Second Law Analysis of Residential Heating Systems

Xinyu Wu

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ABSTRACT

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Effective analysis methods/tools and assessment indicators appear to be essential in the design process, in order to improve the energy performance of buildings. With respect to HVAC system, which contributes to a major fraction of the building energy use and CO_2 emissions, the second law analysis is an appropriate tool to evaluate its performance.

This thesis presents the second law analysis of 20 design alternatives of the HVAC-DHW system of a house located in Montreal. The following design alternatives were considered:

- heating system: (1) electric or hot water baseboard heaters, (2) forced air system, and (3) radiant heating floor;
- heating equipment: (1) electric or gas-fired boiler, (2) ground-source-heat-pump, and (3) air-to-air heat pump;
- heat recovery: (1) air-to-air heat exchanger, (2) earth-tube heat exchanger, and boiler economizer.

Engineering Equation Solver (EES) program was used to perform the second law analysis for both peak design conditions and annual operating conditions. The results show that the HVAC-DHW system using electric baseboard heater and separate electric air

ventilation heater has the lowest exergy efficiencies of 7.4% and 4.1%, respectively, under the above two conditions, while its energy efficiencies are 64.2% and 61.6% respectively under the same conditions. The HVAC-DHW system with the highest exergy efficiency of 23.2% (at peak design conditions) and 10.9% (at annual operating conditions) is composed of: radiant floor heating system; ground source heat pump; air-to-air heat exchanger; earth tube heat exchanger; gas-fired DHW heater.

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Table of Contents

LIST OF FIGURES

LIST OF TABLES

Nomenclature

A	Total floor area of the house (m^2)
A_{tube}	Section area of the earth tube (m^2)
c_{pa}	Specific heat of air at $T=(T_i+T_o)/2$, (kJ/kg·K)
c_{pw}	Specific heat of water (kJ/kg·K)
COP	Coefficient of performance of the heat pump
EIR	Electric input ratio
$\dot{E}$	Energy flow rate (kW)
ECO_2	Total annual equivalent CO_2 emissions due to the energy use in the HVAC-DHW system, (kg CO_2/yr)
ECO_2-1	Annual equivalent CO_2 emissions due to the electricity generation, (kg CO_2/yr)
ECO_2-2	Annual equivalent CO_2 emission due to the on-site fossil fuel use, (kg CO_2/yr)
$\dot{F}$	Electric power of fan (kW)
$FRAC$	Fraction of the hour the GSHP is on
h	Specific enthalpy (kJ/kg)
H	Height of the room, (m)
L	Length of earth tube, (m)
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (kPa)
$\dot{P}$	Electric power of pump (kW)
PLR	Part load ratio
$\dot{Q}$	Heat flow rate (kW)
$\dot{Q}_L$	Heating load of the house (kW)
$RMIN$	Minimum part load ratio
s	Specific entropy (kJ/kg·K)
$\dot{S}$	Rate of entropy (kW/K)
T	Temperature (°C)
TK	Temperature (K)
v	Air velocity (m/s)
$\dot{V}$	Volumetric flow rate (m^3/s)
$\dot{W}$	Rate of work (kW)
$\dot{X}$	Rate of exergy (kW)

Greek symbol

α	Convective heat transfer coefficient, ($W/m^2 \cdot K$)
α_{gas}	Contribution of natural gas to the off-site electricity generation
α_{oil}	Contribution of oil to the off-site electricity generation
α_{coal}	Contribution of coal to the off-site electricity generation
$\alpha_{nuclear}$	Contribution of nuclear energy to the off-site electricity generation
α_{hydro}	Contribution of hydro to the off-site electricity generation

$a_i\,(i=1\text{-}5)$	Equivalent CO_2 emissions for hydro, heavy oil, coal, natural gas, and nuclear power, respectively in the power plants, (kt CO_2/TWh)
β	Air exchange rate of the house (ach)
$\eta_{gboiler}$	Energy efficiency of gas-fired boiler
η_{HE}	Heat transfer efficiency
η_{pp}	Energy efficiency of power plant
η_{trans}	Efficiency of electricity transmission
η_1	Energy efficiency of the system
η_2	Exergy efficiency of the system

Subscripts

a	Air
ashp	Air source heat pump
cap	capacity
comp	Compressor
cond	Condenser
da	Air at ASHP design condition
de	(Exergy) destruction
dg	Ground water at GSHP design condition
Dw	Domestic hot water through the condenser
ea	Exhaust air
ebase	Electric baseboard heater
eboiler	Electric boiler
econ	Boiler economizer
eDHW	Electric DHW tank
eheater	Electric air heater
evap	Evaporator
de	Destruction
flueg	Flue gases
g	Ground
gboiler	Gas-fired boiler
gen	(Entropy) Generation
gshp	Ground source heat pump
hDHW	Warmer side of the hot water DHW tank
HE	Air-to-air heat exchanger
hwHE	Warmer side of water-to-water heat exchanger
i	Inside of the house
in	Air or water flowing in
is	Isentropic compression process
mix	Mixing box
o	Outside of the house
out	Air or water flowing out
pp	Power plant
r	Refrigerant
sp	Space heating

surr	Surroundings
sys	System
tube	Earth tube heat exchanger
underg	Underground heat exchanger
w	Water
warh	Hot water air reheater
wbase	Hot water baseboard heater
wHE	Water-to-water heat exchanger
wheater	Hot water air heater

CHAPTER 1

INTRODUCTION

1.1 Introduction

To improve the energy performance of buildings, effective analysis tools and assessment indicators appears to be essential. Leskinen and Simonson (2000) stated that developing a comprehensive set of tools to assess low exergy technologies, components and systems is an important task of IEA Annex 37. With respect to HVAC system, which contributes to a major fraction of the building energy use and CO_2 emissions, the second law analysis is an appropriate tool to assess and evaluate its performance.

The energy performance of HVAC systems is usually evaluated based on the first law of thermodynamics. In many cases, some energy performance improvements can be estimated through energy analysis, which is based on the application of energy conservation principle; this approach, however, does not distinguish the quality difference in energy sources. As system configurations become more complex and advanced engineering technologies are applied, the energy analysis alone is not adequate to gain a full understanding of all the important aspects of energy utilization processes, and sometimes leads to less important aspects for improvement (Schmidt, 2003). Compared to energy analysis, the exergy analysis can better and accurately show the location of inefficiencies. The results from exergy analysis can also be used to assess and optimize the performance of HVAC systems. In addition, the second law analysis can present a whole picture of the system performance. A number of applications of exergy analysis in HVAC system have shown its effectiveness.

Renewable energy sources and energy efficient technologies may be utilized to improve the effectiveness of energy use in HVAC system. Some HVAC systems adopted energy efficient technologies, and renewable energies, but failed to meet the expected improvement. The reason is that these technologies are not correctly integrated into the system. For a specific project, the selection of renewable energy sources and energy efficient technologies is influenced by many factors, such as local climate, building location, hours of operation, people's habits and cost of energy sources. In order to achieve sustainable design of HVAC system, renewable energies and energy efficient technologies should be integrated in such a manner where entropy generation and exergy destruction are minimized.

In this thesis, the second law analysis is first reviewed. Several design alternatives of HVAC-DHW systems for a residential building and their component models are then generated in Chapters 3 and 4. In Chapters 5 and 6, these design alternatives and the components are evaluated based on their energy and exergy performance and annual equivalent CO_2 emissions.

1.2 Objectives

The first objective of this research is to apply the second law analysis to the assessment of the energy performance of HVAC systems and equipment. This analysis method will be helpful for the designing of HVAC system, the evaluation of new technologies, and the optimization of the energy performance for HVAC-DHW systems and equipment.

The second objective of this research is to integrate energy efficient technologies, 'cascading' energy use strategies and renewable energies into HVAC-DHW systems to reduce the non-renewable energy use for a house in Montreal.

CHAPTER 2

LITERATURE REVIEW

2.1 Exergy analysis and its application

Building is a major energy consumer in modern society, and HVAC systems contribute to a large fraction of the building energy demand and CO_2 emission. Some estimates show that HVAC systems account for approximately 20% of the total energy use globally nowadays (Ren et al, 2002). This situation requires the increase in the efficiency of energy utilization in buildings and HVAC systems. Usually, potential energy performance improvements are estimated through energy analysis methods, which are mostly based on the application of energy conservation principle (first law of thermodynamics); this approach, however, does not distinguish the quality difference in energy sources. As system configurations become more complex and advanced engineering technologies are applied, the energy analysis alone is not adequate to gain a full understanding of all the important aspects of energy utilization processes, and sometimes leads to less important aspects for improvement (Schmidt, 2003).

Exergy is an important thermodynamic concept, which can be used to assess the potential improvements, since it can link the physical or engineering world and the environment, and can express the true efficiency of engineering systems (Dincer et al, 2004). Exergy is a useful concept to evaluate the quality of energy sources. Using exergy, different types of energy sources, such as solar energy, geothermal energy, fossil fuel energy and electricity, can be compared to each other (Dincer et al, 2004). Exergy analysis, which is based on the combination of the first and second laws of thermodynamics, can provide for a better understanding of energy utilization processes and locate the real inefficient areas for improvement (Simpson and Kay, 1989).

2.1.1 Exergy and exergy analysis

The first law of thermodynamics deals with the quantity of energy, and states that the total amount of energy is conserved and cannot be created and destroyed, even though energy may change from one form to another (Cegel and Boles, 2002). However, the first law of thermodynamics does not distinguish energy sources from their qualities. Sometimes, the difference between input energy and output energy is not clear and leads to a confusing situation in the analysis of energy conversion process. Further, the first law of thermodynamics, generally fails to identify losses of work and potential improvements or the effective use of energy in a specific component or process, e.g., in an adiabatic throttling process (Simpson and Kay, 1989).

The second law of thermodynamics exposes the quality of energy and its degradation during a process, and therefore the lost opportunities to do work. The second law of thermodynamics shows that, for some energy forms, only a part of the energy can be converted to work, and explains explicitly that, in an energy conversion process, exergy is consumed or destroyed, and entropy is created (Cegel and Boles, 2002).

From the thermodynamic point of view, exergy is defined as the maximum useful work that could be obtained from a system at a given state in a specified environment. Exergy measures the quality of energy, while entropy measures the molecular disorder of a system. In a process or system, the total amount of exergy is not conserved (except for an ideal or reversible process) but can be destroyed due to internal irreversibilities and heat transfer crossing the system boundaries. The exergy destruction is proportional to the entropy created due to irreversibilities associated with the process (Cegel and Boles, 2002). Table 2.1 summarizes the main differences between energy and exergy.

