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Integrating Air Handling Units in Office Buildings for High Performance

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Integrating Air Handling Units in Office Buildings for High Performance

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Integrating Air Handling Units in Office Buildings for High Performance

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Office buildings consume about fifty-five percent of their total energy use to heat and cool the spaces. As much as thirty percent of the energy consumed in office buildings was found to be wasted. Although the efficiency of heating/cooling energy use in office buildings has increased considerably in general, the total amount of thermal and mechanical energy demand has not decreased. The trend of increasing in building energy consumption will continue due to the expansion of office built area and the associated energy needs.

This study investigates the thermal load features in office buildings and proposes an innovative Integrated Air Handling Unit (IAHU) concept in order to achieve energy savings with conventional office building air handling systems. The corresponding deduction of IAHU for an acceptable Indoor Air Quality (IAQ) and better energy performance is conducted. The system variables and constraints are analyzed in detail to understand the feasibility and operability of IAHU. The control logics and implementation methods are elaborated for typical system layouts. With an IAHU operation, the internal heat gain can be transferred from an interior region into an exterior region in winter. The sensible load and latent coil load can also be decoupled in mild weather.

To evaluate the performance of IAHU for buildings, especially insufficiently sub-metered buildings, a simplified simulation method is proposed. The theoretical modeling

process is provided. Through a case building simulation, it is found that, by converting a Two Air Handling Unit (TAHU) system into an IAHU system, about 14% of thermal energy can be saved for the case building, which is equivalent to a 3.5 MBTU/ft^2 yr saving in the given climate. By transferring the internal heat gain from the interior region to the exterior region, 58% of the total savings, by applying IAHU, can be achieved in winter time and 17% in swing seasons. Another 25% savings comes from the sensible and latent coil load decoupling of using IAHU in summer mild weather.

The study concludes that IAHU can be generalized as an operational method and adopted into new and existing office buildings for high performance.

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DEDICATION

I dedicate this thesis first to my wonderful wife Daihong Yu for all the love and support during the years of study. I would also like to dedicate this thesis to my lovely daughter, Pearl, for bringing me joy and happiness.

献辞

谨将此论文献给我亲爱的妻子，余代红， 感谢她在这些年的学习和生活中给予我的爱和支持。同时也献给我可爱的女儿，余佩儿，她的出现使我的生活有了更多的乐趣。

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NOMENCLATURE

Roman Letter Symbols

A, *AG*	Area [m^2]
ACH	Air change rate [/hr]
C	Contaminant concentration [g/m^3]
CFM	Air volume flow rate [m^3/s]
C_g	Contaminant generating rate [g/s]
C_p	Air specific heat [kJ/kg K]
E	Thermal energy rate[kW]
H, *h*	Enthalpy [kJ], specific enthalpy [kJ/kg]
H	Fan head [Pa]
Load	Zone thermal load [kW]
M	Air mass flow rate [kg/s], Heat gain coefficient [kW/°C]
N	Rotational speed [/s]
Q	Contaminant mass change rate[g/s], Heat gain [kW]
q	Power or load density [W/m^2]
t	Time [s]
T	Temperature [K]
U, *UG*	Heat transfer coefficient [W/m^2 K]
V	Air volume [m^3] , zone volume [m^3]
W	Power [kW]

Greek symbols

β	Outdoor air ratio
$\beta_{i,IAQ}$	Interior zone OA ratio for acceptable IAQ in TAHU mode
$\beta_{e,IAQ,d}$	Exterior zone OA ratio for acceptable IAQ in TAHU mode

λ	Exhaust air ratio from the exterior zone
η	Efficiency
γ	Circulation air ratio
φ	Interior zone supply airflow rate ratio
ξ	Interior zone relief air ratio
δ	Exhaust air ratio from the interior zone
μ	Occupancy ratio, partial load ratio
ω	Normalized rotational speed, humidity ratio [g/kg]
ρ	Density [kg/m^3]

Subscripts or coefficient

a, b, c, d, k, l	Coefficients
adjn	Adjusted, north
adj	Adjusted
c	Cold deck
cc	Cooling coil
cn	Control volume
cr	Critical
d	Design condition
e	Exterior zone
eco	Economizer
eqt	Equipment
F	Floor
fan	Fan
fg	Water vapor
g	Generate

gl	Glass
hc	Heating coil
I	Interior zone
id	Induction unit
in	In
inf	Infiltration
jan ,jul	January, July
l	Latent
ltg	Lighting
m	Motor
min	Minimum
mix	Mixed air
o	Outdoor
oa	Outdoor air
out	Out
pc	Peak cooling
pe	People
ph	Peak heating
r	Room air
r2	Secondary coil
rh	Reheat coil
run	Operation time
s	Supply air, sensible air
shf	Shaft
sol	Solar

thm	Thermal
ts	Transmission with solar effect
wl	Wall

Abbreviations

AHU	Air handling unit
ASHRAE	American society of heating, refrigerating and air-conditioning engineering
CAV	Constant air volume
CC	Cooling coil
CLFTOT	Total cooling load factor
CLTDS	Cooling load temperature difference
DDC	Direct digital control
EA	Exhaust air
EMCS	Energy management control system
EPA	Environmental protection agency
ESL	Energy systems lab
FAS	Fan airflow station
FPS	Fraction of possible sunshine
HC	Heating coil
HRC	Heating recovery chiller
HRV	Heat recovery wheel
HVAC	Heating, ventilation and air-conditioning system
IAHU	Integrating air handling unit
IAQ	Indoor air quality
MSHGF	Maximum solar heat gain factor
OA	Outdoor air

OAHU	Office air handling unit
RA	Room air
RAF	Return air fan
SAF	Supply air fan
SAHU	Single air handling system
SC	Shading coefficient
SLF	Sunlit factor
TAHU	Two dedicated air handling unit
VAV	Variable air volume
VFD	Variable frequency drive
WLHP	Water loop heat pump

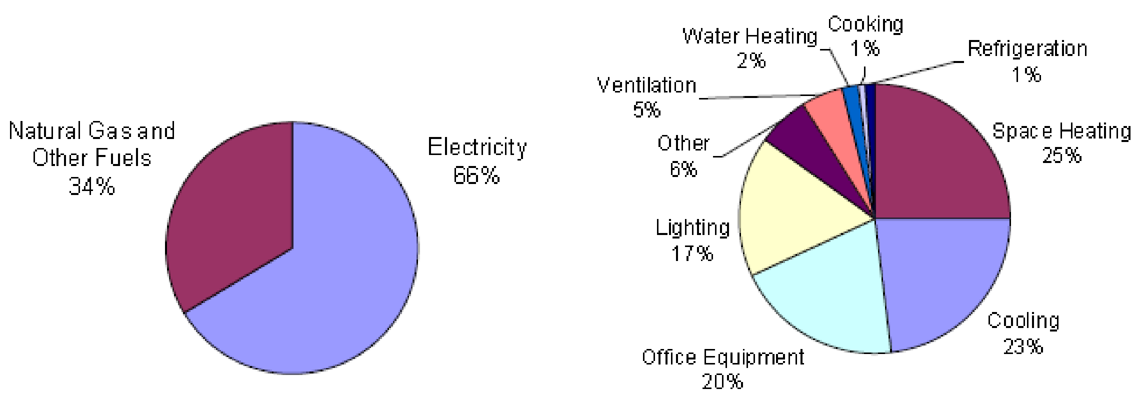
Chapter 1 INTRODUCTION

1.1 Background

1.1.1 General

In commercial buildings the energy needs for building services systems often account for a substantial portion of total energy consumption. Within the commercial sector, office buildings are those with the largest consumption of energy and CO_2 emissions. In the United States, offices account for 17% of the total non-domestic building area and about 18% of the energy use in buildings (Luis et al. 2008).

Across the US, the average annual energy intensity for office buildings is 79.8 kBtu per square foot and the average cost is $1.65 per square foot. Of the total energy consumption, 66% is from electricity and 34% is from natural gas and other fuels. This consumption translates to 15.5 kWh per square foot of electricity and 0.27 therms (32 cubic feet) per square foot of natural gas (CBECS, 2003).



Adopted from *E Source*, 2006

Figure 1-1: Energy Consumption (Left: by Fuel Type, Right: by End Use)

As shown in Fig 1-1, the energy consumption on space heating, ventilation, and air-conditioning (HVAC) represents about fifty five percent of the total use in a typical office building. In another study, it is stated that energy represents about nineteen percent of total expenditures for a typical office building ("Managing Energy Costs in Office Buildings", 2006). As much as 30% of the energy consumed in office buildings was estimated to be wasted in daily operation ("Office Building Energy Use Profile", 2006).

Although the efficiency of heating/cooling energy use in office buildings has increased considerably in general because of better building insulation, component efficiency and automated control, etc, the amount of total thermal and mechanical energy demand for commercial buildings has not decreased. This increasing trend in building energy consumption will continue due to the expansion of office built areas and the associated energy needs.

The above facts clearly mark the HVAC of office buildings an important sector which deserves both management attention and systematic research.

1.1.2 Office building thermal features

The office building is one of the great icons of our modern world. It differs from other commercial buildings due to its diversified building layout and corresponding occupancy and operation. An office building's layout can be a mixture of space divisions, private offices, open plans and auxiliary space, etc. The characteristics of an office building lead to the configuration complexity of HVAC systems. Furthermore, the control and operation of air conditioning systems are also difficult if a high energy performance is desired.

Compared to those of earlier times, modern office buildings have higher internal heat gains since more electricity-powered equipment is employed. The higher heat gains reflect from both the quantity and the density. Computers, copy machines, printers, even data servers are commonplace in office buildings. Although the efficiency and the convenience have been highly improved, a significant amount of heat exhausted by these equipment enters into the air conditioned spaces. If not properly handled, the heat can turn into a thermal load that requires mechanical cooling year around.

In line with the prevailing architectural style, office buildings are built with large glass surfaces and becoming more air-tight with wall curtains and non-operable windows. The area along the perimeter of a typical office building and the area in the core possess very different load characteristics. Normally, perimeter zones are mainly influenced by the outside air (OA) temperature and solar variation, while interior zones are dominated by the internal heat load density and occupancy schedules. The unique thermal feature of office buildings leads to a common practice that heating and cooling exist simultaneously under some circumstances in winter or swing seasons. In the context of this thesis, the definition of “zone” is extended to be the same as “region”, which includes multiple zones that possess the identical thermal characteristics.

Unless specially designed, natural ventilation is almost impossible for most existing and newly built office buildings. Because of the improved closeness, increasing occupancy density and longer residence time, the indoor air quality (IAQ) of modern office buildings is now a major concern. To acquire ventilation air for the occupants, either an individual system, or a system with a proper amount of OA should be installed. ASHRAE standard 62.1-2004 (ASHRAE 62.1, 2004), the code of Ventilation for

Acceptable Indoor Air Quality, specifies the calculation of minimum ventilation rates for an acceptable IAQ in residential and commercial buildings.

Office building spaces are normally open with continuous areas above the ceiling and no floor-to-floor partition walls or locked doors. To achieve the maximum use of area, work stations are created near windows and outer walls. In the center of the building, stair wells, elevators and hallways are located for passage convenience. Therefore, the exterior zones and interior zones do not have obvious and rigid boundaries. The two zones influence each other in a way in which it is difficult to be clearly isolated. The terminals of the systems serving different zones also might be put in the same office room, which causes the ventilation air from different systems in the occupant space mix and network spontaneously.

1.1.3 Office building HVAC systems

The comfort of occupants in office spaces is fundamentally influenced by the thermal and air quality. HVAC systems are designed to provide workers in office buildings with a suitable air temperature, humidity and quality. HVAC systems can vary greatly in complexity, from stand-alone units that serve individual rooms to large, centralized systems serving multiple zones in a building. In project EPA 402-C-06-002 for building assessment survey and evaluation study, the EPA randomly selected 100 public and commercial office buildings, which were built from before 1900 to 2000, in 37 cities and 25 states (“BASE Study”). The majority of existing office buildings have centralized cooling and heating systems, as shown in Fig 1-2.

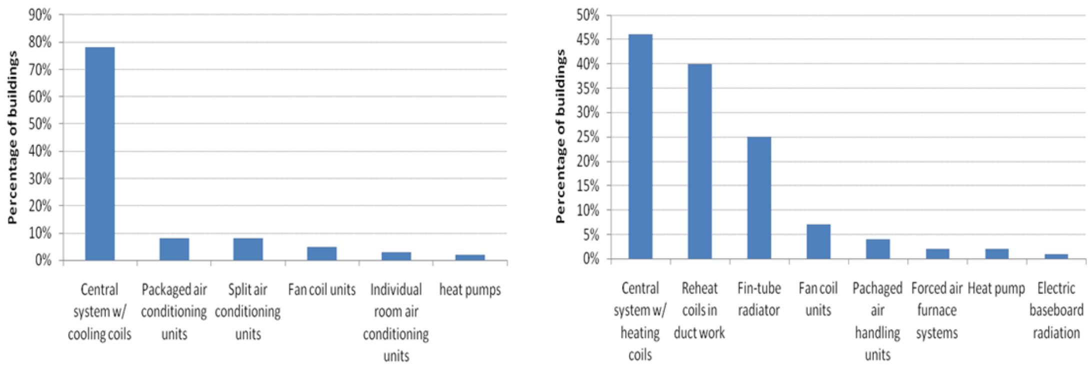


Figure 1-2: Office building cooling/ heating system type

Among them, 98% use mechanical ventilation; 50 out of 141 air handlers are constant air volume (CAV), while the others are variable air volume (VAV). Air based centralized HVAC systems with different terminals and layouts still dominate office buildings for air conditioning because of the associated advantages. A DOE report about energy consumption characteristics of commercial buildings shows that central systems with VAV air handling units (AHUs) are more efficient than packaged systems (Roth et al, 2002). The other reason of the dominance is that a central HVAC system is more flexible in providing air distribution with duct work throughout the entire building.

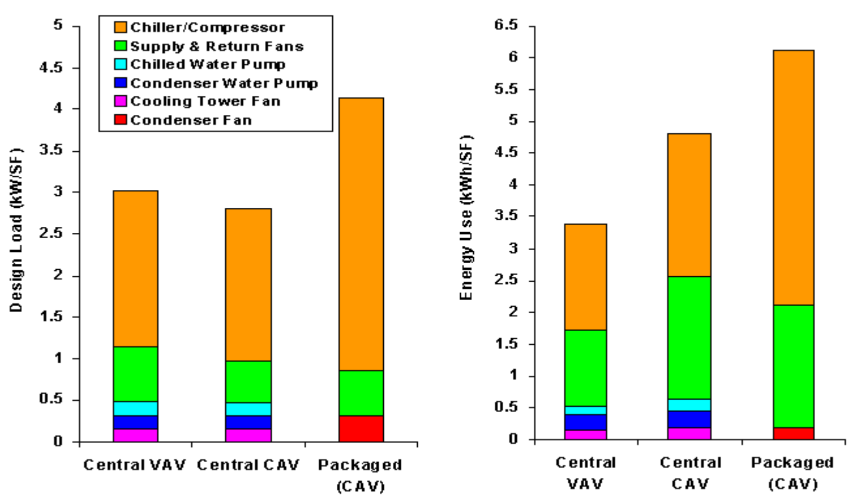


Figure 1-3: Design load and energy use comparison cross systems

There are many different types of air based HVAC systems, which have evolved gradually for better control, energy savings or thermal comfort considerations. A CAV system has the features of simplicity, low cost and reliability. It does not change air delivery rates when building load changes. To accommodate the varying building load for thermal comfort, it either mixes cold air with warm air or use terminal reheats to offset the excessive cooling. The air distribution in the conditioned space can be easily ensured with this type of system.

In many applications, pure CAVs are not energy conservative and are becoming rare in newer construction. Some changes in the system operation and configuration have been adopted to preserve the original simplicity while achieving better energy conservation. Resetting the supply air temperature, supply water temperature or water flow rate can be utilized in a CAV and induction terminal system for building perimeter air conditioning.

The airflow to a conditioned space could be adjusted to the needed rate with a VAV system. The system supplies cold air to the space via terminal damper boxes. Depending on the system layout, the terminal boxes may or may not be equipped with a reheat coil. Additional heat is provided by a reheat coil if the space needs less cooling than that delivered by the supply air at the box’s minimum airflow rate setting. Since the air can be adjusted at the terminal side with both rate and temperature, VAV is considered more energy conservative under most circumstances. It has been used in many applications where the space cooling/heating loads have a wide range variation with occupancy activities and external air conditions.

In a large or multi-story office building, more than one AHU is needed to accommodate the load variances across zones. As stated in the previous section, the office building load can be characterized as interior and exterior. Since the central interior area is less influenced by outside conditions, the space is mainly cooling dominant. The variance of load within the interior zone seldom changes the cooling dominancy throughout the seasons. A constant supply air temperature at 55^{o}F is commonly used so that the building humidity and temperature can be maintained with relatively low fan power consumption. The system's minimum airflow rate varies from 30% to 50%, or higher, which depends on factors including minimum OA intake, mechanical fan features and air circulation requirements.

Circumferential perimeter zones in an office building can also be conditioned by one or more AHUs. Since perimeter zones have strong correlations between building envelopes, outside conditions and orientations, the load across the zones could have significantly different features. The extreme situation is that cooling and heating coexists in the zones. A CAV with induction units and a VAV with terminal reheats are two common types of air-based systems that are applied in perimeter zones for flexibility. The AHUs are controlled to adjust the supply air temperature, or along with the flow rate in VAV, so that cooling/heating penalties can be avoided mostly from the system level.

Deployed along with the mechanical HVAC system is the building automation system (BAS) or energy management control system (EMCS). The first generation of pneumatic controls and then electric controls have been widely replaced by direct digital control (DDC), which emerged in the 1970's. With the communication and implementation advantages of DDC, computer aided EMCS can achieve a high level of automation and optimization with HVAC equipment and systems. It is estimated that energy savings resulting from the installation of an EMCS in a typical commercial building is about 5% (Massachusetts Market Transformation Scoping Study, 1997). Sensors, controllers, actuators and variable frequency drives (VFD) are networked to fulfill the control algorithm in supervisory level controllers.

EMCS with DDC technology provides the basic infrastructure where a building level operational optimization could be implemented in office building HVAC systems instead of local optimization, as studied in this research.

1.1.4 Ventilation control

OA is needed in occupied buildings to first ensure the building space ventilation for an acceptable IAQ, and to also maintain a proper building positive pressure. Depending on outdoor conditions, the air may need to be heated/cooled or humidified/dehumidified before it is distributed into the space. As OA is drawn into a building, the indoor air is re-circulated, exhausted or relieved. OA can be introduced into the spaces by mechanical fans, passive vents or operable windows. Ventilation rates for commercial buildings are codified as part of the requirements in state and federal energy standards in the USA, often based on the recommendations in ASHRAE Standard 62.1.

Natural ventilation, mechanical ventilation and hybrid ventilation are the means to introduce OA. Natural ventilation was the most common ventilation method, which let fresh air come in without thermal processing through operable windows or special designed vents. In this mode, wind pressure and thermal pressure are the main drives to move the air in and out of the building. A special building layout is needed so that OA can travel through the building and maintain the average indoor air temperature and other parameters in an acceptable range. Naturally ventilated buildings usually have a wider air temperature band which might lead to discomfort. Nowadays natural ventilation is no longer the best strategy for office buildings that are air sealed.

Modern office buildings generally use mechanical ventilation systems to introduce OA. The quantity of OA introduced into an AHU is typically controlled by coordinated action of the relief, mixed and outside air dampers. During most times of the year, OA is not suitable for direct air conditioning purposes. A thermal process is needed so that the supply air can accommodate the building's thermal and moisture load. OA is subjected to a minimum limit in order to control the associated thermal energy consumption and provide basic ventilation.

In addition to maintaining an IAQ, OA might also be used as a cooling medium in a swing season or during winter to offset the surplus heat gain from lighting, equipment and occupants. When the enthalpy or temperature of OA is lower than that of the return air, it could be economical to use up to 100% OA. The temperature or enthalpy based economizer control is recommended by ASHRAE as an energy conservation technology for operating air-conditioning systems (ASHRAE, 90.1, 2004).

The minimum OA intake is usually controlled by fixed position OA dampers. In a study conducted by EPA, about 90% percent of the sampled office buildings used this measure while less than 1% had intake airflow monitoring ("BASE Buildings Test Space HVAC Characteristics"). However, a fixed position damper can hardly guarantee the amount of OA intake, especially in a VAV system where the total airflow rate is modulated according to the changing load (Mumma and Wong, 1990). Traditional intake airflow stations seem like a logical solution, but the reality shows that they either result in a large pressure drop or the measurement is far from accurate (Kettler, 1998). Persily et al (2005) reported that in a survey from 1994-1999, of 100 randomly selected large office buildings in the US , the average of measured OA supply rates was 120cfm/person using duct traverse measurements, while the required OA is about 12cfm/person.

Over-ventilated buildings waste energy with little or no benefit to the occupants, while under-ventilated buildings may have significant adverse effects on occupants. An unbalanced OA control will also lead to bad building pressure and moisture control during cooling seasons. ASHRAE 62.1 allows the designer to take credit for “unused” ventilation air returning from the over-ventilated spaces in the system since variable occupant density is usually lower than the designed condition. Indoor air CO_2 levels can be used as the IAQ index to control the building’s overall OA intake.

1.2 Objective and scope

The aim of this research is to:

- Develop an innovative approach for the high performance of office building HVAC systems by virtually integrating AHUs (IAHU);
- Introduce fan-law based airflow measurement methods and instrumentations for better air distribution and building pressure control;
- Analyze parameters and their influences on IAHU performance and develop practical control algorithms for IAHU operation;
- Develop a practical evaluation process for the performance of an IAHU system and demonstrate it with a given building system layout.

IAHU is developed for implementation in existing and new buildings with few or no retrofitting on conventional air-based HVAC systems. VFDs are recommended on all AHU supply and return fans so that a flexible modulation and coordination across AHUs can be accomplished. A DDC based EMCS should be available to enable building level communication and control.

This paper theoretically deduces and investigates the basic theory of IAHU for office buildings to improve energy performance while maintaining the IAQ. Based on IAQ constraints, the optimal year round operation schema are defined. The involved parameters are evaluated in terms of their influences on IAHU. The key implementation elements are introduced followed by a discussion of control for four typical office building system layouts. The evaluation process is then detailed in a simple steady state simulation and demonstrated in a building case.

1.3 Methodology

A special feature of this study is to enhance the operation of existing and new office buildings by integrating the AHUs for better OA intake control with little or no retrofitting. The internal heat gain can be transferred to the external heating area in winter by using IAHU. Meanwhile, the latent load and sensible coil load can possibly be decoupled in summer for further energy savings.

The OA intake and distribution are critical to ensure the success of IAHU in office buildings. The method of controlling fans and dampers based on fan laws is analyzed and the control algorithms for typical office system layouts are described in this study.

The methodology comprises four different ways of studies:

- Literature review;
- Theoretical modeling and deduction;
- Engineering analysis and control algorithm development;
- Mathematical simulation.

A study of office air handling units (OAHU) has been conducted by Dr. Li Song in a PhD dissertation to improve energy savings of HVAC system in office buildings (Song, 2005). However, a piece of duct work is needed to enable the operation of such an improved AHU system. This could potentially be a big obstacle in a real application since such a retrofitting would interrupt the daily operation of an office building. Moreover, the parameters involved in the control are complicated and theoretical. The measurements and calculations deduced in theory for OAHU might not be practical for real implementation.

IAHU is proposed in this study based on OAHU to eliminate the necessity of duct work retrofit in upgrading a two dedicated air handling unit (TAHU). The operation and control algorithm are reasonably simplified to crop major benefits of the system with improved viability. The IAHU theory with the innovative OA intake methods for both an acceptable IAQ and additional energy savings are deduced. The variables in upgrading a TAHU into an IAHU system have been analyzed before the description of a control algorithm and implementation methodology. A simulation process to evaluate an IAHU system is given later and demonstrated with an existing office building.

1.4 Outline of the thesis

The introductory portion, chapter 1, provides the background and the aims of the current study. The description of the chosen methodology is included. A concise literature review with technology related to the study is also provided.

In chapter 2, the conventional office building HVAC system operation is first summarized. The concept and theoretical deduction of IAHU are presented for an acceptable IAQ. The analysis of energy consumption features of an IAHU system is included here as well. The IAHU control is defined with clear operation criteria after the two step analysis.

The parameters involved in the conversion from a TAHU to an IAHU system are studied in chapter 3. Fan law based airflow measurements and the control are introduced to support the implementation of IAHU. Control algorithms for real application are developed based on the theoretical analysis given in chapter 2. The involved instrumentations for an IAHU operation are then concisely discussed.

Chapter 4 provides a simulation process for an IAHU performance evaluation. An existing office building is evaluated with this method as a demonstration.

The discussion and conclusion of IAHU are reviewed in chapter 5.

1.5 Literature review

Due to their closeness and occupant density, most modern office buildings must rely on mechanical HVAC systems to provide OA ventilation and air conditioning for acceptable indoor environments. Centralized air based HVAC systems, such as CAV and VAV, still dominate in existing and new office buildings as conventions for mechanical ventilation and air conditioning. Due to the prior fan energy savings potential, VAV systems are gaining more popularity than CAV. This dominance is not likely to change in the near future.

With the office building load features stated above, in winter time, the core zones require cooling due to large internal heat gain, while the perimeter zones require a large amount of heating to offset the heat loss through building envelopes. The quantitative contradiction can be significant when the cooling and the heating are close to equal. For energy saving purposes, more than one AHU is normally deployed for large area or multistory office buildings, so that the efficiency compromising simultaneous heating and cooling can be attenuated.

It is more energy efficient to transfer the internal heat gain to the perimeter zones in winter and swing seasons. Heat recovery chillers (HRC) have been studied to recycle heat gains (Nichols and Laframboise, 1984). They provide chilled water to cool the air supplied to the interior zone and use condenser water to warm up the air supplied to the exterior zone. However, a heat recovery chiller system needs to operate even during a free cooling season. For most office buildings, such a system is not the best solution to transfer the interior zones heat to the exterior zones.

For the same purposes, a water loop heat pump system (WLHP) was developed (ASHRAE, 2008). The water-to-air heat pump units in each individual zone are distributed hydraulically with a two-pipe water loop. The units in the cooling cycle reject heat into the pipe system, while the units in the heating cycle retrieve heat from the pipe system. Through the transfer, the exterior zone can be heated by the interior zone waste heat with little or no external heat. However, as in most decentralized systems, it has potential IAQ and noise problems. Furthermore, it requires an additional boiler and cooling tower to balance the loads.

An OAHU concept has been proposed and studied by Dr. Li Song to utilize air based systems for transferring internal heat gain (Song, 2005). Two AHUs, which serve the interior and the exterior zone respectively, are duct-linked together on the AHU side. The OA intake and return air of the two AHUs can be manipulated to transfer internal heat gain while maintaining an acceptable IAQ for both zones. The latent cooling and sensible cooling can also be decoupled with this system design to prevent unnecessary reheat in mild weather. Ventilation by taking credits of unused fresh air is well considered for office buildings under the concept of OAHU. With DDC based EMCS, OAHU breaks the conventional operation frame of individual AHU in office buildings.

However, since an OAHU requires duct work retrofitting between two individual AHUs, the application is limited in existing office buildings. Four airflow meters are needed to enable the control, which might also increase the difficulty and resistance of generalization. In addition, the operation scenarios are complicated and suitable mainly for theoretical analysis.

Chapter 2 VIRTUAL AIR HANDLING UNIT INTEGRATION

2.1 Conventional system and operation

A core zone plus perimeter zone layout has been widely accepted for air based HVAC systems in office buildings. To accommodate the different load features of the zones for higher effectiveness, two or more AHUs are needed. From the design to the construction and the operation, an individual AHU in a building is conventionally confined only to itself and the corresponding zones. Supervisory level and local level controllers, actuators, and thermal components involved in the zones of an AHU work accordingly under a defined control algorithm and sequence.

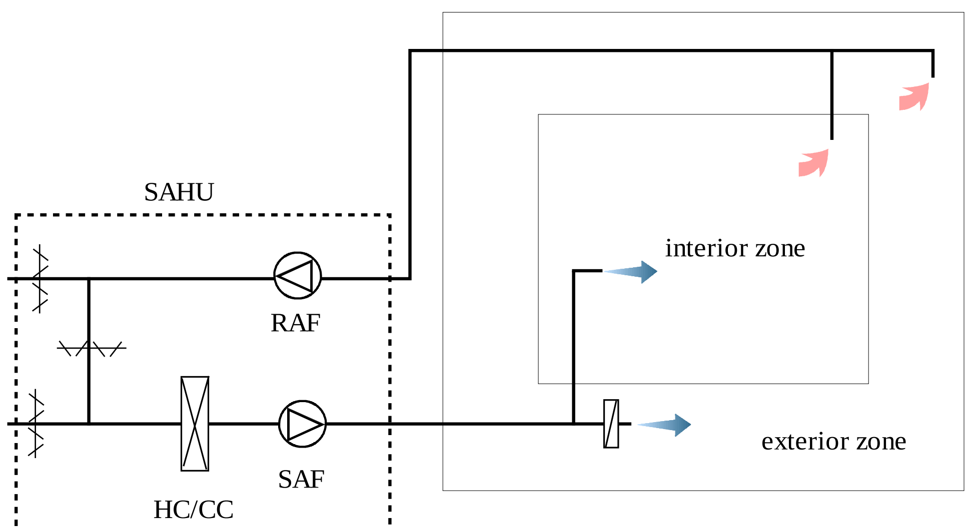


Figure 2-1: Illustration of an SAHU

A single air handling system (SAHU) is a typical air-based centralized system for commercial buildings. One conditioned air stream is distributed into the space for heating, cooling, humidification and dehumidification. Heating and cooling coils are installed in series in the AHU. For an office building, where the interior zone and exterior zone are served by the single system, the AHU working mode is mainly determined by the interior zone. Fig 2-1 illustrates the layout of an SAHU system.

Depending on the zone load features, the terminals in an SAHU system can be as simple as a reheating coil, throttling damper, or combined box. To satisfy the interior and exterior zone, typically only cooled air at a low temperature is supplied from the AHU. The cold air is throttled or reheated if heating is needed by minority zones. The cooling coil, heating coil and air dampers are coordinated to achieve the desired supply air temperature. The OA intake for the building ventilation is mixed with/without the return air before being supplied into the space. As one of its benefits, during an economizer period, free cool OA can be utilized to save mechanical cooling energy.

Since the supply air is used to condition different zones, there exists a compromise when the zones are in opposite modes. The supply air temperature reset for an SAHU system can only be conducted in a manner where the interior cooling demand is not impaired and the fan does not need to blow too hard. Significant reheat energy could be consumed since the terminal reheat applies to permit zone or space control for areas of unequal loading, or to provide heating or cooling for perimeter areas with different exposures (Mcquiston et al, 2000). Therefore, although it is typical and simple, SAHU is not energy efficient for modern office buildings.

To overcome the drawback of an SAHU system for space conditioning, a two dedicated AHU system (TAHU) was later used. An interior zone and an exterior zone have their own dedicated SAHUs; the individual SAHU of TAHU has a similar operation as a single SAHU but with better adjustability. As an air-based HVAC system, a TAHU system inherits the advantages of an SAHU system such as economizer control. Two separate units supply dedicated conditioned air to the interior zone and exterior zone, respectively.

In a TAHU system, the two zones' air conditioning is separated from each other. The supply air temperature of the exterior zone in winter can be reset to a higher value without the restriction from the interior zone which requires cooling year round. Meanwhile, for the interior zone, since the supply air temperature can be maintained at a low set point, the fan power can be saved with VFDs when the load is low. With the separation, one of the main benefits is that a significant amount of unnecessary reheat consumption can be avoided.

Fig 2-2 plots the diagram of a typical TAHU.

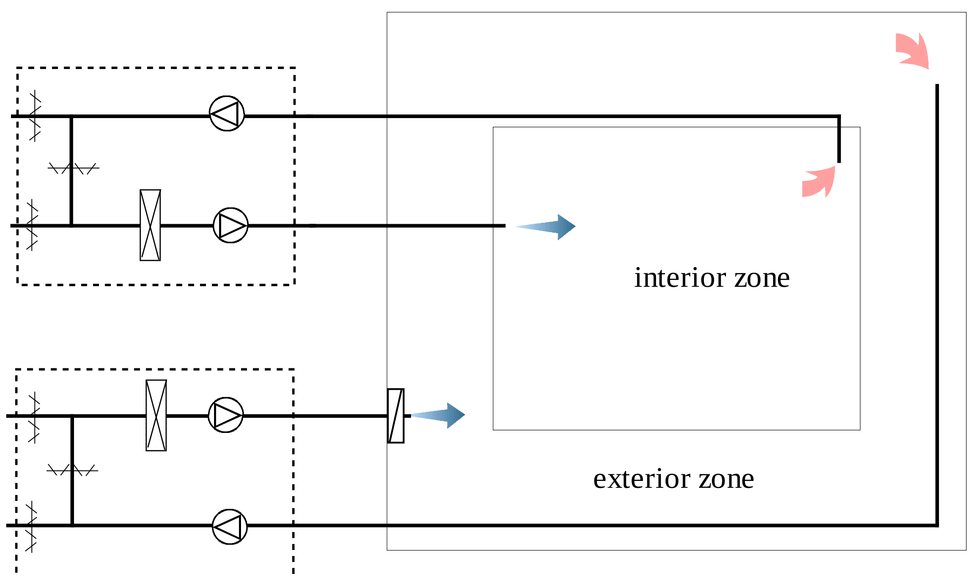


Figure 2-2: Illustration of a TAHU

Not only is the thermal processing of the two zones separated, but so are the OA intake and ventilation. Both AHUs need to ensure the minimum OA intake for the corresponding zones which could be a shortcoming of the system. For humidity control purposes during a humid mild season, the exterior AHU has to use a cooling coil to remove the moisture from the OA regardless of the space sensible load. Reheating will be activated if the conditioned air is cooled, in order to remove the moisture, more than needed to offset the sensible load. In addition, in winter, when the interior zone AHU disposes a substantial amount of heat through exhaust or relief air due to the internal heat gains, external heating is required by the exterior zone.

TAHU's two-zone deployment is better than SAHU in terms of energy conservation, since the adverse interaction of the thermal load between the interior zone and the exterior zone can be largely decoupled. However, there is still potential for energy efficiency improvements on the two points mentioned above:

a. Decouple the moisture load from thermal load for less cooling and heating conflict in humid and mild weather;
b. Recover/transfer the internal heat gain from the interior zone to the exterior zone in winter.

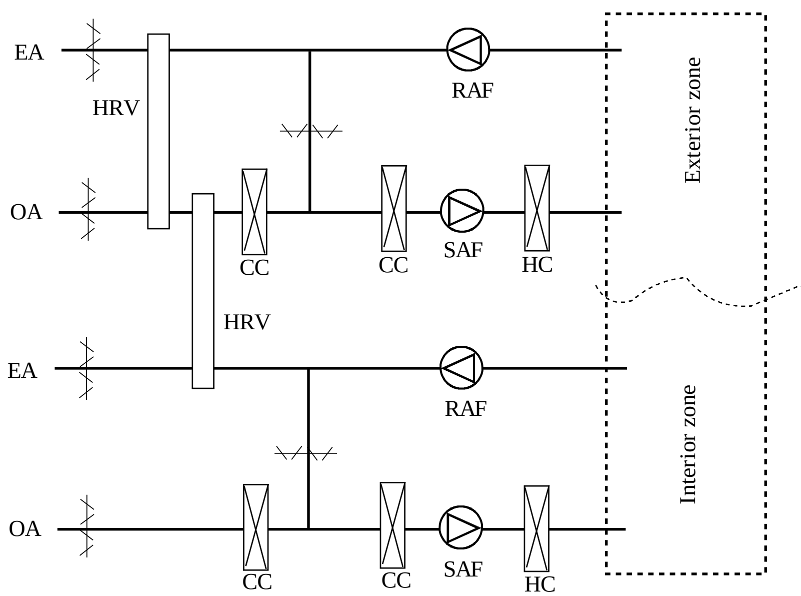


Figure 2-3: Illustration of an upgraded TAHU system

One possible solution is to install another cooling coil for OA intake only, and then use a heat recovery wheel to recycle the heat/cool capacity from the exhaust and relief air. The illustration of such a system is shown in Fig 2-3.

Using a separate cooling coil for OA, as depicted in the figure, can be referred to as a dual path system (Khattar and Brandemuehl, 2002). The original intention was to facilitate AHU's dual function of maintaining a thermal environment and providing necessary ventilation. The ventilation air passes through its dedicated cooling coil to remove the moisture. Since OA is also the main source of the latent load for most office buildings, this modification can ensure effective humidity control.

If the ventilation air is dehumidified to be dry, it can further carry internal latent load. In such a system, the circulation air is provided with another cooling coil to facilitate the control of supply air temperature for the sensible load of the space. Therefore, the latent and sensible load can be decoupled correspondingly.

In addition to this, an air side heat recovery wheel can be mounted on the AHU to recycle otherwise wasted heat from the exhaust air and transfer it to the exterior zone for energy efficiency in winter time. If the latent heat of moisture in the exhaust air is also transferred, it is referred to as an energy recovery wheel. The flexibility of the modified AHU is highly enhanced for better air conditioning control.

For a hypothetical multistory office building, if the total heat from the interior zone can be transferred by using the aforementioned imaginary system, it was illustrated that, no outside heat source or supplemental heat is needed during the occupied periods when the OA temperature is at or above 23°F (ASHRAE Handbook, ch8, 2008). Fig 2-4 illustrates the simplified trend of the available heat. The heat might also be recovered for storage if both zones are in need of cooling. The actual profile varies from building to building.

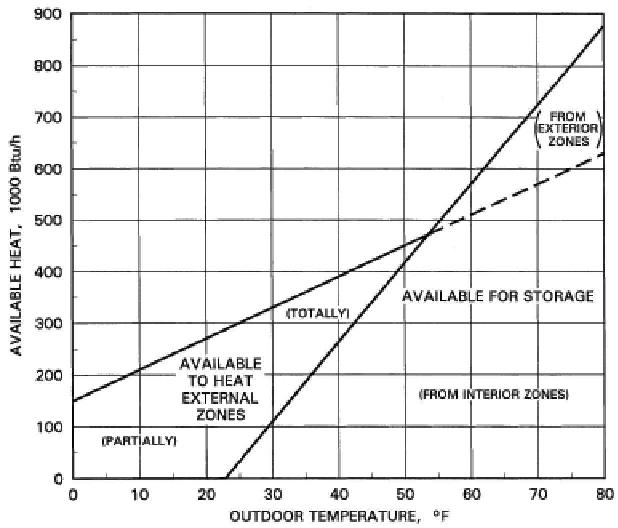


Figure 2-4: Hypothetical building self heating ability

Unlike HRC and WLHP, the imaginary system with two upgrades inherits all the traits of air based systems, and possesses high flexibility and heat recovery capability. However, there are also disadvantages that keep them from wide application in office buildings. The initial cost or cost of system upgrading might be prohibitive. The operation could also be complicated beyond operators' capability to truly achieve the desired energy savings.

