MODELING AND TESTING STRATEGIES FOR EVALUATING VENTILATION LOAD REDUCTION TECHNOLOLGIES

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I. INTRODUCTION

A. Scope

The heating and cooling loads associated with ventilation can contribute significantly to the total energy requirements for a commercial space being conditioned. In recent years, several different approaches have been proposed to reduce ventilation loads including enthalpy exchangers, economizers, demand-control ventilation and ventilation heat recovery heat pumps. However, different technologies may be appropriate for different environments and buildings.

This project will focus on identifying appropriate applications and locations for ventilation load reduction technologies within the state of California. The performance of economizer, enthalpy exchanger, demand-controlled ventilation and heat recovery heat pump technologies will be compared for different types of buildings and locations. For demand-controlled ventilation, field sites are being established in coastal and inland sites in both northern and southern California. Three different building types are being considered with two nearly identical buildings for each location so that direct comparisons between the performance of fixed ventilation and demand-controlled ventilation can be made. Data from the field sites will be compared with simulation results in order to validate computer models. The models will then be used to evaluate the cost savings potential for this technology for other buildings and locations. In addition, the models will also consider economizer, enthalpy exchanger, and heat pump heat recovery technologies. The performance of all these technologies will be compared in terms of their cost effectiveness. As a further validation of the simulation results, an additional field will be established for testing the heat pump heat recovery unit.

B. Purpose of this Report

This progress report presents an overview of the modeling approach and input data to be used in evaluating the energy savings associated with each of the ventilation load reduction technologies. In addition, an overview of the preliminary test plan and field site monitoring setup for the heat pump heat recovery unit is given.

II. VENTILATION LOAD REDUCTION TECHNOLOGIES

A. Economizer

An economizer uses outside air to reduce or eliminate the mechanical cooling required to condition a building. This accessory usually includes an outside air damper, a relief damper, a return air damper, filters, an actuator and linkages. An economizer can be installed with any of the other three ventilation energy savings technologies that will be considered in this study. When the outdoor conditions are suitable, the outdoor air dampers switch from their minimum position (minimum ventilation air) to fully open. For a dry-bulb economizer, this switch point occurs when ambient air is less than a specified value. This switch point should be less than the switch point to return to minimum outside air in order to ensure stable control. The economizer switchover temperature may be significantly lower than the return air temperature (e.g., 10 F lower) in humid climates where latent ventilation loads are significant. However, in dry climates, the switchover temperature may be close to the return temperature (e.g., 75 F). An enthalpy (or wet-bulb) economizer compares the outside and return air enthalpies (or wet-bulb temperatures) in order to initiate or terminate economizer operation. In general, enthalpy economizers yield lower energy costs than dry-bulb economizers, but require a humidity measurement. With either economizer, the outside air damper modulates the flow to maintain a mixed air temperature set point, and when this set point can no longer be achieved, the compressor is engaged (Howel et al., 1998).

B. Enthalpy Exchanger

A rotary air-to-air enthalpy exchanger, sometimes called a heat recovery wheel, is a revolving cylinder filled with an air permeable medium with a large internal surface area for contact with the air passing through it. Adjacent supply and exhaust streams each flow through half the exchanger in a counter-flow pattern as illustrated in Figure 1.

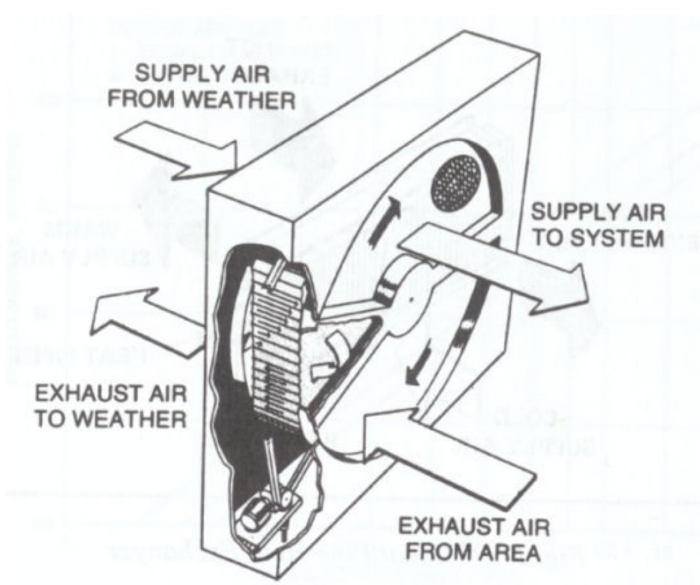

FIGURE 1. FLOW DIRECTION IN AN ENTHALPY EXCHANGER

Sensible heat is recovered as the medium picks up and stores heat from the hot airstream and gives it up to the cold airstream. Latent heat is transferred as the medium condenses moisture from the airstream having the higher humidity ratio, with a simultaneous release of heat. The medium then releases the moisture through evaporation into the airstream with the lower humidity ratio. The enthalpy exchanger medium is fabricated from metal, mineral, or man-made materials and classified as providing either random flow or directionally oriented flow through their structures (Howel et al., 1998). An enthalpy exchanger works for both heating and cooling and can allow for 100% outside air.