Since Carnot stated that the maximum useful work which can be obtained from a heat engine is proportional to the temperature difference, which later led to the second law of thermodynamics, in 1824, the theory of exergy has been developed and applied to a variety of engineering fields (Wall and Gong, 2001a). Gibbs (1928) introduced the concept of available work, including the diffusion term, to account for the extractable work. Keenan (1948) interpreted the extractable work as 'available' energy. In 1953, the term exergy was suggested by Rant to denote technical working capacity (Rant, 1956). Later in 1977, Wall (1977) proposed a concise exergy theory and introduced exergy as a useful concept not only in engineering but also for improving the resource use and reducing environmental destruction.

During the past two decades, there has been increasing interest from scientists, researchers, and engineers in the use of the concept of exergy in the following aspects (Dincer et al, 2004):

- Environmental impact of energy utilization;
- Efficient energy utilization;
- Accurate location of the inefficient areas in energy conversion systems;
- Design of the energy efficient system;
- Quality difference between energy resources.

Table 2.1 Energy versus exergy (Dincer et al, 2004)

Energy	Exergy
The first law of thermodynamics deals with energy.	The second law of thermodynamics deals with both energy and exergy.
Energy is always conserved; it can not be created or destroyed.	Exergy is conserved in a reversible process but it is destroyed in a real process.
Energy is a measure of quantity.	Exergy is a measure of quantity and quality.
Energy is motion or ability to produce motion.	Exergy is work, i.e., ordered motion, or ability to produce work.
Energy change can be calculated from the following formula:	Exergy destruction can be calculated from the following formula:
$\Delta Q = \Delta U + \Delta W$ (kJ)	$X = T_0 \cdot S_{generated}$ (kJ)
where:	where:
ΔQ is added energy as heat to the system, kJ;	X is the exergy destruction, kJ;
ΔU is the change of the internal energy U of the system, kJ;	T_0 is the absolute temperature of the environment, K;
ΔW is extracted energy as work from the system, kJ.	$S_{generated}$ is the entropy generation of the system, kJ/K.

2.1.2 Exergy analysis and its application in HVAC systems

Exergy analysis is based on the combination of the first and second law of thermodynamics, and generally allows the inefficient areas to be better located than does energy analysis (Rosen et al, 2001). In exergy analysis, not only energy related parameters, such as energy efficiency and energy demand, but also exergy related parameters, such as exergy efficiency, entropy generation, exergy destruction, and exergy demand, can be obtained. Unlike the energy analysis in which only the amount of input energy and output energy is needed, the exergy analysis requires, in addition, information such as the temperatures of energy sources and sinks as well as the temperature of environment. For conventional energy technologies, the energy performance can be improved based on the results of energy analysis. For advanced energy technologies, energy analysis alone is not adequate; and the combination of energy analysis and exergy analysis can be expected to evaluate the system energy performance and locate the potential improvement areas (Rosen et al, 2001).

The strategies recommended to improve the efficiency are quite different depending on if they are based on exergy analysis or energy analysis. For example if there is a need for more space heating capacity, an exergy strategy would consist of identifying appropriate energy sources which match the task of space heating in the region and ways of making use of the sources. From energy point of view, the strategy would consist of using highly energy efficient electrical heaters and thus building more power plants (Simpson and Kay, 1989). Clearly, efforts of increasing the energy use efficiencies should focus on the inefficient areas identified by exergy analysis.

Compared to energy analysis, the exergy analysis can identify where an energy source is being used inappropriately, such as the case where electricity is generated from the combustion of fossil fuel and is used in baseboard heaters for space heating. The exergy analysis can also indicate that energy is efficiently used when the quality of the source is matched to the quality demanded by the task. By thermodynamically matching energy sources to tasks, the enormous waste of using high quality energy for low quality tasks can be avoided, and as a result, the increasing social and economic costs of energy production can be decreased (Simpson and Kay, 1989). The exergy analysis can indicate the locations and causes of inefficiency in a process or system accurately. For instance, in the case of electricity generation in thermal plant, exergy losses are associated primarily with internal consumption (combustion and heat transfer components). Therefore, to improve the efficiency, more consideration should be paid on combustion and heat transfer components. If only the energy analysis is applied, heat rejected by condensers accounts for the major energy losses (Rosen, 2001).

Only a few applications of exergy analysis to the HVAC systems were found in the literature. Exergy analyses have been performed to evaluate the energy performance of heating systems and their components in the Annex 37 of International Energy Agency (IEA, 2002). The results have shown that: there is enormous waste of exergy, when electricity is used for space heating; low quality energy tasks such as space heating can be provided more efficiently and less expensively by other means, such as geothermal energy and solar energy; ground source heat pumps are an excellent way to make use of the low quality heat from the ground to provide for the low quality energy demand of space heating; electricity can be used to perform other tasks that require a high quality energy source, such as driving electric motors (Leskinen and Simonson 2000).

The Annex 37 of International Energy Agency reached the following conclusions concerning the space heating system with boiler (IEA, 2002):

- Improving the building heating efficiency is more beneficial than improving the efficiency of a boiler, in order to decrease the total exergy consumption.
- The efficiency improvement of the boiler and its sub-systems has meaningful results only together with the improvement of building heating system.
- The selection of HVAC components such as heat pumps and boilers should be made in a consistent manner with each other to improve the whole system exergy efficiency.

In order to promote the rational use of energy by means of facilitating and accelerating the use of low valued and renewable energy sources for space heating and cooling of buildings, the concept of low exergy heating or cooling system is defined in the IEA annex 37 (Leskinen and Simonson, 2000) as: “a system, which allows the use of low valued energy as the source.” This means a system where the heat or cold supply temperatures are as close as possible to the desired room temperature. The definition of low exergy system provides guidance for the design in buildings and HVAC systems with respect to the energy use.

Kanoglu et al. (2004) analyzed an experimental open-cycle desiccant cooling system and showed that the desiccant wheel has the greatest percentage of total exergy destruction followed by the heating system. Li et al. (2004) analyzed the exergy performance of a district heating system, and found that if the heating mode is switched from boiler heating or electric heating to a mode, which makes use of renewable energy or waste energy, the fossil fuel consumption and emissions, can be reduced significantly. Asada and Takeda (2002) found that the ceiling radiant cooling system with well water is not exergy efficient because of its relatively large electricity consumption by pumps. Badescu (2002) found that in a vapor compression heat pump, most exergy losses occur during the compression and condensation process. Ren et al. (2002) evaluated the performance of evaporative cooling schemes and showed that the regenerative evaporative cooling has the best performance, and the effectiveness of indirect evaporative heat exchange has great importance in improving the exergy efficiency of regenerative scheme.

Recently, the exergy modeling techniques have been used to evaluate the performance of energy conversion system. The exergy modeling techniques have been applied to various industrial sectors and thermal processes (Dincer et al, 2004). In relation to the energy analysis of buildings, Rosen et al. (2001) expressed the opinion that one major weakness in the building modeling and simulation is the lack of using the second law analysis and exergy modeling techniques.

2.1.3 Exergy and environment

In addition to its application in the improvement of efficiency of thermal system, exergy can also be used to assess the environmental impacts. The concept of exergy and the related concept of entropy can offer a better understanding of the depletion of natural resource and corresponding environmental impacts and therefore give an opportunity to improve the resource use and to reduce the environmental impacts (Rosen and Dincer, 2001; Rosen and Dincer, 1999; Zhang and Reistad, 1998; Finnveden and Ostlund, 1997; Shukuya and Komuro, 1996; Ayres and Martinas, 1995; Szargut, 1978; Wall, 1977;).

Wall (1977) outlined the basic idea of incorporating the concept of exergy into the accounting of natural resources. Energy and material resources were treated in terms of exergy, and exergy analysis was proposed as a method for calculating the total performance of a product or a service. Szargut (1978) suggested that "the index of cumulative consumption", which means the loss of exergy of deposit resources, can be redefined as an index of ecological costs. Ayres and Martinas (1995) stated that in the case of a waste residual, exergy could be regarded as the potential for doing harm to the environment by causing undesirable effects on the environment. Shukuya and Komuro (1996) presented a new important concept of the "Exergy-entropy process". According to this concept, all processes of biological systems, engineering systems or even the global environmental systems can be regarded as "exergy-entropy processes". An "exergy-entropy process" consists of a series of four steps: "exergy is first supplied; its portion is consumed; entropy is generated accordingly; and entropy is finally dumped into the environment." For a process, exergy destruction represents the resource depletion while entropy generation expresses the environmental impacts. Finnveden and Ostlund (1997) have successfully introduced exergies of natural resources into the methodology of environmental life cycle assessment. Zhang and Reistad (1998) used a method for the overall evaluation of energy conversion systems. In this method, exergy was used as an overall evaluation measure of the performance of energy conversion systems over their life time including global warming aspects. Rosen and Dincer (1999) presented an application of exergy analysis to waste emissions. They concluded that exergy can make a substantial contribution to the evaluation of environmental problems. Wall and Gong (2001a) stated that, if exergy resource is used at a faster rate than it can be renewed, natural resource would be degraded and finally depleted; if the entropy disposed to the environment is greater than the ability of the environment to absorb it, the environment would be damaged.

Rosen and Dincer (2001) presented the linkage between exergy and environment from three aspects:

- On the very fundamental level, exergy destruction and entropy generation indicate the chaos creation or the destruction of order in a system, which is a form of environmental damage.
- The degradation of resources found in nature, which is another form of environmental damage, can be measured by exergy destruction. The exergy deposit on the earth represents our precious natural resources.

- The exergy associated with waste emissions can be viewed as a potential damage to environment. This is because if an amount of exergy is brought to equilibrium with the environment, it will cause change.

The above presented studies underlined that the concept of exergy could successfully links the fields of energy, environment, and sustainable development, and is gradually being adopted as a useful tool in the development and design of a sustainable society. Although the relationship between exergy and environment can address environment concern, its use in the environment assessment is still in its infancy and much effort needs to be devoted to exploit its full benefits. Therefore, in order to achieve sustainable development, updating the knowledge of energy use and the physical resource use in engineering even in the society is necessary (Wall and Gong, 2001b).

2.2 Energy performance indicators for green and sustainable buildings

2.2.1 Energy and Sustainable development

In 1987, a report known as the Brundtland Report was published by the United Nations (WCED, 1987) in which sustainable development was defined as: "sustainable development meets the needs of the present without compromising the ability of future generations to meet their own needs." Search for sustainability covers the solutions of the environmental, economic and social aspects. In the environmental aspect, currently, energy use plays an important role in the definition of some important sustainability indicators (Todd et al., 2001).

Energy is essential for the social and economic growth. It provides basic needs and services such as heating, cooling, cooking, lighting, and transportation and is a critical production factor in all sectors of industry. The production and use of energy also cause environmental damage at all levels - local, regional and global. For example, combustion of fossil fuels leads to air pollution by particulates and oxides of sulphur, nitrogen, and etc.; hydropower often causes environmental damage due to the submergence of large areas of land; and global climate change associated with the increasing concentration of greenhouse gases in the atmosphere has become a worldwide concern today. Natural resource depletion, accumulation of wastes, deforestation, water pollution and land disturbance are further examples of energy-related environmental concerns (IAEA and IEA, 2001).