Meanwhile, the operating energy cost for the system could be higher than a normal TAHU system since the flexibility is obtained by increasing the system configuration complexity. The economizer operation, additional pressure drop, heat exchanger maintenance and heat recovery efficiency degradation all count in the overall evaluation of this system's performance.

Besides thermal conditioning, the building pressure and IAQ control is also of high importance and can be influenced by the operation of HVAC systems. The OA intake through AHUs is utilized to fulfill both purposes.

For an SAHU, the whole building is treated as one zone. The OA intake from the AHU is provided into the space to dilute the building air contaminants. The indoor air is considered to have a uniform freshness and contaminant concentration. Unless the system is running in an economizer mode, the OA intake is mainly intended to achieve an acceptable IAQ. The flow rate difference between the OA and the relief and exhaust air pressurizes the entire building and prevents unwanted moisture infiltration. In an SAHU system, since there is only one OA inlet, the airflow balance is relatively easy and simple.

With the separation of interior zone and exterior zone, two or more AHUs might be deployed in an office building. The OA intake from an individual AHU is utilized to dilute the air contaminants in the dedicated zone. The building pressure is controlled by maintaining the difference of the total OA intake and the overall air relieved and exhausted from all the AHUs. Since OA intake varies in different modes, building pressure control could be a tough job which requires extra care. The ins and outs need to be accurately measured and balanced to ensure a desired airflow rate difference for proper building pressure.

A conventional operation rarely coordinates the different AHUs in the building. Therefore building pressure control can hardly be satisfied, especially when VAV technology is applied and the airflow rate varies along with the changing zone load. The OA intake of AHUs maintained by coordinating dampers could be far away from the desired value.

2.2 IAHU as a system approach

As reviewed in the previous sections, a TAHU system has its advantages over HRC and WLHP as a popular air-based HVAC system for office buildings. Separating the interior zone and exterior zone reduces the possible compromise between heating/cooling due to the otherwise adverse interaction of thermal load between the two zones. To further improve the system performance, dual path cooling coils for circulation air and OA are needed to decouple the latent cooling and sensible cooling. Heat recovery wheels could be mounted to squeeze the energy savings, provided that the interior zone heat gain are to be recovered for the exterior zone heating load.

OAHU is a new HVAC system proposed by Dr. Song that consists of two conventional AHUs, illustrated in Fig 2-5.

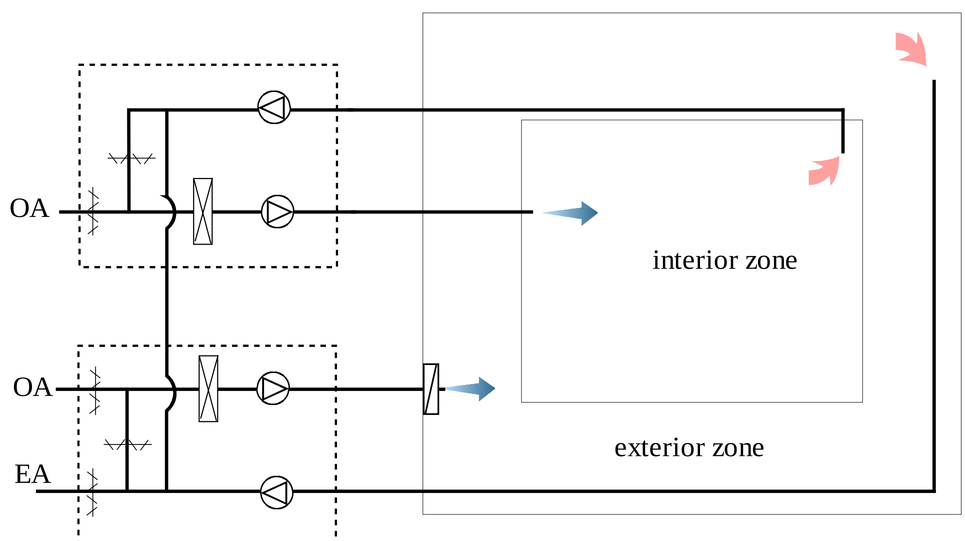


Figure 2-5: Schematic of an OAHU system

The purpose was to accomplish optimal energy performance through integrating all above features into one air-based system:

1. Separate the interior zone and exterior zone supply air;

2. Decouple the latent cooling and the sensible cooling;
3. Recover internal heat gain from the interior zone for the exterior zone heating.

To fulfill all of the three aforementioned features, a piece of duct work connection between the two AHUs is needed in an OAHU system; however, in some buildings, it is not easy to make such a duct connection, especially when the mechanical rooms are apart in the building. The link also brings in more complicated control since two systems are physically tied up with additional inlets and outlets. Meanwhile, the zone air temperature set point was not discussed in the OAHU operation.

2.2.1 IAHU description

IAHU, which surpasses OAHU in several aspects, is developed in this study. Firstly, there is no retrofit or physical duct work required, while all of the benefits of OAHU remain. The control algorithm is tailored so that the conversion is practical. In addition, differentiating the zone air temperature set points in IAHU is proposed as part of its future features for transferring heat gains.

Compared to OAHU, IAHU has four advantages:

1. Without the need of remodeling the duct work, the system configuration of IAHU is much simpler with almost zero retrofitting cost. This aids in customer acceptance of the IAHU upgrading concept.
2. The implementation of IAHU is more feasible with the fan airflow station (FAS) technique developed by the Energy Systems Lab at the University of Nebraska-Lincoln (Liu et al, 2005, Wang, et al, 2007). In OAHU, it is difficult to determine the amount of re-circulated air from the interior zone to the exterior zone since two fans have more than two inlets/outlets. Additional

airflow rate measurements are required. With IAHU, the airflow rate information can be deduced using FAS since the fans have no more than two inlets/outlets.

3. The control algorithm is much simplified in IAHU with less system variables, so that it is more practical in real project applications.
4. In some cases, the heat transfer capacity in IAHU is higher with differentiated zone air temperature set points. The achieved energy savings in winter is not limited to the heat transfer for warming up the OA intake.

In humid weather, a dual path is emulated in an IAHU system to decouple the sensible and latent cooling. OA is introduced in one AHU and the cooling coil is cooled to a temperature lower than both the OA and room air (RA) dew point so that the moisture can be effectively removed to control the indoor humidity. More latent cooling capacity can be provided by the conditioned dry air.

The building sensible load is covered by controlling the discharge air temperature of the circulating air in the other AHU. Depending on the actual condition of the existing HVAC system, the AHU for either the interior zone or the exterior zone can be chosen as latent cooling AHU. Under most circumstances, the interior zone AHU will be selected since the interior zone is always cooling dominant and the load is relatively constant. A low supply air temperature is desired so that the fan energy consumption can be low with less air flow.

In winter and swing season, it is likely that the exterior zone requires heating due to the heat loss through the building envelope. At the same time, cooling is needed in the interior zone because of the internal heat gain from lightings, equipment and occupants' activities.

The OA is warmed up by passing through the interior zone and then pushed into the exterior zone for ventilation. In this scenario, the OA intake required to remove the internal heat load in the interior zone could be more than the OA required for ventilation. The credit of the interior zone ventilation air is used for the exterior zone. By doing this, smaller external heating source is needed by the exterior zone and a significant amount of heat can be saved. Fig 2-6 provides the illustration of the IAHU concept. When the zone temperature set point is properly set, the internal heat gain can also be partially transferred to benefit the exterior zone.

In other conditions, if the constraints listed in the following deduction for IAHU could not be satisfied, an IAHU system is operated as a conventional TAHU system.

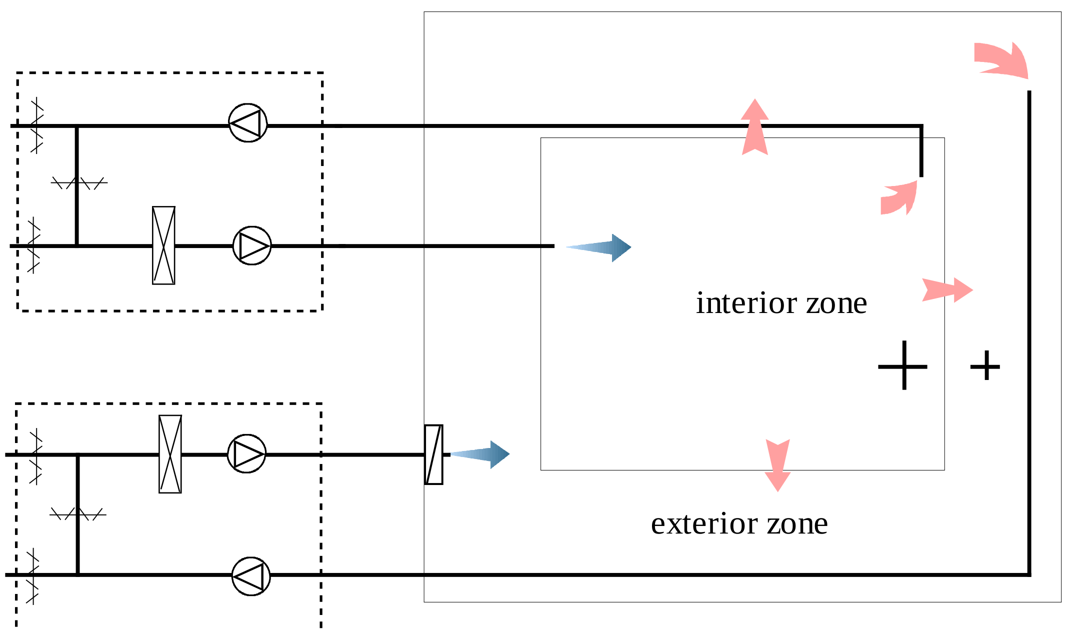


Figure 2-6: Schematic of IAHU

In IAHU mode, the OA required by the entire building is mainly introduced through one AHU only. The zone ventilation is ensured by either getting direct OA intake via its AHU or receiving high quality circulation air from another zone.

At the same time, three conditions regarding the redistributed air should be satisfied to ensure the prior performance of an IAHU system:

1. The IAQ in each zone should be maintained with adequate air ventilation;
2. The amount of total OA intake and released RA should be balanced;
3. The total energy consumption should be optimized.

A theoretical analysis is given in the following to obtain the constraints and expression for the optimized IAHU operation. To facilitate the description and deduction, Fig 2-7 provides the two zone model for IAHU with notations included. AHUs are solidified into two AHUs: one for the interior zone and the other for the exterior zone.

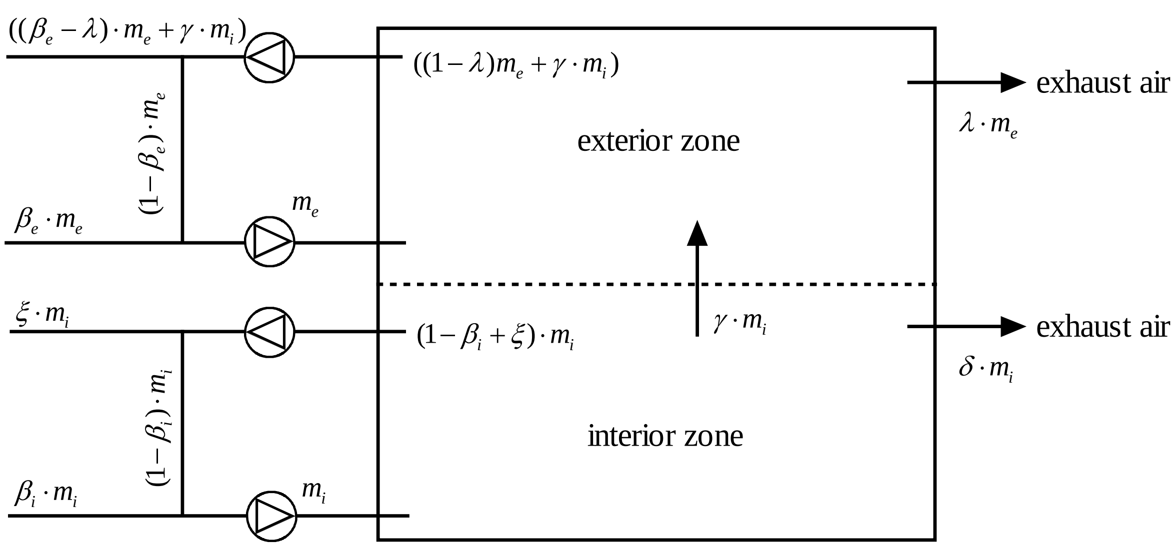


Figure 2-7: Denoted IAHU for two zones simplification

The interior zone supply air flow rate is m_i and the one of the exterior zone is m_e. The Greek symbols in the chart are normalized air flow rate ratios. The OA air intake ratio β for a zone is defined by the ratio of the OA flow rate of that zone to the associated AHU supply air flow rate. The relived air ratio is denoted as ξ and the circulation air between the two zones is γ. The exhausted air ratios for the interior zone and the exterior zone are given as δ and λ, respectively.

2.2.2 IAQ considerations for IAHU

The total airflow rate from the two AHUs is defined as m and the emission rate of indoor air contaminants as C_g. For a control volume, the net in and out mass difference, and the mass generated within the volume is equal to the change of contaminants within the control volume.

Taking the interior zone (dashed in Fig 2-7) as an example, the contaminants' mass coming from OA is:

$$Q_{in} = \beta_i m_i C_o \qquad (2\text{-}1)$$

The mass leaving the zone is:

$$Q_{out} = -\beta_i m_i C_i \qquad (2\text{-}2)$$

The source of the contaminants in the interior zone is $C_{g,i}$. The change rate of the contaminants within the zone is:

$$Q_{cn} = \frac{\partial C_i}{\partial t} \qquad (2\text{-}3)$$

Under the mass conservation law, we can calculate the balance equation as:

$$\frac{-\beta_i m_i C_i + \beta_i m_i C_o + C_{g,i}}{V_i} = \frac{\partial C_i}{\partial t} \tag{2-4}$$

where V_i is the control volume of the interior zone.

A similar mass balance deduction holds for the exterior zone:

$$\frac{-\beta_e m_e C_e + \beta_e m_e C_o + C_{g,e}}{V_e} = \frac{\partial C_e}{\partial t} \tag{2-5}$$

In the previous equations, the interior zone and the exterior zone are treated as one which takes in OA from and releases RA into, the outside individually. There is no interaction of air or contaminants between the two zones.

Suppose the OA intake is only used to remove the contaminants and maintain the IAQ requirement, the following OA ratio for the corresponding zone should be ensured:

$$\beta_{i,IAQ} = \frac{\frac{\partial C_i}{\partial t} V_i + C_{g,i}}{m_i (C_{i,IAQ} - C_o)} \tag{2-6}$$

$$\beta_{e,IAQ,d} = \frac{\frac{\partial C_e}{\partial t} V_e + C_{g,e}}{m_e (C_{e,IAQ,d} - C_o)} \tag{2-7}$$

In the analysis below, the interior zone is chosen as the primary OA intake zone for an integrated operation. To clarify the scenarios, we adopt *i,IAQ* as the subscript to denote the directly relevant variables when the interior zone is operated on its own. In addition, *e,IAQ,d* is used to describe a similar circumstance for the exterior zone when an individual OA intake is considered. The difference holds for the rest of this section.

In a steady state, there is no concentration change for either zone. This means that $\frac{\partial C_i}{\partial t} = 0$ and $\frac{\partial C_e}{\partial t} = 0$. Therefore, the individual OA intake to the interior zone and exterior zone are simplified respectively as:

$$\beta_{i,IAQ} = \frac{C_{g,i}}{m_i(C_{i,IAQ} - C_o)} \tag{2-8}$$

$$\beta_{e,IAQ,d} = \frac{C_{g,e}}{m_e(C_{e,IAQ,d} - C_o)} \tag{2-9}$$

For any arbitrary OA intake ratio, the steady state mass balance should be true in order to maintain an acceptable contaminant concentration. For the interior zone, this means:

$$\beta_i m_i C_o + C_{g,i} - \beta_i m_i C_i = 0 \tag{2-10}$$

The mass balance also holds for the total amount of transferred, relieved and exhausted air from the interior zone, so that the zone pressure can be balanced:

$$\beta_i = \gamma + \delta + \xi \tag{2-11}$$

Submit equation (2-11) into (2-10):

$$\beta_i m_i C_o + C_{g,i} - (\gamma + \delta + \xi) m_i C_i = 0 \tag{2-12}$$

Now, to fulfill an IAHU operation for the benefits, one zone will be designated as the primary OA intake zone. Here we take the interior zone with $C_i \leq C_e$. The still-fresh RA transferred from the interior zone to the interior zone is defined as γm_i.

The supplementary OA intake from the exterior zone AHU itself is $\beta_{e,IAQ} m_e$. The OA is to ensure that the IAQ requirement is met if additional direct OA is needed.

Based on the mass balance of contaminants, the following equation can be established for the exterior zone:

$$\beta_{e,IAQ} m_e C_o + C_{g,e} + \gamma m_i C_i - (\beta_{e,IAQ} m_e + \gamma m_i) C_e = 0 \qquad (2\text{-}13)$$

To further simplify the analysis, we normalize several variables. The total conditioned airflow rate supplied into the building is:

$$m = m_i + m_e \qquad (2\text{-}14)$$

The interior zone total airflow rate ratio is:

$$\varphi = \frac{m_i}{m} \qquad (2\text{-}15)$$

For CAV applications, φ is considered as a constant. However, in many real office buildings, VAV is widely used. In this study, this ratio is generalized as a changing variable.

Substituting (2-15) into (2-13), we have:

$$\beta_{e,IAQ} m_e C_o - \beta_{e,IAQ} m_e C_e + \gamma \varphi m C_i - \gamma \varphi m C_e + C_{g,e} = 0 \qquad (2\text{-}16)$$

From equation (2-10), it is known that:

$$C_i = C_o + \frac{C_{g,i}}{\beta_i m_i} \qquad (2\text{-}17)$$

Replacing C_i in (2-16) with (2-17), we obtain:

$$\beta_{e,IAQ} m_e C_e - \beta_{e,IAQ} m_e C_o + \gamma \varphi m C_e - \gamma \varphi m C_o - \gamma m_i \frac{C_{g,i}}{\beta_i m_i} = C_{g,e} \qquad (2\text{-}18)$$

Rearranging the equation to find the expression for $\beta_{e,IAQ}$:

$$\beta_{e,IAQ} = \frac{1}{m_e(C_e - C_o)}\left[C_{g,e} + \gamma\varphi m(C_o - C_e) + \gamma\frac{C_{g,i}}{\beta_i}\right] \tag{2-19}$$

Equations (2-8) and (2-9) can be further rearranged as:

$$C_{g,i} = (C_i - C_o)\beta_i m_i = (C_{i,IAQ} - C_o)\beta_{i,IAQ} m_i \tag{2-20}$$

$$C_{g,e} = \left(C_{e,IAQ,d} - C_o\right)\beta_{e,IAQ,d} m_e \tag{2-21}$$

Substituting them into equation (2-19), and solving for the exterior zone direct OA intake:

$$\beta_{e,IAQ} = \frac{\left(C_{e,IAQ,d} - C_o\right)\beta_{e,IAQ,d} m_e + \frac{\gamma}{\beta_i}(C_{i,IAQ} - C_o)\beta_{i,IAQ} m_i}{(C_e - C_o)(1-\varphi)m} - \frac{\gamma\varphi}{1-\varphi} \tag{2-22}$$

The supplementary OA intake $\beta_{e,IAQ}$ from the exterior zone AHU is used to maintain the same level of IAQ as that in the interior zone. In the previous equation, the exterior zone contaminant concentration C_e should be a value that can ensure an acceptable IAQ:

$$C_e = C_{e,IAQ,d} = C_{i,IAQ} \tag{2-23}$$

With it, equation (2-22) is further simplified:

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} - \frac{\gamma\varphi}{1-\varphi} + \frac{\varphi}{(1-\varphi)}\frac{\beta_{i,IAQ}}{\beta_i}\gamma \tag{2-24}$$

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} - \frac{\gamma\varphi}{1-\varphi}(1 - \frac{\beta_{i,IAQ}}{\beta_i}) \tag{2-25}$$

To this point, the general expression is acquired, which governs the additional OA intake from the exterior zone AHU in IAHU to maintain the IAQ in the exterior zone.

In the following, we will deduce the threshold of the direct OA intake $\beta_e(\beta_{e,IAQ,d}, \beta_{i,IAQ}, \varphi, \beta_i, \gamma)$ from the exterior zone AHU which is needed to maintain the exterior zone IAQ.

Among the five independent variables, the first two are determined by equations (2-8) and (2-9). φ is a variable determined mainly by the space load and supply air temperature. The contaminants generated within zones are assumed to be invariant; therefore the minimum OA flow rate is considered as a constant. $\beta_{e,IAQ,d}$ and $\beta_{i,IAQ}$ could be constant in CAV and variables in VAV.

In a CAV/VAV system operation, the first three variables can be calculated out as knowns, if the airflow rates are known or properly measured. The most free variables are β_i and, γ which is $(\beta_i - \delta - \xi)$.

The ratio of transferred air γ could be modulated, with the interior zone OA intake β_i adjusted, in an IAHU operation to crop the desired benefits. Next, the general relationship between γ and β_i will be briefly deduced for several different conditions.

1: $\gamma = 0$, no transfer air between the two zones

This is similar to the normal operation for conventional office AHUs. There is no interaction between the two AHUs.

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} \tag{2-26}$$

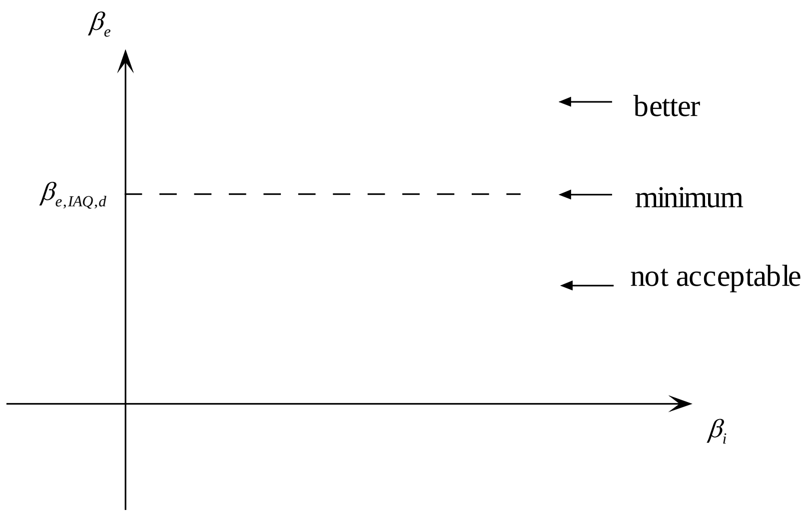


Figure 2-8: OA intake for normal operation

2: $\gamma = \beta_i - \delta$, no direct exhaust from the interior zone

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} - \frac{\varphi}{1-\varphi}\left(1 - \frac{\beta_{i,IAQ}}{\beta_i}\right)(\beta_i - \delta) \tag{2-27}$$

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} - \frac{\varphi}{1-\varphi}\left(\beta_i - \beta_{i,IAQ}\right) + \frac{\delta\varphi}{1-\varphi}(1 - \frac{\beta_{i,IAQ}}{\beta_i}) \tag{2-28}$$

To lower down the direct OA intake from the exterior zone AHU, the absolute value of the second item in equation (2-28) is expected to larger and that of the third smaller.

The partial derivative of equation (2-28) on β_i is:

$$\frac{\partial \beta_{e,IAQ}}{\partial \beta_i} = -\frac{\varphi}{1-\varphi} + \frac{\delta\varphi}{1-\varphi}\frac{\beta_{i,IAQ}}{{\beta_i}^2} = \frac{\varphi}{1-\varphi}(\delta\frac{\beta_{i,IAQ}}{{\beta_i}^2} - 1) \tag{2-29}$$

It can be seen that the result is negative, as is the second derivative. β_i and $\beta_{e,IAQ}$ have an inverse correlation as a concave curve. This means when β_i increases, $\beta_{e,IAQ}$ monotonically decreases. This can be illustrated as:

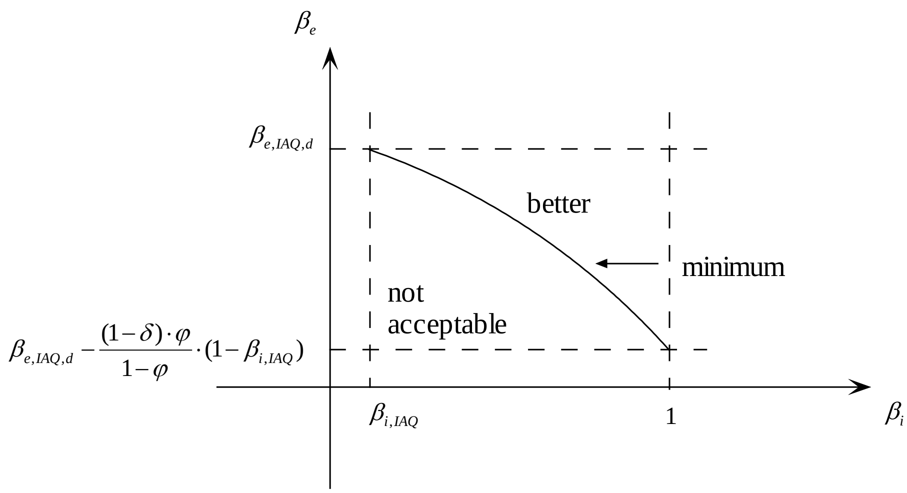


Figure 2-9: OA intake with no direct exhaust from the interior zone

3: $\gamma = \beta_i$ ($\delta = 0$, $\xi = 0$), no air relived or exhausted from the interior zone

Under this circumstance, equation (2-25) becomes:

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} + \frac{\varphi}{1-\varphi}(\beta_{i,IAQ} - \beta_i) \tag{2-30}$$

Meanwhile, it follows that $\frac{\partial \beta_{e,IAQ}}{\partial \beta_i} = -\frac{\varphi}{1-\varphi}$. This is a monotonically decreasing function with a constant slope. The drawing can be illustrated as below:

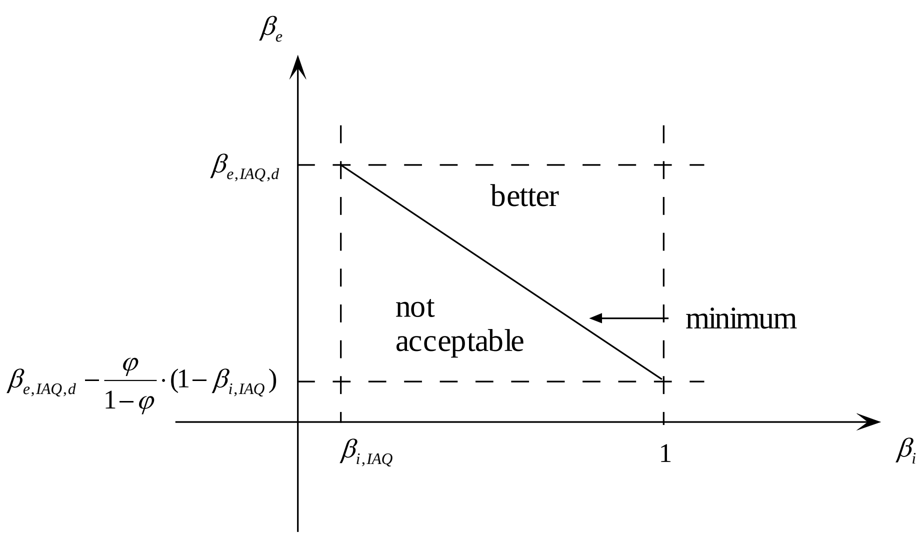


Figure 2-10: OA intake with no direct exhaust or relief from interior zone

4: $\gamma m_i + \beta_e m_e = m_e$, IAHU is converted into OAHU

Submitting the condition into equation (2-25), the following can be obtained:

$$\beta_{e,IAQ} = \beta_{e,IAQ,d} - \frac{\varphi}{1-\varphi}\left(1 - \frac{\beta_{i,IAQ}}{\beta_i}\right)\left(\frac{(1-\beta_{e,IAQ})(1-\varphi)}{\varphi}\right) \tag{2-31}$$

Further rearrange the above equation, we obtain:

$$\beta_{e,IAQ} = 1 - \beta_i\left(\frac{1-\beta_{e,IAQ,d}}{\beta_{i,IAQ}}\right) \tag{2-32}$$

The relationship between β_i and $\beta_{e,IAQ}$ is a line with a smaller slope compared to the previous one, which means, with the same β_i, $\beta_{e,IAQ}$ of the exterior zone is higher.

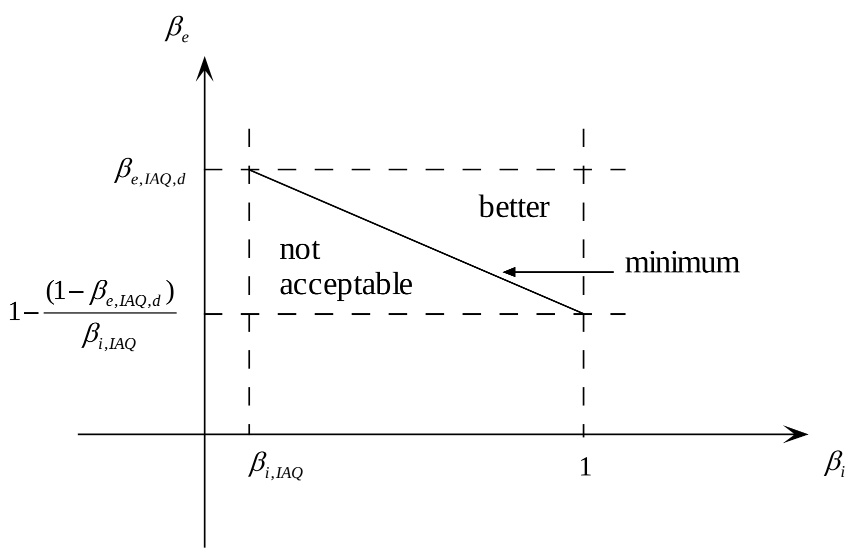


Figure 2-11: OA intake for OAHU

By combining the analysis and comparing the four charts (Fig 2-8 to Fig 2-11), it is easy to see that, with the same interior zone OA intake ratio, β_i, the required OA intake for the exterior zone $\beta_{e,IAQ}$ decreases with an increasing air recirculation from the interior zone γ. This point is important in determining the control algorithm for an IAHU system.

It is possible that, in a real application where even the interior zone OA intake β_i is 100% without any direct relief, the exterior zone still requires individual OA intake through its AHU when the interior zone airflow rate ratio φ is too low.

Under this extreme circumstance, equation (2-28) will be the following with $\beta_i = 1$ and $\beta_{e,IAQ} = 0$ as the boundaries:

$$\beta_{e,IAQ,d} - \frac{\varphi}{1-\varphi}\left(1-\beta_{i,IAQ}\right) + \frac{\delta\varphi}{1-\varphi}\left(1-\beta_{i,IAQ}\right) = 0 \qquad (2\text{-}33)$$

Reorganize the equation,

$$\beta_{e,IAQ,d} - \frac{(1-\delta)\varphi}{1-\varphi}\left(1-\beta_{i,IAQ}\right) = 0 \qquad (2\text{-}34)$$

Solving for φ, and we find:

$$\varphi_{cr} = \frac{\beta_{e,IAQ,d}}{\beta_{e,IAQ,d} + (1-\delta)(1-\beta_{i,IAQ})} \qquad (2\text{-}35)$$

2.2.3 Energy considerations for IAHU

The objective of IAHU is to lower the thermal energy consumption and ensure an acceptable or even improved IAQ by optimizing the building OA intake and allocation.

The cost function of thermal energy consumption for cooling and heating the building with two AHUs can be expressed as:

$$E_{thm} = E_{i,hc} + E_{i,cc} + E_{i,rh} + E_{e,hc} + E_{e,cc} + E_{e,rh} \qquad (2\text{-}36)$$

where, the individual components for both zones are:

$$E_{hc} = \max\left(0, m(h_c - h_{mix})\right) \qquad (2\text{-}37)$$

$$E_{cc} = \max(0, m(h_{mix} - h_c)) \tag{2-38}$$

$$E_{rh} = \max(0, m(h_s - h_c)) \tag{2-39}$$

To facilitate the interpretation, Fig 2-12 gives the system configuration with thermal components included:

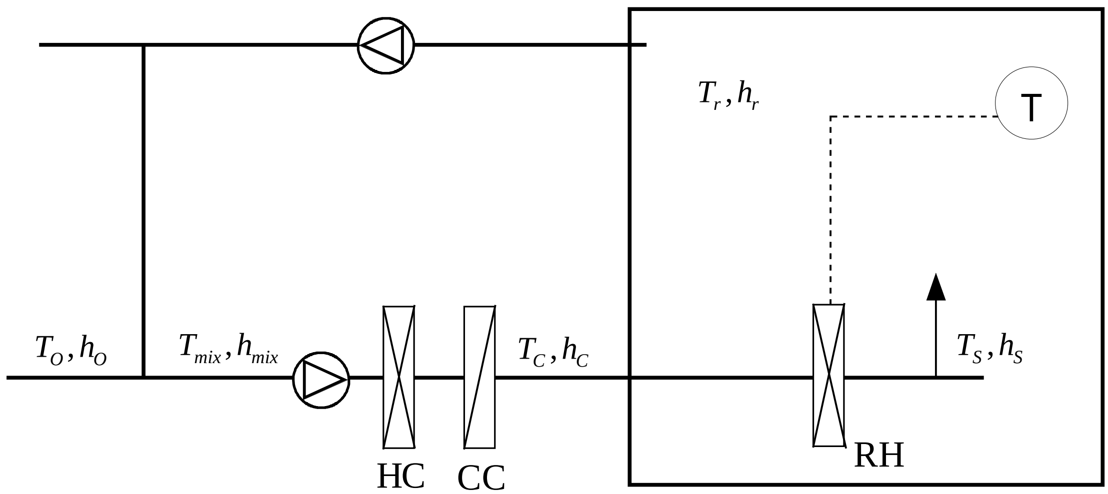


Figure 2-12: Thermal components of a typical AHU

For a real building operation, this is a varying constrained optimization problem.

First of all, the OA ratio should always be no less than the minimum in non-economizer seasons as deduced in the previous section.

$$\beta_i \in [\beta_{i,IAQ}, 1] \tag{2-40}$$

$$\beta_e \in [\max(\beta_{e,IAQ}, 0), 1] \tag{2-41}$$

In swing seasons, if a zone is operating in cooling mode, it is likely that more than IAQ required OA should be introduced into the space. The economizer OA intake can be simply deduced from:

$$(1 - \beta_{eco})T_r + \beta_{eco}T_{oa} = T_c \tag{2-42}$$

That is:

$$\beta_{eco} = \frac{T_r - T_c}{T_r - T_{oa}} \tag{2-43}$$

The economizer OA intake constraints for the zones are:

$$\beta_i \in [\beta_{i,eco}, 1] \tag{2-44}$$

$$\beta_e \in [\beta_{e,eco}, 1] \tag{2-45}$$

The economizer constraints can only be applied when the corresponding supply air temperature satisfies the system's minimum airflow rate without trigging on reheating. For a large system, due to the zones' diversity, an economizer is an idealized cooling dominant operation. For example, in a perimeter zone CAV, it is obvious that there is no clear cut β_{eco} unless the supply air temperature is continuously adjusted according to the real load.

In addition, there is another constraint on the supply air temperature. In order to maintain the RA humidity, the supply air humidity ratio in cooling mode should not be higher than that of the RA. In other words, the air temperature should not be higher than the RA design dew point in cooling mode.

$$T_c < T_{r,dew} \tag{2-46}$$

The system thermal energy control strategy for an optimal consumption is highly related to the OA condition and OA intake.

The following different scenarios are analyzed to obtain the operation strategy for IAHU:

A: $T_{oa} \leq T_{i,c}$

B: $T_{i,c} \leq T_{oa} \leq T_{e,c}$

C: $T_{e.c} \leq T_{oa} \leq T_{r}$, dry   Winter or swing season

D: $T_{e,c} \leq T_{oa} \leq T_{r}$, humid

Summer

E: $T_{r} \leq T_{o,a}$

A brief deduction for each scenario is provided below.

The cost/objective functions, which are always non-negative, are defined in the sub domain of $0 \leq \gamma \leq \beta_i \leq 1$ on (β_i, γ) plane.

A: $T_{oa} \leq T_{i,c}$

It is winter and OA is colder than the interior zone cold deck air temperature. Free cooling is available. The possible components of thermal energy consumption include a heating and reheat. The cost function can be rewritten as:

$$E_{thm} = E_{i,hc} + E_{i,rh} + E_{e,hc} + E_{e,rh} \tag{2-47}$$

Referring to the above system drawing and notes, the heating energy for the interior zone is:

$$E_{i,hc} = m_i C_p (T_{i,c} - T_{i,mix}) \tag{2-48}$$

The reheat, if there is any, is:

$$E_{i,rh} = m_i C_p \left(T_s - T_{i,c}\right) \tag{2-49}$$

The mixed air temperature is a function of OA intake and, OA and RA temperature:

$$T_{mix} = T_{oa}\beta + (1 - \beta)T_r \tag{2-50}$$

Combining equation (2-50), (2-49) and (2-48), we obtain:

$$E_{i,rh} = m_i C_p \left(T_{i,c} - T_r\right) + m_i C_p \beta_i (T_r - T_{oa}) + m_i C_p (T_s - T_r) + m_i C_p \left(T_r - T_{i,c}\right) \tag{2-51}$$

It is known that the third term in equation (2-51) is the zone load, and the first term cancels out the last term. Therefore, the final expression of reheat is:

$$E_{i,rh} = m_i C_p \beta_i (T_r - T_{oa}) - Load_i \tag{2-52}$$

where $Load_i$ is an absolute value.

The same deduction holds for the perimeter zone but the load is a heating load. Thus, the final thermal energy consumption of the entire system is:

$$E_{thm} = m_i C_p \beta_i (T_r - T_{oa}) - Load_i + m_e C_p \beta_e (T_r - T_{oa}) + Load_e \tag{2-53}$$

To minimize the thermal consumption, we want the two terms with the OA ratio β to be minimized. Since OA is the cooling source for the interior zone, β_i is constrained and defined by the maximum of (2-40) and (2-44). Therefore, the OA intake for the exterior zone is a dependent variable.

