International Energy Agency
Programme on
Energy in Buildings and Communities

Annex 59: High Temperature Cooling & Low Temperature Heating in Buildings

Final Report
I. Guide book of new analysis method for HVAC system

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Preface

The International Energy Agency

The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Co-operation and Development (OECD) to implement an international energy programme. A basic aim of the IEA is to foster international co-operation among the 30 IEA participating countries and to increase energy security through energy research, development and demonstration in the fields of technologies for energy efficiency and renewable energy sources.

The IEA Energy in Buildings and Communities Programme

The IEA co-ordinates international energy research and development (R&D) activities through a comprehensive portfolio of Technology Collaboration Programmes. The mission of the IEA Energy in Buildings and Communities (IEA EBC) Technology Collaboration Programme is to develop and facilitate the integration of technologies and processes for energy efficiency and conservation into healthy, low emission, and sustainable buildings and communities, through innovation and research. (Until March 2013, the IEA EBC Programme was known as the IEA Energy Conservation in Buildings and Community Systems Programme, ECBCS.)

The R&D strategies of the IEA EBC Programme are derived from research drivers, national programmes within IEA countries, and the IEA Future Buildings Forum Think Tank Workshops. These R&D strategies aim to exploit technological opportunities to save energy in the buildings sector, and to remove technical obstacles to market penetration of new energy efficient technologies. The R&D strategies apply to residential, commercial, office buildings and community systems, and will impact the building industry in five areas of focus for R&D activities:

- Integrated planning and building design
- Building energy systems
- Building envelope
- Community scale methods
- Real building energy use

The Executive Committee

Overall control of the IEA EBC Programme is maintained by an Executive Committee, which not only monitors existing projects, but also identifies new strategic areas in which collaborative efforts may be beneficial. As the Programme is based on a contract with the IEA, the projects are legally established as Annexes to the IEA EBC Implementing Agreement. At the present time, the following projects have been initiated by the IEA EBC Executive Committee, with completed projects identified by (*) and joint projects with the IEA Solar Heating and Cooling Technology Collaboration Programme by (☼):

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Annex 2: Ekistics and Advanced Community Energy Systems (*)
Annex 3: Energy Conservation in Residential Buildings (*)
Annex 4: Glasgow Commercial Building Monitoring (*)
Annex 5: Air Infiltration and Ventilation Centre
Annex 6: Energy Systems and Design of Communities (*)
Annex 7: Local Government Energy Planning (*)
Annex 8: Inhabitants Behaviour with Regard to Ventilation (*)
Annex 9: Minimum Ventilation Rates (*)
Annex 10: Building HVAC System Simulation (*)
Annex 11: Energy Auditing (*)
Annex 12: Windows and Fenestration (*)
Annex 13: Energy Management in Hospitals (*)
Annex 14: Condensation and Energy (*)

Annex 15: Energy Efficiency in Schools (*)
Annex 16: BEMS 1- User Interfaces and System Integration (*)
Annex 17: BEMS 2- Evaluation and Emulation Techniques (*)
Annex 18: Demand Controlled Ventilation Systems (*)
Annex 19: Low Slope Roof Systems (*)
Annex 20: Air Flow Patterns within Buildings (*)
Annex 21: Thermal Modelling (*)
Annex 22: Energy Efficient Communities (*)
Annex 23: Multi Zone Air Flow Modelling (COMIS) (*)
Annex 24: Heat, Air and Moisture Transfer in Envelopes (*)
Annex 25: Real time HVAC Simulation (*)
Annex 26: Energy Efficient Ventilation of Large Enclosures (*)
Annex 27: Evaluation and Demonstration of Domestic Ventilation Systems (*)
Annex 28: Low Energy Cooling Systems (*)
Annex 29: ☼ Daylight in Buildings (*)
Annex 30: Bringing Simulation to Application (*)
Annex 31: Energy-Related Environmental Impact of Buildings (*)
Annex 32: Integral Building Envelope Performance Assessment (*)
Annex 33: Advanced Local Energy Planning (*)
Annex 34: Computer-Aided Evaluation of HVAC System Performance (*)
Annex 35: Design of Energy Efficient Hybrid Ventilation (HYBVENT) (*)
Annex 36: Retrofitting of Educational Buildings (*)
Annex 37: Low Exergy Systems for Heating and Cooling of Buildings (LowEx) (*)
Annex 38: ☼ Solar Sustainable Housing (*)
Annex 39: High Performance Insulation Systems (*)
Annex 40: Building Commissioning to Improve Energy Performance (*)
Annex 41: Whole Building Heat, Air and Moisture Response (MOIST-ENG) (*)
Annex 42: The Simulation of Building-Integrated Fuel Cell and Other Cogeneration Systems (FC+COGEN-SIM) (*)
Annex 43: ☼ Testing and Validation of Building Energy Simulation Tools (*)
Annex 44: Integrating Environmentally Responsive Elements in Buildings (*)
Annex 45: Energy Efficient Electric Lighting for Buildings (*)
Annex 46: Holistic Assessment Tool-kit on Energy Efficient Retrofit Measures for Government Buildings (EnERGo) (*)
Annex 47: Cost-Effective Commissioning for Existing and Low Energy Buildings (*)
Annex 48: Heat Pumping and Reversible Air Conditioning (*)
Annex 49: Low Exergy Systems for High Performance Buildings and Communities (*)
Annex 50: Prefabricated Systems for Low Energy Renovation of Residential Buildings (*)
Annex 51: Energy Efficient Communities (*)
Annex 52: ☼ Towards Net Zero Energy Solar Buildings (*)
Annex 53: Total Energy Use in Buildings: Analysis and Evaluation Methods (*)
Annex 54: Integration of Micro-Generation and Related Energy Technologies in Buildings (*)
Annex 55: Reliability of Energy Efficient Building Retrofitting - Probability Assessment of Performance and Cost (RAP-RETRO) (*)
Annex 56: Cost Effective Energy and CO_2 Emissions Optimization in Building Renovation (*)
Annex 57: Evaluation of Embodied Energy and CO_2 Equivalent Emissions for Building Construction (*)
Annex 58: Reliable Building Energy Performance Characterisation Based on Full Scale Dynamic Measurements (*)
Annex 59: High Temperature Cooling and Low Temperature Heating in Buildings (*)
Annex 60: New Generation Computational Tools for Building and Community Energy Systems (*)
Annex 61: Business and Technical Concepts for Deep Energy Retrofit of Public Buildings (*)
Annex 62: Ventilative Cooling (*)
Annex 63: Implementation of Energy Strategies in Communities (*)

Annex 64: LowEx Communities - Optimised Performance of Energy Supply Systems with Exergy Principles (*)
Annex 65: Long-Term Performance of Super-Insulating Materials in Building Components and Systems
Annex 66: Definition and Simulation of Occupant Behavior in Buildings (*)
Annex 67: Energy Flexible Buildings
Annex 68: Indoor Air Quality Design and Control in Low Energy Residential Buildings
Annex 69: Strategy and Practice of Adaptive Thermal Comfort in Low Energy Buildings
Annex 70: Energy Epidemiology: Analysis of Real Building Energy Use at Scale
Annex 71: Building Energy Performance Assessment Based on In-situ Measurements
Annex 72: Assessing Life Cycle Related Environmental Impacts Caused by Buildings
Annex 73: Towards Net Zero Energy Resilient Public Communities
Annex 74: Competition and Living Lab Platform
Annex 75: Cost-effective Building Renovation at District Level Combining Energy Efficiency and Renewables
Annex 76: ☼ Deep Renovation of Historic Buildings Towards Lowest Possible Energy Demand and CO_2 Emissions
Annex 77: ☼ Integrated Solutions for Daylight and Electric Lighting
Annex 78: Supplementing Ventilation with Gas-phase Air Cleaning, Implementation and Energy Implications
Annex 79: Occupant Behaviour-Centric Building Design and Operation
Annex 80: Resilient Cooling

Working Group - Energy Efficiency in Educational Buildings (*)
Working Group - Indicators of Energy Efficiency in Cold Climate Buildings (*)
Working Group - Annex 36 Extension: The Energy Concept Adviser (*)
Working Group - HVAC Energy Calculation Methodologies for Non-residential Buildings
Working Group - Cities and Communities
Working Group - Building Energy Codes
Working Group - International Building Materials Database

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Contents

Nomenclature

A	heat or mass transfer area (m^2)
COP	coefficient of performance (dimensionless)
c_p	specific heat capacity (kJ/kg·ºC)
E_n	entransy (kW·K)
E_x	exergy (kW)
ΔE_n	entransy dissipation (kW·K)
m	mass flow rate (kg/s)
Δp	pressure drop (kPa)
Q (q)	heat flux or cooling capacity (kW)
R	equivalent thermal resistance (K/W)
T	absolute temperature (K)
ΔT	temperature difference (ºC)
t	temperature (ºC)
U	heat transfer coefficient (kW/m^2·ºC)
$\dot{V}$	volume flow rate (m^3/s)
W	power consumption (kW)
Greek symbols	
ξ	unmatched coefficient (dimensionless)
ρ	density (kg/m^3)
η	thermodynamic perfectness (dimensionless)
ε	efficiency of fans or pumps (dimensionless)
Subscripts	
a	air
ac	air-conditioning system
CH	chiller (mechanical refrigeration cycle)
c	low temperature fluid
c ($cond$)	condenser
CH	chiller
$conv$	convection
des	exergy destruction
dis	entransy dissipation
e ($evap$)	evaporator
h	high temperature fluid
hp	heat pump
$ideal$	ideal system
in	inlet state
mix	mixing process
out	outlet state
r	radiant floor
ra	radiation
sa	supply air
tot	total heat
w	water

PART 1

1. Introduction and brief history of HVAC systems

Since primordial times human beings looked for solutions able to modify indoor climate conditions and to provide better and more comfortable living spaces. Examples of basic and rough technologies, able to slightly modify and control the hygrothermal conditions inside confined space and buildings, can be found in most of the ancient civilizations like Egyptians, Greeks, Chinese and Romans [1]. Since that era, two were the main problems that people wanted to address: provide some heating during the cold season and cool down the environment during summer periods. As a further and secondary measure, the handling of the relative humidity was sometimes of interest. At the beginning of the history people just used fire for heating. The second concern that emerged was that of indoor air quality. By the late 1880s, rules of thumbs for ventilation were proposed [2].

The next step in people expectations was that of cooling; primarily for food conservation and, just after that, for comfort. In 1851 Dr. John Gorrie patented the first refrigeration machine for industrial purposes. By the beginning of the 1900s engineers started to think using refrigeration systems for cooling the buildings with the aim of providing comfort for human beings. With this last evolution came the realization that just cooling the air was not enough and that serious issues about moisture might happen when the built environment is only cooled and relative humidity is not controlled. It started the era in which it was considered mandatory to cool and dehumidify the air during the summer periods. At first, the term "air conditioning" was used to identify the comfort cooling of buildings, but it gradually changed its meaning during the time. Today "air conditioning" or, better, "full air conditioning" [2] denotes and identify the total control of:

- Temperature,
- Relative humidity,
- Ventilation
- Filtration of airborne particles
- Air movement in the occupied space,

while the acronym "HVAC" is typically used where only some of the above listed items are being controlled.

It is worth noting that the space heating may or may not include the humidification of air (most of the residential buildings are only heated, but the latent load is seldom controlled) while the space cooling practically always includes a dehumidification process of some sort. Moreover, the technologies and measures that were typically used to heat and cool a building were substantially different (and, frequently, they are still different). For these reasons the heating and cooling systems will be examined separately in the following part of this paragraph. In particular, the attention will be focused on the general configuration of the installations and on the type and usual temperature levels of the heat carrier fluids.

Heating systems

For centuries and millennia, fire based apparatuses have been used for heating purposes. At the origins, and as long as the nineteen century, direct fire heating systems (stoves, braziers, fireplaces) were the most widespread method to – roughly – control the indoor air temperature in confined spaces. Looking at the history it is also possible to find more sophisticated systems, which can be considered as precursors of the modern installations. For example, Romans developed the so – called hypocaustum a precursor of the heating radiant systems. It was a kind of central heating system based on a combustion furnaces that produced hot flue gases. These fumes were then flown inside purposely realized cavities in walls and floors and warmed up the indoor environment by means of convection and radiation.

At the origins the home heating was done by means of braziers and fireplaces. These appliances were simple and easy to install, however they were rough, inefficient and relatively unsafe. The combustion happened in a non-confined volume and was uncontrolled. Flue gasses were frequently released inside the living space and the correct management of the fuel and of the combustion air was not possible. The first measure was, therefore, that of "boxing up" the combustion reaction inside a purposely provided space, thus physically separating the heat production from the people. It is the birth of the fire stove. In 1742 Franklin invented the cast iron stove, but brick and earthenware stoves were already in use. This first step allowed to improve the air quality inside the buildings, to enhance (to some extent) the efficiency of the heating systems and to provide a more safety way to warm the houses.

At the beginning of the '800 more developed solutions appeared. A furnace for heating air was used in England. This installation had pipes inside which hot air could be distributed even in large spaces and factories. Nevertheless, even in these "sophisticated" configurations, the space heating was still represented by a direct fire heating systems and both the type of heat carrier fluid and its temperature levels did not changed significantly. If we look at all these apparatuses through the perspective of the temperature level and type of the heat carrier fluid, we can see some common features:

- the fluid was essentially constituted by hot fumes,
- the temperature levels of the heat carrier fluids were of the order of magnitude of some hundreds of degree Centigrade (roughly between 300 °C to 800 °C)
- the indoor air temperature (that was very dependent on the climate context) was less than 20 °C (due to the poor performance of the building envelopes of that time and the inefficiencies of the installations).

The first evolution of the heating systems was driven by the growth of the size of the buildings. In fact, one of the main constraints deriving from the use of stove, fireplaces and direct fire heating was represented by the great difficulty of satisfactorily control the winter temperature inside big building or complex of buildings. To overcome this barrier the next step was represented by central heating systems. The end of the 19th century sees the invention of cast iron radiators that open the doors to a mass diffusion of central heating plants. In these systems a coal fired boiler in the basement of the building produced steam or hot water. Such heat carrier fluids were then distributed in every room and released to the environment by means of radiators. Natural (free) convection phenomena (based on buoyancy effects) were used to move the heat carrier fluids through ducts from the basement furnace to the heating terminals in the rooms above. Natural convection was adopted because, at that time, electricity was not widespread and fans and pumps were expensive and unreliable. This configuration would dominate home central heating until 1935, when the introduction of the first forced air furnace using coal as a heat source used the power of an electric fan to distribute the heated air through ductwork within the home [3]. The necessity of providing a sufficient pressure difference between the supply and return of the fluid and the inefficiencies of the heat exchange in cast iron radiators implied the adoption of high temperatures of the hot water and/or the use of steam instead of water. That meant to have heat carrier fluids with temperature of about 90~120 °C (pressurized steam). At the same time, the possibility of installing bigger and more functional plants, combined with a better thermal insulation of walls and windows, allowed an improved control of the indoor air temperature in buildings and was therefore possible to provide a more comfortable temperature of about 20 °C.

Shortly thereafter, coal and cast iron radiators were replaced by gas and oil fired versions of forced air furnaces. Thanks to the technological improvement of the pumps, the heating systems switched from natural to forced convection. This allowed on the one hand to improve the heat exchange in the terminals and, on the other, to free the configuration of the plant (that no more needs to have a furnace necessarily located in the basement and the heating terminals above). As a result the heating systems started to be adopted, almost always, hot water as heat carrier fluid and to keep the supply temperature at about 80 °C (a typical return temperature was of about 60 °C). The general configuration of the heating systems did not change as long as the end of the 20th and the turn of the 21st century, even if the temperature level of the hot water slightly changed over the time. However, a tendency of decreasing the hot water temperature was shown. Between the nineties and the end of the 20th century, laws were issued in many countries that imposed a maximum hot water temperature of 60 °C (being the typical return temperature of about 40 °C).

With the turn of the millennium new technologies began to significantly penetrate the market of the heating systems. Specifically, radiant systems (floor and, later, ceilings) started to become popular and to challenge the position of the more traditional radiators. Though this technology was discovered and applied for the first time around the 1930, the scarce knowledge about the thermal comfort of people and the poor technological development of material and components (which made the system unreliable) prevented their widespread use for long time. It was necessary to wait for the years 1990 – 2000 to see a significant growth of such technologies. Particular importance in the development of radiant floor has to be ascribed to the improvement of the knowledge about the thermal comfort and, specifically, about the problems of local thermal discomfort. As well known (see e.g. ISO 7730 [4]), the maximum allowable surface temperature to satisfy the comfort conditions has to be of 29 °C. Such requirement has two important consequences: firstly it solved the problems of health and comfort related to the pristine applications (the systems adopted during the period 1930 – 1970 were operated at high temperatures, well above the recommended limit, and determined serious concerns for comfort and health), secondly it pushed towards lower temperature of the heat carrier fluids.

In a radiant floor the inlet temperature of the hot water has to be kept at values of about 35 – 32°C. If the dawn of radiant floor was fostered by considerations related to: better comfort, possibility of achieving local heating, no space occupation due to cumbersome heating terminals (like radiators), the possibility (or better, the necessity) of operating these extended terminals with hot water with low temperatures made this solution perfect to be matched with other emerging technologies. Such technologies are those that throw ourselves to the present. In particular, thanks to the possibility given by radiant floor (and, to less extent, to radiant ceilings) to operate at lower temperature levels (around 30 °C instead of 60 °C), a new way of producing heat became (and is becoming more and more) popular. It is the so-called "thermodynamic heating". Instead of producing heat by means of a combustion reaction, as it has been done for millennia, now, it is possible to deliver heat at the desired temperature level by means of a thermal machine. e.g. using an Heat Pump (HP).

Heat pumps can, in theory, work with whatsoever temperature of the heat carrier fluid, but their efficiency increases tremendously as much as the supply temperature decreases. This features explains why HP are typically coupled to radiant systems. The increased attention paid to a sustainable development, the necessity of further improving the energy efficiency of buildings and the goal of enhancing the exploitation of Renewable Energy Sources (RES) , makes, and will make, the use of heat pumps more and more attractive in the near future.

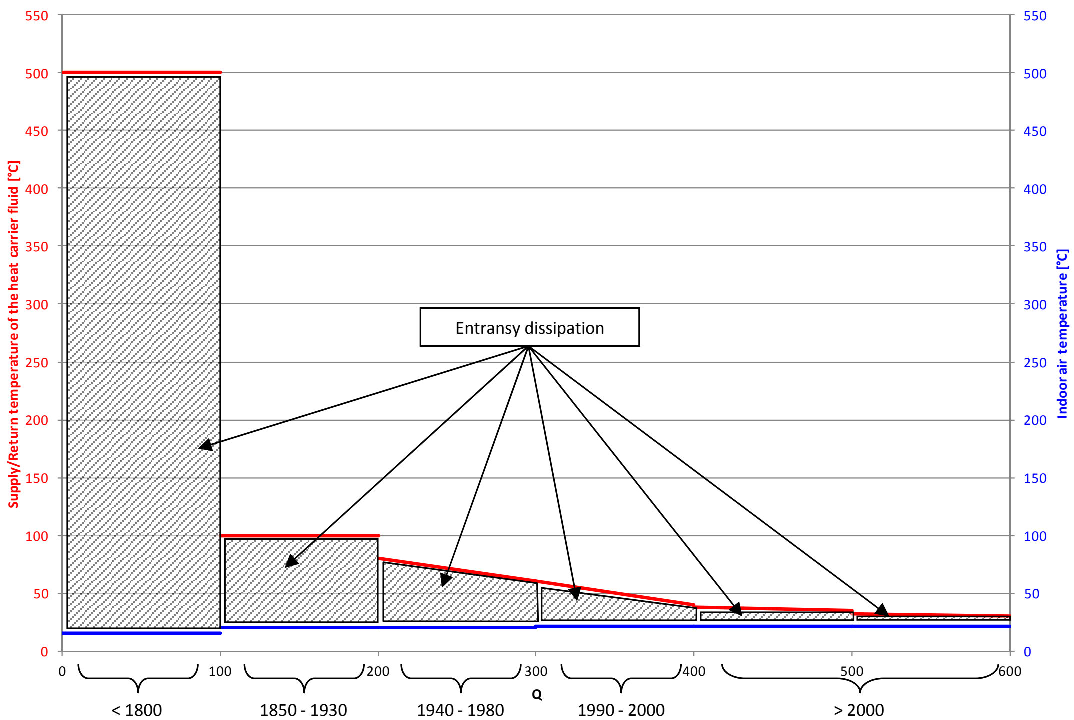


Fig. 1 Qualitative Temperature - Power chart for heating systems – Evolution over the time of the loss (qualitative trend)

In fact, these appliances operate absorbing electricity, that today can be produced by RES (like wind and solar photovoltaic), became more efficient during the time and, finally, can be used in combination with other measures to further improve their efficiency (like, e.g. , coupling them with the ground and/or ground water). This means that today and in the future one can envisage that the temperature of the heat carrier fluid will be around the minimum value possible, that is around 30 - 32 °C.

Cooling systems

The first examples of primordial cooling systems date back to Egyptian's and Indians who invented some sort of fans powered by man. It is also known that Leonardo da Vinci conceived a fan driven by a water wheel [1]. All these apparatuses were not exactly "cooling systems", they only raised the air speed and, hence, enhanced the convective heat transfer between the person and the indoor air. This measure provided a better thermal comfort through the increase of the heat exchange between the human body and the environment, but it did not modify the air temperature. During the history more sophisticated systems were used worldwide, like the wind catcher ("wind towers") or solar chimneys that used either the dynamic effect of the wind or the buoyancy phenomena to create a relatively high ventilation air flow rate. Again, such technologies allowed only to increase the air speed over the body of the occupants and/or removed part of the internal and solar loads, but did not modified the indoor temperature.

One of the first example of a system aimed at modifying the air temperature is represented by the so-called "stanze dello scirocco" in Sicily. Between the XVI and XVIII century, specially engineered rooms were built in the basements of the aristocratic palaces. These rooms were typically carved into the rock; a suitable chimney created an air flow by means of natural (wind and buoyancy driven) ventilation and a spring provided fresh water. The air flow, produced by means of natural ventilation, was cooled and humidified by flowing over the spring water and all along the wet surfaces of the rocky walls. It is one of the first examples of evaporative cooling.

The first attempts to cool objects and volumes at a temperature significantly below the ambient one were based on evaporation phenomena. Around year 1750 Benjamin Franklin and John Hadley (Cambridge University) discovered that evaporation of various liquids (alcohol and other volatile fluids) can cool objects at a point below the freezing of water.

However, it is only in the last hundred and fifty years that dedicated machineries were developed to artificially manufacture the indoor climate ("manufactured air" was the original definition of the method of controlling the relative humidity in textile mills) [5]. The beginning of the modern history of air conditioning starts with the engineering advances of the Industrial revolution.

In 1820 Michael Faraday discovered that it was possible to "generate cold" by compressing and then expanding ammonia. As already mentioned, the precursor of an air conditioning system is the apparatus invented by Dr. John Gorrie in 1830[1] for cooling

[1] The system was then patented in 1851.

hospital rooms in Florida. His machine was essentially a fan which blew across a container of ice, which was made by a compression cycle [5], [6]. In this case there wasn't yet a heat carrier fluid, as it is intended today, but the "source" used to cool the air (e.g ice) was a thermostat at 0 °C. The first fan coil dehumidifying system was made by a company called "Buffalo Forge" in the early 1900s. The same company made the first spray type air conditioner device [1]. Nevertheless, the first example of a modern comfort cooling systems is considered the machinery installed at the New York Stock Exchange Building in 1902 (designed by Alfred Wolff) [2]. In the same year:

- the first office building with an air-conditioning system was built in Kansas City, Missouri. It was the Armour Building, in which each room was individually controlled with a thermostat that operated dampers in the ductwork, making it also the first office building to incorporate individual "zone" control of separate rooms [7].
- Willis Carrier designed a spray driven temperature and humidity controlled system as well as a system for offices, apartments, hotels, and hospitals [5].

Carrier is universally recognized as the father of air conditioning and his company became synonymous with air conditioning excellence. He built his "Apparatus for Treating Air" where chilled cooling coils were used to cool the air and lower the humidity to as low as 55%. The device was even able to adjust the humidity level to a desired setting creating what would become the modern air conditioner. His studies were not only oriented toward solving the mechanical/technological issues, but he also gave fundamental contributions' in the theoretical study of moist air. In 1911 he presented to the American Society of Mechanical Engineers the 'Rational Psychometric Formula', which is still used today by the air conditioning industry.

In 1906 Stuart Cramer coined the term "air conditioning" for an appliance he developed for a textile farm.

But it is during the 1920s and 1930s that the mass-market refrigeration and air conditioning systems boomed. Self-contained refrigerators and air conditioning units started to be marketed in 1914. "Frigidaire" commercialized a central air conditioning appliance for homes in 1931, following its "room cooler" presented in 1929. Carrier made two fundamental breakthrough in 1922. The first one consisted in replacing the coolant ammonia with the much safer dichloroethylene [5]. The second was a significant reduction of the size of the chilling units.

CFCs were developed in 1928 and trade-marked as "Freon" in 1930.

From 1917 to 1930 movie theatres were the next focus of the industry, giving the average person the opportunity to experience for the first time the air conditioning [5].

In 1931 Schultz & Sherman invented the individual room air conditioner, that sits on a window ledge (a configuration that became ubiquitous). This system was commercialized an year later, but its cost (between 120'000 to 600'000 $ at the value of today) limited the spread of this machine only to wealthy people [6].

It is worth noting that, starting from the rough and primeval equipments, the history of cooling the buildings is closely intertwined with the US actuality and its timeline is strongly connected with the development of the electric energy. It is no coincidence that the first coal fired electric power plant opened in New York in 1882 and the first example of air conditioning systems appeared at the beginning of 1900s.

In fact, the first case of a nation building a power plant to cover “summer peak” of the electric energy demand happened in 1942 in the US.

The residential air conditioning, however, remained a luxury item for wealthy people until the post-world war II economic boom [3].

It is during the 1950s that the mass market of air conditioning devices exploded in the US and, slowly, it started to expand in the rest of the world.

The early 1950s saw the introduction of residential thru-windows and central air conditioning units. In 1946 it was estimated that 30’000 room air conditioners were manufactured; they became 74’00 in 1948 [5]. By 1953 room air conditioning sales exceeded 1 million of units. By 1998 shipments of air conditioners and reversible heat pumps score the record of 6.2 millions of units [3].

Today, it is estimated that about 20% of the electricity generated in the US is required for serving the HVAC systems [3].

The technological development that occurred between the end of the II world war and the present day has been mainly focused at improving the energy efficiency of the machineries and at lessening the environmental impact of the fluid used as coolants.

In 1980 Toshiba introduced in the market, for the first time, an inverter type compressor able to cope more efficiently the part loads (about 30 % better compared to single speed units) [6].

In 1987 the United Nations Montreal Protocol established to phase out the use of stratospheric ozone depleting substances, like the chlorofluorocarbon or CFC refrigerants used in a huge number of refrigeration and air conditioning systems. In 1995 the Clean Air act imposes to cease the manufacture of CFC-based chillers.

At the beginning of the 20th century radiant systems started to penetrate the market and to be used not only for heating, but also for cooling. Obviously the first concerns related with these technologies was that of avoiding local condensation of water vapor. To prevent such occurrence the heat carrier fluid temperatures must be significantly higher than those usually adopted, until that time (that is around 7 °C, if the secondary coolant is considered), for the air conditioning systems. As a rules of thumb values of the supply temperature between 17-19 °C can be assumed as a first guess for many climate conditions around the world.

This breakthrough allowed to operate chillers and reversible heat pumps at working conditions more favorable, to improve their energy efficiency and to reduce the temperature differences of the heat exchanges. Nevertheless, it has to be considered that the possibility of working at higher temperature levels only applies in practice if the chiller is servicing just the radiant panels. In fact, if dehumidification is required, one still needs to produce an heat carrier fluid at a lower temperature, as it will discussed further on.

Finally, technology based on desiccant cooling started to be investigated and applied. Such technology represents a great change in the way “cold” is produced, since it is not based on compression cycles. One of the main advantages is that the heat for the regeneration process is required at low exergy content. This feature make the system favourable to be coupled to solar thermal installations.

As it is possible to see, the history of air cooling/conditioning is, by far, shorter than that of heating systems.

If we consider only those appliances that were able to actually cool the environment below the outdoor temperature (thus neglecting all those devices aimed at just improving the air movement), the starting date could be located around 1900s. It is worth noting that at the beginning the HVAC systems were only used to cool the air. One of the first air condition system, appeared on the market at the dawn of the 20th century, was able to cool down the air of approximately 6 °C, but since it did not controlled the relative humidity people were complaining about the indoor thermal comfort conditions. These machineries, as previously observed, made frequently use of ice to treat the air. This means to have a cold source at a temperature of 0 °C, whilst the desired temperature level of the indoor air could be located around 26 °C.

However, it was soon realized that a proper air treatment should comprise both the cooling and dehumidification of the air.

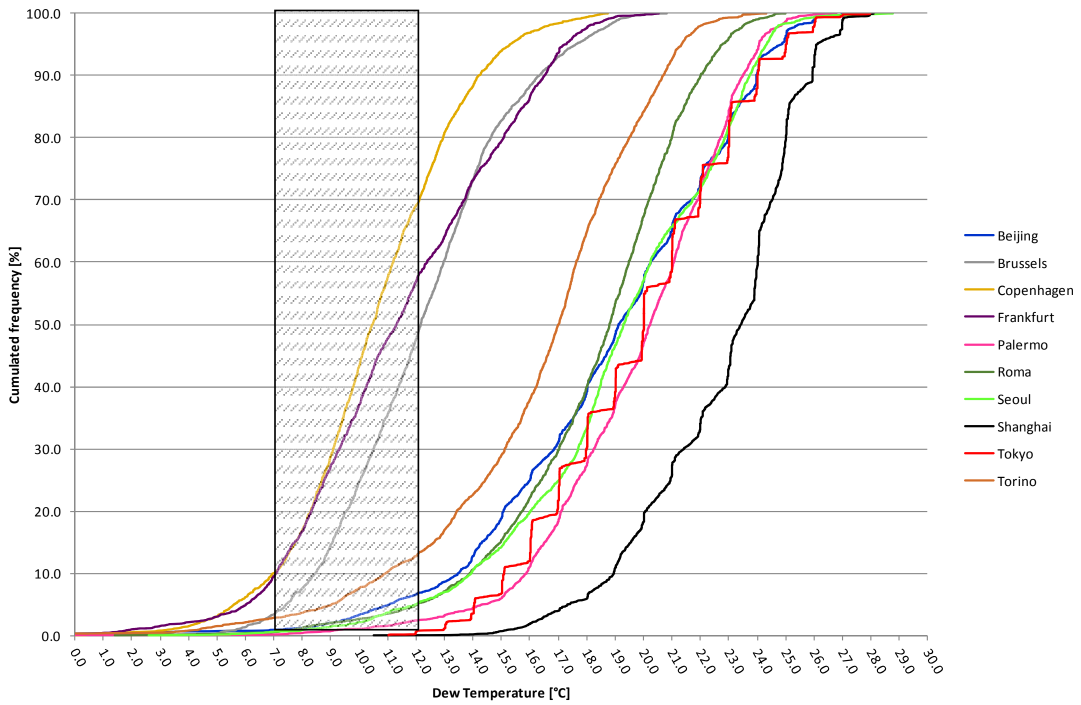


Fig. 2 Cumulated frequency of dew temperature for the 10 locations considered in Annex 59 for the period between 1st June to 15th September (Dashed area = typical supply/return temperatures of the heat carrier fluid).

Such necessity has an important consequence as far as the temperature levels of the heat carriers fluids are concerned. In fact, the most straightforward, easy to implement and cheap method to dehumidify the air is to cool the air at a point in which the water vapour condense. That is, one needs to reach a temperature below the dew point of the outdoor air.

If we look at the outdoor air conditions considered in Annex 65, we can see that the dew point temperatures typically ranges between 5 °C and 20 °C (figure 2).

It follows, that in order to be sure to cool the air at a temperature below the condensation condition of the water vapour, the supply temperature of the heat carrier fluid (usually water and glycol, at least in central air conditioning systems) has to be kept well below 10 °C. From these facts it comes out the usual habit of designing cooling coil with a supply water temperatures of about 7 °C and corresponding return temperatures of about 12 °C. In more modern air handling units, in order to improve the energy efficiency of chillers, one tries to operates between 8 °C and 13 °C. But the physical constraint of reaching the dew point temperature hinders any possibility of switching to higher operating temperatures (at least for many climatic contest).

The consequence is that even if radiant floor/ceiling systems could be operated at significantly higher temperature levels, thus reducing the consumed temperature differences, one must still have a cold source at about 7~8°C in order to be able to control the relative humidity. In practice this means that the chiller is usually producing the whole flow rate of the heat carrier fluid at the lowest temperature needed(e.g. 7°C) and then the quota of the flow rate that is used inside the radiant panels is warmed at a suitable temperature (around 17°C~19°C) by means of a mixing process. Such procedure ends up in a huge consumed temperature difference rate. The only solution to this problem is to completely change the way dehumidification is performed, that is, doing a separate control of the temperature and relative humidity. Some appliances are being marketed that perform such operation, as it will be illustrated in this Annex project.

