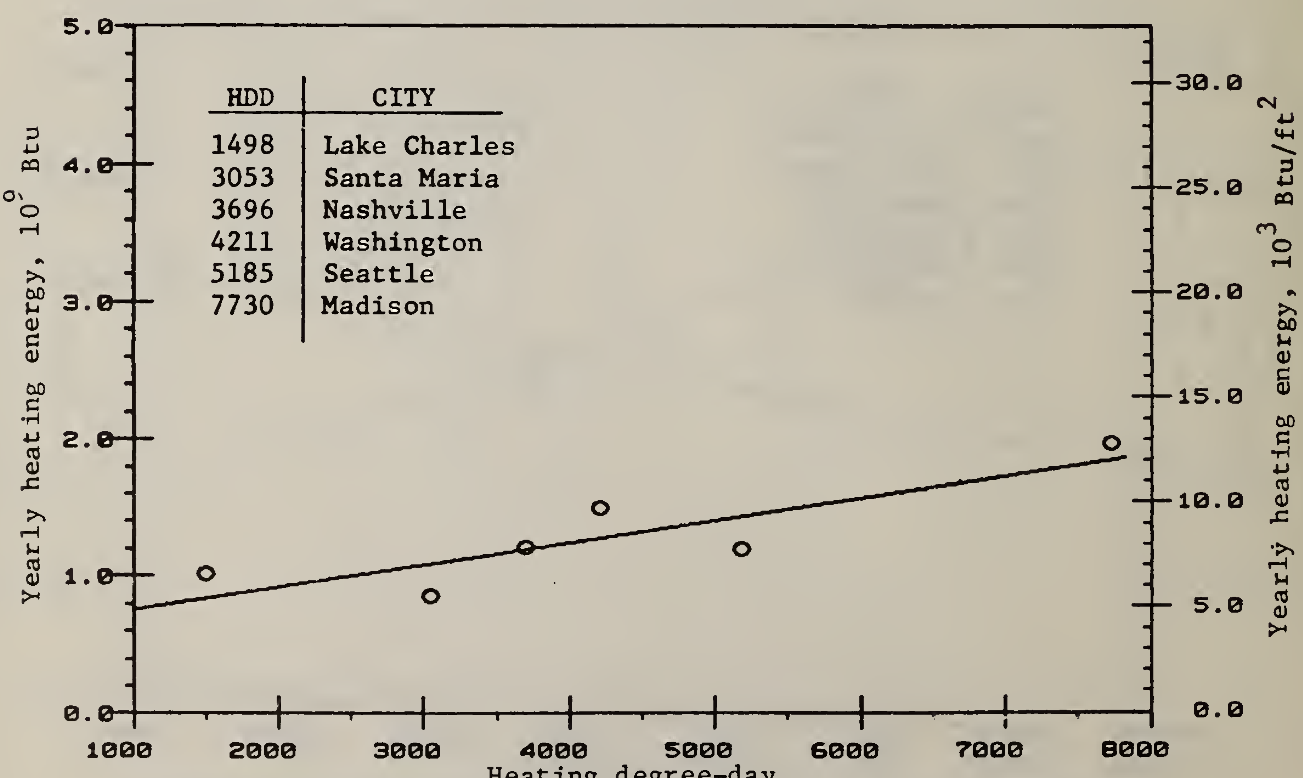


Figure 25. Heating energy consumption of applying VAV to perimeter zones (Case 6)