The total amount of OA intake for the building is given as equation (2-54):

$$f = \beta_{e,IAQ,d} \cdot (1 - \varphi) - \gamma\varphi \left(1 - \frac{\beta_{i,IAQ}}{\beta_i}\right) + \varphi \cdot \beta_i \tag{2-54}$$

It is easy to find that this function does not have extremum on the definition domain, since the discriminant is always zero for all points (B.Demidovich, 1989):

$$\Delta = AC - B^2 \tag{2-55}$$

where $A = f''_{\gamma\gamma}$, $B = f''_{\gamma\beta_i}$ and $B = f''_{\beta_i\beta_i}$.

Therefore, the extremum exists only at the boundary. Meanwhile, the partial derivative on β_i has a form as:

$$\frac{\partial f}{\partial \beta_i} = \varphi\left(1 - \frac{\gamma\varphi\beta_{i,IAQ}}{{\beta_i}^2}\right) \tag{2-56}$$

Since $\beta_i \geq \beta_{i,IAQ}$ and $\gamma \leq \beta_i$, it is constantly positive and the extremum of the function is the lower boundary. The partial derivative on γ is negative and has a form as:

$$\frac{\partial f}{\partial \gamma} = -\varphi\left(1 - \frac{\beta_{i,IAQ}}{\beta_i}\right) \tag{2-57}$$

The minimum is the point where β_i takes the low boundary and γ takes the upper boundary.

$$\beta_i = \max\left(\beta_{i,IAQ}, \beta_{i,eco}\right) \tag{2-58}$$

$$\beta_e = \max\left(\beta_{e,IAQ}, 0\right) \tag{2-59}$$

<u>B:</u> $T_{i,c} \leq T_{oa} \leq T_{e,c}$

In this scenario, the interior zone is in a cooling mode while the exterior zone in either a cooling or a heating mode. The cost function is different:

$$E_{thm} = E_{i,cc} + E_{i,rh} + E_{e,hc} + E_{e,rh} \tag{2-60}$$

$$E_{i,cc} + E_{i,rh} = m_i C_p\left(T_{i,mix} - T_{i,c}\right) + m_i C_p\left(T_{i,s} - T_{i,c}\right) \tag{2-61}$$

$$E_{e,hc} + E_{e,rh} = m_e C_p\left(T_{e,c} - T_{e,mix}\right) + m_e C_p\left(T_{e,s} - T_{e,c}\right) \tag{2-62}$$

T_{mix} is given by equation (2-50). After the substitution and rearrangement, the cost function can be put as:

$$\begin{aligned} E_{thm} = (T_r - T_{oa})\left(m_e C_p \beta_e - m_i C_p \beta_i\right) + 2 m_i C_p\left(T_r - T_{i,c}\right) + Load_e \\ - Load_i \end{aligned} \tag{2-63}$$

To minimize the thermal energy consumption, we should decrease the value of the first term, which means $\beta_i \uparrow, and\ \beta_e \downarrow$. This is beneficial to the entire system, since the corresponding OA ratios for IAQ also have an inverse correlation in IAHU. Thus, we have the equivalent objective function:

$$f = \beta_{e,IAQ,d} \cdot (1-\varphi) - \gamma\varphi\left(1 - \frac{\beta_{i,IAQ}}{\beta_i}\right) - \varphi \cdot \beta_i \tag{2-64}$$

Similar to condition A, the extremum lies at the boundary instead of on the inside. The partial derivatives on both variables are negative. Therefore the upper boundary value on both β_i and γ minimizes the energy consumption.

$$\beta_i = 1 \tag{2-65}$$

$$\beta_e = \max\left(\beta_{e,IAQ}, 0, \left(\beta_{e,eco}, if\ applicable\right)\right) \tag{2-66}$$

C & D: $T_{e,c} \leq T_{oa} \leq T_r$

In this condition, both zones are in a cooling mode. It is possible that the OA enthalpy is higher than that of room air. A dehumidification might be involved in the air conditioning process. The thermal energy consumption is defined in terms of air enthalpy.

The interior zone thermal energy consumption is:

$$E_{i,thm} = m_i\left(h_{i,mix} - h_{i,c}\right) + m_i\left(h_{i,s} - h_{i,c}\right) \tag{2-67}$$

The exterior zone thermal energy consumption is given by a similar expression:

$$E_{e,thm} = m_e\left(h_{e,mix} - h_{e,c}\right) + m_e\left(h_{e,s} - h_{e,c}\right) \tag{2-68}$$

The mixed air enthalpy for both AHUs is:

$$h_{mix} = \beta h_{oa} + (1-\beta)h_r \tag{2-69}$$

Taking the interior zone for a deduction, we have:

$$E_{i,thm} = m_i\left(\beta h_{oa} + (1-\beta)h_r - h_{i,c}\right) + m_i\left(h_{i,s} - h_{i,c}\right) \tag{2-70}$$

Since $\left(h_{i,s} - h_{i,c}\right) = \left(h_{i,s} - h_{i,r}\right) + \left(h_{i,r} - h_{i,c}\right)$, replacing the last term in the previous equation:

$$\begin{aligned} E_{i,thm} &= m_i\left(\beta h_{oa} + (1-\beta)h_r - h_{i,c}\right) + m_i\left(h_{i,s} - h_r\right) + m_i\left(h_r - h_{i,c}\right) \\ &= 2m_i\left(h_r - h_{i,c}\right) + m_i\beta_i(h_{oa} - h_r) - Load_i \end{aligned} \tag{2-71}$$

A similar expression can be obtained for the exterior zone thermal energy consumption. The total building thermal energy consumption under this condition is:

$$\begin{aligned} E_{thm} &= 2m_i\left(h_r - h_{i,c}\right) + 2m_e\left(h_r - h_{e,c}\right) - Load_i - Load_e \\ &\quad + (h_{oa} - h_r)(m_i\beta_i + m_e\beta_e) \end{aligned} \tag{2-72}$$

For ***C***: dry air condition, we have $h_{o,a} \leq h_r$, therefore, the last term is negative. This means the higher the OA intake, the less the energy consumption. The equivalent objective function is:

$$f = -\beta_{e,IAQ,d} \cdot (1-\varphi) + \gamma\varphi\left(1 - \frac{\beta_{i,IAQ}}{\beta_i}\right) - \varphi \cdot \beta_i \tag{2-73}$$

This is an opposite function to that of condition A. The upper boundary of β_i and lower boundary γ gives the minimum.

$$\beta_i = \beta_e = 1 \tag{2-74}$$

For ***D***: humid air condition, we have $h_{o,a} \geq h_r$.

With conventional cooling coil dehumidification, to dehumidify mixed air and remove air moisture with a chilled water coil, the mixed air should be processed to the corresponding dew point of the calculated supply air point before being distributed into the space. A complication could occur in humid mild weather when the sensible load and latent load do not balance as illustrated in the psychrometric chart below. The conditioned air needs to be reheated from point *L* to *O* in order to attain an acceptable RA status *N* in the diamond, or the space will be overcooled to *N'*. Under this circumstance, a decoupled sensible load and latent load processing are desired. The analysis for an optimum system operation is conducted briefly.

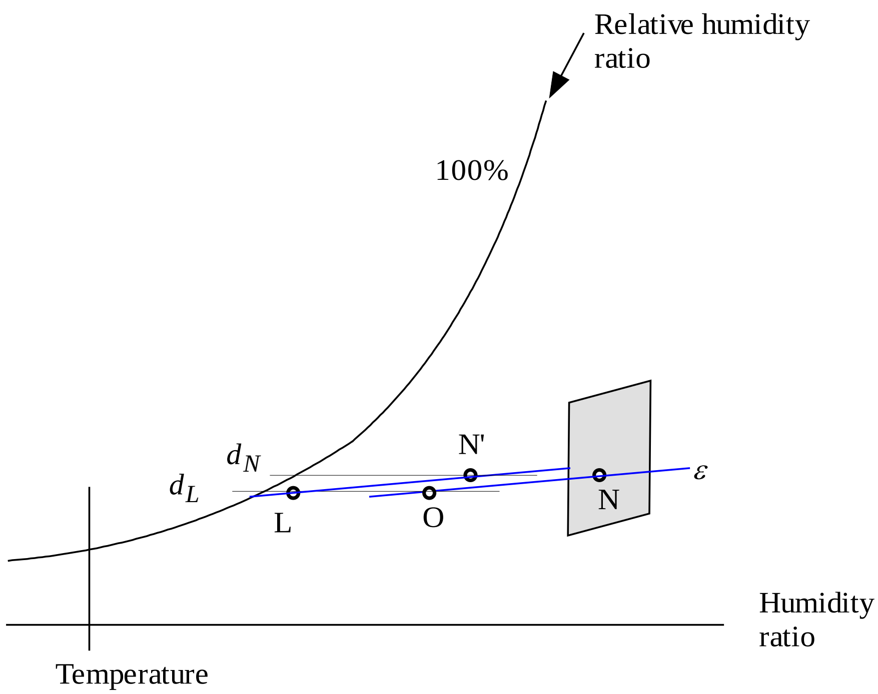


Figure 2-13: Thermal process of partial load condition

Equation (2-72) holds for the humid condition. The controllable parts are recollected as below:

$$f = 2m_e(h_r - h_{e,c}) + (h_{oa} - h_r)(m_i\beta_i + m_e\beta_e) \tag{2-75}$$

During a normal operation in a system, the first term is a fixed value with a given supply airflow rate since $h_{e,c}$ is fixed by the RA dew point for the purpose of dehumidification. The total airflow rate is modulated to meet the building load. It is easy to find that the minimum points exist at the low boundary of the two OA ratios, with $\beta_i = \beta_{i,IAQ}$, and correspondingly $\beta_e = \beta_{e,IAQ,d}$. There is no controlled air transfer between the two zones. IAHU is equivalent to TAHU.

But when a partial load happens in humid mild weather, the following inequity exists:

$$\left(h_{e,s} = h_r - \frac{Load_e}{m_e}\right) \geq h_{e,c} \quad (2\text{-}76)$$

In this condition, the airflow rate is first decreased to the minimum threshold. If the sensible load keeps dropping after this point, a potential reheat is needed in the system which wastes energy to balance the load discrepancy. The additional cost is likely to be eliminated if $h_{e,c}$ can be readjusted to cover the sensible load only. This happens when β_e decreases to zero and there is no latent load added on to the supply air. With this operation, the zone sensible load and latent load are decoupled.

During such a decoupling process in IAHU, the first term in equation (2-75) decreases and the second term increases. The evaluation of the question becomes a comparison of the savings and the cost in the two terms.

The normalized energy saving from the first term is:

$$f_1 = 2\left(h_{e,s} - h_{e,c}\right)(1 - \varphi) \quad (2\text{-}77)$$

And the extra energy cost of the second term, $(h_{oa} - h_r)((m_i\beta_i + m_e\beta_e) - (m_i\beta_{i,IAQ} + m_e\beta_{e,IAQ,d}))$, can be normalized as:

$$f_2 = (h_{oa} - h_r)(\varphi\beta_i - \varphi\beta_{i,IAQ} - (1-\varphi)\beta_{e,IAQ,d}) \tag{2-78}$$

In this operation, $h_{e,s}$ is determined by the real time sensible load in the conditioned exterior zone.

The optimum can be found by comparing f_1 and f_2, but it increases the implementation complexity. A reasonable simplification is needed.

$f_1 \geq f_2$ is generally, if not always, true in humid mild weather, because the ratio term $2(1-\varphi)$ in f_1, is much bigger than the ratio term in f_2, $(\varphi\beta_i - \varphi\beta_{i,IAQ} - (1-\varphi)\beta_{e,IAQ,d})$. In this operation mode, if there is no direct relief and exhaust air from the interior zone for simplification, the optimum points can be found by setting $\beta_{e,IAQ}$ in equation (2-25) to zero.

$$\beta_i = \frac{\gamma\varphi\beta_{i,IAQ}}{\gamma\varphi - \beta_{e,IAQ,d}\cdot(1-\varphi)} \cong \frac{\beta_{e,IAQ,d}\cdot(1-\varphi) + \beta_{i,IAQ}\cdot\varphi}{\varphi} \tag{2-79}$$

$$\beta_e = 0 \tag{2-80}$$

When the required OA intake ratio is more than 30% (adjustable) of the supply airflow rate in the exterior zone, IAHU can be considered unfeasible. When this happens, IAHU becomes TAHU.

E: $T_r \leq T_{oa}$

Since the OA temperature is greater than the RA temperature, both zones are in a cooling mode. In most climates and normal circumstances, the OA enthalpy is higher than that of RA. A dehumidification is needed. The analysis is identical to the one in condition ***D*** for humid weather.

During a normal operation, the minimum points are the boundary for the two systems:

$\beta_i = \beta_{i,IAQ}$ and $\beta_e = \beta_{e,IAQ,d}$.

When the exterior zone sensible load is low in mild weather, a decoupling operation is desired to avoid unnecessary reheat. Equations (2-79) and (2-80), and their analysis are applicable.

To this point, a theoretic deduction of IAHU and its year round operation is presented. Space ventilation and energy savings are considered to ensure the superior performance after the conversion of TAHU. This brings in benefits beyond the normal interior zone and exterior zone TAHU by synergizing the AHUs when proper conditions occur. Excessive heating and reheat in a conventional operation can be reduced largely by shifting the internal heat gain or decoupling latent load from sensible load processing.

The operation of IAHU is collected in the following table:

Table 2-1: IAHU operation scenarios

No.	**T_{oa} Condition**		**Interior OA**	**Exterior OA**
A	$T_{oa} \leq T_{i,c}$		$\beta_i = \max\,(\beta_{i,IAQ}, \beta_{i,eco})$	$\beta_e =$ $\max\,(\beta_{e,IAQ}, 0)$
B	$(T_{i,c}, T_{e,c}]$		$\beta_i = 1$	$\beta_e =$ $\max\,(\beta_{e,IAQ}, 0, \beta^*{}_{e,eco})$
C	$(T_{e,c}, T_r]$, $h_{o,a} < h_r$		$\beta_i = 1$	$\beta_e = 1$
D	$(T_{e,c}, T_r]$, $h_{o,a} \geq h_r$	mild weather	$\beta_i =$ $\left[\dfrac{\gamma\varphi\beta_{i,IAQ}}{\gamma\varphi - \beta_{e,IAQ,d} \cdot (1-\varphi)}\right], \left[\beta_{i,IAQ}\right]$	$\beta_e = [0], \left[\beta_{e,IAQ,d}\right]$
		normal operation	$\beta_{i,IAQ}$	$\beta_e = \beta_{e,IAQ,d}$

E	$T_r < T_{oa}$	mild weather	$\beta_i = \left[\frac{\gamma\varphi\beta_{i,IAQ}}{\gamma\varphi - \beta_{e,IAQ,d} \cdot (1-\varphi)}\right], \left[\beta_{i,IAQ}\right]$	$\beta_e = [0], \left[\beta_{e,IAQ,d}\right]$
		normal operation	$\beta_{i,IAQ}$	$\beta_e = \beta_{e,IAQ,d}$

* if applicable

As shown in the deduction and the table above, knowledge the supply air flow rate, outside air intake, airflow distribution, as well as the switching points is critical to IAHU systems, especially for a VAV based AHU system. In a real building, the information has rarely been utilized other than as an equipment status indicator.

In the following chapter, the system variables and control algorithm, as well as the implementation methodology, are further analyzed in detail. The control algorithm is later adopted in a simulation for a government office building in Omaha to,

1. Investigate the implementation details involved in an IAHU operation;
2. Illustrate the methodologies to attain the airflow rate related parameters;
3. Verify the new conservative office building HVAC system approach.

Chapter 3 IAHU Control Algorithm and Implementation

IAHU is introduced in Chapter 2 to improve conventional TAHU for energy conservation. It evolves from OAHU with considerations on less retrofitting and more feasible control algorithms. No additional physical duct work connection is required in an IAHU system. The conditioned indoor air is manipulated and reallocated within the space between the two individual AHUs. The operation scenarios are also simplified to an applicable level.

To fulfill the benefits of converting a TAHU system into IAHU, there are two issues to be solved before we implement the control:

1. Among the variables, what is the relationship between the independent variables? How are they decided and in what sequence?
2. How do we obtain the variables during a normal operation?

3.1 Variable analysis

There are five critical independent variables in the deduction of an IAHU operation: β_e ($\beta_{e,IAQ,d}$, $\beta_{i,IAQ}$, φ, β_i, γ). When the system is operated in an economizer mode, two more OA ratio variables, $\beta_{i,eco}$ and $\beta_{e,eco}$, should be considered as the constraints. In addition, a feasible OA intake to the interior zone in mild weather should be defined to enable thermal decoupling operations. All of the variables are normalized to be airflow rate ratios for IAHU deduction and theory analysis. In a real application, knowledge the absolute airflow rate values is important because the ratios are calculated based on these quantities.

$\beta_{e,IAQ,d}$ and $\beta_{i,IAQ}$:

$\beta_{e,IAQ,d}$ and $\beta_{i,IAQ}$ are the OA intake ratios in the exterior zone and the interior zone respectively. The two variables are defined for the purpose of removing indoor contaminants and ensure an acceptable IAQ when the AHUs operate separately as a TAHU system. They are generally regarded as design constants with a fixed value, for example 10% or 15%. In real applications, the amount of OA intake can be ensured by installing an airflow rate station in the OA duct; or, more commonly, a fixed minimum OA damper openness is arbitrarily assigned.

In a real building, since the building occupancy varies, the flow rate of required ventilation air is actually also a changing variable, as is the OA ratio in a CAV operation. For a VAV TAHU, although the airflow rate changes along with the occupancy, the OA ratio to maintain an acceptable IAQ can be considered as a constant. The reason for this is that the occupants and their activities in buildings create not only thermal load which influences the total airflow rate but also create demands for ventilation air. Building occupancy was found to have a linear proportional relationship to the difference between the real time energy consumption on the lighting and equipment, and the minimum value of the consumption (Abushakra and Knebel, 2008).

One of IAHU system's advantages over the conventional TAHU system is that the interior zone air can be re-circulated to the exterior zone because of the freshness. In winter, the heat that is required in a TAHU system to warm up the cold OA is saved in an IAHU system. In summer, cooling energy that is needed to remove the moisture in the humid OA can also be saved in an IAHU system by decoupling the sensible and latent loads.

Therefore, for a given occupancy and OA condition, $\beta_{e,IAQ,d}$ also defines the quantity of thermal energy needed to process the OA intake in a TAHU system. This thermal energy consumption might be saved by converting it into an IAHU system. $\beta_{i,IAQ}$ does not have a direct impact on the system's energy saving capacity after transforming a TAHU into an IAHU system; however, it does lay out the low limit for $\beta_{i,eco}$ in winter free cooling mode.

In a real time operation of IAHU, when the OA latent load is shifted from the exterior zone to the interior zone, the two variables, along with the two zone airflow ratios (φ and (1- φ)), jointly determine the feasibility of the conversion. The overall cooling demand might be beyond the cooling capacity of the cooling coil in the interior zone AHU. Under this circumstance, part of the OA for the exterior zone should be introduced directly from the AHU serving the exterior zone, or the system must be kept as a TAHU system.

Both variables are included in the expression for the exterior zone's OA intake in IAHU. The derivatives of the variables are given as below:

$$\frac{\partial \beta_{e,IAQ}}{\partial \beta_{e,IAQ,d}} = 1 \tag{3-1}$$

$$\frac{\partial \beta_{e,IAQ}}{\partial \beta_{i,IAQ}} = \frac{\gamma\varphi}{(1-\varphi)\beta_i} \tag{3-2}$$

It can be seen that the derivatives of both variables are positive; therefore these variables have a positive correlation with $\beta_{e,IAQ}$. In other words, $\beta_{e,IAQ}$ increases if either of the two variables increases. The value of the two variables sets the low limit for $\beta_{e,IAQ}$.

In general, the two variables have the highest freedom in an IAHU operation since they interact with each other and are not influenced by the other three variables. They are considered to be adjustable parameters in real project implementations. High β_{IAQ} creates a high demand on β_i in an IAHU operation, which might make IAHU unsuitable. Therefore, if possible, it is recommended to reevaluate real time β_{IAQ} when a building is partially occupied.

$\beta_{i,eco}$ and $\beta_{e,eco}$:

When a zone is in a cooling mode, and the OA temperature is lower than the zone AHU cooling coil set point, OA can be directly mixed with the zone return air for cooling purposes. The amount of OA intake for the economizer control is given by equation (2-43). The measurements of the OA temperature and RA temperature are needed to determine the value of the two variables.

When the exterior zone is also in a cooling mode and the OA is cold enough to enable free cooling, an IAHU system is identical to a TAHU system. Since cooling does not consume any thermal energy in this condition, it is not necessary to re-circulate air from the interior zone to the exterior zone and there is no energy savings by doing so. The purpose of OA intake is to satisfy both the thermal and respiratory needs of the exterior zone. $\beta_{e,eco}$ only influences the OA intake ratio for the exterior zone and has no impact on the interior zone OA.

$\beta_{i,eco}$ is jointly defined by the indoor thermal load of the interior zone and the OA temperature during a free cooling season. When the OA temperature is low, OA is mixed with the return air and then supplied into the space. For most OA temperatures and the same zone cooling load, $\beta_{i,eco}$ is greater than $\beta_{i,IAQ}$. Fig 3-1 shows that, even when the OA temperature drops to 0°F degree, the OA ratio is above 0.25. This provides extra credit to the freshness of the interior zone air in winter, so that it can be circulated into the exterior zone for ventilation.

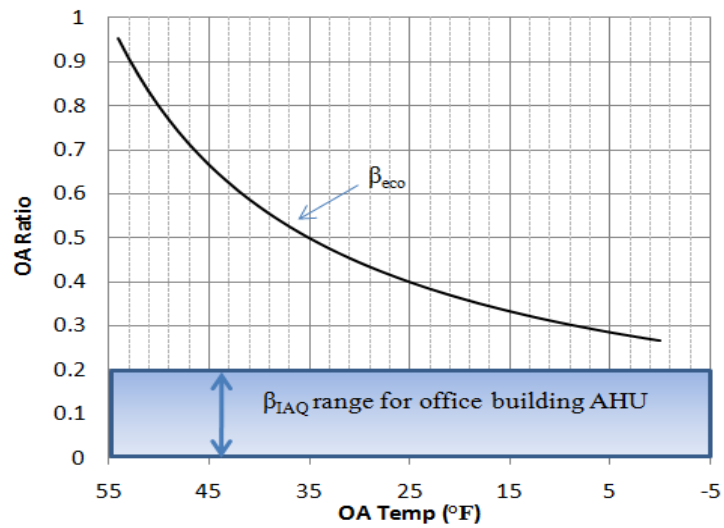


Figure 3-1: OA ratio for economizer

With a relatively constant thermal load in the interior zone, it is possible to further save the fan power energy in IAHU with VFDs by reducing the supply air temperature until the fan airflow drops to a certain value (e.g. 50%). The airflow rate and fan power consumption vs. the supply air temperature is plotted in Fig 3-2.

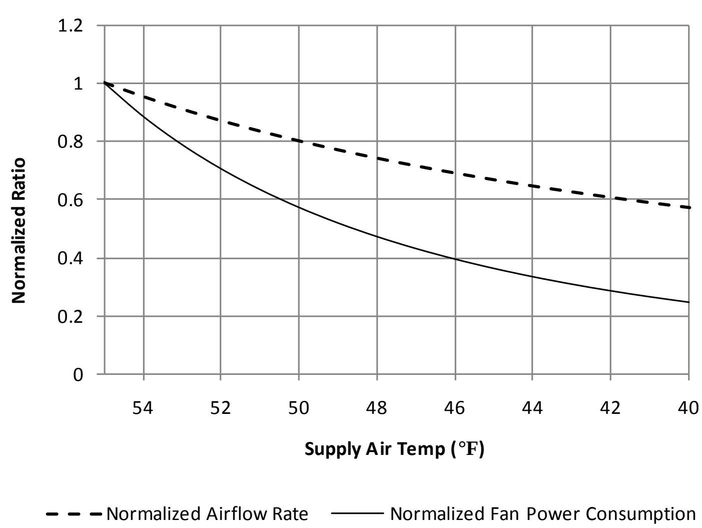


Figure 3-2: Normalized airflow and power consumption with different SA temp

When the supply air temperature decreases from 55°F to 50°F in a free cooling mode, the supply airflow rate is about 85% of that in a normal operation, and the power consumption could be reduced by about 40%. At 46°F, the fan power can be saved by 60%. If the diffusers do not draft the air, a reasonably low SA temperature in winter is beneficial to an IAHU system.

$\beta_{i,eco}$ and $\beta_{e,eco}$ work as constraints in the setting of $\beta_{e,IAQ}$. The real time desired value of the two variables is calculated based on the measurements of air temperature sensors. The actual value can be further verified by using an airflow station on the OA side and FAS on the fans. The two variables are independent to other variables in IAHU, and they only appear and need concerns in cold winter.

φ and airflow rate:

In IAHU, the exterior zone OA intake ratio is defined by equation (2-25) for IAQ consideration. In the expression, the interior zone supply airflow rate ratio, φ, is included as one of the most important variables. The value of φ has a dynamic impact on the real time OA ratio $\beta_{e,IAQ}$.

The interior zone airflow influences the feasibility of re-circulating air from the interior zone to the exterior zone. Unlike the other variables in the optimized exterior zone OA intake ratio expression, it generally cannot be arbitrarily manipulated. The absolute value is jointly determined by the zone thermal load and the supply air temperature. The supply airflow rate, as well as the ratio, is a dependent variable in the thermal process of IAHU.

As an airflow rate ratio, φ is influenced by two airflow rates: that of the interior zone and that of the exterior zone. The air conditioning mode of the interior zone is determined mainly by the internal load, which might be regarded as a constant cooling source. The exterior zone is highly influenced by the OA temperature. Therefore, the airflow ratio qualitatively has the trend to decrease when the OA temperature increases in summer. In winter, the airflow ratio decreases with a small slop when the OA temperature drops.

The derivative of the equation (2-25) over φ is given in equation (3-3):

$$\frac{\partial \beta_{e,IAQ}}{\partial \varphi} = (\frac{\beta_{i,IAQ}}{\beta_i} - 1)\frac{\gamma}{(1-\varphi)^2} \tag{3-3}$$

From the expression, it can be seen that the value is usually negative since β_i is generally greater than $\beta_{i,IAQ}$ in an IAHU system. Therefore, φ has an inverse correlation with $\beta_{e,IAQ}$. In other words, with given value of the other variables, a higher value φ gives a lower value $\beta_{e,IAQ}$.

Fig 3-3 illustrates the change pattern of $\beta_{e,IAQ}$ and φ with arbitrary variables in two cases.

Case 1: $\beta_{e,IAQ,d} = 0.1$, $\gamma = 0.25$, $\beta_{i,IAQ} = 0.1$, $\beta_i = 0.3$

Case 2: $\beta_{e,IAQ,d} = 0.1$, $\gamma = 0.4$, $\beta_{i,IAQ} = 0.1$, $\beta_i = 0.5$

With Case 1, when the supply airflow ratio is 0.4 or more, there is no direct OA intake requirement on the exterior zone to satisfy the IAQ. If the OA intake to the interior zone is higher than 0.5, the minimum supply airflow ratio is less than 0.25.

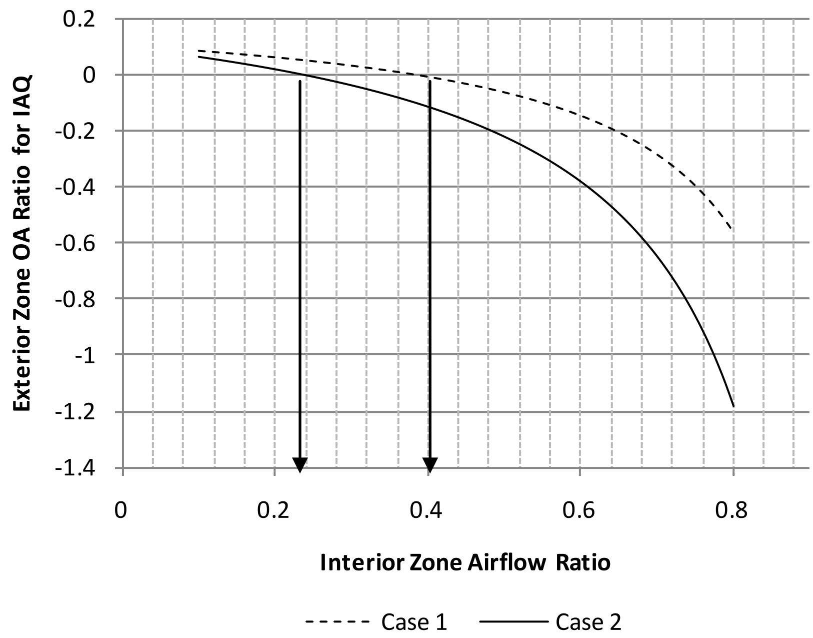


Figure 3-3: Exterior zone OA ratio for two cases

With the arbitrarily picked value for the variables, $\beta_{e,IAQ}$ decreases and becomes negative when the interior zone airflow is high (i.e. 0.4 in case 1). That means more than the needed amount of fresh air is circulated into the exterior zone. In a real operation of IAHU, a tradeoff evaluation between the ventilation and energy savings should be conducted to decide the allocation ratio γ. When the air quality constraint is satisfied, indoor air should only be re-circulated across the zones if it brings in more energy benefits. This situation happens in winter when more recirculation air can bring both freshness and heat from the interior zone to the exterior zone. This point will become clear in the control algorithm analysis.

In summary, φ is an important independent variable in deciding $\beta_{e,IAQ}$. It cannot be arbitrarily selected unless it is necessary and proper to bring in energy savings. With a given thermal load, it also results in a corresponding change of the supply air temperature. The airflow rate information should be obtained in a timely manner in an IAHU operation to ensure the exterior zone is properly ventilated with enough OA.

***β_i* and *γ*:**

These two are the most critical variables in the operation of an IAHU system. They have the highest freedom of value adjustment and are naturally correlated with the relation $0 \leq \gamma \leq \beta_i \leq 1$. In other words, they constrain mutually for each other. To obtain an acceptable IAQ of the exterior zone, we can either re-circulate more air from the interior zone with a given OA intake ($\beta_i \rightarrow, \gamma \uparrow$), or keep the re-circulation but take more OA ($\gamma \rightarrow, \beta_i \uparrow$).

The derivative of β_i has been studied in the last chapter for several different operations. The common expression is:

$$\frac{\partial \beta_{e,IAQ}}{\partial \beta_i} = -\frac{\gamma \varphi}{(1-\varphi)} \frac{\beta_{i,IAQ}}{\beta_i^{\,2}} \tag{3-4}$$

The negative derivative means that β_i is inversely correlated to $\beta_{e,IAQ}$. Since, in most mechanical heating/cooling seasons, the OA intake consumes energy, less OA is generally desired if the IAQ can be maintained with that amount of OA. In winter, because the interior zone has a constant cooling load in most office buildings, the OA intake is determined by the interior zone thermal load instead of an IAQ ventilation requirement. Under this circumstance, β_i evolves to $\beta_{i,eco}$, which is generally higher than $\beta_{i,IAQ}$ as is illustrated in Fig 3-1 for different OA temperatures based on equation (2-43).

In IAHU, to ensure the IAQ in both zones, β_{IAQ} for the interior zone is improved since the ventilation air must also be re-circulated into the exterior zone. With 10%, 15%

and 20% OA intake ratios for the two zones in a conventional TAHU system, the combined OA ratio in an IAHU operation is plotted in Fig 3-4. The plot shows that, by using IAHU and with OA being introduced from the interior zone only, β_{IAQ} could be much greater than the design ratio.

As analyzed in Chapter 2, IAHU saves building thermal energy consumption by shifting the OA intake into part of the AHUs under certain circumstances. The most appropriate situation happens in winter when the exterior zone needs heating while the interior zone needs cooling, and in summer when there is an unbalanced sensible load and latent load in the exterior zone. For winter applications, OA is beneficial but constrained by the interior zone thermal cooling load. For summer applications, OA is not welcome but needed for ventilation.

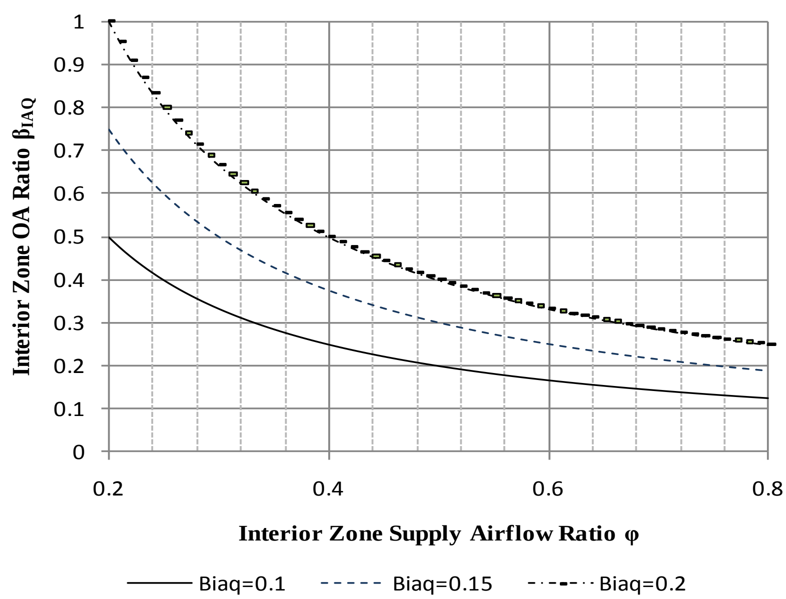


Figure 3-4: Interior zone OA ratio with different β_{IAQ}

The OA intake ratio β_i has a mutual dependency on the airflow ratio (γ) re-circulated from the interior zone to the exterior zone as illustrated in the last section for different scenarios.

The derivative of $\beta_{e,IAQ}$ on γ has the following expression:

$$\frac{\partial \beta_{e,IAQ}}{\partial \gamma} = \left(\frac{\beta_{i,IAQ}}{\beta_i} - 1\right)\frac{\varphi}{(1-\varphi)} \tag{3-5}$$

It can be seen that the derivative is always negative since β_i is greater than $\beta_{i,IAQ}$ in IAHU. This means that $\beta_{e,IAQ}$ decreases when γ increases. It is easy to understand from physics that more re-circulated fresh air from the interior zone naturally reduces the direct OA demand through the exterior zone AHU.

In the previous chapter, most concerns were placed on the IAQ related deduction with a symbolic γ. In an IAHU system, the re-circulated air extracted from the interior zone to the exterior zone is of a given value. The amount of air from the interior zone is allocated through the return air fan and the exhaust fan of the exterior zone.

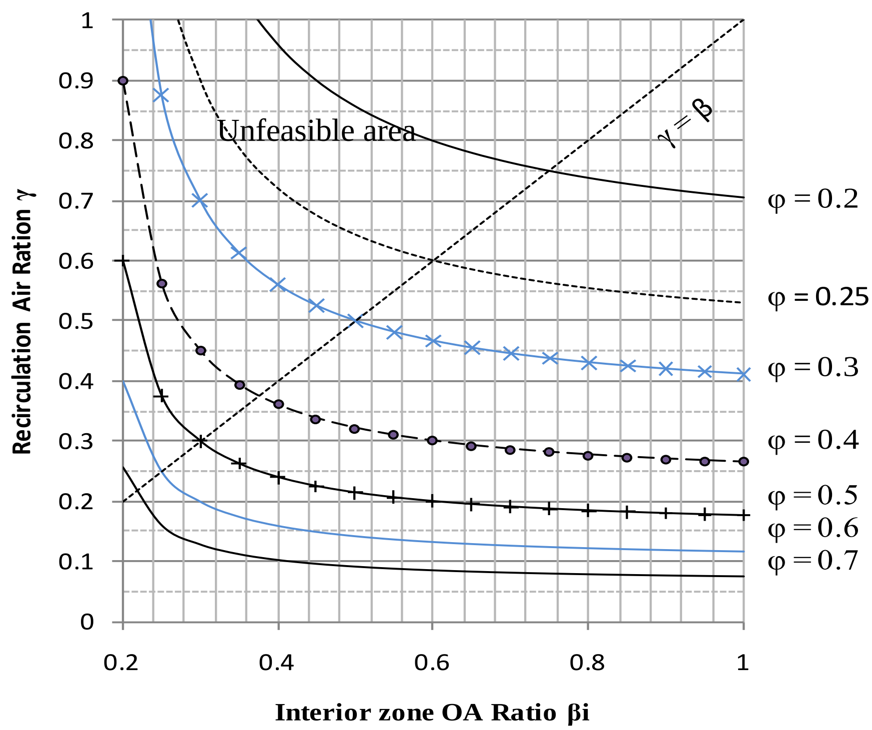


Figure 3-5: Minimum recirculation air ratio with different βi

With equation (2-25), if we assume $\beta_{e,IAQ}$ is zero, we obtain the following relationship between γ and β_i:

$$\gamma = \frac{(1-\varphi)\beta_{e,IAQ,d}}{(1-\frac{\beta_{i,IAQ}}{\beta_i})\varphi} \tag{3-6}$$

With $\beta_{e,IAQ,d} = \beta_{i,IAQ} = 0.15$, Fig 3-5 depicts γ and β_i based on equation (3-6).

With a given interior zone airflow rate ratio φ, when the OA intake ratio of the interior zone is small ($\beta_i \leq 0.4$, for example), the recirculation air ratio γ increases quickly in order to attain an acceptable IAQ in the exterior zone if β_i decreases. If γ increases into the infeasible area ($\gamma \geq \beta_i$), an IAHU system has to take at least part of the OA directly from the exterior zone AHU to satisfy the IAQ.

In an IAHU winter operation, since the RA temperature is suggested to be six Fahrenheit degrees (adjustable) higher than that of the exterior zone, when it is proper, a larger γ provides more thermal energy savings recovered from the interior zone to the exterior zone. However, γ has to be constrained in the range $\gamma \leq \beta_i$. When the OA temperature drops, both variables and the heat transfer capability, decrease. The trend roughly follows that of β_{eco}.

An amount of air with the ratio $\eta\cdot(1-\varphi)$ is later exhausted from the exhaust fans in the exterior zone to balance the amount of re-circulated air that comes from the interior zone. Part of this air, $\gamma\cdot\varphi - \eta\cdot(1-\varphi)$, goes through the return fan. Concern arises only if the exhaust fan of the exterior zone cannot remove the additional air, and there is a significantly more power consumption added to the RA fan. The influence of additional return fan power consumption is considered in the control algorithm section for a reasonable balance but it is not critical in thermal energy analysis, as generally the power consumption of a return air fan is much smaller than that of a supply air fan; the return fan power of the interior zone decreases at the same time; and the additional power usage is beneficial in winter to the exterior zone.