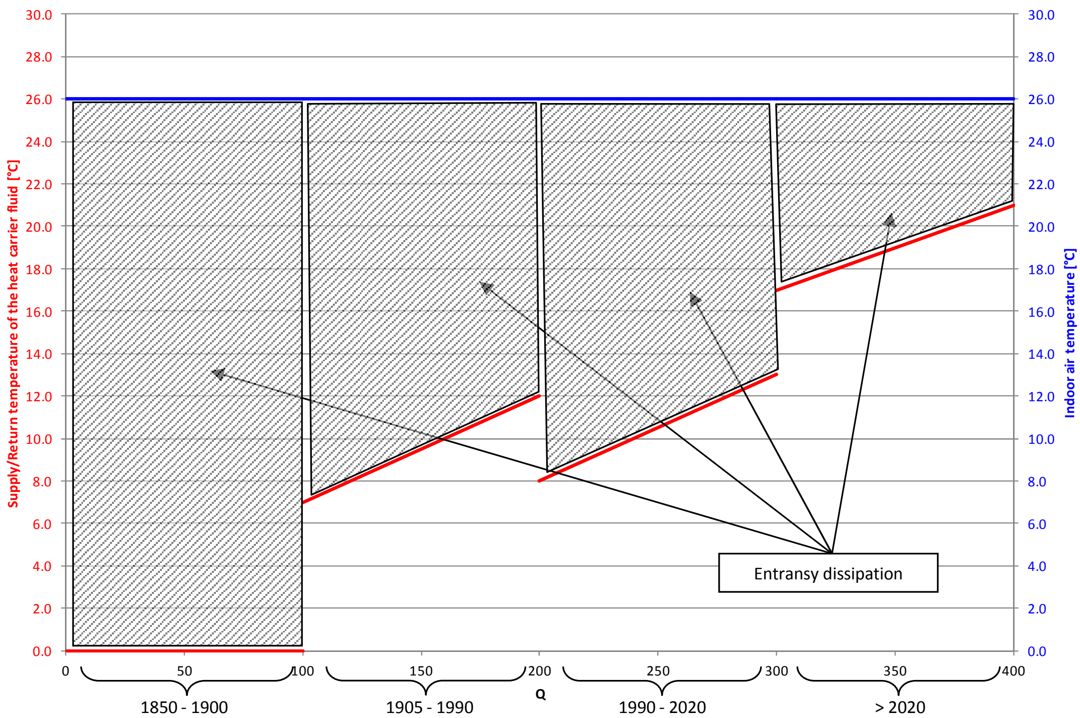


Fig. 3 Qualitative Temperature - Power chart for cooling systems – Evolution over the time of the loss (qualitative trend)

2. Typical heating/cooling systems in various countries

2.1 Heat generation systems

Boilers

Boiler systems burn usually gas and oil. In countries with a wide and diffused gas network (e.g. Italy) heat production by means of hot water boilers is the dominant solution for local heat generation, though heat pump are gaining wider and wider part of the market. In other countries, where there are wide district heating networks (e.g. Denmark or China), boilers usually operates only in households, production buildings and offices located further from the bigger agglomerations, where district heating network is not present.

In Italian old buildings boilers were centralized, with some potential problems of consumption metering and control efficiency. For these reasons, between the eighties and the nineties, small heating capacity individual boilers (less than 30 kW) started to be preferred. These small boilers had a low production efficiency and required individual flues which are complex to build. Nowadays, the new technologies of centralized boilers (e.g. thermostatic valves, variable flow rate pumps, heat meters and efficient control systems) solved the majority of the previous problems. Hence, standards and regulation force designers to use centralized heat generators or alternative energy sources.

In the latest years, the heat transfer fluid is produced at lower temperatures in boilers. This occurs because of the increasing use of condensation boilers. Moreover, considering more stringent regulations on the building envelope energy performance, the terminal units (usually oversized) work very well also with lower temperatures.

Heat pumps

Heat pumps could be used to produce hot water or hot air (DX units) with appropriate temperature levels for heating. Different types of heat pumps can be found as a function of the fluid/systems from which the heat is extracted to which is released: air (ambient or exhaust) to air (AA); air to water (A-W); water to water (water source heat pump system is becoming popular in China) (W-W); ground to air (very rarely) or water (GSHP). A further classification includes the type of heat transfer fluids (HTF): air (for direct or indirect use) and water (hydronic systems). Within these categories it is also possible to distinguish whether the heat pumps are "reversible", i.e. capable of being operated alternately in cooling and heating modes, or if they can work only in the heating function. Sometimes, in case of saturated vapor compression heat pumps, the cycle is driven by endogenous engines.

As far as absorption heat pumps are concerned, a first, important, division is between those that make use of direct combustion and those that make use of waste heat. In Danish context, heat pumps as well as boiler systems are usually used where district heating is absent. During recent years a trend can be observed, where old oil boilers are changed with heat pump systems. The majority of the heat pumps installed in last years was A-A heat pumps, but also W-W heat pumps with ground heat exchanger are being installed more and more. A-A units are often installed in buildings or spaces where peak demands are significant and where a discontinuous occupancy profile is expected.

In Korea, many of office buildings use reversible heat pumps to satisfy both heating and cooling demand. Almost all of these heat pumps are powered by electric energy and use air as heat source. Coupling heat pump systems with radiant terminals (TABS, floor heating, …) gives a particular advantage due to the low temperature of the HTF required by such terminals. This heating concepts, therefore, facilitate the integration of renewable energy sources into the buildings and allow to increase the COP values of the HP. Moreover, thanks to the lower working temperatures, they determine a lower entropy production and/or a lower temperature difference.

2.2 Systems controlling only indoor temperature

These systems are widely used in residential buildings and sometimes in tertiary buildings, when a lower quality of the indoor environment quality is sufficient. They are designed to accomplish a single function – either space heating or cooling. Typically, they cannot manage IAQ and UR (unless they are coupled with other systems).

District heating and CHPs

The fuel used in the power and heat plants can be: natural gas, biogas, solar panels, geothermal sources, wind, waste heat, biomass, coal, wastes. A good share of district heating is produced in cogeneration mode with electricity (CHPs), using waste heat from production processes and burning of wastes. In other cases a generation with a boiler gas-based or coal-based is necessary. The heat carrier fluid adopted in winter for the district heating is usually represented by superheated water or vapor. The fluid reaches different utilities and exchanges power through a local heat exchanger or through a direct connection. During the non-heating season, the temperature can be controlled on the basis of the outside temperature and wind conditions (e.g Denmark). In this way the same return temperature (lower than 60°C) can be obtained without modifying the efficiency of the CHP also during non-heating periods.

In Denmark 62 % of all houses are connected to district heating. Below 50 % of heat is produced by fossil fuels (coal, oil and natural gas). In Italy these percentages are strongly dependent on the city. Some cities have large district heating systems (e.g. Torino, 75 % of the city is covered by the district heating system), some other do not have a district heating system at all.

District cooling

District cooling is based on the same principle of district heating, but it is less widespread. In Denmark, district cooling is mainly coupled with ventilation systems. So far examples of radiant systems fed by district cooling are not available. In particular, district cooling in Copenhagen is obtained through heat exchangers, absorption and compression chillers. Some examples of district cooling are also shown in Japan and China.

Free cooling

In general, the most energy efficient solution for cooling is to limit or avoid mechanical cooling, implementing as much as possible passive or free cooling solutions. In some countries (e.g. Denmark) free cooling strategies can be performed via different sources. They could be realized by ground heat exchangers and also by sea water. Moreover, it is possible to combine these strategies with radiant systems (TABS, floor cooling). Possibilities about the exploitation of free cooling are strongly dependent on the local climate conditions

2.3 Terminal units

Radiant floors

In new constructions, especially in case of residential buildings, radiant floors are widely used for heating and in some cases - combined with primary air - also for cooling. During the eighties they were made with large pipes of iron or steel embedded in the attic. They were used with high water temperatures (50°C-40°C) and high flow rates. This led to discomfort due to warm floor and radiant asymmetry. Moreover, due to their intrinsic high thermal inertia several problems to follow the high variability of loads during the middle seasons happened. Nowadays, the radiant panels are made in polyethylene (rarely in copper) integrated in the floating floor. A lower flow rate and supply carrier temperature are used. In this way the thermal inertia is reduced, so that the radiant panels can be efficiently managed. They work very well when coupled with low temperature generation systems, such as condensing boilers, heat pumps or thermal solar systems.

They require large installation areas to satisfy the cooling/heating loads. In some rooms, when a fast heating is required, radiant floors are not suitable (e.g. bathrooms); in these cases it is necessary to replace radiative panels with radiators. This fact leads to two thermal levels of the heating carrier fluid (which become three levels considering also DHW). A central adjustment group for every housing unit is needed to mix water and obtain different temperatures. Such mixing, however, is not energy efficient and determines larger entropy production and/or temperature difference losses. Moreover, the need of having higher temperature levels leads to a lower COP of the heat pumps.

Radiant ceilings

Construction characteristics of radiant ceilings are usually of two types: ceilings with metal panels and ceilings made of gypsum board with low diameter plastic pipes embedded. They are being used more and more frequently for cooling applications, but in this case they need to be coupled with a primary air system which allows the indoor relative humidity to be controlled in order to prevent surface condensation and/or discomfort. When they are used for heating, radiant ceilings work in a similar way to the radiant floors, but they generally work with a lower temperature difference between supply and return (from 38°C to 35°C. This diminishes the losses for irreversibility).

Radiators

Radiators are only used for heating. Generally, they are made in aluminum or cast iron. Even though radiators are actually less efficient than radiant surfaces, they have a lower thermal inertia that make it possible to better follow the outside temperature trend during middle seasons. Moreover, when coupled with thermostatic valves, they allow temperature and comfort in individual rooms to be adjusted. The difference between supply and return temperature inside the device is about 20°C. The water supply temperature depends on the boiler and on the typology of the radiator: about 85°C/80°C in old systems and about 65°C/60°C in new condensing boilers. Compared to extended surface terminals (e.g. radiative panels) they work at higher temperatures and, therefore, are prone to higher losses for irreversibility (e.g entropy production/temperature difference losses).

Fan coil units (FCU)

Fan coil units (FCU) are widely used in offices and service buildings as heating and cooling units, due to their low cost and easy design. Their major advantage is their low thermal inertia which allows to promptly control the indoor environment and to follow the change of the outdoor boundary conditions combining heating and cooling functions. The coil of the heat exchanger of the FCU can be of two different types: 2 pipes (a unique coil is used both for heating and cooling. Just one function– e.g. either cooling or heating – can be exploited at a time) or 4 pipes (a coil is dedicated to the heating and another coil is dedicated to the cooling; in this last case FCU can work alternatively, in every period of the year, in cooling and heating mode). For heating applications fan coil units work with supply water temperature at about 50°C and return water at 40°C.

For cooling design, guidelines suggest to use a working temperature difference of 7°C (from 14°C to 7°C) in order to reduce the water flow rate and the pumping power. Moreover they suggest to slightly increase the supply water temperature also to avoid any condensation occurrence and indoor humidity modifications. The cooling water is produced through a chiller, where the temperature of the refrigerating fluid is about 2°C-5°C in the evaporator and about 35°C in the condenser. For heating applications fan coil units work with supply water temperature at about 50°C and return water at 40°C. This high temperature difference is required to keep small the size of the system.

Fan heaters

In the past fan heaters were widely used in factories and workshops where big heated rooms were common. They are now partially being replaced by radiant panels. They work only as heating terminals, but during the warm season they are used as simple fans to move the air and increase thermal comfort. They are often preferred to primary air/fan coil systems - which allow also the air cooling – for their higher thermal power output, due to the greater working temperature difference (from 75° to 55° or 85°C to 65°C compared to fan coil with temperature from 50°C to 40°C). Working with higher temperature difference, they are also less efficient and cause higher losses for irreversibility.

Packaged air conditioner

In many countries households having a cooling system are conditioned with a DX split or multi-split air conditioner (in many cases, however, just one or two rooms are served by the DX split system). DX unit split systems do not require a complex system and they are quite easy to install. Packaged air conditioner is composed by pipes with refrigerating fluid that pass through an outdoor compressor and an indoor evaporator (called split). The majority of these systems can be used as reversible heat pump. However, especially in residential buildings, they are used just for cooling so they are coupled with other traditional heating systems. In Korea, many offices adopt this system using reversible heat pump to perform heating and cooling. In Italy, DX unit split system is a direct expansion system that commonly appears in shops and offices. Lately it is becoming frequent also in residential buildings. Almost of all heat pumps are powered with electric energy and use air as heat source.

Multi-split type air conditioner

Multi-split type air conditioning was spreading rapidly in the last decade. It is combined with outdoor air processor. It is popular that outdoor air handling unit and total heat exchanger are used as outdoor air processor. When a desiccant air handling unit is used as outdoor air processor, evaporative temperature of refrigerant of multi-split air conditioner can be raised to treat only sensible heat load. VRV multi split systems shows significantly higher energy efficiency than traditional A-A split systems.

2.4 HVAC systems

HVAC systems allow a total air conditioning (control of temperature, relative humidity and IAQ).

Mixed systems

Mixed systems combine a central air handling unit (AHU) with a water based terminal unit placed in a single room. The AHU has the task of controlling relative humidity, indoor air quality and, partly, the sensible heat load, while the hydronic local room system balances the remaining part of the sensible heat loads. An alternative to the traditional mixed system, is the temperature and humidity independent control (THIC) system which is developing very quickly nowadays and it is finding several applications especially in China and Japan.

Mixed system - Primary air

The outdoor air can be supplied directly in the AHU from the outdoor or pre-treated with the exhaust air in a mixing chamber. During winter, the primary air is supplied at a neutral or slightly colder temperature respect to indoor temperature (in the range 20°C-18°C). Even though this fact causes a slight sensible heat load to balance, it allows a major flexibility, a better distribution of the temperature in rooms and an efficient treatment of the relative humidity in the AHU.

During summer, the primary air covers also a little part of the sensible load. For this reason the primary air is supplied in the rooms at an inlet temperature of 18°C-16°C lower than the indoor comfort temperature (26 °C). The middle seasons are the most hardly periods to accurately control the indoor microclimatic conditions. Indeed, the uncertainty due to the changing loads does not allow the use of a constant inlet temperature. However, in the middle season a chance to reduce the energy consumption through a free cooling exist.

Mixed systems - Low temperature terminals

In the various rooms the sensible heat load is balanced by local terminal units which allow, together with the primary air, to complete the air conditioning process. The terminal units may be low temperature difference devices (e.g. radiant floors, radiant ceilings, chilled beams) or high temperature difference devices (e.g. fan coil units). These last being less advisable form the efficiency point of view, since they imply higher ireversibilities.

During the heating season radiant floors and ceilings work in a similar way than the stand-alone mode. During the cooling season the reduction of the specific humidity in the primary air system allows to reduce the condensing risk on the cooled surfaces. In the last few years radiant ceilings have been widely used in Italy combined with primary air treatments, especially in office buildings and healthcare structures. Typically, the heat power emission provided to the space is about 70-80 W/m^2

A chilled beam is another type of terminal for HVAC systems. Pipes of water are passed through a "beam" (the heat exchanger) either integrated into standard suspended ceiling systems or suspended at a short distance from the ceiling of a room. As the beam chills the air around it, the air becomes denser and falls to the floor. It is replaced by warmer air moving up from below, causing a constant flow of convection and cooling the room. There are two types of chilled beams: passive types and the active type (also called an "induction diffuser") with a dedicated fan in the beam. Anyway, in cooling season a low temperature difference between the terminal and the ambient is required to avoid the surface condensation. This fact means a low cooling power exchange. For this reason in most common cases the high temperature mixed system are preferred to these system for cooling purposes.

Mixed systems - High temperature terminals

The high temperature mixed systems use fan coil units (FCU) as terminals. They are the most common configuration for space cooling, especially in Asian climates (e.g. Japan, China, Korea). The outdoor air can be supplied directly in the AHU from the outdoor (in this case the system is called fan coil units + outdoor air or FCU+OA), or pre-treated with the exhaust air in a mixing chamber (in this case the system is called fan coil units + all air). Fan coil unit (FCU) with VAV (variable air volume) or CAV (constant air volume) systems is the most common air conditioning system in Korea.

The operation of FCU in mixed system is exactly the same of the standalone FCU. Further attention should be kept to the temperature of the exchange coil to avoid the excessive reduction of relative humidity, which, is already controlled by the primary air coming from the AHU. Fan coil unit (FCU) with outdoor air (OA) system is the most common air conditioning system in Japan and Korea. The FCU remove the heating load of perimeter zone and OA system remove the cooling load and ventilation load of interior zone. AHU mainly treats internal heat gain (generated from human body, lighting and equipment) and fresh air load. FCU treats the heat gain or heat loss thorough building envelop. AHU supply cold air all year around and FCU supply cold air in summer and hot air in winter. Therefore, it is important to determine set point temperature for interior and perimeter zone to avoid mixing loss between cold air from AHU and hot air from FCU.

All air systems

VAV (variable air volume) or CAV (constant air volume) systems are a common types of centralized air-conditioning systems all over the world. There are two design modes: classic for single utilities (e.g. a conference hall) or combined with post-cooling or post-heating systems for utilities with different thermal levels or different loads (e.g. some special departments of hospitals). Dual conduits are excessively expensive and have been abandoned since many years.

When the maximum load occurs, the temperature difference between inlet air and room comfort temperature is about 10°C. This means supplying air at 16-14°C. It is not recommended to supply air at a lower temperature due to problems of air distribution, cold draft and potential local condensation. In Denmark, to provide cooling, passive measures are in general adopted. If they are not sufficient to maintain indoor temperature within allowed boundaries, then mechanical cooling is introduced. In Italy, VAV are gradually disappearing from the market (just some are still used in healthcare structures). CAV systems are still used but they are less widespread than mixed systems.

All air system: Perimeter-less system

Perimeter-less system is an improved version of all air system. Air flow window is one of the most common perimeter-less system in new high performing buildings. The air flow window is comprised of a window shade to install between two pieces of glass. The room air passes between two pieces of glass and is drawn out by the upper part. It prevents heat load through the window to enter the room.

All air system: Underfloor air conditioning system

The under floor air conditioning system uses underfloor chamber of free access floor. Conditioned air is supplied through the diffuser which is mounted on the floor and return through the inlet of the ceiling. Air conditioning for occupied zone which is approximately 1.7 m above the floor can be realized by under floor air conditioning system.

3. Reducing losses to achieve energy-efficient HVAC systems

To remove heat from an indoor source to outdoor sink, a series of processes are necessary for a central air-conditioning system, and loss in each process compose the total loss of the system. Given a certain heat exhaust, reducing the loss of driving force and temperature difference in each section will contribute to decreasing the total temperature difference. The interrelationship between the reduction in total system temperature difference and performance enhancement will be analyzed in the following.

When an appropriate outdoor heat sink is selected for heat exhaust, reducing the system loss may result in a better use of the natural cold source. Assuming an appropriate outdoor heat sink temperature, the difference between indoor temperature and the sink $\Delta T_{s\text{-}s}$ may meet the total driving temperature difference in every section, which makes it possible to use the natural cold source directly to fulfill the demand of air-conditioning system. Heat exhaust from the indoor heat source to outdoor heat sink is shown in Fig. 3-1 when using a natural cold source; additionally, the intermediate loss ΔE_n equals $Q_{load}\cdot(T_{in}\text{-}T_{sink})$. Assuming that heat exhaust Q_{load} and indoor temperature T_{in} are fixed, reducing the intermediate loss will be beneficial to achieving a higher temperature level of the sink, and thus an extended temperature range of the natural cold source would be available. If the natural cold source changes with time, such as changing outdoor wet bulb temperature, reducing ΔE_n will also give impetus to achieving a higher temperature level of the natural cold source, and thus an extended duration of using the natural cold source to exhaust heat would be available.

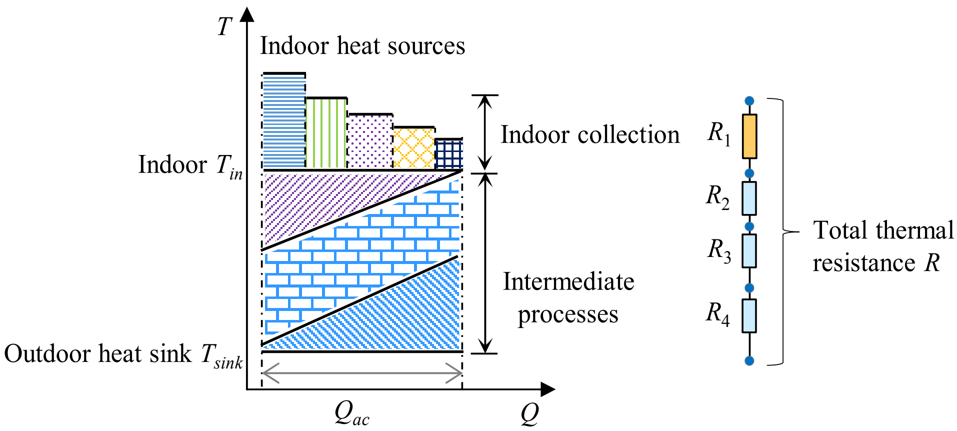


Fig. 3-1 Heat exhaust using a natural cold source

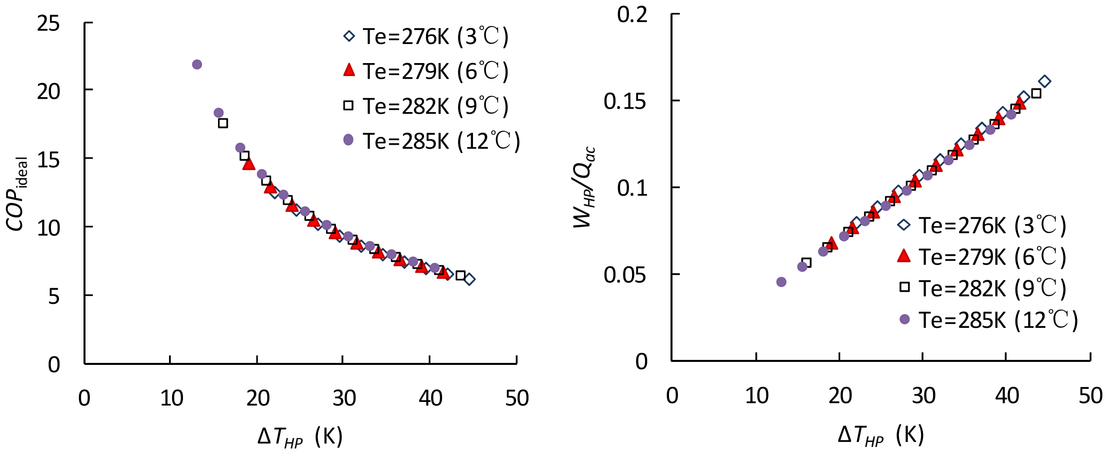
When an appropriate outdoor heat sink is not available directly, a mechanical refrigeration cycle is necessary to provide heat transfer with a driving force. The *COP* of the vapor compression refrigeration cycle can be calculated using Eq. (3-1), where T_e and T_c are the evaporation temperature and condensation temperature, respectively. The thermodynamic perfectness η is restricted by the manufacturing and processing technology, and is about 0.5–0.75 under the current technological level.

$$COP=\eta \cdot \frac{T_e}{\Delta T_{HP}} = \eta \cdot \frac{T_e}{T_c - T_e} = \eta \cdot COP_{ideal} \tag{3-1}$$

The COP_{ideal} is the performance coefficient of the ideal Carnot refrigeration cycle under given evaporation and condensation temperatures, as shown in Fig. 3-1(a). Taking the heat pump refrigeration cycle for instance, the power consumption corresponding to cooling capacity per unit W_{HP}/Q_{ac} is expressed as

$$\frac{W_{HP}}{Q_{ac}} = \Delta T_{HP} \cdot \frac{1}{\eta \cdot T_e} \tag{3-2}$$

where the evaporation temperature T_e is expressed in kelvin and ranges from 276 K to 285 K in common HVAC systems. According to Eq. (3-2), the power consumption is approximately proportional to ΔT_{HP}, as shown in Fig. 3-1(b).



(a) (b)

Fig. 3-2 Effect of evaporating temperature on chiller performance: (a) COP_{ideal}; (b) power consumption corresponding to cooling capacity per unit

Air-conditioning systems comprise indoor collection, heat transfer, handling of heat and moisture, and other processes. These individual processes are relatively independent although they interact with one another. By reducing the consumption of driving temperature difference in each section and transfer losses, performance enhancement can be achieved in each section; furthermore, the total driving temperature difference can be decreased and system performance improvement can be expected.

It should be noted that system performance is restricted by the pump, fan, and other transportation equipment aside from cold source equipment. Because of the restriction imposed by transportation power consumption, air volume and water volume in the system are finite, which renders a fluid temperature difference in the transport process and leads to transfer losses. Compared with power consumption in refrigeration cycles, transportation power consumption is a disparate task with a disparate identity, which should be taken into consideration in system performance analysis.

PART 2

1. Introduction: tasks and transfer processes of HVAC system

1.1 Basic tasks of HVAC systems

The main purposes of an HVAC system are to maintain a suitable indoor temperature and humidity and provide sufficient fresh and clean air to the inhabitants. In the cooling condition, solar radiation and other heat/humidity sources bring the cooling/humidity load into the indoor environment. As shown in Fig. 1-1, the indoor cooling load can be removed from the indoor environment to the outdoor environment either through the building envelope (passive channel) or by the HVAC system (active channel). The driving force of the heat transfer through the building envelope is the temperature difference, which is determined by the indoor and outdoor conditions. When the temperature of the outdoor condition is not sufficiently low, power consumption is required to provide the required temperature difference between the indoor and outdoor environments to remove heat or moisture in the HVAC system.

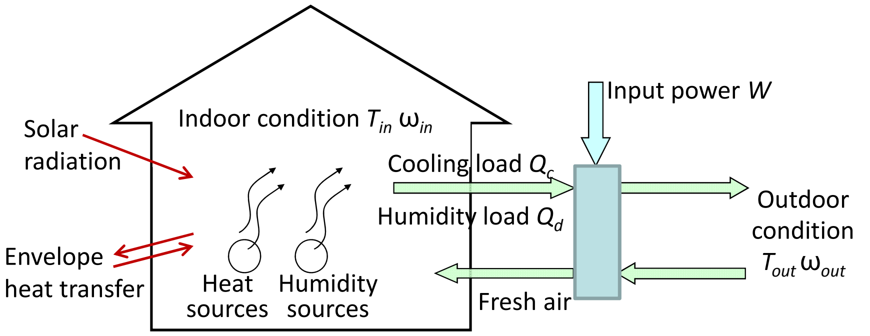


Fig. 1-1 Concept diagram of HVAC system

Given a determinate heat source and humidity source with a stable requirement from the indoor environment, a lower heat sink temperature combined with a larger temperature difference between the indoor heat source and outdoor heat sink indicates that a greater driving force exists for removal of heat from the indoor source to the outdoor sink; in other words, when a greater temperature difference for transfer, ΔT, is available for consumption in the various processes of an air-conditioning system, heat exhaust and moisture transport would be more attainable by the system. Conventional outdoor heat sinks and their temperatures for an air-conditioning system are displayed in Fig. 1-2. In the process of heat release, appropriate selection of heat sink contributes to an expected increase in the driving force from source to sink. Thus, enhancement of air-conditioning system performance would be achieved. This way of enhancement may be validated when the outdoor dew temperature is selected as the heat sink by using the indirect evaporative cooling method, which enables a higher temperature level of the additional cold source, and consequently effective exploitation of the natural cold source.

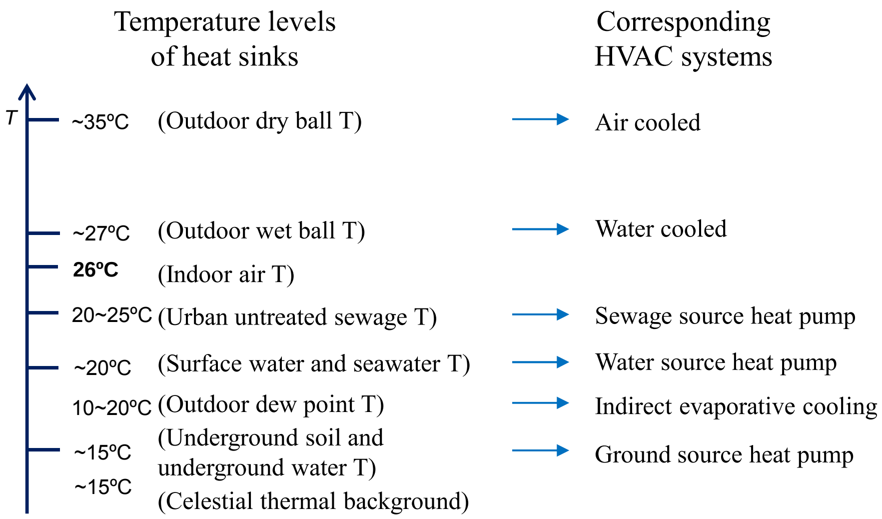


Fig. 1-2 Typical outdoor heat sinks available for air-conditioning system

When the cooling load cannot be removed by a single passive system, additional heat needs to be removed by an active system consisting of an air-conditioning system. Such a system includes systems for heat transfer processes as well as heat–work conversion processes, such as heat pumps and refrigeration systems. The heat transfer process and the heat–work conversion process should be described with corresponding parameters and indices, respectively. Taking an active air-conditioning system as an example (Fig. 1-3), the system consists of a series of processes, including the collection and transfer of indoor heat and moisture to the outdoor heat sink. The additional heat/moisture produced by indoor heat/humidity sources is handled by the air-conditioning system and removed to the proper outdoor sinks, which realizes the transfer of heat and moisture from indoor sources to outdoor sinks.

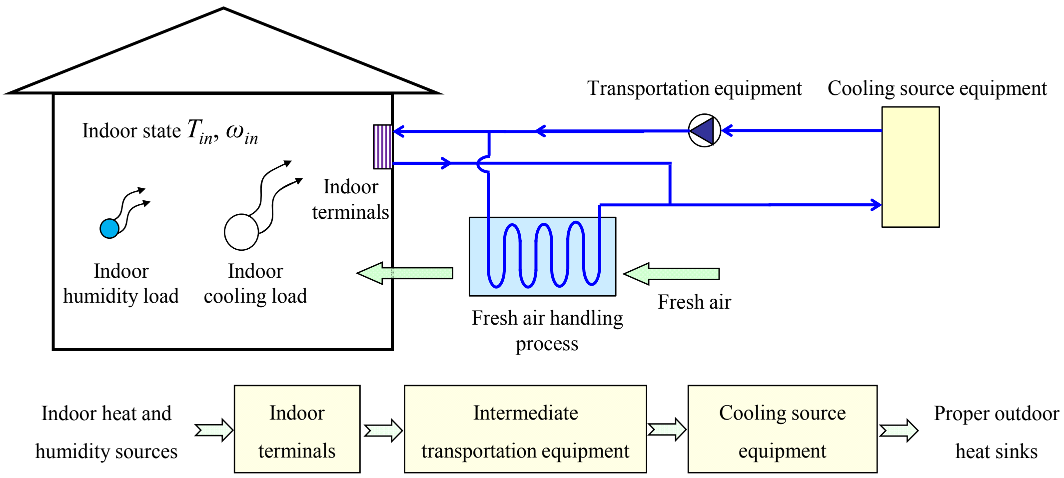
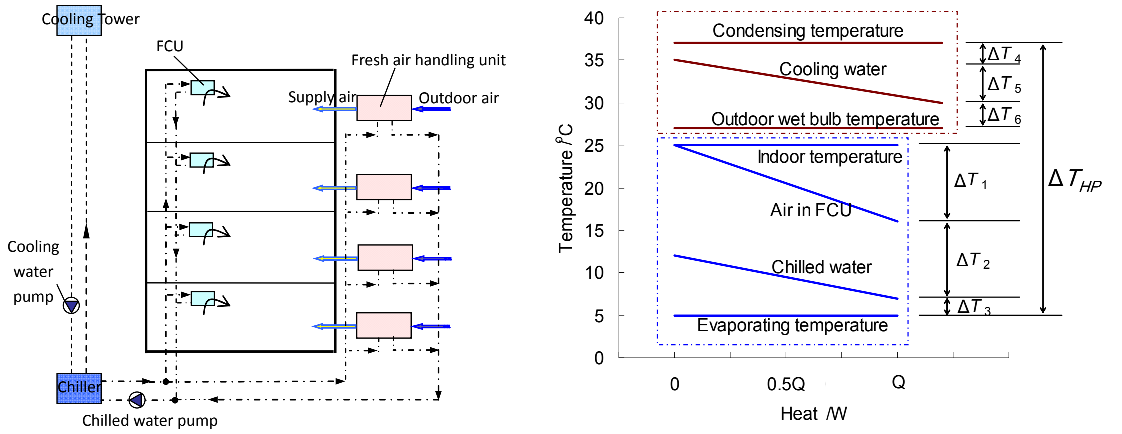


Fig. 1-3 Main processes of active air-conditioning system

1.2 Transfer processes and Δ*T* existing in HVAC system

An outdoor air system with FCUs (fan coil units) can be taken as an example. Fig. 1-4 shows the main processes of a typical air-conditioning system between indoor heat/humidity sources and outdoor sinks. Terminal units of the air-conditioning system collect the additional indoor heat and moisture, the energy transportation process transports the heat and moisture to cold source equipment, the cold source equipment removes the heat and moisture to outdoor heat sinks. To meet the indoor health requirements, air-conditioning systems are expected to supply an adequate amount of fresh air. The fresh air handling process is always carried out along with the heat and mass transfer of humid air. A chiller produces chilled water at 7°C, and the chilled water is supplied to the FCUs and the outdoor air-handling units to remove both the sensible load and the latent load of the building. The exhaust heat from the condenser is then released into the outdoor environment through the cooling tower.



(a) (b)

Fig. 1-4 Typical air-conditioning system: (a) operating principle; and (b) processes

From the main processes of an air-conditioning system shown in Fig. 1-3, a conclusion can be drawn that an air-conditioning system mainly consists of transfer and transform processes with a certain heat sink, in which the transfer comprises indoor collection, heat transfer, handling of heat and moisture, and other processes. During the heat and humidity exhaust process, deficiency in the driving forces of temperature and humidity difference results in a requirement for a mechanical refrigeration cycle or heat pump to supplement the driving forces; hence heat–work conversion is contained within the whole system. The different processes with respective tasks in an air-conditioning system are relatively independent. Meanwhile, individual processes are closely connected and jointly constitute the entire process in the air-conditioning system.

Fig. 1-4(b) shows the temperature level of each component in the handling process. This *T*–*Q* (temperature–heat flux) chart is a depiction of heat transfer processes, with the temperature level as the ordinate and the heat flux of the transfer process as the x-coordinate. The temperature difference between the outdoor wet bulb temperature (cooling tower is adopted) and the indoor air temperature is only approximately 2 K; however, the temperature difference between the condenser and evaporator sides of the chiller (ΔT_{CH}) is as high as 32 K. A larger temperature difference in the chiller means that a lower COP_{CH} and greater power input (W_{CH}) of the chiller are required for the same cooling capacity Q_{CH}, as calculated by Eq. (1-1):

$$W_{CH} = \frac{Q_{CH}}{COP_{CH}} = \frac{Q_{CH}}{\eta \cdot T_{evap}}\left(T_{cond} - T_{evap}\right) = \frac{Q_{CH}}{\eta \cdot T_{evap}} \cdot \Delta T_{HP} \qquad (1\text{-}1)$$

where T_{cond} and T_{evap} represent the condensation temperature and the evaporation temperature, respectively, of the chiller (in K), and η is the thermodynamic perfectness degree, which is the ratio of the *COP* of the actual chiller to that of the ideal Carnot chiller at the same T_{cond} and T_{evap}.