Globally, the demand for energy is increasing with social and economic development. Much of the world's energy resources are used in ways that are not sustainable. Furthermore, the control of the atmospheric emissions of greenhouse and other gases and substances will increasingly rely on the improvement of efficiency of energy conversion, transmission, distribution and end use, and on environmentally friendly energy systems, particularly those using renewable sources of energy (IAEA and IEA, 2001).

The depletion of non-renewable natural resources (energy and materials) and environmental impacts are the most important environmental issues under recent consideration by the building and energy industries. ASHRAE released its Building Sustainability Position Statement to encourage the design of HVAC systems to "provide safe, healthy and comfortable indoor environments while simultaneously limiting their impact on the earth's natural resource." (ASHRAE, 2002). Sustainability is the most important challenge to modern building industry, and the appropriate design, construction, and operation of HVAC systems can play a critical role to achieve sustainability.

2.2.2 Energy performance indicators for green and sustainable buildings

Although the objectives of sustainable development are very broad, we still need a set of indicators to measure and monitor important changes and significant progress towards the achievement of these objectives. For energy sector, the provision of adequate indicators of energy use in accordance to social, economic, and environmental developmental needs, is an important element towards sustainable development.

A sustainability indicator should be easy to understand and be an unambiguous quantity. Within the Organization for Economic Co-operation and Development (OECD) an indicator is defined as: "a parameter or a value derived from parameters, which points to, provides information about, and describes the state of a phenomenon/environment/area, with a significance extending beyond that directly associated with a parameter value" (OECD, 1994). Indicators are bits of information that should reflect the status of large systems. They provide a way of seeing the "big picture" by looking at a smaller piece of it. They tell us to which direction a system is going: getting better or worse or staying the same. Sustainable development indicators are used to determine whether a development of a system is towards or away from sustainability. Complex problems of sustainable development require integrated or interlinked sets of indicators, or an aggregation of the indicators themselves (Wall and Gong, 2001b).

During the past few years, several tools and indicators were developed to measure environmental and energy impacts of buildings. Most development came from countries such as Canada, United States, France, Germany, Denmark, United Kingdom and The Netherlands (Editorial, 2001). So far, most building assessment indicators and tools have focused on incremental environmental improvements designed to produce 'green' or 'greener' buildings. Now, there is an increasing need to bring the sustainability concept into the assessment tools and indicators (Todd et al., 2001).

Green building practice is supposed to achieve improvement in the environmental performance of building relative to current practice or requirements. Thus, green performance is most described in relative terms. Compared to the green concept, sustainability is a much broader concept. It has environmental, social and economic dimensions, embraces all facets of human activity and its scale spans from local actions through to redressing the major inequities that exist between developed and developing nations. Sustainability is very useful when applied on a global scale. The assessing progress towards sustainability needs absolute performance values. However, the considerable practical overlap between 'green' and 'sustainability' suggests that they can be reconciled within a single assessment tool (Cole, 1999).

BREEAM, developed by the Building Research Establishment in UK, was the first assessment system implemented and it has served as a model for many other systems, such as HK-BEAM in Hong Kong and BREEAM-Canada (Dickie and Howard, 2000). In this system, carbon dioxide production due to energy consumption is selected as the energy performance indicator, since CO_2 emissions have a direct environmental impact and allows the type of primary fuel to be taken into consideration. Annual equivalent CO_2 emissions due to the building operation are calculated, and then compared to the standard emission value to award credits (Dickie and Howard, 2000).

LEED (Leadership in Energy and Environmental Design) green building rating system (USGBC, 2002), uses two energy performance indicators which are easy to use as design and assessment tools:

a) Points of energy performance, which are awarded compared to the prerequisite standard regulated by ASHRAE/IESNA Standard 90.1-2002. For instance, one point is awarded if the design energy use of a new building is lower than the prerequisite standard by 15%.
b) Points of use of renewable energy, which are awarded based on the increasing level of self-supply through renewable energies. For instance, if at least 10% of the building energy use is supplied through the use of on-site renewable energy systems, one point is awarded.

GBC (Green Building Challenge) framework and its application software GBTool (Cole, and Larsson, 2002), evaluates the energy performance of buildings using several indicators. These indicators were not scored on a regionally adaptable scale, but instead were presented as absolute numbers to reflect global sustainability and to permit international comparisons. These indicators are the following: total net consumption of primary embodied energy, in GJ; Net annualized consumption of primary embodied energy, in MJ; Net annual consumption of primary energy for building operations, in MJ; Net annual consumption of primary non-renewable energy for building operations, in MJ; Net annualized primary embodied energy and annual operating primary energy, in MJ; Net annual GHG emissions from building operations, in kg CO_2 equivalent.

Another indicator is the energy ecological footprint, which is defined as the land area required for sequestering the amount of CO_2 emissions from burning fossil fuels, for buffering the radiation from nuclear power, and for building dams to generate hydrological electricity (Wackernagel et al, 1999a). Because its physical unit is very familiar to the general public and policy makers, energy ecological footprints can raise public and political awareness of environmental impacts. These aspects give the energy ecological footprint analysis wide application potential. Energy ecological footprint can also be used to assess the sustainability of current human activities, and also an educational tool for public awareness and decision-making (Wackernagel et al, 1999b). However, energy ecological footprint is a relatively new indicator developed, and there is still a great need for further development of its application for building environmental assessment.

Affordance is an indicator that expresses the performance of a building design in relation to a context (Ries and Mahdavi, 2001). For a specific context, there is an allowance level and affordance can determine whether an impact meets the allowance level. For instance, allocation and use of energy could be expressed in terms of an affordance indicator. Areas with low energy availability would have a low affordance. The advantage of this type of indicator is that when emissions are related to a context or eco-system type over a period of time, the ability of the ecosystem to sustain the impacts from the emissions can be included in the indicator.

Exergy as a thermodynamic property is founded on physical principles and gives a consistent theoretical value to energy and material resources with respect to the environment. It also has close linkage with environmental concerns as discussed above. Several research works have shown that exergy has a great potential to be used as an ecological, environmental and sustainability indicator (Wall and Gong, 2001b).

Szargut (1978, 1997) suggested that the loss of exergy of deposit resources can be defined as an index of ecological costs. Wall (1993) proposed that the exergy of emissions can be used as an indicator of environmental effects. Cornelissen (1997) proposed a method called Exergetic Life Cycle Analysis (ELCA), where the exergy destruction is used as a single criterion for the depletion of natural resources. Jorgensen and Nielsen (1998) emphasized that exergy could be used as an ecological indicator, as it expresses energy with its quality. It measures the energy that can do work. An investigation of exergy as an ecological indicator was presented by Gong (1999). In this study it is noted that exergy can be further applied as a useful concept in the environmental field. A research project called 'Exergy as an Environmental Indicator' is underway at the University of California, Berkeley (Wall and Gong 2001b).

From the above presented reference, one can conclude that the concept of exergy has the potential to link the fields of energy, environment, and sustainable development. It is gradually adopted as a useful tool in the development and design of a sustainable society. So far, it was mainly used as a complement to the energy analysis, to describe, analyze and improve energy systems and processes. However, it is a much more useful concept, and could be applied across the breadth of ecosystem and industrial application (Wall and Gong 2001a). By adopting exergy based tools and indicators, we will get "an exacting and demanding process" (Wall and Gong 2001b).

2.3 Integrated HVAC system, energy efficient technologies, and renewable energies

2.3.1 Integrated HVAC system

In order to reduce the nonrenewable energy use and minimize the environmental impact in HVAC field, making use of the low valued energy and waste energy resources is a feasible way. In residential buildings, except for lighting and electric appliances, most residential energy can be classified as so-called low temperature level energy ranging from 5 to 60°C (Saitoh and Fujino, 2001). For example, space heating, cooling, and domestic hot water are categorized into this range, and can be supplied with natural renewable energies such as geothermal, solar and wind energies as well as waste energy resources. Low valued energy and waste energy resources are abundant in all countries; however they are not effectively used. Almost 75% of the available waste and alternative energy resources have a temperature below 75 °C (Kilkis, 2002). As a result, developing integrated HVAC systems which can be directly coupled to low valued energy resources without the impact on our precious environment is necessary.

The indoor environment is a highly complicated system in which some parameters can be controlled and some cannot. In order to significantly increase the energy performance of the HVAC systems in buildings, while satisfying the requirements of a comfortable indoor environment, it is necessary to use a combination of energy efficient technologies, energy cascading use strategies, and renewable energy technologies. This requires an integrated design approach, where the different low valued energies to be used are considered to be integral parts of the system. The complexity of building performance requires that each technology cannot be considered in isolation from the rest of the design (Kilkis, 2002). To significantly reduce the total energy use, the designers of these buildings therefore need to find the optimum combinations of technologies for each specific case.

Today, the major obstacle in large-scale utilization of sustainable and alternative energy resources in buildings is the lack of available heating, cooling and air-conditioning systems that can be directly coupled with low temperature energy resources. Experience gained from energy-efficient and sustainable buildings demonstrates very clearly that this ambitious goal requires integrated design processes. This situation is specifically true for the HVAC system design, but an absence of knowledge and experience of integrated approaches remains widespread in the traditional planning community (Kilkis, 2000). Therefore, there is a need to identify, develop, apply, demonstrate, and document design knowledge base for new and alternative HVAC system, which can be integrated with existing and potentially new alternative energy resources without any compromise.

2.3.2 Renewable energy sources and energy efficient technologies

There are a number of renewable energy sources available in nature such as solar energy, geothermal energy, wind energy, bioenergy, ocean energy, and hydropower. For a particular project, the selection of renewable energy will be influenced by many factors such as the local climate, building location, hours of operation and cost of non-renewable energy sources including unit price, service cost and infrastructure cost.