Feasible β_i for thermal decoupling in mild weather:

In the deduction of IAHU operation during summer partial load, it has been stated that shifting OA intake might not save energy in conditions D and E. Introducing a partial load ratio, $\mu = \frac{h_r - h_{e,s}}{h_r - h_{e,c}}$, the normalized cost function of IAHU, combining equation (2-77) and (2-78), is:

$$f = (h_{oa} - h_r)\left(\varphi\beta_i - \varphi\beta_{i,IAQ} - (1-\varphi)\beta_{e,IAQ,d}\right) - 2\left(h_r - h_{e,c}\right)(1-\mu)(1-\varphi) \tag{3-7}$$

Correspondingly, with all other variables defined, β_i should satisfy the following equation to avoid a positive additional cost:

$$\beta_{i,eng} \leq \frac{2\left(h_r - h_{e,c}\right)(1-\mu)(1-\varphi) + (h_{oa} - h_r)(\varphi\beta_{i,IAQ} + (1-\varphi)\beta_{e,IAQ,d})}{(h_{oa} - h_r)\varphi} \tag{3-8}$$

It can be seen that with no air relieved or exhausted from the interior zone, equation (3-8) holds true unless it violates any other mechanical constraints. The circulated air is preferred to be less than 30% (the ratio *x* is adjustable) of the exterior zone total airflow rate. Therefore, a realistic OA intake ratio for the interior zone in conditions D and E takes the constraint:

$$\beta_i = \beta_{i,min} \leq \frac{x(1-\varphi)}{\varphi} \tag{3-9}$$

where $\beta_{i,min} = \frac{\beta_{e,IAQ,d}(1-\varphi)+\beta_{i,IAQ}\varphi}{\varphi}$.

The chart is plotted below for x= 0.35:

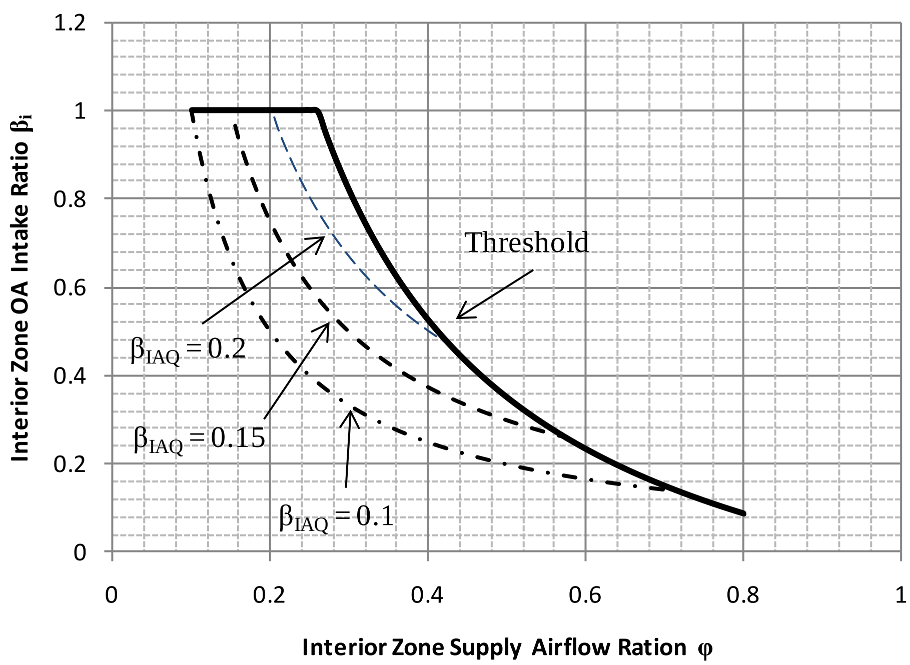


Figure 3-6: Feasible β_i with different φ, x= 0.35

The feasibility of IAHU thermal decoupling in mild weather depends on the real time interior zone supply airflow rate ratio and circulation capability between the two zones. For example, when the interior zone airflow ratio is 0.6 and the minimum OA intake for both zones is equal to or above 0.15, IAHU is not suitable to decouple the thermal load. For TAHUs, there are minimum interior zone supply airflow ratios where the corresponding β_{IAQ} in IAHU is 1. If the interior zone air ratio drops to lower than the critical points, summer thermal decoupling is not feasible either.

The chart also partially provides the information about the difference between IAHU and OAHU in terms of feasibility and capability. Take β_{IAQ} = 0.15, for instance, when the interior zone supply airflow ratio is higher than 0.58, an OAHU system surpasses an IAHU system for mild weather thermal decoupling. And in winter, the heat transfer ability could be impaired in an IAHU when β_i is lower than 0.3, which happens when the OA temperature is lower than 10°F. The lower the interior zone supply airflow ratio, the closer an IAHU system is to an OAHU system.

3.2 Control algorithm and implementation

In the previous deduction, the variables are normalized into ratios. The number of AHUs in an office building is also simplified into two AHUs: one for the interior zone and one for the exterior zone. In a real project, there could be different AHU-to-zone relationships and the utilization of IAHU should consider all possible constraints. A proper control of IAHU requires knowledge of all the related variables. This section deals with the general control algorithm and implementation guidelines for IAHU.

The table below summarizes the variables and some main operation points in IAHU:

Table 3-1: Collection of IAHU variables

Item	**Ratio**	**Quantity**	**Obtaining Means**	**Info**
1	μ_i	N_i	f (time, day)	Occupancy of the interior zone
2	μ_e	N_e	f (time, day)	Occupancy of the exterior zone
3	$\beta_{i,IAQ}$	$Q_{i,IAQ}$	f (μ_i ,N_i, m_i)	Originally required OA intake for the interior zone
4	$\beta_{e,IAQ,d}$	$Q_{e,IAQ,d}$	f (μ_e, N_e, m_e)	Originally required OA intake for the exterior zone

5	$\beta_{i,eco}$	$Q_{i,eco}$	f ($T_{i,r}$, $T_{i,c}$, T_{oa}, m_i, $Load_i$)	Economizer OA intake for the interior zone
6	$\beta_{e,eco}$	$Q_{e,eco}$	f ($T_{e,r}$, $T_{e,c}$, T_{oa}, m_e, $Load_e$)	Economizer OA intake for the exterior zone
7	φ	m_i	f (m, $T_{i,c}$, $Load_i$)	Interior zone airflow
8	1- φ	m_e	f (m, $T_{e,c}$, $Load_e$)	Exterior zone airflow
9	β_i	Q_i	f ($\beta_{i,IAQ}$, $\beta_{e,IAQ,d}$, γ, φ, m_i)	OA intake from the interior zone
10	γ	Q_{cir}	f (β_i, $\beta_{i,IAQ}$, $\beta_{e,IAQ,d}$, φ, m_i)	Recirculation air from the interior zone to the exterior zone
11	ξ	$Q_{i,ref}$	f (β_i, γ, δ, m_i)	Relief air from the interior zone
12	δ	$Q_{i,exh}$	f (β_i, γ, m_i)	Exhaust air from the interior zone
13	η	$Q_{e,exh}$	f (β_e, γ, m_i, m_e)	Exhaust air from the exterior zone
14	T_{oa}, $T_{i,r}$, $T_{e,r}$		measurement	OA and room air temperature
15	$T_{i,c}$, $T_{e,c}$		define and calculate	Control cold deck temperature
16	VFD, $VFD_{e,min}$, $VFD_{i,min}$, $m_{e,d}$		measurement and design information	Variable frequency drive and exterior zone designed airflow rate

The AHUs in an IAHU system could be either CAV or VAV. In a CAV system, there is no VFD installed. To accommodate varying loads, the supply air temperature is adjusted. The distribution of the air is accomplished by modulating the dampers and the number of exhaust fans. Because the redistribution of air between the interior zone and the exterior zone is mainly between the two return fans, the operation of the supply fans will not be modified. The supply air temperature of the AHUs in an IAHU system is subject to the same algorithm used in a CAV based TAHU system. Airflow meters are needed to monitor the corresponding airflow rate info on the outlets of the duct work. The details will be covered in Chapter 4 for IAHU evaluation.

In a VAV system, VFDs are installed on all, or at least most, of the AHU fans. The modulation of VFDs on the return fans is mainly utilized to achieve the air distribution among the AHUs and across the building. The dampers are supportive to the

VFD modulation. Fan Airflow Station® (FAS) developed by the Energy Systems Lab (ESL) at the University of Nebraska-Lincoln is to be implemented for more accurate and reliable monitoring on airflow rates. Additional airflow meters are recommended on the OA inlets of AHUs. The exhaust fans are controlled to support the IAHU operation.

3.2.1 Airflow measurements

The critical variables, especially the ratios, in IAHU require knowledge of the airflow rates for the interior zone and the exterior zone. The building air is guided to flow from the interior zone to the exterior zone and then to the outdoor. The pressure gradient is achieved by maintaining the airflow rates and their differences.

Airflow rate measurement in HVAC may be obtained by measuring the velocity or the air stream dynamic pressure. The velocity is then configured onsite with the coefficients set to account for the duct area and installation influence to obtain the volumetric information of airflow. The preferred method of measuring duct volumetric flow is the pitot-tube traverse average (ASHRAE handbook, ch37, 2007). Other supplementary measures can be made based on CO_2 concentration or energy balance for mixture plenums when the air quantities cannot be determined accurately by volumetric measurements.

An FAS utilizes the fan laws to establish the pre-determined relationship of fan head and fan airflow rate under a given fan speed (Liu et al, 2005). If the fan curve is flat in the range of airflow, the fan power consumption and fan speed can be used to access the actual fan airflow rate (Wang and Liu, 2007). The application process of developing an FAS is briefly described here.

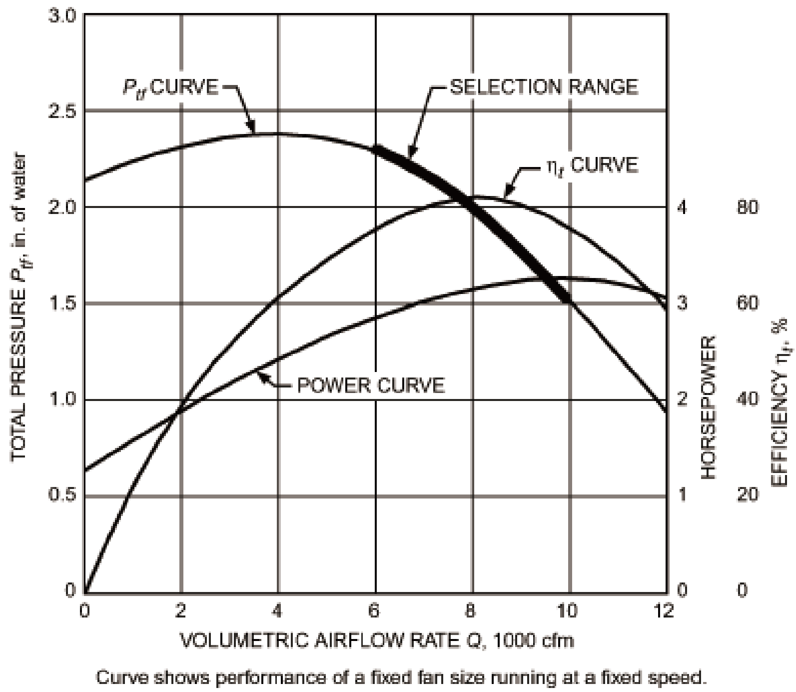


Figure 3-7: Typical fan performance curve

A fan performance curve shows the relationship between the quantities of air delivered by a fan and the pressure generated at various air quantities. The curve also shows the power consumption of a given quantity of airflow. The total fan head or fan power can be regressed as a function of the airflow rate using the design fan performance curve. A typical fan performance curve is illustrated in Fig 3-7 (ASHRAE handbook, ch20, 2008).

The methods to obtain a fan curve at full speed by applying different system resistance are illustrated in the diagram below (ASHRAE handbook, ch20, 2008):

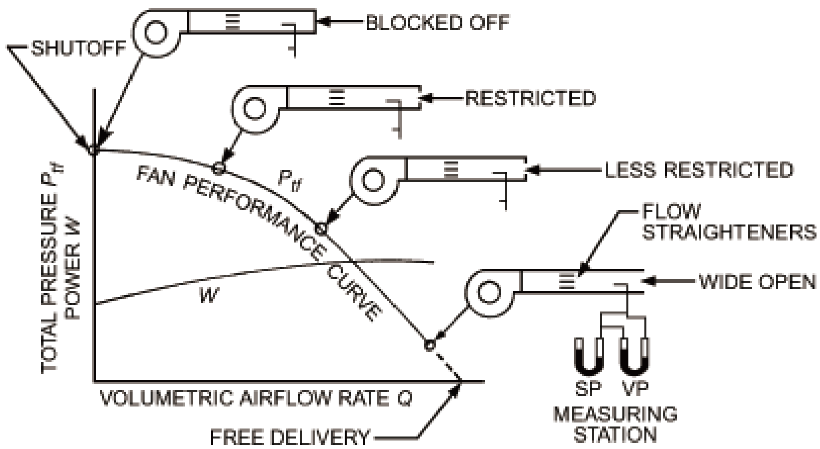


Figure 3-8: Fan curve measurement process

The fan laws relate the performance variables of any dynamically similar series of fans. The variables are fan size, rotational speed, gas density, volume airflow rate, pressure, power and mechanical efficiency. The basic set of theoretical fan law for the same size fan is given below:

Table 3-2: Basic set of fan laws

Law No.	Dependent Variables		Independent Variables
1a	$CFM_1 = CFM_2$	X	(N_1/N_2)
1b	$H_1 = H_2$	X	$(N_1/N_2)^2\ \rho_1/\ \rho_2$
1c	$W_1 = W_2$	X	$(N_1/N_2)^3\ \rho_1/\ \rho_2$

Both the fan head-based and the fan power-based FAS utilize the fan curves and fan laws to regress the airflow rate.

Assume the fan head-airflow curve can be expressed using a second order polynomial equation under the full speed for the total fan head and fan power respectively:

$$H = a_0 + a_1 CFM + a_2 CFM^2 \tag{3-10}$$

Combining fan law (1b) and the second order polynomial equation (3-7), for a partial speed N_2, the following equations hold:

$$\omega = \frac{N_2}{N_1} \tag{3-11}$$

$$H = \frac{H_\omega}{({N_\omega}/{N_1})^2} = {H_\omega}/{\omega^2} \tag{3-12}$$

$$CFM = \frac{CFM_\omega}{({N_\omega}/{N_1})} = {CFM_\omega}/{\omega} \tag{3-13}$$

Placing equation (3-10) and (3-11) into equation (3-7), the fan head in a partial speed expresses the following:

$$H_\omega = \omega^2(a_0 + a_1 CFM/\omega + a_2(\frac{CFM}{\omega})^2) \tag{3-14}$$

The fan airflow rate at different fan rotation speeds can also be deduced as:

$$CFM_\omega = \frac{\left(-a_1 \pm \sqrt{a_1^2 - 4a_2(a_0 - \frac{H_\omega}{\omega^2})}\right)\omega}{2a_2} \tag{3-15}$$

In order to work, a fan curve based FAS requires the signals of the fan head across the fan and the fan rotational speed. The fan head can be obtained by installing a probe or pitot-tube across the fan. The fan speed is either obtained by utilizing a tachometer or based on fan VFD commands. A linear relationship with a constant intercept is found in most applications. Caution should be exercised when a VFD reaches the minimum percentage.

When the fan curve for the fan head and airflow rate is flat in a partial load, equation (3-13) might introduce a large error. Under this circumstance, a fan power based FAS should be referred in order to obtain an accurate fan airflow rate. Since fan power is difficult to measure, some assumptions have to be reasonably made to obtain the airflow rate.

The power in a fan curve is the fan power input to the fan shaft (W_{shf}). The fan power (W) in the fan law (1c) is the fan power to the air. The actual curve should be adjusted by integrating the fan power curve and the fan efficiency curve (η_{fan}).

$$W = W_{shf}\eta_{fan} \tag{3-16}$$

The relationship of the various powers is illustrated below:

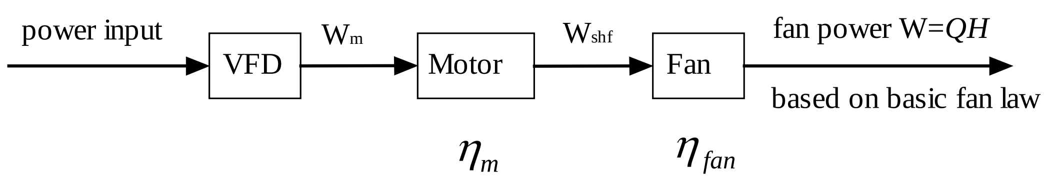


Figure 3-9: Different powers in fan application

Motor power is the power delivered into the shaft plus the motor losses. Three-phase induction motors are widely used in HVAC systems. For induction motors, based on motor theory, the motor losses can be expressed as a function of both the fan speed and the fan power. The deduction is given in literature (Wang and Liu, 2007, Liu, 2006).

The following equation correlates the required fan power to the measureable motor power and motor speed based on the motor theory:

$$W_m = (W_{shf} + b\omega^3) + c + d(W_{shf} + b\omega^3)^2 \tag{3-17}$$

where b, c, and d are factors determined by the motor size and configuration.

Solving for W_{shf}, we can obtain :

$$W_{shf} = \frac{-1+\sqrt{1-4d(c-W_m)}}{2d} - b\omega^3 \qquad (3\text{-}18)$$

Assuming the fan power is a linear function of the fan airflow, combining fan law, we have the power expression:

$$W = k_0\omega^3 + k_1\omega^2 CFM \qquad (3\text{-}19)$$

Also assuming the fan efficiency curve is approximately linear at the operation range:

$$\eta_{fan} = l_0 + l_1\frac{CFM}{\omega} \qquad (3\text{-}20)$$

Combining the equation from (3-16) to (3-19), we have the power-based FAS:

$$CFM_\omega = \frac{l_0\left(\frac{-1+\sqrt{1-4d(c-W_m)}}{2d} - b\omega^3\right) - k_0\omega^3}{(k_1+bl_1)\omega^2 - l_1\frac{-1+\sqrt{1-4d(c-W_m)}}{2d\omega}} \qquad (3\text{-}21)$$

3.2.2 Control flowchart

In the deduction of the IAHU concept and algorithms, the supply air temperature and the supply airflow of each individual AHU are not specified. The two variables are mainly determined by the zone load and the original TAHU system configuration. They do not violate the analysis on the IAHU OA intake to achieve energy savings and an acceptable IAQ in both zones. They can be viewed as the characteristics of an individual AHU system and are not changed because of a new IAHU operation. Zone loads vary along with the space conditioning means and/or AO temperature. For example, the supply air temperature might be reset in winter when the zone load drops to a specified threshold. The supply airflow rate could be constant in a CAV, or subject to a low limit in a VAV to ensure the air circulation.

There are four main combinations of single duct air based systems in commercial buildings:

1. CAV in both zones;

2. CAV in the interior zone, VAV in the exterior zone;

3. VAV in the interior zone, CAV in the exterior zone;

4. VAV in both zones.

Different types of AHUs have different limitations on the supply air temperature and supply airflow rate. This could lead to slightly different control in an IAHU system because of the AHUs' different features and the limitations they incur. In an IAHU operation, the OA temperature should be used as one of the most important indexes in the control algorithm to categorize which scenario the IAHU system is running in. This approach also matches the operation strategies of most office buildings and can be accepted by onsite facility engineers. The partial load status of the exterior zone should also be well defined and monitored to support the categorization.

The control algorithm of an IAHU system adds the OA intake and air distribution logic to the operational control of the original system.

The following part specifies the control for the aforementioned four AHU combinations.

CAV in both zones:

Because of its low initial cost and simple controls, CAVs are still used in office buildings, mainly in small premises. A CAV system accommodates the change of zone load by varying the supply air temperature or modulating terminal reheats to offset the change.

The corresponding IAHU flowchart for a dual CAV can be generalized as:

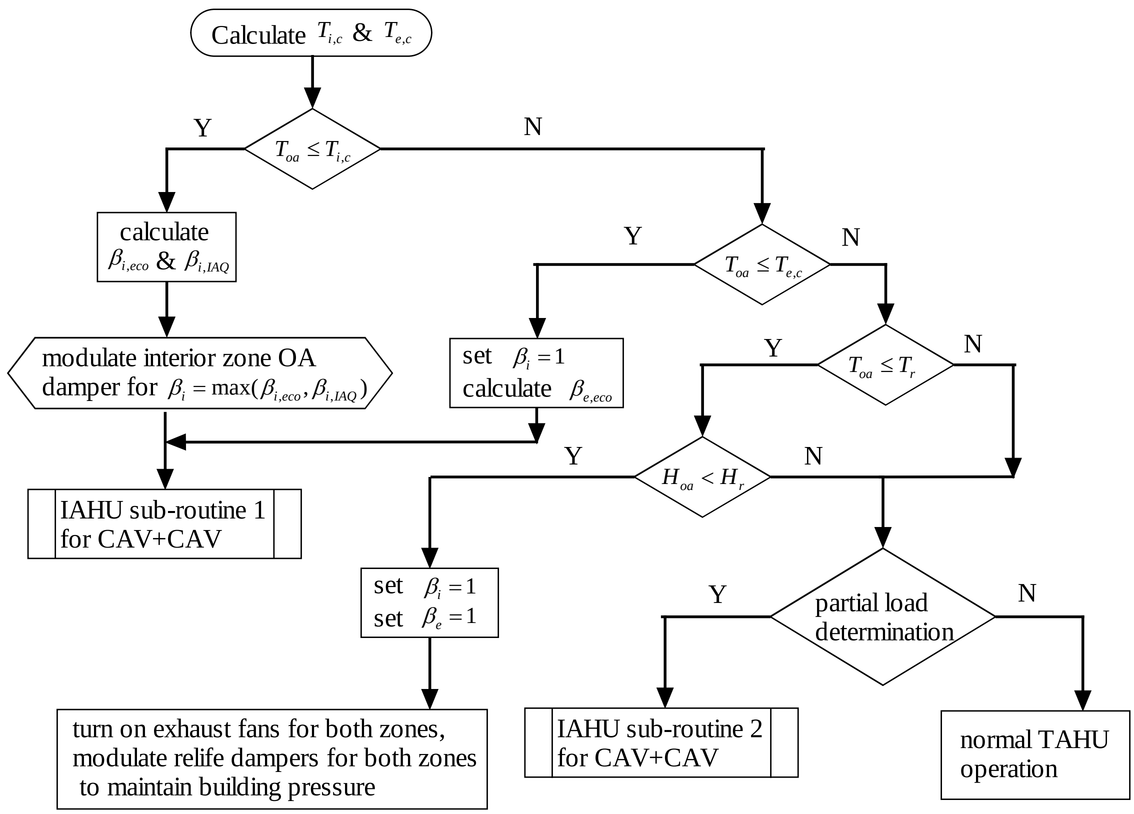


Figure 3-10: Main control algorithm for CAV in both zones

The AHU cold deck temperatures are reset based on the zone load in the two CAVs. The signal of terminal reheats (i.e. more than 20% of reheats are on), the return air temperature (i.e. 4°F less than the set point) and the duration (i.e. longer than 10 minutes) can be utilized to assist the determination in the supervisory control.

Sub-routine 1 in Fig 3-10 mainly checks whether the circulation air is in the redefined limits of IAHU when the OA is favorable to the interior zone. Exhaust fans and relief dampers remain off unless the air must be extracted directly from the interior zone. The relief dampers are modulated for a proper value, as given in the chart, to maintain the desired recirculation airflow rate γ into the exterior zone.

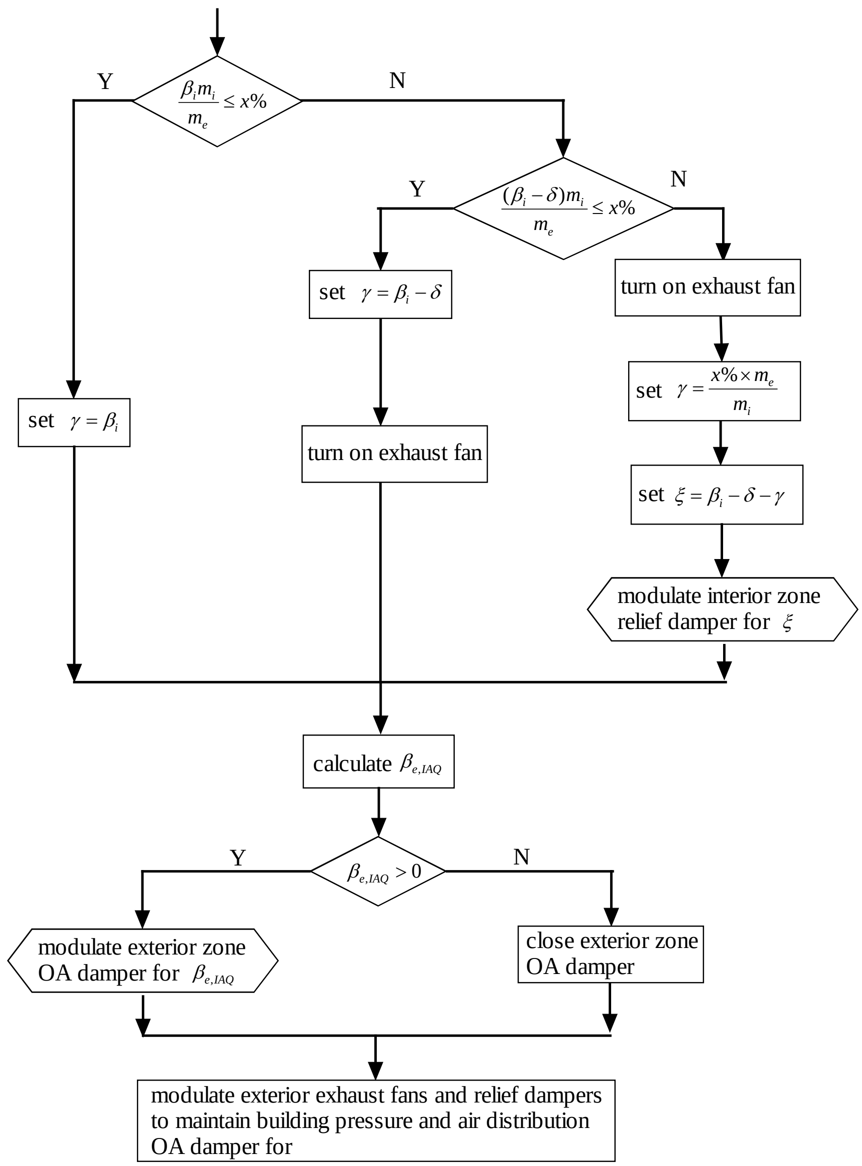


Figure 3-11: Sub-routine 1 for CAV system

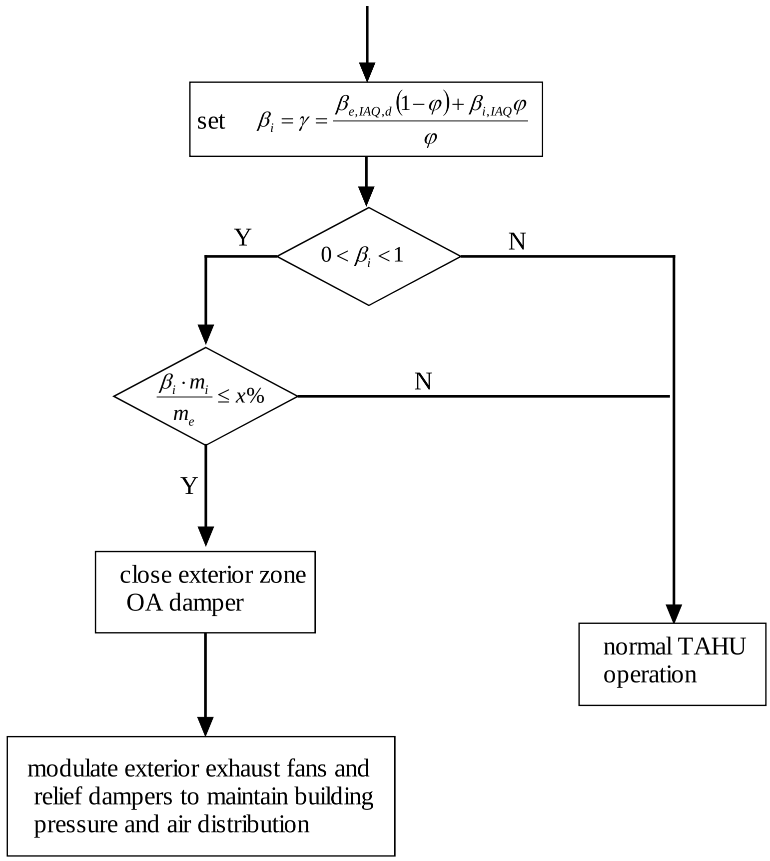


Figure 3-12: Sub-routine 2 for CAV system

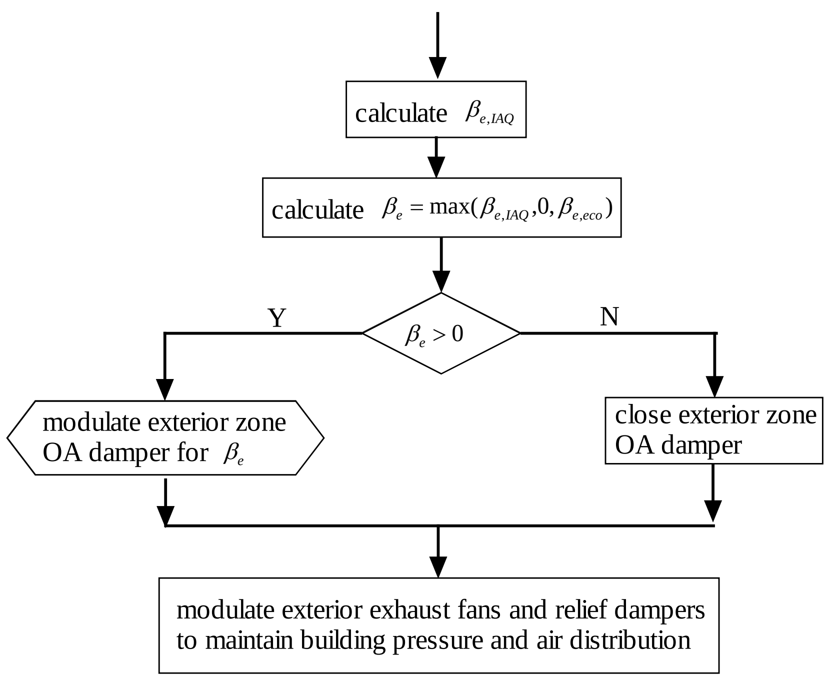


Figure 3-13: Sub-routine 3 for CAV system

In sub-routine 2, the feasibility of an IAHU system is determined. When it is not feasible due to the limit of the circulation airflow rate, an IAHU system is turned back into a normal TAHU system. Otherwise, the exterior zone OA damper and modulate relief dampers should be shut off for proper building pressure. Thermal decoupling will be triggered once the feasibility is verified.

Sub-routine 3 specifies the control of the exterior zone AHU. When a direct OA intake is needed, the exterior zone OA damper is modulated to satisfy the requirement. Otherwise, the OA damper should be closed. The exhaust fans and relief dampers in the exterior zone are controlled to maintain building pressure and air distribution.

CAV + VAV:

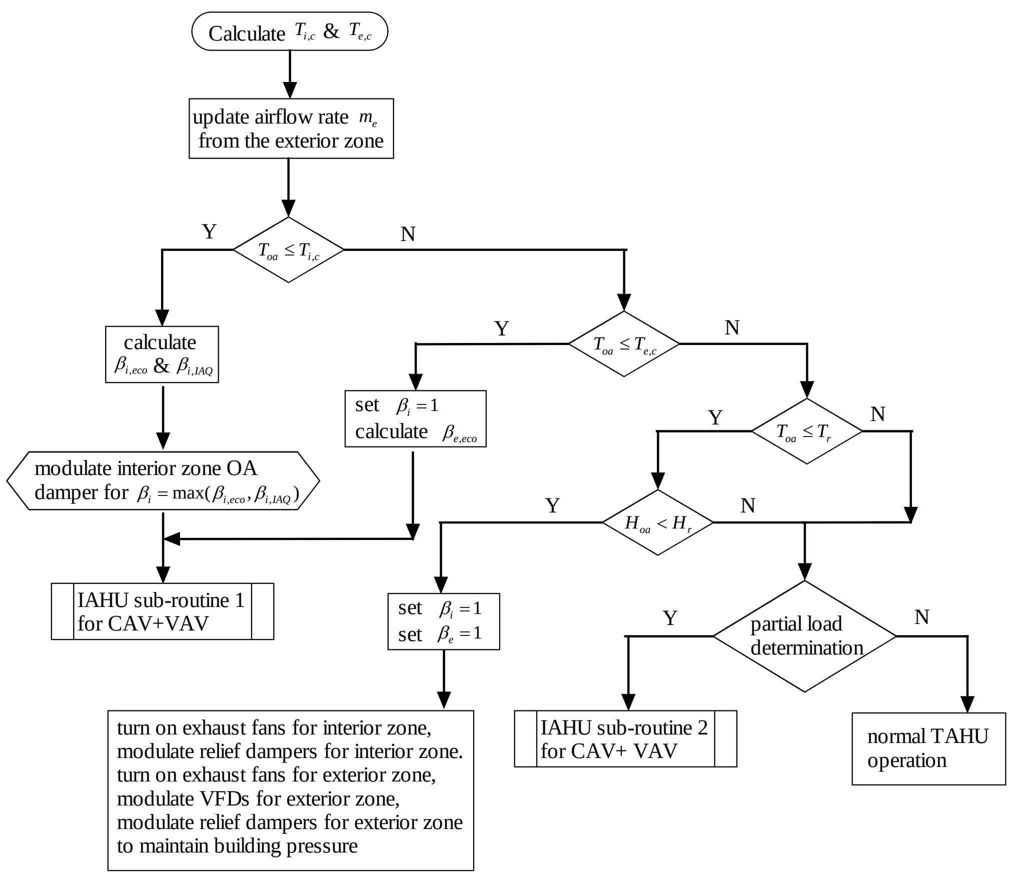


Figure 3-14: Main control algorithm for CAV + VAV

For a CAV + VAV combination, the exterior zone has the ability to adjust the supply airflow rate based on the real time zone thermal load. Therefore, the airflow rate needs to be updated from time to time via the FAS. The restriction on the air recirculation could be relaxed since there are VFDs installed on the supply and return fans.

The main control algorithm updates the real-time supply airflow rate from the exterior zone FAS. The cold deck temperature will not be reset in this VAV application unless it is needed to ensure the air circulation in the space when the supply airflow rate decreases to the minimum (i.e. 50%). Fan power can be saved by around 90% with this practice. The determination of the exterior zone partial load status in VAV can be made based on the supply fan airflow rate and the room air temperature. For example, it might be judged as a partial load status if the room air temperature is sensed to be lower than the set point (i.e. 4°F) and the supply airflow rate reaches the minimum.

The sub-routine 1 of a CAV+VAV is similar to the standard one given in the previous section for a CAV system. The only difference lies on the control of the exterior zone as given in sub-routine 3.

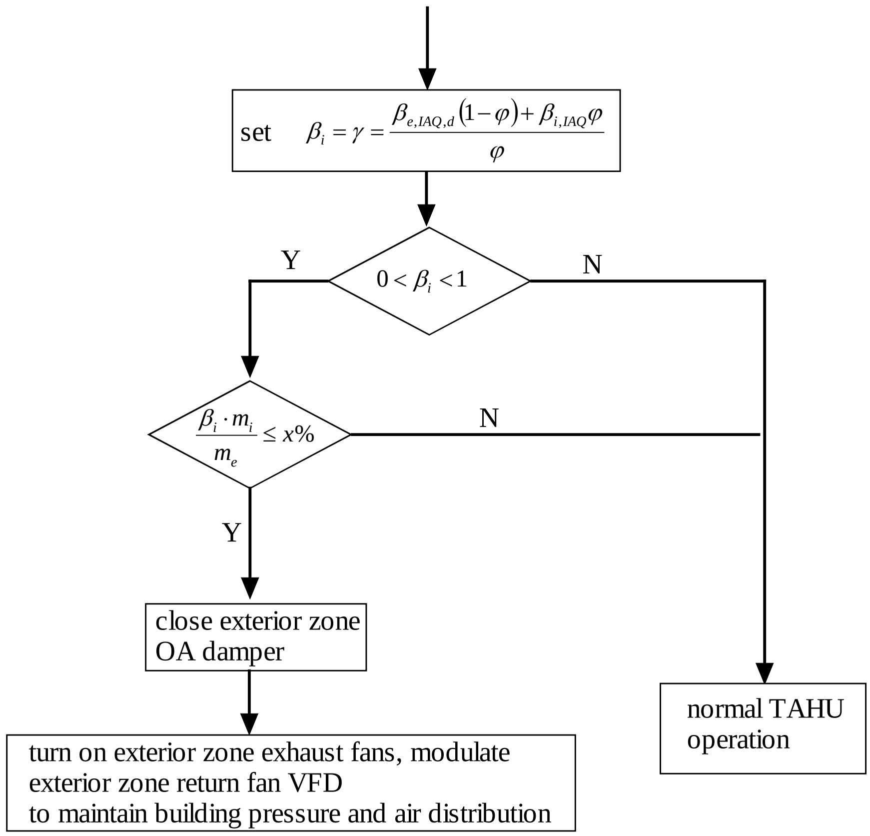


Figure 3-15: Sub-routine 2 for CAV+VAV

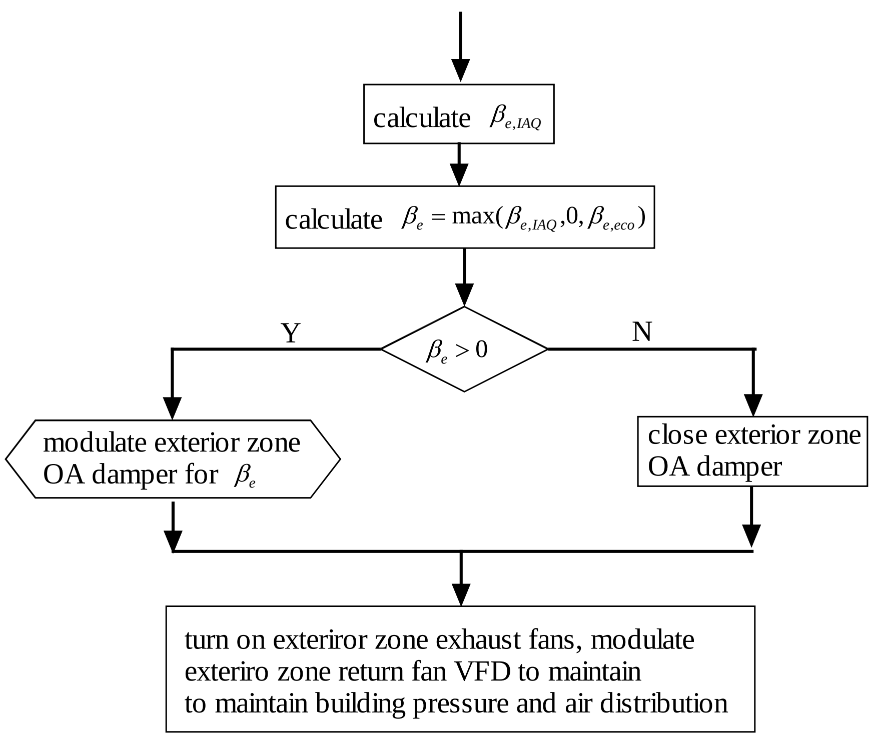


Figure 3-16: Sub-routine 3 for CAV+VAV

For the IAHU partial load operation in mild weather, the exterior zone exhaust fans and return fan VFDs are controlled to ensure the airflow across the zones and then to ensure the building pressure. If a direct OA introduction is needed from the exterior zone, the OA damper in the exterior zone is also modulated to satisfy the IAQ.