C. Demand Controlled Ventilation

The energy requirements to heat or cool a building can be reduced by modulating ventilation air in response to the number of occupants in the building at any given time. This can be accomplished by controlling the ventilation air to maintain a specific CO_2 level within the building. This strategy is referred to as demand-controlled ventilation (DCV). Brandemuehl and Braun (1999) performed a simulation study for a number of different buildings and locations and showed that as much as 20% savings in electrical energy for cooling are possible with demand-controlled ventilation. The savings in heating energy associated with demand-controlled ventilation are generally much larger, but are strongly dependent upon the building type and occupancy schedule. Significantly greater savings are possible for buildings with highly variable occupancy schedules and relatively large internal gains. However, the overall cost effectiveness of DCV has not been evaluated and the savings have not been documented in the field.

D. Ventilation Heat Pump Heat Recovery

Carrier's Energy Recycler® accessory, available for 3 to 12.5 ton rooftop units, introduces a technique to help reduce the total load on the primary HVAC system by outside air pre-treatment. Figure 2 illustrates operation of the Energy Recycler® using some example design cooling conditions. In the cooling season, the Energy Recycler® cools and possibly dehumidifies outside air entering the unit, allowing for larger quantities of outside air. The heat is rejected into the exhaust air from the building. The room air is used to cool the condenser coil and thus allows the condenser to operate at a lower temperature than the ambient. During heating season, the Energy Recycler® operates in reverse as a heat pump to extract heat from the exhaust air and pre-heat the outside air. The application of a ventilation heat pump heat recovery units leads to a lower load on the primary equipment. However, the unit requires energy and the overall economics are not known.

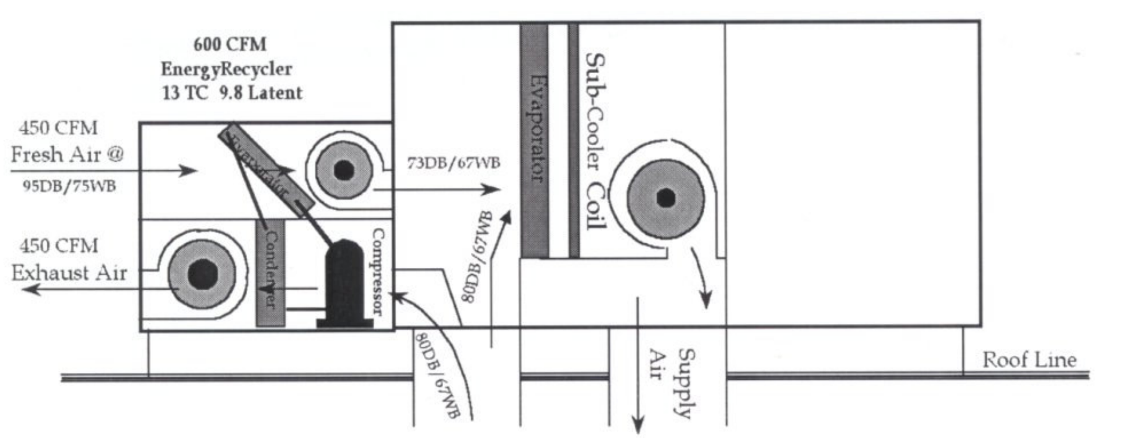

FIGURE 2. ENERGY RECYCLER® SCHEMATIC ATTACHED TO ROOFTOP UNIT

II. SIMULATION APPROACH

The simulations will be performed for a variety of small commercial building types that utilize packaged air conditioning and heating equipment. A computer simulation model is being developed for estimating the energy requirements and life cycle economic impact for the different ventilation load reduction technologies. The model is based upon the tool previously developed by Brandemuehl and Braun (1999). Figure 3 shows a flow diagram of the computer simulation model to be implemented for evaluating these different methods.