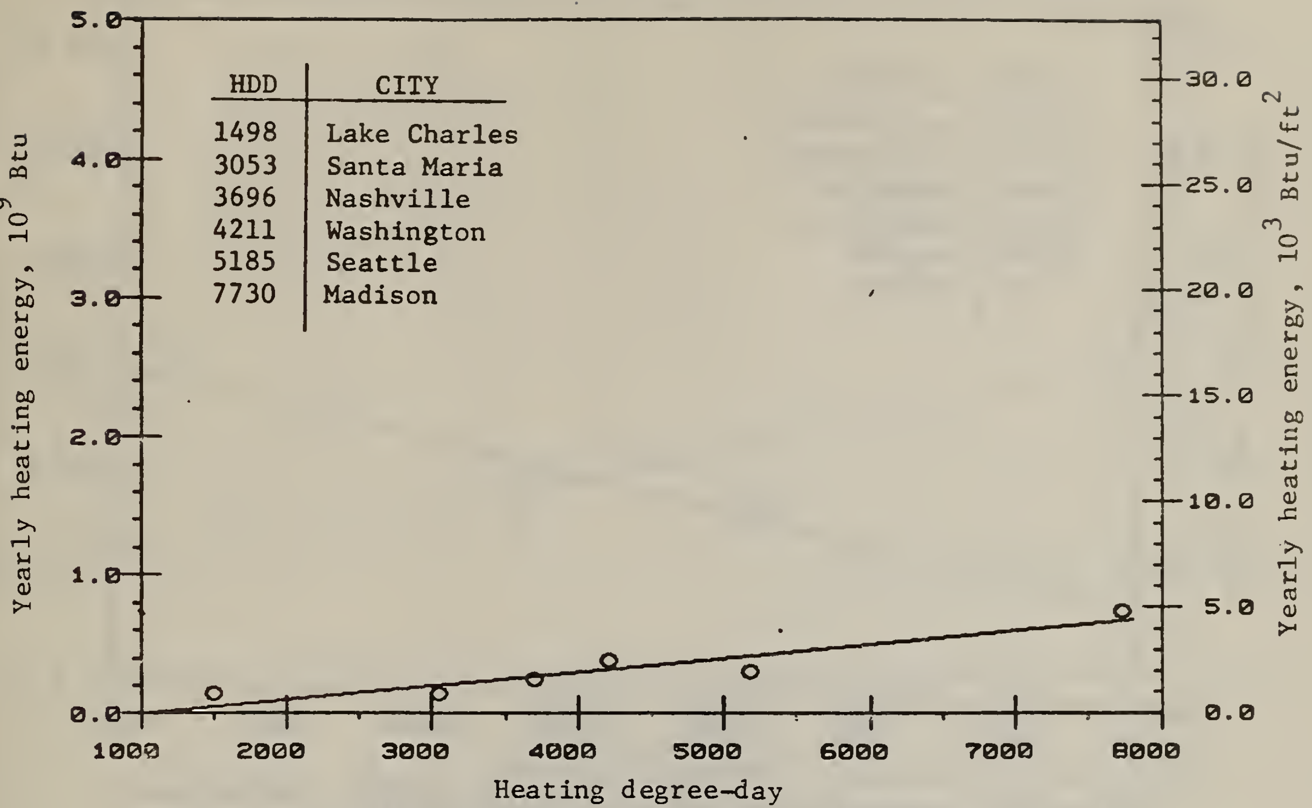


Figure 26. Heating energy consumption of applying VAV to all zones (Case 7)