Experiments and research on various energy efficient technologies and their combinations have been carried out throughout the world in recent years. Watts and IIp (2002) studied the feasibility of an energy self-sufficient office with the design integrating biomass, photovoltaic, solar thermal panel, wind turbine and seasonal heat storage. Nowak et al. (2002) discussed a building in Poland, which will be built to demonstrate maximal efficiency, and minimal cost and environmental impacts. This project was designed to use a diversity of energy sources: classical sources of energy, such as gas, and renewable sources including solar energy, gas obtained from biomass, ground energy, waste water and energy regained from ventilation and conditioning. Mazzei et al. (2002) simulated a HVAC system, which integrated the technologies of desiccant cooling and heat recovery in a retail store in Italy, using 3 different software codes, and found that the operating savings is up to 87%, compared to the conventional conditions. Gieseler et al. (2002) studied the cost efficiency of ventilation systems for low-energy buildings with earth-to-air heat exchange and heat recovery and concluded that, in cold climates, the savings of earth heat exchanger in combination with heat recovery unit are substantial, because an appropriate earth heat exchanger can substitute the defroster and the corresponding electric energy. Hamada et al. (2001) studied a low energy home constructed in Japan, using a ground source heat pump, photovoltaic modules, a solar collector, wind mills, stack ventilation, an earth heat exchanger, and exhaust heat recovery. The results show that the annual purchased energy use of this low energy home was reduced by 87.5% and CO_2 emissions by 77%, compared to a typical home. Saitoth and Fujino (2001) studied an energy-independent residential house built in Japan, which used only one sixth of fossil energy and reduced CO_2 emissions by 90%, compared to the conventional house. This house integrated energy efficient technologies such as solar collector, photovoltaic modules, underground heat storage tank, and heat pump. Torcellini et al. (1999) defined an integrated energy design process and showed an actual building whose energy costs were reduced by 63%. These energy savings were achieved by incorporating daylighting, passive solar heating and cooling and energy efficiency strategies into the building design. Zhang et al. (1997) proposed a new type of residence heated by solar energy in winter, and cooled by earth tubes in summer. The simulation results showed that in Tokyo, the room temperature could be maintained at less than 28 °C in summer and more than 18 °C during most hours in winter without HVAC equipment.

In Canada, Natural Resources Canada (NRCan) launched the Advanced Houses program in 1991. The goal of Canada's Advanced Houses program was to build houses that are far more energy efficient, environmentally benign, and healthy for occupants than typical Canadian houses. Under the Advanced Houses Program, 10 prototype homes were constructed across Canada, in which total purchased energy consumption was only 25 percent of a conventional house. The energy efficient technologies and strategies adopted by these houses includes ground source heat pump, radiant heating floor, photovoltaic modules, solar collector, wind turbines, heat recovery from exhaust air and greywater, heat storage, and "free" cooling. In addition, in Hamilton advanced house, a design strategy which consists of the integration of space heating, domestic water heating, ventilation, and heat recovery, is shown to be very effective (Mayo and Sinha,1996).

Table 2.2 presents the energy efficient technologies reviewed from the literature.

Table 2.2 Examples of energy efficient technologies

Name of technology	Reference
Solar collector	Watts and IIp, 2002; Hamada et al, 2001; Saitoh and Fujino, 2001; Zhang et al, 1997; Mayo and Sinha, 1996.
Photovoltaic module	Watts and IIp, 2002; Hamada et al, 2001; Saitoh and Fujino, 2001; Zhang et al, 1997.
Ground source heat pump (GSHP) . • Open loop • Closed loop	Kilkis, 2002; Hamada et al, 2001; Saitoh and Fujino, 2001; Gerbasi, 2000; Mayo and Sinha, 1996.
Air source heat pump	Hamada et al, 2001.
Earth heat exchanger • Air-to-earth heat exchanger • Water-to-earth heat exchanger	Gieseler et al, 2002; Hamaha et al, 2001; Gerbasi, 2000; Zhang et al, 1997; Mayo and Sinha, 1996.
Wind power generator	Kilkis, 2002; Watts and IIp, 2002; Hamada et al, 2001; Mayo and Sinha, 1996.
Radiant heating/cooling system • Radiant floor • Radiant panel	Kilkis, 2002; Hamada et al, 2001; Gerbasi, 2000; Mayo and Sinha, 1996.
Heat storage • Seasonal heat storage • Daily heat storage	Watts and IIp, 2002; Saitoth and Fujino, 2001; Mayo and Sinha, 1996.
Natural ventilation	Hamada et al 2001; Torcellini et al, 1999; Mayo and Sinha, 1996.
Evaporative cooling	Ren et al, 2002; Mayo and Sinha, 1996.
Heat recovery unit • Air-to-air heat exchanger • Water-to–water heat exchanger • Water-to–air heat exchanger	Gieseler, 2002; Watts and IIp, 2002; Gerbasi, 2000; Mayo and Sinha, 1996.
Desiccant cooling	Kanoglu et al, 2004; Mazzei et al, 2002.

2.4 Conclusions

Exergy is an important concept to find the real improvement; it measures the quality of energy sources; it links the physical, engineering world, and the environment, and expresses the true efficiency of engineering systems.

Exergy analysis is based on a combination of the first and second laws of thermodynamics, and can provide for a better understanding of energy utilization processes and locate the real inefficient areas for improvement. In the typical design process of energy efficient HVAC system, energy analysis is highly relied to improve energy use efficiency and environmental performance. In some complex cases, it is not adequate to reflect on the complexity of the HVAC system and the interaction between HVAC system and the building and other systems. Therefore, it can be expected that the exergy analysis combined with energy and entropy analysis should have great applications in the HVAC field.

Indicators to measure sustainability are required. The above literature review shows that all the energy performance indicators for buildings are based on energy and energy analysis. Although some literatures stated that exergy has close linkage with environment and could be a sound environmental indicator, no comprehensive study has emerged in the review to develop or create sustainability indicators based on exergy and exergy analysis (Wall and Gong, 2001b). Clearly, there is a need to develop sustainability indicators based on the second law analysis.

In order to meet the complicated requirements of the indoor environment and make full use of low valued and waste energies, integrated design processes is necessary. Today, the major problem is the lack of available effective integrated heating, cooling and air-conditioning systems and evaluation methods.

CHAPTER 3

DEVELOPMENT OF MODELS OF HVAC COMPONENTS

One objective of this research is to evaluate the performance of integrated HVAC and DHW (domestic hot water) systems. The following components are candidates for the integration within the HVAC-DHW systems.

- earth tube heat exchanger (ETHEx);
- air-to-air heat exchanger (AAHEx);
- water-to-water heat exchanger (WWHEx);
- radiant heating floor (RADF);
- electric or hot water baseboard heater;
- electric or hot water air heater;
- electric or hot water air reheater;
- ground source heat pump (GSHP);
- air source heat pump (ASHP);
- gas-fired boiler with and without economizer;
- electric boiler;
- fan;
- pump;
- domestic hot water tank.

This chapter presents the development of the mathematical models of the above mentioned components, used in the second law analysis. These mathematical models are developed based on the first and second law of thermodynamics. Figure 3.1 shows the position of the above mentioned components in the integrated HVAC–DHW systems of a house.

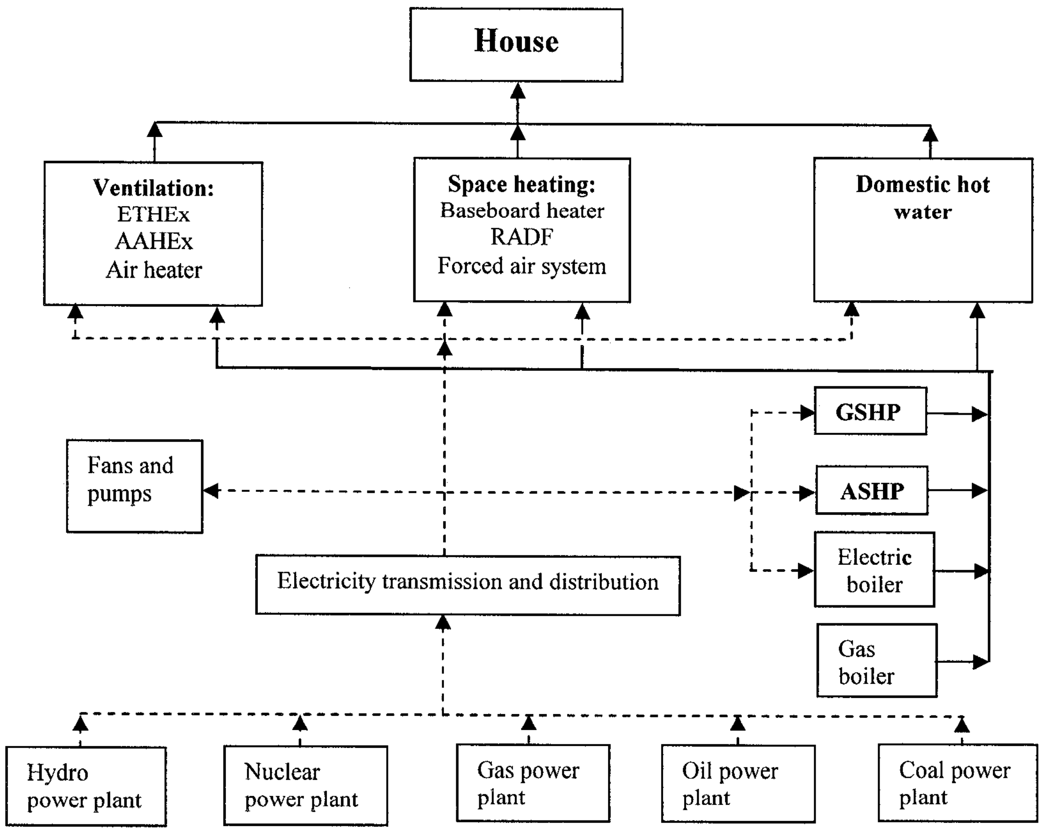

Figure 3.1 Flowchart for the integrated HVAC-DHW system design

Power generation is included in this analysis to reflect the use of primary resources. For instance, in Quebec the contribution of energy sources to the off-site electricity generation is: hydro-electricity 96.7%; natural gas 1.1%; oil 1.1%; nuclear 1.1% (Baouendi, 2003). The overall energy efficiency of the power plant is assumed to be as follows: coal-fired power plant: 37% (Rosen, 2001); natural gas-fired power plant 43.1% (AIE, 1988); oil-fired power plant 33% (Kannan, 2004); nuclear power plant: 30%

(Rosen, 2001); hydro power plant: 80% (Ileri and Gurer, 1998). The transmission and distribution loss is 14%, while the rest 86% is supplied to the end users (Zhang, 1995).

3.1 Second law analysis

3.1.1 Mass balance

Each HVAC component can be modeled as an open thermodynamic system, which can exchange heat, work, and mass with its surroundings. Based on the *principle of mass conservation* applied to an open thermodynamic system, the mass balance equation is:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{sys} \qquad \text{(kg/s)} \qquad (3\text{-}1)$$

where

$\dot{m}_{in}$ is the mass flow rate entering the system, kg/s;

$\dot{m}_{out}$ is the mass flow rate leaving the system, kg/s;

$\Delta \dot{m}_{sys}$ is the mass change rate within the system, kg/s.

In the case of steady-flow process, the mass balance equation is expressed as:

$$\dot{m}_{in} = \dot{m}_{out} \qquad \text{(kg/s)} \qquad (3\text{-}2)$$

3.1.2 Energy balance

The first law of thermodynamics, known as the *principle of energy conservation,* states that energy can neither be created nor destroyed. For an open thermodynamic system, the energy balance equation is:

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{sys} \qquad \text{(kW)} \qquad (3\text{-}3)$$

where

$\dot{E}_{in}$ is the energy flow rate entering the system by heat, work, and mass, kW;

$\dot{E}_{out}$ is the energy flow rate leaving the system by heat, work, and mass, kW;

$\Delta\dot{E}_{sys}$ is the energy change rate within the system, kW.