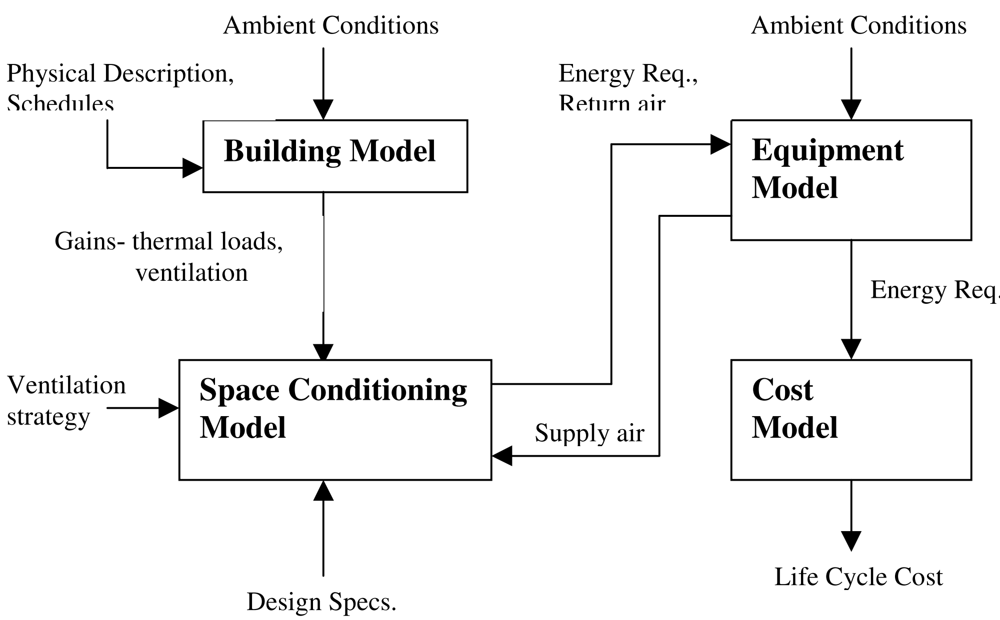

FIGURE 3. FLOW DIAGRAM OF MODELING APPROACH

The model will calculate hourly energy requirements for a particular building type and system type and then use this data to determine the total cost of HVAC operation. The building model will predict the thermal gains to or from the zone based upon transient heat transfer from outside walls and internal sources. The space-conditioning model will solve mass and energy balances for the zone air and then determine return air conditions for the equipment model. The zone air humidity, dry-bulb temperature, and CO2 concentration will be calculated at each hour within the space conditioning model. The ventilation and return air will be mixed according to the ventilation technique being analyzed. The equipment model will use mixed air conditions and the sensible cooling requirement to determine the average supply air conditions. These entering mixed air conditions and supply air conditions will be determined iteratively using a nonlinear equation solver. The energy used by the equipment model will be calculated and used as an input in determining the life cycle cost for each system.

The cost model will incorporate current electricity rates in California and equipment costs to estimate the life cycle cost of the HVAC system for each ventilation control technique. From this economic data, comparisons can be made between all the different combinations of location and building type. The length of the economic analysis will be varied to reflect different potential decision makers.

The nonlinear equation solver to be used in this study is an HVAC building/ energy simulation program called TRNSYS (1996), developed by the Solar Energy Laboratory at the University of Wisconsin-Madison. TRNSYS is a transient systems simulation program with a modular structure. It recognizes a system description language in which the user specifies the components that constitute the system and the manner in which they are connected. The TRNSYS library includes many of the components commonly found in thermal energy systems, as well as component routines to handle input of weather data. The modular structure of TRNSYS gives the program tremendous flexibility and facilitates the addition to the program of mathematical models that are not included in the standard TRNSYS library. An electronic simulation of the previously mentioned ventilation control strategies can thus be added to the TRNSYS library. With this computer simulation in place, several different combinations of location and building type can be simulated to evaluate the performance of all ventilation control strategies.

A. Building Model

The TYPE 56: “Multi-Zone Building” component from the TRNSYS library will be used for the building model. This component models the thermal behavior of a building having up to 25 thermal zones. This is a very detailed model of a building that is built up from individual descriptions of wall layers, windows, internal gain schedules, etc. The model solves individual transient conduction through walls and considers long-wave radiation exchanges within the space. Model inputs include separate hourly heating and cooling setpoints and the model outputs the required heating or cooling rates necessary to maintain the setpoints.

B. Space-Conditioning Model

The space-conditioning model determines return air conditions for the equipment model. The zone sensible heat gain or loss and the specified zone temperature setpoint determines the required average supply air temperature. Given the supply airflow rate and the supply air temperature, the thermal load requirements for the equipment model are determined by the mixed air conditions. These mixed air conditions depend on the ventilation control strategy implemented.

When the DCV control strategy is enabled, a minimum flow rate of ventilation air is determined that will keep the CO_2 concentration in the zone at or below a specified level (Brandemuehl and Braun, 1999). In the absence of DCV, ventilation percentages are based on design conditions for each specific building type from the ASHRAE Standard 62-1999. Table 1 shows the parameters used to estimate the minimum ventilation rates according to building type.

TABLE 1. ASHRAE MINIMUM VENTILATION REQUIREMENTS

Parameter	*Office*	*Retail*	*School*	*Restaurant*	*Hotel*	*Super-market*
Minimum Ventilation per Person, cfm	20	10*	15	20	15	15
Maximum Design Occupancy for minimum ventilation flow, P/1000 ft^2	7	20	50	70	30	8

*Retail store minimum ventilation is based upon an average of 0.25 cfm/ft^2 for upper and lower floors.