The energy consumption of an actual chiller is always lower than that of an ideal Carnot chiller operating at the same T_{cond} and T_{evap}. As indicated in Eq. (1-1), the thermodynamic perfectness η of an actual chiller expresses the difference between the actual and ideal chillers. For common centrifugal water chillers, η is approximately 0.6–0.7, which means that the energy efficiency of the actual chiller is much lower than that of the ideal chiller. According to Eq. (1-1), the power consumption of the chiller (W_{CH}) is proportional to the temperature difference $\Delta T_{CH} = T_{cond} - T_{evap}$ in addition to the variance of η. In other words, a decrease in ΔT_{CH} will directly result in a decrease in W_{CH}. In addition to the performance discrepancies between the ideal Carnot chiller and actual chiller, how is ΔT_{CH} consumed or restricted in the actual air-conditioning system shown in Fig. 1-4?

ΔT_{CH} is restricted by the integrated processes for handling sensible load and moisture in an actual air-conditioning system, as shown in Fig. 1-4. Integrated processes for handling sensible load and moisture are adopted in actual systems. The same cooling source produces chilled water, which simultaneously dehumidifies and cools the building in actual systems indicated by Fig. 1-4. However, the required cooling source temperatures for sensible load and moisture are quite different. In theory, the temperature requirement for removing indoor sensible load is that the cooling source temperature is lower than that of the heat source. If the indoor sensible load is removed from indoor air, the required cooling source temperature is the dry bulb temperature of indoor air in theory, whereas if condensing dehumidification is adopted for removing indoor moisture, the required cooling source temperature is the dew point temperature of indoor air. In a conventional air-conditioning system, the same cooling source is utilized for removing indoor sensible load and moisture, restricting the operating temperature difference of the water chiller.

ΔT_{CH} is also caused by the limited water or air flow rates in an actual air-conditioning system. As indicated by Fig. 1-4, there are various air or water circulation processes driven by fans or pumps in the air-conditioning system, including the air circulation of FCU and the chilled water circulation between the evaporator and the FCU. The energy consumption of fans (W_a) and pumps (W_w) can be written as follows:

$$W_a = \frac{\dot{V}_a \Delta p_a}{\varepsilon_a} = Q_a \cdot \frac{\Delta p_a}{\varepsilon_a c_{p,a} \rho_a} \cdot \frac{1}{\Delta T_a} \qquad (1\text{-}2)$$

$$W_w = \frac{\dot{V}_w \Delta p_w}{\varepsilon_w} = Q_w \cdot \frac{\Delta p_w}{\varepsilon_w c_{p,w} \rho_w} \cdot \frac{1}{\Delta T_w} \qquad (1\text{-}3)$$

where $\dot{V}$ is the volumetric flow rate, Δp is the head or supplied pressure, ε is the efficiency of fans or pumps, c_p is the specific heat, ρ is the density, and ΔT is the temperature difference between the inlet and outlet of the circulation fluid. Q is the cooling/heating capacity provided by the transportation system, and subscripts a and w represent air and water, respectively.

where $\dot{V}$ is the volumetric flow rate, Δp is the head or supplied pressure, ε is the efficiency of fans or pumps, c_p is the specific heat, ρ is the density, ΔT is the temperature difference between the inlet and outlet of the circulating fluid, Q is the cooling/heating capacity provided by the transportation system, and subscripts a and w represent air and water, respectively.

If the flow rate of the working fluid is infinite, ΔT_a or ΔT_w can be reduced to zero but the power consumption of the transportation system will be too high. Therefore, limited flow rates are adopted in actual systems, and the energy consumption of the transportation system must be included. Reducing the ΔT_{CH} is also restricted by the power consumption of the transportation system. Taking the FCU in Fig. 1-4 as an example, increasing the air flow rate leads to a decrease in ΔT_1 for a given cooling/heating capacity Q; ΔT_{CH} then decreases, indicating a performance improvement of chiller, but fan power consumption for air circulation increases as indicated by Eq. (1-2). On the other hand, a larger ΔT_1 can help save the energy consumption for air circulation, but this would lead to an increase in ΔT_{CH} and result in an almost exactly proportional increase in the energy consumption of chiller. Therefore, the determination of ΔT_a or ΔT_w in the circulation system requires balancing the energy consumption of the transportation system with that of the chiller.

There are many heat exchangers (or heat and moisture exchangers) in an air-conditioning system. For a given cooling/heating capacity Q, increasing the heat transfer area is a feasible method for reducing the temperature difference in heat transfer. For example, ΔT_3 and ΔT_4 shown in Fig. 1-4 decrease with an increase in heat transfer areas of the evaporator and condenser. If the heat transfer areas are infinite, ΔT_3 and ΔT_4 approach 0, which means an increase in T_{evap} and a decrease in T_{cond}, i.e., a lower ΔT_{CH}. However, the heat transfer area could be regarded as proportional to the initial cost and therefore never be infinite in an actual system.

In summary, the large temperature difference ΔT_{CH} of an actual air-conditioning system (shown in Fig. 1-4) is restricted by the integrated processes for handling sensible load and moisture, the performance of actual chillers, the energy consumption of fans and pumps, and the input heat or mass transfer capacity. Table 1-1 shows how ΔT_{CH} is consumed during the entire air-handling process as well as the input of the system. The heat or mass transfer capacity expresses the initial cost or input of the air-conditioning system, and the energy consumption of fans/pumps and chiller expresses the operating energy cost. Because of the aforementioned reasons, the working temperature difference of the chiller ΔT_{CH} is far from that of the ideal process.

Table 1-1 Temperature differences in typical air-conditioning system shown in Fig. 1-4

Handling process	Temperature difference	Input of the system		
		Heat or mass transfer capacity	Energy consumption of fans and pumps	Energy consumption of compressor in chiller
FCU	ΔT_1		Fan consumption W_{FCU}	
	ΔT_2	Transfer area of FCU	Pump consumption W_{CWP}	
	ΔT_3	Transfer area of evaporator		
Outdoor air handler	ΔT_{1a}		Fan consumption W_{FAN}	
	ΔT_{2a}	Transfer area of cooling coil	Pump consumption W_{CWP}	
	ΔT_3	Transfer area of evaporator		
Condensing heat exhaustion	ΔT_4	Transfer area of condenser		
	ΔT_5		Pump consumption W_{CTP}	
	ΔT_6	Transfer area of cooling tower	Cooling tower consumption W_{CT}	
Chiller	ΔT_{CH}			Compressor consumption W_{CH}

Notes: CWP-chilled water pump; CTP-cooling water pump; CT-cooling tower

1.3 Temperature influence on efficiency of heating and cooling sources

As indicated in section 1.2, there are various transfer processes in an HVAC system and temperature differences are caused by limited flow rates, limited transfer capacity, etc. The temperature level or temperature difference of the circulating fluid significantly affects the energy performance of the HVAC system. Energy efficiency of the heating/cooling sources is also related to the operating temperature levels. To reduce the temperature differences existing in the various transfer processes, a "high temperature cooling" or "low temperature heating" system could be realized, which improves the energy performance of cooling/heating sources. In the following, the influence of temperature on the energy performance of cooling/heating sources is investigated.

1.3.1 Temperature influence on performance of cooling sources

Increasing the chilled water temperature for air conditioning can benefit in the following ways: using underground water circulation to provide cooling around average annual temperature; using indirect evaporative cooling to provide chilled water between web bulb and dew point temperatures; and increasing the *COP* of mechanical chillers. Increasing the required cooling source temperature offers the possibility to utilize natural cooling sources, e.g., deep phreatic water, ground heat exchangers, and direct or indirect evaporative cooling methods in certain dry regions. Increasing the chilled water temperature also helps to lengthen the duration of utilizing natural cooling sources. If no natural cooling sources can be adopted directly as the cooling source, a vapor compression refrigeration system can be utilized instead. Owing to the increased evaporation temperature of the vapor compression refrigeration cycle, the operating temperature difference ΔT_{HP} of the mechanical chiller will be significantly reduced, which improves the *COP* of the chiller.

If there is no natural cooling source to be utilized directly, high temperature cooling can improve the *COP* of a mechanical chiller, because of a significant increase in the evaporation temperature. In a conventional system, the chilled water temperature is approximately 7ºC (45ºF). For a high temperature cooling system, the chilled water temperature increases to approximately 16–18ºC (61–64ºF). Fig. 1-5(a) shows the ideal *COP*s for a conventional water chiller and high temperature water chiller. As indicated by this figure, increasing the required chilled water temperature helps to increase the evaporation temperature. Thus, the operating temperature difference ΔT_{CH} is reduced, leading to a significant improvement in the *COP* of the chiller. The performance of a water chiller is related to the load factor and cooling water temperature, and a simulation model is established on the basis of experimental results of the chiller. The energy performance of centrifugal water chillers with varying load ratio, shown in Fig. 1-5(b), shows that the *COP* of a water chiller with a higher supplied water temperature is significantly superior to that with a lower temperature. The *COP* of the high temperature water chiller is more than 30% higher than that of the conventional type.

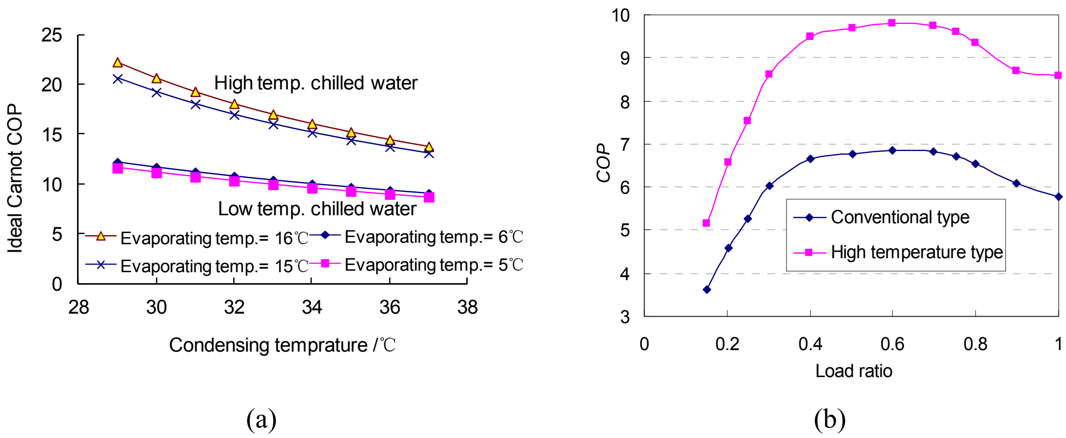


Fig. 1-5 Performance comparison of centrifugal water chillers with different supply water temperatures: (a) COP of ideal chillers; and (b) COP of actual chillers.

1.3.2 Examples of temperature influence on performance of heating sources

Decreasing the heating temperature for space heating can benefit in the following ways: increasing the *COP* if a heap pump is utilized as the heating source; expanding the range of industrial waste heat available for heating; increasing the efficiency of condensation gas boilers; and increasing the efficiency of CHP (combined heat and power) sources.

Low temperature heating can promote the utilization of energy of various grades such as industrial waste heat and solar energy. Fig. 1-6 takes a non-ferrous metal smelter in northern China as an example and presents a comparison of industrial waste heat sources on the basis of different temperatures. As the results show, waste heat sources with a temperature less than 100ºC constitute 70% and only 14% of the sources have a temperature higher than 200ºC. The temperature of waste heat that has recycling feasibility is mainly less than 100ºC. Therefore, low temperature exhaust heat occupies a large proportion of industrial waste heat. On the other hand, the distribution of industrial waste heat is often dispersed and the parameters of heat and temperature are uneven, so we need to integrate these various grades of low temperature heat in order to apply them to heat-supply systems. A lower hot water temperature of heat supply will benefit the demand of using industrial waste heat as a heat source for winter heating and integration of industrial waste heat sources.

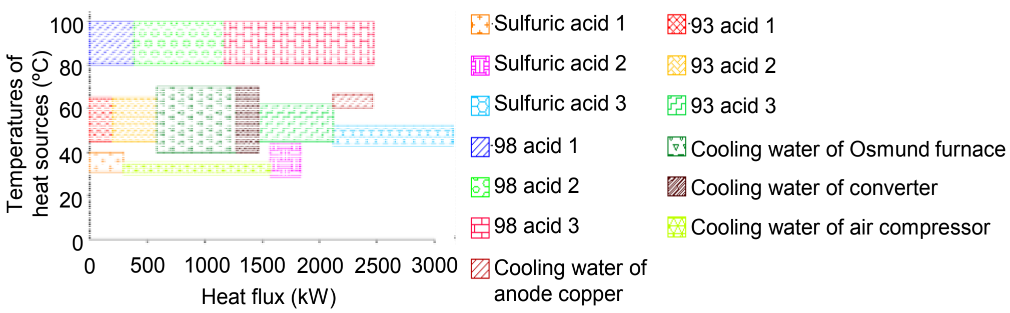
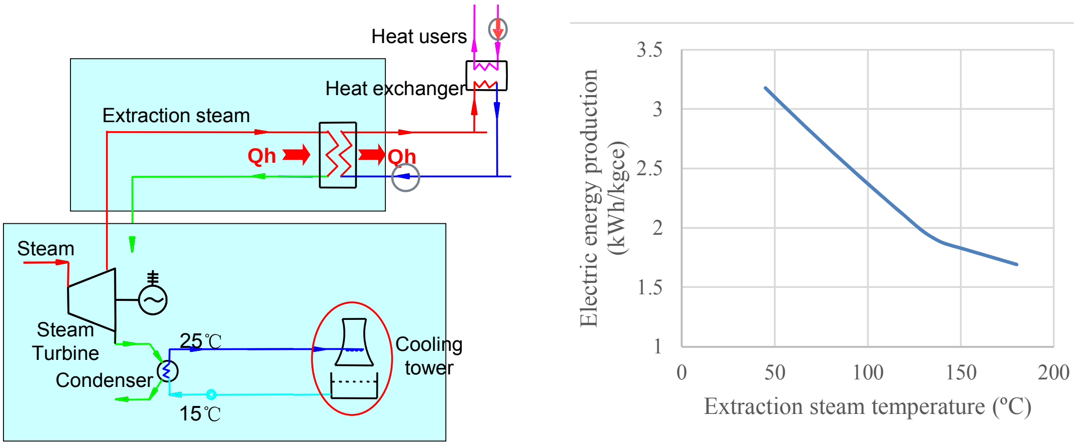


Fig. 1-6 Waste heat distribution of some copper plant

Cogeneration is another common form of central heating, with its operating principle shown as Fig. 1-7(a). While utilizing cogeneration units as the heat source of a heat-supply system, coal consumption of the heat-supply process depends on the heat-supply temperature. Hence, a lower heat-supply temperature means a lower exhaust parameter of cogeneration units and a smaller impact of extraction process on the power generation of cogeneration units. The impact of extraction process on the generated energy can also be decreased through some technological means, such as using heat pumps (even a turbine) operated in low vacuum to reduce the capacity of suction. Fig. 1-7(b) shows the impact of extraction steam temperature for heat supply on electricity generation in cogeneration. As shown in the figure, a lower extraction steam temperature helps to increase the electricity generation. As a result, the reduction of heat-supply temperature will contribute to improve the performance for electricity generation and improve the overall performance of heat-supply and generation process.



(a) (b)

Fig. 1-7 (a) operating principle of cogeneration system and (b) effect of extraction steam temperature for heat supply on electricity generation in cogeneration

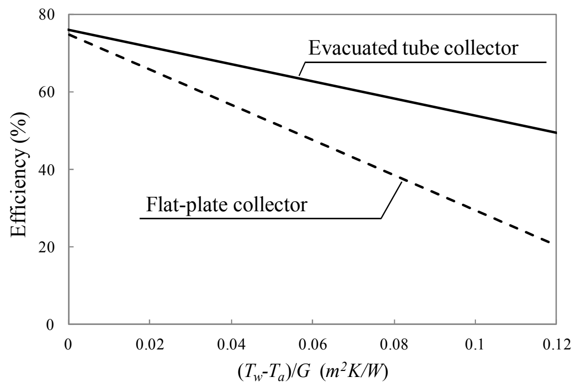
In the cogeneration units that use an absorption heat pump to recycle condenser waste heat in power plants, the key to reducing the grades of heat source and improving the efficiency and economics is to increase the proportion of condensation steam utilization; this can be achieved by lowering the demand for heat-supply temperature. Taking a primary network heat-supply system as an example, Table 1-2 gives the variations in extraction heat and condensation steam waste heat recovery with a varying return water temperature and a constant supply water temperature of 130ºC. The total heat supply is 100 MW, extraction parameter is 0.4 MPa, and exhaust parameter is 40ºC. It can be concluded from the table that the proportion of condensation steam waste heat recovery gradually increases as the temperature of the primary network decreases. Therefore, by reducing the return water temperature, the heat-supply temperature can be reduced and the effect of condensation steam waste heat recovery can be improved considerably.

Table 1-2 Relationship between waste-heat recovery of condensation steam and return water temperature in cogeneration

Supply water temperature/ºC	Return water temperature/ºC	Extraction heat /MW	Condenstion steam heat/MW	Total heat/MW
130	70	92.5	7.5	100
	60	87.5	12.5	
	50	82.1	17.9	
	40	73.0	27.0	
	30	65.4	34.7	
	20	59.8	40.2	

If a solar collector is adopted as the heating source, similar conclusions to industrial waste heat can be obtained. With a decrease in the water temperature on the user side (i.e., circulating water temperature inside the collector), heat loss between the solar collector and the environment will be decreased. Then the solar collector's efficiency will be improved, and there will be more heat for heating for a certain collector area. Taking the two common types of solar collectors as examples, Fig. 1-8 illustrates the influence of normalized operating temperature on the solar collector's efficiency, where T_w is the average water temperature of the solar collector (ºC), T_a is the ambient temperature (ºC), and G is the solar radiation intensity (W/m^2). As indicated by the efficiencies of the flat-plate collector and evacuated tube collector varying with the normalized temperature difference, a lower average water temperature inside the collector (corresponding to low temperature heating system) results in a higher efficiency of the solar collector.

(a)



(b)

Fig. 1-8 Temperature influence on performance of solar collectors: (a) solar air collector; and (b) efficiencies of solar collectors.

If an air-source heat pump, a water-source heat pump, or a ground-source heat pump is adopted as the heating source, decreasing the required temperature for heating helps to lower the condensation temperature of the heat pump cycle, leading to an improvement in the energy performance. For the evaporation temperatures T_e of -5ºC and 5ºC for a heat pump cycle, Fig. 1-9 illustrates the *COP* of the heat pump with varying supply water temperature, where the condensation temperature is treated as 2ºC higher than the supply water temperature and the thermodynamic perfectness of the heat pump cycle is 0.55 to simplify analysis. This figure indicates that a lower required supply water temperature results in a lower operating temperature difference of the heat pump cycle ΔT_{HP} and a higher *COP*.

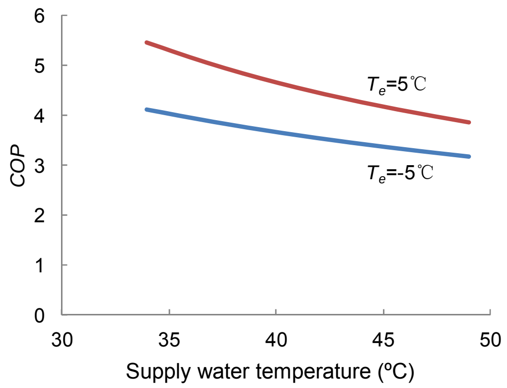


Fig. 1-9 Influence of supply water temperature on energy performance of heat pump

2. Basic definition and method of entransy theory

To improve the performance of HVAC system, a conventional solution is to calculate its performance with certain input parameters and then adjust the input parameters for performance improvement. It's a kind of optimizing method through black box, without clearly understanding of the HVAC system. To focus on the essential of indoor thermal built environment, thermal analysis according to thermodynamic parameters are emphasized and thermal principles are to be proposed for system optimization in this Annex 59. The thermal analysis method is to identify the losses occurring in the HVAC system and try to find approaches for performance optimization through reducing loss. As indicated by the current HVAC systems of different countries in Appendix A, temperature differences mainly exists in the internal heat or moisture transfer processes. Reducing the temperature differences consumed by these internal processes will help to reduce the entire temperature difference required by the heating/cooling source. As indicated by the analysis in Chapter 1, lowing the temperature requirements helps to improve the cooling/heating sources' energy performances. In thermal analysis method, both heating/cooling capacity Q and temperature level T or temperature difference ΔT are concerned to calculate the internal losses. Entropy (the same theoretical basis with exergy) and entransy are two common thermal parameters. Fig. 2-1 illustrates a simple comparison of these two parameters. As indicated by this figure, entropy is a theoretical parameter based on the second law of thermodynamics, which is suitable for characterization of heat-work transformation. Entransy is a parameter for analyzing the heat transfer process, focusing on the transfer ability. Thus in this annex project, entransy is chosen as the theoretical tool to investigate the internal losses of HVAC system.

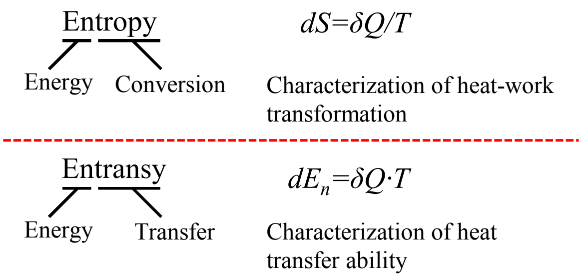


Fig. 2-1 Thermal parameters adopted for analysis

2.1 Basic definition of entransy

Entransy definition

The concept of entransy originates from an analogy between heat conduction and electric conduction (Guo ZY et al., 2007); entransy corresponds to electric potential. The definition of entransy is based on the heat conduction equation, which is also the basis of entransy analysis.

For heat conduction problems with an inner heat source, the thermal energy conservation equation is

$$\rho c_p \frac{\partial T}{\partial \tau} = \nabla \cdot (k\nabla T) + \dot{q} \tag{2-1}$$

By multiplying Eq. (2-1) with *T*, Eq. (2-2) is obtained:

$$\rho c_p T \frac{\partial T}{\partial \tau} = \nabla \cdot (k\nabla T)T + \dot{q}T \tag{2-2}$$

Eq. (2-2) can be rewritten as

$$\frac{\partial E_n}{\partial \tau} = \nabla \cdot (Tk\nabla T) - \Delta E_{n,dis} + \dot{q}T \tag{2-3}$$

where

$$E_n = \frac{1}{2}\rho c_p T^2 \tag{2-4}$$

This quantity is referred to as entransy, the capacity to transfer thermal energy that results from both energy and temperature level. The left side of Eq. (2-3) is the time variation of entransy. The first term on the right side of Eq. (2-3) is the entransy transfer capacity associated with heat transfer, the second term is the entransy dissipation rate written as Eq. (2-5), and the third term is the entransy input rate by the inner heat source.

$$\Delta E_{n,dis} = k\left|\nabla T\right|^2 \tag{2-5}$$

In a reversible heat transfer process, entransy dissipation is equal to zero. For example, the heat transfer area of the counter-flow heat exchanger is infinite and the heat capacity flow rates of the two sides are equal.

$\dot{E}n_{rev}$ is the entransy flow rate of a reversible heat transfer system, and $\dot{E}_n$ is the entransy flow rate of an actual heat transfer system.

$$\frac{d\dot{E}n_{rev}}{d\tau} = \sum_{i=1}^{n}(T_i - T_0)\dot{Q}_i + \sum_{in}\frac{1}{2}\dot{m}c_p(T-T_0)^2 - \sum_{out}\frac{1}{2}\dot{m}c_p(T-T_0)^2 \tag{2-6}$$

$$\frac{d\dot{E}n}{d\tau} = \sum_{i=1}^{n}(T_i - T_0)\dot{Q}_i + \sum_{in}\frac{1}{2}\dot{m}c_p(T-T_0)^2 - \sum_{out}\frac{1}{2}\dot{m}c_p(T-T_0)^2 - \iiint_{\Omega} k\left|\nabla T\right|^2 dV \tag{2-7}$$

The balance equation of entransy flow rate can be established based on the thermal energy conservation equation:

$$\dot{E}n_{rev} - \dot{E}_n = \Delta E_{n,dis} \tag{2-8}$$

Therefore, in HVAC system the entransy balance equation is written as Eq. (2-9). The supplied and obtained entransy should be distinguished based on the purpose of heat transfer, while entransy dissipation is inevitable in actual heat transfer processes.

$$E_{n,in} - \Delta E_{n,dis} = E_{n,out} \tag{2-9}$$

The irreversibility of heat transfer processes can be measured by either exergy destruction or entransy dissipation, but with different meanings. Exergy destruction (or entropy generation) indicates a loss of the capacity to convert heat into work. Entransy dissipation indicates a loss of the capacity of heat transfer for the purposes of heating and cooling.

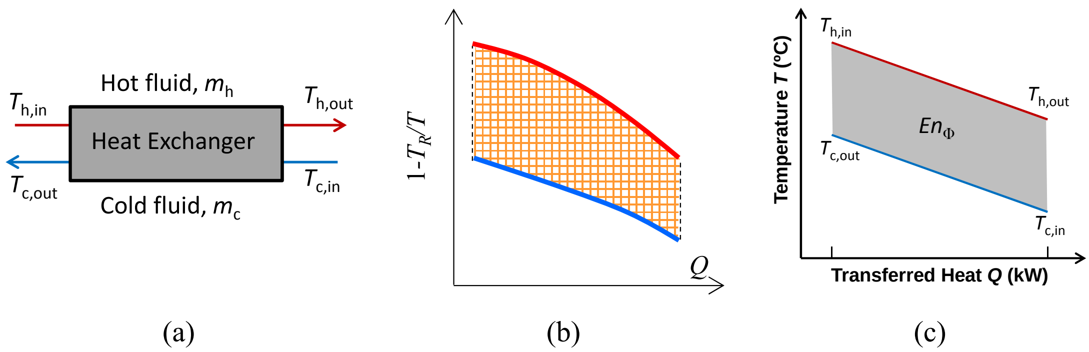


Fig. 2-2 T-Q diagram of counter-flow heat exchanger: (a) Scheme diagram; (b) exergy destruction; and (c) entransy dissipation.

For the counter-flow heat exchanger as shown in Fig. 2-2, the entransy dissipation during heat transfer process is Eq. (2-10), which can be depicted by Fig. 2-2(c).

$$\Delta E_{n,dis} = \int_0^Q (T_h - T_c)dQ \tag{2-10}$$

Entransy dissipation can be depicted in a *T*–*Q* diagram as Fig. 2-2(c). The horizontal axis of the *T*–*Q* diagram is the heat transferred in the heat exchanger, and the vertical axis is the temperature of hot and cold fluid corresponding to the heat transfer. The two lines will be straight if the specific heat of the fluid is constant, and the area between the two lines represents the entransy dissipation.

The *T*–*Q* diagram is a clear and straightforward way to describe heat exchange processes in HVAC systems. Temperature (*T*) and heat (*Q*) are equally important factors in HVAC systems, and they can be illustrated simultaneously in *T*–*Q* diagrams.

Equivalent thermal resistance

The traditional definition of thermal resistance is the ratio of temperature difference to transferred heat, as shown in Eq.(2-11), which is suitable for heat transfer between one heat source and one heat sink:

$$R = \frac{\Delta T}{Q} \tag{2-11}$$

In buildings, heat is transferred among several heat sources and heat sinks, which makes it impossible to define thermal resistance by temperature difference. The equivalent thermal resistance defined by entransy, written as Eq. (2-12), will address this issue:

$$R = \frac{\Delta E_{n,dis}}{Q^2} \tag{2-12}$$

To remove a certain amount of heat from the indoor environment to the outdoor environment, reducing entransy dissipation or reducing the equivalent thermal resistance is the key to achieving high-temperature cooling.

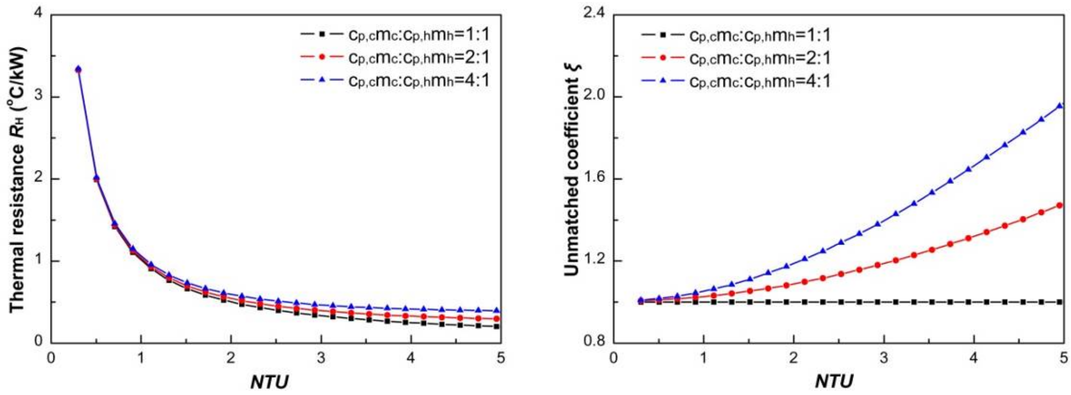
For counter-flow heat exchanger, as shown in Fig. 2-2(a), the thermal resistance can be expressed as Eq. (2-13), where ξ is the flow unmatched coefficient:

$$R = \frac{\xi}{UA} \tag{2-13}$$

In Eq. (2-13), *1/UA* represents the resistance caused by limited heat transfer capacity. The thermal resistance can be reduced by improving the heat transfer coefficient or increasing the heat transfer area. Flow unmatched coefficient ξ represents an increase of thermal resistance if the calorific capacities of the fluids are different.

If the temperatures of both fluids change during heat transfer, as shown in Fig. 2-2(c), flow unmatched coefficient ξ can be expressed as Eq.(2-14). The flow unmatched parameter is always greater than or equal to 1. When the calorific capacities of the two fluids are equal, the flow unmatched parameter is equal to 1. Fig. 2-3 shows the variation of thermal resistance and the unmatched parameter as a function of NTU of the heat exchanger.

$$\xi = \frac{P}{2}\frac{e^P + 1}{e^P - 1}, \text{where } P = UA \cdot \left(\frac{1}{c_{p,h}\dot{m}_h} - \frac{1}{c_{p,c}\dot{m}_c}\right) \tag{2-14}$$



(a) (b)

Fig. 2-3 Heat transfer in counter-flow heat exchanger: (a) Heat resistance; and (b) unmatched coefficient.

2.2 Common descriptions of transfer and mixing dissipations

Heat exchangers

Heat exchangers are the most common components in conventional HVAC systems, as shown in Fig. 2-2(a). In addition to water-water or water-air heat exchangers, evaporators and condensers in heat pumps are essentially heat exchangers as well.

Entransy dissipation ($\Delta E_{n,dis}$) is the loss in heat transfer ability in the heat exchanger due to irreversible heat transfer. Entransy dissipation, the shaded area between the two lines in the *T*-*Q* diagram shown in Fig. 2-2(c), can be expressed as the integration during the heat transfer process, written as Eq. (2-10). If the specific heat of fluids remains constant, entransy dissipation can be simplified as Eq.(2-15), where Q is the total transferred heat:

$$\Delta E_{n,dis} = \frac{1}{2}(T_{h,in} + T_{h,out} - T_{c,in} - T_{c,out})Q \quad (2\text{-}15)$$

Mixing process

The mixing of fluids of different temperatures occurs frequently in HVAC systems. Because mixing is an irreversible process, the entransy dissipation of a mixing process describes the loss of heat transfer capacity during mixing. Fig. 2-4(a) illustrates the mixing of two streams of fluids, e.g., when fresh air mixes with return air in the air-handling process. If the indoor environment can be considered as an infinite heat sink compared to the supply air, then a special case of mixing occurs, as shown in Fig. 2-4(b).

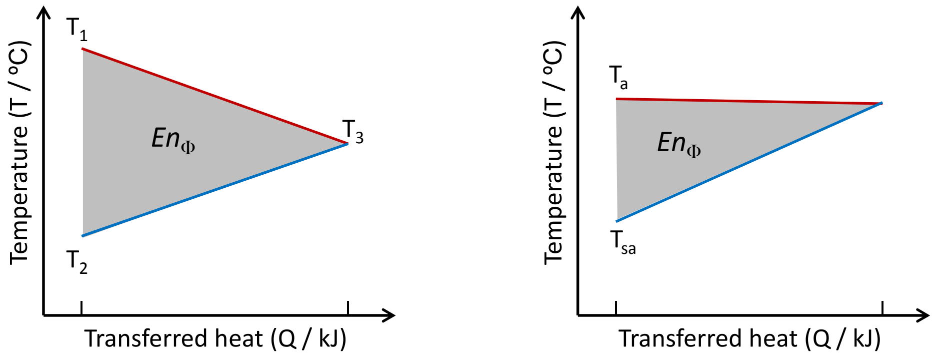
The entransy dissipation of a mixture can be expressed as Eq. (2-16), which is the shaded area between the two lines in Fig. 2-4(a):

$$\Delta E_{n,dis} = \frac{1}{2} c_p \frac{\dot{m}_1 \dot{m}_2}{\dot{m}_1 + \dot{m}_2} (T_1 - T_2)^2 \quad (2\text{-}16)$$

If the specific heat of fluids remains constant, the entransy dissipation illustrated in Figs. 2-4(a) and (b) can be simplified as Eqs. (2-17) and (2-18), respectively, where Q is the total transferred heat.

$$\Delta E_{n,dis} = \frac{1}{2}(T_1 - T_2)Q \quad (2\text{-}17)$$

$$\Delta E_{n,dis} = \frac{1}{2}(T_a - T_{sa})Q \quad (2\text{-}18)$$



(a) (b)

Fig. 2-4 Entransy dissipation of (a) mixture of two streams of fluids; and (b) mixture of supply air into room.

<u>Convective heat transfer between indoor air and heat sources</u>

Convective heat transfer between indoor air and heat sources occurs when a part of the indoor air is heated by a heat source and then mixes into the indoor environment. Entransy dissipation of the process is the sum of the entransy dissipation in the convective heat transfer and the mixing, also shown in Fig. 2-5.

$$\Delta E_{n,dis,i} = \frac{1}{2} q_{h,i} (2T_{h,i} - T_a - T_i) + \frac{1}{2} q_{h,i} (T_a + T_i - 2T_a) \quad (2\text{-}19)$$

The total entransy dissipation from heat sources to the indoor air can be expressed by Eq. (2-20):

$$\Delta E_{n,dis} = \sum_{i=1}^{n} q_{h,i} (T_{h,i} - T_a) \quad (2\text{-}20)$$

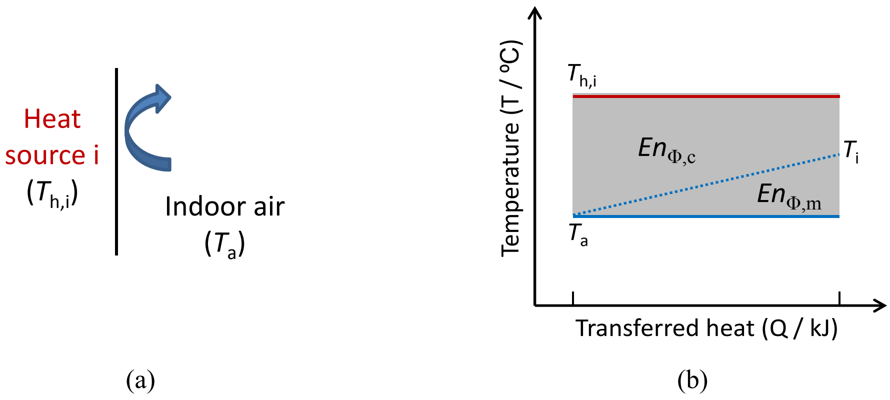


Fig. 2-5 Convective heat transfer between indoor air and heat source: (a) Schematic diagram; and (b) T-Q diagram.