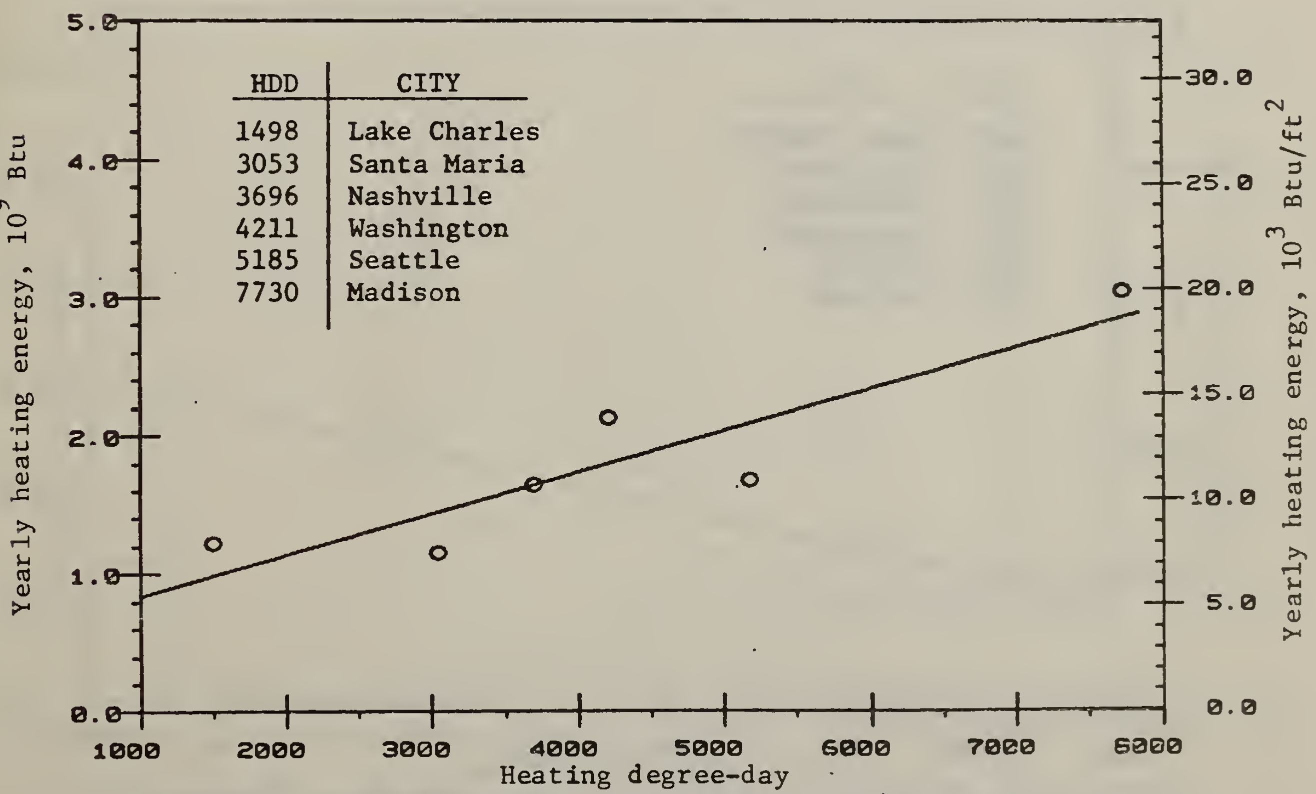


Figure 27. Heating energy consumption of applying lowered space heating temperature and temperature economy cycle (Case 8)

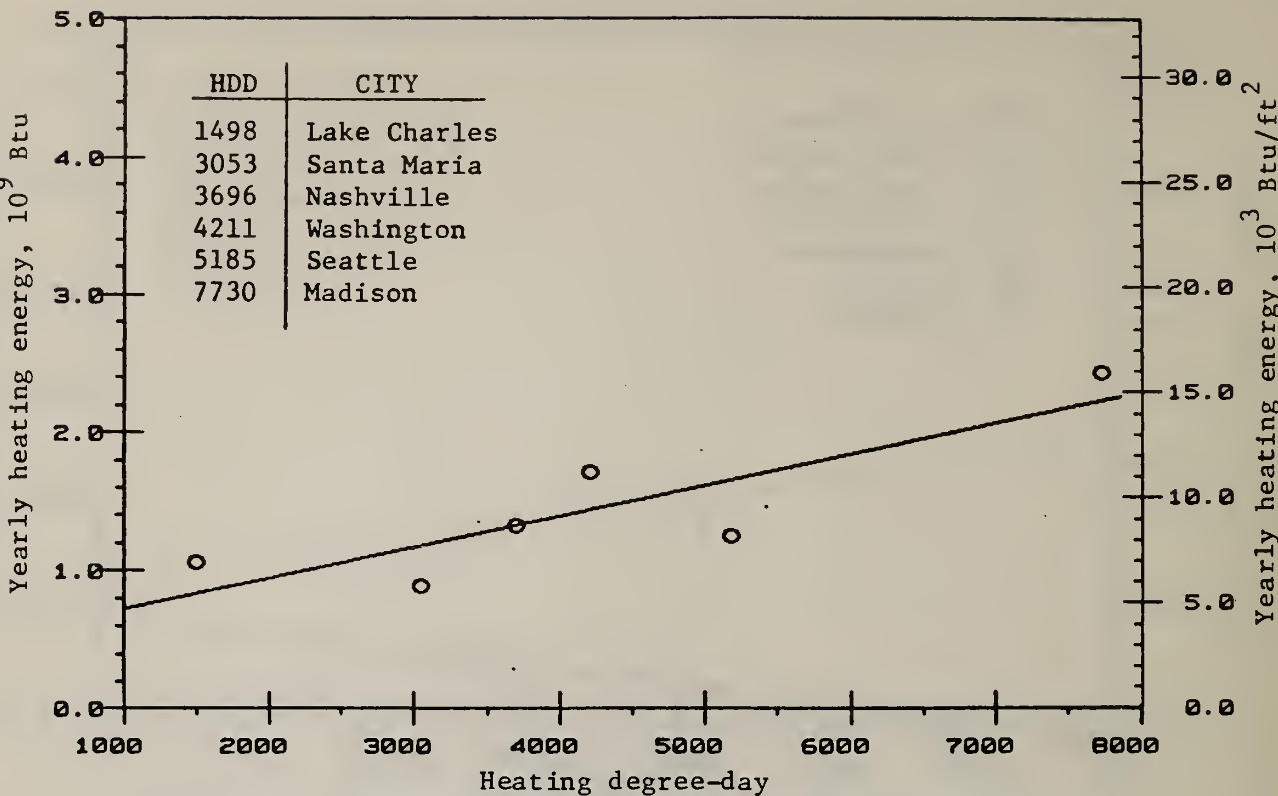


Figure 28. Heating energy consumption of applying lowered space heating temperature, temperature economy cycle, and outside air resetting supply air temperature (Case 9)