For known ventilation flow, zone temperature, and ambient conditions, steady-state mass and energy balances will be applied to the zone and air distribution system to determine average values over each timestep for the return and zone air CO_2 concentration and humidity ratio. These calculations will be based on a fully-mixed zone model, modified by an air exchange effectiveness to account for partial short-circuiting of the supply air to the ceiling return.

Within the TRNSYS environment, the space-conditioning model will be a custom TYPE component that will interact with the TYPE 56 model through inputs and outputs

C. Equipment Model

Packaged rooftop air conditioner with on/off controls will be simulated in this study. The model will use the return air and ambient air conditions to determine the average supply air conditions for the space-conditioning model. The analysis will include air conditioners with gas furnaces and electric auxiliary heat. The supply fan will be on during all hours of occupancy, and the compressor or heater will cycle on and off as necessary to maintain the zone temperature at its set point. Models for a direct expansion air conditioner will taken from the ASHRAE Secondary Toolkit (Brandemuehl, et al., 1993) and adapted for this project. The secondary toolkit contains a library of subroutines and functions that have been debugged and documented. The direct expansion and heat pump models are based upon correlations used in DOE 2.1E. These models estimate capacity (cooling or heating) and power consumption as a function of mixed air and ambient conditions for typical devices. The outputs are scaled according to capacity and efficiency values that are specified for ARI rating conditions. Both high and moderate efficiency units will be considered in this study. For cooling, both sensible and total cooling capacities are determined. Iteration with the space-conditioning model is required, since the space humidity level is determined by the moisture removal rate of the equipment, which is affected by the mixed air humidity.

Models for a heat pump will also be taken from the ASHRAE Secondary Toolkit and adapted for modeling the heat pump heat recovery unit. Laboratory test data will be taken over a wide range of conditions and used to adjust coefficients of the model.

D. Cost Model

The cost model will consider utility and initial equipment costs to determine life-cycle costs (including inflation, alternative investments, taxes, financing, depreciation, maintenance, etc.). Utility rate information will be gathered for each location considered, including energy and demand rates. The life-cycle costs for different ventilation load technologies will be compared leading to an overall assessment.

III. SIMULATION INPUT DATA

A. Selected Locations

TMY2 (NREL, 1995) data for a number of locations in and near California will be used in the simulation studies. The National Renewable Energy Laboratory, NREL, has extracted data from the National Solar Research Data Base, NSRDB, for the years of 1961 to 1990 to produce the Typical Meteorological Year, or TMY weather data. TMY data is a set of hourly values of solar radiation and meteorological elements for a one-year period. It consists of months selected from individual years and concatenated to form a complete year. TMY2 data is a more recent version that was completed in March of 1994. Two minor errors that affected about 10% of the original TMY data stations were corrected in this version.

For this study, locations were selected from the available TMY2 data that are representative of diverse climates acrorss California. The selected cities are shown in Table 2.

TABLE 2. CALIFORNIA AND NEVADA CITIES FOR TRNSYS SIMULATIONS

	Latitude		Longitude		
City	Deg	Min	Deg	Min	Elev. (m)
Arcata	N40	59	W124	06	69
Bakersfield	N35	25	W119	03	150
Daggett	N34	52	W116	47	588
Fresno	N36	46	W119	43	100
Los Angeles	N33	56	W118	24	32
Sacramento	N38	31	W121	30	8
San Diego	N32	44	W117	10	9
San Francisco	N37	37	W122	23	5
Santa Maria	N34	54	W120	27	72
Reno, NV	N39	30	W119	47	1341
Las Vegas, NV	N36	05	W115	10	664

Arcata, San Francisco, Santa Maria, Los Angeles and San Diego are on the west coast of California proceeding from the north to south. These areas have very temperate climates averaging around 80°F and 40 to 50% relative humidity during the summer season. During winter months, the mean temperature drops to the low 40's and perhaps on occasion the upper 30's. Sacramento, Fresno, Bakersfield, and Baggett are inland from the west coast, approximately in the middle of the state. These areas are much hotter in the summer season, especially Bakersfield and Baggett. Las Vegas and Reno, Nevada, were both chosen to represent the eastern border area of California. Las Vegas temperatures range from the 20's during the winter and above 100°F during the summer. Climates near Reno are in the high 90's during the summer and lower teens in the winter.

B. Buildings

Brandemuehl and Braun (1999) considered four different types of buildings in their study: office, large retail store, school, and sit-down restaurant. Descriptions for these buildings were obtained from prototypical descriptions of commercial buildings developed by Lawrence Berkeley National Laboratory (Huang and Franconi, 1995). Table 3 gives data to describe these buildings. The current study will expand upon the previous work in that the cost effectiveness of DCV and other ventilation load reduction technologies will be considered and compared.