Heat transfer between radiant floors and indoor environment

The total heat flux (q_{tot}) of a radiant floor is the sum of the convective heat flux (q_{conv}) and the radiant heat flux (q_{ra}). Convective heat flux represents the convective heat exchange between the floor surface and the indoor air. Radiant heat flux represents the radiant heat transfer between the floor surface and the indoor surfaces (e.g., building envelope and furniture).

$$q_{tot} = q_{conv} + q_{ra} \qquad (2\text{-}21)$$

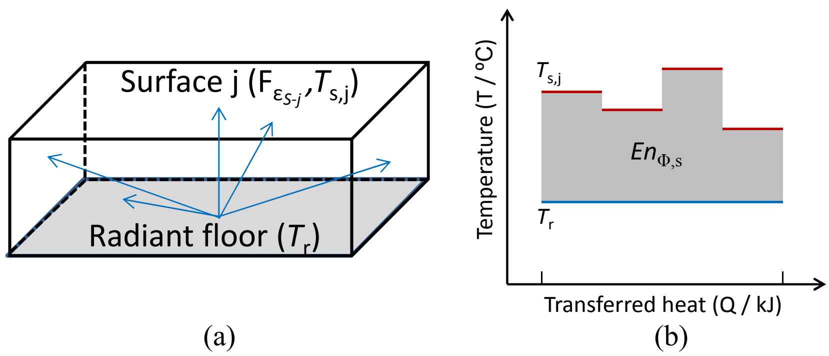


Fig. 2-6 Radiant heat transfer between radiant floor and indoor surfaces:

(a) Schematic diagram; and (b) T-Q diagram.

As shown in Fig. 2-6(a), the radiant heat exchange of the radiant floor with indoor surface j is calculated by

$$q_{r,j} = \sigma F_{\varepsilon_{s-j}}((T_{s,j} + 273.15)^4 - (T_r + 273.15)^4) \qquad (2\text{-}22)$$

$$F_{\varepsilon_{s-j}} = \frac{1}{[(1-\varepsilon_s)/\varepsilon_s] + (1/F_{s-j}) + (A_s/A_j)[(1-\varepsilon_j)/\varepsilon_j]} \quad (2\text{-}23)$$

where F_{s-j} represents the view factors of each surface and $F_{\varepsilon_{s-j}}$ represents the radiation interchange factors. Therefore, the entransy dissipation of a radiant heat exchanger can be expressed as

$$\Delta E_{n,dis,r} = \sum_{j=1}^{n} q_{r,j}(T_{s,j} - T_r) \quad (2\text{-}24)$$

For convective heat exchange, q_{conv} is calculated by the room air temperature (T_a) and the corresponding convective heat exchange coefficient (h_{conv}).

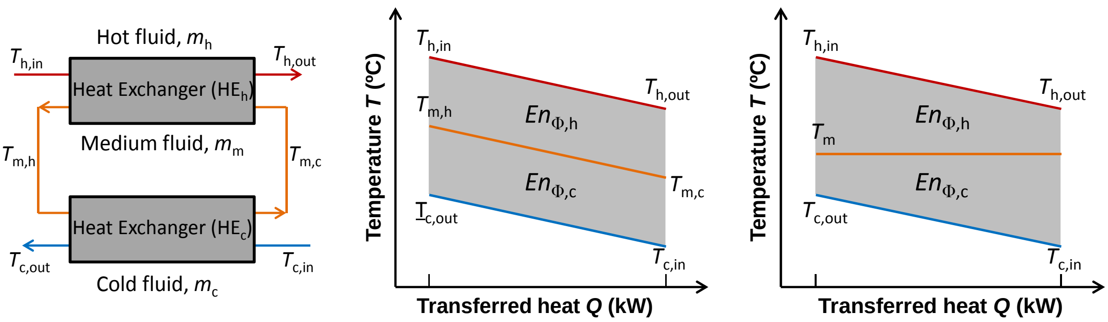
$$q_{conv} = h_{conv}(T_a - T_r) \quad (2\text{-}25)$$

Entransy dissipation of a convective heat exchanger can be expressed as

$$\Delta E_{n,dis,c} = q_{conv}(T_a - T_r) \quad (2\text{-}26)$$

Entransy analysis of heat exchangers in series

Fig. 2-7(a) shows two heat exchangers connected in series in order to transfer heat from a hot fluid to a cold fluid through a fluid medium. Fig. 2-7(b) depicts the T–Q diagram of these heat exchangers in series. If the fluid medium is a refrigerant, there will be a horizontal line in the T–Q diagram, as shown in Fig. 2-7(c).



(a) (b) (c)

Fig. 2-7 Heat exchangers in series: (a) Schematic diagram; (b) with cooling fluid; and (c) with evaporators.

The purpose of connecting heat exchangers in series is either to heat the cold fluid or to cool the hot fluid. Let us take the case of cooling a hot fluid from $T_{h,in}$ to $T_{h,out}$ as an example. The question will be how to optimize the heat capacity of the fluid medium and allocate the heat transfer area in order to obtain the highest temperature of the cold fluid in the condition of constant heat transfer capacity ($(UA)_h + (UA)_c = const$) for a certain heat capacity of the cold fluid ($c_{p,c}\dot{m}_c$). For heat transfer problems, the entransy balance equation indicates

$$\min(\Delta E_{n,dis}) \Leftrightarrow \min(E_{n,in}) \Leftrightarrow \max(T_{c,in}) \tag{2-27}$$

The total equivalent thermal resistance is the sum of thermal resistance of heat exchanger HE_{h} and heat exchanger HE_{c}:

$$R = R_h + R_c = \frac{\Delta E_{n,dis,h}}{Q^2} + \frac{\Delta E_{n,dis,c}}{Q^2} = \frac{\xi_h}{UA_h} + \frac{\xi_c}{UA_c} \tag{2-28}$$

The optimization of the heat capacity of the fluid medium and the allocation of the heat transfer area can be obtained by solving Eq. (2-29):

$$\frac{\partial R}{\partial (c_{p,m}\dot{m}_m)} = 0，\frac{\partial R}{\partial (UA)_h} = 0 \tag{2-29}$$

Fig. 2-8 gives an example of free cooling by outdoor air. For different flow rates of fluid medium, the total UA of the two heat exchangers will reach the lowest value when the flow rates of the three fluids match each other. As shown in Fig. 2-9, the flow unmatched coefficient equals 1.

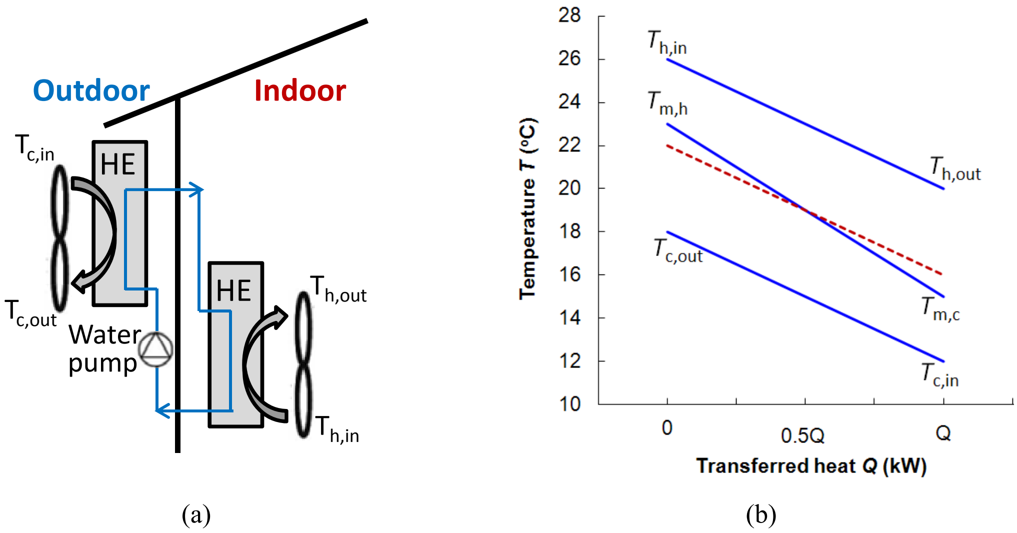


Fig. 2-8 Example of heat exchangers in series (water circulation):

(a) Schematic diagram; and (b) T-Q diagram.

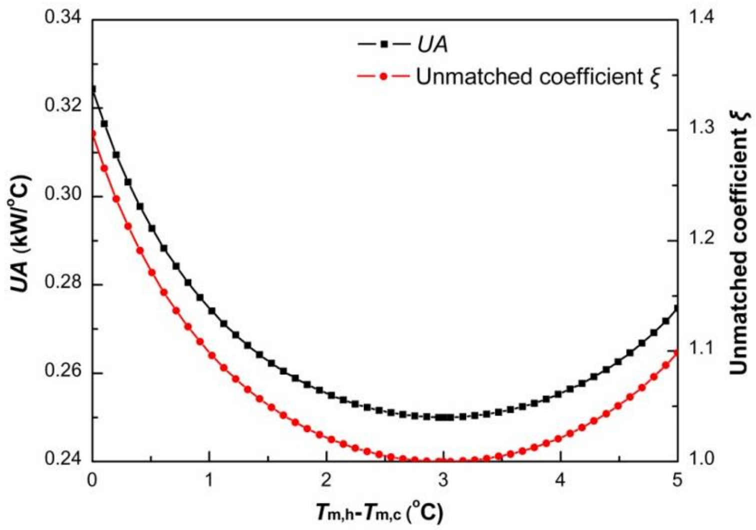
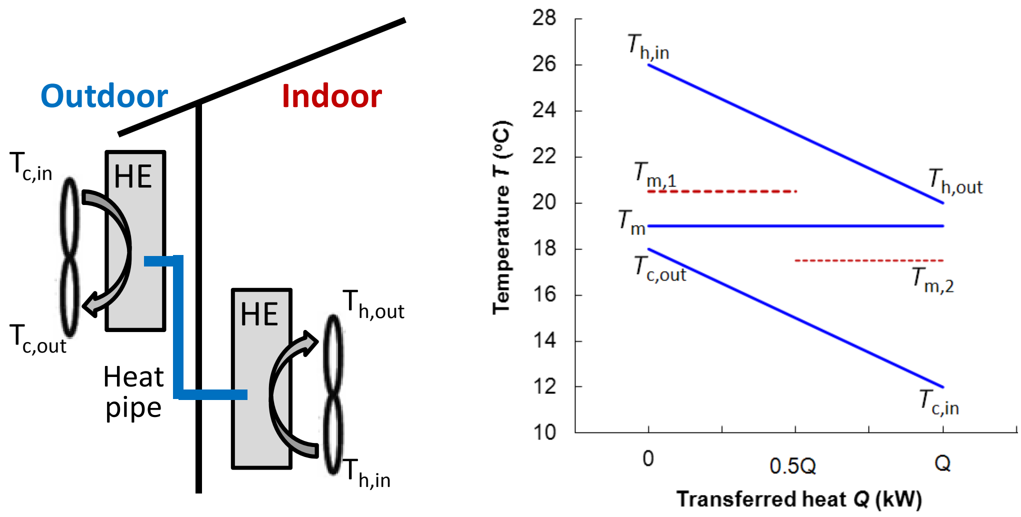


Fig. 2-9 UA and unmatched coefficient ξ of water circulation.

If the fluid medium is a heat pipe as shown in Fig. 2-10, the heat transfer is achieved by evaporation and condensation of the refrigerant. The flow rate of the fluid medium cannot match that of the hot fluid and cold fluid; for example the flow unmatched coefficient ξ of a single-stage heat pipe is 1.297. The flow unmatched coefficient ξ can be reduced by employing a multistage heat pipe as shown in Fig. 2-10(b). The *UA* is changed from 0.324 kWºC to 0.253 kWºC when a single-stage heat pipe is separated into four stages, as shown in Fig. 2-11.



(a) (b)

Fig. 2-10 An example of heat exchangers in series (heat pipe): (a) schematic diagram; and (b) T-Q diagram.

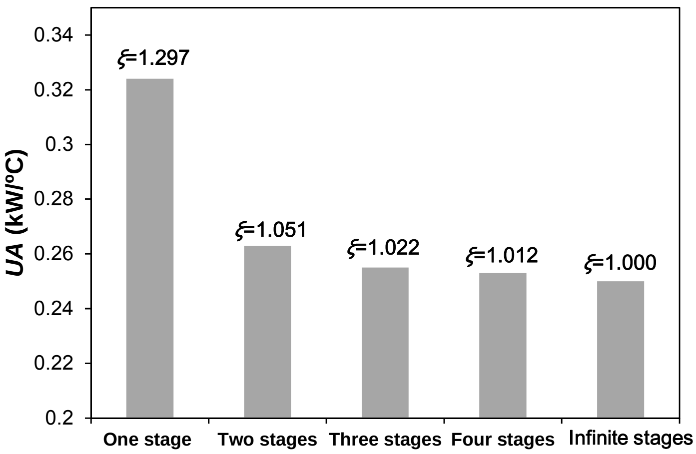


Fig. 2-11 UA and unmatched coefficient ξ of multistage heat pipe

2.3 Novel perspective of entransy analysis method

The core task of establishing a suitable indoor temperature and humidity environment is to remove the additional heat and moisture from the indoor to the outdoor. This process is regarded as composing a complex system consisting of passive building envelops and active air-conditioning systems. The indoor cooling/humidity load can be removed from the indoor environment to the outdoor environment passively through building envelops by heat transfer and infiltration; this process is a heat transfer process instead of a heat–work conversion process. Entransy is a thermal parameter for analyzing the heat transfer process. Through entransy dissipation and the equivalent thermal resistance, the characteristics of the heat transfer process can be elucidated effectively. Reducing the entransy dissipation helps to decrease the driving temperature difference of passive systems such as building envelops with a fixed cooling load. Therefore, the heat exhaust requirement may be met with a smaller temperature difference between indoor and outdoor ($T_{\mathrm{room}} - T_0$), the adjustable range of building envelops may be enlarged, the available duration of using building envelops to remove heat may be extended, and the input of the active system may be reduced.

Fig. 2-12 describes the change in the analysis method of system performance optimization. In the conventional analysis method, called external optimization, for a given system process, each input parameter of the system is adjusted to obtain the optimal output parameter, e.g., highest system efficiency. Generally, several comparative system processes are indicated empirically and then the optimized system performance of each system process is achieved by the method described above; hence, the optimal system process can be elected. However, the methodology in this report transforms the conventional external optimization into a method whereby the internal losses of the system processes are reduced, namely, a method of decreasing the temperature difference based on the causes of consumption of the temperature difference to guide the system performance optimization and reduction in energy consumption.

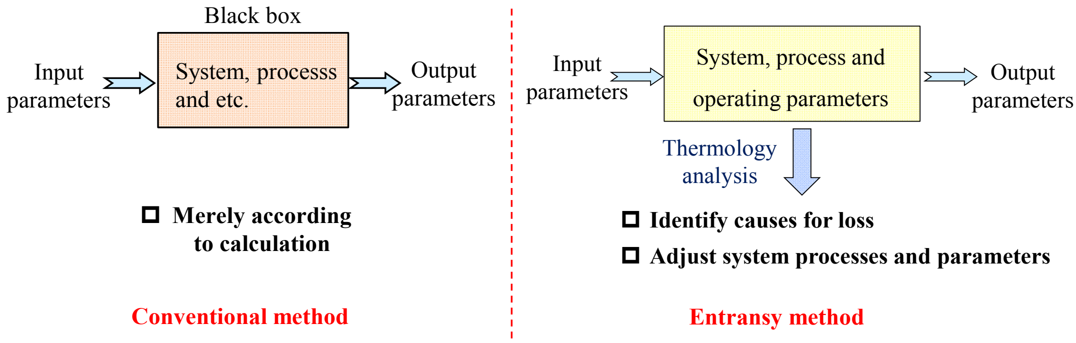


Fig. 2-12 Analysis methods of system performance optimization

3. Entransy analysis of HVAC system

3.1 Entransy dissipation in HVAC system

The processes in a typical central air-conditioning system are represented in a *T–Q* diagram to analyze the transfer properties of the whole system. For the FCU+OA system (fan coil unit combined with outdoor air processing) of an air-cooled condenser, the *T–Q* diagram shown in Fig. 3-1 describes the fan coil process. The heat transfer from FCU to evaporation temperature includes heat collection by indoor FCU, heat transfer from indoor air to chilling water and to evaporator, and heat transfer from condenser to outdoor dry bulb temperature. The difference in heat flux between the evaporator side and condenser side equals the input work of the refrigeration cycle, and the surrounding area of all heat transfer processes corresponds to the entransy dissipation in heat transfer.

The aim of heat exhaust of this system is to remove heat Q from indoor temperature to outdoor sink (difference between outdoor dry bulb temperature and indoor temperature is $\Delta T_{S\text{-}S}$). The theoretical minimum entransy dissipation ΔE_n is the product of Q and $\Delta T_{S\text{-}S}$, with the actual temperature difference ΔT_{HP} much greater than $\Delta T_{S\text{-}S}$, and a much higher entransy dissipation of the whole heat exhaust than the theoretical minimum. Because there are various intermediate processes in heat exhaust, each process consumes a certain temperature difference or causes entransy dissipation. Taking heat transfer on the condenser side (air-cooled) for illustration, when the refrigerant transfers heat to the air, the total entransy dissipation $\Delta E_{n,c}$, which consists of dissipation in heat transfer in the condenser and mixing entransy dissipation due to mixing of exhaust air of the condenser with ambient air, can be calculated using Eq. (3-1):

$$\Delta E_{n,c} = Q_c \cdot (\Delta T_5 + \Delta T_6) \tag{3-1}$$

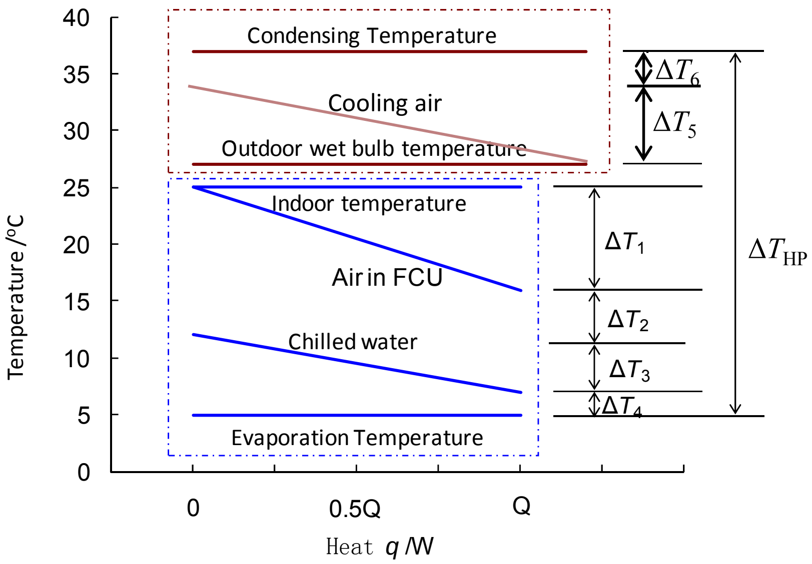


Fig. 3-1 T-Q diagram of air-cooled process in air-cooled condenser

Similarly, entransy dissipation for each process on the evaporator side in Fig. 3-1 can be formulated using heat Q and the temperature difference. The total transfer entransy dissipation of all the processes from indoor temperature to evaporation temperature can be expressed as Eq. (3-2).

$$\Delta E_{n,e} = Q \cdot (\Delta T_1 + \Delta T_2 + \Delta T_3 + \Delta T_4) \qquad (3\text{-}2)$$

Therefore, the required driving temperature difference, ΔT_{HP}, of a heat pump can be expressed as Eq. (3-3).

$$\Delta T_{HP} = \Delta T_{s-s} + \frac{\Delta E_{n,c}}{Q} + \frac{\Delta E_{n,e}}{Q} \qquad (3\text{-}3)$$

For the FCU+OA system of a water-cooled condenser, the T–Q diagram shown in Fig. 3-2 describes the FCU process. Similar to the air-cooled system in Fig. 3-1, entransy dissipation on both the evaporator and condenser sides in the water-cooled system can be formulated by the correlation between heat exhaust and the temperature difference. A refrigeration cycle is therefore expected to provide the required driving temperature difference, because there are various intermediate processes in heat exhaust and each process consumes a certain temperature difference, such as $\Delta T_1 - \Delta T_6$ in Fig. 3-1 and $\Delta T_1 - \Delta T_7$ in Fig. 3-2, or causes entransy dissipation, such as total entransy dissipation on the condenser side $\Delta E_{n,c}$ and total entransy dissipation on the evaporator side $\Delta E_{n,e}$. The existence of intermediate processes eventuate a significant increase in the temperature difference and entransy dissipation during the actual heat exhaust, demanding a larger temperature difference ΔT_{HP} in the cycle than $\Delta T_{S\text{-}S}$. A less entransy dissipation on the condenser and evaporator sides means a smaller driving temperature difference ΔT_{HP} and better performance of the heat pump.

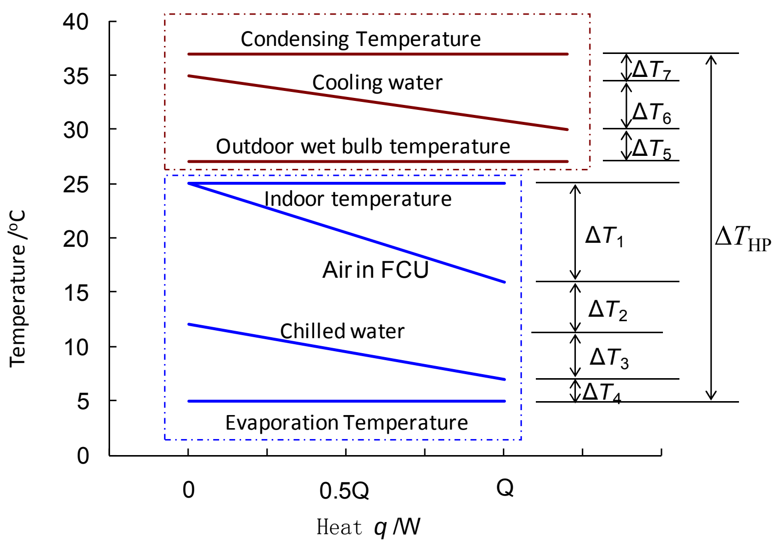


Fig. 3-2 T-Q diagram of air-cooled processing for a water-cooled condenser

3.2 Focus on T-Q diagram and thermal resistance

3.2.1. T-Q diagram and thermal resistance from heat sources to heat sinks

An active air-conditioning system contains a series of processes from collection, transportation, and cooling source equipment. The temperature difference ΔT and the humidity difference $\Delta\omega$ (or the water vapor pressure difference Δp) are the driving force of the heat transfer and mass transfer, respectively. The driving force is consumed in every process of the system to realize the heat and mass transfer. When the temperature level of the outdoor heat sink is appropriate or the temperature difference ΔT between the indoor heat source and the outdoor heat sink meets the requirement of the driving temperature difference ΔT_{total} consumed in the total process, the active air-conditioning system can use the natural cooling source directly to remove the additional indoor heat without any mechanical refrigeration cycle. Entransy dissipation analysis can provide an effective guide for performance improvement of the system. As shown in Eq. (3-4), the loss in every process of the active air-conditioning system contributes to the total loss (entransy dissipation) of the system. Reducing the consumed driving force (temperature difference) and the entransy dissipation (or the equivalent thermal resistance) in every process helps in decreasing the total entransy dissipation of the system.

$$\begin{aligned}\Delta E_{n,\mathrm{dis}} &= \Delta E_{n,1} + \Delta E_{n,2} + \ldots \Delta E_{n,\mathrm{n}} \\ &= Q_{ac} \cdot \Delta \overline{T}_{\mathrm{total}} = Q_{ac} \cdot (\Delta \overline{T}_1 + \Delta \overline{T}_2 + \ldots \Delta \overline{T}_{\mathrm{n}})\end{aligned} \tag{3-4}$$

Fig. 3-3 shows the T–Q diagram of the processes of removing the cooling load from indoor heat sources to outdoor heat sinks with natural cooling sources. With a fixed cooling load Q_{ac}, reducing the entransy dissipation or the equivalent thermal resistance in every process helps in decreasing the total thermal resistance of the system, decreasing the total entransy dissipation $\Delta E_{n,\mathrm{dis}}$, and decreasing the required driving temperature difference $\Delta \overline{T}_{\mathrm{total}}$ of the system. Therefore, for an active air-conditioning system using natural sources to remove heat, the required temperature level of the outdoor heat sinks (natural cooling sources) can be decreased and the range of available natural cooling sources and the working duration of natural cooling sources can be extended by reducing the entransy dissipation in every process.

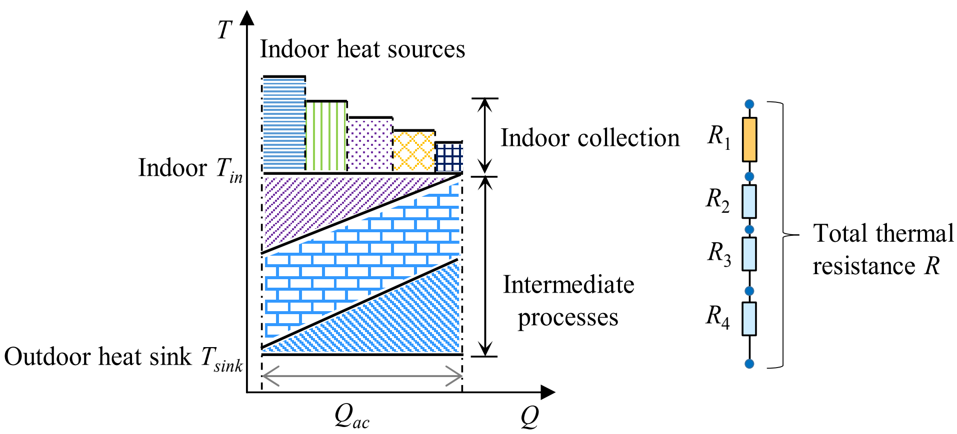


Fig. 3-3 Losses in all processes of an active air-conditioning system

When outdoor heat sinks cannot be used directly to remove heat, the active air-conditioning system needs to establish mechanical refrigeration systems along with heat–work conversion processes to provide the driving force for the heat transfer processes. According to the processes of the active air-conditioning system shown in Fig. 1-4, the heat pump (or the refrigeration) cycle only exists in the cooling source equipment and the other processes are mostly heat transfer processes with no heat–work conversion involved. The heat–work conversion is considered to be analyzed by the thermal parameter exergy, whereas the heat transfer processes are considered to be analyzed by the thermal parameter entransy.

3.2.2. T-Q diagram and thermal resistance in HVAC system

Active HVAC systems typically include several terminal devices and processes; hence complex transfer networks are constructed to meet various requirements of processing cooling or heating load in multiple terminal devices. Entransy dissipation and equivalent thermal resistance may be applied in analyzing complex heat transfer networks consisting of terminal devices connected in series or parallel. Taking an active heat and air-conditioning system for instance, Fig. 3-4(a) illustrates a system that contains multiple terminal devices and the process from water chilling units to FCU terminal equipment on different floors, where only sensible heat is considered to simplify analysis. The process contains mixing of air supply from FCU and indoor air, heat transfer between water and air in FCU, mixing of different chilled water outlet flows, and mixing of return and supply chilled water in the refrigerator side bypass. Because of mixing and limited heat transfer capacity, entransy dissipation and thermal resistance are inherent in the above processes. In the serial–parallel thermal resistance network shown in Fig. 3-4(b), heat transfer takes place from indoor air temperature (T_{in}) to average chilled water temperature (T_m) in the described process, in which $T_{in,1}$, $T_{in,2} \ldots T_{in,\mathrm{n}}$ refer to indoor air temperature, $\overline{T}_{a,1}$, $\overline{T}_{a,2} \ldots \overline{T}_{a,\mathrm{n}}$ refer to the average temperature of supply air from FCU and return air, and $\overline{T}_{m,1}$, $\overline{T}_{m,2} \ldots \overline{T}_{m,\mathrm{n}}$ refer to the average temperature of supplied chilled water and return water in FCU. The thermal resistance network includes the mixture thermal resistance $R_{mix,a}$ of supply from FCU and indoor air, thermal resistance R_{FCU} of water–air heat transfer process in fan coils, mixture thermal resistance $R_{mix,w}$ from each fan coil branch to converge at the main return pipe (the mean temperature of each branch $\overline{T}_{m,1}$, $\overline{T}_{m,2} \ldots \overline{T}_{m,\mathrm{n}}$ mix into $\overline{T}_m$).

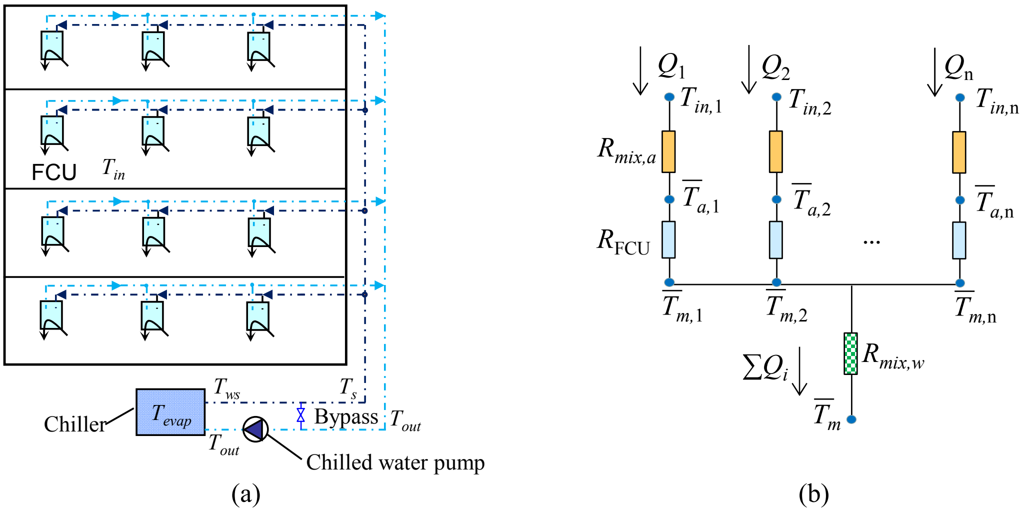


Fig. 3-4 Complex heat transfer network in active heating and air-conditioning system: (a) operating schematic; and (b) network of thermal resistance ($T_{in} \to \overline{T}_m$)

The process involving three FCU terminal devices shown in Fig. 3-5 further illustrates the heat transfer from indoor air temperature T_{in} to the average chilled water temperature ($\overline{T}_m$), and then to evaporation temperature of chiller units T_{evap}. $T_{in,1}$, $T_{in,2}$, and $T_{in,3}$ correspond to the three FCU terminal devices. The temperature of the supplied chilled water (T_s) entering each FCU changes to $T_{out,1}$, $T_{out,2}$, and $T_{out,3}$ after heat transfer with air; then all of the FCU outlet water mixes to a uniform temperature T_{out}. The chilled return water on the refrigerator side bypass pipe (T_{out}) and outlet water from refrigerator (T_{ws}) will mix to temperature T_s. The temperature of chilled water changes from T_{out} to T_{ws} by heat transfer with the refrigerant in the evaporator. From the analysis of the T–Q diagram above, entransy dissipation caused by mixing and heat transfer can be obtained in the complex heat transfer process.

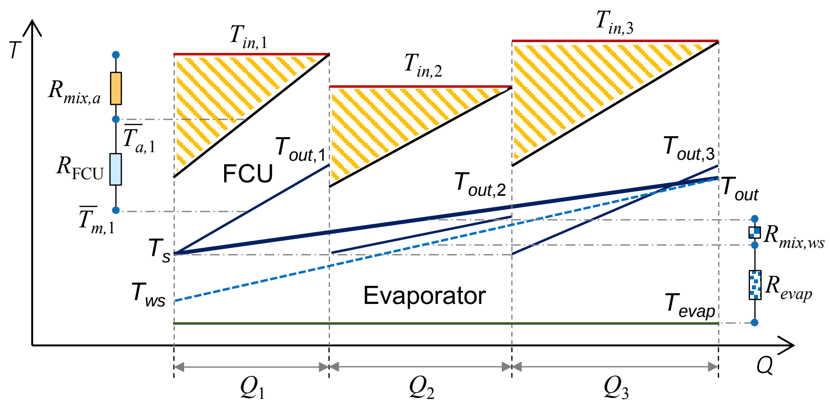


Fig. 3-5 T-Q diagram of heat transfer process with multiple terminal devices

Fig. 3-6(a) further illustrates the network depicted in Fig. 3-4(a), which comprises mixture thermal resistance and heat transfer thermal resistance. The mixture thermal resistance exists when supply air from FCU mixes with indoor air, or when the return chilled water in different branches mixes to the total return water temperature, or when there is mixing in the chilled water bypasses. The node temperatures all correspond to the average fluid temperature during the heat transfer process when an equivalent thermal resistance is applied in heat transfer analysis in Fig. 3-4(b), whereas in the network shown in Fig. 3-6(a), the adjusted resistance E is introduced to represent the process between the average supply chilled water temperature ($\overline{T}_{m,1}, \overline{T}_{m,2} \ldots \overline{T}_{m,\mathrm{n}}$) and each branch to the return water temperature, as expressed by Eq. (3-5).

$$E = -\frac{1}{2c\dot{m}} \tag{3-5}$$

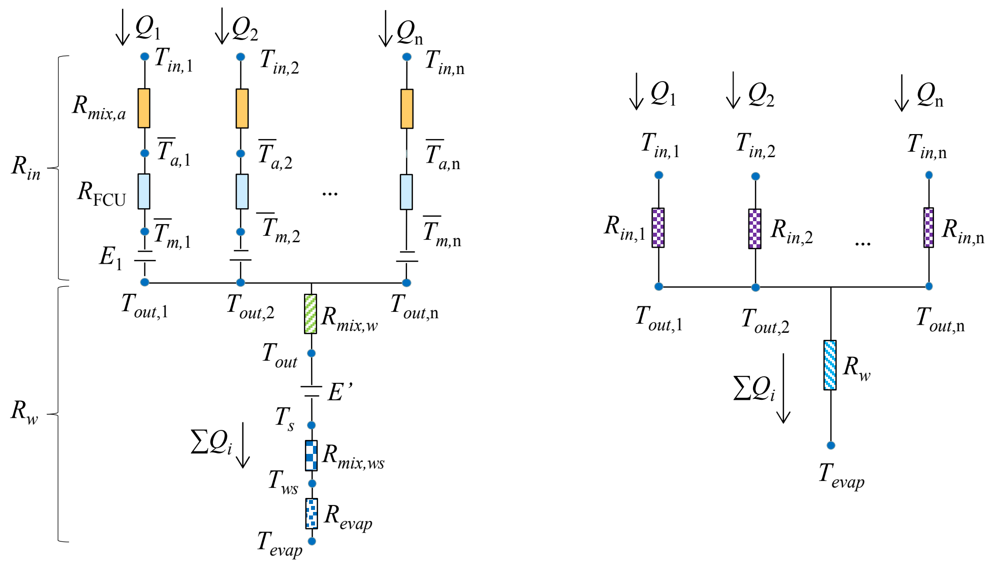


Fig. 3-6 Transfer network of thermal resistance in complex heat transfer process: (a) Transfer network of thermal resistance ($T_{in} \rightarrow T_{evap}$); and (b) simplified transfer network of thermal resistance ($T_{in} \rightarrow T_{evap}$).