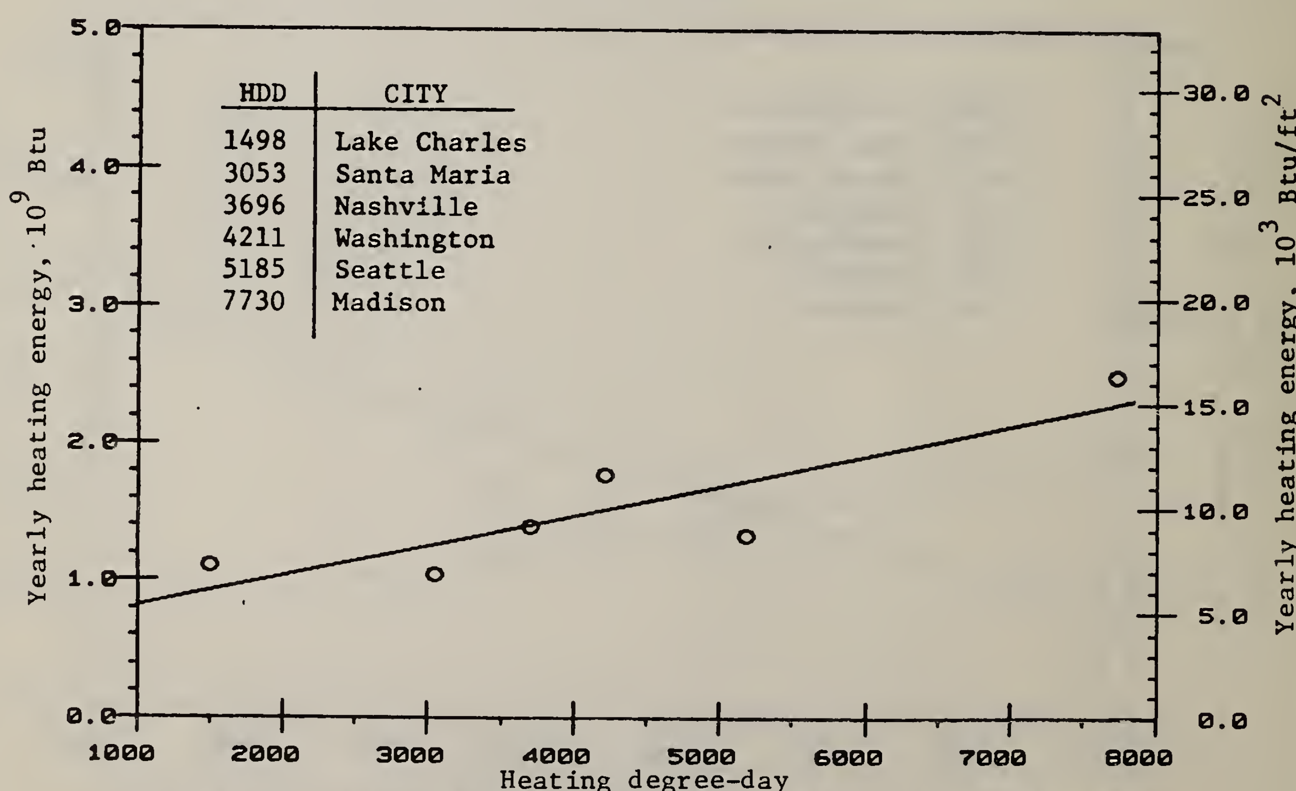


Figure 29. Heating energy consumption of applying lowered space heating temperature, enthalpy economy cycle, and outside air resetting supply air temperature (Case 19)