Table 3: Prototypical Building Characteristics use by Brandemuehl and Braun (1999)

Characteristic	Office	Large Retail	School	Sit-Down Restrnt.
Floor area (ft^2)	6600	80,000	9,600	5250
Floors	1	2	2	1
Percent glass	15	15	18	15
Window R-value	1.6	1.7	1.7	1.5
Window shading coeff.	0.75	0.76	0.73	0.80
Wall R-value	5.6	4.8	5.7	4.9
Roof R-value	12.6	12.0	13.3	13.2
Wall material	Masonry	Masonry	Masonry	Masonry
Roof material	Built-up	Built-up	Built-up	Built-up
Weekday hours (hrs/day)	11	14	Varies	17
Weekend hours (hrs/day)	5	14	Varies	17
Equipment power (W/ft^2)	0.5	0.4	0.8	2.0
Lighting power (W/ft^2)	1.7	1.6	1.8	2.1

Four additional building types from the LBL report will be considered in the current study: small retail stores, hotels, supermarkets, and middle schools. Tables 4, 5, 6 and 7 give data that describe these buildings. All of the simulated buildings will utilize packaged air conditioning equipment with a natural gas electric heater. For supermarkets, both old and new buildings will be simulated. The construction of this building type has changed dramatically over the last 30 years. However, many older buildings still are in commission and could be retrofit with ventilation load reduction technologies.

The LBL study consulted the 1989 CBECS (EIA, 1992) to determine total floor area for each building type, vintage, and climatic zone, the percentages of floor area heated or cooled, and the total energy use of the building type. The building shell characteristics and schedules were derived from the LBL study; however, the LBL study derived the data from a previous study conducted by (Huang et al., 1990) along with updates from the 1989 CBECS.

In addition to the buildings from the LBL study, the field site buildings will also be simulated. Site-specific data necessary for simulating system performance is currently being gathered (see report on the Description of Field Sites for Deliverables 2.1.1a and 3.1.1a). Once all data has been gathered from the field sites, this information will serve to validate the computer simulation model before any HVAC simulations are conducted for other buildings and locations. For DCV, the field sites have been chosen with two nearly identical buildings for each site. This will allow some degree of side-by-side testing for comparison of fixed minimum ventilation and DCV. However, more importantly, the test data will be used for validating the models and the predicted savings. Then, the improved models can be used to evaluate savings for the other technologies and locations.

TABLE 4. CHARACTERISTICS OF A MODELED SMALL RETAIL STORE

	Parameters
FLOOR-AREA	
Building area (ft^2)	6400
Floors	1
SHELL	
Percent Glass	15
Window R-value	1.67
Window shading co-efficient	0.84
Wall R-value	4.83
Roof R-value	12.04
Wall material	masonry
Roof material	built-up
OCCUPANCY	
Occupancy (ft^2/pers)	1635
Weekday hours (hrs/day)	12
Weekend hours (hrs/day)	4
EQUIPMENT	
Power density (W/ft^2)	0.50
Full Eqp hours (hrs/yr)	3480
LIGHTING	
Power density (W/ft^2)	1.7
Full lighting hours (hrs/yr)	4412
SYSTEM AND PLANT CHARACTERISTICS	
System type	Packaged single-zone w/ economizer
Heating plant	Gas furnace
Cooling plant	Direct expansion

TABLE 5. CHARACTERISTICS OF MODELED HOTEL PROTOTYPES

	Large hotels	Small hotels (Motels)
FLOOR-AREA		
Building area (ft^2)	250000	12000
Floors	10	2
SHELL		
Percent Glass	35	21
Window R-value	1.67	1.71
Window shading co-efficient	0.74	0.76
Wall R-value	6.16	5.32
Roof R-value	14.00	13.16
Wall material	masonry	masonry
Roof material	built-up	shingle/ siding
OCCUPANCY		
Occupancy (ft^2/pers)	210	120
Weekday hours (hrs/day)	24	24
Weekend hours (hrs/day)	24	24
EQUIPMENT		
Power density (W/ft^2)	0.72	0.69
Full Eqp hours (hrs/yr)	2722	2826
LIGHTING		
Power density (W/ft^2)	1.18	1.06
Full lighting hours (hrs/yr)	5157	3443
SYSTEM AND PLANT CHARACTERISTICS		
System type	Packaged single-zone w/ economizer	Packaged single-zone w/ economizer
Heating plant	Gas furnace	Gas furnace
Cooling plant	Direct expansion	Direct expansion