For each branch in the diagram, the adjusted resistance E between the average supply chilled water temperature and return chilled water temperature respectively are

$$E_{w,1} = -\frac{1}{2c\dot{m}_{w,1}},\ E_{w,2} = -\frac{1}{2c\dot{m}_{w,2}},\ \ldots\ldots,\ E_{w,n} = -\frac{1}{2c\dot{m}_{w,n}} \tag{3-6}$$

For the process in bypass installed in the chilled water return and supply main pipe, the adjusted resistance E_{ws} between the chilled water temperature T_{ws} and the average chilled water temperature in the evaporator is

$$E_{ws} = -\frac{1}{2c\sum \dot{m}_{w,i}} \tag{3-7}$$

Therefore, for the process from indoor air temperature $T_{in,i}$ to chilled water outlet temperature in FCU for branch i in Fig. 3-4(a) (i = 1,2,……,n), the heat flux is Q_i and the total thermal resistance of the branch can be formulated as Eq. (3-8):

$$\text{Branch 1: } R_{in,1}{=}R_{mix,a,1}+R_{FCU,1}+E_{w,1}$$

$$\text{Branch i: } R_{in,i}{=}R_{mix,a,i}+R_{FCU,i}+E_{w,i} \tag{3-8}$$

$$\text{Branch n: } R_{in,n}{=}R_{mix,a,n}+R_{FCU,n}+E_{w,n}$$

For branch *i*, the mixture thermal resistance $R_{mix,a,I}$ of air supply from FCU and indoor air as well as the thermal resistance of air–water heat transfer in FCU $R_{FCU,I}$ are as follows:

$$R_{mix,a,i}=\frac{1}{2c\dot{m}_{a,i}},\ R_{FCU,i}=\frac{1}{2}(\frac{1}{c\dot{m}_{a,i}}-\frac{1}{c\dot{m}_{w,i}})\cdot\frac{e^{[NTU(1-C_r)]}+1}{e^{[NTU(1-C_r)]}-1} \tag{3-9}$$

With regard to chilled water supplied from different FCU branches to the chiller unit (T_{evap}), as shown in Fig. 3-6(a), the mixture thermal resistance is present in the processes where outlet water at different temperatures $T_{out,i}$ mix into return water at a uniform temperature T_{out}; the adjusted resistance *E'* can be applied to represent the process from chilled outlet water temperature T_{out} to supply water temperature T_s. The bypass of chilled return water and supply water may elicit the mixture thermal resistance $R_{mix,ws}$. Additionally, heat transfer between the refrigerant and chilled water on the evaporator side may also cause thermal resistance, namely R_{evap}. To summarize, the total thermal resistance in the described process can be formulated as Eq. (3-10):

$$R_w{=}R_{mix,w}+E'+R_{mix,ws}+R_{evap} \tag{3-10}$$

The mixture thermal resistance $R_{mix,w}$ for the processes where outlet water from different FCU branches mix into return water at a uniform temperature is expressed as Eq. (3-11), where $R'_{mix,w}$ can be calculated by the method mentioned in section 2.2. To be more specific, the mixture thermal resistance R_{mix} equals the ratio of $\sum Q_{mix,i}$ squared to $\sum Q_i$ squared, where $\sum Q_{mix,i}$ represents the heat flux in the mixture process and $\sum Q_i$ represents as the total heat handled by the terminal FCU. Similarly, the mixture thermal resistance $R_{mix,ws}$ caused when the chilled return water bypasses the supply water can be calculated using the computing method in Eq. (3-11). The adjusted resistance *E'* between the chilled return and supply water at the temperatures of T_{out} and T_s, respectively, as well as the thermal resistance on the evaporation side of chilling units represented by R_{evap} can be formulated as in Eq. (3-12).

$$R_{mix,w}{=}\frac{\Delta E_{n,mix,w}}{(\sum Q_{\mathrm{i}})^2}{=}\frac{(\sum Q_{mix,w,\mathrm{i}})^2}{(\sum Q_{\mathrm{i}})^2}\cdot R'_{mix,w} \tag{3-11}$$

$$E'{=}\frac{1}{c\dot{m}_w},\ R_{evap}{=}\frac{T_{ws}-T_{evap}}{\sum Q_{\mathrm{i}}} \tag{3-12}$$

Accordingly, for a complex heat transfer process containing multiple terminal devices mentioned above, the heat resistance $R_{in\text{-}rw,\mathrm{i}}$ can represent the heat transfer process in different branches between indoor air temperature $T_{in,i}$ and outlet chilled water temperature $T_{in,i}$. $R_{in\text{-}rw,i}$

combined with R_{mix}, R_{cm}, and $R_{s\text{-}e}$ can describe the thermal resistance characteristics of the entire transfer network. For this heat transfer process, the transfer network of thermal resistance is shown in Fig. 3-6 (b), which represents the heat transfer from the temperature of each indoor outlet $T_{in,i}$ and evaporation temperature T_{eva}, with the total thermal resistance calculated by Eq. (3-13).

$$R_{总} = \sum \frac{Q_i^2}{(\sum Q_i)^2} \cdot R_{in,i} + R_w \tag{3-13}$$

Given the above equations for thermal resistance in the entire transfer network and in each section, it may be concluded that for the heat transfer process from different indoor temperatures T_{in} to evaporation temperature of chilling unit T_{evap}, there exist two types of thermal resistance, namely, mixture thermal resistance and heat transfer thermal resistance. The mixture thermal resistance is caused by the mixing of air (or chilled water) at different temperatures, such as $R_{mix,a}$ and $R_{mix,w}$ mentioned before. The heat transfer thermal resistance is caused by the heat transfer between two fluids, for example, R_{FCU} for water-to-air heat transfer process in FCU and R_{evap} on the evaporation side. By analyzing the thermal resistance in each section, an approach to reducing the total system thermal resistance can be obtained. Therefore, it is recommended to reduce the number of unnecessary sections in heat transfer to attain a lower thermal resistance. Avoiding mixing is another effective way to attain less mixture thermal resistance. For example, avoiding mixing of chilled water outlet flows at different temperatures contributes to lowering the mixture thermal resistance $R_{mix,w}$, and avoiding mixing of chilled supply and return water in the bypass pipe contributes to reducing the mixture thermal resistance $R_{mix,ws}$. Additionally, by ameliorating the heat transfer between water and air in FCU and matching the properties of heat transfer between refrigerant and chilled water on the evaporation side, the heat transfer thermal resistance R_{FCU} and R_{evap} can be reduced to enhance the heat transfer performance of the system main circuit.

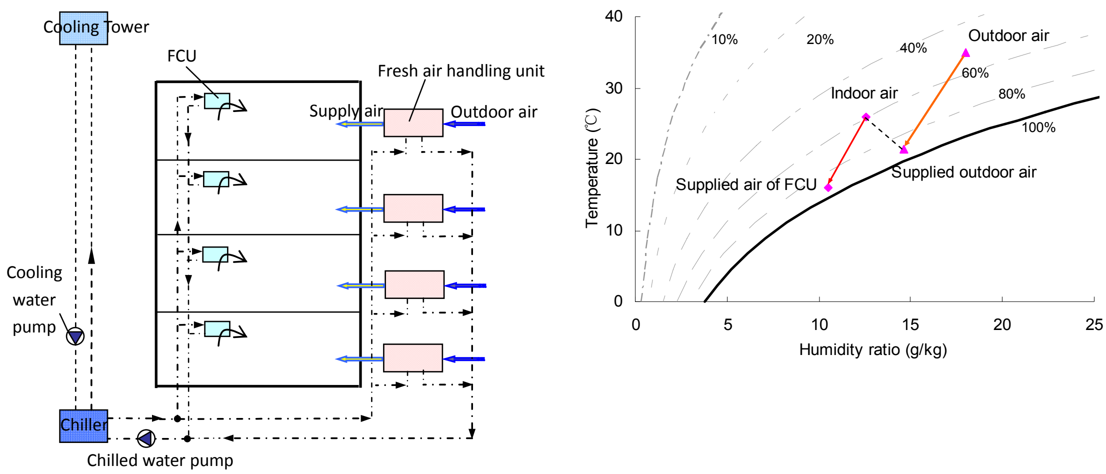
Appendix A Typical heating/cooling systems in various countries

A.1 China

a) Cooling systems

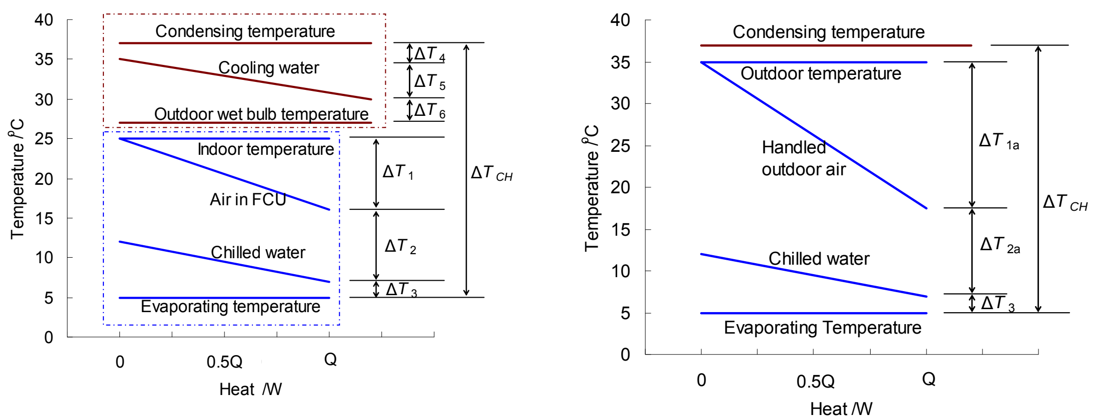
FCU+OA system

Fan coil unit (FCU) and outdoor air (OA) system is the most common air conditioning system in China. Fig. A-1 gives its system principle and Fig. A-2 shows the *T-Q* figures for the FCU and OA handling processes.



(a) (b)

Fig. A-1. Typical air-conditioning system (FCU+OA system): (a) schematic of the system; and (b) psychrometric chart of the air handling process.

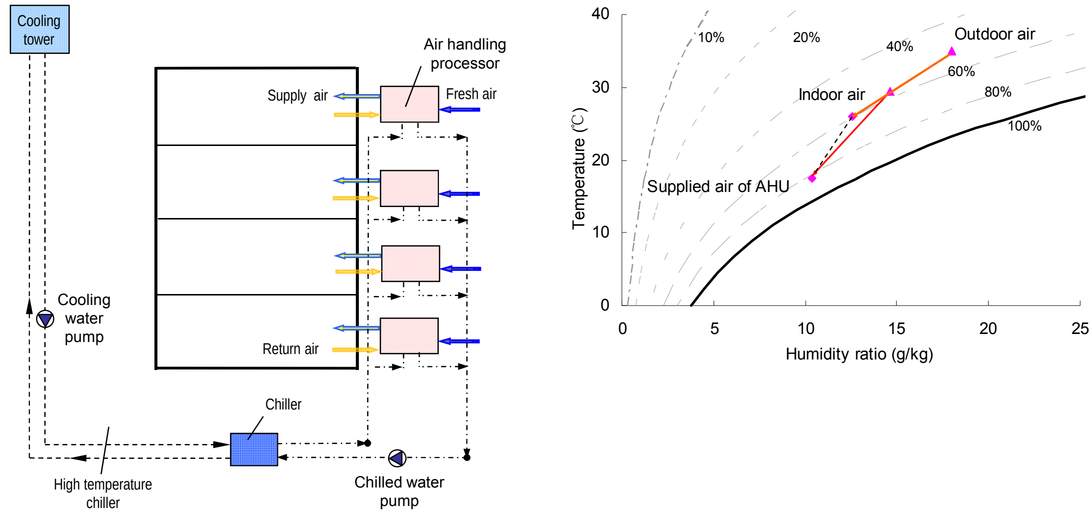


(a) (b)

Fig. A-2. Temperature difference (T-Q figure) of the typical air-conditioning system (FCU+OA): (a) the FCU subsystem; and (b) the outdoor air handling subsystem.

All air system

VAV (variable air volume) or CAV (constant air volume) system is also a common type of centralized air-conditioning system in China. Fig. A-3 gives its system principle and Fig. A-4 shows the *T-Q* figure for the air handling process.



(a) (b)

Fig. A-3. Typical air-conditioning system (all air system): (a) schematic of the system; and (b) psychrometric chart of the air handling process

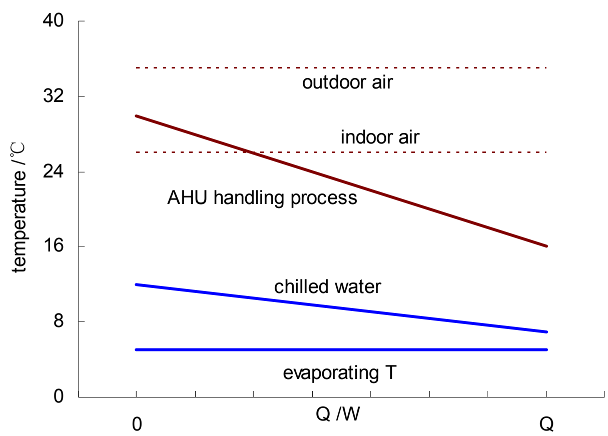
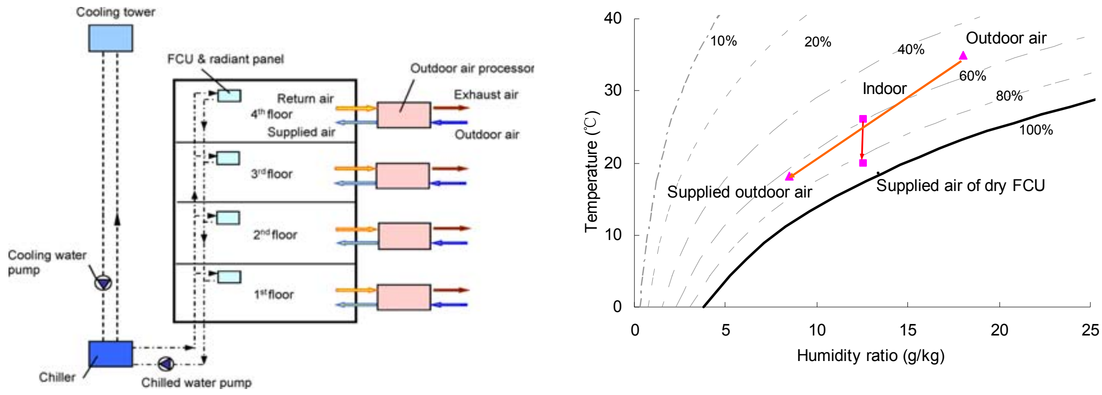


Fig. A-4. Temperature difference (T-Q figure) of the typical air-conditioning system (all air system).

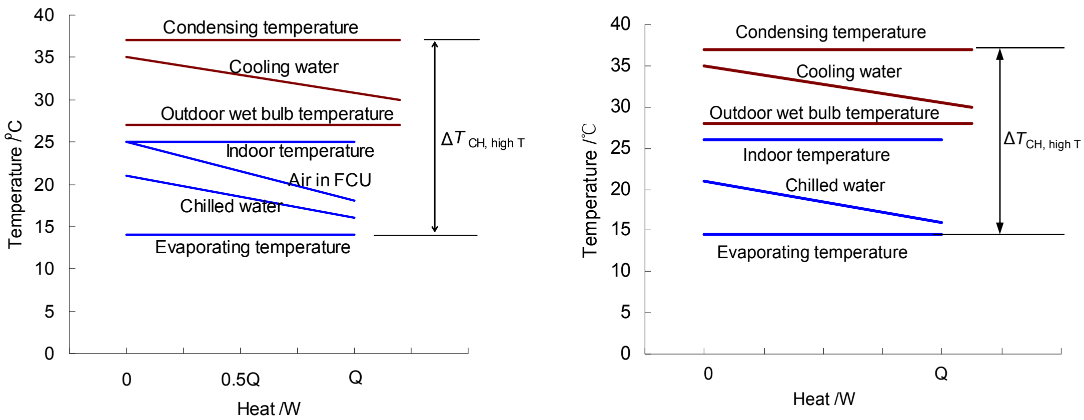
THIC system

Temperature and humidity independent control (THIC) system is developing very quickly nowadays. The system operating principle is shown as Fig. A-5 and *T-Q* figure of the temperature control subsystem is listed in Fig. A-6.



(a) (b)

Fig. A-5. The THIC air-conditioning system: (a) typical outdoor air system with FCUs; and (b) psychrometric chart of the air handling process.



(a) (b)

Fig. A-6. Temperature difference (T-Q figure) of a temperature control subsystem of THIC system: (a) dry FCU; and (b) radiant terminal.

b) Heating systems

CHP system

Combined Heat and Power (CHP) system is commonly used in China for power and district heating. Fig. A-7 gives the operating schematic of a coal-fired CHP system and Fig. A-8(a) shows the *T-Q* figure for this system.

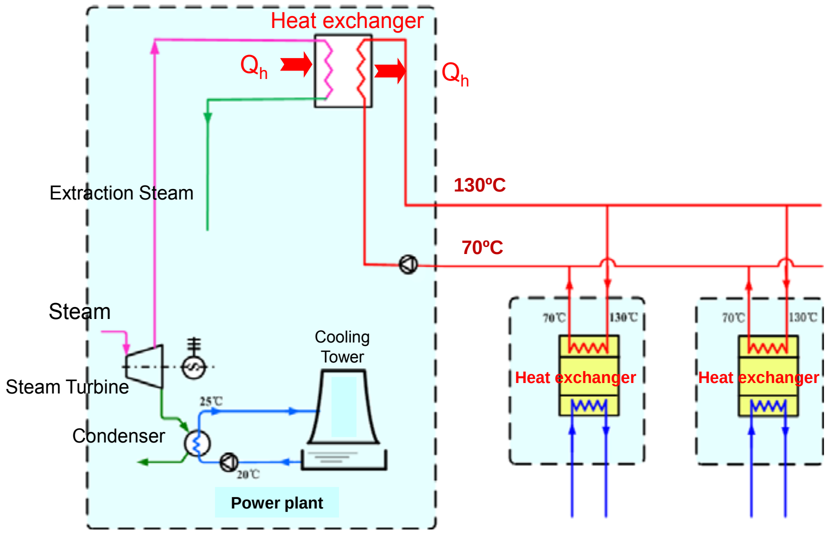


Fig. A-7. Conventional heating system with Co-generation (Coal-fired).

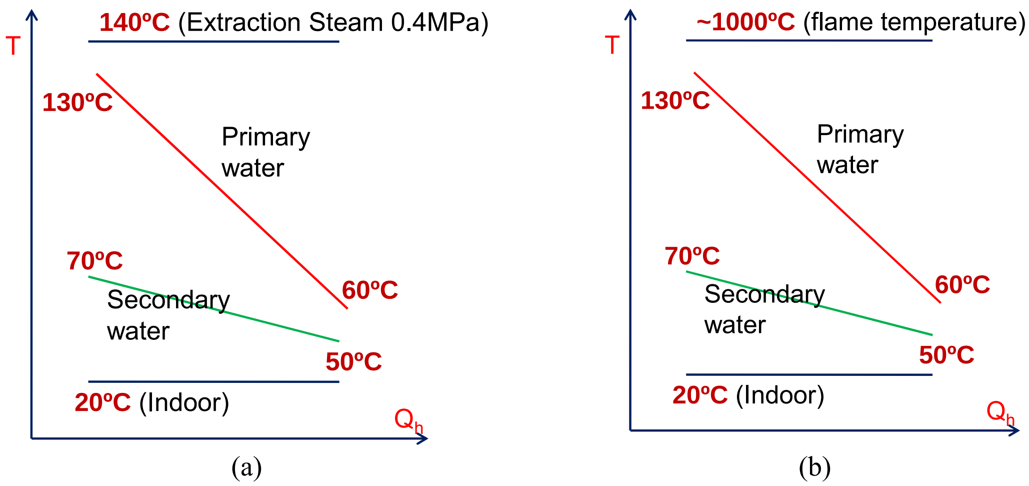


Fig. A-8. Temperature difference (T-Q figure) of conventional heating system: (a) CHP system (Coal-fired); and (b) boiler heating system.

Boiler system

The boiler system is also used in China for district heating. Fig. A-9 gives its operating schematic and Fig. A-8(b) shows the *T-Q* figure.

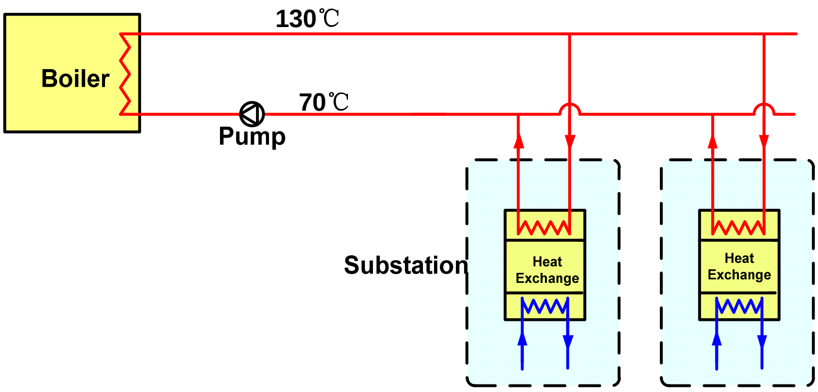


Fig. A-9. Conventional boiler heating system.

Heat pump system

Heat pump could be used to produce hot water with appropriate temperature for heating and Fig. A-10 gives the system schematic. The fluid providing heat for the evaporator could be air or water and nowadays the water source heat pump system is becoming popular in China. Figs. 2-11 (a) and (b) gives the *T-Q* figures of water source heat pump systems with FCU terminal and radiant terminal respectively.

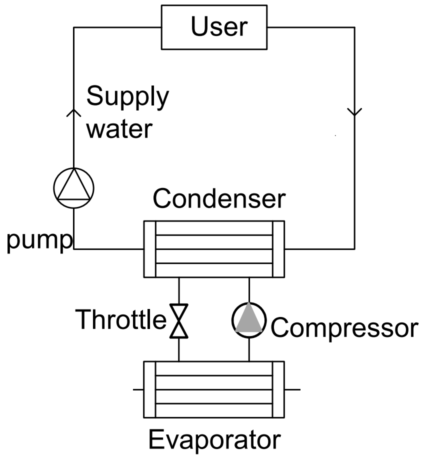


Fig. A-10. Heat pump system for heating.

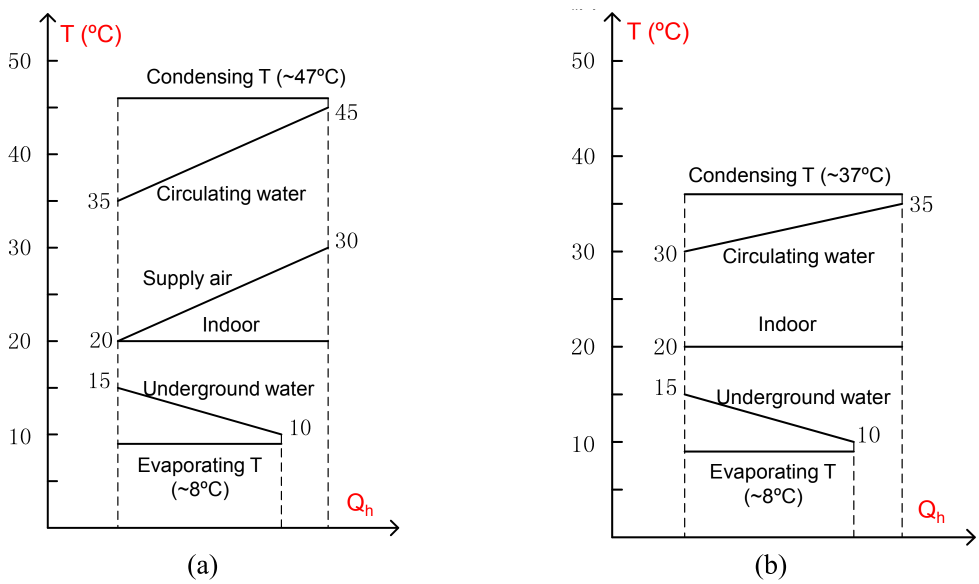


Fig. A-11. Temperature difference (T-Q figure) of the heat pump system for heating: (b) FCU; and (b) radiant terminal.

A.2 Italy

a) Introduction

Italy is a very difficult country to outline in a single climatic model. Indeed, its great extension in latitude (about 12°) involves a variety of climatic conditions which range from cold mountain climates of the northern Alps regions to warmer climates of the insular and peninsular areas of southern Italy (e.g. In winter conditions the climatic zone in Italy range from 600 dd to 4600 dd). Furthermore, during middle-season both heating and cooling loads are significant and they alternate frequently. This is due to the geographic location of the country. For this reason, it is necessary to design flexible systems capable of responding to both cooling and heating peak loads as well as capable of following their high variability during the mid-seasons. Moreover, the extremely variable boundary conditions are the reason why there is not a unique system design solution recurring throughout the country.

In general, the traditional systems are designed to balance either the heating or the cooling load. When a single system for both loads is used, it is generally designed to balance the summer peak load since it is usually higher than the winter one. Indeed, the evolution of building techniques following the guidelines of the Northern European countries has led to the reduction of energy loads during the winter season. Therefore, the air conditioning systems in new and refurbished buildings result usually slightly oversized in winter. For this reason they can work with reduced thermal temperature gradients for balancing the thermal heating loads.

Residential buildings in 2009 amounted to 29,618,828 units of which 11,852,080 single or duplex building, 7,372,677 characterized by 3 to 8 dwellings and 10,394,092 over the 9 dwellings (therefore assimilated to condominium complexes). Approximately 13,300,000 of these buildings are located in the Centre-South and Islands (climate zone W - warmer according to the European classification). It must be borne in mind also that 21,782,741 building units were built before 1981 therefore potentially subject to refurbishment or at least to a renovation of the heating systems.

With regard to the existing heating installations, about 6 million of units have not a fixed installation while about 14 million are equipped with a local system, the remaining about 10 million units have centralized system and are concentrated mainly in Piedmont, Lombardy and Lazio (large condominium buildings). The type of fuel used is gas for 48.9% of them, liquid for 29.4% and solid for 14.5 %. The systems can be categorized according to a first distinction:

Systems controlling only indoor temperature in cooling and heating modes, but not the relative humidity and the indoor air quality;

HVAC systems allowing a complete air conditioning (controlling temperature, relative humidity, IAQ).

b) Systems controlling only indoor temperature

These systems are widely used in residential buildings and sometimes in tertiary buildings. They are generally designed to accomplish a single function – either heating or cooling – and sometimes for controlling the indoor temperature in cooling and heating seasons two systems are combined together (e.g. radiator for heating and DX unit for cooling).

HEATING	Radiators Radiant floor Radiant ceilings Fan heaters
COOLING/HEATING	Fan coil units DX unit split systems

Combustion boiler heat production and district heating

In old buildings boilers were centralised. They caused several problems of consumption metering and of control efficiency. For these reasons, between the eighties and the nineties, small heating capacity individual boilers (less then 30 kW) started to be preferred. These boilers had a low production efficiency and, for safety reasons, they required individual flues which were complex to build. Hence, nowadays some Italian national and regional regulations (e.g. DPR 59-2009 and DGR 46-11968 2009) encourage designers to use centralised heat generators for buildings that require both heating capacity over 100 kW or for buildings with more than 4 dwellings. This is due to the higher efficiency of well designed centralised boilers compared to the individual ones. Moreover, compared to the past the new technologies solved the majority of the problems the centralised boilers used to have (e.g. thermostatic valves, variable flow rate pumps, heat meters and efficient control systems).

Furthermore, in the latest years the heat carrier is produced at lower temperatures. This occurs because of the increasing use of condensation boilers and renewable energy sources, such as solar thermal panels and heat pumps. Moreover considering more stringent regulations on the building envelope energy performance, the terminal units (usually oversized designed) work very well also with lower temperatures.

Between 2000 and 2005, 49% of units housing has changed the heat generator (boiler) and / or radiators and in the same period, 22% have changed the DHW heater and / or air cooling conditioning. The other half of the units housing still are equipped with an old or very old heat combustion generators. Besides, district heating is gradually expanding in Italy, starting from big cities to little towns. The major district heating distribution is in Turin (750.000 users and more than 820 km of pipes) and the second one is in Brescia. On the other hand, district cooling finds only seldom applications (Brescia and some towns in Emilia-Romagna region) and it always covers restricted cooling areas.

In winter for district heating the heat carrier fluid is usually superheated water which reaches different utilities and exchanges power through a local heat exchanger. In the past there were some examples in which the carrier fluid reached directly the room terminals (one is the Politecnico of Turin), but they cause several problems of excessive heating, so they are gradually being abandoned. Winter supply temperature of the district heating carrier is about 110°C-120°C. In summer instead – when the exchanged loads are lower – it is reduced to 90°C in order to have the same return temperature (lower than 60°C) without modifying the efficiency of the CHP. The supply water in the building secondary loop of the heat exchanger is at 55°C and the return water is about 65°C. This temperature difference becomes higher after periods when the circuit is turned off (about 45°C). This could lead to some power peak loads. For this reason, Italian article 9 of the law DPR 412/93 allows users to not respect limits of maximum operation time. However, users usually treat district heating in the same way of traditional systems.

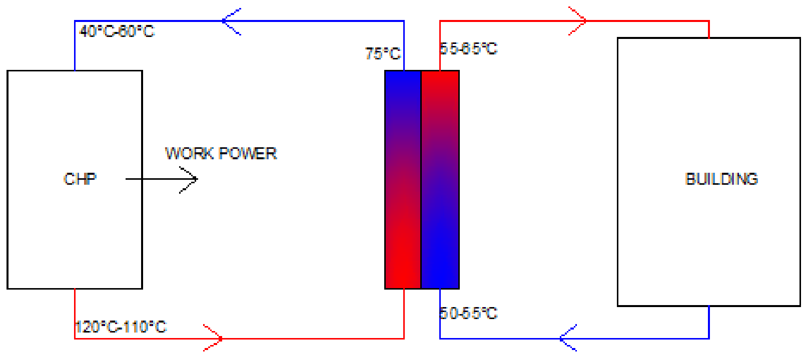


Fig. A-12 DHW typical system with CHP production

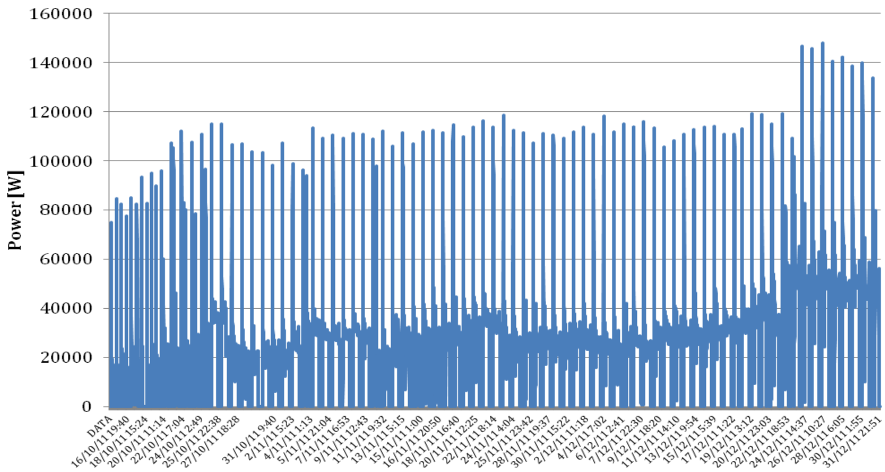


Fig. A-13 Power trend for a typical residential building with DHW in Turin. Data monitored in 2011 from Monetti, Fabrizio, Filippi - Politecnico di Torino.

Heat pumps

In Italy 92% of households with a cooling system has a DX split or multi-split air conditioner (these statistics involve only one or a few zones of each housing unit, then the percentage figure is misleading), 5.2% an hydronic system with fan coils and 2.6% a portable air conditioner. About 95% of these systems use reversible heat pump. The nomenclature of heat pumps distinguishes two basic categories according to the thermodynamic cycle on which they are based: saturated vapor compression or absorption heat pumps.

Within these categories also distinguish whether they are "reversible", i.e. capable of operated alternately in both cooling and heating modes, or when they work only in the heating function. A further classification includes three basic types according to the energy source with low enthalpy used (all renewables): air, any kind of water, ground. For each of them is a further distinction according to the heat transfer fluids: air (for direct or indirect use) and water (hydronic systems).

For absorption heat pumps also exists an important division between those that make use of direct combustion and those who make use of waste heat. For the saturated vapor compression heat pumps exists also the possibility that the cycle is driven by endogenous engines.