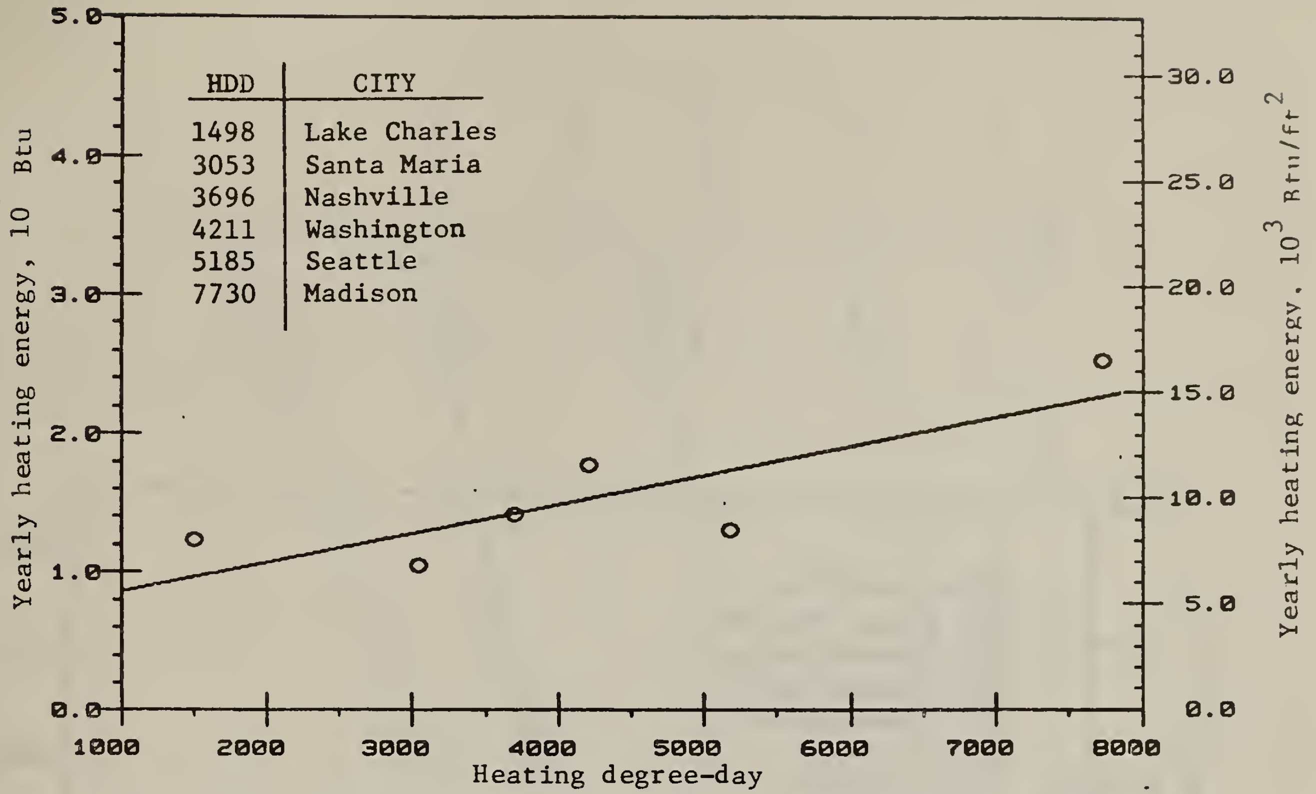


Figure 30. Heating energy consumption of applying lowered space heating temperature, enthalpy economy cycle, outside air resetting supply air temperature, and VAV to perimeter zones (Case 11)