TABLE 6. CHARACTERISTICS OF MODELED SUPER-MARKETS

	Supermarket	
	old	new
FLOOR-AREA		
Building area (ft^2)	21300	21300
Floors	1	1
SHELL		
Percent Glass	15	15
Window R-value	1.51	1.60
Window shading co-efficient	0.82	0.79
Wall R-value	3.3	5.8
Roof R-value	9.2	11.8
Wall material	masonry	masonry
Roof material	shingle/ siding	shingle/ siding
OCCUPANCY		
Occupancy (ft^2/pers)	227	227
Weekday hours (hrs/day)	18	18
Weekend hours (hrs/day)	18	18
EQUIPMENT		
Power density (W/ft^2)	1.20	1.20
Full Eqp hours (hrs/yr)	5168	5168
LIGHTING		
Power density (W/ft^2)	2.4	2.4
Full lighting hours (hrs/yr)	7816	7816
SYSTEM AND PLANT CHARACTERISTICS		
Numer of systems	5 (office, storage, deli, bakery, sales)	
System type	Constant-vol. single-zone	Variable-air vol. single-zone
Heating plant	Gas furnace	
Cooling plant	Direct expansion	

TABLE 7. CHARACTERISTICS OF MODELED MIDDLE SCHOOL PROTOTYPE

	Parameters
FLOOR-AREA	
Building area (ft^2)	136000
Floors	1
SHELL	
Percent Glass	6
Window R-value	1.39
Window shading co-efficient	0.85
Wall R-value	2.38
Roof R-value	7.56
Wall material	masonry
Roof material	metal surface
OCCUPANCY	
Occupancy (ft^2/pers)	2085
Weekday hours (hrs/day)	12
Weekend hours (hrs/day)	4
EQUIPMENT	
Power density (W/ft^2)	0.30
Full Eqp hours (hrs/yr)	6462
LIGHTING	
Power density (W/ft^2)	0.8
Full lighting hours (hrs/yr)	3638
SYSTEM AND PLANT CHARACTERISTICS	
System type	Packaged single-zone w/ economizer
Heating plant	Gas furnace
Cooling plant	Direct expansion

IV. TESTING

A. Overview

Two distinct types of testing will be conducted for the DCV and ventilation heat pump heat recovery projects. First of all, the Carrier heat pump heat recovery unit will be tested in the laboratory over a wide range of conditions to be encountered in the field. These data will be used to build performance maps for the unit that will be integrated in the simulation tool. Secondly, field tests will be performed for DCV and heat pump heat recovery. An overview of the data flow for the testing and evaluation phase of these projects is given in Figure 4.