Summarizing, the heat pumps may be (considering the heat sources): • AA, form air (ambient or exhaust) to air; • A-W, from air to water; • W-W, from water (for each source) water; • W-W, from water to water; • GSHP, from ground to air (very rarely) or water.

The statistics in the following presented about the spread of heat pumps consider only the systems used to satisfy totally or partially (for the main possible portion) the heating demand of the building. In fact, surveys will include the so-called binary heat pumps, i.e those combined with a traditional heat generator which must however be designed to satisfy only the demand peaks for heating.

In the EHPA (European heat pumps agency) surveys the AA heat pumps are excluded with the exception of those installed in Sweden. Anyway the Italian association manufacturers of equipment HVAC (Anima/CoAer), estimate that more than 10% of air to air heat pumps are actually used for the total heating. Then this meaning that at least 10% should be included in the statistics.

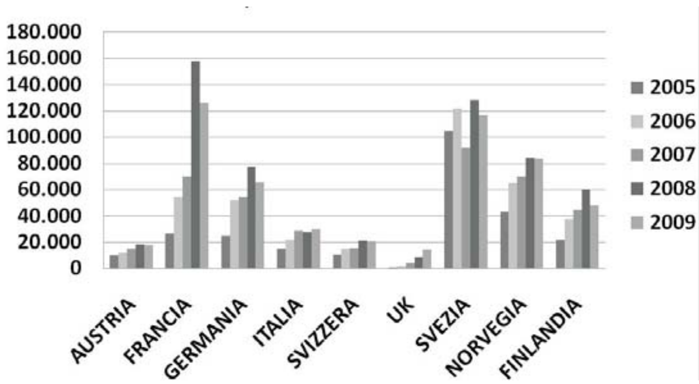


Fig. A-14 EHPA, Heat pumps for country, 2005-2009.

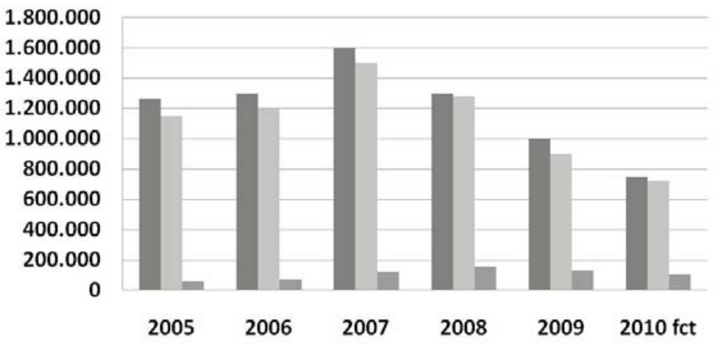


Fig. A-15 Italy number of Air to air heat pumps units (split system and VRF[2]).

(First bar is related to total number of units, second to the units only for cooling, and third to the units used also for heating)

[2] Variable Refrigerant Flow

The installation AA heat pumps is remarkable and, it is always around 92% of installed systems. However only about 10% is to be used as "total" or "binary" heating systems. VRF systems are in great development and have achieved considerable sums around 15,000 condensing units used. Recent surveys affirm of a total stock of air -to-air heat pumps working in Italy of over 1 million units .This number cannot be interpreted as " housing units" heated, but only as a " living spaces " heated . The heat capacity can be considered a weighted average around 3.2 kWth. For Roof-Top heat pumps in the following presented a real total heating can be considered.

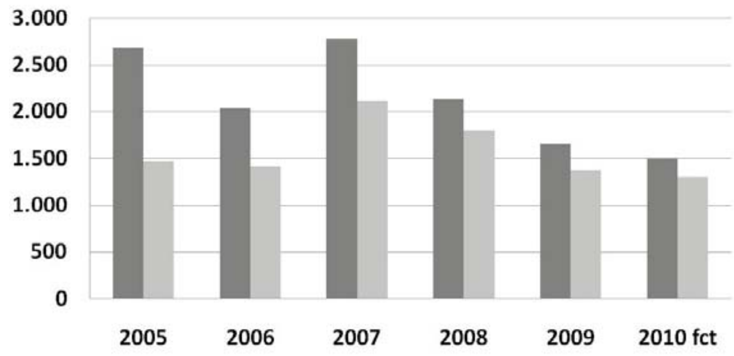


Fig. A-16 Italy number of heat pumps on roof top units. (First bar is related to total number of units, second bar to the units used for cooling and heating)

The following two graphs relate to the Chilling according to PdC suitable especially for hydronic systems. The capacity-weighted average of these groups can be calculated with greater accuracy because the measurements are made for bands capacity - up to 17 kWt nominal, 18 to 50, from 51 to 100, from 101 to 200, 201 to 350, 351 to 500, 501 to 700, 701 to 900 and more than 900 kWth. An average capacity of 75 kWth is acceptable.

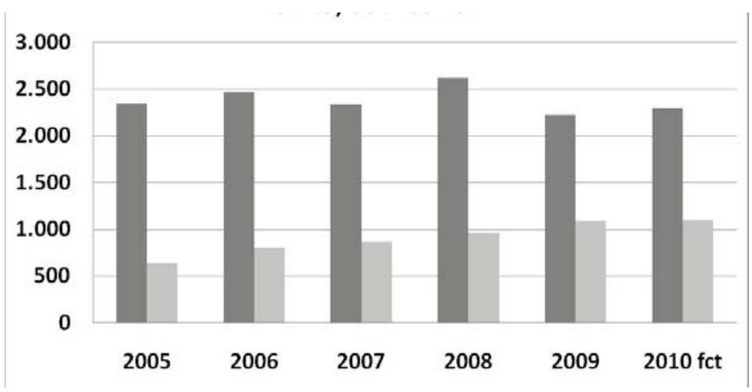


Fig. A-17 Italy number of hydronic heat pumps. (First bar is related to total number of units, second bar to the units used for cooling and heating)

Statistics shown in the graphs you need to add 647 PdC brine-water reversible, ie GSHP, as detected by the Group of COAER Heat Pumps.

Radiators

Radiators are only used for heating. Generally, they are made in aluminum or cast iron.

In the past radiators were the most common heating device in the Italian houses. Nowadays in the new constructions they are gradually disappearing, replaced by radiant panels which take up less space and are considered more efficient. For this reason, in the North of Italy radiators are used just in social housing. Even though radiators are actually less efficient than radiant floors, they have some advantages for the Italian climate: their lower thermal inertia (mainly for aluminum radiator). Allows the trend of the outside temperature to be followed during the strict middle seasons and, when coupled with thermostatic valves, they allow temperature and comfort in individual rooms to be adjusted. In this way can result a high emission and control efficiency.

The difference between supply and return temperature inside the device is about 20°C. The water supply temperature depends on the boiler and on the typology of the radiator: about 85°C/80°C in old systems and about 65°C/60°C in new condensing boilers.

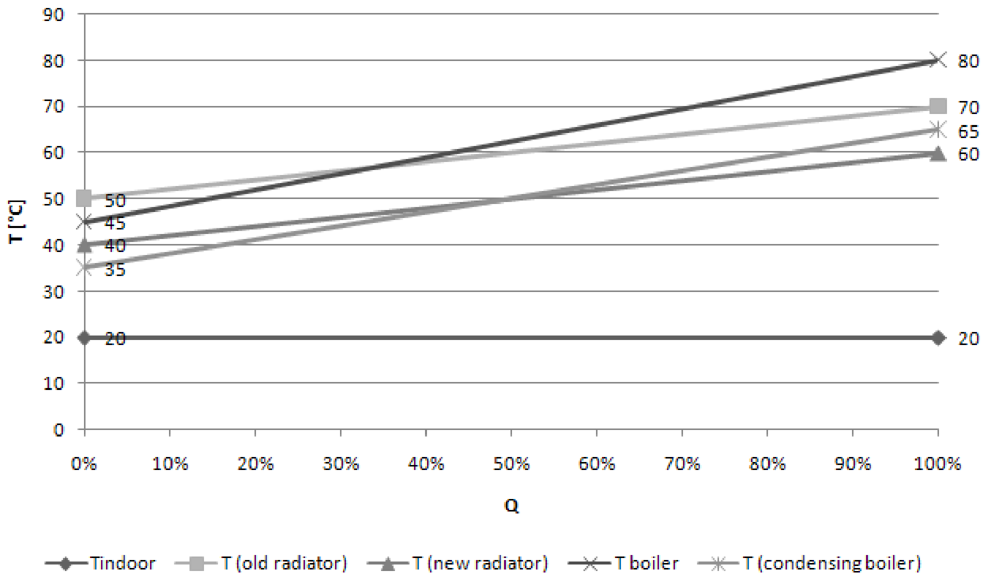


Fig. A-18 Temperature-Power chart for radiators

Radiant floors

In new constructions radiant floors are mainly used for heating and in some cases - combined with primary air - also for cooling (this configuration will be discussed in a following chapter). During the eighties they were made with large pipes of iron or steel embedded in the attic. They were used with high water temperatures (50°C-40°C) and high flow rates. This led to discomfort due to warm floor and radiant asimmetry. Moreover, the thermal inertia was very high and it caused several problems in following the high variability of loads during the middle seasons.

Nowadays the radiant panels are made in polyethylene (rarely in copper) integrated in the floating floor. A lower flow rate and supply carrier temperature are used (from 38°C to 30°C). In this way the thermal inertia is reduced, so that the radiant panels can be efficiently used also at the Italian latitudes. However, in order to avoid stratification phenomena the supply carrier temperature should be less than 45°C. They work very well coupled with low temperature generation systems, such as condensing boilers, heat pumps or thermal solar systems. They have become the most widely used terminal in new residential houses and in some tertiary applications for their greater efficiency and the smaller space they occupy. Radiant panels have two main problems. The first one is that – compared with radiators with thermostatic valves – the control of the terminals in each room is very difficult without the use of expensive electronic two and three valve systems.

Secondly, they require large installation areas to satisfy the loads. In some rooms a fast heating is needed and these areas are not available (e.g. bathrooms), so it is necessary to replace panels with radiators. This fact leads to two thermal levels of the heating carrier fluid (which become three levels considering also DHW). A central adjustment group for every housing unit is needed to mix water and obtain different temperatures.

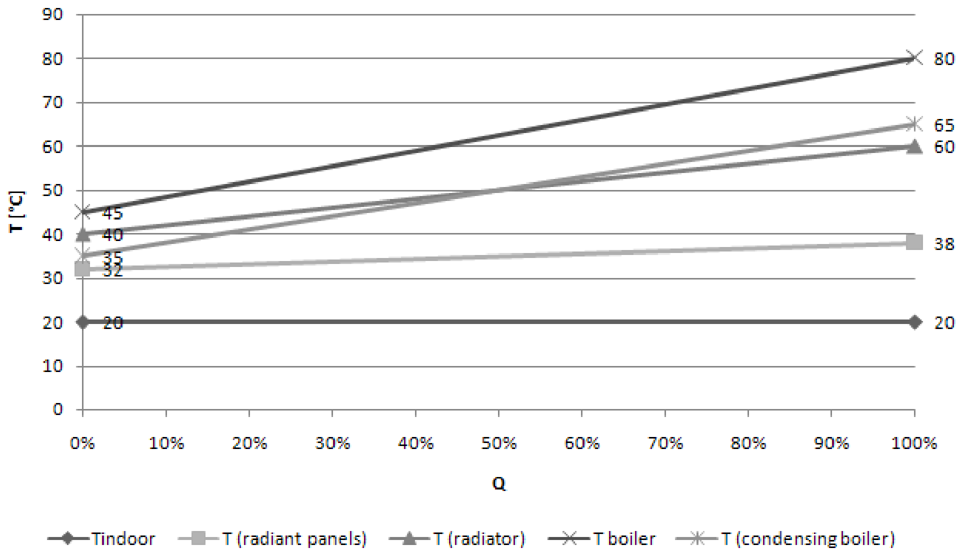


Fig. A-19 Temperature-Power chart for radiant floors

Radiant ceilings

Construction characteristics of radiant ceilings are usually of two types: ceilings with metal panels characterized by copper pipes or thermal diffusers and ceilings gypsum board panels characterized by low diameter plastic pipes. Radiant ceilings are used rarely for heating purpose, due to the fact that they are a new technology and it is not very common to find labor able to assemble them. Moreover a less emission efficiency is usually obtained comparing to the radiant floors. They are appearing more and more frequently for cooling applications, but in this case they need to be coupled with a primary air system which allows the indoor humidity control to prevent surface condensation. When they are used for heating, radiant ceilings work in a similar way to the radiant floors, but they generally work with a lower temperature difference between supply and return (from 38°C to 35°C).

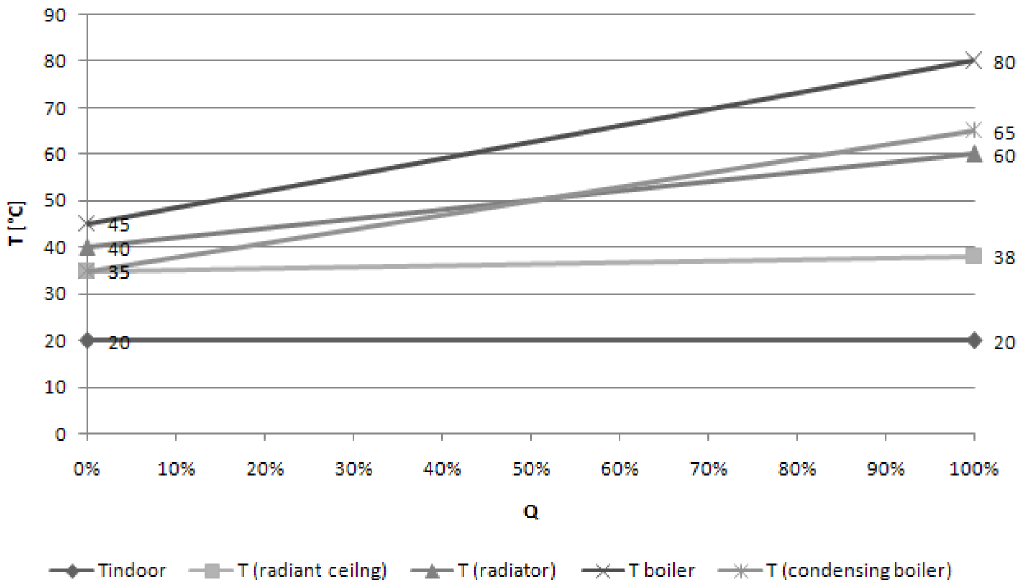


Fig. A-20 Temperature-Power chart for radiant ceilings

Fan heaters

In the past fan heaters were widely used in factories and workshops where big heated rooms were common. They are now partially being replaced by radiant panels. They work only as heating terminals, but during the warm season they are used as simple fans to move the air and increase thermal comfort.

They are often preferred to primary air/fan coil systems - which allow also the air cooling – for their higher thermal power, due to the greater working temperature difference (from 75° to 55° or 85°C to 65°C compared to fan coil with temperature from 50°C to 40°C). For this reason, smaller devices may produce the same amount of heat of a bigger fan coil (e.g. a typical size for a fan heater is 20-50 kW thermal power and a typical size for a fan coil unit is 5-6 kW).

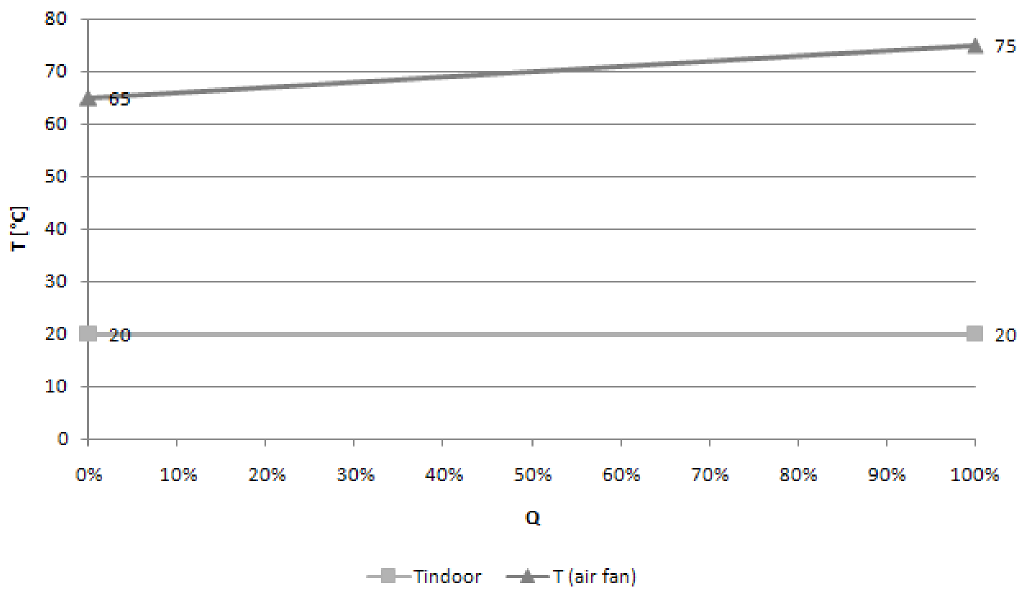


Fig. A-21 Temperature-Power chart for fan heaters

Fan coil units

Fan coil units are widely used in offices and service buildings as heating and cooling units, due to their low cost and easy design. When they are not coupled with primary air (this mixed system is described in a following chapter) they are mainly used for summer cooling and only seldom for winter heating. It is a system that usually create a low health indoor quality (problems with dust in the filters and bacteria in condensed water), low heat diffusion efficiency and high levels of noise. Their major advantage is their low thermal inertia which allows to quickly condition rooms and to follow the change of the outdoor boundary conditions combining heating and cooling functions.

For cooling applications they generally cool the indoor air from 25°C-26°C to 14°C-16°C. The cooling battery is supplied by chilled water with working temperature difference of 5°C (from 12°C to 7°C). Nowadays design guidelines suggest to use a working temperature difference of 7°C (from 14°C to 7°C) in order to reduce the water flow rate and the pumping power. Moreover they suggest to slightly increase the supply water temperature also to avoid any condensation occurrence and indoor humidity modifications. However this solution is rarely adopted in Italy, because it should also oblige to increase the dimension of the device.

The cooling water is produced through a chiller, where the temperature of the refrigerating fluid is about 2°C-5°C in the evaporator and about 35°C in the condenser.

For heating applications fan coil units work with supply water temperature at about 50°C and return water at 40°C. This high temperature difference is required to contain the dimension of the system. In Italy, in places where the climate is very cold and the buildings are rarely occupied (e.g. a ski resort) a low flow rate of heat carrier is maintained during the whole heating season (with the fan switched off) to avoid freezing risks.

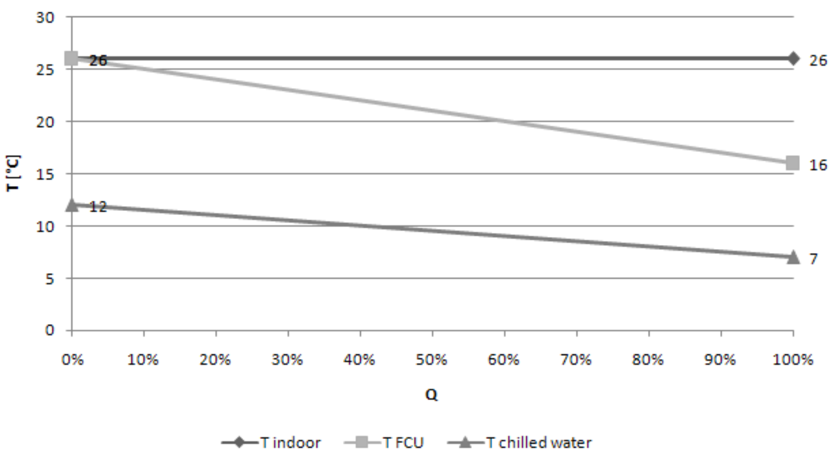


Fig. A-22 Temperature-Power chart for FCUs – cooling mode.

A single heat pump may produce both heat and cool, otherwise a separate boiler and refrigerating unit may be used. The battery of the heat exchanger of the fan coil can be of two different types:

2 Pipes: a unique battery is used both for heating and cooling. No cooling possibilities during heating period and vice versa;

4 Pipes: a battery is dedicated to the heating and another battery is dedicated to the

cooling. They can work alternatively in every period of the year.

DX unit split system

DX unit split system is a direct expansion system that in Italy commonly appears in shops and offices. Lately it is becoming frequent also in residential buildings. DX unit split systems do not require a complex system and they are quite easy to install. Their heat pumps allow to do both heating and cooling. Usually, though, especially in residential buildings, they are used just for cooling so they are coupled with other traditional heating systems.

A split system is composed by pipes with refrigerating fluid that pass through an outdoor compressor and an indoor evaporator (called split). During heating periods this process can be reversed.

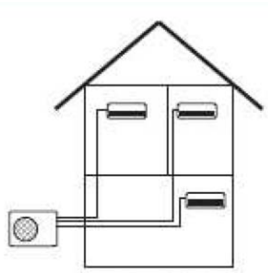


Fig. A-23 Traditional DX units split system (AA)

c) HVAC systems

HVAC systems allow a total air conditioning (control of temperature, relative humidity and IAQ). In Italy, there are typically two types: Mixed systems (90% of the market); all air systems (10% of the market).

Primary air in mixed systems

Mixed systems combine a central air handling unit with one of the previous described system placed in a single room. The AHU has the task of controlling relative humidity and indoor air quality, while the hydronic local room system balances the sensible heat loads (cooling or heating).

During winter, the primary air (external ventilation air) is supplied through diffusers in the ambient. This primary air has a neutral or slightly colder temperature respect to indoor temperature (the indoor comfort temperature is 20°C, the inlet temperature is in the range 20°C-18°C). Even though this fact causes a slight sensible heat load to balance, it permits a major flexibility, a better distribution of the temperature in rooms and an efficient treatment of the relative humidity in the AHU.

The typical temperature difference for heat carrier fluid in the heating coil of the AHU is from 50°C of the supply water to 45°C of the return water.

The cooling water is produced through a heat pump, where the temperature of the refrigerant fluid is about 2°C-5°C in the evaporator and about 50-55°C in the condenser.

During summer, the primary air covers also a little part of the sensible load. For this reason the primary air is supplied in the rooms with a temperature difference of 8-10°C between the indoor comfort temperature at 26°C and the inlet temperature at about 18°C-

16°C. The typical temperature difference in the cooling coil of the AHU is from 7°C of the supply water to 12°C of the return water. The cooling water is produced through a chiller, where the temperature of the refrigerating fluid is about 2°C-5°C in the evaporator and about 35°C in the condenser.

The middle seasons, which generally are characterized by outside temperature ranging between 16°C and 30°C, are the most hardly periods to accurately control the indoor microclimatic conditions.

Indeed, the uncertainty due to the changing loads does not allow the use of a constant inlet temperature. Therefore, it is necessary to control the inlet air temperature through models as a function of the outside temperature, as it is shown in the following chart (certain refined systems can modulate the air temperature also on the basis of the solar irradiance).

Moreover in the middle season a chance to reduce the energy consumption through a free cooling exist.

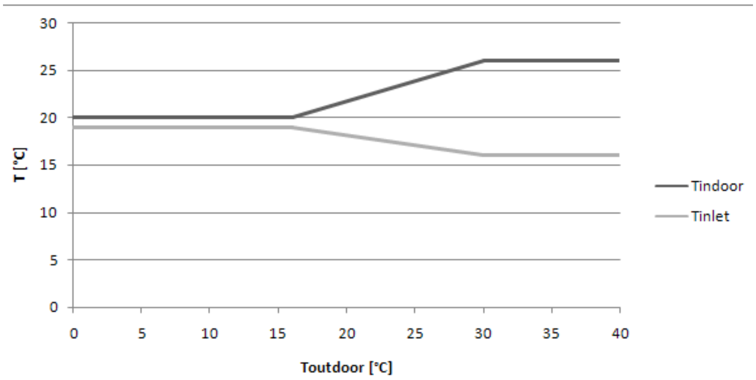


Fig. A-24 Trend of the inlet and the comfort temperature related to the outside temperature

In the various rooms the sensible heat load is balanced by local terminal units which allow together with the primary air to complete the air conditioning process. They may be the following devices:

LOW TEMPERATURE DIFFERENCE	Radiant floors Radiant ceilings Chilled beams
HIGH TEMPERATURE DIFFERENCE	Fan Coil Units

Low temperature differences in the first three systems are needed more to avoid the surface condensation than for efficiency reasons. All of these systems require a heat pump or a combined boiler/refrigerator unit to work.

Radiant floors in mixed systems

During the heating season the radiant floors coupled with the primary air work in the same way as previously described. The only difference is the dimension of the radiant surfaces (the temperature difference is always 8°C). The control of relative humidity allows to use radiant floors also in cooling periods avoiding surface condensation. The supply water temperature in the radiant panels for cooling is commonly 20°C. The temperature difference (water supply-water return) inside the panels is 2°C-3°C, which means a low cooling power exchange (no more than 20-25 W/m2). In Italy the use of radiant floors in cooling applications is very limited and usually radiant ceiling panels are preferred.

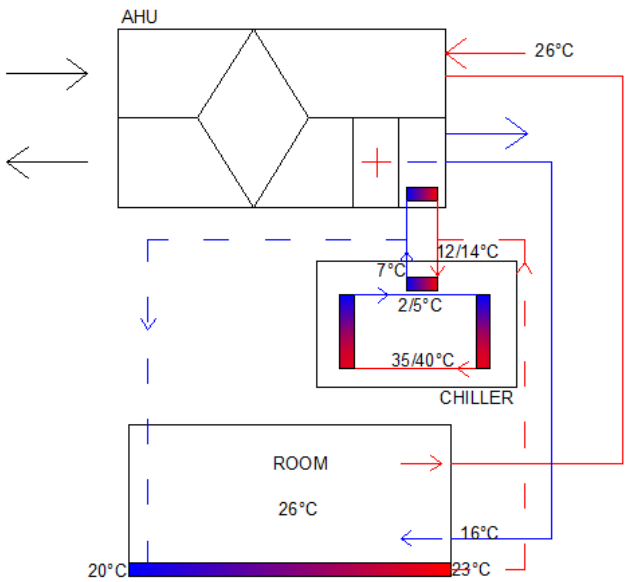


Fig. A-25 Radiant floor in a mixed system during cooling season

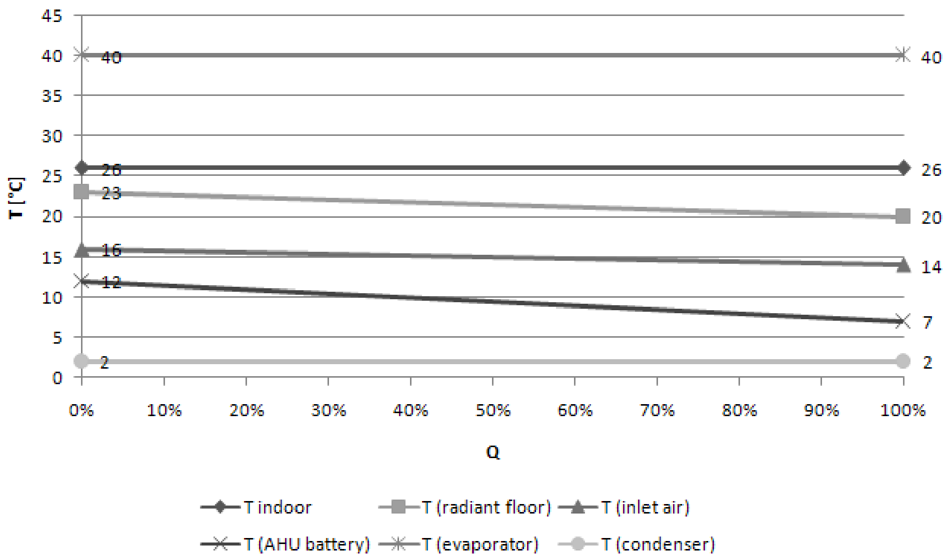


Fig. A-26 Temperature-Power chart for radiant floors during heating season

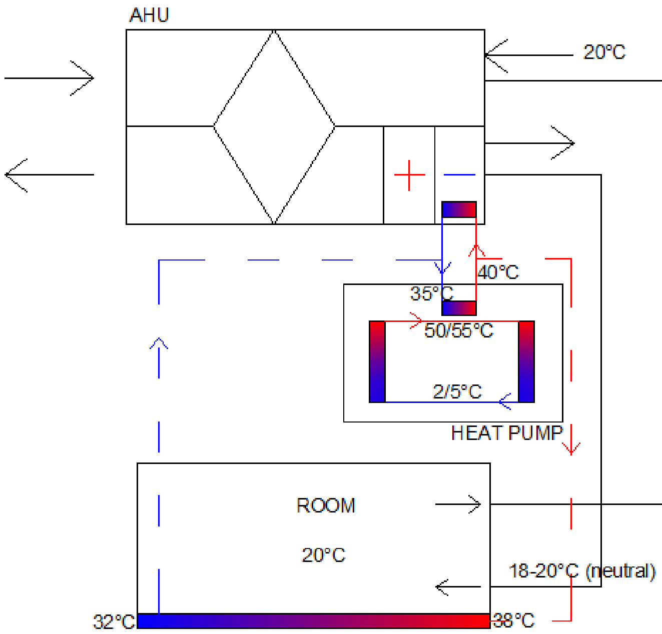


Fig. A-27 Radiant floor in a mixed system during heating season

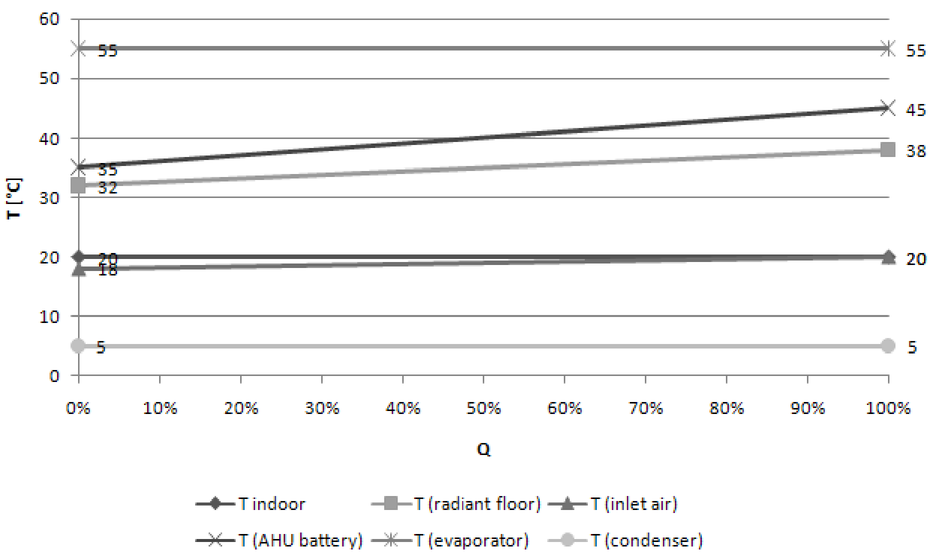


Fig. A-28 Temperature-Power chart for radiant floors during heating season

Radiant ceilings in mixed systems

In the last few years radiant ceilings have been widely used in Italy combined with primary air treatments, especially in office buildings and healthcare structures. During heating season the functioning is exactly the same as without the coupling with primary air.

The supply temperature of the carrier fluid in cooling period is 17°C and the return temperature is 20°C or 19°C (it depends on the dimension of the pipes, which influences the flow rate). Typically, the heat power emission provided to the space is about 70-80 W/m2. It is important to account for the correct exchanging surface. Where located lights and primary air diffusers there are not radiant panels.

There are also versions of radiant ceilings with suspended panels that works with efficiency of 120-130 W/m2 (e.g.: Intesa-San Paolo skyscraper in Turin). This is due to the fact they exchange heat also towards above.