VAV+ CAV:

For a VAV + CAV combination, the interior zone is equipped with VFDs to adjust the supply airflow rate based on the zone load. The supply air temperature is typically 55°F for the interior zone until the airflow hits the low threshold (i.e. 50%). The combination is very common in office buildings where VAV boxes are used in the interior zone, and fin tube induction terminals are deployed along the external walls.

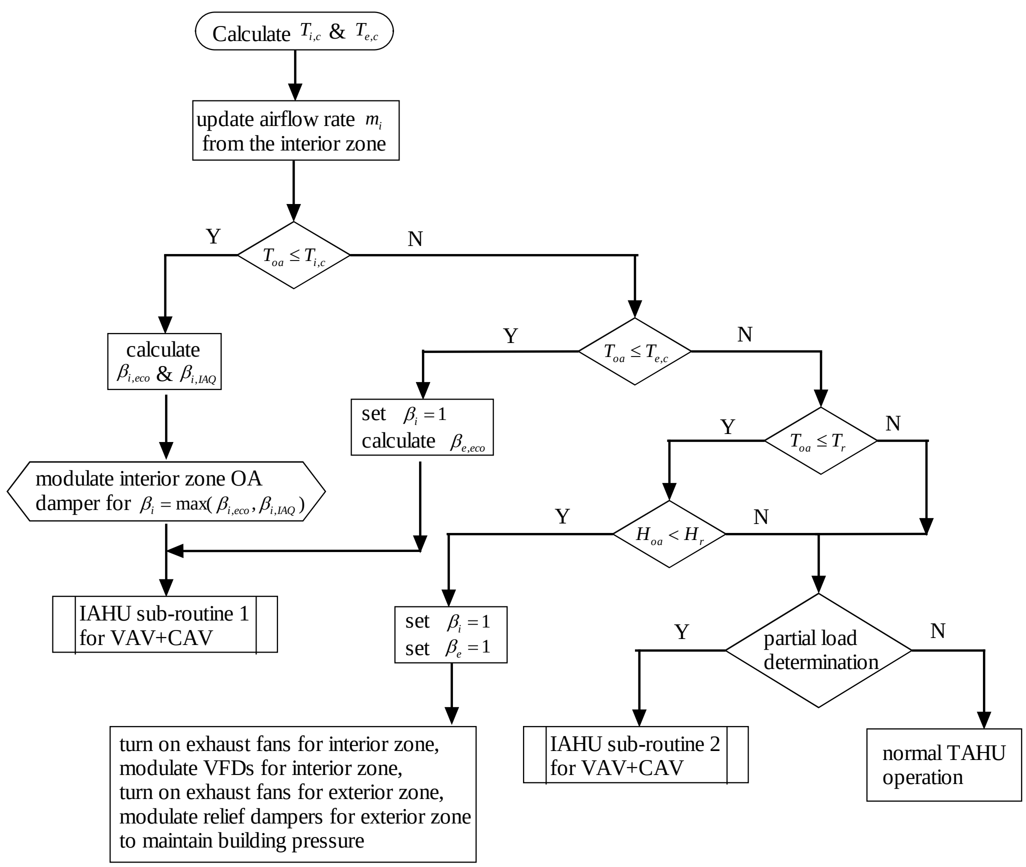


Figure 3-17: Main control algorithm for VAV+CAV

The airflow rate is read from the FAS of the interior zone supply fans. The airflow rate ratios might vary during the operation course in a day. The supply air temperature of the exterior zone might be a linear reset based on the OA temperature. The sub-routine for air circulation is similar to the standard one given in the CAV section, but the VFD on the return fan of the interior zone is modulated to ensure the airflow across the zones.

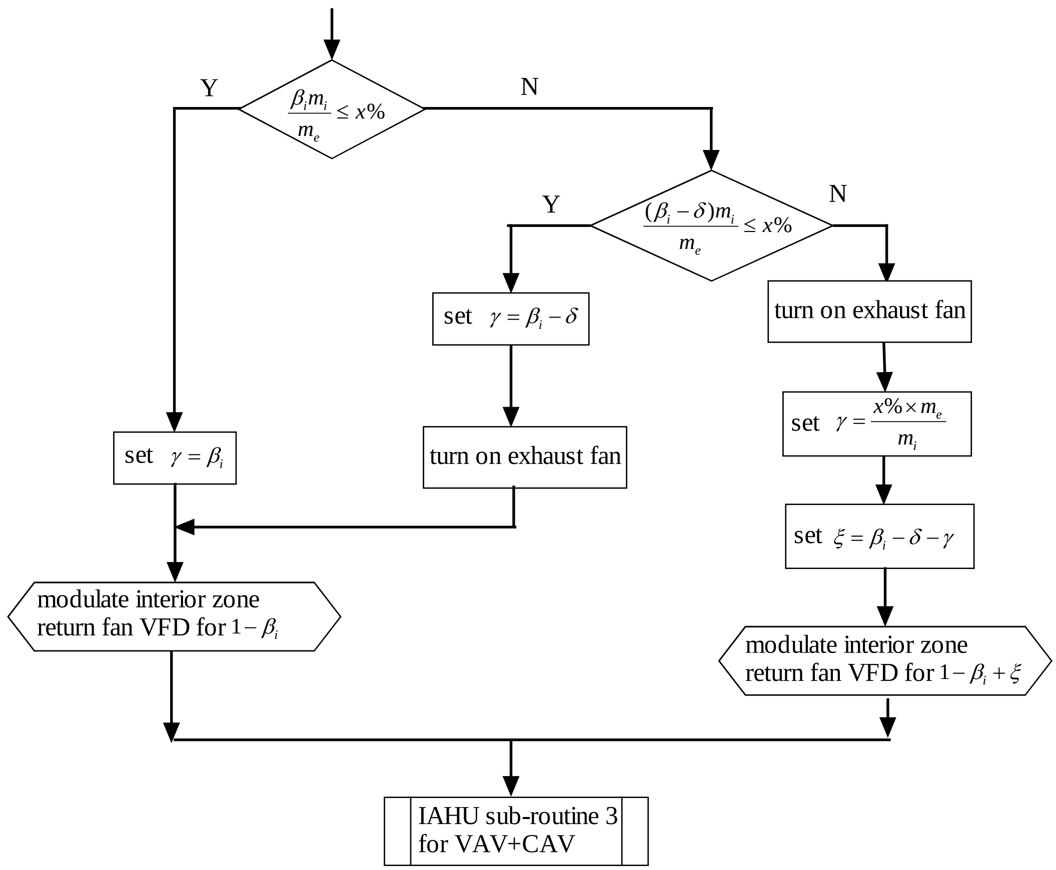


Figure 3-18: Sub-routine 1 for VAV+CAV

The sub-routine 2 and 3 for the exterior zone control in the VAV+CAV layout are similar to the standard one given in the CAV section.

VAV in both zones:

This combination is also common in commercial buildings. The AHUs have the capability to adjust airflow with the VFDs to accommodate the changing internal loads. Typically, the cold deck temperature remains 55°F until a reset is needed when the zone load makes the supply airflow drop to the minimum. The supply air temperature might be gradually reset in winter to ensure the airflow rate is above the minimum. The airflow across all the VFD fans needs to be measured and coordinated so that the IAHU operation maintains proper air distribution across the zones. The return fan VFDs serve as the central means for the IAHU to maintain both the airflow across the zones and the building pressure.

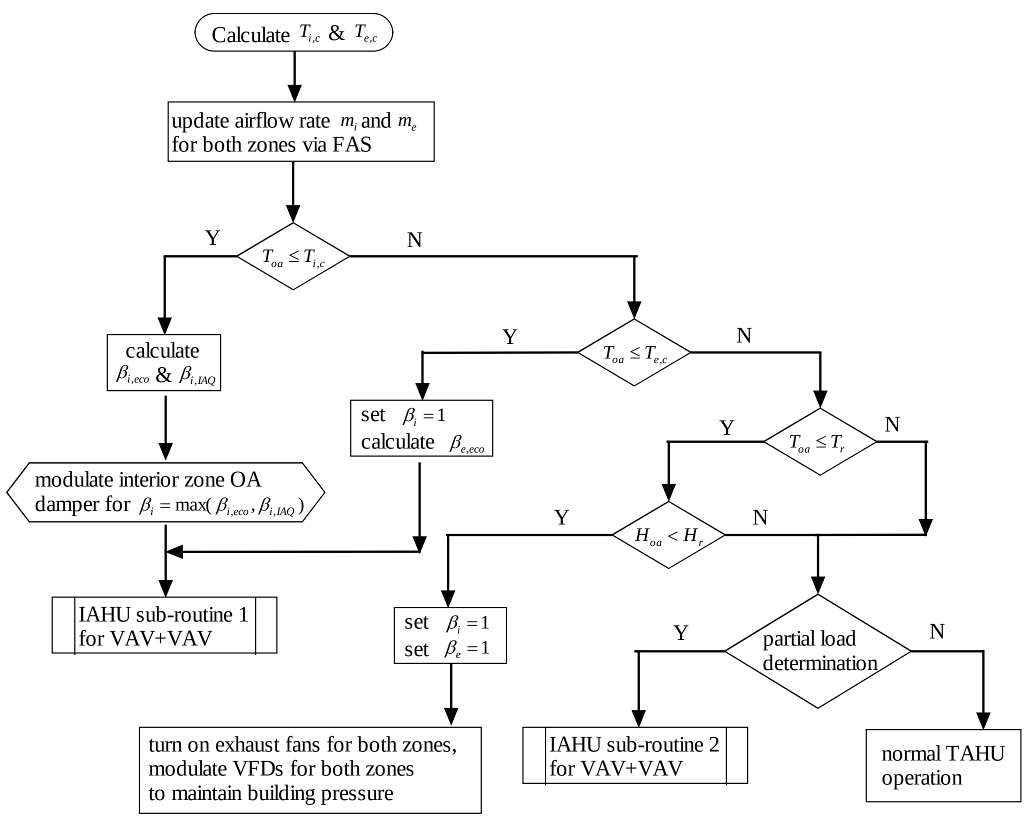


Figure 3-19: Main control algorithm for VAV system

The sub-routine 1 for the control of the equipment in the interior zone is identical to the one given in the VAV+CAV application. The sub-routine 2 for the control of the exterior zone is identical to the one given in the CAV+VAV application. The FAS installed on the return fans are utilized to fulfill the desired air circulation across the zones and the proper building pressure.

IAHU control algorithm summary

An IAHU system differs from a conventional TAHU system mainly because of its innovative control of the building OA intake. The freshness of air in one zone is circulated to the other zones for ventilation. The original zone conditioning modes are not changed in most operations, with one exception. In winter, the interior zone air temperature set point could be slightly higher than that of the exterior zone to improve the heat transfer capability of the circulated air.

The IAHU control algorithm adds control over the airflow modulating equipment to the original TAHU thermal logics to ensure the proper OA allocation and building pressure. The general control algorithm, regardless of the original AHU configuration, can be summarized for the previous system types:

1. Determine the proper fan speed and cold deck temperature for the individual zone. The supply airflow should be kept higher than the minimum to ensure indoor air circulation.
2. Obtain the information of the OA temperature, OA enthalpy, and fan airflow rates via the instruments. Update the supply fan airflow ratio for the two zones. If multiple AHUs exist in one zone, sum up the total before calculating the ratio.
3. Determine the scenario of IAHU based on Table 2-1.

4. Determine the feasibility of IAHU with the OA intake from the interior zone AHUs. If it is feasible, determine the distribution of the OA intake among the interior zone AHUs. Otherwise, preserve original TAHU operation and return to step 1.
5. Calculate the ratio of circulated air between the two zones. Calculate the OA intake from the exterior zone AHUs. Modulate the OA dampers on the interior zone AHUs to introduce in the desired flow rate of the OA. Modulate the OA dampers on the exterior zone AHUs to introduce in the calculated flow rate.
6. Coordinate airflow control components on the interior zone AHUs (return fan VFDs if they are equipped, exhaust fans, and relief dampers) to ensure the air circulation between the interior zone and the exterior zone. Coordinate airflow control components on the exterior zone AHUs (return fan VFDs if equipped, exhaust fans, and relief dampers) to ensure the air circulation and building pressure.
7. Switch back to normal operation if any of the conditions do not support the given scenario: the OA temperature, the OA humidity, the airflow ratio, or the OA ratio.

3.2.3 Instrumentation

To develop and utilize FAS in real building operations, the manufacture fan curve should be verified onsite. Fan head, fan power and fan airflow rates are linked via the fan curves and fan laws. The instrumentation for implementation is briefly discussed here.

The detailed process can be found in literature (Wang and Liu, 2007, Liu, 2003 and Liu 2006).

A fan speed is one of the key parameters that should be obtained to shift the fan curves. For most VAV systems, VFDs are installed and commanded to vary the fan speed. A linear relationship with a minimum rotational speed intercept can be obtained by utilizing a laser tachometer for a one-time measurement. After that, the fan speed is determined by utilizing the EMCS command. Special attention should be paid to the minimum EMCS command and the corresponding VFD frequency.

A fan head is the static pressure difference across a fan. It should be obtained by using the differential pressure transducer with either current or voltage signals. The manufacturer fan curve should be referred to in selecting the right pressure range. The design static fan head should be around 2/3 of the full scale of a pressure transducer. The most typical range is 5 in w.g. for HVAC applications. An accuracy of ±0.5% or higher can be achieved by brand pressure transducers. The pressure tap should be installed near the fan inlet and outlet. The relative AMCA standard should be referred to in determining the pressure probe location and installation for accurate measurements.

Power measurements can be achieved by using a power meter or sensor. This type of instrument is available with accuracies of 1% of the full scale for volts, amps and power factors and 2% of the full scale for watts. VFDs also have reliable output for the power related variables and can be fed into EMCS for sharing.

Chapter 4 IAHU PERFORMANCE EVALUATION

In the previous chapters, the concept and structure of IAHU has been introduced and discussed. The operation to ensure IAQ and energy savings is theoretically analyzed. The control algorithms are also developed for four typical system layouts, followed by the discussion of key implementation issues. As one of its main traits, IAHU can be integrated into Continuous Commissioning© (CC) to retrofit an office building for better energy and IAQ performance.

Since an IAHU operation is subject to the system static and dynamic constraints, the actual performance in terms of the quantity of energy saving differs from system to system. An energy evaluation of IAHU helps to identify the opportunity before the adoption and assert the results after the implementation.

An energy analysis plays an import role in developing an optimum HVAC and building CC methodology. High cost-effectiveness is always desired; advanced CC algorithms for energy savings can be evaluated through detailed measurements before and after retrofit implementation in real building energy systems. However, a non-intrusive metering takes effort and can be time consuming, especially in modern commercial buildings where the systems are becoming larger and more complex. Meanwhile, detailed measurements before upgrades are generally not available, so such a direct comparison with real data could be challenging or problematic.

Real buildings are seldom well-metered to clearly identify the cause-effect of the CC measures and are subject to many changing and non-repetitive excitements, from the indoor occupants' activities to the outdoor weather. CC typically includes a set of measures designed to improve the building energy performance. A clear-cut of interlacing implementation and evaluation is hardly practical.

One alternative and also supplementary method is to use building energy system modeling and simulation. Compared to a direct real time implementation and measurement, this method is much cheaper, safer, and can be controlled to discern the cause-effect of a single CC measure. Energy savings can be predicted with the model assisted by engineering analysis. The algorithm can be further improved if the preliminary results in a simulation are not as predicted.

From the deduction in the previous chapters, it has been shown that the IAHU control algorithm and saving quantity mainly rely on the synergetic relationship of heat gain, sensible and latent load between the interior and exterior zones. With the clear mathematic expressions of OA intake amount and allocation, a theoretical energy saving can be predicted for a given building and climate. Zone load information is needed to perform a reliable energy consumption calculation for different operation strategies.

In this section, building energy simulation methods are briefly reviewed to identify a practical approach for the study of IAHU.

4.1 Introduction

"Numerous building energy calculation procedures have emerged since the late 1960s. The methods range from simple degree day procedures to comprehensive program/ coding computations." (Knebel, 1983). The time step can decrease to minutes if the finest system dynamic response is desired for a study. However, a simulation with a small time step is very time costly, while the results might not necessarily be more accurate than a simplified method. On the other hand, the time constants of building energy systems are much greater than minutes.

In building energy system modeling, for most general purposes, hourly time series simulation is considered sufficient enough. The modeling methodology can be categorized into two types: a forward (classical) approach and data-driven (inverse) approach (ASHRAE Handbook, ch19, 2009). Hybrid utilization might be found to be more flexible and useful in building energy related applications.

Forward modeling of building energy use begins with a physical description of the building system or components of interest. The objective is to obtain the system performance and the output variables with known structures and parameters of the system when the system is subject to given variables. For further classification, forward modeling can be either dynamic or steady state. For dynamic forward modeling, fundamental engineering principles are employed to describe the dynamic response of the building, system and components of interest. Time series of the inputs and outputs are produced with the simulation. An advance control might also be imposed in the complete metrics of a building system.

The major dynamic simulation software including EnergyPlus, DOE-2, eQuest, Trnsys, etc, are based on this approach. Although procedures might vary from one to one in their degree of complexity, three common elements are normally involved: computation of space load, computation of secondary equipment load, and computation of primary equipment energy requirements. Since the model has been coded as software, users are mainly responsible for the detailed inputs and interpretation of final results. Only the given accessible fields in the model can be changed, therefore the flexibility of analysis is compromised and limited to general philosophy. As indicated in ASHRAE standard 90.1, energy calculations lead to an economic analysis to establish the cost-effectiveness of conservation measures. This approach is normally used to design and size HVAC systems and have begun to be used for modeling existing buildings after calibration (Haberl and Culp, 2005).

Forward modeling can also be steady state analysis-based. Unlike a dynamic forward method, a steady state forward model is usually more handy and simplified without necessarily losing accuracy. A complicated code or program package is not needed here. The thermodynamics that govern the system and components are simplified in a way that maintains the acceptable accuracy while removing the complexity. Typical steady state forward methods include simplified energy analysis using the modified bin method, the traditional ASHRAE bin method and change point models, etc. The cause and effect in the building system analysis is much easier and more flexible with this approach since users are usually responsible for the model and simulation set up. An innovative algorithm or operation for energy saving can be investigated with this method.

Data-driven modeling, sometimes termed inverse modeling, relies on known and measured input and output variables. The purpose is to determine a mathematical description of the system and to estimate system parameters. To develop an inverse model, one must assume a physical configuration of the building or system, and then identify the parameter of interest using statistical analysis (Rabl and Riahle, 1992). Once obtained and verified, the model is simple and useful to predict the system outputs in the future with a set of new inputs if the involved system characteristics are not changed.

Like forward modeling, data-driven modeling can be also either dynamic or steady state. Advanced methodology is used to develop a dynamic data-driven model. It is capable of capturing dynamic effects such as thermal mass which traditionally needs first-principles with differential equations. But typically to gain the benefit of this approach, a large amount of data and detailed measurements are needed to train or tune the model, which is a gray or black box to the user. The models are usually complex for applications. The examples are equivalent thermal network analysis, ARMA models, Fourier series models, machine learning and artificial neural networks, etc.

A steady state data-driven model is simple and has limited applications. It can be single-variate, multivariate, polynomial, or physical. Monthly utility billing data and OA temperatures are generally used for such a regression. Several more variables, like solar radiation, OA humidity, occupancy (as dummy variables) etc., may be used to better capture the system characteristics, but they add a need for reliable inputs. A steady state data-driven model is insensitive to dynamic effects and is used mainly with monthly or daily data.

Data driven inverse modeling requires reliable data before and after the implementation of a retrofit in order to evaluate its performance, and for future performance prediction. Hard coded forward models have effective dynamic simulation performance but low flexibility. The internal mechanisms are predefined and modularized. The simulation cannot be performed when the conventional system characteristics are changed.

In an IAHU operation, the original TAHU system characteristic is changed since the air flow in one zone is manipulated into another zone. A hybrid steady state model is one option to evaluate the performance of an IAHU system in a given building.

4.2 Evaluation Methodology

Building energy simulation methods are briefly reviewed above. Detailed forward dynamic modeling is stiff and cumbersome without necessarily providing us more accuracy if the inputs are not sufficiently accurate. A data-driven model is not suitable for evaluating an innovative algorithm before its implementation. A simplified hourly steady state modeling method is adopted here to perform a handy but reliable evaluation of IAHU in a commercial building.

4.2.1 System simplification

Many researchers have pointed out that a large commercial building can be modeled as two zones: an internal one and a perimeter one. Knebel (1983) presented the idea to simplify energy analysis using a modified BIN method. Katipamula and Claridge (1993) modeled a commercial building as two zones and considered it adequate for large commercial building load simulation. Liu and Song, et al, (2004) applied a similar idea to a simplified building modeling and asserted it could provide good hourly average thermal load. Liu and Claridge (1995, 1998) showed very accurate results by using the simplified zoning in a case study. The studies support the claim that a two-zone model works well with properly defined interior and exterior zones for large commercial buildings.

The rules for simplification and simulation are followed to support the evaluation. Correspondingly, the zone HVAC systems are also modeled as two systems for the central AHUs and the terminals. It follows the previous deduction for IAHU if there are more than two AHUs serving one zone. In a real application, the calculated proper OA flow rate in an IAHU system can just be distributed among the interior zone AHUs and the exterior AHUs. The physics and deduction of IAHU will not change.

The AHUs serving zones with similar features typically have identical system configurations and operation strategies. An IAHU operation does not change the similarity of AHUs serving the same zone. For example, there are either all VAVs or all CAVs for the interior or exterior zone respectively. The OA intake is specified with the same algorithm and ratio. This is also true for the free cooling economizer control.

The zone occupancy, peak load and load profile for individual AHU conditioned space in one of the two zones are similar. The consolidated AHUs' airflow rate is the sum flow rate of each individual AHU. The cold and hot deck set points can be determined using an airflow weighted average value if the individual AHUs are different sizes.

4.2.2 Inputs and variables

Inputs are needed to drive the model and simulation. In addition to the inputs and variables defined in table 3-1, which are mainly for the system side, inputs for the building load are also needed. Some parameters, e.g. building geometry, location, CLF, that are involved to conduct a simulation, are not listed to maintain the simplicity. The parameters can be found in ASHRAE handbooks.

Table 4-1: Inputs and variables for simulation

Item	Symbol	Definition	Unit
Building			
1	A_F	Building conditioned floor area	Ft^2 (m^2)
	$A_{F,in}$	Conditioned interior area	
	$A_{F,ex}$	Conditioned exterior area	
2	U	Heat transfer coefficient for walls and roof	Btu/hr F Ft^2 (W/m^2 K)
3	UG	Heat transfer coefficient for windows	Btu/hr F Ft^2 (W/m^2 K)
4	AG	Window area	Ft^2 (m^2)
5	A	Walls and roof area	Ft^2 (m^2)
6	q_{ltg}	Maximum lighting power density	W/ Ft^2 (W/m^2)
7	q_{eqt}	Maximum equipment power density	W/ Ft^2 (W/m^2)
8	$q_{pe,s}$	Maximum sensible load density from occupants	W/ Ft^2 (W/m^2)
9	$q_{pe,l}$	Maximum latent load density from occupants	W/ Ft^2 (W/m^2)
10	ACH_{inf}	Infiltration air change rate	/hr
11	V	Zone volume	Ft^3 (m^3)
System			
Refer to Table 3-1			

The inputs listed in the table for building load can be determined from design drawings, air balance reports, or building automation systems.

4.2.3 Load simulations

A building zone thermal load includes the external heat gain and the internal heat gain. Windows and opaque surfaces are the interfaces between the internal environment and the external environment. The simulation can be divided into the following parts:

Solar radiation through glass:

In the reviewed literature, the seasonal variation of the solar load is considered to be a constant or more accurately approximated by a linear relationship with the OA temperature. The linearization is obtained by associating the January solar contribution to the winter design OA temperature and the July solar contribution to the summer design OA temperature. The following equations are used to calculate the window glass solar heat gain in January and July:

$$Q_{sol,jan} = \frac{\sum_{i=1}^{N} MSHGF_i \times AG_{i,adj} \times SC_i \times CLFTOT_i \times FPS_{jan}}{24 \times A_F} \tag{4-1}$$

$$Q_{sol,jul} = \frac{\sum_{i=1}^{N} MSHGF_i \times AG_{i,adj} \times SC_i \times CLFTOT_i \times FPS_{jul}}{t_{run} \times A_F} \tag{4-2}$$

MSHGF is the maximum solar heat gain factor for orientation i for the corresponding month at the latitude in literature [2]. The glass total cooling load factor, CLFTOT, and fraction of possible sunshine, FPS, are given in tables 3-2 and 3-1 in literature [1].

The slope is determined then by:

$$M_{sol} = \frac{(Q_{sol,jul} - Q_{sol,jan})}{(T_{pc} - T_{ph})} \tag{4-3}$$

The solar radiation heat gain through windows can be simulated by using the linear relationship:

$$Q_{sol} = M_{sol} \times \left(T_{oa} - T_{ph}\right) + Q_{sol,jan} \tag{4-4}$$

External shading, such as overhangs and fins, is also considered to improve the accuracy of solar radiation calculation. Table 3-5 through 3-9 in literature [1] can be referred to in determining the solar load factor (SLF) and adjusting the window area. $AG_{i,adjn}$ are noted as north facing windows.

$$AG_{i,adj} = AG_i \times SLF_i \tag{4-5}$$

$$AG_{i,adjn} = AG_i \times (1 - SLF_i) \tag{4-6}$$

Transmission load:

The transmission load comes from the windows, the roof and the external walls. The contribution through glass is given as:

$$Q_{T,gl} = \frac{\sum_i^N (AG_i \times UG_i)(T_{oa} - T_r)}{A_{F,i}} \tag{4-7}$$

The air to air temperature difference causes heat transfer through the opaque surfaces.

$$Q_{T,wl} = \frac{\sum_i^N UA_i(T_{oa} - T_r)}{A_{F,i}} \tag{4-8}$$

In addition, the external surface temperature increase due to solar effects should be considered to avoid the underestimation of the transmission load. A similar OA temperature linearization is applied to account for the effect.

$$Q_{ts,jul} = \frac{\sum_{i=1}^N A_i U_i \times CLTDS_{jul} \times K \times FPS_{jul}}{A_F} \tag{4-9}$$

$$Q_{ts,jan} = \frac{\sum_{i=1}^N A_i U_i \times CLTDS_{jan} \times K \times FPS_{jan}}{A_F} \tag{4-10}$$

$$M_{ts} = \frac{(Q_{ts,jul} - Q_{ts,jan})}{(T_{pc} - T_{ph})} \tag{4-11}$$

$$Q_{ts} = M_{ts} \times \left(T_{oa} - T_{ph}\right) + Q_{ts,jan} \tag{4-12}$$

Cooling load temperature differences, CLTD, are provided in the ASHRAE handbook (2009).

Infiltration load:

Depending on the building operation, the infiltration might be considered for both the exterior and the interior zone. It serves as a direct sensible and latent load source when the OA enters the building. The expression can be:

$$Q_{inf,s} = ACH_{inf} \times V \times C_p \times (T_r - T_{oa})\rho \tag{4-13}$$

$$Q_{inf,l} = ACH_{inf} \times V \times h_{fg} \times (w_r - w_o)\rho \tag{4-14}$$

For good mechanical ventilated commercial office building, a positive pressure is usually controlled. Under this circumstance, the infiltration rate, ACH, is assigned as 0.

Internal load:

The internal load constitutes a significant proportion of the total loads in commercial office buildings because of the increasing density of electrical equipment, e.g. printers, computers, lighting. Cooling load factors, CLF, which can be found in the ASHRAE handbook (2009), should be used to account for the fact that all the internal heat gains do not appear as cooling loads instantaneously.

$$Q_{ltg} = A \times q_{ltg} \times CLF \tag{4-15}$$

$$Q_{eqt} = A \times q_{eqt} \times CLF \tag{4-16}$$

$$Q_{pe,s} = A \times q_{pe,s} \times CLF \tag{4-17}$$

$$Q_{pe,l} = A \times q_{pe,l} \times CLF \tag{4-18}$$

Occupants also contribute latent load into the zone. Liu, et al, discussed the process of introducing discount factors of rated power, diversity factors and load discount factors if a calibration evaluation is desired (Liu and Song, 2004).

With the expressions for the individual sectors, the total zone load for the internal and the external zone can be simulated in a program:

$$Q = Q_{sol} + Q_{ts} + Q_{T,gl} + Q_{T,wl} + Q_{ltg} + Q_{eqt} + Q_{pe,s} \tag{4-19}$$

$$w = Q_{pe,l}/h_{fg} \tag{4-20}$$

4.2.4 System simulations

For air-based commercial office buildings, the components are illustrated in the deduction of IAHU for energy considerations in chapter 2. The main thermal components are heating coils and cooling coils. With different AHU configurations, the simulation might vary slightly due to the different constraints imposed and the different corresponding operations in the system level simulation.

For instance, in a CAV, the airflow rate is a constant constraint to the simulation. For a VAV, the minimum airflow rate is a constraint. The control flow chart in chapter 3 is the main part of the simulation for an IAHU operation that determines the energy consumption of a system. To evaluate the energy saving of IAHU, the original TAHU system configuration and operation should be simulated for a comparison.

In a system, mixing, heating and cooling (sensible and latent) are the basic thermal processes for both TAHU and IAHU, as depicted in Fig 2-12.

The mixing air status can be simulated by using:

$$T_{mix} = T_{oa}\beta + (1 - \beta)T_r \tag{4-21}$$

$$h_{mix} = h_{oa}\beta + (1 - \beta)h_r \tag{4-22}$$

$$w_{mix} = w_{oa}\beta + (1-\beta)w_r \tag{4-23}$$

The heating coil energy consumption is simulated as:

$$E_{hc} = 60 \times m \times C_p \times (T_r - T_{mix}) \tag{4-24}$$

The cooling energy consumption of the main AHU is:

$$E_{cc} = 60 \times m \times (h_{mix} - h_c) \tag{4-25}$$

The reheat coil is simulated as:

$$E_{rh} = 60 \times m \times C_p \times (T_s - T_c) \tag{4-26}$$

The zoom air humidity is determined as a steady state:

$$w_r = \frac{w + \frac{m}{\rho}w_c + ACH_{inf}Vw_{oa}}{\frac{m}{\rho} + ACH_{inf}V} \tag{4-27}$$

The return air humidity w_r and the supply air humidity w_c are coupled when the coil is dry. Under this circumstance, the supply air humidity equals the mix air humidity. An iteration with equation (4-23) and (4-27) is needed, or the following equation (4-28) can be used, to solve for the room air humidity.

$$w_r = \frac{w_{oa}\frac{m}{\rho}\beta + w + ACH_{inf}Vw_{oa}}{\frac{m}{\rho}\beta + ACH_{inf}V} \tag{4-28}$$

In some conventional systems, there are induction units deployed along the external walls. The induction coil energy consumption is simulated as:

$$E_{id} = 60 \times n \times m \times (h_{r2} - h_r) \tag{4-29}$$

Additional governing equations are needed to obtain the room air humidity for this special system layout.

In winter, both coils are in a heating mode, the humidity ratio does not change across the coils, and equation (4-28) is applicable. In summer, either the primary cooling coil or the secondary cooling coil can be dry or wet. The return air humidity w_r, the supply air humidity w_c and the induction air humidity w_{id} are coupled under this circumstance. After the induction unit, the room air humidity is expressed as:

$$w_r = \frac{w_{id} n \frac{m}{\rho} + \frac{m}{\rho} w_c + w + ACH_{inf} V w_{oa}}{\frac{m}{\rho} + n \frac{m}{\rho} + ACH_{inf} V} \tag{4-30}$$

If the primary coil is dry, it needs to be reevaluated as:

$$w_r = \frac{w_{id} n \frac{m}{\rho} + \frac{m}{\rho} \beta + w + ACH_{inf} V w_{oa}}{\frac{m}{\rho} \beta + n \frac{m}{\rho} + ACH_{inf} V} \tag{4-31}$$

If the induction coil is wet, equation (4-31) is enough to obtain the room air humidity. If the induction coil is also dry, equation (4-28) is utilized to evaluate the room air humidity. When either coil is supposed to provide heating to the space, there is no dehumidification.

After obtaining the room air humidity, equation (4-23) should be reevaluated to solve for the mixed air humidity.

To this end, a steady-state air-based system simulation structure for building demand side energy consumption (to the secondary system) is presented for performance evaluation. An hourly performance and the system states, e.g. the various air humidities, can be predicted with the simulation.

4.2.5 Simulation procedure

With the governing equations for the building and components, a supervisory control is needed to direct the operation and performance. The rules and algorithm given in Table 2-1 and the flow charts in Chapter 3 for the corresponding system configuration, should be incorporated into the model to accomplish the simulation. For evaluation purposes, a baseline with the existing conventional TAHU should also be simulated. The difference between the two will be the energy saving potential that might be achieved by introducing an IAHU operation. The analysis can be utilized to direct the implementation and verify it in a CC® process.

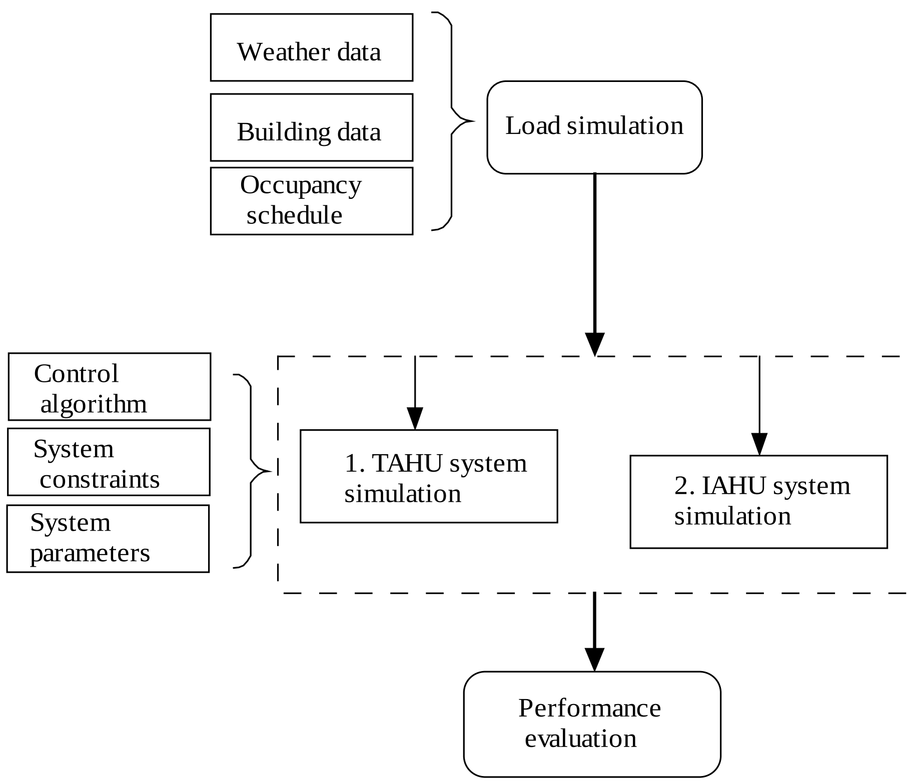


Figure 4-1: IAHU evaluation procedure based on simulation

The simulation procedure for the evaluation can be depicted as in Fig 4-1.

The weather data, building data, as well as the occupancy schedule are fed into the simulation engine to predict the building load. It is assumed that the room air is well mixed and the physical condition is satisfied through the system's dynamic PI control loops. Afterward, the TAHU and IAHU system simulation with different control algorithms, system constraints, and system parameters imposed is used to determine the system energy consumption with the same user side demand. The result is then compared and organized for performance evaluation.

4.3 Evaluation of IAHU in a case building

A case building located in Omaha, Nebraska is selected to demonstrate the IAHU operation and its performance.

4.3.1 Building and system

The selected building is a government office building. From the 3rd floor to the 12th floor is the office area is identical in layout. The long side of the building faces the west and the east. Tinted double layer glazes are installed with aluminum frames along the external walls. Internal blinds are also equipped for shading.

Outside view from south east **Inside view of an office**

Figure 4-2: Pictures of the case building

The office area is occupied by about 620 people, and operated an average 60 hours per week. The occupants' density is about 0.0033 /ft^2. The space is conditioned by three AHUs. The main heating and cooling sources are district supplied chilled water and steam. A building automation system from INET is deployed in the building and is available for customers to program the control of the main equipment.

A single duct VAV system, with 60 electrical reheat equipped terminal boxes, serves the interior zone. Two constant air systems, with 495 induction terminals each, are in charge of the perimeter space along the external wall. One is for the east and north sides, and the other one is for the west and south sides. The layout is a typical TAHU and the system consolidation can be applied. All supply and return fans are equipped with VFDs. Four constant fans are installed to exhaust air from the rest rooms.

The room air is returned via the ceiling plenum without ductwork through evenly deployed slot diffusers. The light panels have either three slots for return in the internal zone or four slots for return in the external zone.