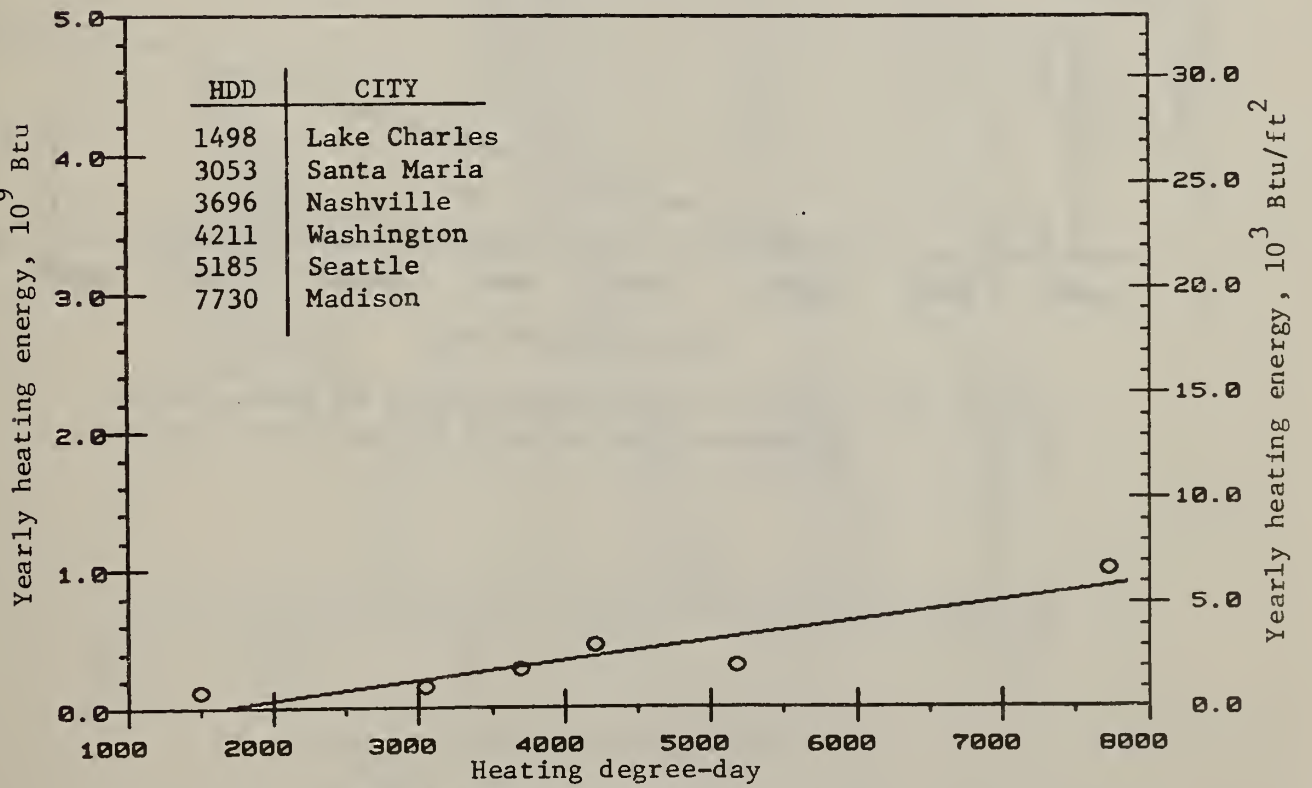


Figure 31. Heating energy consumption of applying lowered space heating temperature, enthalpy economy cycle, outside air resetting supply air temperature, and VAV to all zones (Case 12)

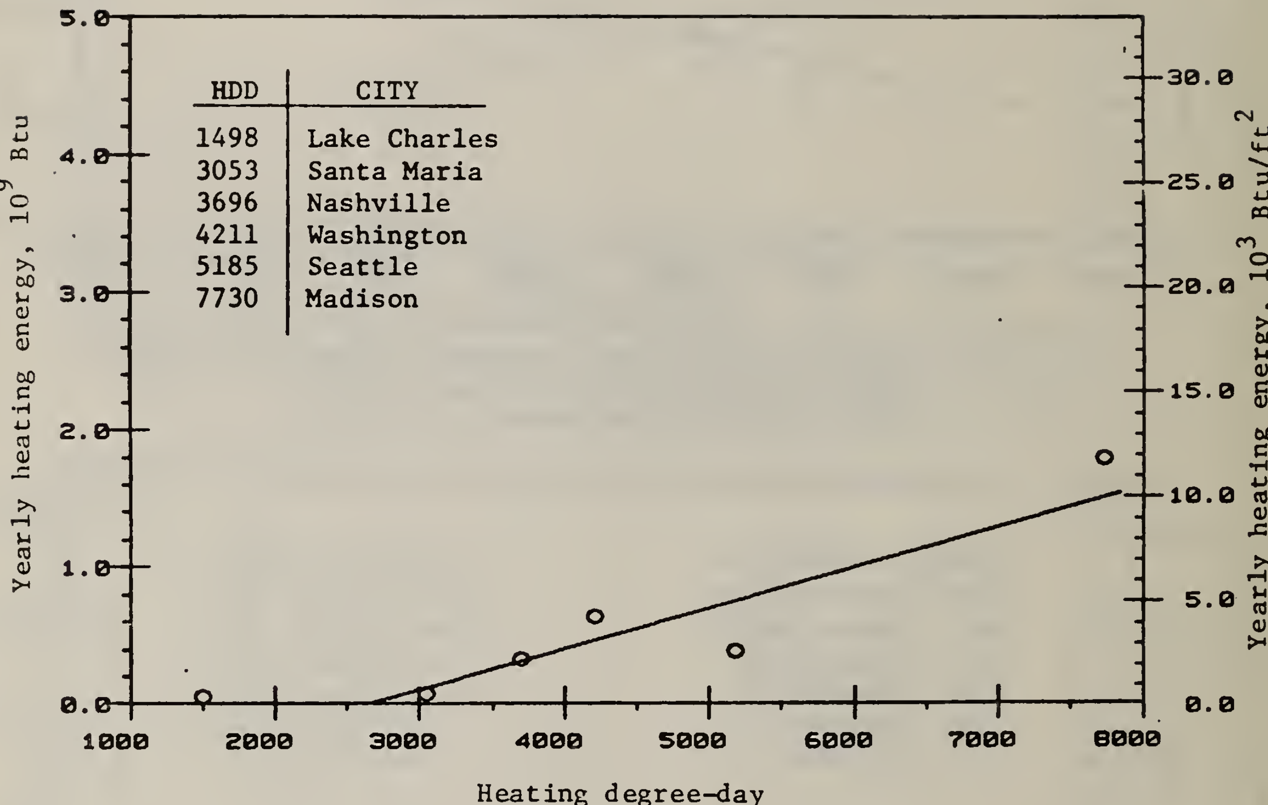


Figure 32. Heating energy consumption of DX system and DX system applying enthalpy economy cycle (Cases 13 and 14)