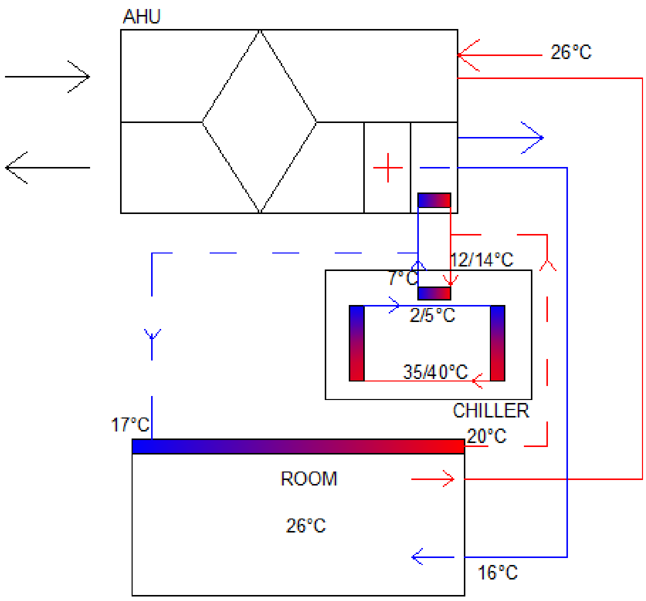


Fig. A-29 Radiant ceiling in a mixed system during cooling season

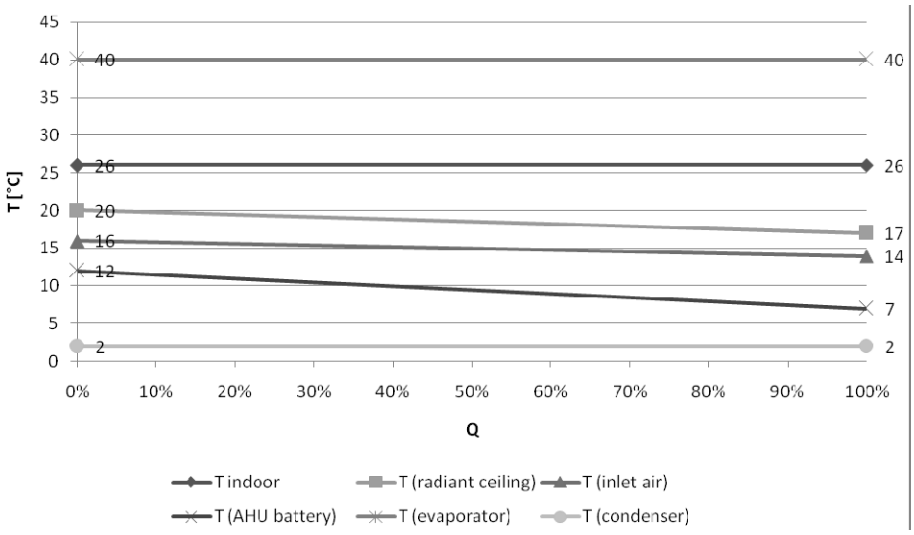


Fig. A-30 Temperature-Power chart for radiant ceilings during cooling season

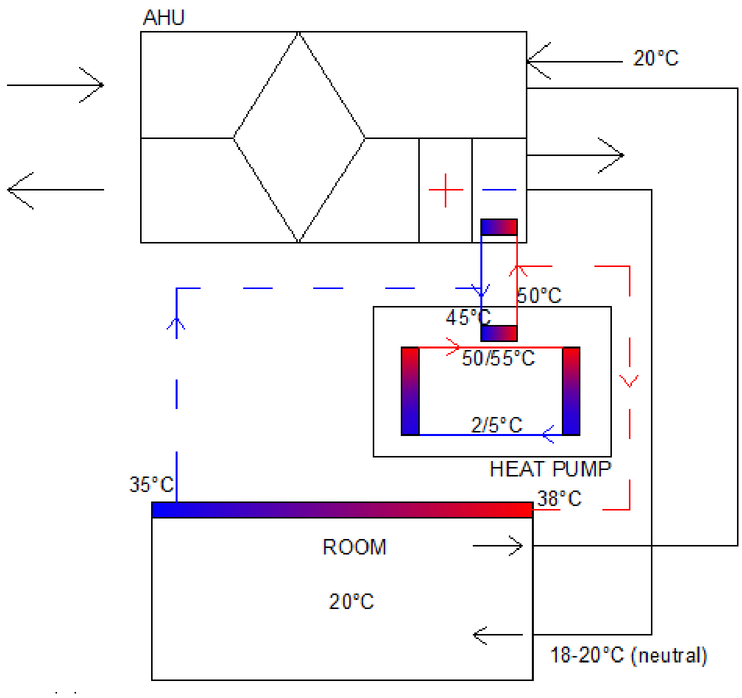


Fig.2-31 Radiant ceiling in a mixed system during heating season

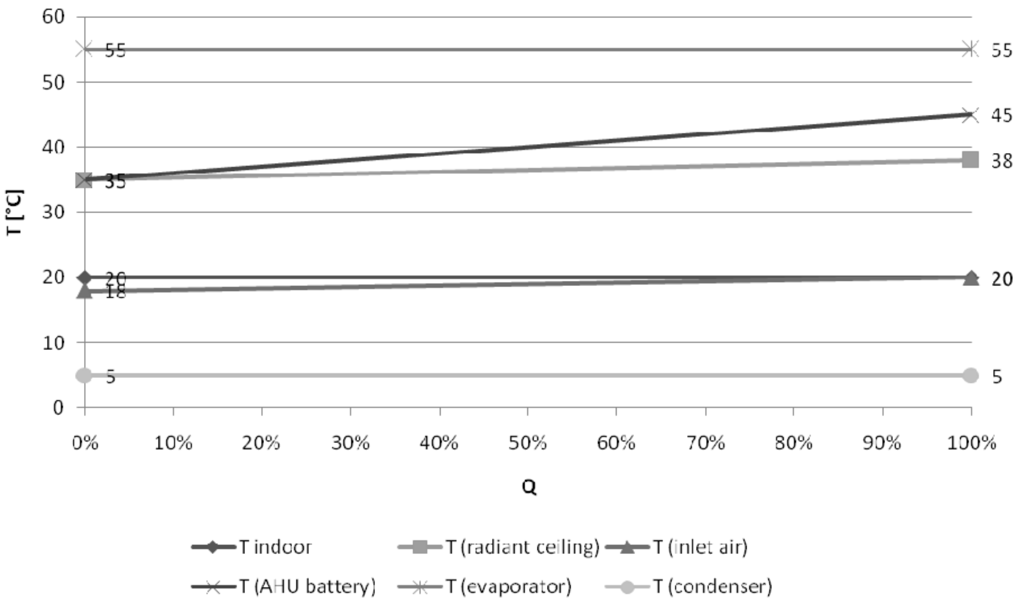


Fig. A-32 Temperature-Power chart for radiant ceilings during heating season

Chilled beams in mixed systems

A chilled beam is a type of HVAC system designed to heat or cool buildings. Pipes of water are passed through a "beam" (the heat exchanger) either integrated into standard suspended ceiling systems or suspended a short distance from the ceiling of a room. As the beam chills the air around it, the air becomes denser and falls to the floor. It is replaced by warmer air moving up from below, causing a constant flow of convection and cooling the room. There are two types of chilled beams: passive types and the active type (also called an "induction diffuser") with a dedicated fan in the beam.

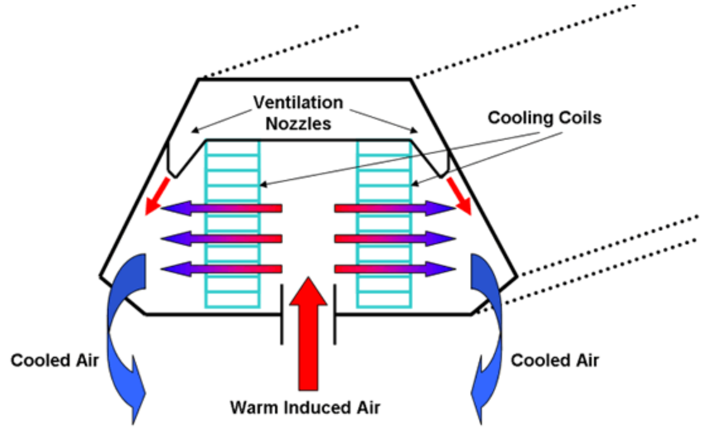


Fig. A-33 Typical chilled beam

In Italy this system is as diffused as radiant ceilings. Chilled beams work with temperatures comparable with radiant ceilings except for the fact that in heating periods they require a slightly more extended temperature difference (from 38°C to 32°C). Some studies have shown that chilled beams work better with lower temperature than radiant ceilings.

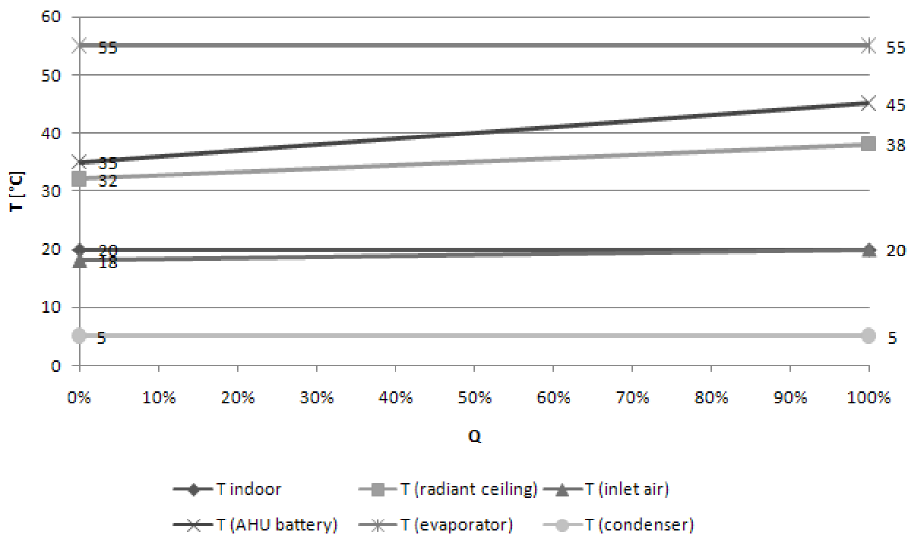


Fig. A-34 Temperature-Power chart for chilled beams during heating season

Fan coil units in mixed systems

The functioning of fan coil units in mixed system is exactly the same of the standalone fan coils. Further attention should be kept to the temperature of the exchange battery, to avoid the treatment of the relative humidity (which, in this case, is already controlled by the primary air coming from the AHU). Unlike the fan coils without primary air, this mixed solution is also used during heating season without causing excessive problems of low relative humidity and bad temperature distribution.

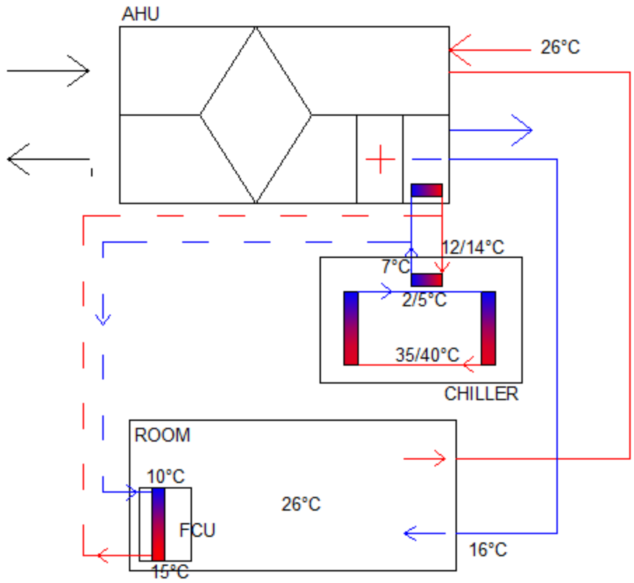


Fig. A-35 FCU in a mixed system during cooling season

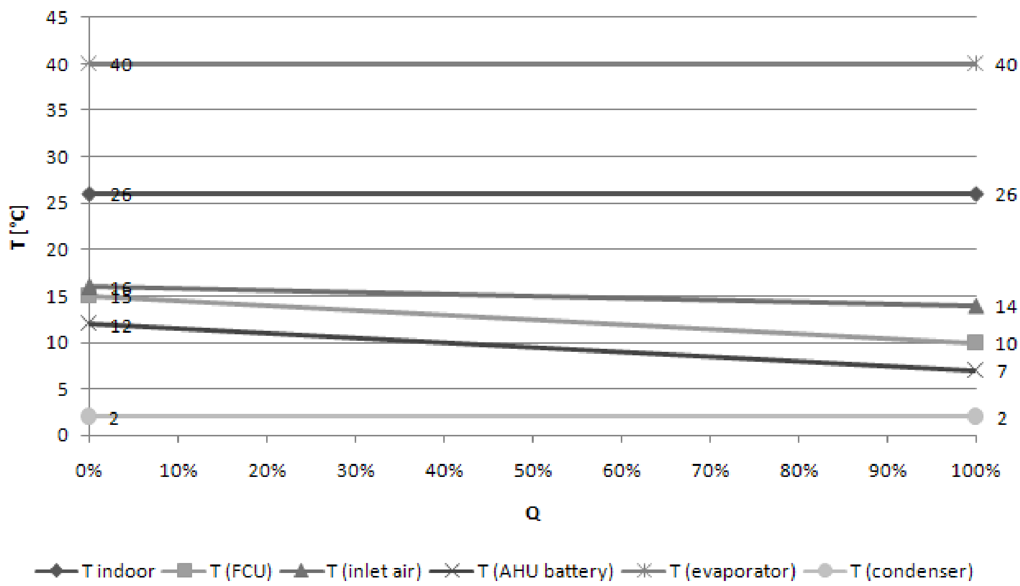


Fig. A-36 Temperature-Power chart for FCUs during cooling season

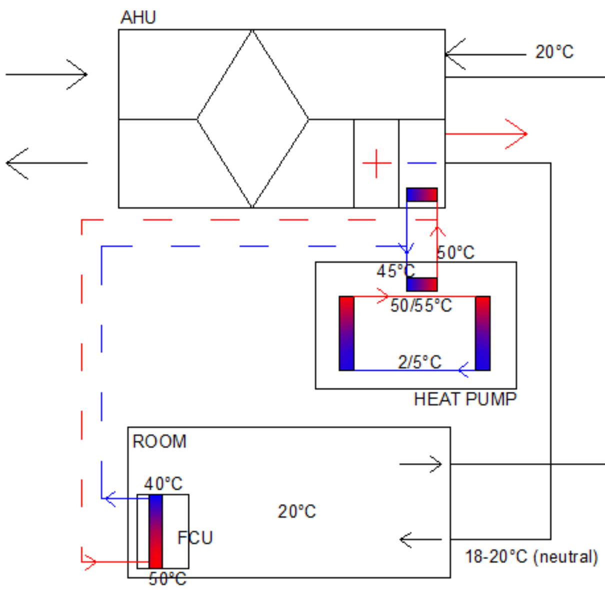


Fig. A-37 FCU in a mixed system during heating season

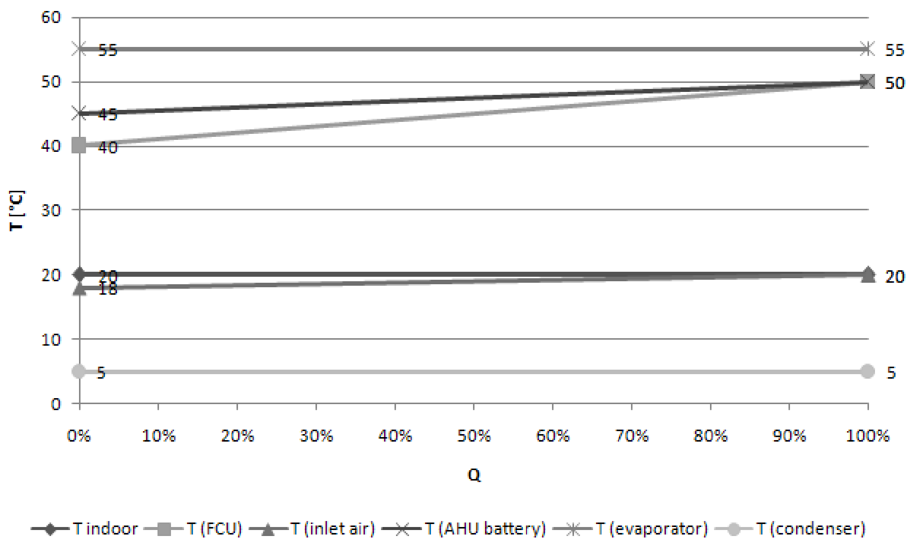


Fig. A-38 Temperature-Power chart for FCUs during heating season

All air system

During the nineties in Italy, VAV system were seldom used in healthcare structures. Since then their use has gradually disappeared from the market.

On the other hand CAV systems are still used but in rare applications. There are two design modes: classic for single utilities (e.g. a conference hall) or combined with post-cooling or post-heating systems for utilities with different thermal levels or different loads (e.g. some special departments of hospitals). Dual conduits are excessively expensive and have been abandoned for many years.

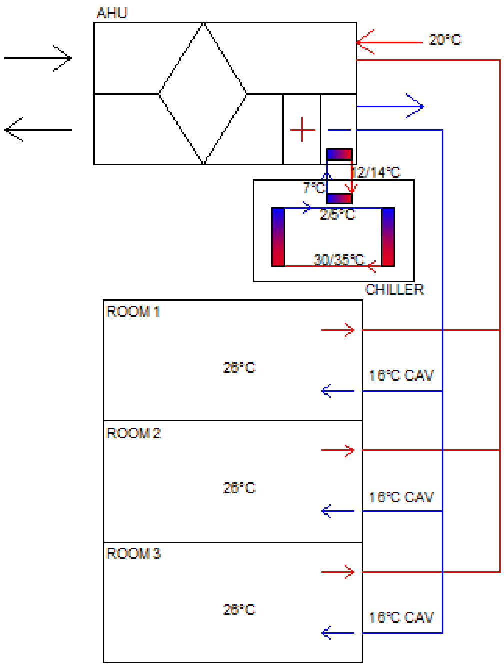


Fig. A-39 Traditional CAV system.

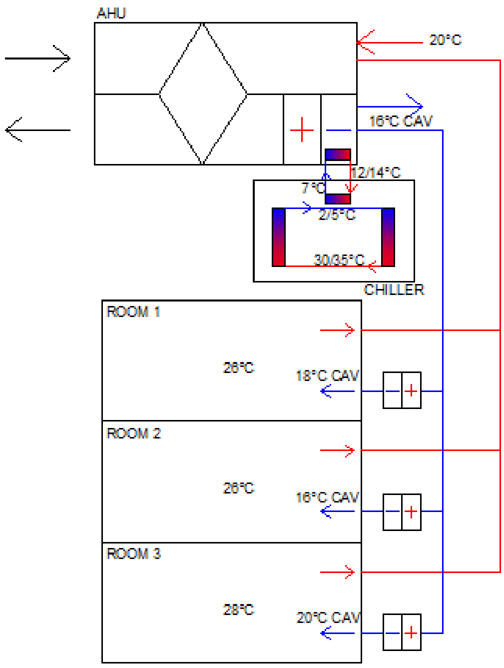


Fig. A-40 CAV system with post-heating. Rooms 1 and 2 have the same indoor temperature but different loads. Room 2 and 3 have different indoor temperatures.

During cooling season a typical Italian CAV system uses the temperature difference schematically shown in the following chart:

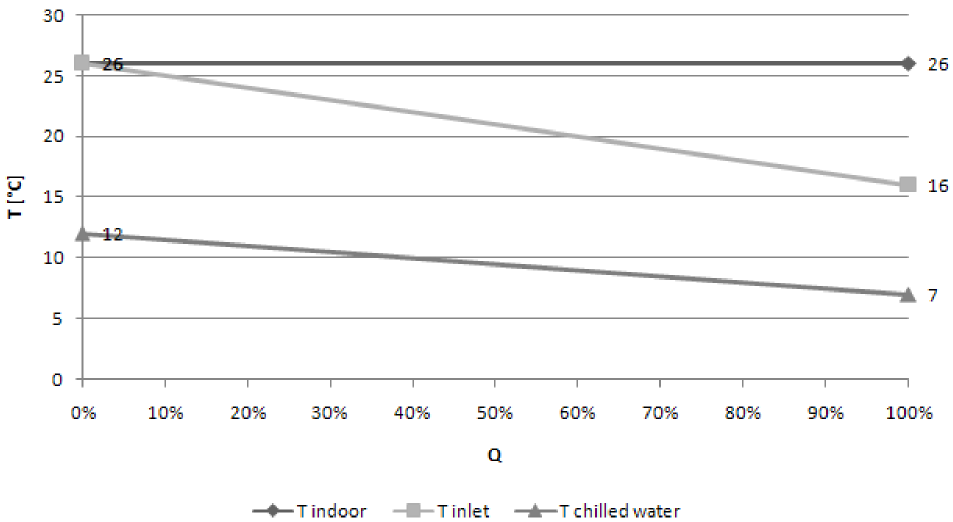


Fig. A-41 Temperature-Power chart for all-air systems during cooling season

Usually, temperature differences between refrigerant fluid and water carrier fluid of 5°C are used in the evaporator. However, CAV systems are not designed on this temperature difference, but on their efficiency and the dimension of their coils. When the maximum load occurs, the temperature difference between inlet air and room comfort temperature is about 10°C. This means supplying air at 16-14°C. It is not recommended to supply air at a lower temperature due to problems of temperature diffusion and potential local condensation.

A.3 Korea

<Cooling>

FCU + All air system

Fan coil unit (FCU) with all air system (VAV (variable air volume) or CAV (constant air volume)) is the most common air conditioning system in Korea. Fig. A-42 shows a diagram of FCU and OA handling system and Fig. A-43 shows the *T-Q* figures for the FCU and All air system.

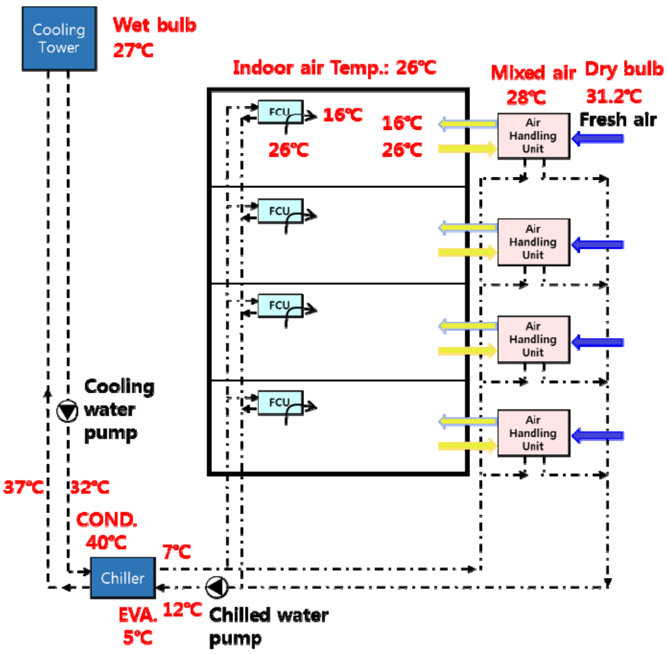
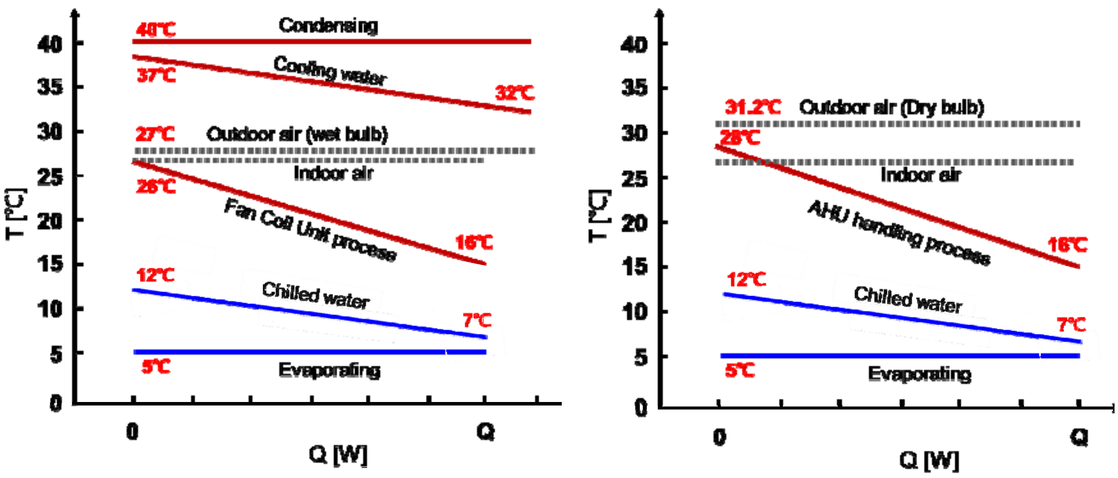


Fig. A-42 Diagram of FCU and all air system (Cooling)



(a) (b)

Fig. A-43 Temperature difference (T-Q figure) of the FCU and all air system (Cooling): (a) the FCU system and cooling tower; and (b) the All air system.

Packaged air conditioner (Heat Pump (Cooling mode))

In Korea, many of households with a cooling system has a DX split or multi-split air conditioner (in many cases, however, just one or two rooms are served by the DX split system.). In addition, many of offices take this system using reversible heat pump to perform heating and cooling. Almost of all heat pumps applied in Korea use electric energy and utilize air as heat source. Fig. A-44 shows a diagram and Fig. A-45 gives the *T-Q* figures of a packaged air conditioner (heat pump (cooling mode)) system.

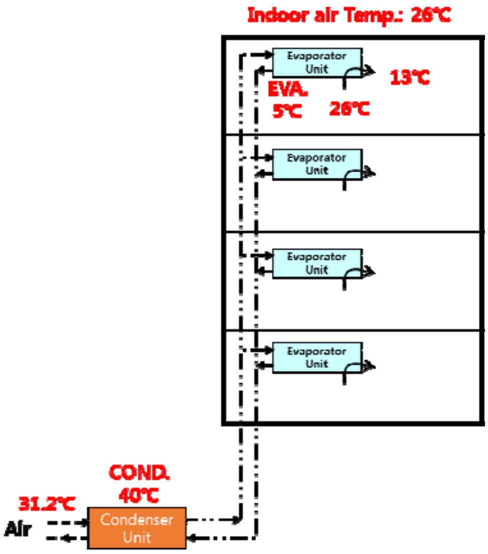


Fig. A-44 Diagram of the packaged air conditioner (heat pump (cooling mode)).

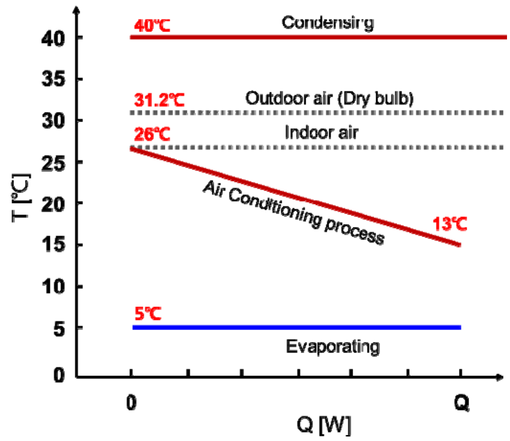


Fig. A-45 Temperature difference (T-Q figure) of the packaged air conditioner (heat pump (cooling mode)).

<u>All air system</u>

VAV (variable air volume) or CAV (constant air volume) system is also a common type of centralized air- conditioning system in Korea. Fig. A-46 shows a diagram and Fig. A-47 gives the *T-Q* figures of an air handling process.

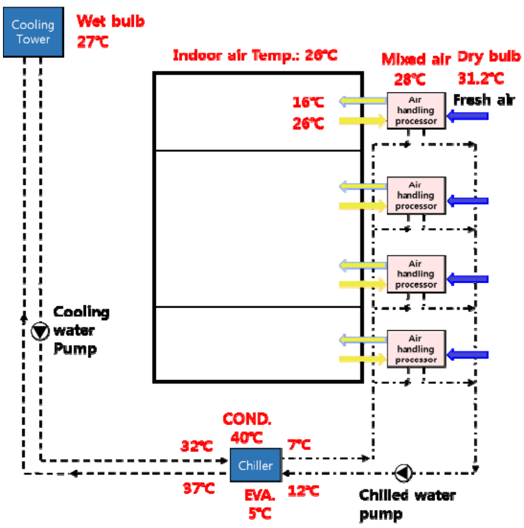


Fig. A-46 Diagram of the all air system.

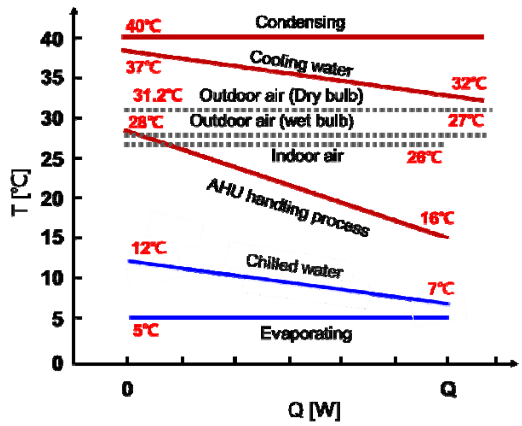


Fig. A-47 Temperature difference (T-Q figure) of the all air system.

FCU+OA system

Fan coil unit (FCU) and outdoor air (OA) system is one of air conditioning system in Korea. Fig. A-48 shows a diagram of FCU and OA handling system and Fig. A-49 shows the *T-Q* figures for the FCU and OA handling processes.

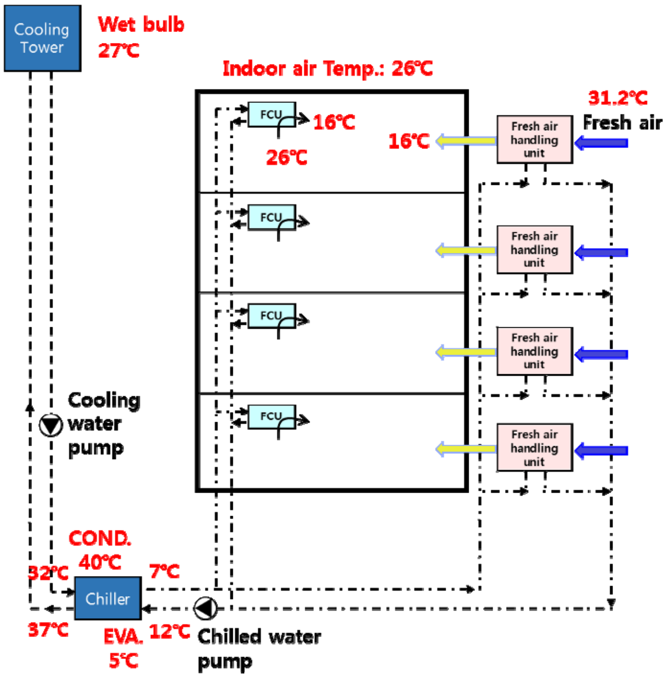


Fig. A-48 Diagram of FCU and OA handling system (Cooling)

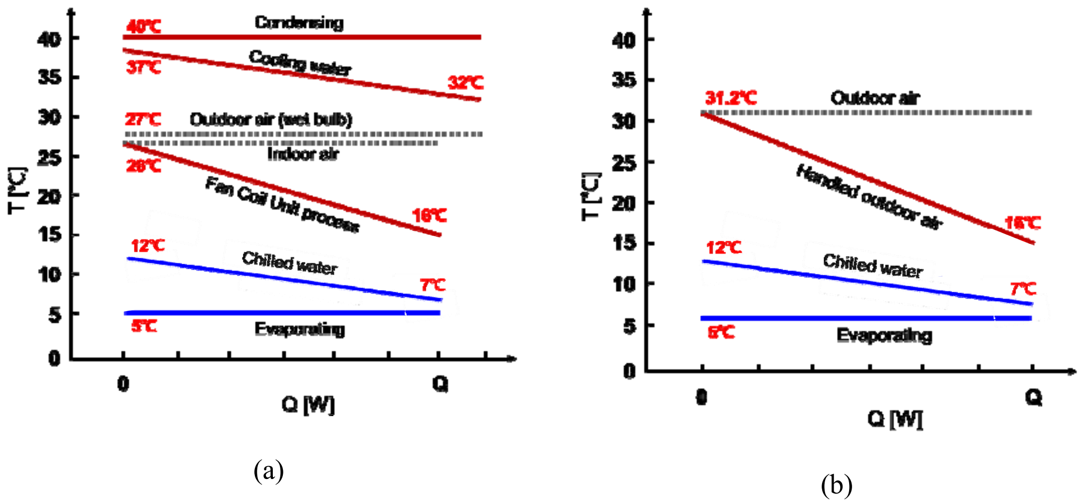


Fig. A-49 Temperature difference (T-Q figure) of the FCU and OA handling system (Cooling): (a) the FCU system and cooling tower; and (b) the outdoor air handling system.

<Heating>

FCU+OA system

Fan coil unit (FCU) with outdoor air (OA) system is the most common air conditioning system in Korea. The FCU remove the heating load of perimeter zone and OA system remove the cooling load and ventilation load of interior zone. Fig. A-50 shows a diagram and Fig. A-

51 shows the *T-Q* figures for the FCU and OA system.

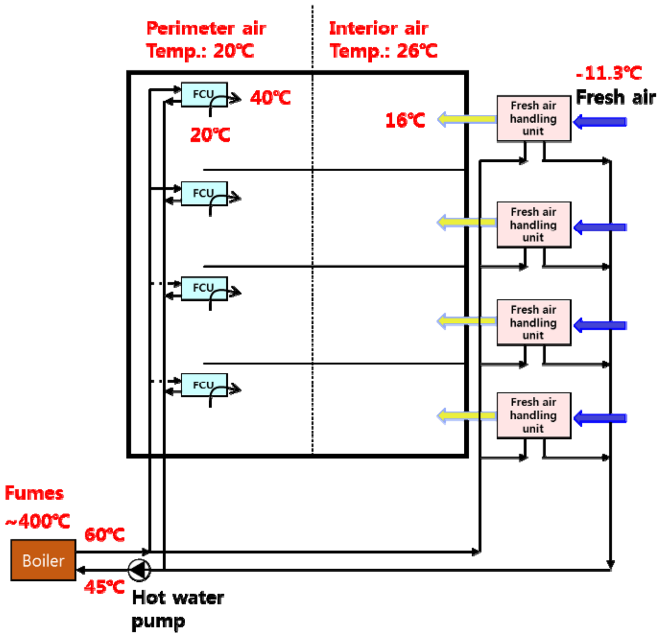


Fig. A-50 Diagram of FCU (Heating) and OA system (Cooling).

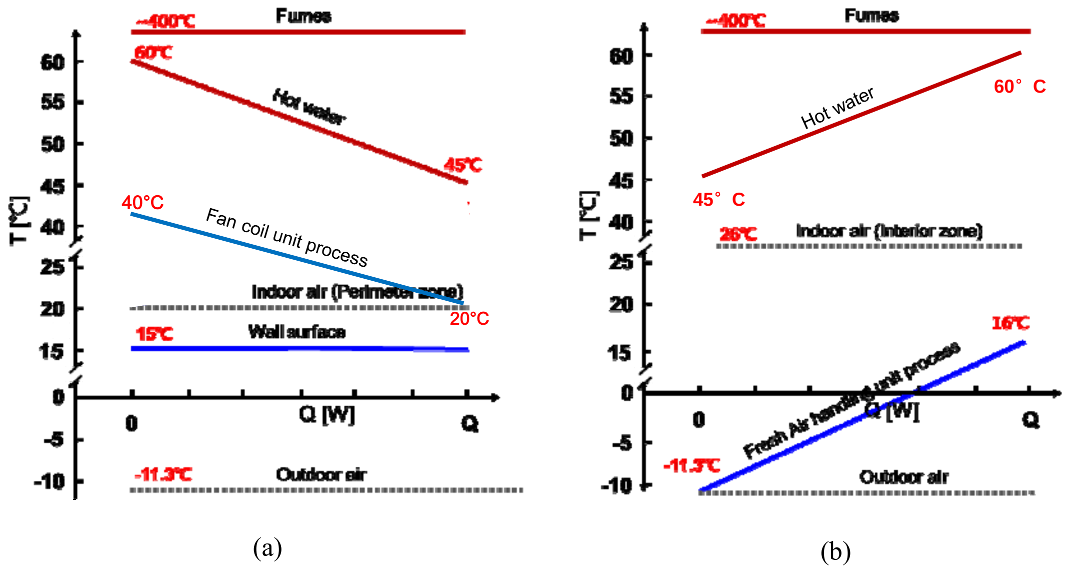


Fig. A-51 Temperature difference (T-Q figure) of (a) the FCU (Heating) and (b) the OA system (Cooling).

<u>Heat pump system (Heating mode)</u>

In Korea, many of offices take this system using reversible heat pump to perform heating and cooling. Almost of all heat pumps applied in Korea use electric energy and utilize air as heat source. Fig. A-52 shows a diagram and Fig. A-53 gives the *T-Q* figures of an air source heat pump systems with DX terminal.