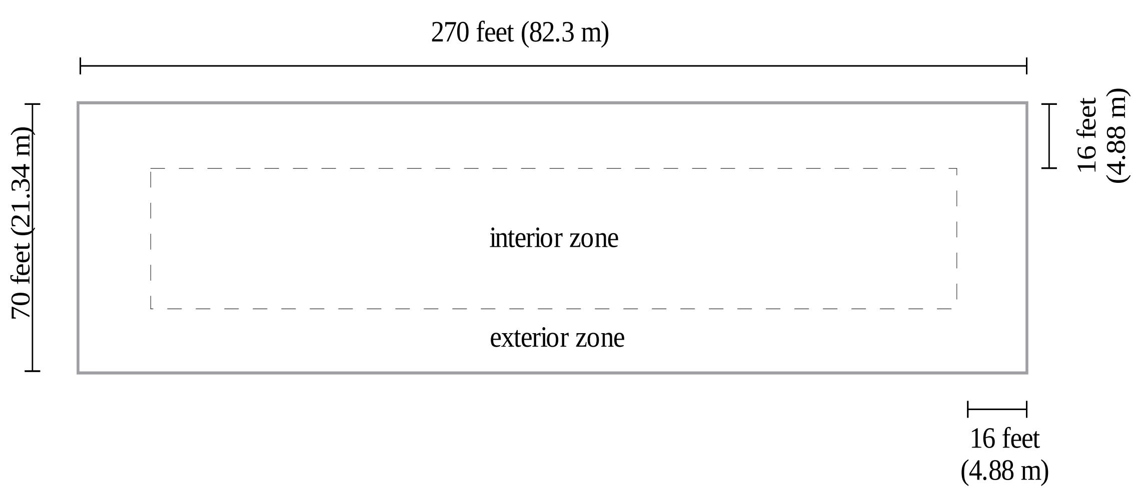


Figure 4-3: Typical floor layout of the office building

Figure 4-3 illustrates the space layout of the office building. The floor area is 18900 ft^2 per floor. The conditioned air in the internal zone is distributed via slot diffusers along one side of the lighting panel. The density of the panels is even across the space except for the elevator, stairs and rest room area.

4.3.2 Control and operation as TAHU

The existing TAHU has normal, conventional air system algorithms as below:

System ON/OFF:

Both systems have a warm up/cool down control for the average season. Facility engineers start the system at 4:00 am and shut it down at 10:00 pm from Monday to Friday in normal operation. During the hot summer and cold winter, the system is kept on constantly 24/7.

Supply air temperature:

AHU1 is a year around cooling only system serving the interior zone. The discharge air temperature is set at 55ºF (adjustable). For AHU2-3, the discharge air temperature is set at 55ºF (adjustable) when the OAT is greater than 70ºF, 75ºF when the OAT is less than 55ºF. A linear reset is applied when the OAT is in between.

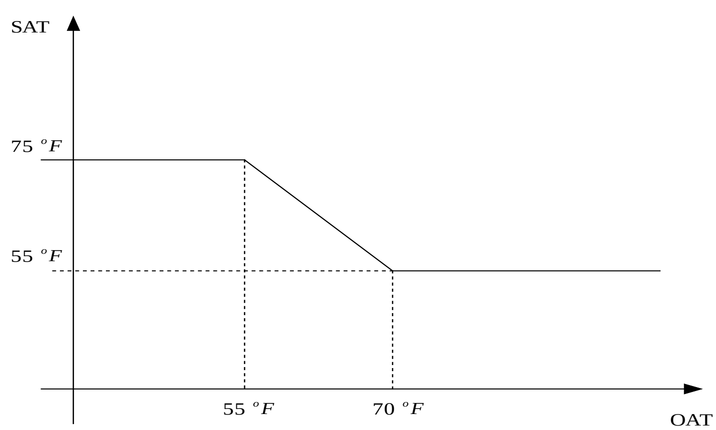


Figure 4-4: Supply air temperature for AHU2-3

Outdoor air intake:

The systems satisfy the minimum OA intake requirement in summer and winter via the individual AHUs.

Economizer:

An air temperature based economizer is used for the systems. It is enabled when the OA temperature is below 70ºF. The return air dampers and the OA dampers are controlled to maintain the mixed air temperature set point 55ºF (adjustable), provided the minimum OA intake is satisfied.

Operation:

The TAHU operation regarding to the OA intake can be summarized in the following table.

Table 4-2: TAHU operation of OA intake

No.	T_{oa} Condition	Interior OA (AHU1)	Exterior OA (AHU)
A	$T_{oa} \leq T_{i,c}$	$\beta_i = \max(\beta_{i,IAQ,d}, \beta_{i,eco})$	$\beta_e = \beta_{e,IAQ,d}$
B	$(T_{i,c}, T_{e,c}]$	$\beta_i = 1$	$\beta_e = \max(\beta_{e,IAQ,d}, \beta_{e,eco})$
C	$(T_{e,c}, T_r]$,	$\beta_i = 1$	$\beta_e = 1$
E	$T_r < T_{oa}$	$\beta_i = \beta_{i,IAQ}$	$\beta_e = \beta_{e,IAQ,d}$

4.3.3 Outdoor information

The TMY3 weather data from the Department of Energy for EnergyPlus is used for the simulation. Omaha, in climate zone 5A, has cold winters and hot summers. The hottest temperature reaches above 100ºF while the lowest decreases to -11ºF in the data file. The long winter time and occasionally mild weather in swing season and summer provide opportunities for IAHU to exert its energy saving capability.

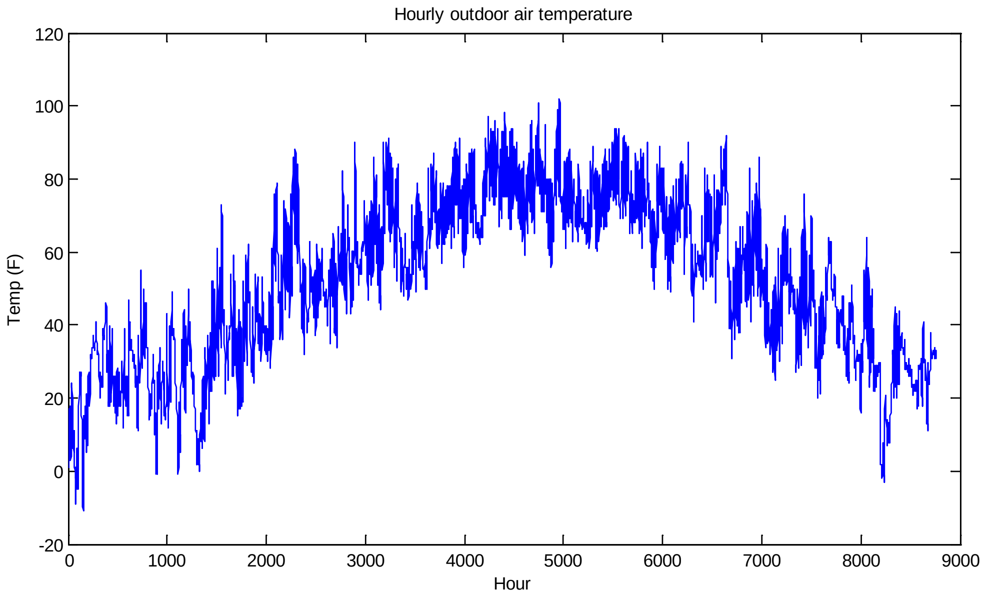


Figure 4-5: Hourly outdoor air temperature in Omaha

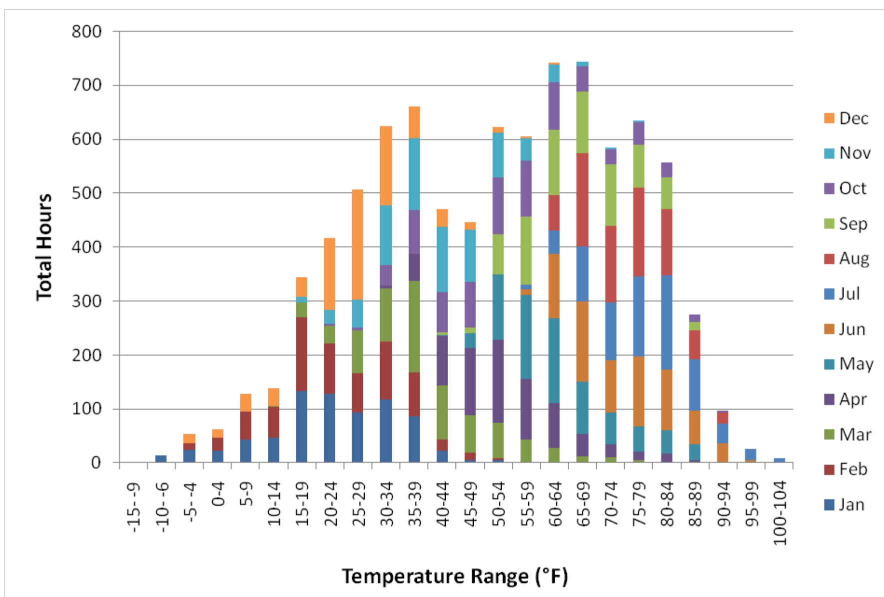


Figure 4-6: Outdoor air temperature BIN in Omaha

The corresponding hours for the location are sorted according to the IAHU operation scenarios. The results are collected in the table below:

Table 4-3: OA condition hours during occupied time (T_r set as 75ºF)

No.	OA Condition	Hours
A	$T_{oa} \leq T_{i,c}$	1508

B	$(T_{i,c}, T_{e,c}]$	224	
C	$(T_{e,c}, T_r]$, $h_{o,a} < h_r$	169	
D	$(T_{e,c}, T_r]$, $h_{o,a} \geq h_r$	488	448, mild weather
E	$T_r < T_{oa}$	896	

4.3.4 Simulation inputs and process

The building and system information are collected from the building drawings, onsite observation and control documentations. The designed indoor air temperature is 75ºF in summer and 72ºF in winter. The system design air flow rate for AHU1 is 90696 cfm and 54000 cfm for AHU2-3 respectively. The total OA intake is set based on the ASHRAE standard: 5*620 + 180000*0.06 = 13900 cfm, which is 16.7% of the total designed air flow rate. Correspondingly, AHU1 requires 6636.5 CFM (7.5%) and AHU2-3 requires 7263.5 CFM (14%) of OA.

The main simulation inputs of the building features are collected below:

Table 4-4: Main inputs of the building

Symbol	Info		
Building conditioned floor area	180000 Ft^2 (16722.5 m^2)		
	Interior zone	85940 Ft^2 (8318.5 m^2)	
	Exterior zone	94060 Ft^2 (8738.4 m^2)	
External Wall	East	18337 Ft^2 (1703.6 m^2)	0.148 Btu/hr F Ft^2 (0.842 W/m^2 K)
	West	24423 Ft^2 (2269 m^2)	
	North	30000 Ft^2 (2787 m^2)	
	South	30000 Ft^2 (2787 m^2)	
Windows	East	17550 Ft^2 (1630 m^2)	0.633 Btu/hr F Ft^2 (3.6 W/m^2 K)
	West	12480 Ft^2 (1160 m^2)	
	North	2600 Ft^2 (242 m^2)	
	South	2600 Ft^2 (242 m^2)	
Roof	18000 Ft^2 (1672.3 m^2)		0.099 Btu/hr F Ft^2 (0.559 W/m^2 K)
Maximum lighting power density	1.45 W/Ft^2 (15.61 W/m^2)		

Maximum equipment power density	1 W/Ft^2 (10.78 W/m^2)	
Maximum occupants density	0.0034 / Ft^2	
Infiltration air change rate	0 ACH	
Minimum airflow rate ratio	AHU1: 30%	
	AHU2-3: 100%	

The simulation is conducted in Matlab. The functions are coded as a main program, an internal load simulation, an external load simulation, two mechanical system simulations, and several supportive functions. The subroutines for IAHU decision making are also individually coded for better readability. The system states for the air temperatures and air humidity are simulated hourly. The codes are included in Appendix A. Other supportive materials, including drawings, pictures, etc, are provided in Appendix B.

4.3.5 Results and analysis

The hourly simulation is performed for the normal TAHU operation and the improved IAHU operation with the process and parameters provided in the above sections. Part of the simulation results for the discharge air humidity, room air humidity, etc are collected in appendix B for reference.

The savings are divided into four groups corresponding to the IAHU operation A, B, C and D/E mild weather. From Table 4-3, we might roughly weigh the IAHU saving potential. The descending sequence is A, E, D, B and C in terms of the duration. Since an IAHU operation does not have a retrofit cost, the adoption of the improved operation is of high flexibility. In Omaha, NE, in conditions A and D, IAHU should be first considered in CC® to achieve energy savings.

The simulated annual heating and cooling energy savings are collected in the table below.

Table 4-5: Energy saving performance of IAHU in a case building, x=0.4

OA conditions	Total Heating (MMBTU)	Total Cooling (MMBTU)	Savings (MMBTU)	Percentage (%)	Normalized savings (BTU/ft^2 yr)
A	1566.7	34.6	368.9	23	2049
B	32.4	53.8	70.4	82	391
C	95.1	615	32.3	5	179
D/E	89.8	1981	162.6	8	903
Total				**14**	**3523**

From the table, it can be seen that the absolute energy savings mainly come from OA conditions A and D/E for the case building. For heating seasons, applying the IAHU operation saves about 2049BTU/ft^2 yr, equivalent to 23% of the overall heating energy consumption. Considering that heating is usually provided by steam/hot water, or electricity, the saving percentage is significant. The next biggest savings comes from the thermal load decoupling in mild weather condition, where 163 MMBTU of thermal energy is saved. The normalized overall annual energy savings of using IAHU in this building is 3387 BTU/ft^2. The two together contribute nearly 80% of savings. In total, IAHU operation saves 14% of thermal energy for heating and cooling the building.

To compare the difference between an IAHU system and an OAHU system, a set of parametric runs is conducted using different circulation ratio γ. Since it must fall in the range between 0 and β_i, the energy saving difference between the two AHUs is subject to the constraints illustrated in Figure 3-5, from the OA intake ratio and airflow ratio between the two individual AHUs. Using a duct work connection in an OAHU might not necessarily yield greater savings once the savings is already fulfilled with a smaller value of circulation air ratio. With the same other simulation parameters, the difference of energy savings are plotted in the figure below.

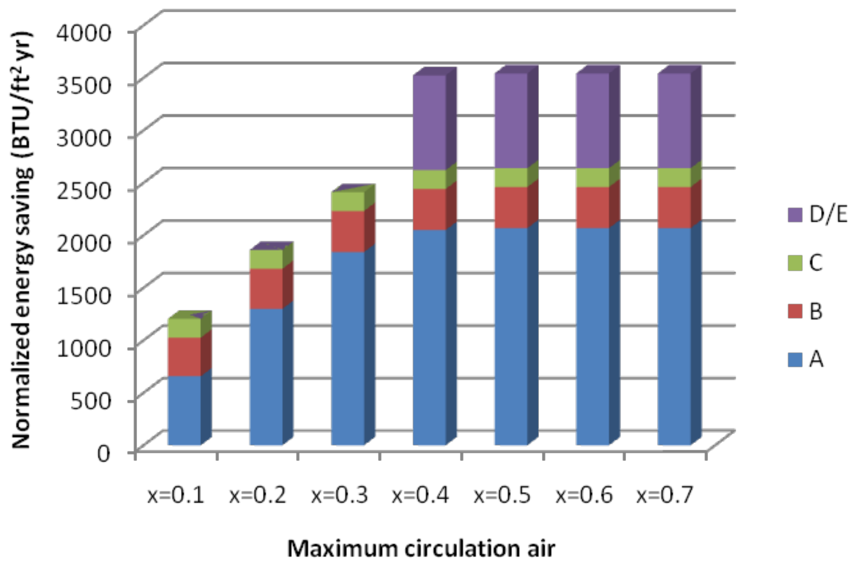


Figure 4-7: Normalized energy saving of using IAHU for different circulation ratio x

When the two zones are in different air conditioning modes, the OA intake from the interior zone has a positive effect on the system for heat transferring. The OA intake is warmed up by going through the interior zone. The energy savings acquired by using IAHU in cold winter (condition A) is given in Table 4-5 and Fig 4-7. With an increasing circulation air ratio *x* from 0.1 to 0.4, the overall energy savings of warming up the OA intake also increases until the OA is fully satisfied through the dedicated interior zone AHU.

It is worthy to mention that additional heat can be transferred from the interior zone to the exterior zone, provided the interior zone air temperature is higher than that of the exterior zone. This is stated in Chapter 1as one of the IAHU’s traits. The amount of the additional energy savings is a function of the room air temperature difference and the amount of circulation air between the two zones. Although a large value should be avoided for comfort consideration, a reasonable difference is recommended to not only bring in more energy savings in IAHU, but also avoid unnecessary zone fighting between the two zones that are in opposite modes.

As also shown in the simulation results, the sensible and latent load decoupling feature has a dependency on the circulation air percentage in IAHU. The maximum is around 40% for this selected building. If the ratio limit is set lower than this value, the interior zone has to introduce more OA in summer to satisfy the IAQ requirement of the exterior zone. Since there is an associated penalty of taking in more OA in summer, IAHU should be run as TAHU under this limit condition. The overall energy savings by utilizing IAHU finally saturates when the re-circulated ratio is about 40%. Allowing more air recirculation does not yield additional savings for x equal to or above 0.5 as given in Fig 4-7.

Chapter 5 CONCLUSION AND DISCUSSION

Modern office buildings have higher internal heat gains compared to decades ago, as more and more electricity-powered equipment enters office buildings, which results in an increase in both the quantity and density. Office building heating, ventilation and air-conditioning systems, mainly air based, have evolved from single air handling unit system (SAHU) to two air handling unit system (TAHU), constant air volume system (CAV) to variable air volume system (VAV) for better energy performance. However, the air handling units (AHUs) in a building are mainly operated individually for designated zones.

This study investigated the thermal load features in office buildings and proposed an Integrated Air Handling Unit (IAHU) concept for an integrated operation in order to achieve benefits with little to zero retrofitting. The internal heat gain can be transferred from an interior zone into an exterior zone in winter, the sensible load and latent load can also be decoupled in mild weather. With no additional duct remodeling, the integrated operation is much easier to be accepted by customers for generalization than Office Air Handling Unit (OAHU).

In this study, the deduction of IAHU is theoretically conducted for an acceptable indoor air quality (IAQ) and better energy performance. Five scenarios based on the outdoor air conditions are defined for an annual operation. The system variables and constraints are investigated in detail to comprehend the feasibility and operability of IAHU. The implementation methods and issues of airflow rate information are addressed based on Fan Airflow Station (FAS) developed by Energy Systems Lab (ESL) at the

University of Nebraska-Lincoln (UNL). The control flow charts are then provided in Chapter 3 for different AHU combinations. IAHU can be easily integrated into Continuous Commissioning® measures to achieve energy savings in most existing and new office buildings.

To evaluate the performance of IAHU, a case building in Omaha is chosen as the simulation base. An hourly steady state modeling method is considered suitable for the evaluation of the innovative algorithm. The corresponding simulation method and procedure for the building and system are then elaborated upon. The simulation is later conducted in Matlab.

The simulation results demonstrate that converting the existing TAHU system into an IAHU system can achieve about 14% thermal energy savings for the case building. If normalized, about 3.5 MBTU/ft^2 yr can be saved for this case building in the given climate. By transferring the internal heat gain from the interior region to the exterior region, 58% of the total savings, by applying IAHU, can be achieved in winter and 17% achieved in swing seasons. Another 25% savings comes from the sensible and latent load decoupling with IAHU in summer mild weather.

In the previous chapters, IAHU is presented as a system consisting of two AHUs to facilitate the theory deduction and interpretation. The implementation of IAHU in a real office building might have some practical challenges. For example, the plenum above the ceiling is blocked by fire walls between zones. Under these conditions, the building cannot be just simplified into two consolidated zones. IAHU theory should be adopted in a smaller scale among the AHUs. Additional analysis might be necessary to evaluate the utilization of IAHU.

There are several topics that can be conducted as the future work:

1. The algorithm can be implemented in the case building for performance verification. Since the savings are categorized accordingly, IAHU can be performed step by step with proper tuning.
2. An IAHU simulation based control in a real building can be further investigated with a tool, e.g. the building control virtual test bed (BCVTB) provided by Lawrence Berkley National Lab.
3. Research on automated IAHU logic generation, diagnostics and troubleshooting for different building systems can be performed.