In the case of steady-flow process, where there is no energy storage within the system, the energy balance equation is expressed as:

$$\dot{E}_{in} = \dot{E}_{out} \qquad \text{(kW)} \tag{3-4}$$

3.1.3 Entropy balance

The second law of thermodynamics introduces a new property called entropy, which measures the molecular disorder of a system. Therefore, entropy can be used to evaluate the quality of energy source. Entropy can be created but can not be destroyed. The *increase of entropy principle* states that, for any system, the entropy change during a process is equal to the net entropy transfer through the system boundary and the entropy generated within this system. This principle is expressed by the entropy balance equation (Cegel and Boles, 2002):

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta\dot{S}_{sys} \qquad \text{(kW/K)} \tag{3-5}$$

$$\dot{S}_{in} = \dot{m}_{in} \cdot s_{in} + \frac{\dot{Q}_{in}}{TK_{sys}} \qquad \text{(kW/K)} \tag{3-6}$$

$$\dot{S}_{out} = \dot{m}_{out} \cdot s_{out} + \frac{\dot{Q}_{out}}{TK_{surr}} \qquad \text{(kW/K)} \tag{3-7}$$

where

$\dot{S}_{in}$ is the entropy flow rate entering the system by heat and mass, kW/K;

$\dot{m}_{in}$ is the mass flow rate entering the system, kg/s;

s_{in} is the specific entropy of the mass entering the system, kJ/kg·K;

$\dot{Q}_{in}$ is the heat flow rate entering the system, kW;

TK_{sys} is the temperature of the system, K;

$\dot{S}_{out}$ is the entropy flow rate leaving the system by heat and mass, kW/K;

$\dot{m}_{out}$ is the mass flow rate leaving the system, kg/s;

s_{out} is the specific entropy of the mass leaving the system, kJ/kg·K;

$\dot{Q}_{out}$ is the heat flow rate leaving the system, kW;

TK_{surr} is the temperature of the surroundings, K;

$\dot{S}_{gen}$ is the entropy generation rate within the system, kW/K;

$\Delta\dot{S}_{sys}$ is the entropy change rate within the system, kW/K.

Work is an organized form of energy. It is free of disorder or entropy. Therefore, there is no entropy transfer associated with energy transfer as work (Cegel and Boles, 2002).

For the steady-flow process, where there is no entropy storage within the system, the entropy balance is written as:

$$\dot{S}_{in} + \dot{S}_{gen} = \dot{S}_{out} \qquad \text{(kW/K)} \qquad (3\text{-}8)$$

Therefore, for an isolated system composed of the HVAC components and the surroundings: $\dot{S}_{gen} = \dot{S}_{out} - \dot{S}_{in} > 0$ (kW/K) (3-9)

3.1.4 Exergy balance

Exergy is defined as the maximum useful work that could be obtained from a system at a given state with respect to a reference environment (dead state) (Cegel and Boles, 2002). In a process or a system, the total amount of exergy is not conserved but destroyed due to internal irreversibilities, such as friction loss and heat transfer due to temperature difference. In this study, the irreversibilities due to heat loss in the piping and duct systems are neglected.

In a thermodynamic system, exergy can be transferred to or from a system in three forms: heat, work and mass flow, which are recognized at the system boundaries. The exergy transfer by heat $\dot{X}_{heat}$ is expressed as (Cegel and Boles, 2002):

$$\dot{X}_{heat} = \left(1 - \frac{TK_0}{TK_{sys}}\right) \cdot \dot{Q} \qquad \text{(kW)} \qquad (3\text{-}10)$$

where:

$\dot{Q}$ is the heat transfer rate crossing the system boundaries, kW;

TK_o is the environmental temperature, K;

TK_{sys} is the temperature of the system, K.

In the case of mechanical work or electricity crossing the system boundaries, exergy transfer $\dot{X}_{work}$ (kW) equals the electricity or mechanical work $\dot{W}$ (kW).

In the case of mass flow crossing the system boundaries, exergy transfer by mass $\dot{X}_{mass}$ is:

$$\dot{X}_{mass} = \dot{m}x \qquad \text{(kW)} \tag{3-11}$$

where:

$\dot{m}$ = mass flow rate crossing the system boundaries, kg/s;

x = exergy per unit mass, kJ/kg.

For a flow stream, the unit mass exergy x can be expressed as (Cegel and Boles, 2002):

$$x = (h - h_0) - TK_0(s - s_0) \qquad \text{(kJ/kg)} \tag{3-12}$$

The exergy change of a flow stream is:

$$\Delta x = x_2 - x_1 = (h_2 - h_1) - TK_0(s_2 - s_1) \qquad \text{(kJ/kg)} \tag{3-13}$$

where:

TK is temperature, K;

h is specific enthalpy, kJ/kg;

s is specific entropy kJ/kg·K.

The subscript "0" indicates the environmental dead state and subscripts "1" and "2" indicate different states of the flow stream.

The *decrease of exergy principle* states that the exergy change of a system during a process is equal to the difference of the net exergy transfer through the system boundary and the exergy destruction within the system boundary as a result of irreversibilities.

$$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{de} = \Delta\dot{X}_{sys} \qquad \text{(kW)} \qquad (3\text{-}14)$$

where

$\dot{X}_{in}$ is the exergy flow rate entering the system, kW;

$\dot{X}_{out}$ is the exergy flow rate leaving the system, kW;

$\dot{X}_{de}$ is the exergy destruction rate within the system, kW;

$\Delta\dot{X}_{sys}$ is the exergy change rate within the system, kW.

In the case of steady-flow process, where there is no change of exergy within the system, the exergy balance equation can be written as:

$$\dot{X}_{in} = \dot{X}_{de} + \dot{X}_{out} \qquad \text{(kW)} \qquad (3\text{-}15)$$

Exergy destruction $\dot{X}_{de}$ in a process can be calculated as product of the entropy generation $\dot{S}_{gen}$ in the same process and the reference environment temperature TK_o:

$$\dot{X}_{de} = TK_0 \cdot \dot{S}_{gen} \qquad \text{(kW)} \qquad (3\text{-}16)$$

Wepfer et al (1979) stated that for a system, such as a HVAC system, the steady-flow exergy balance can also be expressed as:

$$\dot{X}_{supplied} = \dot{X}_{useful} + \dot{X}_{de} + \dot{X}_{lost} \qquad \text{(kW)} \qquad (3\text{-}17)$$

The exergy supplied to the system $\dot{X}_{supplied}$ is partially destroyed inside the system due to the irreversibilies $\dot{X}_{de}$, partially delivered to the outside with the effluents $\dot{X}_{lost}$ and partially used by the system $\dot{X}_{useful}$.

3.15 Efficiency

The energy efficiency of a component or a system is defined as:

$$\eta_I = \frac{\dot{E}_{useful}}{\dot{E}_{supplied}} = 1 - \frac{\dot{E}_{lost}}{\dot{E}_{supplied}} \qquad (3\text{-}18)$$

where

$\dot{E}_{useful}$ is the rate of useful energy needed by a component or a system to function properly, regardless of its source, W;

$\dot{E}_{supplied}$ is the rate of primary energy used by a component or a system, including both on-site and off-site primary energy, kW;

$\dot{E}_{lost}$ is the rate of energy loss in a component or a system; kW.

In order to improve the energy efficiency η_I, the amount of energy lost with the component or system should be reduced, for a given $\dot{E}_{supplied}$.

The exergy efficiency, which provides the realistic measure of performance of engineering system, can be expressed in the following forms (Cegel and Boles, 2002):

$$\eta_2 = \frac{\dot{X}_{useful}}{\dot{X}_{supplied}} = 1 - \frac{\dot{X}_{de} + \dot{X}_{lost}}{\dot{X}_{supplied}} \tag{3-19}$$

where

$\dot{X}_{useful}$ is the rate of useful exergy needed by a component or a system to function properly, regardless of its source, W;

$\dot{X}_{supplied}$ is the rate of exergy provided to a component or a system, both on-site and off-site, kW;

$\dot{X}_{de}$ is the exergy destruction rate within a component or a system, kW;

$\dot{X}_{lost}$ is exergy loss rate through effluents, kW.

To improve the exergy efficiency η_2, the effort should be put on reducing the amount of exergy destroyed inside a component or a system and the amount lost through the effluents.

3.2 Component models

A HVAC system is considered to be a combination of blocks that can interact with other blocks and their surroundings through the heat, work, and mass transfer. A block represents a component of the system. The boundary of a block is dependable on the analysis.

The first step in the application of the second law analysis to a HVAC system is the representation of the system by a series of blocks. Inputs and outputs of each block represent the main process parameters. Once a block diagram is generated with the system boundaries defined, it is possible to assess the mass, energy, entropy and exergy balances based on first principle and correlation–based models from product data.

3.2.1 Earth tube heat exchanger (ETHEx)

Figure 3.2 shows the earth tube heat exchanger which is employed to make use of the geothermal energy to preheat the outdoor air, which is brought into the house for ventilation purposes.

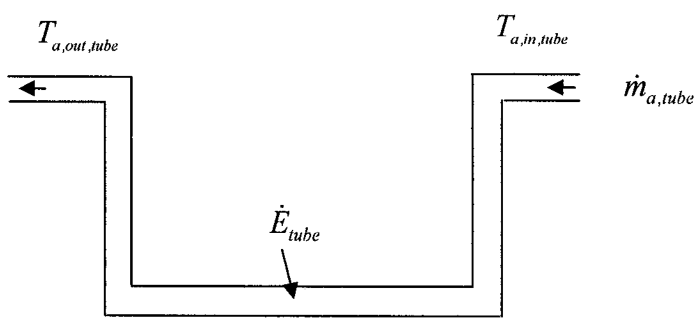

Figure 3.2 Earth tube heat exchanger

<u>Energy balance</u>

$$\dot{E}_{tube} = \dot{m}_{a,tube} \cdot c_{pa} \cdot (T_{a,out,tube} - T_{a,in,tube}) = \alpha \cdot A_{tube} \cdot L \cdot (T_{ground} - T_{a,tube}) / 1000 \quad \text{(kW)} \qquad (3\text{-}20)$$

where

$\dot{E}_{tube}$ is the heat transfer rate between earth and the air in the earth tube, kW;

$\dot{m}_{a,tube}$ is the mass flow rate of air in the earth tube, kg/s;

c_{pa} is the specific heat of air at $T = (T_i + T_o)/2$, kJ/kg·K ; it is assumed constant in the calculation;

$T_{a,in,tube}$ is the temperature of air entering the tube, °C;

$T_{a,out,tube}$ is the temperature of air leaving the tube, °C;