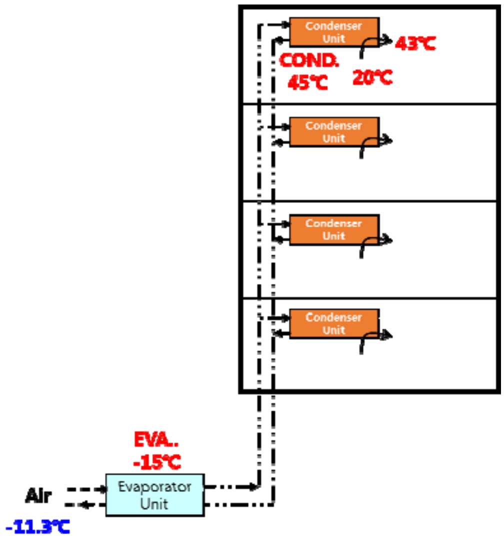


Fig. A-52 Diagram of air source heat pump systems with DX terminal (Heating mode)

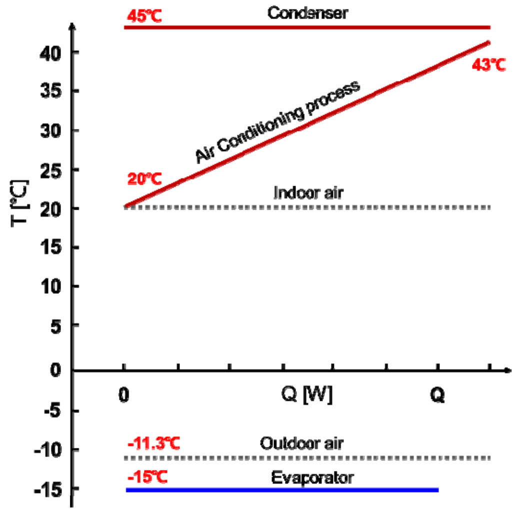


Fig. A-53 Temperature difference (T-Q figure) of the air source heat pump systems with DX terminal (Heating mode)

<u>Boiler + Radiant floor system</u>

Boiler + Radiant system in Korea burn usually gas and oil. These are usually units operating in households. Fig. A-54 shows a diagram and Fig. A-55 gives the T-Q figures of the boiler + Radiant floor system.

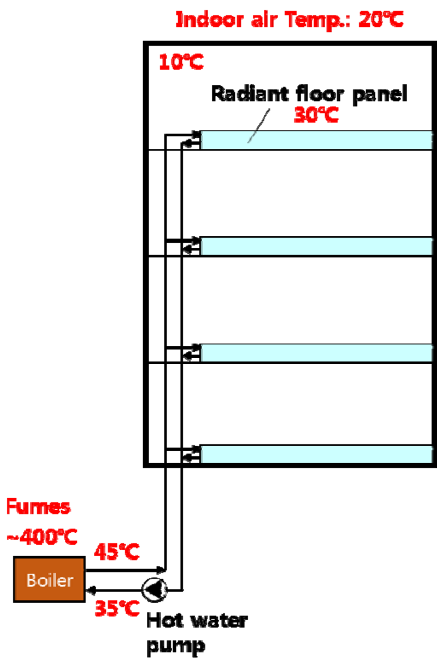


Fig. A-54 Diagram of boiler system and radiant floor system.

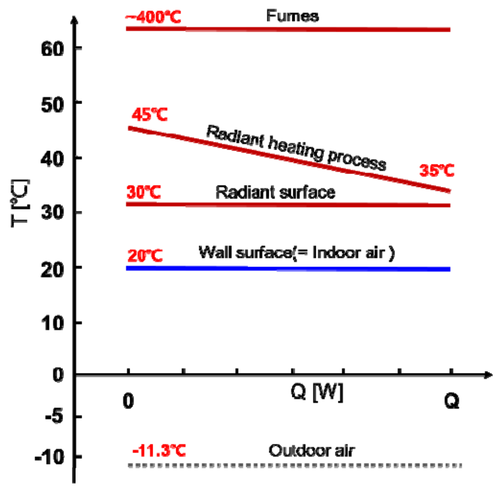


Fig. A-55 Temperature difference (T-Q figure) of the Boiler system.

A.4 Japan

1. AHU+FCU system

Air handling unit (AHU, handling outdoor fresh air) and Fan coil unit (FCU) system has been common air conditioning system in Japan, which is the same as FCU+OA system in China discussed in Appendix A.1. Especially when the depth of the room is long this system is often adopted. Fig. A-56 gives its system principle. AHU mainly treats internal heat gain (generated from human body, lighting and equipment) and fresh air load. FCU treats the heat gain or heat loss thorough building envelop. AHU supply cold air all year around and FCU supply cold air in summer and hot air in winter. Therefore it is important to determine set point temperature for interior and perimeter zone to avoid mixing loss between cold air from AHU and hot air from FCU.

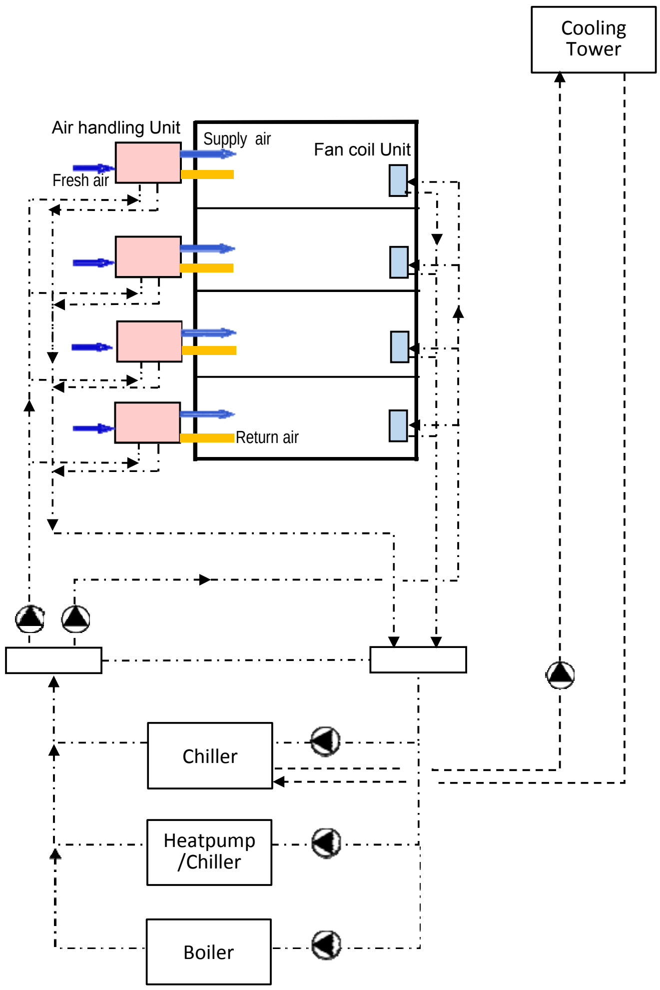


Fig. A-56 Schematic of AHU+FCU system

2. AHU system (All air system : Perimeter-less system)

Recently thermal performance of building envelop is improved by using high performance window system, insulation and solar shading. It results in reduction of heat load through building envelop and sometimes it isn’t necessary to install FCU. Usually FCU can contain little number of coil, it is slightly difficult to maintain large temperature difference through the coil. Therefore if not having to install FCU the performance of the system can be improved.

Air flow window is the one of the perimeter-less system. The air flow window is comprised of a window shade to install between two pieces of glass. The room air passes between two pieces of glass and is drawn out by the upper part. It prevents heat load through the window to enter the room. Fig. A-57 shows the schematic of air flow window.

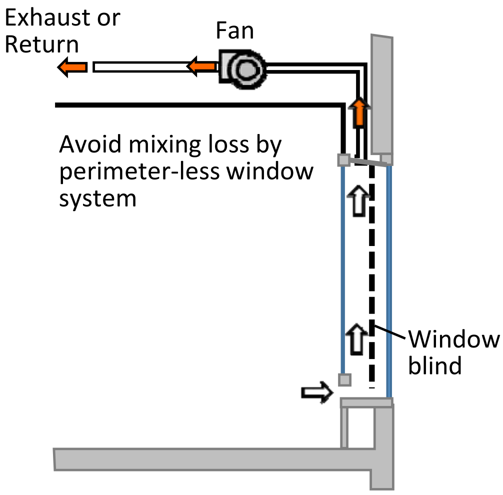


Fig. A-57 Schematic of air flow window.

3. Underfloor air conditioning system

The under floor air conditioning system uses underfloor chamber of free access floor. Conditioned air is supplied through the diffuser which is mounted on the floor and return through the inlet of the ceiling. Air conditioning for occupied zone which is approximately 1.7m above the floor can be realized by under floor air conditioning system. Fig. A-58 shows the outline of under floor air conditioning system.

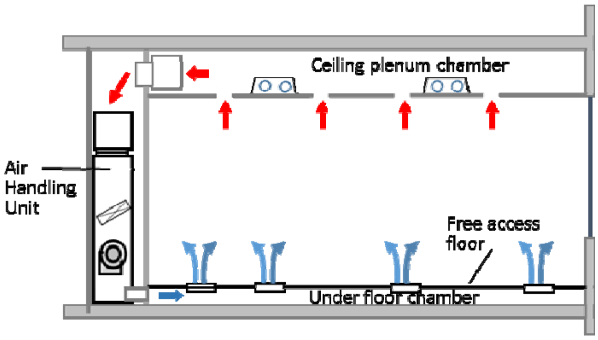


Fig. A-58 Outline of under floor air conditioning system.

4. Multi-split type air conditioner

Multi-split type air conditioner (i.e. VRF) was spreading rapidly for these ten years. It is combined with outdoor air processor. It is popular that outdoor air handling unit and total heat exchanger are used as outdoor air processor. When desiccant air handling unit is used as outdoor air processor, evaporative temperature of refrigerant of multi-split sir conditioner can be raised to treat only sensible heat load.

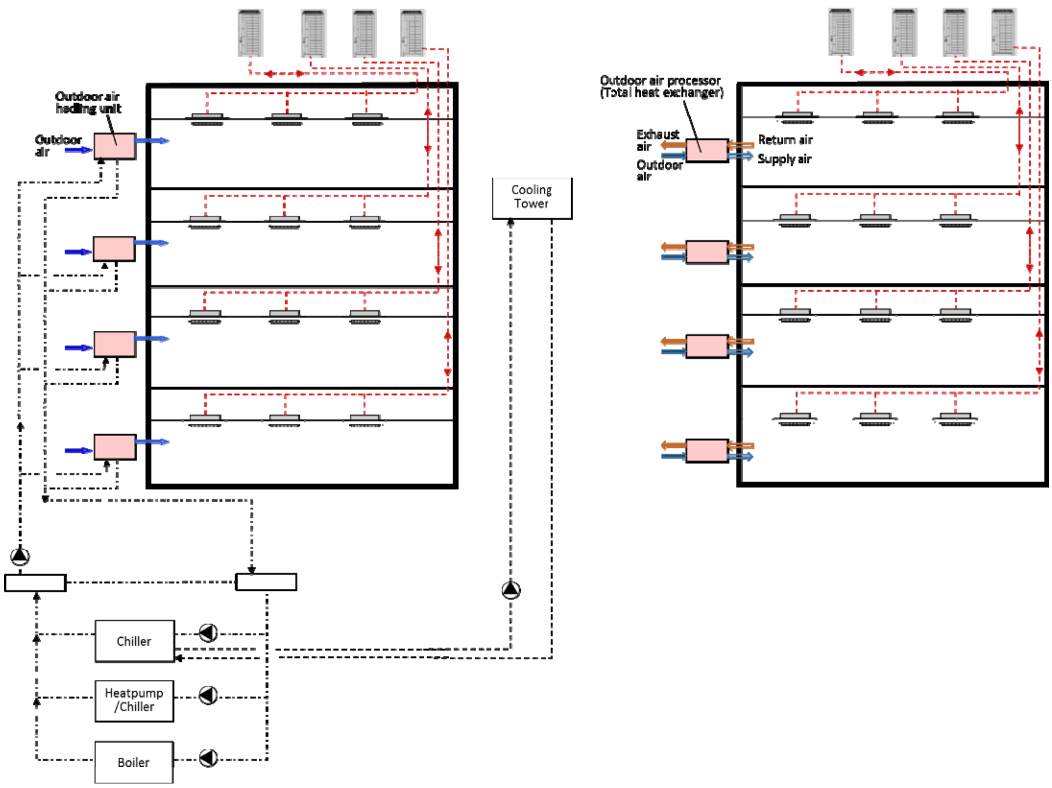


Fig. A-59 Schematic of multi-split air conditioning system.

A.5 Denmark

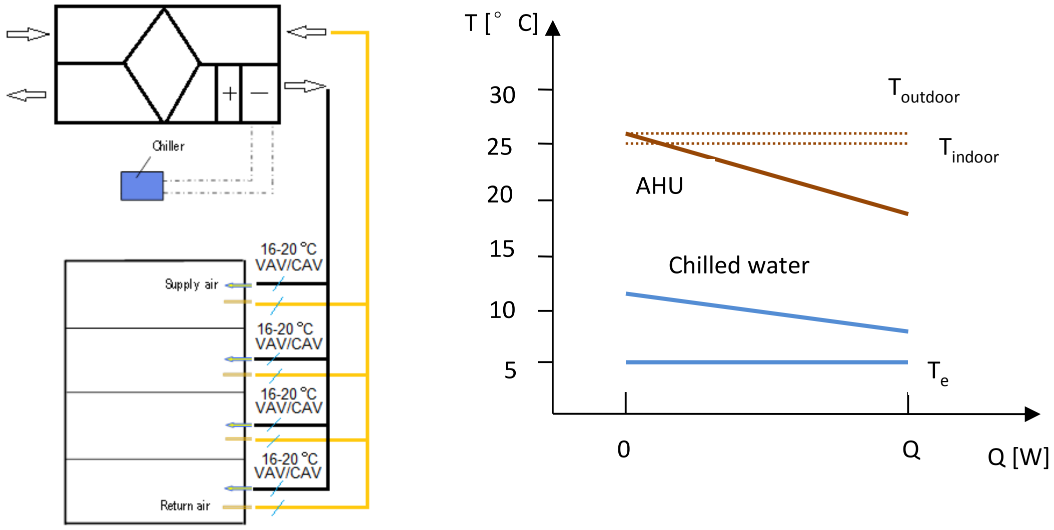
2.5.1 Cooling systems

In general, cooling in Danish context is mostly focused on avoiding mechanical cooling and implementing as much as possible passive or free cooling solutions. The reason for that is the very tight energy policy that enforces that buildings are within energy frames. The target for the building energy frames is to significantly reduce energy use in buildings by the year 2020. The mechanical cooling of the building is one of the components that is taken into account when defining energy use in the building and when defining if certain building fulfills its energy frame. Moreover, it is assumed that mechanical cooling is driven by the electrical power and is equal to that power. What is more, in Denmark electrical energy is given higher quality than, for example, energy used for hearting, and therefore presently every unit of electrical energy is multiplied by factor of 2.5. For that reason, if mechanical cooling is used in the building it is very difficult to fulfill energy frames and receive permission for building.

Nevertheless, if necessary in some cases mechanical cooling is used in the buildings and in this chapter are presented most popular systems.

a) All air system

VAV (variable air volume) or CAV (constant air volume) systems are a common type of centralized air-conditioning system in Denmark. Fig. A-60(a) gives its system principle and Fig. A-60(b) shows the T-Q figure for the air handling process. If passive measures are not sufficient to maintain indoor temperature within allowed boundaries then mechanical cooling is introduced.



(a) (b)

Fig. A-60. Typical air conditioning VAV/CAV system: (a) system principle; and (b) Temperature difference (T-Q figure) of the typical air-conditioning system (all air system).

b) FCU + OA system

Ventilation air supplied by the air handling unit (AHU) and cooling supplied by the fan coil units (FCU) there where cooling is required.

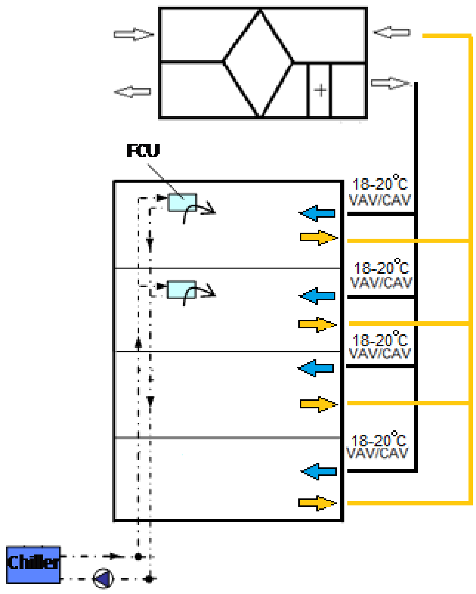


Fig. A-61 Ventilation by air handling unit (AHU) and cooling supplied by the fan coil units (FCU)

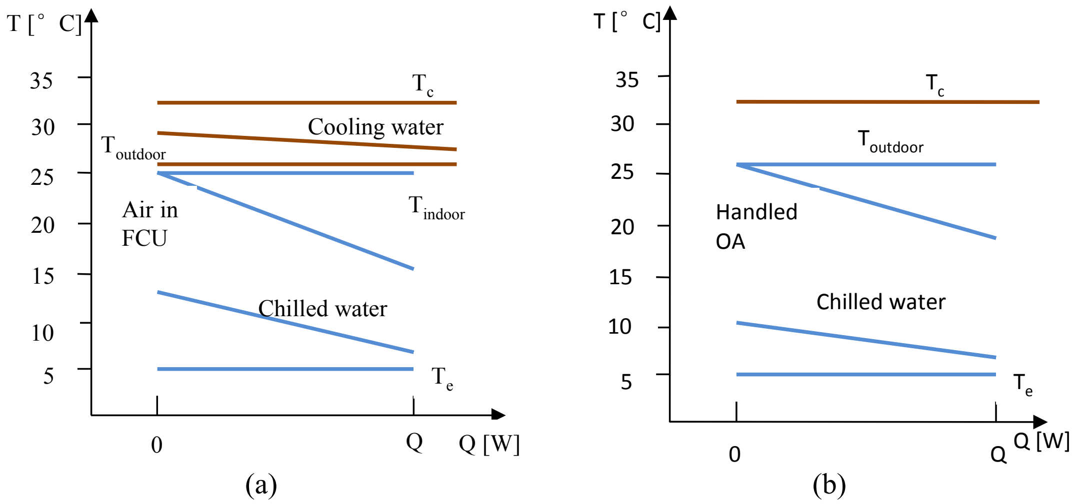


*Fig. A-62. Temperature difference (*T-Q *figure) of the typical air-conditioning system (FCU+OA): (a) The FCU system; and (b) The outdoor air handling system*

c) <u>Other cooling systems</u>

District cooling

Even though district cooling in Denmark is not as widely needed as district heating, it has its particular application locations. It is based on the same principle as district heating. Some of the application areas so far have been; a project in central Copenhagen area, Kongens Nytorv square and at the Technical University of Denmark.

In particular, district cooling in Copenhagen is obtained by heat exchangers, absorption and compression chillers. District cooling is mainly used in ventilation systems, so far an example of a radiant system coupled to district cooling hasn't been encountered.

Free cooling

It is possible to obtain free cooling (i.e. eliminating heat pump) via different sources. It could be realized by ground heat exchangers and also by sea water (one of the district cooling projects in Copenhagen uses sea water as a pre-cooling means). It is possible to combine this system with radiant systems (TABS, floor cooling).

<u>2.5.2 Heating system</u>

a) <u>District heating</u>

In Denmark 62% of all houses are connected to district heating. The fuel used in the power and heat plants is: natural gas, biogas, solar panels, geothermal sources, wind, waste heat, biomass, coal, wastes. Presently over 95% of district heating is produced in cogeneration with electricity, as waste heat from production processes and burning of wastes. Below 50% of heat is produced of fossil fuels (coal, oil and natural gas).

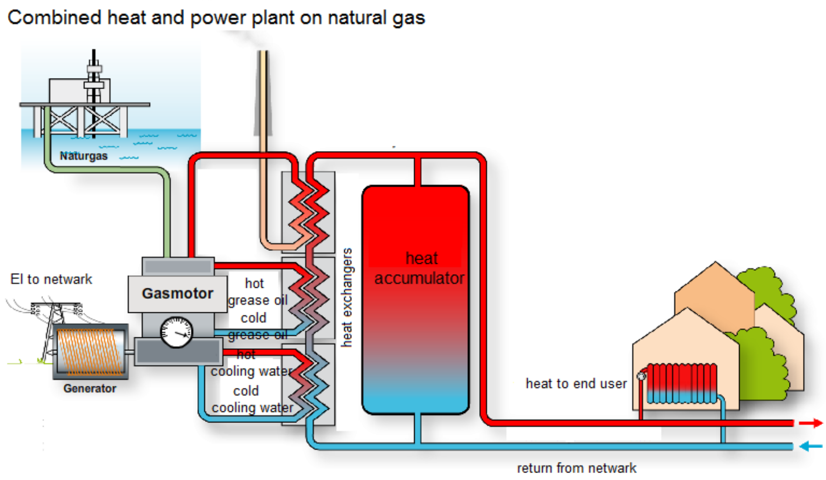


Fig. A-63 Schematic drawing of combined heat and power production and district heating

Table 2-1 District heating temperatures for some chosen Danish cities

	During heating season (at -12°C)			**Outside heating season (15.05-15.09)**		
	To	From	dT**	To	From	dT
	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
Frederikshavn	60-80	35	min 25	#	#	#
Aalborg	65/75 (max 95 °C)	30/35	min 35	#	#	#
Aarhus	60	30	30	#	#	#
Vejle	70-80	max 40	> 30	#	#	#
Esbjerg	70 (max 95)	max 40 (never >50 °C)	min 30	60	max 30	min 30
Odense	70 (max 95)	30	40	60	20	
Copenhagen	95 (max 110)	45	50	65	35	30

* Many district heating networks have introduced punishing quota if dT<30°C

** Water required cooling is the annual average

Inlet primary water temperature usually does not exceed 70°C and in the most regions necessary cooling of water within building's installation is required to be higher than 30°C. Additionally, in Denmark can be distinguished heating season and non-heating season. During the non-heating season temperature is adjusted depending on the outside temperature and wind condition.

Table 2-2 Primary and secondary water temperature for heating season.

	Direct connection	**Indirect connection**			
	Primary water	Primary water	Secondary water[a]	Secondary water[b]	Secondary water[c]
	[°C]	[°C]	[°C]	[°C]	[°C]
In	70	70	60	70	65
Out	40	40	30	40	35

[a]If heat exchanger temperatures are known
[b]If heat exchanger temperatures are not known
[c]Dimensioning parameters of many new installed radiators

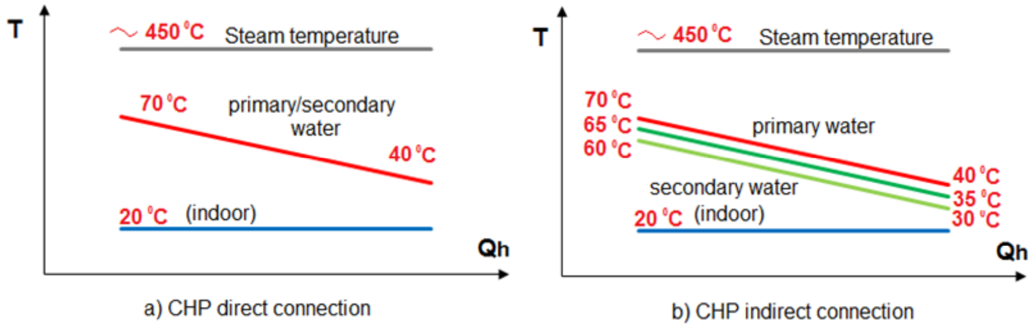


Fig. A-64 Temperature difference (T-Q figure) of conventional heating system

Connection to households can be direct or indirect. In Figs. 2-65~2-68 are presented the most common connections between district heating network and building.

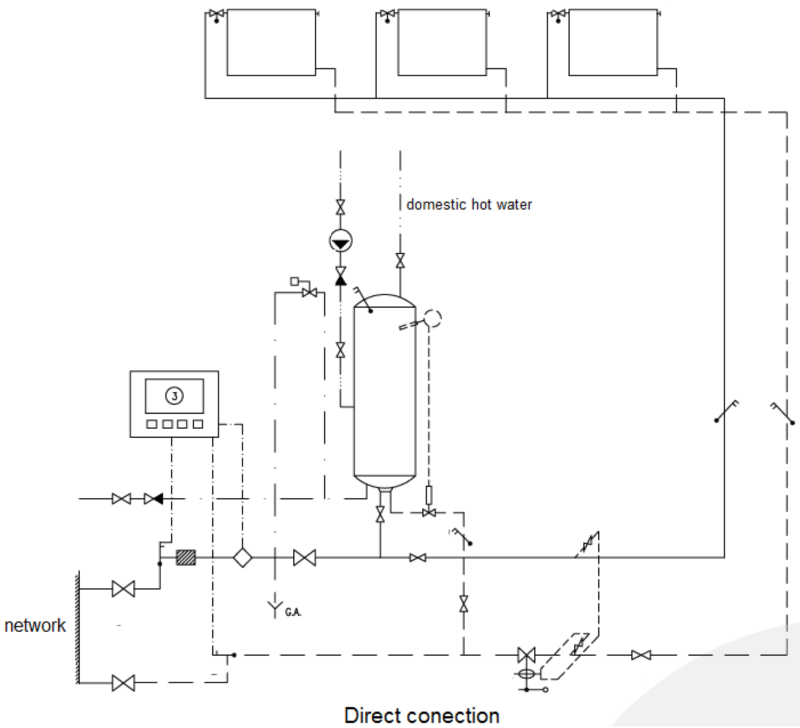


Fig. A-65 Direct connection

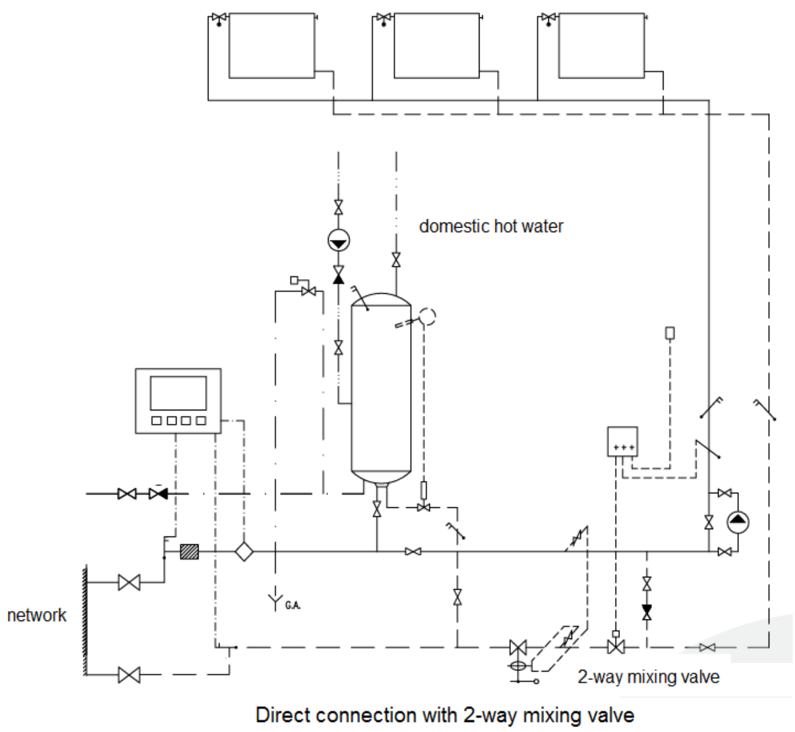


Fig. A-66 Direct connection with 2-way mixing valve

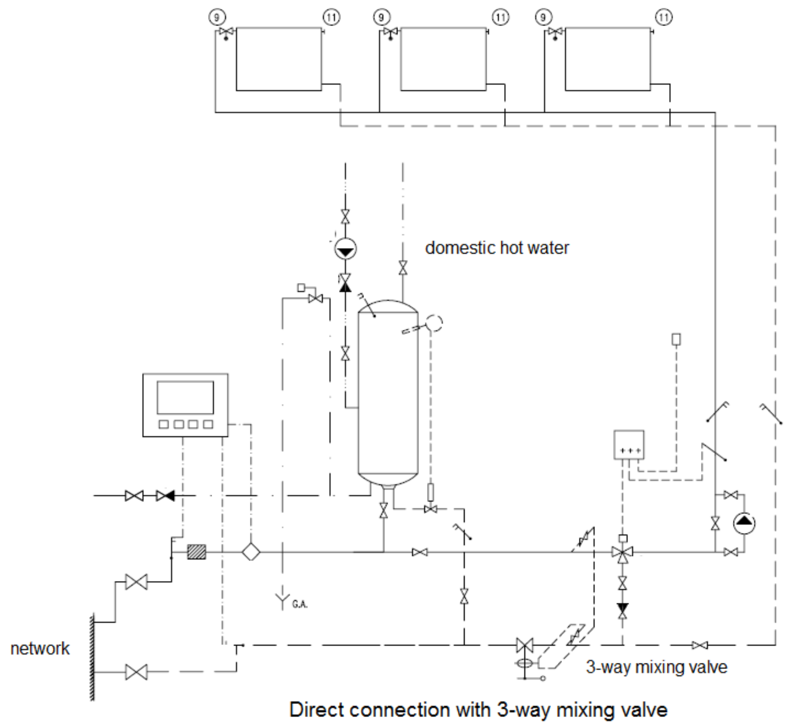


Fig. A-67 Direct connection with 3-way mixing valve

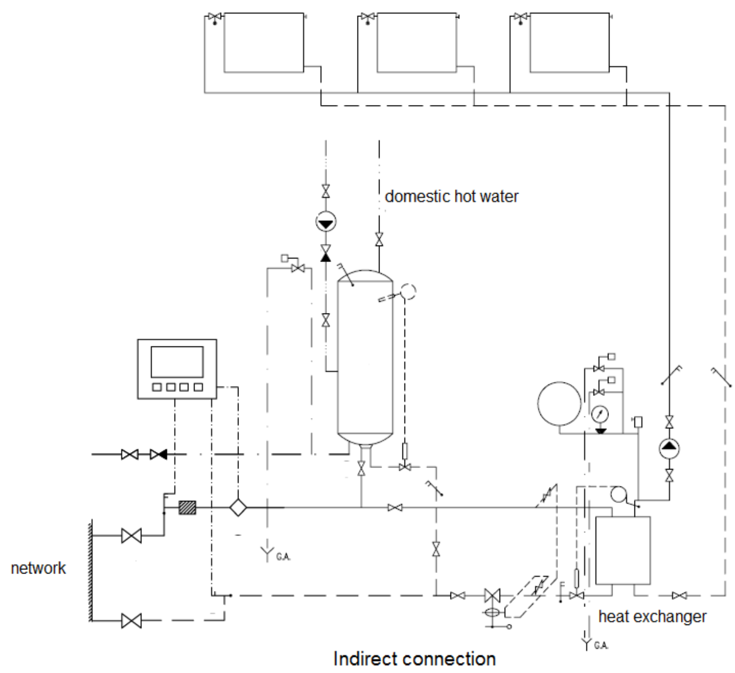


Fig. A-68 Indirect connection.

b) Boiler system

Boiler systems in Denmark burn usually gas and oil. These are usually units operating in households, production buildings and offices located further from the bigger agglomerations where district heating network does not reach.

Table 2-3 Inlet and outlet temperatures to installations with boilers

	Gas boilers		**Oil boilers**	
	Large water capacity boilers	Small water capacity boilers	Large water capacity boilers	Small water capacity boilers
	Primary/secondary water			
	[°C]	[°C]	[°C]	[°C]
In	70	55	70	55
Out	45	45	40	40

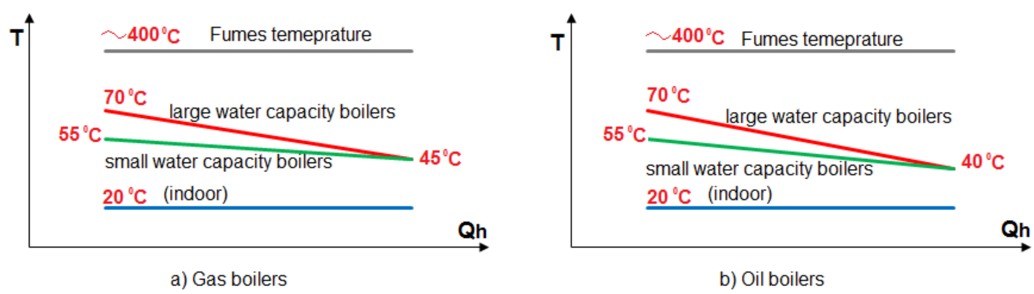


Fig. A-69 Temperature difference (T-Q figure) of conventional boiler system

c) Heat pump system

In Danish context heat pumps like boiler systems are usually used there where district heating does not reach. During recent years a tendency can be observed in changing old oil boilers to heat pump systems. In the last year over 20.000 heat pumps was sold in which majority was air to air heat pumps but also more and more water to water heat pumps with ground heat exchanger are installed. Air to air units are often installed in buildings or spaces where peak demands are significant and where discontinuous occupation is expected.

Even though ground coupled heat pump systems are more advantageous from the point of view that ground temperatures are more stable, though sometimes limitations can be faced due to regulations or spacing issues.

Coupling of heat pump systems with radiant terminals (TABS, floor heating) gives a particular advantage due to the low temperature heating concept which enables integration of renewable energy resources into heating system and enables higher *COP* values than in systems with convective radiators.

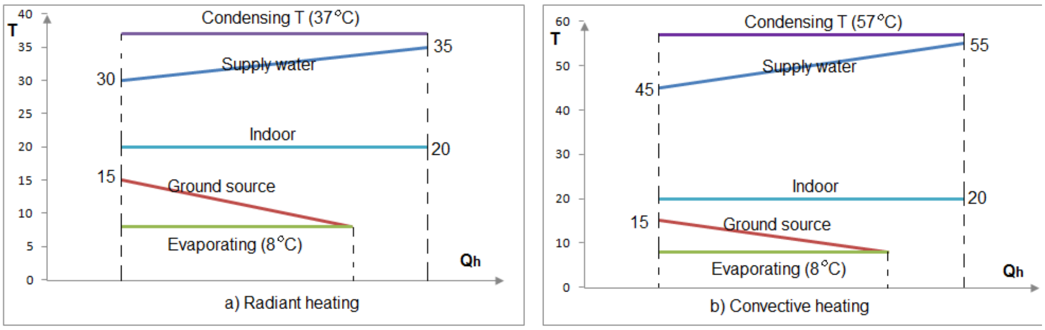


Fig. A-70 Temperature difference (T-Q figure) of the heat pump system for heating

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