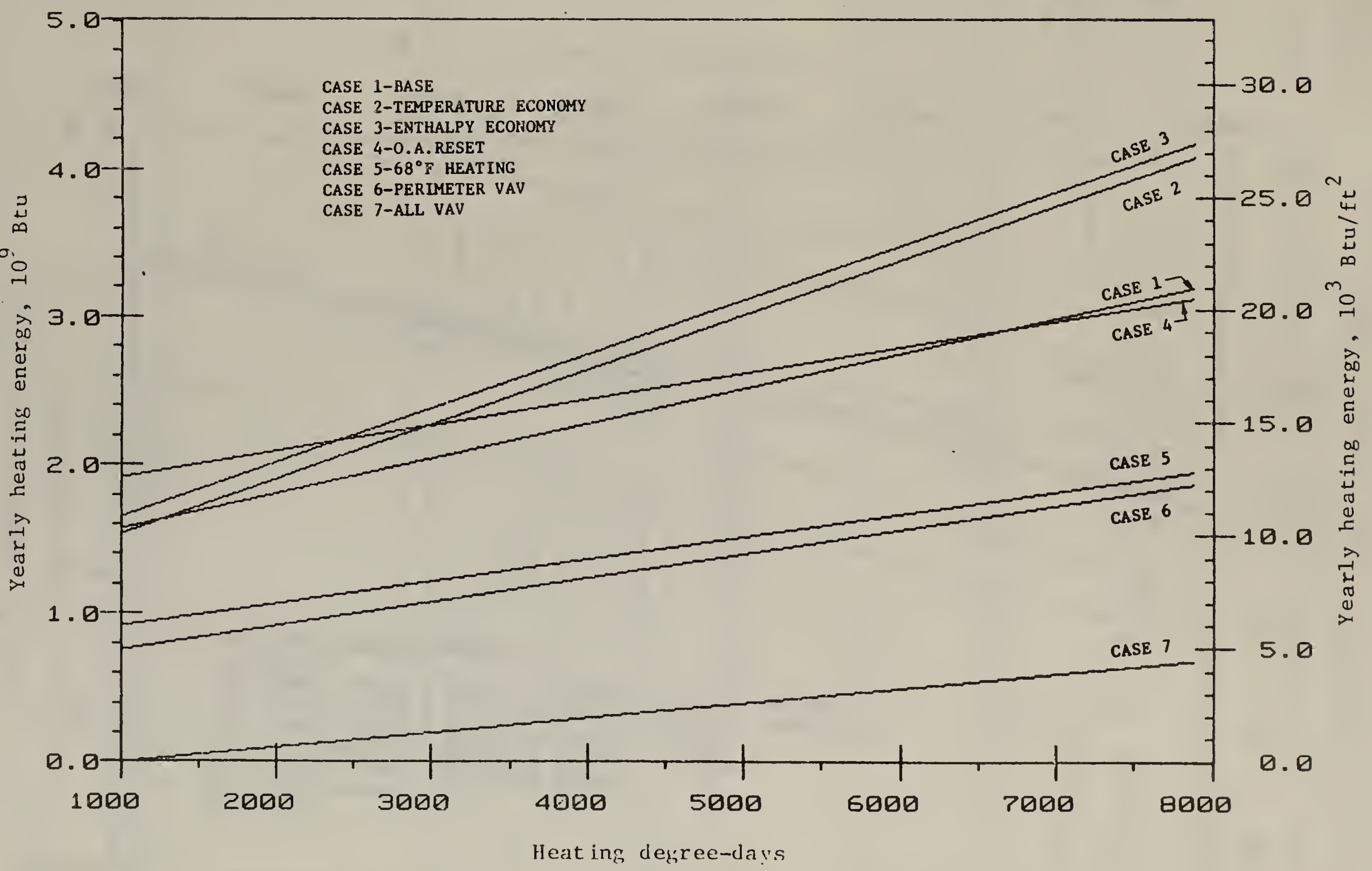


Figure 33. Heating energy consumption of all single control strategy cases

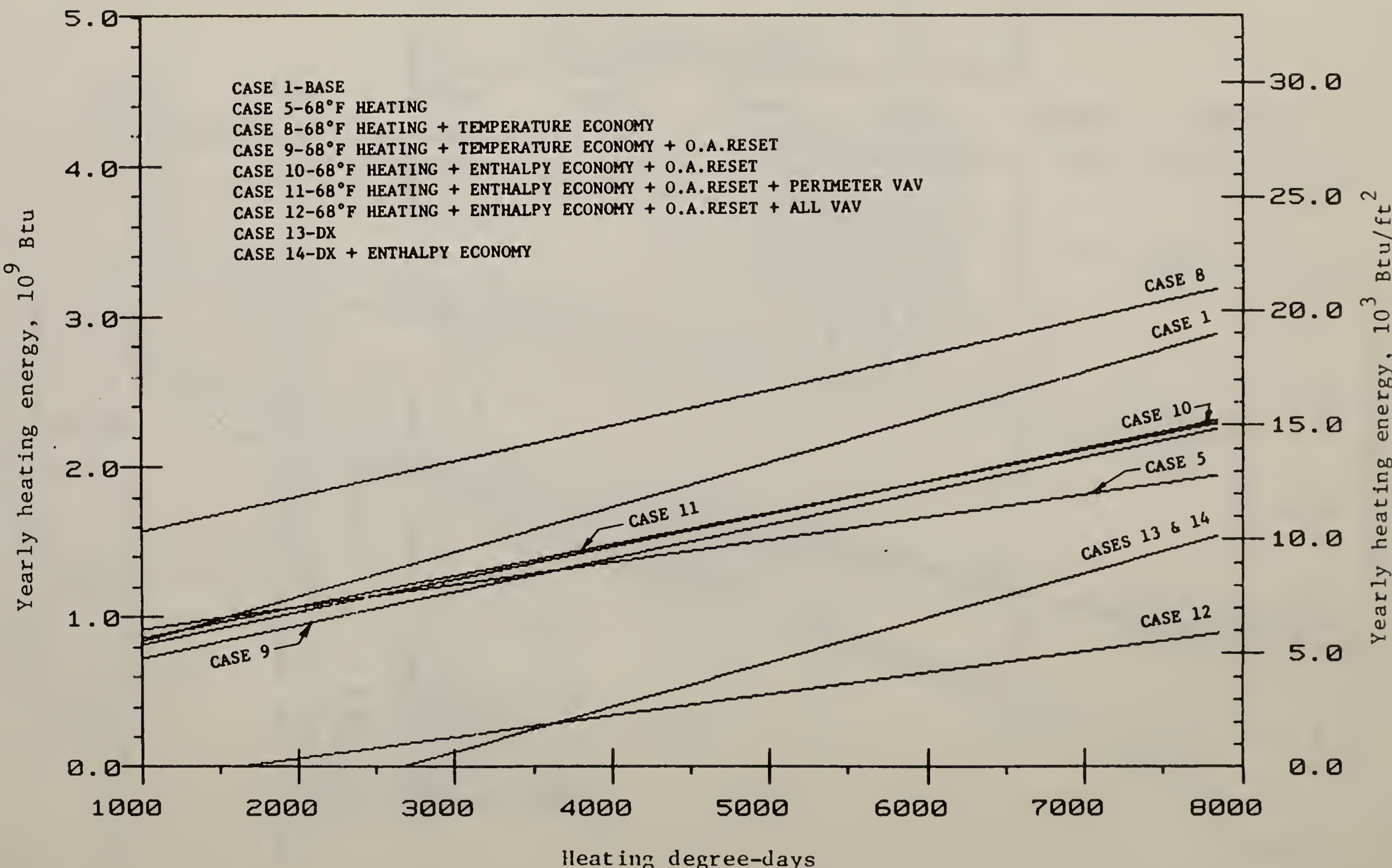


Figure 34. Heating energy consumption of all combined control strategy cases

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10. SUPPLEMENTARY NOTES

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

11. ABSTRACT *(A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here)*

A comparative analysis is made of the thermal performance of selected HVAC systems and control strategies commonly employed in large retail stores. The comparisons are made for six geographical locations representing wide climatic variations within the continental United States. Hour-by-hour simulations with the BLAST computer program were used to obtain the yearly heating, cooling and fan energy consumption of a two-story large retail store. The HVAC systems simulated were constant volume reheat, variable air volume, and with direct expansion coils. The control stragegies tested were dry bulb temperature economy cycle, enthalpy economy cycle, supply air temperature resetting, lowered space heating temperature, VAV zoning variations, and the combinations of these strategies. The results of these simulations were given and discussed. Substantial energy consumption differences were shown.

12. KEY WORDS *(Six to twelve entries; alphabetical order; capitalize only proper names; and separate key words by semicolons)*

Building control strategies; building energy conservation; building thermal performance; HVAC systems

13. AVAILABILITY

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