α is the heat transfer coefficient at the tube inside surface, W/m^2·K, which is determined as a function of air velocity (Hokoi et al, 1998),

$$\alpha = 6.15 + 4.18 \cdot v \quad (\text{W/m}^2\cdot\text{K}) \tag{3-21}$$

v is the air velocity in the earth tube heat exchanger,

$$v = \dot{V}_{a,fresh} / A_{tube} \quad (\text{m/s}) \tag{3-22}$$

A_{tube} is the section area of the earth tube, m^2;

$\dot{V}_{a,fresh}$ is the volume flow rate of the air in the earth tube, m^3/s;

$$\dot{V}_{a,fresh} = A \cdot H \cdot \beta / 3600 \quad (\text{m}^3/\text{s}) \tag{3-23}$$

A is the total floor area of the house, m^2;

β is the ventilation air change rate of the house, ach;

H is the height of the room, m;

L is the length of the earth tube, m;

T_{ground} is the temperature of earth deep in the ground, °C;

$T_{a,tube}$ is the average temperature of earth tube heat exchanger, °C.

$$T_{a,tube} = (T_{a,in,tube} + T_{a,out,tube}) / 2 \quad (°\text{C}) \tag{3-24}$$

Entropy generation

$$\dot{S}_{gen,tube} = \dot{m}_{a,tube} \cdot (s_{a,out,tube} - s_{a,in,tube}) - \dot{E}_{tube} / TK_{a,tube} \quad (\text{kW/K}) \tag{3-25}$$

where

$\dot{S}_{gen,tube}$ is the entropy generation rate within the earth tube heat exchanger, kW/K;

$s_{a,in,tube}$ is the specific entropy of the air entering the tube kJ/kg·K, at $T = T_{a,in,tube}$, $P = 101$ kPa;

$s_{a,out,tube}$ is the specific entropy of air leaving the tube kJ/kg·K, at $T = T_{a,out,tube}$, $P = 101$ kPa.

$$TK_{a,tube} = 273.15 + (T_{a,in,tube} + T_{a,out,tube})/2 \quad \text{(K)} \qquad (3\text{-}26)$$

Exergy destruction

$$\dot{X}_{de,tube} = TK_o \cdot \dot{S}_{gen,tube} \qquad \text{(kW)} \qquad (3\text{-}27)$$

where

$\dot{X}_{de,tube}$ is the exergy destruction rate within the earth tube, kW;

TK_o is the reference environmental temperature, (K).

The operation of the earth tube heat exchanger is controlled in the following sequence in the winter heating mode: when the outdoor air temperature is lower than the ground temperature, the ventilation air is drawn through the earth tube; otherwise, the tube is closed, and the ventilation air is brought directly to the HVAC system. The summer cooling mode is not included in this study.

3.2.2 Air-to-air heat exchanger (AAHEx)

An air-to-air heat exchanger is used to recover heat from the exhaust air and preheat the incoming ventilation air (Figure 3.3).

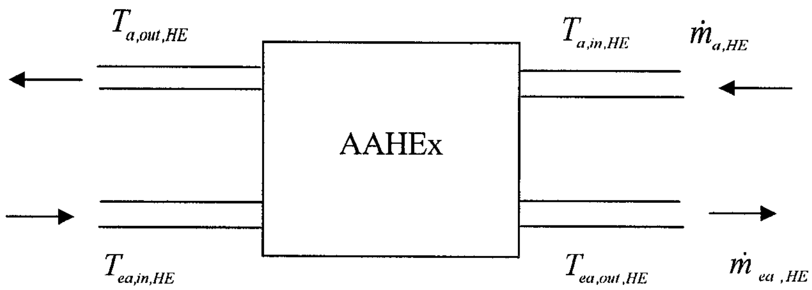

Figure 3.3 Air-to-air heat exchanger

A balanced ventilation system is assumed to be installed in the house. Therefore, the mass flow rate of incoming ventilation air and exhaust air in the heat exchanger are equal:

$$\dot{m}_{a,HE} = \dot{m}_{ea,HE} \quad \text{(kg/s)} \tag{3-28}$$

Energy balance:

$$\dot{m}_{a,HE} \cdot c_{pa} \cdot (T_{a,out,HE} - T_{a,in,HE}) = \dot{m}_{ea,HE} \cdot c_{pa} \cdot (T_{ea,in,HE} - T_{ea,out,HE}) \text{ (kW)} \tag{3-29}$$

where

$\dot{m}_{a,HE}$ is mass flow rate of the ventilation air in the AAHEx, kg/s;

$\dot{m}_{ea,HE}$ is mass flow rate of the exhaust air in the AAHEx, kg/s;

$T_{a,in,HE}$ is the temperature of ventilation air entering the AAHEx, °C;

$T_{a,out,HE}$ is the temperature of ventilation air leaving the AAHEx, °C;

$T_{ea,in,HE}$ is the temperature of exhaust air entering the AAHEx, °C;

$T_{ea,out,HE}$ is the temperature of exhaust air leaving the AAHEx, °C;

The sensible heat recovery efficiency (η_{HE}) is given by:

$$\eta_{HE} = \frac{\dot{Q}_{recovered}}{\dot{Q}_{max,potential}} \tag{3-30}$$

where

$\dot{Q}_{recovered}$ is the heat flow rate recovered from the exhaust air stream, and assumed to be equal to the heat flow rate transferred to the ventilation air; there are no heat losses between the two streams.

$\dot{Q}_{max,potential}$ is the maximum potential heat recovery rate which can be obtained when the exhaust air leaves the heat exchanger at the temperature of incoming ventilation air.

Therefore, the sensible heat recovery efficiency (η_{HE}) can be rewritten as:

$$\eta_{HE} = (T_{a,out,HE} - T_{a,in,HE})/(T_{ea,in,HE} - T_{a,in,HE}) = (T_{ea,in,HE} - T_{ea,out,HE})/(T_{ea,in,HE} - T_{a,in,HE}) \tag{3-31}$$

<u>Entropy generation:</u>

$$\dot{S}_{gen,HE} = \dot{m}_{a,HE} \cdot (s_{a,out,HE} - s_{a,in,HE}) + \dot{m}_{ea,HE} \cdot (s_{ea,out,HE} - s_{ea,in,HE}) \quad \text{(kW/K)} \tag{3-32}$$

where

$\dot{S}_{gen,HE}$ is the entropy generation rate within the AAHEx, kW/K;

$s_{a,in,HE}$ is the specific entropy of the ventilation air entering the AAHEx, kJ/kg·K, at $T = T_{a,in,HE}$, $P = 101$ kPa;

$s_{a,out,HE}$ is the specific entropy of the ventilation air leaving the AAHEx, kJ/kg·K, at $T = T_{a,out,HE}$, $P = 101$ kPa;

$s_{ea,in,HE}$ is the specific entropy of the exhaust air entering the AAHEx ,kJ/kg·K, at $T = T_{ea,in,HE}$, $P = 101$ kPa;

$s_{ea,out,HE}$ is the specific entropy of the exhaust air leaving the AAHEx, kJ/kg·K, at $T = T_{ea,out,HE}$, $P = 101$ kPa;

$\dot{S}_{gen,HE}$ is the entropy generation rate within the AAHEx, kW/K;

Exergy destruction:

$$\dot{X}_{de,HE} = TK_o \cdot \dot{S}_{gen,HE} \qquad \text{(kW)} \tag{3-33}$$

where $\dot{X}_{de,HE}$ is the exergy destruction rate within the AAHEx, kW.

3.2.3 Electric air heater

An electric air heater is employed to heat the supply air to the required supply air temperature before the air enters the house.

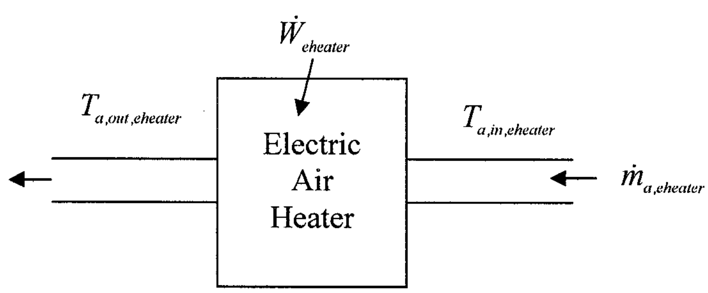

Figure 3.4 Electric air heater

Energy balance:

$$\dot{W}_{eheater} = \dot{m}_{a,eheater} \cdot c_{pa} \cdot (T_{a,out,eheater} - T_{a,in,eheater}) \quad \text{(kW)} \tag{3-34}$$

where

$\dot{W}_{eheater}$ is the electric power input of the electric air heater, kW;

$\dot{m}_{a,eheater}$ is the mass flow rate of the air through the electric air heater, kg/s;

$T_{a,in,eheater}$ is the temperature of the air entering the electric air heater, ^{o}C;

$T_{a,out,eheater}$ is the temperature of the air leaving the electric air heater, ^{o}C.

Entropy generation

$$\dot{S}_{gen,eheater} = \dot{m}_{a,eheater} \cdot (s_{a,out,eheater} - s_{a,in,eheater}) \quad \text{(kW/K)} \tag{3-35}$$

where

$\dot{S}_{gen,eheater}$ is the entropy generation rate within the electric air heater, kW/K;

$s_{a,in,eheater}$ is the specific entropy of the air entering the electric air heater kJ/kg·K, at $T = T_{a,in,eheater}$, $P = 101$ kPa;

$s_{a,out,eheater}$ is entropy of the air leaving the electric air heater, kJ/kg·K, at $T = T_{a,out,eheater}$, $P = 101$ kPa.

Exergy destruction

$$\dot{X}_{de,eheater} = TK_{o} \cdot \dot{S}_{gen,eheater} \quad \text{(kW)} \tag{3-36}$$

where $\dot{X}_{de,eheater}$ is the exergy destruction rate within the electric air heater, kW.

3.2.4 Hot water air heater

A hot water air heater is an alternative to electric air heater. It is used to heat the supply air to the required supply air temperature before the air enters the house. Figure 3.5 shows the hot water air heater with its inputs and outputs.