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Appendix A:

```
% this is the main function to calculate the load and energy
consumption
% for TAHU and IAHU system
clear all;
originalData=importdata('Omaha_Weather.csv');
%originalData=importdata('Pittsburgh_Weather.csv');
%% control parameters
W2BTUhr=3.412;Tic=55;summerPoint=55;CFMRatio_in=0.30;CFMRatio_out=1;
CFM_in_design=90696;% design airflow rate for the interior zone
CFM_ex_design=54000;% design airflow rate for the exterior zone
Ti_sm=75;Ti_in_wt=75;Ti_ex_wt=75;
% yearly indoor air temperature for interior zone and the exterior zone
AirCapRou60=1.06; % 60*rou*Cp: Btu min/ft^3 F hr . Energy Btu/hr=
AirCap60*CFM*delt
SAT_in_summer=55;SAT_in_winter=60;
SAT_in=ones(8760,1).*SAT_in_summer;
SAT_ex_summer=55;SAT_ex_winter=Ti_ex_wt;
Ti_in= (originalData.data(:,4)> summerPoint).*Ti_sm +
(originalData.data(:,4)<= summerPoint).*Ti_in_wt;
Ti_ex= (originalData.data(:,4)> summerPoint).*Ti_sm +
(originalData.data(:,4)<= summerPoint).*Ti_ex_wt;
SAT_ex=(originalData.data(:,4)>70).*SAT_ex_summer+...
    (originalData.data(:,4)<
55).*SAT_ex_winter+(originalData.data(:,4)<=70).*...
    (originalData.data(:,4)>= 55).*round(148-
(4/3).*originalData.data(:,4));

OAinIAQ=6636.5; % 12%
OAexIAQd=7263.5; % 27%
AirDensity=0.0736; % lbm/cfm
Ocu_Schedule=[zeros(7,1);ones(9,1);zeros(8,1)]; % occupancy
scheduleones(24,1);%
Ocu_Density=[zeros(7,1);ones(9,1);zeros(8,1)]; % occupancy density
OAinIAQ=OAinIAQ.*Ocu_Density;
OAexIAQd=OAexIAQd.*Ocu_Density;
Tec=60;Trc=75;

%% this is the explanation of the functions
% Env_Load_Coeffs.glass=[M_Glass_sol Qsol_Jan_PerArea];
% M_Glass_sol: W/sqf K, Qsol_Jan_PerArea: W/sqf:
% Env_Load_Coeffs.int=[Qt_Int M_Int_Sol Qts_IntRoof_Jan];
% Env_Load_Coeffs.ext=[Qt_Ext M_Ext_Sol Qts_Ext_Jan];
% M_Glass_sol*(Weather_Data(:,4)-Tphc)+Qsol_Jan_PerArea;
OAinIAQY=[];OAexIAQdY=[];
for i=1:365
    OAinIAQY=[OAinIAQY;OAinIAQ];
    OAexIAQdY=[OAexIAQdY;OAexIAQd];
end

%% get the basic equation coefficients
Tpc=100;%92
```

```
Tph=30;% outside design temperature
Bld_Width=70;Bld_Length=270;%ft
Side_Width=16; %ft
Ele_Area=900;
Floor_Area=Bld_Length*Bld_Width-Ele_Area; %sft
Int_Area=(Bld_Width-(2*Side_Width))*(Bld_Length-(2*Side_Width))-
Ele_Area/2;% sft
Ext_Area=Floor_Area-Int_Area; % sft
Area=[Floor_Area Int_Area Ext_Area]; % area of each floor to calcuate
the load/ sft
interLoad=intLoad(Area,Ocu_Density'); % 24 hours repeat pattern, W/sft
EnvCoeff=Envelop(Area);
TotalArea= Area*10; % 10 story, sft
%% data interpretation
%Weather_Data column 1: month (1-12)
%Weather_Data column 2: day (1 to 31, change may happen)
%Weather_Data column 3: hours (1 to 24)
%Weather_Data column 4: dry bulb (F)
%Weather_Data column 5: wet bulb (F)
%Weather_Data column 6: dewpoint (F)
%Weather_Data column 7: Relative humidity ration (%)
%Weather_Data column 8: HumidityRatio (lbmw/lbma)
%Weather_Data column 9: Solar global horizontal (W/m2)
%Weather_Data column 10: Solar direct norm (W/m2)
%Weather_Data column 11: Wind speed (MPH)
%Weather_Data column 12: Wind direction (degree)
% LoadData column 1-3: month, day and hours
% LoadData column 4: Glass solar load, exterior zone W
% LoadData column 5: Wall transmission load, exterior zone W
% LoadData column 6: All envelope (Wall, column, window and roof all
included) transmission load due to solar effect, exterior zone W
% LoadData column 7: Internal sensible load, exterior zone W
% LoadData column 8: Exterior zone sensible load summary, W
% LoadData column 9: roof transmission load, interior zone W
% LoadData column 10: Solar load through roof, interior zone W
% LoadData column 11: Internal sensible load, interior zone W
% LoadData column 12: Interior zone load summary, W
% LoadData column 13: Exterior zone latent load, W
% LoadData column 14: Interior zone latent load, W

%% basic load simulation, in W
LoadData=originalData.data(:,1:3);
LoadData(:,4)=((originalData.data(:,4)-
Tph)./1.8.*EnvCoeff.glass(1)+EnvCoeff.glass(2)).*TotalArea(3); % solar
glass
LoadData(:,5)=(originalData.data(:,4)-
Ti_ex)./1.8.*EnvCoeff.ext(1).*TotalArea(3); % transmission
LoadData(:,6)=((originalData.data(:,4)-
Tph)./1.8.*EnvCoeff.ext(2)+EnvCoeff.ext(3)).*TotalArea(3); % opaque
solar
inload_ExtZoneSen=[];% repeat 365 times for the same schedule of
internal load
for i=1:365

inload_ExtZoneSen=[inload_ExtZoneSen;interLoad.ext(:,2).*TotalArea(3)];
end
LoadData(:,7)=inload_ExtZoneSen;
```

```
LoadData(:,8)=LoadData(:,4)+LoadData(:,5)+LoadData(:,6)+LoadData(:,7);  %
W

LoadData(:,9)=(originalData.data(:,4)-
Ti_in)./1.8.*EnvCoeff.int(1).*TotalArea(2);
LoadData(:,10)=((originalData.data(:,4)-
Tph)./1.8.*EnvCoeff.int(2)+EnvCoeff.ext(3)).*TotalArea(2);
inload_IntZoneSen=[];
for i=1:365

inload_IntZoneSen=[inload_IntZoneSen;interLoad.int(:,2).*TotalArea(2)];
end
LoadData(:,11)=inload_IntZoneSen;
LoadData(:,12)=LoadData(:,9)+LoadData(:,10)+LoadData(:,11);

inload_ExtZoneLat=[];% repeat 365 times for the same schedule of
internal load
for i=1:365

inload_ExtZoneLat=[inload_ExtZoneLat;interLoad.ext(:,3).*TotalArea(3)];
end
LoadData(:,13)=inload_ExtZoneLat;

inload_IntZoneLat=[];% repeat 365 times for the same schedule of
internal load
for i=1:365

inload_IntZoneLat=[inload_IntZoneLat;interLoad.int(:,3).*TotalArea(2)];
end
LoadData(:,14)=inload_IntZoneLat;

%% calculate outdoor air humidity and anthalpy
OAhumDesnity=originalData.data(:,8); % lbmw/lbma
OAanthalpy=0.24.*originalData.data(:,4)+OAhumDesnity.*...
    (0.444*originalData.data(:,4)+970);
% Btu/lbm
% it can be a negative value if the air temp is below 0 C degree

%% control and energy simulation
%% Now Start the IAHU simulation
% SAT of the interior zone is the same as TAHU, SAT of the exterior
zone is
% optimized under the control algorithm.
% meanwhile, the outside air intake ratio is dynamic according to the
IAHU
% algorithm.  the initialization should be made at the begining
CCLAW_asump_in=ones(8760,1).*0.0082; % lbw/lba , 55F, 90%
Ocu_Schedule_in=[];
for i=1:365
    Ocu_Schedule_in=[Ocu_Schedule_in;Ocu_Schedule];
end
Schedule_in= Ocu_Schedule_in;
Loadsen_in=LoadData(:,12).*W2BTUhr; % Btu/hr
Loadlat_in=LoadData(:,14).*W2BTUhr; % Btu/hr
QCS_in=zeros(8760,1);
QCL_in=zeros(8760,1);
```

```
QH_in=zeros(8760,1);
QRH_in=zeros(8760,1);

RAW_in=ones(8760,1).*0.0082;
CCLAW_in=ones(8760,1).*0.0082;
MAT_in=Ti_in; % F
DAT_in=Ti_in; % F
MAW_in=RAW_in; % lbw/lba
SA_CFM_Required_in=(Loadsen_in./(AirCapRou60.*(Ti_in-SAT_in)));
SA_CFM_in=max(SA_CFM_Required_in,CFM_in_design*CFMRatio_in);

OA_IAQ_in=OAinIAQY; % vector
k=(Ti_in-SAT_in)./(Ti_in-originalData.data(:,4));
OA_eco_beta_in=((k>0).*(k<=1).*k+(k>1).*(originalData.data(:,4)<Trc)).*
Schedule_in;
%acceptable Economizer has a positive ratio, Trc=70
OA_IAQ_beta_in=OA_IAQ_in./SA_CFM_in.*Schedule_in;
OA_beta_in=max(OA_eco_beta_in,OA_IAQ_beta_in);
OA_CFM_in=OA_beta_in.*SA_CFM_in;



CCLAW_asump_in_I=ones(8760,1).*0.0082; % lbw/lba , 55F, 90%
Ocu_Schedule_I=[];
for i=1:365
    Ocu_Schedule_I=[Ocu_Schedule_I;Ocu_Schedule];
end
Schedule_I= Ocu_Schedule_I;
Loadsen_in_I=LoadData(:,12).*W2BTUhr; % Btu/hr
Loadlat_in_I=LoadData(:,14).*W2BTUhr; % Btu/hr
QCS_in_I=zeros(8760,1);
QCL_in_I=zeros(8760,1);
QH_in_I=zeros(8760,1);
QRH_in_I=zeros(8760,1);

RAW_in_I=ones(8760,1).*0.0082;
CCLAW_in_I=ones(8760,1).*0.0082;
SAT_in_I=SAT_in;
Ti_in_I=Ti_in;
MAT_in_I=Ti_in; % F
DAT_in_I=Ti_in; % F
MAW_in_I=RAW_in_I; % lbw/lba
SA_CFM_Required_in_I=(Loadsen_in_I./(AirCapRou60.*(Ti_in_I-SAT_in_I)));
SA_CFM_in_I=max(SA_CFM_Required_in_I,CFM_in_design*CFMRatio_in);

OA_IAQ_in_I=Schedule_I.*OAinIAQY; % based on schedule
k=(Ti_in_I-SAT_in_I)./(Ti_in_I-originalData.data(:,4));
OA_eco_beta_in_I=((k>0).*(k<=1).*k+(k>1).*(originalData.data(:,4)<Trc))
; % acceptable Economizer has a positive ratio, Trc=70
OA_IAQ_beta_in_I=OA_IAQ_in_I./SA_CFM_in_I;
OA_beta_in_I=max(OA_eco_beta_in_I,OA_IAQ_beta_in_I);
OA_CFM_in_I=OA_beta_in_I.*SA_CFM_in_I;

induct_Ratio=3;
CCLAW_asump_ex=ones(8760,1).*0.0082; % lbw/lba , 55F, 90%
Ocu_Schedule_ex=[];
```

```
for i=1:365
    Ocu_Schedule_ex=[Ocu_Schedule_ex;Ocu_Schedule];
end
Schedule_ex= Ocu_Schedule_ex;
Loadsen_ex=LoadData(:,8).*W2BTUhr; % Btu/hr
Loadlat_ex=LoadData(:,13).*W2BTUhr; % Btu/hr
QCS_ex=zeros(8760,1);
QCL_ex=zeros(8760,1);
QH_ex=zeros(8760,1);
QRCS_ex=zeros(8760,1);
QRCL_ex=zeros(8760,1);
QRH_ex=zeros(8760,1);

RAW_ex=ones(8760,1).*0.0082;
CCLAW_ex=ones(8760,1).*0.0082;
RA2W_ex=CCLAW_ex; % lbw/lba
MAW_ex=ones(8760,1).*0.0082; % lbw/lba
DAW_ex=ones(8760,1).*0.0082;% lbw/lba
MAT_ex=Ti_ex; % F
RA2T_ex=Ti_ex; % F degree
DAT_ex=Ti_ex; % F
SA_CFM_ex=ones(8760,1).*36000;
RA2_CFM_ex=SA_CFM_ex.*induct_Ratio;

OA_IAQd_ex=OAexIAQdY; % based on schedule
OA_IAQd_beta_ex=OA_IAQd_ex./SA_CFM_ex.*Schedule_ex;

k=(Ti_ex-SAT_ex)./(Ti_ex-originalData.data(:,4));
OA_eco_beta_ex=(((k>0).*(k<=1).*k+(k>1).*(originalData.data(:,4)<Trc)))
.*Schedule_ex; % acceptable Economizer has a positive ratio, Trc=70
OA_beta_ex=max(OA_eco_beta_ex,OA_IAQd_beta_ex);
OA_CFM_ex=OA_beta_ex.*SA_CFM_ex;

induct_Ratio=3;
CCLAW_asump_ex_I=ones(8760,1).*0.0082; % lbw/lba , 55F, 90%
Loadsen_ex_I=LoadData(:,8).*W2BTUhr; % Btu/hr
Loadlat_ex_I=LoadData(:,13).*W2BTUhr; % Btu/hr
QCS_ex_I=zeros(8760,1);
QCL_ex_I=zeros(8760,1);
QH_ex_I=zeros(8760,1);
QRCS_ex_I=zeros(8760,1);
QRCL_ex_I=zeros(8760,1);
QRH_ex_I=zeros(8760,1);

RAW_ex_I=ones(8760,1).*0.0082;
CCLAW_ex_I=ones(8760,1).*0.0082;
RA2W_ex_I=CCLAW_ex_I; % lbw/lba
MAW_ex_I=ones(8760,1).*0.0082; % lbw/lba
DAW_ex_I=ones(8760,1).*0.0082;% lbw/lba
Ti_ex_I=Ti_ex;
SAT_ex_I=SAT_ex;
MAT_ex_I=Ti_ex; % F
RA2T_ex_I=Ti_ex; % F degree
DAT_ex_I=Ti_ex; % F
SA_CFM_ex_I=ones(8760,1).*36000;
RA2_CFM_ex_I=SA_CFM_ex_I.*induct_Ratio;
```

```
OA_IAQd_ex_I=Schedule_I.*OAexIAQdY; % based on schedule
OA_IAQd_beta_ex_I=OA_IAQd_ex_I./SA_CFM_ex_I;
OA_IAQ_beta_ex_I=zeros(8760,1);

k=(Ti_ex_I-SAT_ex_I)./(Ti_ex_I-originalData.data(:,4));
OA_eco_beta_ex_I=(((k>0).*(k<=1).*k+(k>1).*(originalData.data(:,4)<Trc)
)); % acceptable Economizer has a positive ratio, Trc=70
OA_beta_ex_I=max(OA_eco_beta_ex_I,OA_IAQd_beta_ex_I);
OA_CFM_ex_I=OA_beta_ex_I.*SA_CFM_ex_I;

MoveAir_CFM=zeros(8760,1);
Gama=zeros(8760,1).*Schedule_I;
Fai=SA_CFM_in_I./(SA_CFM_in_I+SA_CFM_ex_I).*Schedule_I;
Dlt=zeros(8760,1);
Kexi=zeros(8760,1);
Nanda=zeros(8760,1);

X=1;
Delt_CFM=3000;% 3000 cfm exhaust from the interior zone

ACase1HtSaving=0;ACase1ClSaving=0;
ACase2HtSaving=0;ACase2ClSaving=0;
ACase3HtSaving=0;ACase3ClSaving=0;
ACase4HtSaving=0;ACase4ClSaving=0;
ACase1Ht=0;ACase1Cl=0;
ACase2Ht=0;ACase2Cl=0;
ACase3Ht=0;ACase3Cl=0;
ACase4Ht=0;ACase4Cl=0;

III=[];AAA=[];
AAB=[];
for i=1:8760
    if (Schedule_I(i)>0)
        %%
        if (originalData.data(i,4)<=Tic) % Tic is 55 F
            %% case 1

[Gama(i),Dlt(i),Kexi(i)]=sub1(OA_beta_in_I(i),SA_CFM_in_I(i),...
                SA_CFM_ex_I(i),X,Delt_CFM/SA_CFM_in_I(i));

OA_IAQ_beta_ex_I(i)=beIAQ(OA_IAQd_beta_ex_I(i),Gama(i),Fai(i),...
                OA_IAQ_beta_in_I(i),OA_beta_in_I(i));
            OA_beta_ex_I(i)=max(OA_IAQ_beta_ex_I(i),0);
            OA_CFM_ex_I(i)=OA_beta_ex_I(i)*SA_CFM_ex_I(i);
            MoveAir_CFM(i)=Gama(i)*SA_CFM_in_I(i);
            %%

resulinI=intVAV(OA_CFM_in_I(i),OAhumDesnity(i),originalData.data(i,4),S
AT_in_I(i),...

CCLAW_asump_in_I(i),SA_CFM_in_I(i),Ti_in_I(i),Loadsen_in_I(i),Loadlat_i
n_I(i));
            MAT_in_I(i)=resulinI(1);
            MAW_in_I(i)=resulinI(2);
            CCLAW_in_I(i)=resulinI(3);
```

```
            DAT_in_I(i)=resulinI(4);
            RAW_in_I(i)=resulinI(5);
            QCS_in_I(i)=resulinI(6);
            QCL_in_I(i)=resulinI(7);
            QH_in_I(i)=resulinI(8);
            QRH_in_I(i)=resulinI(9);
            %%

resul=extCAV(OA_CFM_ex(i),OAhumDesnity(i),originalData.data(i,4),...
            SAT_ex(i),CCLAW_asump_ex(i),SA_CFM_ex(i),RA2_CFM_ex(i),...
            Ti_ex(i),Loadsen_ex(i),Loadlat_ex(i));
            MAT_ex(i)=resul(1);
            MAW_ex(i)=resul(2);
            CCLAW_ex(i)=resul(3);
            RA2T_ex(i)=resul(4);
            RA2W_ex(i)=resul(5);
            DAT_ex(i)=resul(6);
            DAW_ex(i)=resul(7);
            RAW_ex(i)=resul(8);
            QCS_ex(i)=resul(9);
            QCL_ex(i)=resul(10);
            QH_ex(i)=resul(11);
            QRCS_ex(i)=resul(12);
            QRCL_ex(i)=resul(13);
            QRH_ex(i)=resul(14);
            %%

resultexI=extCAVI(OA_CFM_ex_I(i),OAhumDesnity(i),originalData.data(i,4)
,...

SAT_ex_I(i),CCLAW_asump_ex_I(i),SA_CFM_ex_I(i),RA2_CFM_ex_I(i),...

Ti_in_I(i),Ti_ex_I(i),RAW_in_I(i),Loadsen_ex_I(i),Loadlat_ex_I(i),MoveA
ir_CFM(i));
            MAT_ex_I(i)=resultexI(1);
            MAW_ex_I(i)=resultexI(2);
            CCLAW_ex_I(i)=resultexI(3);
            RA2T_ex_I(i)=resultexI(4);
            RA2W_ex_I(i)=resultexI(5);
            DAT_ex_I(i)=resultexI(6);
            DAW_ex_I(i)=resultexI(7);
            RAW_ex_I(i)=resultexI(8);
            QCS_ex_I(i)=resultexI(9);
            QCL_ex_I(i)=resultexI(10);
            QH_ex_I(i)=resultexI(11);
            QRCS_ex_I(i)=resultexI(12);
            QRCL_ex_I(i)=resultexI(13);
            QRH_ex_I(i)=resultexI(14);
            ACase1Ht=ACase1Ht+QH_ex(i)+QRH_ex(i);
            ACase1Cl=ACase1Cl+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i);
            ACase1HtSaving=ACase1HtSaving+QH_ex(i)+QRH_ex(i)-
(QH_ex_I(i)...
                +QRH_ex_I(i));

ACase1ClSaving=ACase1ClSaving+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i)-(QCS_ex_I(i)+QCL_ex_I(i)+QRCS_ex_I(i)+...
```

```
                QRCL_ex_I(i));
        end
        %% case 2
        if (Tic<originalData.data(i,4))&&(originalData.data(i,4)<= Tec)
            % III=[III;OA_beta_in_I(i)];
            if X>0 % the sameving is brought by IAHU
            if LoadData(i,8)<0 % exterior heating needed, no direct
outside air intake
                AAA=[AAA,i];

[Gama(i),Dlt(i),Kexi(i)]=sub1(OA_beta_in_I(i),SA_CFM_in_I(i),...
                SA_CFM_ex_I(i),X,Delt_CFM/SA_CFM_in_I(i));

OA_IAQ_beta_ex_I(i)=beIAQ(OA_IAQd_beta_ex_I(i),Gama(i),Fai(i),...
                OA_IAQ_beta_in_I(i),OA_beta_in_I(i));
                OA_beta_ex_I(i)=max(OA_IAQ_beta_ex_I(i),0);
                SAT_ex_I(i)=Ti_ex_I(i);
                OA_CFM_ex_I(i)=OA_beta_ex_I(i)*SA_CFM_ex_I(i);
                MoveAir_CFM(i)=Gama(i)*SA_CFM_in_I(i);
            else % cooling savings regardless of X

                Gama(i)=0;
                Dlt(i)=Delt_CFM/SA_CFM_in_I(i);
                Kexi(i)=1-Dlt(i);
                MoveAir_CFM(i)=0;
                SAT_ex_I(i)=Ti_ex_I(i)-
Loadsen_ex_I(i)/(AirCapRou60*SA_CFM_ex_I(i));
                k=(Ti_ex_I(i)-SAT_ex_I(i))./(Ti_ex_I(i)-
originalData.data(i,4));

OA_eco_beta_ex_I(i)=(k>0)*(k<=1)*k+(k>1).*(originalData.data(i,4)<Trc);

OA_beta_ex_I(i)=max(OA_eco_beta_ex_I(i),OA_IAQd_beta_ex_I(i));
                OA_CFM_ex_I(i)=OA_beta_ex_I(i).*SA_CFM_ex_I(i);
            end
            end
%

resulinI=intVAV(OA_CFM_in_I(i),OAhumDesnity(i),originalData.data(i,4),S
AT_in_I(i),...

CCLAW_asump_in_I(i),SA_CFM_in_I(i),Ti_in_I(i),Loadsen_in_I(i),Loadlat_i
n_I(i));
            MAT_in_I(i)=resulinI(1);
            MAW_in_I(i)=resulinI(2);
            CCLAW_in_I(i)=resulinI(3);
            DAT_in_I(i)=resulinI(4);
            RAW_in_I(i)=resulinI(5);
            QCS_in_I(i)=resulinI(6);
            QCL_in_I(i)=resulinI(7);
            QH_in_I(i)=resulinI(8);
            QRH_in_I(i)=resulinI(9);


resul=extCAV(OA_CFM_ex(i),OAhumDesnity(i),originalData.data(i,4),...
            SAT_ex(i),CCLAW_asump_ex(i),SA_CFM_ex(i),RA2_CFM_ex(i),...
```

```
            Ti_ex(i),Loadsen_ex(i),Loadlat_ex(i));
            MAT_ex(i)=resul(1);
            MAW_ex(i)=resul(2);
            CCLAW_ex(i)=resul(3);
            RA2T_ex(i)=resul(4);
            RA2W_ex(i)=resul(5);
            DAT_ex(i)=resul(6);
            DAW_ex(i)=resul(7);
            RAW_ex(i)=resul(8);
            QCS_ex(i)=resul(9);
            QCL_ex(i)=resul(10);
            QH_ex(i)=resul(11);
            QRCS_ex(i)=resul(12);
            QRCL_ex(i)=resul(13);
            QRH_ex(i)=resul(14);


resultexI=extCAVI(OA_CFM_ex_I(i),OAhumDesnity(i),originalData.data(i,4)
,...

SAT_ex_I(i),CCLAW_asump_ex_I(i),SA_CFM_ex_I(i),RA2_CFM_ex_I(i),...

Ti_in_I(i),Ti_ex_I(i),RAW_in_I(i),Loadsen_ex_I(i),Loadlat_ex_I(i),MoveA
ir_CFM(i));
            MAT_ex_I(i)=resultexI(1);
            MAW_ex_I(i)=resultexI(2);
            CCLAW_ex_I(i)=resultexI(3);
            RA2T_ex_I(i)=resultexI(4);
            RA2W_ex_I(i)=resultexI(5);
            DAT_ex_I(i)=resultexI(6);
            DAW_ex_I(i)=resultexI(7);
            RAW_ex_I(i)=resultexI(8);
            QCS_ex_I(i)=resultexI(9);
            QCL_ex_I(i)=resultexI(10);
            QH_ex_I(i)=resultexI(11);
            QRCS_ex_I(i)=resultexI(12);
            QRCL_ex_I(i)=resultexI(13);
            QRH_ex_I(i)=resultexI(14);
            ACase2Ht=ACase2Ht+QH_ex(i)+QRH_ex(i);
            ACase2Cl=ACase2Cl+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i);
            ACase2HtSaving=ACase2HtSaving+QH_ex(i)+QRH_ex(i)-
(QH_ex_I(i)...
                +QRH_ex_I(i));

ACase2ClSaving=ACase2ClSaving+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i)-(QCS_ex_I(i)+QCL_ex_I(i)+QRCS_ex_I(i)+...
                QRCL_ex_I(i));
        end
        % case 3
        if  (originalData.data(i,4)> Tec)&&(originalData.data(i,4)<=Trc)%
from 60 to 70 F
           % III=[III;OA_beta_ex_I(i)];
            Enthp=0.24*Ti_ex_I(i)+0.0082*(0.444*Ti_ex_I(i)+970);
            if (X>0)
                if (OAanthalpy(i)<Enthp)
            OA_beta_ex_I(i)=1;
```

```
            OA_CFM_ex_I(i)=OA_beta_ex_I(i)*SA_CFM_ex_I(i);
            Gama(i)=0;
            Dlt(i)=Delt_CFM/SA_CFM_in_I(i);
            Kexi(i)=1-Dlt(i);
            MoveAir_CFM(i)=0;
            SAT_ex_I(i)=Ti_ex_I(i)-
Loadsen_ex_I(i)/(AirCapRou60*SA_CFM_ex_I(i));
                end
            end
%

resulinI=intVAV(OA_CFM_in_I(i),OAhumDesnity(i),originalData.data(i,4),S
AT_in_I(i),...

CCLAW_asump_in_I(i),SA_CFM_in_I(i),Ti_in_I(i),Loadsen_in_I(i),Loadlat_i
n_I(i));
            MAT_in_I(i)=resulinI(1);
            MAW_in_I(i)=resulinI(2);
            CCLAW_in_I(i)=resulinI(3);
            DAT_in_I(i)=resulinI(4);
            RAW_in_I(i)=resulinI(5);
            QCS_in_I(i)=resulinI(6);
            QCL_in_I(i)=resulinI(7);
            QH_in_I(i)=resulinI(8);
            QRH_in_I(i)=resulinI(9);


resul=extCAV(OA_CFM_ex(i),OAhumDesnity(i),originalData.data(i,4),...
            SAT_ex(i),CCLAW_asump_ex(i),SA_CFM_ex(i),RA2_CFM_ex(i),...
            Ti_ex(i),Loadsen_ex(i),Loadlat_ex(i));
            MAT_ex(i)=resul(1);
            MAW_ex(i)=resul(2);
            CCLAW_ex(i)=resul(3);
            RA2T_ex(i)=resul(4);
            RA2W_ex(i)=resul(5);
            DAT_ex(i)=resul(6);
            DAW_ex(i)=resul(7);
            RAW_ex(i)=resul(8);
            QCS_ex(i)=resul(9);
            QCL_ex(i)=resul(10);
            QH_ex(i)=resul(11);
            QRCS_ex(i)=resul(12);
            QRCL_ex(i)=resul(13);
            QRH_ex(i)=resul(14);


resultexI=extCAVI(OA_CFM_ex_I(i),OAhumDesnity(i),originalData.data(i,4)
,...

SAT_ex_I(i),CCLAW_asump_ex_I(i),SA_CFM_ex_I(i),RA2_CFM_ex_I(i),...

Ti_in_I(i),Ti_ex_I(i),RAW_in_I(i),Loadsen_ex_I(i),Loadlat_ex_I(i),MoveA
ir_CFM(i));
            MAT_ex_I(i)=resultexI(1);
            MAW_ex_I(i)=resultexI(2);
            CCLAW_ex_I(i)=resultexI(3);
```

```
            RA2T_ex_I(i)=resultexI(4);
            RA2W_ex_I(i)=resultexI(5);
            DAT_ex_I(i)=resultexI(6);
            DAW_ex_I(i)=resultexI(7);
            RAW_ex_I(i)=resultexI(8);
            QCS_ex_I(i)=resultexI(9);
            QCL_ex_I(i)=resultexI(10);
            QH_ex_I(i)=resultexI(11);
            QRCS_ex_I(i)=resultexI(12);
            QRCL_ex_I(i)=resultexI(13);
            QRH_ex_I(i)=resultexI(14);
            ACase3Ht=ACase3Ht+QH_ex(i)+QRH_ex(i);
            ACase3Cl=ACase3Cl+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i);
            ACase3HtSaving=ACase3HtSaving+QH_ex(i)+QRH_ex(i)-
(QH_ex_I(i)...
                +QRH_ex_I(i));

ACase3ClSaving=ACase3ClSaving+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i)-(QCS_ex_I(i)+QCL_ex_I(i)+QRCS_ex_I(i)+...
                QRCL_ex_I(i));
        end

    %
            Enthp=0.24*Ti_ex_I(i)+0.0082*(0.444*Ti_ex_I(i)+970);
            case4=(originalData.data(i,4)>
Tec)&&(originalData.data(i,4)<=Trc)&&...
                (OAanthalpy(i)>Enthp);

        if (case4||(originalData.data(i,4)> Trc))
            %% case 4 and 5 together
            if (Loadsen_ex_I(i)< AirCapRou60*SA_CFM_ex_I(i)*...
                    (Ti_ex_I(i)-SAT_ex_I(i)))
                BiTemp=OA_IAQd_beta_ex_I(i)*(1-
Fai(i))/Fai(i)+OA_IAQ_beta_in_I(i);
                GamaTemp=BiTemp;
                if (GamaTemp<=(X*(1-
Fai(i))/Fai(i)))&&(BiTemp>=0)&&(BiTemp<=1)
                    Gama(i)=GamaTemp;
                    OA_beta_in_I(i)=max(OA_IAQ_beta_ex_I(i),0);
                    OA_beta_ex_I(i)=0;
                    SAT_ex_I(i)=Ti_ex_I(i);
                else
                    Gama(i)=0;
                    Dlt(i)=Delt_CFM/SA_CFM_in_I(i);
                end
            else
                % either not mild weather or IAHU not possible
                %OA_CFM_ex_I(i)=OA_beta_ex_I(i)*SA_CFM_ex_I(i);
                Gama(i)=0;
                Dlt(i)=Delt_CFM/SA_CFM_in_I(i);
                Kexi(i)=1-Dlt(i);
                MoveAir_CFM(i)=0;
                OA_beta_ex_I(i)=OA_IAQd_beta_ex_I(i);
            end
```

```
resulin=intVAV(OA_CFM_in(i),OAhumDesnity(i),originalData.data(i,4),SAT_
in(i),...

CCLAW_asump_in(i),SA_CFM_in(i),Ti_in(i),Loadsen_in(i),Loadlat_in(i));
            MAT_in(i)=resulin(1);
            MAW_in(i)=resulin(2);
            CCLAW_in(i)=resulin(3);
            DAT_in(i)=resulin(4);
            RAW_in(i)=resulin(5);
            QCS_in(i)=resulin(6);
            QCL_in(i)=resulin(7);
            QH_in(i)=resulin(8);
            QRH_in(i)=resulin(9);


resulinI=intVAV(OA_CFM_in_I(i),OAhumDesnity(i),originalData.data(i,4),S
AT_in_I(i),...

CCLAW_asump_in_I(i),SA_CFM_in_I(i),Ti_in_I(i),Loadsen_in_I(i),Loadlat_i
n_I(i));
            MAT_in_I(i)=resulinI(1);
            MAW_in_I(i)=resulinI(2);
            CCLAW_in_I(i)=resulinI(3);
            DAT_in_I(i)=resulinI(4);
            RAW_in_I(i)=resulinI(5);
            QCS_in_I(i)=resulinI(6);
            QCL_in_I(i)=resulinI(7);
            QH_in_I(i)=resulinI(8);
            QRH_in_I(i)=resulinI(9);


resul=extCAV(OA_CFM_ex(i),OAhumDesnity(i),originalData.data(i,4),...
            SAT_ex(i),CCLAW_asump_ex(i),SA_CFM_ex(i),RA2_CFM_ex(i),...
            Ti_ex(i),Loadsen_ex(i),Loadlat_ex(i));
%
resul=extCAVI(OA_CFM_ex(i),OAhumDesnity(i),originalData.data(i,4),...
%
SAT_ex(i),CCLAW_asump_ex_I(i),SA_CFM_ex_I(i),RA2_CFM_ex(i),...
%
Ti_ex(i),Ti_ex(i),RAW_in_I(i),Loadsen_ex(i),Loadlat_ex(i),0);
            MAT_ex(i)=resul(1);
            MAW_ex(i)=resul(2);
            CCLAW_ex(i)=resul(3);
            RA2T_ex(i)=resul(4);
            RA2W_ex(i)=resul(5);
            DAT_ex(i)=resul(6);
            DAW_ex(i)=resul(7);
            RAW_ex(i)=resul(8);
            QCS_ex(i)=resul(9);
            QCL_ex(i)=resul(10);
            QH_ex(i)=resul(11);
            QRCS_ex(i)=resul(12);
            QRCL_ex(i)=resul(13);
            QRH_ex(i)=resul(14);
```

```
resultexI=extCAVI(OA_CFM_ex_I(i),OAhumDesnity(i),originalData.data(i,4)
,...

SAT_ex_I(i),CCLAW_asump_ex_I(i),SA_CFM_ex_I(i),RA2_CFM_ex_I(i),...

Ti_in_I(i),Ti_ex_I(i),RAW_in_I(i),Loadsen_ex_I(i),Loadlat_ex_I(i),MoveA
ir_CFM(i));
            MAT_ex_I(i)=resultexI(1);
            MAW_ex_I(i)=resultexI(2);
            CCLAW_ex_I(i)=resultexI(3);
            RA2T_ex_I(i)=resultexI(4);
            RA2W_ex_I(i)=resultexI(5);
            DAT_ex_I(i)=resultexI(6);
            DAW_ex_I(i)=resultexI(7);
            RAW_ex_I(i)=resultexI(8);
            QCS_ex_I(i)=resultexI(9);
            QCL_ex_I(i)=resultexI(10);
            QH_ex_I(i)=resultexI(11);
            QRCS_ex_I(i)=resultexI(12);
            QRCL_ex_I(i)=resultexI(13);
            QRH_ex_I(i)=resultexI(14);
              ACase4Ht=ACase4Ht+QH_ex(i)+QRH_ex(i);
              ACase4Cl=ACase4Cl+QCS_ex(i)+QCL_ex(i)+QRCS_ex(i)...
                +QRCL_ex(i);
            ACase4HtSaving=ACase4HtSaving+QH_ex(i)+QRH_ex(i)-
(QH_ex_I(i)...
                +QRH_ex_I(i));

ACase4ClSaving=ACase4ClSaving+QRCS_ex(i)+QRCL_ex(i)+QCS_ex(i)...
                +QCL_ex(i)+QCS_in(i)+QCL_in(i)-
(QRCS_ex_I(i)+QRCL_ex_I(i)+QCS_ex_I(i)...
                +QCL_ex_I(i)+QCS_in_I(i)+QCL_in_I(i));
        end
        ACooling_ex=sum(QRCS_ex)+sum(QRCL_ex);
        ACooling_ex_I=sum(QRCS_ex_I)+sum(QRCL_ex_I);
    end
end
TotalHeat_TAHU=sum(QH_ex+QRH_ex+QH_in_I+QRH_in_I);
TotalCool_TAHU=sum(QCS_ex+QRCS_ex+QCL_ex+QRCL_ex+QCS_in_I+QCL_in_I);
AAAA=[ACase1Ht,ACase1Cl,ACase1ClSaving-ACase1HtSaving;...
    ACase2Ht,ACase2Cl,ACase2ClSaving-ACase2HtSaving;...
    ACase3Ht,ACase3Cl,ACase3ClSaving-ACase3HtSaving;...
    ACase4Ht,ACase4Cl,ACase4ClSaving-ACase4HtSaving];

TotalHeatingSaving=ACase1HtSaving+ACase2HtSaving+ACase3HtSaving+...
    ACase4HtSaving;
TotalCoolingSaving=ACase1ClSaving+ACase2ClSaving+ACase3ClSaving+...
    ACase4ClSaving;
%
AATotalHeating_I=(sum(QH_in_I)+sum(QRH_in_I)+sum(QH_ex_I)+sum(QRH_in_I)
)/TotalArea(1);
%
AATotalCooling_I=(sum(QCS_in_I)+sum(QCL_in_I)+sum(QCS_ex_I)+sum(QCL_ex_
I)+...
%      sum(QRCS_ex_I)+sum(QRCL_ex_I))/TotalArea(1);
% Qheating_in_I=QH_in_I+QRH_in_I;
```

```
% Qcooling_in_I=QCS_in_I+QCL_in_I;
% Qheating_ex_I=QH_ex_I+QRH_ex_I;
% Qcooling_ex_I=QCS_ex_I+QCL_ex_I+QRCS_ex_I+QRCL_ex_I;
% Qheating_I=Qheating_in_I+Qheating_ex_I;
% Qcooling_I=Qcooling_in_I+Qcooling_ex_I;
% monthI.a=originalData.data(:,1);
% energyI.a=[Qheating_I,Qcooling_I];
% MonthlyEnergyI=Monthly(monthI,energyI);
% figure(5);scatter(originalData.data(:,4),-
(Qheating_I),5,'*','r');hold on;
% figure(6);scatter(originalData.data(:,4),Qcooling_I,5,'*','b');hold
on;
% figure(7);bar([1:12],-MonthlyEnergyI(:,2),'r');hold on;
% figure(8);bar([1:12],MonthlyEnergyI(:,3),'b');hold on;
% figure(3);h=bar([-
(MonthlyEnergy(:,2)),MonthlyEnergy(:,3)],'grouped');hold on;


function Env_Load_Coeffs = Envelop(area)
%% a proper formate of BIN data for simplified application
originalData=importdata('Omaha_Weather.csv');
[hours, cols]=size(originalData.data);
Ti_I_Summer=76;Ti_I_Winter=76;
Ti_E_Summer=76;Ti_E_Winter=70;
sqf2sqm=0.09290304;btuh2w=0.29307107;
%% data interpretation
%Weather_Data column 1: month (1-12)
%Weather_Data column 2: day (1 to 31, change may happen)
%Weather_Data column 3: hours (1 to 24)
%Weather_Data column 4: dry bulb (F)
%Weather_Data column 5: wet bulb (F)
%Weather_Data column 6: dewpoint (F)
%Weather_Data column 7: Relative humidity ration (%)
%Weather_Data column 8: HumidityRatio (lbmw/lbma)
%Weather_Data column 9: Solar global horizontal (W/m2)
%Weather_Data column 10: Solar direct norm (W/m2)
%Weather_Data column 11: Wind speed (MPH)
%Weather_Data column 12: Wind direction (degree)
%% Building Basic Info
% Bld_Width=70;Bld_Length=270;%ft
% Side_Width=16; %ft
% Floor_Area=Bld_Length*Bld_Width; %ft
% Int_Area=(Bld_Width-(2*Side_Width))*(Bld_Length-(2*Side_Width));
% Ext_Area=Floor_Area-Int_Area;
% Floor_Area=18000; % sft
% Int_Area=8594; % sft
% Ext_Area=9406; % sft
Floor_Area=area(1); % sft
Int_Area=area(2); % sft
Ext_Area=area(3); % sft
East_Wdow=1755;West_Wdow=1248;North_Wdow=260;South_Wdow=260; % sft
U_Wdow=3.596*1.2;% W/m2 K
West_Wall=2442;East_Wall=1833.8;North_Wall=300;South_Wall=300;% sft,
each floor
Int_Roof=Int_Area;
Ext_Roof=Floor_Area-Int_Roof;
```

```
U_Wall=0.8418;U_Roof=0.559;% W/m2 K
West_Con=317.33;East_Con=317.33;North_Con=466.67;South_Con=466.67;
U_Con=0.45;% W/m2 K
% Density_Ocup=0.002;Lit=540;Density_Eqp=2;Unit_Lit=40;
%
Ext_Ocup=Density_Ocup*Ext_Area;Ext_Lit=Lit*Ext_Area/Floor_Area*Unit_Lit;
% Ext_Eqp=Density_Eqp*Ext_Area;
%
Int_Ocup=Density_Ocup*Int_Area;Int_Lit=Lit*Int_Area/Floor_Area*Unit_Lit;
% Int_Eqp=Density_Eqp*Int_Area;
AG=[North_Wdow East_Wdow South_Wdow West_Wdow];
Wall_Area=[North_Wall East_Wall South_Wall West_Wall];
Column_Area=[North_Con East_Con South_Con West_Con];
U_Value=[U_Wall U_Con U_Wdow U_Roof];
%% Solar Through Glass Linear function of To
MSHGF_Jul=[38 216 109 216];%N, E, S, W
Ac_Hr=18;
SC_Jul=[0.55 0.55 0.55 0.55];
CLFTOT_Jul=[11.57 5.46  6.43 5.46];
FPS_Jul=[0.78 0.78 0.78 0.78];
SLFD_Jul=[1 0.65 0.07 0.65];
Norht_AG2_Jul=AG(1,1)*SLFD_Jul(1,1)+sum(AG(1,2:4).*(1-SLFD_Jul(1,2:4)));
AG2_Jul=[Norht_AG2_Jul AG(1,2:4).*SLFD_Jul(1,2:4)];
Qsol_Jul=sum(MSHGF_Jul.*SC_Jul.*CLFTOT_Jul.*FPS_Jul.*AG2_Jul)*btuh2w;
Qsol_Jul_PerArea=Qsol_Jul/Floor_Area/Ac_Hr;
MSHGF_Jan=[20 154 254 154];%N, E, S, W
SC_Jan=[0.55 0.55 0.55 0.55];
CLFTOT_Jan=[11.57 5.46  6.43 5.46];
FPS_Jan=[0.61 0.61 0.61 0.61];
SLFD_Jan=[1 0.55 0.7 0.55];
Norht_AG2_Jan=AG(1,1)*SLFD_Jan(1,1)+sum(AG(1,2:4).*(1-SLFD_Jan(1,2:4)));
AG2_Jan=[Norht_AG2_Jan AG(1,2:4).*SLFD_Jan(1,2:4)];
Qsol_Jan=sum(MSHGF_Jan.*SC_Jan.*CLFTOT_Jan.*FPS_Jan.*AG2_Jan)*btuh2w; %
W
Qsol_Jan_PerArea=Qsol_Jan/Floor_Area/24;% W/sft, winter divided by 24
Tpc=100;% 92
Tph=30;% outside design temperature -2
Tpcc=(Tpc-32)/1.8;
Tphc=(Tph-32)/1.8;
M_Glass_sol=(Qsol_Jul_PerArea-Qsol_Jan_PerArea)/(Tpcc-Tphc);% W/K sqf
%Sol_Load=M_Glass_sol*(Weather_Data(:,4)-Tphc)+Qsol_Jan_PerArea;%solar
load for each floor, W/sqf
%% Opaque surfaces, due to solar contribution, Linear Function of To
CLTD_Jan=[0 6 21 6 3 3]./1.8;% from F degree to K degree
CLTD_Jul=[5 15 10 15 22 22]./1.8;% from F degree to K degree
K_Color=[0.83 0.83 0.83 0.83 0.75 0.75];
Qts_Wall_Jul=sum((FPS_Jul.*K_Color(1:4).*CLTD_Jul(1:4).*U_Value(1,1).*W
all_Area)')*sqf2sqm;% W/ K
Qts_Wall_Jan=sum((FPS_Jan.*K_Color(1:4).*CLTD_Jan(1:4).*U_Value(1,1).*W
all_Area)')*sqf2sqm;% W/ K
Qts_Colum_Jul=sum((FPS_Jul.*K_Color(1:4).*CLTD_Jul(1:4).*U_Value(1,2).*
Column_Area)')*sqf2sqm;% W/ K
Qts_Colum_Jan=sum((FPS_Jan.*K_Color(1:4).*CLTD_Jan(1:4).*U_Value(1,2).*
Column_Area)')*sqf2sqm;% W/ K
Qts_ExtRoof_Jul=(FPS_Jul(1)*K_Color(1,6)*CLTD_Jul(1,6)*U_Value(1,4)*Ext
_Roof)*sqf2sqm/10;% W/ K
```

```
Qts_ExtRoof_Jan=(FPS_Jan(1)*K_Color(1,6)*CLTD_Jan(1,6)*U_Value(1,4)*Ext
_Roof)*sqf2sqm/10;% W/ K
Qts_IntRoof_Jul=(FPS_Jul(1)*K_Color(1,6)*CLTD_Jul(1,6)*U_Value(1,4)*Int
_Roof)*sqf2sqm/10;% W/ K
Qts_IntRoof_Jan=(FPS_Jan(1)*K_Color(1,6)*CLTD_Jan(1,6)*U_Value(1,4)*Int
_Roof)*sqf2sqm/10;% W/ K
Qts_IntRoof_Jul_PerArea=Qts_IntRoof_Jul/Floor_Area; % W/ K sft
Qts_IntRoof_Jan_PerArea=Qts_IntRoof_Jan/Floor_Area; % W/ K sft
Qts_Ext_Jul=Qts_Wall_Jul+Qts_Colum_Jul+Qts_ExtRoof_Jul; % W/ K
Qts_Ext_Jan=Qts_Wall_Jan+Qts_Colum_Jan+Qts_ExtRoof_Jan; % W/ K
Qts_Ext_Jul_PerArea=Qts_Ext_Jul/Floor_Area;  % W/ K sft
Qts_Ext_Jan_PerArea=Qts_Ext_Jan/Floor_Area; % W/K sft
M_Ext_Sol=(Qts_Ext_Jul_PerArea-Qts_Ext_Jan_PerArea)/(Tpcc-Tphc);% W/K
sft
M_Int_Sol=(Qts_IntRoof_Jul_PerArea-Qts_IntRoof_Jan_PerArea)/(Tpcc-
Tphc);% W/K sft
%% Transmission Load, due to air-to-air difference
Qt_Wdow=U_Value(1,3)*sum(AG)*sqf2sqm; % W/K
%% Opaque surfaces, due to air-to-air difference
Qt_Wall=U_Value(1,1)*sum(Wall_Area)*sqf2sqm; % W/ K
Qt_Colum=U_Value(1,2)*sum(Column_Area)*sqf2sqm; % W/ K
Qt_IntRoof=U_Value(1,4)*Int_Roof*sqf2sqm/10; % W/ K
Qt_ExtRoof=U_Value(1,4)*Ext_Roof*sqf2sqm/10; % W/ K
Qt_Int=Qt_IntRoof;% W/K sft
Qt_Ext=Qt_Wall+Qt_Colum+Qt_ExtRoof+Qt_Wdow;% W/ K
Qt_Int_PerArea=Qt_Int/Int_Area;% W/K sft
Qt_Ext_PerArea=Qt_Ext/Ext_Area;% W/K sft
%
%% start outputs
Env_Load_Coeffs.glass=[M_Glass_sol Qsol_Jan_PerArea]; % W/sft K,  W/sft
Env_Load_Coeffs.int=[Qt_Int_PerArea M_Int_Sol
Qts_IntRoof_Jan_PerArea]; % W/sft K, W/sft K, W/ sft K
Env_Load_Coeffs.ext=[Qt_Ext_PerArea M_Ext_Sol Qts_Ext_Jan_PerArea]; %
W/sft K,




function
resl=extCAV(OA_CFM,OAW,OA_temp,SAT_set,CCLW,SA_CFM,RA2_CFM,Ti_set,LoadS
en,LoadLat)
% a subfunction for induction unit CAV simulation
AirCapRou60=1.06; % 60*rou*Cp: Btu min/ft^3 F hr . Energy Btu/hr=
AirCap60*CFM*delt
W2BTUhr=3.412;
OABeta=OA_CFM/SA_CFM;
induRatio=RA2_CFM/SA_CFM;
QCL=0;QCS=0;QH=0;heat=0;QRH=0;QRCS=0;QRCL=0;

MAT=OABeta*OA_temp+(1-OABeta)*Ti_set;
RA2T=Ti_set;DAT=SAT_set;
RAW=CCLW;RA2W=CCLW;DAW=CCLW;
MAW=CCLW;CCLAW=CCLW;

k=0;
```

```
j=0;
if (abs(MAT-SAT_set))>0.01
k=AirCapRou60*SA_CFM*(MAT-SAT_set); % primary sensible load
end
j= LoadSen-AirCapRou60*SA_CFM*(Ti_set-SAT_set);  % secondary sensible
load

if (OA_temp>55) % dehumidification might be needed
    QCS=(k>1)*k;
    QH= (k<-1)*k;
    QRCS= (j>1)*j;
    QRH= (j<-1)*j;

    if j>1 % secnodary cooling and possible dehumidification
        RAW=(RA2W*RA2_CFM+CCLAW*SA_CFM+...
       (LoadLat)/4840)/(RA2_CFM+SA_CFM);
        MAW=RAW+OABeta*(OAW-RAW);
        if (MAW<CCLAW)||(k<=0)
            RAW=(RA2W*RA2_CFM+OABeta*SA_CFM*OAW+...
                LoadLat/4840)/(RA2_CFM+OA_CFM);
            MAW=RAW+OABeta*(OAW-RAW);
            CCLAW=MAW;
            if (RAW<RA2W)||(j<=0)
                if OA_CFM>0
                RAW=(LoadLat)/(4840*SA_CFM*OABeta)+OAW;
                end
                RA2W=RAW;
            else
                RA2W=CCLW;
            end
            MAW=RAW+OABeta*(OAW-RAW);
            CCLAW=MAW;
        else
            if (RAW<RA2W)||(j<=0)
                RAW=(LoadLat)/(4840*SA_CFM)+CCLW;
                MAW=RAW+OABeta*(OAW-RAW);
            end
        end
        QCL= 4840*SA_CFM*(MAW-CCLAW)*(k>1); % Btu/hr
        QRCL= 4840*RA2_CFM*(RAW-RA2W); % Btu/hr
    else % secondary heating, no dehumidification
        RAW=(CCLAW*SA_CFM+LoadLat/4840)/(SA_CFM);
        MAW=RAW+OABeta*(OAW-RAW);
        if (MAW<CCLAW)||(k<=0) % if the primary no dehumidification
either
            if OA_CFM>0
            RAW=(LoadLat)/(4840*SA_CFM*OABeta)+OAW;
            end
            RA2W=RAW;
            MAW=RAW+OABeta*(OAW-RAW);
            CCLAW=MAW;
        end
        QRCL=0;
        QCL= 4840*SA_CFM*(MAW-CCLAW); % Btu/hr
        DAW=(RA2W*induRatio+CCLAW)/(1+induRatio);
    end
```

```
    RA2T= j/(AirCapRou60*RA2_CFM)+ Ti_set;
    DAT= (RA2T*induRatio+SAT_set)/(1+induRatio);
else % no dehumidification
    QCS=(k>1)*k;
    QH= (k<-1)*k;
    QRCS= (j>1)*j;
    QRH= (j<-1)*j;
    RA2T= j/(AirCapRou60*RA2_CFM)+Ti_set;
    DAT= (RA2T*induRatio+SAT_set)/(1+induRatio);
    if OABeta>0
    RAW=(LoadLat)/(4840*SA_CFM*OABeta)+OAW;% lbw/lba
    end
    RA2W=RAW;
    MAW=RAW+OABeta*(OAW-RAW);
    CCLAW=MAW;
    DAW=(RA2W*induRatio+CCLAW)/(1+induRatio);
    QRCL=0;QCL=0;
end

resl=[MAT,MAW,CCLAW,RA2T,RA2W,DAT,DAW,RAW,QCS,QCL,QH,QRCS,QRCL,QRH];
end




function
resl=extCAVI(OA_CFM,OAW,OA_temp,SAT_set,CCLW,SA_CFM,RA2_CFM,Ti_in_set,T
i_ex_set,RAW_in,LoadSen,LoadLat,Move_CFM)
% a subfunction for induction unit CAV simulation
AirCapRou60=1.06; % 60*rou*Cp: Btu min/ft^3 F hr . Energy Btu/hr=
AirCap60*CFM*delt
W2BTUhr=3.412;
OABeta=OA_CFM/SA_CFM;
induRatio=RA2_CFM/SA_CFM;
QCL=0;QCS=0;QH=0;heat=0;QRH=0;QRCS=0;QRCL=0;

MAT=OABeta*OA_temp+(1-OABeta)*Ti_ex_set;
RA2T=Ti_ex_set;DAT=SAT_set;
RAW=CCLW;RA2W=CCLW;DAW=CCLW;
MAW=CCLW;CCLAW=CCLW;

k=0;
j=0;
if (abs(MAT-SAT_set))>0.01
k=AirCapRou60*SA_CFM*(MAT-SAT_set);
end
j= LoadSen-AirCapRou60*SA_CFM*(Ti_ex_set-SAT_set)-
AirCapRou60*Move_CFM*...
    (Ti_ex_set-Ti_in_set);

if (OA_temp>55)
    QCS=(k>1)*k;
    QH= (k<-1)*k;
    QRCS= (j>1)*j;
    QRH= (j<-1)*j;
```

```
    if j>1  % secnodary cooling and possible dehumidification
        RAW=(RA2W*RA2_CFM+CCLAW*SA_CFM+RAW_in*Move_CFM+...
       (LoadLat)/4840)/(RA2_CFM+SA_CFM+Move_CFM);
        MAW=RAW+OABeta*(OAW-RAW);
        if (MAW<CCLAW)||(k<=0)
            RAW=(RA2W*RA2_CFM+OABeta*SA_CFM*OAW+RAW_in*Move_CFM+...
                LoadLat/4840)/(RA2_CFM+OA_CFM+Move_CFM);
            MAW=RAW+OABeta*(OAW-RAW);
            CCLAW=MAW;
            if (RAW<RA2W)||(j<=0)
                if (OA_CFM+Move_CFM)>0
                RAW=(LoadLat/4840+OA_CFM*OAW+Move_CFM*RAW_in)/...
                    (OA_CFM+Move_CFM);
                end
                RA2W=RAW;
            else
                RA2W=CCLW;
            end
            MAW=RAW+OABeta*(OAW-RAW);
            CCLAW=MAW;
        else
            if (RAW<RA2W)||(j<=0)
                RAW=(LoadLat/4840+CCLW*SA_CFM+Move_CFM*RAW_in)/...
                    (SA_CFM+Move_CFM);
                MAW=RAW+OABeta*(OAW-RAW);
            end
        end
        QCL= 4840*SA_CFM*(MAW-CCLAW); % Btu/hr
        QRCL= 4840*RA2_CFM*(RAW-RA2W); % Btu/hr
    else % secondary heating, no dehumidification
        RAW=(CCLAW*SA_CFM+LoadLat/4840+Move_CFM*RAW_in)/...
            (SA_CFM+Move_CFM);
        MAW=RAW+OABeta*(OAW-RAW);
        if (MAW<CCLAW)||(k<=0) % if the primary no dehumidification
either
            if (OA_CFM+Move_CFM)>0

RAW=(LoadLat/4840+OA_CFM*OAW+RAW_in*Move_CFM)/(OA_CFM+Move_CFM);
            end
            RA2W=RAW;
            MAW=RAW+OABeta*(OAW-RAW);
            CCLAW=MAW;
        end
        QRCL=0;
        QCL= 4840*SA_CFM*(MAW-CCLAW); % Btu/hr
        DAW=(RA2W*induRatio+CCLAW)/(1+induRatio);
    end
    RA2T= j/(AirCapRou60*RA2_CFM)+Ti_ex_set;
    DAT= (RA2T*induRatio+SAT_set)/(1+induRatio);
else % no dehumidification
    QCS=(k>1)*k;
    QH= (k<-1)*k;
    QRCS= (j>1)*j;
    QRH= (j<-1)*j;
    RA2T= j/(AirCapRou60*RA2_CFM)+Ti_ex_set;
    DAT= (RA2T*induRatio+SAT_set)/(1+induRatio);
```

```
    if (OA_CFM+Move_CFM)>0
    RAW=(LoadLat/4840+OA_CFM*OAW+RAW_in*Move_CFM)/(OA_CFM+Move_CFM);%
lbw/lba
    end
    RA2W=RAW;
    MAW=RAW+OABeta*(OAW-RAW);
    CCLAW=MAW;
    DAW=(RA2W*induRatio+CCLAW)/(1+induRatio);
    QRCL=0;QCL=0;
end

resl=[MAT,MAW,CCLAW,RA2T,RA2W,DAT,DAW,RAW,QCS,QCL,QH,QRCS,QRCL,QRH];
end
```

```
function interLoad=intLoad(area,OcuDensity)
% zone type B, refer to ASHRAE Handbook 1997, CHP 28, Table 35-38
Floor=area(1);
Int=area(2);
Ext=area(3);
density_people_in=0.00344.*OcuDensity;% #/sft
density_people_ex=0.00344.*OcuDensity;% #/sft
density_equip_in=1; % W/sft 2.2
density_equip_ex=1; % W/sft 2.2
density_Light_in= 0.028571429; % fixture/sft
density_Light_ex= 0.028571429; % fixture/sft
CLF_peop_Ocp=[0 0 0 0 0 0 0 0 0 0.65 0.75 0.81 0.89 0.91 0.93 0.95 0.31
0.22 0.17 0.13 0 0 0 0];
CLF_Lit_Ocp=[0 0 0 0 0 0 0 0 0 0.86 0.93 0.96 0.97 0.98 0.98 0.98 0.98
0.98 0.98 0.98 0 0 0 0];
CLF_Unocp=ones(1,24);
%% for interior zone
% occupied hours (9-17)
Light_Ocp_in = density_Light_in*50*0.9;% w/sft
Equip_Ocp_in = density_equip_in*1*0.95;% w/sft
Peop_Ocp_Sens_in= density_people_in.*75*0.95;% w/sft
Peop_Ocp_Lat_in= density_people_in.*75*0.95;% w/sft
%  unoccupied hours (18-8)
Light_Unocp_in = density_Light_in*40*0.1;% w/sft
Equip_Unocp_in = density_equip_in*1*0.05;% w/sft
Peop_Unocp_Sens_in= density_people_in.*75*0.05;% w/sft
Peop_Unocp_Lat_in= density_people_in.*75*0.05;% w/sft
% over 24 hour course
Light_in=(Light_Unocp_in.*CLF_Unocp+Light_Ocp_in.*CLF_Lit_Ocp)';
Equip_in=(Equip_Unocp_in.*CLF_Unocp+Equip_Ocp_in.*CLF_peop_Ocp)';
Peop_sens_in=(Peop_Unocp_Sens_in.*CLF_Unocp+Peop_Ocp_Sens_in.*CLF_peop_
Ocp)';
Peop_lat_in=(Peop_Unocp_Lat_in.*CLF_Unocp+Peop_Ocp_Lat_in.*CLF_peop_Ocp
)';
Time=[1:1:24]';
%% for exterior zone
% occupied hours (9-17)
Light_Ocp_ex = density_Light_ex*50*0.9;% w/sft
Equip_Ocp_ex = density_equip_ex*1*0.95;% w/sft
```

```
Peop_Ocp_Sens_ex= density_people_ex.*75*0.95;% w/sft
Peop_Ocp_Lat_ex= density_people_ex.*75*0.95;% w/sft
%  unoccupied hours (18-8)
Light_Unocp_ex = density_Light_ex*40*0.1;% w/sft
Equip_Unocp_ex = density_equip_ex*1*0.05;% w/sft
Peop_Unocp_Sens_ex= density_people_ex.*75*0.05;% w/sft
Peop_Unocp_Lat_ex= density_people_ex.*75*0.05;% w/sft
% over 24 hour course
Light_ex=(Light_Unocp_ex.*CLF_Unocp+Light_Ocp_ex.*CLF_Lit_Ocp)';
Equip_ex=(Equip_Unocp_ex.*CLF_Unocp+Equip_Ocp_ex.*CLF_peop_Ocp)';
Peop_sens_ex=(Peop_Unocp_Sens_ex.*CLF_Unocp+Peop_Ocp_Sens_ex.*CLF_peop_
Ocp)';
Peop_lat_ex=(Peop_Unocp_Lat_ex.*CLF_Unocp+Peop_Ocp_Lat_ex.*CLF_peop_Ocp
)';
Time=[1:1:24]';
%% interior zone
Int_sens_in=Light_in+Equip_in+Peop_sens_in; % W/sft
%Int_sens_in=(Light+Equip+Peop_sens)*Int/Floor; % W/sft
Int_lat_in=Peop_lat_in; % W/sft
%Int_lat_in=Peop_lat*Int/Floor; % W/sft
%% exterior zone
Int_sens_ex=Light_ex+Equip_ex+Peop_sens_ex;% W/sft
%Int_sens_ex=(Light+Equip+Peop_sens)*Ext/Floor; % W/sft
Int_lat_ex=Peop_lat_ex; % W/sft
%Int_lat_ex=Peop_lat*Ext/Floor; % W/sft

%% overall output
interLoad.int=[Time, Int_sens_in, Int_lat_in];
interLoad.ext=[Time, Int_sens_ex, Int_lat_ex];




function
resl=intVAV(OA_CFM,OAW,OA_temp,SAT_set,CCLW,SA_CFM,Ti_set,LoadSen,LoadL
at)
% a subfunction for single duct VAV simulation
AirCapRou60=1.06; % 60*rou*Cp: Btu min/ft^3 F hr . Energy Btu/hr=
AirCap60*CFM*delt
W2BTUhr=3.412;
OABeta=OA_CFM/SA_CFM;
QCL=0;QCS=0;QH=0;heat=0;QRH=0;

MAT=OABeta*OA_temp+(1-OABeta)*Ti_set;

if abs((MAT-SAT_set))>0.01
heat=AirCapRou60*SA_CFM*(MAT-SAT_set); % Btu/hr
end
if (MAT-SAT_set)<0
QH=heat;
else
QCS=heat;
end

k=LoadSen-AirCapRou60*SA_CFM*(Ti_set-SAT_set);
```

```
if abs(k)>0.01
    QRH=k;
end

if (OA_temp>55)
RAW=(LoadLat)/(4840*SA_CFM)+CCLW;
MAW=RAW+OABeta*(OAW-RAW);
CCLAW=CCLW;

    if (MAW<CCLAW)||(heat<=0)
        if OABeta>0
        RAW=(LoadLat)/(4840*SA_CFM*OABeta)+OAW;
        end
        MAW=RAW+OABeta*(OAW-RAW);
        CCLAW=MAW;
    end
QCL=4840*SA_CFM*(MAW-CCLAW); % Btu/hr
else
    if OABeta>0
    RAW=(LoadLat)/(4840*SA_CFM*OABeta)+OAW;
    end
    MAW=RAW+OABeta*(OAW-RAW);
    CCLAW=MAW;
    QCL=0;
end

DAT=-QRH/(AirCapRou60*SA_CFM)+SAT_set;

resl=[MAT,MAW,CCLAW,DAT,RAW,QCS,QCL,QH,QRH];
end
```