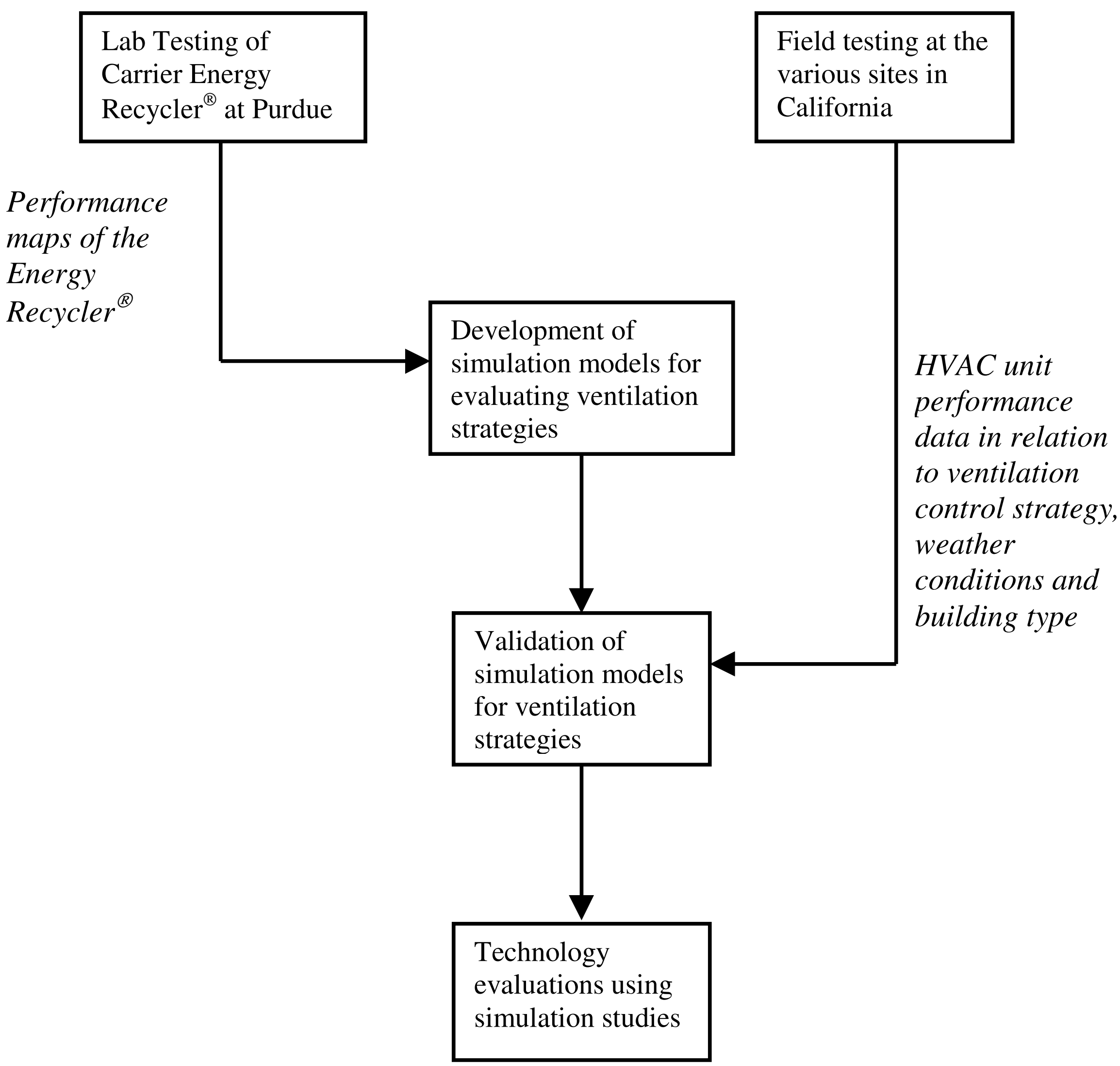

FIGURE 4. LAB AND FIELD TESTING DATA FLOW

B. Lab Testing of the Carrier Energy Recycler® Heat Pump

Project 4.2 is intended to demonstrate the savings potential for application of ventilation recovery heat pumps. This will be done primarily using simulation studies for various building and climate types found throughout California. To develop the simulation model, it is necessary to have accurate performance data for the ventilation recovery heat pump. Therefore, the first phase of Project 4.2 will focus on laboratory testing of a representative unit from Carrier. The environmental chambers at the Ray W. Herrick Laboratories will be used for this testing.

Carrier Corporation, as a sponsor of this program, has provided one of their Energy Recycler® units. This same unit will be used for both laboratory testing and field testing. The unit size was selected based on a field test site at a school that utilizes a Carrier 6-ton rooftop unit with gas heating. This unit was shipped to Purdue in late February of 2001. (See the photo in Figure 5.)

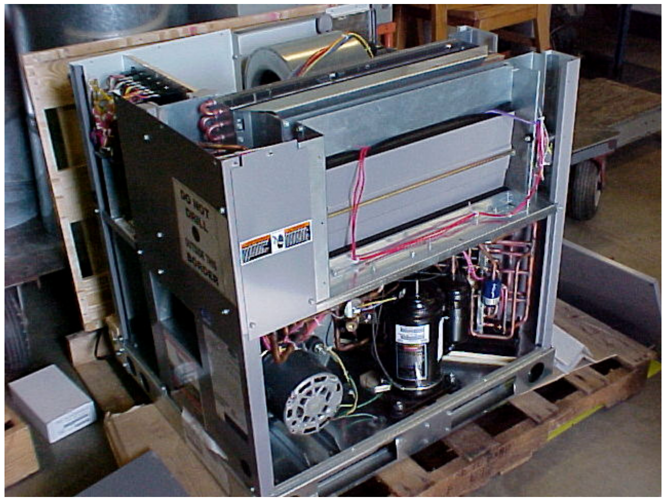

FIGURE 5. CARRIER ENERGY RECYCLER® HEAT PUMP AT HERRICK LAB
(SIDES REMOVED FOR CLARITY)

The ventilation heat pump is scheduled for testing at Herrick Laboratory beginning in May of 2001. The testing will result in a performance map of the unit that covers the complete expected operating envelope for the ambient and return air states. The expected range of the operating conditions for cooling and heating mode testing are given in Table 8. It is only necessary to vary humidity for the evaporator air stream (outside air for cooling mode and return air for heating mode) since performance is relatively independent of humidity when moisture is not condensed.

TABLE 8. OPERATING ENVELOPE FOR LAB TESTING OF THE CARRIER ENERGY RECYCLER® HEAT PUMP

Cooling Mode	
Ambient Temperature	50° to 120° F
Ambient Humidity	10% to 100%
Return Air Temperature	55° to 90° F
Return Air Humidity	Not varied
Heating Mode	
Ambient Temperature	-10° to 55° F
Ambient Humidity	Not varied
Return Air Temperature	50° to 80° F
Return Air Humidity	30% to 80%

The model will correlate sensible and total cooling capacity and power consumption as a function of the entering states and flow rates. The model will then be incorporated into the system model.