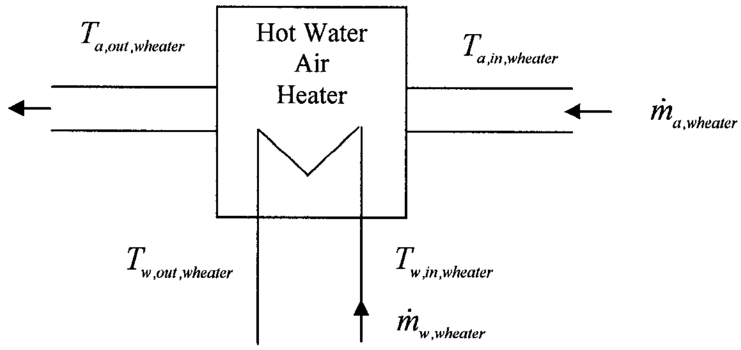

Figure 3.5 Hot water air heater

<u>Energy balance</u>

$$\dot{Q}_{wheater} = \dot{m}_{w,wheater} \cdot c_{pw} \cdot (T_{w,in,wheater} - T_{w,out,wheater}) = \dot{m}_{a,wheater} \cdot c_{pa} \cdot (T_{a,out,wheater} - T_{a,in,wheater}) \quad \text{(kW)} \qquad (3\text{-}37)$$

where

$\dot{Q}_{wheater}$ is the heat transfer rate from hot water to the air in the hot water air heater, kW;

$\dot{m}_{a,wheater}$ is the mass flow rate of the air through the hot water air heater, kg/s;

$\dot{m}_{w,wheater}$ is the mass flow rate of the hot water through the hot water air heater, kg/s;

$T_{a,in,wheater}$ is the temperature of the air entering the hot water air heater, ^{o}C;

$T_{a,out,wheater}$ is the temperature of the air leaving the hot water air heater, ^{o}C;

$T_{w,in,wheater}$ is the temperature of the water entering the hot water air heater, ^{o}C;

$T_{w,out,wheater}$ is the temperature of the water leaving the hot water air heater, ^{o}C.

Entropy generation

$$\dot{S}_{gen,wheater} = \dot{m}_{a,wheater} \cdot (s_{a,out,wheater} - s_{a,in,wheater}) + \dot{m}_{w,heater} \cdot (s_{w,out,wheater} - s_{w,in,wheater}) \text{ (kW/K)} \quad (3\text{-}38)$$

where

$\dot{S}_{gen,wheater}$ is the entropy generation rate within the hot water air heater, kW/K;

$s_{w,in,wheater}$ is the specific entropy of the water entering the hot water air heater (kJ/kg·K), at $T = T_{w,in,wheater}$, $P = 101$ KPa;

$s_{w,out,wheater}$ is the specific entropy of the water leaving the hot water air heater (kJ/kg·K), at $T = T_{w,out,wheater}$, $P = 101$ KPa;

$s_{a,in,wheater}$ is entropy of the air entering the hot water air heater kJ/kg·K, at $T = T_{a,in,wheater}$, $P = 101$ kPa;

$s_{a,out,wheater}$ is entropy of the air leaving the hot water air heater kJ/kg·K, at $T = T_{a,out,wheater}$, $P = 101$ kPa.

Exergy destruction

$$\dot{X}_{de,wheater} = TK_o \cdot \dot{S}_{gen,wheater} \text{ (kW)} \quad (3\text{-}39)$$

where $\dot{X}_{de,wheater}$ is the exergy destruction rate within the hot water air heater, kW.

3.2.5 Electric baseboard heaters

Electric baseboard heaters are commonly used in Quebec for space heating in winter. Figure 3.6 shows the diagram of an electric baseboard heater used for the analysis.

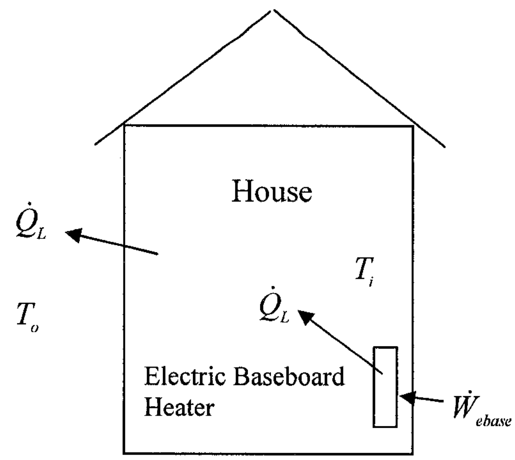

Figure 3.6 Electric baseboard heater

Energy balance

$$\dot{W}_{ebase} = \dot{Q}_L \quad \text{(kW)} \tag{3-40}$$

where

$\dot{W}_{ebase}$ is the electric power input for the electric baseboard heater, kW;

$\dot{Q}_L$ is the heating load of the house, kW.

Entropy generation

$$\dot{S}_{gen,ebase} = \dot{Q}_L / TK_i \quad \text{(kW/K)} \tag{3-41}$$

where

$\dot{S}_{gen,ebase}$ is the entropy generation rate within the electric baseboard heater, kW/K;

$TK_i = 273.15 + T_i$ (K);

T_i is the indoor air temperature, °C.

Exergy destruction

$$\dot{X}_{de,ebase} = TK_o \cdot \dot{S}_{gen,ebase} \qquad \text{(kW)} \tag{3-42}$$

where $\dot{X}_{de,ebase}$ is the exergy destruction rate within the electric baseboard heater, kW.

3.2.6 Hot water baseboard heater

Hot water baseboard heaters are used for space heating. Figure 3.7 shows the analysis diagram of a hot water baseboard heater.

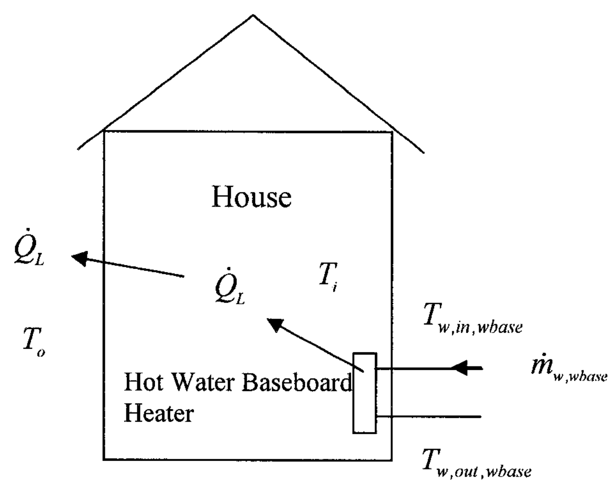

Figure 3.7 Hot water baseboard heater

Energy balance

$$\dot{Q}_L = \dot{m}_{w,wbase} \cdot c_{pw} \cdot (T_{w,in,wbase} - T_{w,out,wbase}) \quad \text{(kW)} \tag{3-43}$$

$\dot{m}_{w,wbase}$ is the mass flow rate of water through the hot water baseboard heater, kg/s;

$T_{w,in,wbasae}$ is the temperature of water entering the hot water baseboard heater, °C;

$T_{w,out,wbasae}$ is the temperature water leaving the hot water baseboard heater, °C.

Entropy generation

$$\dot{S}_{gen,wbase} = \dot{m}_{wbase} \cdot (s_{w,out,wbase} - s_{w,in,wbase}) + \dot{Q}_L / TK_i \qquad \text{(kW/K)} \qquad (3\text{-}44)$$

where

$\dot{S}_{gen,wbase}$ is the entropy generation rate within the hot water baseboard heater, kW/K;

$s_{w,in,wbase}$ is the specific entropy of water entering the heater (kJ/kg·K), at $T = T_{w,in,wbase}$, $P = 101$ KPa;

$s_{w,out,wbase}$ is the specific entropy of water leaving the heater (kJ/kg·K), at $T = T_{w,out,wbase}$, $P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,wbase} = TK_o \cdot \dot{S}_{gen,wbase} \quad \text{(kW)} \qquad (3\text{-}45)$$

where $\dot{X}_{de,wbase}$ is the exergy destruction rate within the hot watetr baseboard heater, kW.

3.2.7 Natural gas-fired boiler

A natural gas-fired boiler is commonly used for space heating, ventilation air heating, and domestic hot water heating. Figure 3.8 presents the analysis diagram of natural gas-fired boiler.

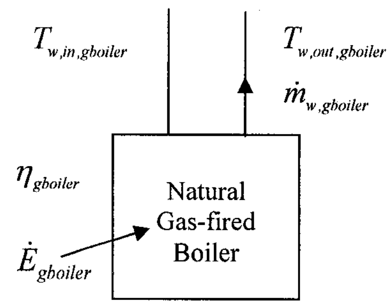

Figure 3.8 Nature gas-fired boiler

Energy balance

$$\dot{E}_{gboiler} = \dot{m}_{w,gboiler} \cdot c_{pw} \cdot (T_{w,out,gboiler} - T_{w,in,gboiler}) / \eta_{gboiler} \quad \text{(kW)} \tag{3-46}$$

where

$\dot{E}_{gboiler}$ is the rate of the primary energy supplied to the boiler, kW;

$\dot{m}_{w,gboiler}$ is the mass flow rate of the water going through the hot boiler, kg/s;

c_{pw} is the specific heat of water at $T = 80$ °C, $P = 101$ kPa, kJ/kg·K ; it is assumed constant in the calculation;

$T_{w,in,gboiler}$ is the temperature of the water entering the boiler, °C;

$T_{w,out,gboiler}$ is the temperature of the water leaving the boiler, °C;

$\eta_{gboiler}$ is the energy efficiency of the boiler.

Entropy generation

$$\dot{S}_{gengboiler} = \dot{m}_{w,gboiler} \cdot (s_{w,outgboiler} - s_{w,ingboiler}) - \dot{E}_{gboiler} / TK_{flame} + \dot{E}_{gboiler} \cdot (1 - \eta_{gboiler}) / TK_o \quad \text{(kW/K)} \tag{3-47}$$

where

$\dot{S}_{gen,gboiler}$ is the entropy generation rate within the natural gas-fired boiler, kW/K;

$s_{w,in,gboiler}$ is specific entropy of the water entering the boiler kJ/kg·K, at $T = T_{w,in,gboiler}$, $P = 101$ KPa;

$s_{w,out,gboiler}$ is the specific entropy of the water leaving the boiler kJ/kg·K, at $T = T_{w,out,gboiler}$, $P = 101$ KPa.

TK_{flame} is the temperature of the flame, K. TK_{flame} depends on many factors, such as, air and fuel temperature, oxygen content of the air. In this study, the adiabatic flame temperature TK_{flame} is assumed to be 2200 K for both on-site and off-site combustion (Bennett, 2002). The last term of equation (3-47) gives the entropy generation which corresponds to the energy losses due to the flue gas through chimney.

Exergy destruction

$$\dot{X}_{de,gboiler} = TK_o \cdot \dot{S}_{gen,gboiler} \quad \text{(kW)} \tag{3-48}$$

where $\dot{X}_{de,gboiler}$ is the exergy destruction rate within the natural gas-fired boiler, kW.

3.2.8 Natural gas-fired boiler with economizer

An economizer is used to recover heat energy from the exhaust flue gases and preheat the domestic hot water, and thus, it reduces the energy used for the domestic hot water heating. Figure 3.9 shows the natural gas-fired boiler with an economizer with its inputs and outputs.