Appendix B:

Omaha is in climate zone 5A, latitude: 41.3, longitude: -95.9, heating: -2ºF (99%), cooling: 92ºF (1%).

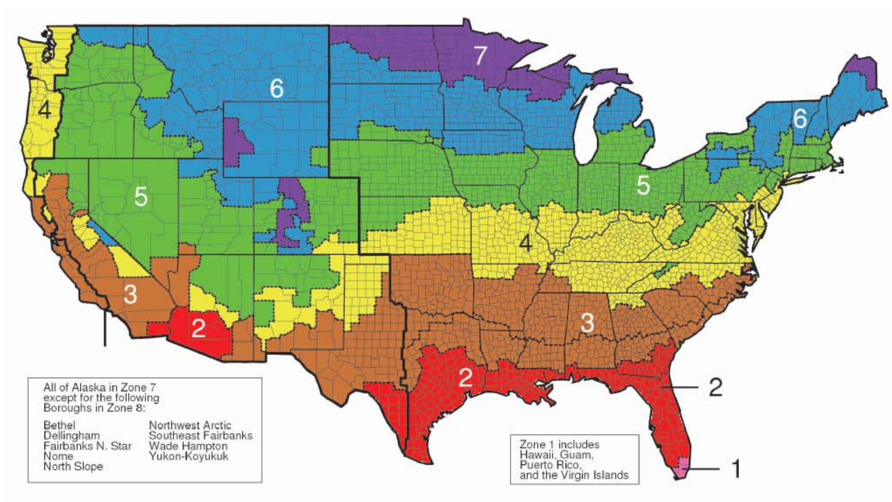

Figure B-0-1 DOE Climate zone map. Source: ASHRAE 90.1-2004

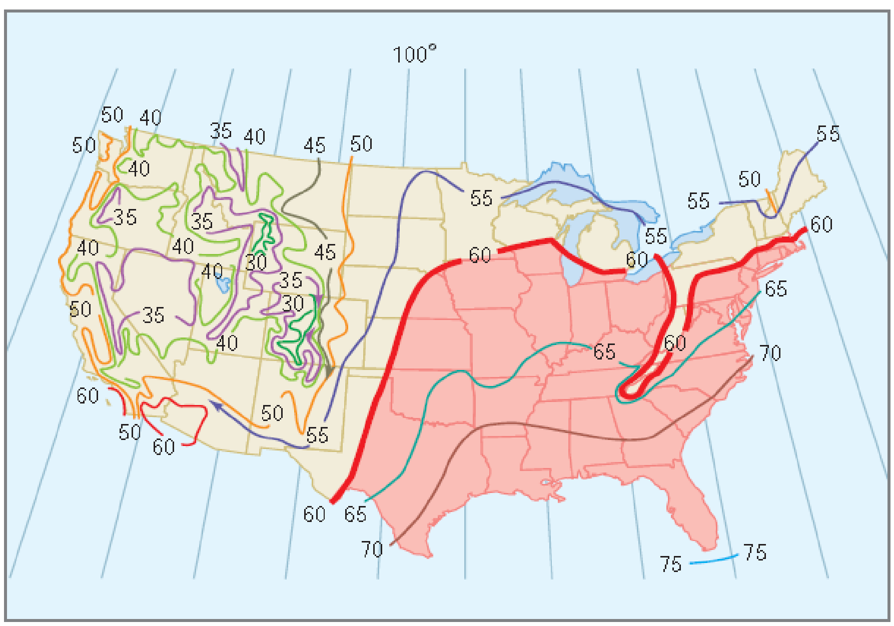

FIGURE 1. Mean dew-point temperature isolines for August (1946 to 1965).
Source: Climatic Atlas of the United States.

Figure B-0-2 Mean dew-point temperature isolines for August

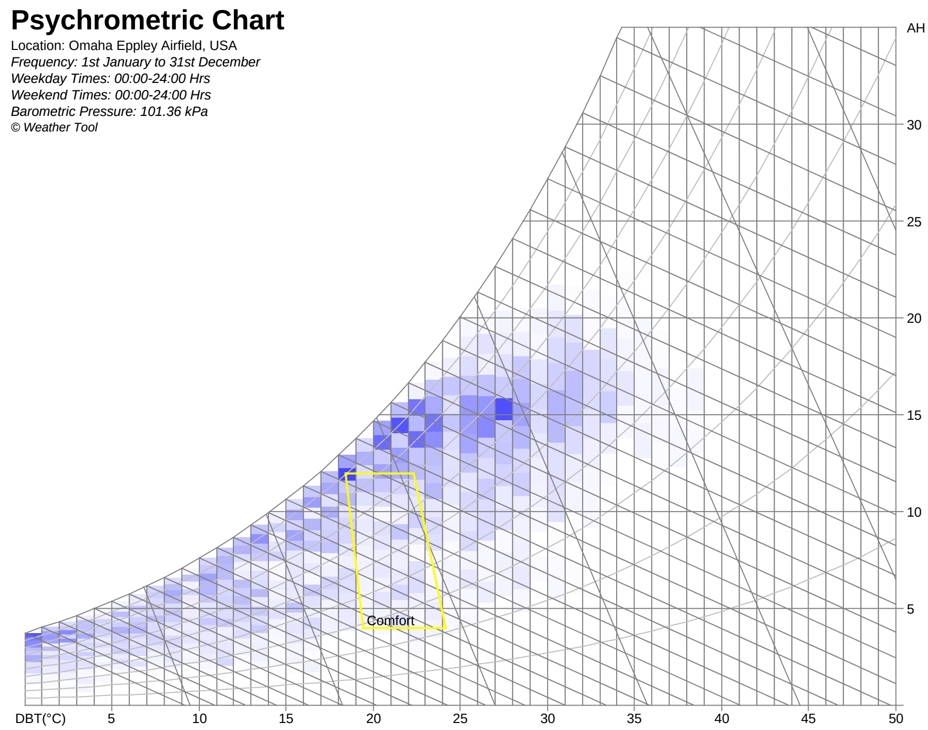

Figure B-0-3: Omaha OA condition psychrometric chart, yearly

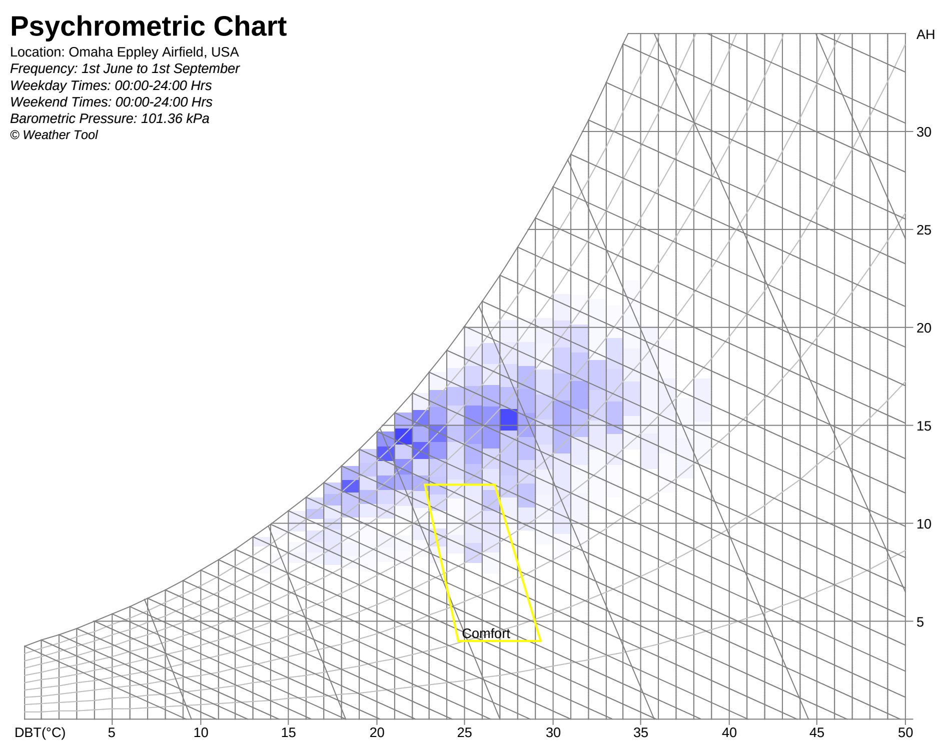

Figure B-0-4: Omaha OA condition psychrometric chart, summer

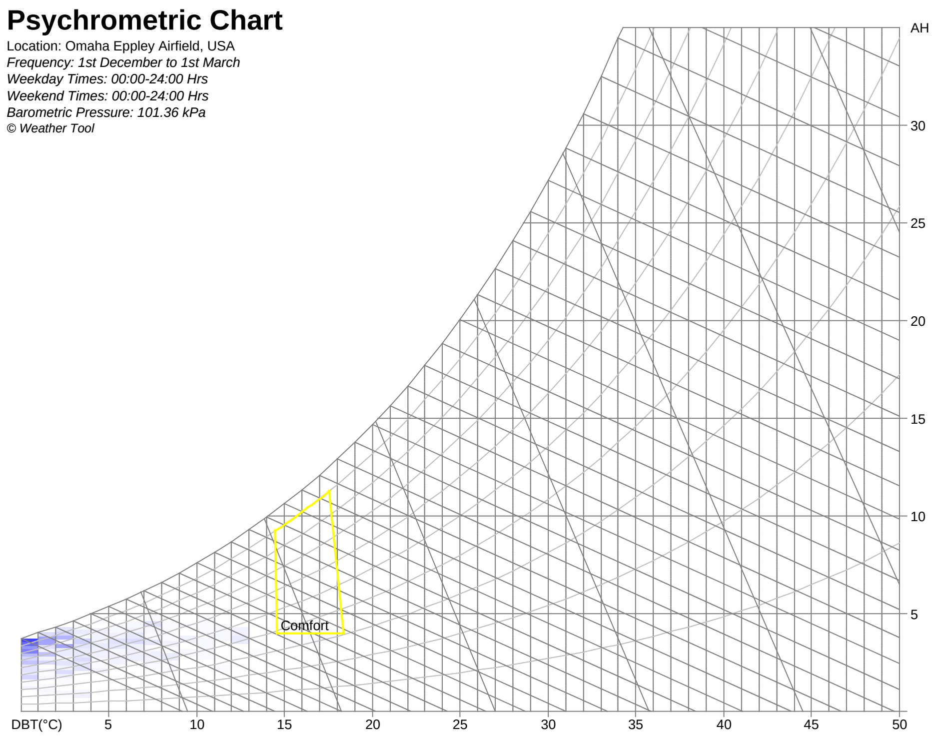

Figure B-0-5: Omaha OA condition psychrometric chart, winter

Figure B-0-6: Case building side view

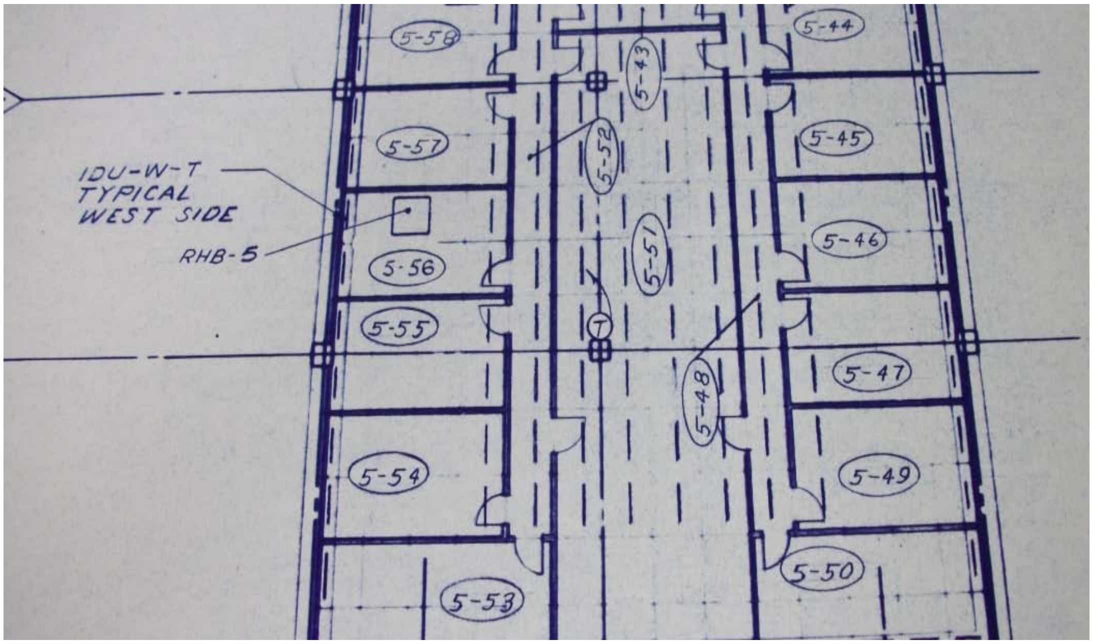

Figure B-0-7: Floor drawing for slot diffusers and induction units- 1

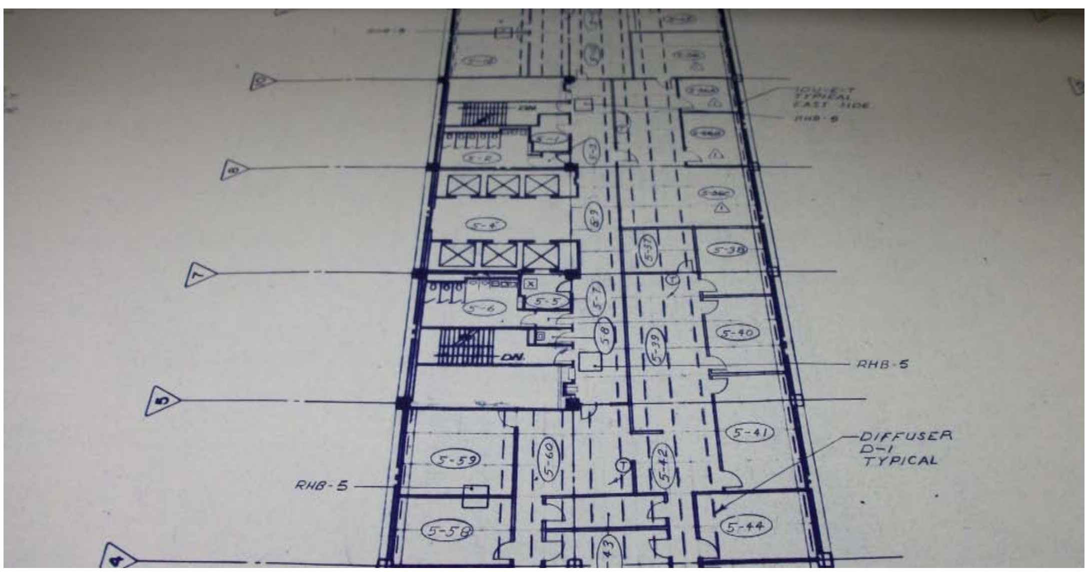

Figure B-0-8: Floor drawing for slot diffusers and induction units- 2

Figure B-0-9: One mechanical room view

Figure B-0-10: An induction unit

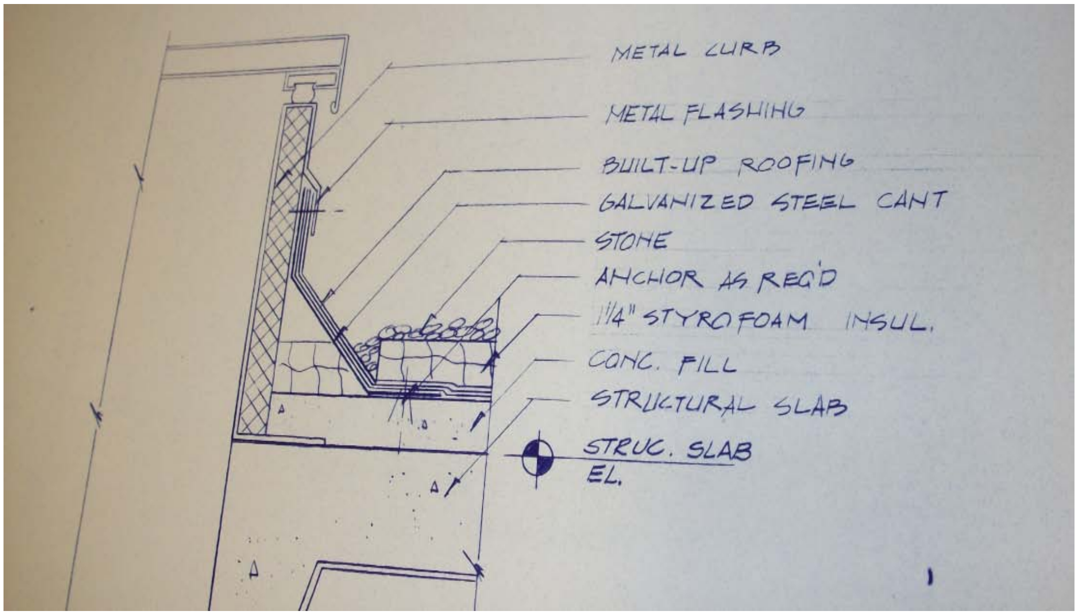

Figure B-0-11: The roof construction

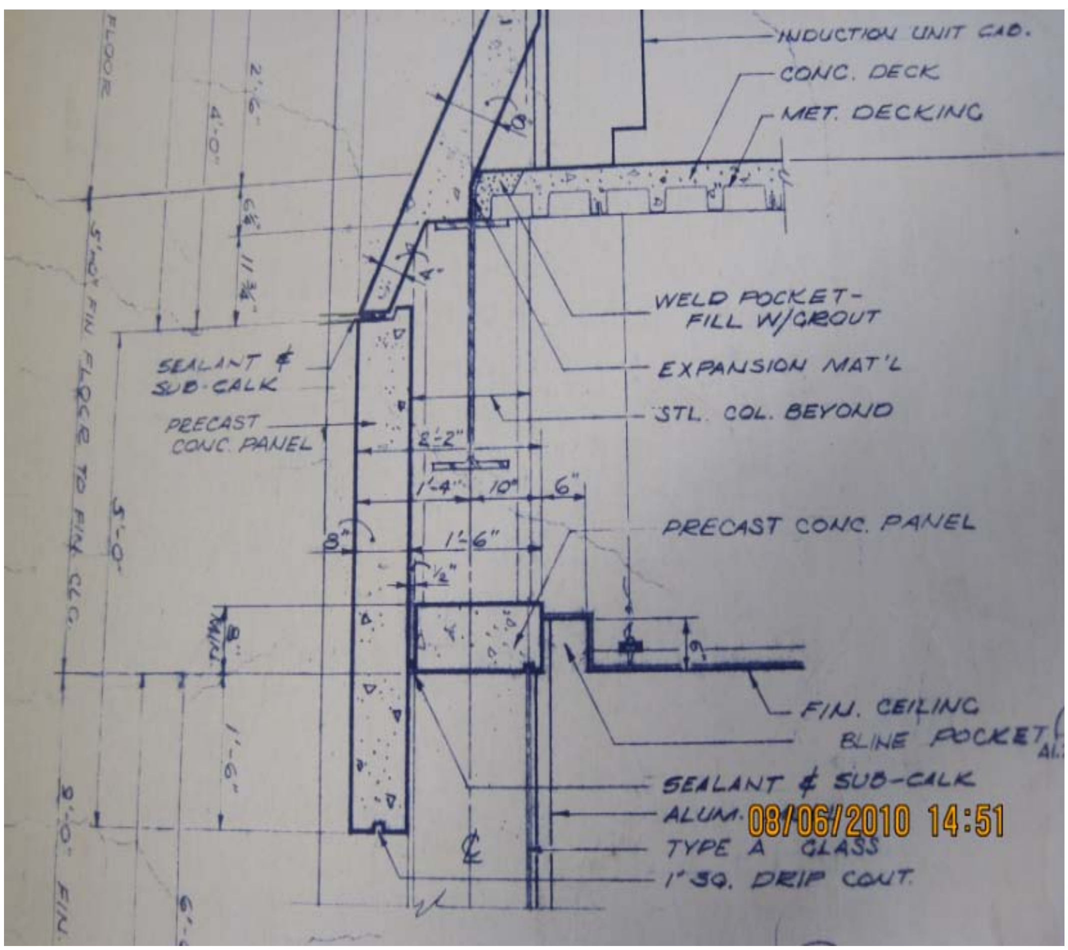

Figure B-0-12: The building wall layout

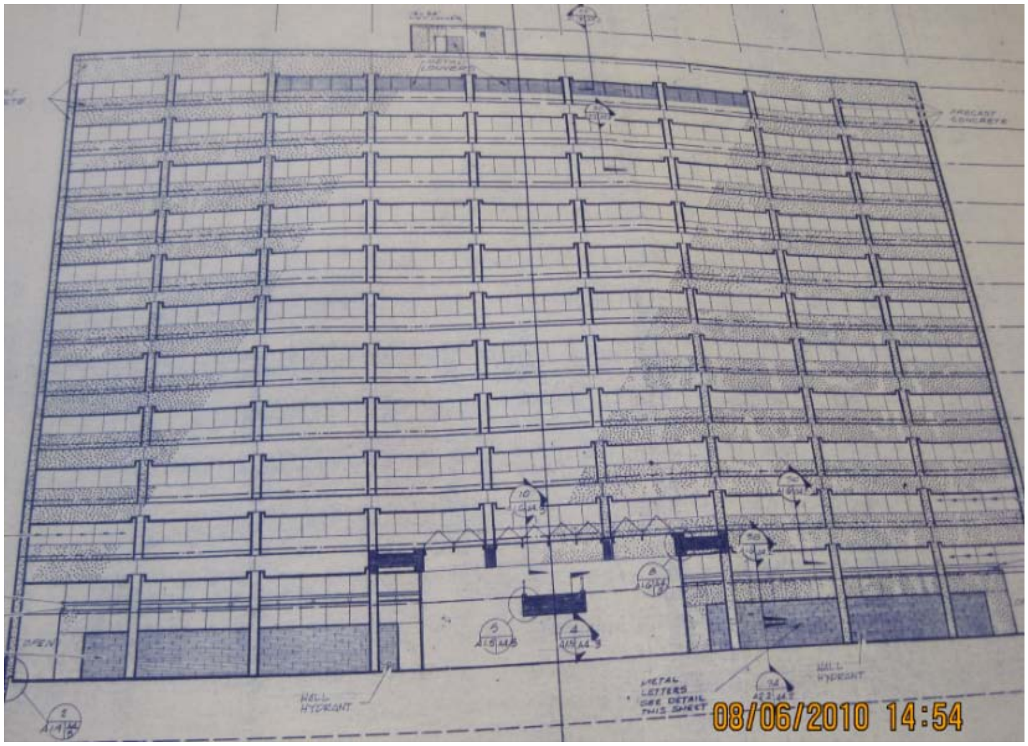

Figure B-0-13: The building elevation

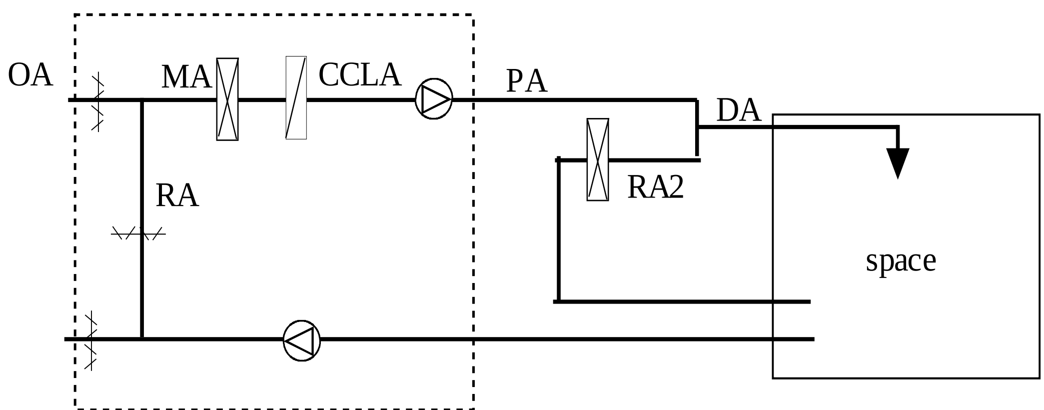

Figure B-0-14: CAV induction unit system illustration

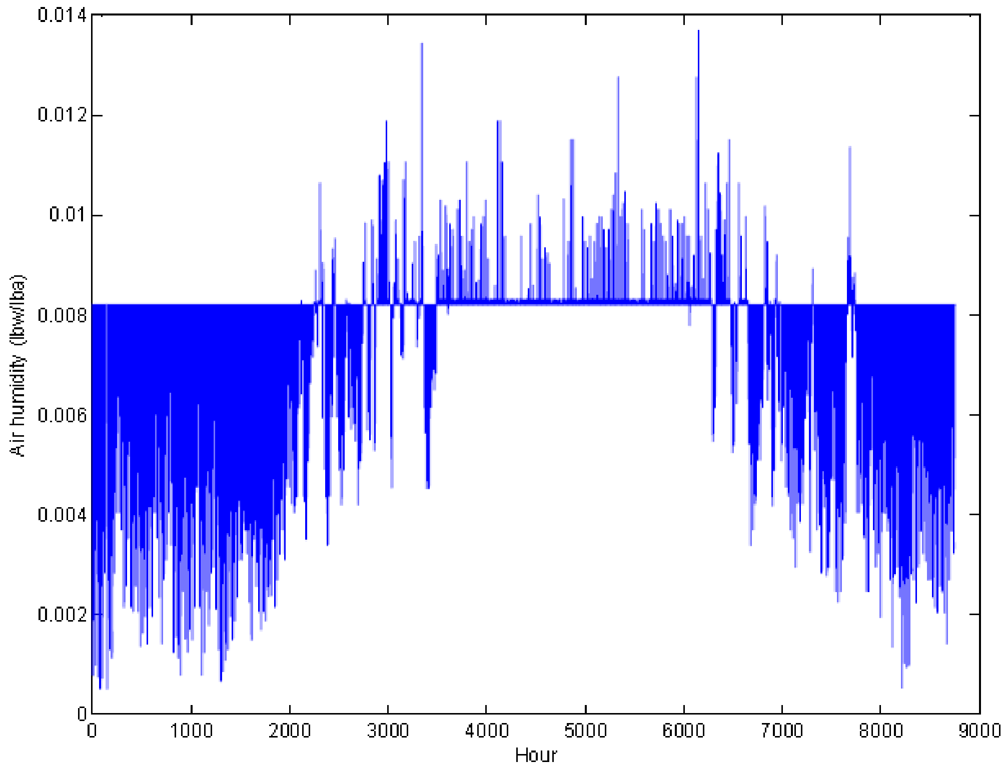

Figure B-0-15: Simulated hourly room air humidity, with x=0.4

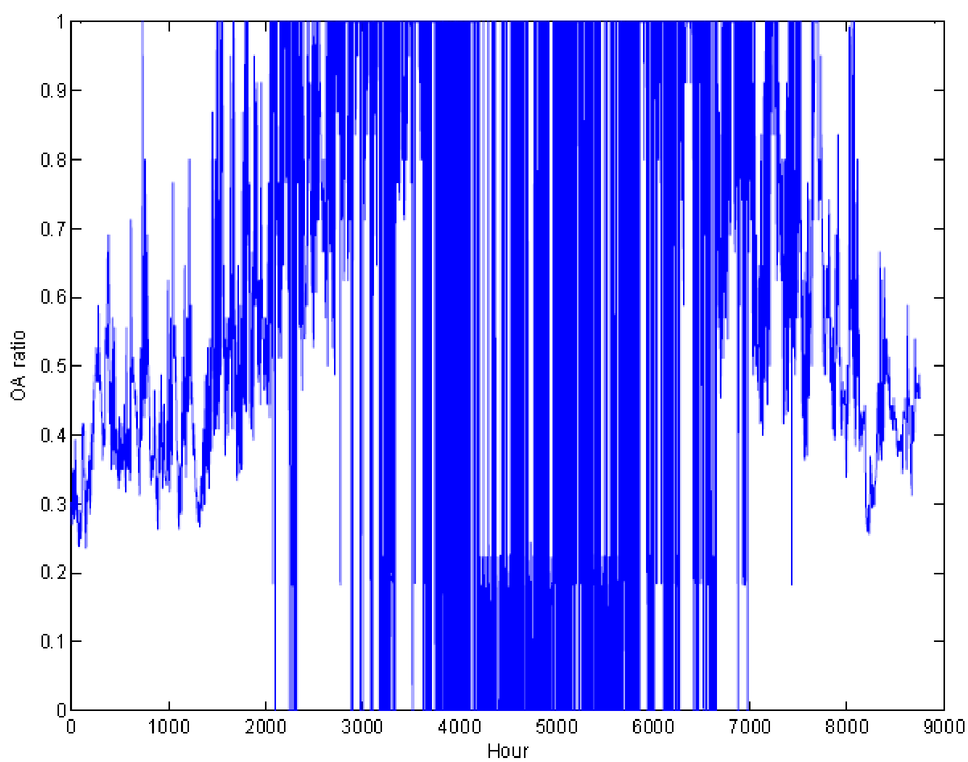

Figure B-0-16: OA intake ratio for the interior zone AHU under IAHU mode, x=0.4

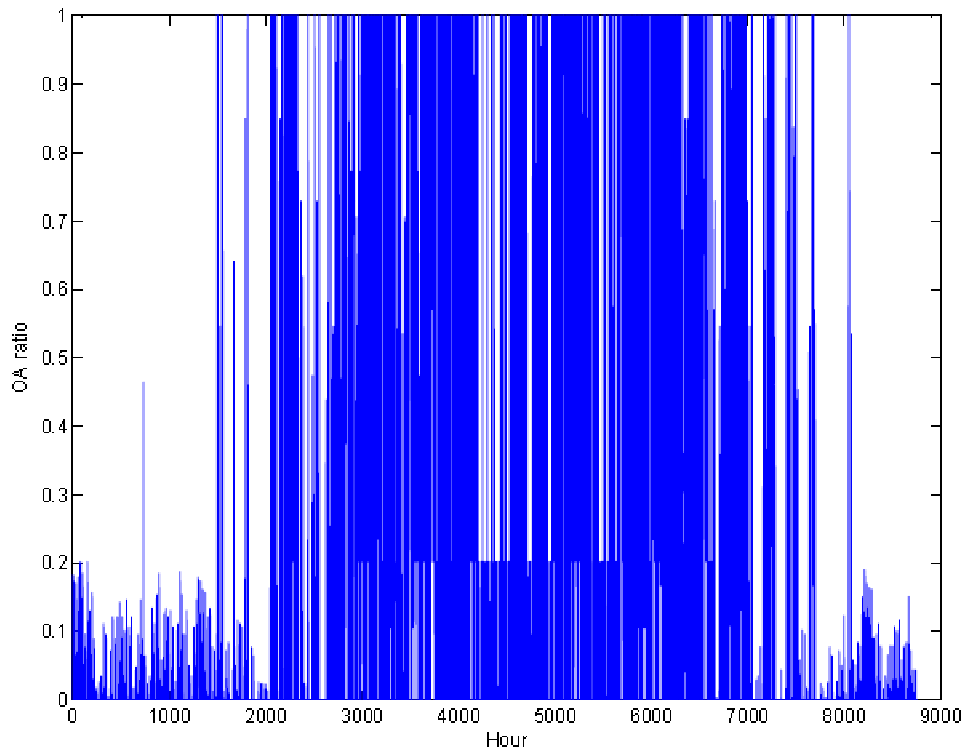

Figure B-0-17: OA intake ration for the exterior zone AHU under IAHU mode, x=0.4

Table B-0-1: Historical energy consumption for the whole case building

Chilled Water and Steam Data						
Date Read	**Month**	**Days**	**Ton-hr**	**Cost (CW)**	**Steam,lbs**	**Cost (Steam)**
01/31/05	Jan	31	0	$ 100.00	1,893,900	$ 34,655.69
02/28/05	Feb	28	0	$ 100.00	1,305,800	$ 28,935.43
03/31/05	Mar	31	0	$ 100.00	1,154,000	$ 22,660.52
04/30/05	Apr	30	36,948	$ 8,191.37	289,551	$ 9,331.85
05/31/05	May	31	86,286	$ 23,817.32	129,115	$ 3,607.12
06/30/04	Jun	30	145,556	$ 36,592.01	350,100	$ 5,993.82
07/31/04	Jul	31	205,766	$ 58,529.35	228,825	$ 5,191.56
08/31/04	Aug	31	178,265	$ 42,651.74	133,113	$ 2,441.03
09/30/04	Sep	30	140,764	$ 40,447.57	102,459	$ 1,791.62
10/31/04	Oct	31	38,403	$ 13,673.84	214,831	$ 3,193.79
11/30/04	Nov	30	6,314	$ 3,692.97	664,000	$ 9,965.23
12/31/04	Dec	31	0	$ 814.91	1,331,400	$ 23,272.36