C. Field Testing

Field test data will be gathered at a total of 13 different sites in California: Twelve of the test sites are the ones being set up for joint evaluation of the demand controlled ventilation and the gathering of data for field evaluation of the fault detection and diagnostics algorithms. A detailed discussion of these sites and the test plan is included in the separate report: "Description of Field Test Sites".

The 13th site is for the heat pump heat recovery project. This site will be at one of the school districts (Woodland Joint Unified) where the modular schoolrooms are being monitored for DCV. The site selected is at the Junior High School for this district, and it has a 6-ton Carrier rooftop unit with gas heating.

The field testing for the ventilation recovery heat pump will involve two phases. The first phase, initiated in March 2001, was to install a Virtual Mechanic monitoring system on the existing rooftop unit at the California site. Performance data on this unit and the conditioned space will be collected for use in developing a baseline for the unit before installation of the Energy Recycler®. Once the laboratory testing is completed, the heat pump will be installed in the field and the second phase of the field testing initiated. It is anticipated that the field installation will occur during the July-August of 2001 time frame.

Table 9 gives a detailed list of the field test data for the Energy Recycler® as it will be set up for baseline data gathering. Additional sensors will be added when the Energy Recycler® is installed this summer. Detailed lists of test data for the other twelve field sites is contained in the Purdue report titled “Description of Field Test Sites”.

TABLE 9. DATA LIST FOR FIELD TESTING OF THE VENTILATION RECOVERY HEAT PUMP

Channel #	Data Point
SENSOR CHANNELS	
Power Transducer Channels	
1	Unit voltage, L1
2	Unit voltage, L2
3	Unit voltage, L3
4	Unit total current, L1
5	Not Used
6	Unit total current, L3
Other Analog Input Data	
7	SPARE - (Use later with heat pump)
8	SPARE - (Use later with heat pump)
9	SPARE - (Use later with heat pump)
10	SPARE - (Use later with heat pump)
11	SPARE - (Use later with heat pump)
12	SPARE - (Use later with heat pump)
13	SPARE - (Use later with heat pump)
14	SPARE - (Use later with heat pump)
15	Mixed air temperature
16	Return air temperature
17	Supply air temperature, before heater
18	Supply air temperature, after heater
19	Condenser inlet air temperature
20	Condenser outlet air temperature
21	Suction line temperature, rooftop unit
22	Discharge line temperature, rooftop unit
23	SPARE - (Use later with heat pump)
24	SPARE - (Use later with heat pump)
25	Evaporation temperature, rooftop unit
26	Condensation temperature, rooftop unit
27	Outdoor air temperature
28	Outdoor air humidity
29	Building zone temperature A
30	Building zone temperature B
31	Building zone temperature C
32	Building zone temperature D
CALCULATED DATA CHANNELS	
33-50	NOT USED
51-56	NOT USED

TABLE 9. DATA LIST FOR FIELD TESTING OF THE
VENTILATION RECOVERY HEAT PUMP (CONT'D)

Channel	Data Point
57	superheat, stage 1
58	subcooling, stage 1
59	evaporating temperature, stage 1
60	condensing temperature, stage 1
61	condensing temperature over ambient (CT-AIC), stage 1
62	NOT USED
63	NOT USED
64	NOT USED
65	NOT USED
66	NOT USED
67	evaporator temperature difference (RA-SA)
68	NOT USED
69	NOT USED
70	unit power (kW)
71	unit KWh
72	unit MWh
73	compressor 1 power (kW)
74	compressor 1 KWh
75	compressor 1 MWh
76	compressor Vent Heat Pump Unit power (kW)
77	compressor Vent Heat Pump Unit KWh
78	compressor Vent Heat Pump Unit MWh
79	digital input 1, supply fan, run time (8 hours)
80	digital input 1, supply fan, run time (seconds)
81	digital input 2, cooling 1, run time (8 hours)
82	digital input 2, cooling 1, run time (seconds)
83	digital input 3, cooling Vent HP, run time (8 hours)
84	digital input 3, cooling Vent HP, run time (seconds)
85	digital input 4, heat 1, run time (8 hours)
86	digital input 4, heat 1, run time (seconds)
87	digital input 5, heat Vent Heat pump, run time (8 hours)
88	digital input 5, heat Vent Heat pump, run time (seconds)
89	digital input 6 run time (8 hours)
90	digital input 6 run time (seconds)
91	time since reset accumulators (8 hours)
92	time since reset accumulators (seconds)
93	up time (8 hours)
94	up time (seconds)
95	board temperature (F)
96	board battery voltage (V)

TABLE 9. DATA LIST FOR FIELD TESTING OF THE VENTILATION RECOVERY HEAT PUMP (CONT'D)

Digital Channels

1	Supply fan contact (fan om / fan off)
2	Low voltage control signal for compressor main unit
3	Low voltage control signal for compressor, heat pump
4	Heating mode signal
5	
6	

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