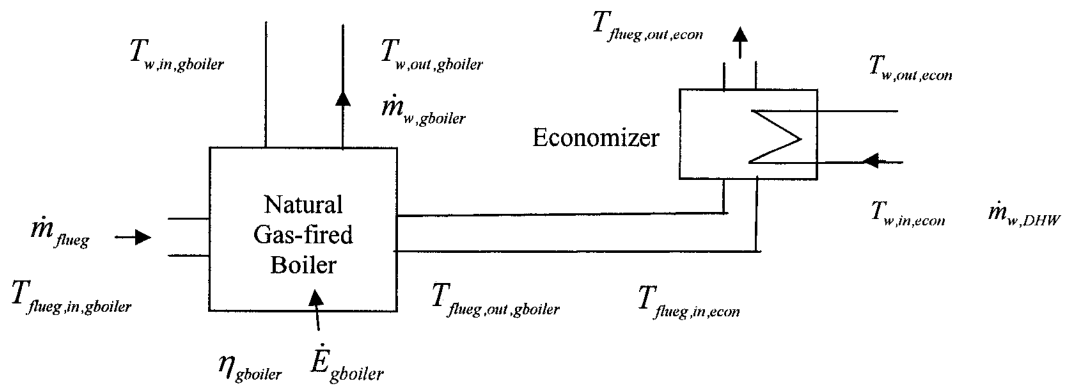

Figure 3.9 Natural gas-fired boiler with economizer

The energy balance within the boiler

$$\dot{E}_{gboiler} = \dot{m}_{w,gboiler} \cdot c_{pw} \cdot (T_{w,out,gboiler} - T_{w,in,gboiler}) + \dot{m}_{flueg} \cdot c_{pa} \cdot (T_{fluegout,gboiler} - T_{fluegin,gboiler}) \text{ (kW)} \quad (3\text{-}49)$$

$$\dot{E}_{gboiler} \cdot (1 - \eta_{gboiler}) = \dot{m}_{flueg} \cdot c_{pa} \cdot (T_{flueg,out,gboiler} - T_{flueg,in,gboiler}) \text{ (kW)} \quad (3\text{-}50)$$

where

$\dot{m}_{flueg}$ is the mass flow rate of the flue gas through the boiler, kg/s;

$T_{flueg,in,gboiler}$ is the temperature of the flue gas entering the boiler, °C;

$T_{flueg,out,gboiler}$ is the temperature of the flue gas leaving the boiler, °C;

Entropy generation within the boiler

$$\dot{S}_{gen,gboiler} = \dot{m}_{w,gboiler} \cdot (s_{w,out,gboiler} - s_{w,in,gboiler}) - \dot{E}_{gboiler} / TK_{flame} + \dot{m}_{flueg} \cdot (s_{flueg,out,gboiler} - s_{flueg,in,gboiler}) \text{(kW/K)} \quad (3\text{-}51)$$

where

$\dot{S}_{gen,gboiler}$ is the entropy generation rate within the natural gas-fired boiler, kW/K;

$s_{flueg,in,gboiler}$ is the specific entropy of the flue gas (kJ/kg·K), at $T = T_{flueg,in,gboiler}$, $P = 101$ KPa;

$s_{flueg,out,gboiler}$ is the specific entropy of the flue gas (kJ/kg·K), at $T = T_{flueg,out,gboiler}$, $P = 101$ KPa.

<u>Exergy destruction within the boiler</u>

$$\dot{X}_{de,gboiler} = TK_o \cdot \dot{S}_{gen,gboiler} \quad \text{(kW)} \tag{3-52}$$

where $\dot{X}_{de,gboiler}$ is the exergy destruction rate within the natural gas-fired boiler, kW.

<u>Energy Balance within the economizer</u>

$$\dot{m}_{flueg} \cdot c_{pa} \cdot (T_{flueg,in,econ} - T_{flue,out,econ}) = \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,econ} - T_{w,in,econ}) \quad \text{(kW)} \tag{3-53}$$

$$\eta_{econ} = \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,econ} - T_{w,in,econ}) / \dot{m}_{flueg} \cdot c_{pa} \cdot (T_{flue,in,econ} - T_{w,in,econ}) \quad \text{(kW)} \tag{3-54}$$

where

η_{econ} is the sensible heat recovery rate of the economizer.

<u>Entropy generation within the economizer</u>

$$\dot{S}_{gen,econ} = \dot{m}_{w,DHW} \cdot (s_{w,out,econ} - s_{w,in,econ}) + \dot{E}_{gboiler} \cdot (1-\eta_{gboiler}) \cdot (1-\eta_{econ}) / TK_o + \dot{m}_{flueg} \cdot (s_{flueg,out,econ} - s_{flueg,in,econ}) \quad \text{(kW/K)} \tag{3-55}$$

where

$\dot{S}_{gen,econ}$ is the entropy generation rate within the economizer, kW/K;

$\dot{m}_{w,DHW}$ is the mass flow rate of the domestic hot water, kg/s;

$s_{flueg,in,econ}$ is the specific entropy of the flue gas entering the economizer (kJ/kg·K), at $T = T_{flueg,in,econ}$, $P = 101$ KPa;

$s_{flueg,out,econ}$ is the specific entropy of the flue gas leaving the economizer(kJ/kg·K), at $T = T_{flueg,out,econ}$, $P = 101$ KPa.

$s_{w,in,econ}$ is the specific entropy of water entering the economizer (kJ/kg·K), at $T = T_{w,in,econ}, P = 101$ KPa;

$s_{w,out,econ}$ is the specific entropy of water leaving the economizer (kJ/kg·K), at $T = T_{w,out,econ}, P = 101$ KPa.

Exergy destruction within the economizer

$$\dot{X}_{de,econ} = TK_o \cdot \dot{S}_{gen,econ} \quad \text{(kW)} \tag{3-56}$$

where $\dot{X}_{de,econ}$ is the exergy destruction rate within the economizer, kW.

3.2.9 Electric boiler

An electric boiler is used to produce hot water for space heating. Figure 3.10 shows the diagram of electric boiler for analysis.

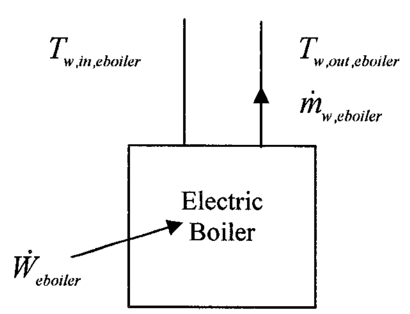

Figure 3.10 Electric boiler

Energy balance

$$\dot{W}_{eboiler} = \dot{m}_{w,eboiler} \cdot c_{pw} \cdot (T_{w,out,eboiler} - T_{w,in,eboiler}) \quad \text{(kW)} \tag{3-57}$$

where

$\dot{W}_{eboiler}$ is the electric power input to the boiler, kW;

$\dot{m}_{w,eboiler}$ is the mass flow rate of the water through the boiler, kg/s;

$T_{w,in,gboiler}$ is the temperature of the water entering the boiler, °C;

$T_{w,out,gboiler}$ is the temperature of the water leaving the boiler, °C.

Entropy generation

$$\dot{S}_{gen,eboiler} = \dot{m}_{w,eboiler} \cdot (s_{w,out,eboiler} - s_{w,in,eboiler}) \quad \text{(kW/K)} \quad (3\text{-}58)$$

where

$\dot{S}_{gen,eboiler}$ is the entropy generation rate within the electric boiler, kW/K;

$s_{w,in,eboiler}$ is the specific entropy of the water entering the boiler (kJ/kg·K), at $T = T_{w,in,eboiler}$, $P = 101$ KPa;

$s_{w,out,eboiler}$ is the specific entropy of the water leaving the boiler (kJ/kg·K), at $T = T_{w,out,eboiler}$, $P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,eboiler} = TK_o \cdot \dot{S}_{gen,eboiler} \quad \text{(kW)} \quad (3\text{-}59)$$

where $\dot{X}_{de,eboiler}$ is the exergy destruction rate within the electric boiler, kW.

3.2.10 Electric DHW tank

This component is used to heat domestic hot water electrically. Analysis can be conducted based on the analysis diagram (Figure 3.11).

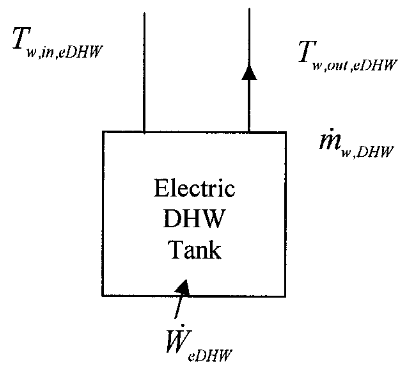

Figure 3.11 Electric DHW tank

Energy balance

$$\dot{W}_{eDHW} = \dot{m}_{w,DHW} \cdot C_{pw} \cdot (T_{w,out,eDHW} - T_{w,in,eDHW}) \quad \text{(kW)} \tag{3-60}$$

where

$\dot{W}_{eDHW}$ is the electric power input to the DHW tank, kW;

$\dot{m}_{w,DHW}$ is the mass flow rate of the domestic hot water, kg/s;

$T_{w,in,eDHW}$ is the temperature of the water entering the DHW tank, ^{o}C;

$T_{w,out,eDHW}$ is the temperature of the water leaving the DHW tank, ^{o}C.

Entropy generation

$$\dot{S}_{gen,eDHW} = \dot{m}_{w,DHW} \cdot (s_{w,out,eDHW} - s_{w,in,eDHW}) \quad \text{(kW/K)} \tag{3-61}$$

where

$\dot{S}_{gen,eDHW}$ is the entropy generation rate within the electric DHW tank, kW/K;

$s_{w,in,eDHW}$ is the specific entropy of the water entering the electric DHW tank (kJ/kg·K), at $T = T_{w,in,eDHW}$, $P = 101$ KPa;

$s_{w,out,eDHW}$ is the specific entropy of the water leaving the electric DHW tank (kJ/kg·K), at $T = T_{,out,eDHW}$, $P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,eDHW} = TK_o \cdot \dot{S}_{gen,eDHW} \quad \text{(kW)} \tag{3-62}$$

where $\dot{X}_{de,eDHW}$ is the exergy destruction rate within the electric DHW tank, kW.

3.2.11 Hot water DHW tank

A hot water DHW tank is a water-to-water heat exchanger, which is used to produce domestic hot water. The heat source is the high temperature water from boiler (Figure 3.12).

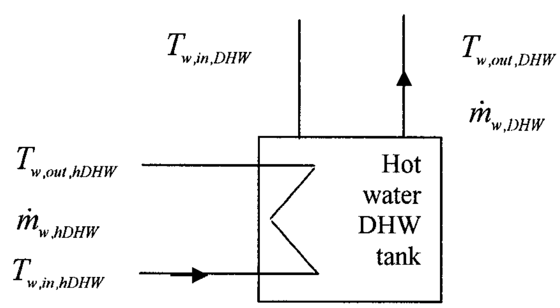

Figure 3.12 Hot water DHW tank

Heat balance

$$\dot{Q}_{DHW} = \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) = \dot{m}_{w,hDHW} \cdot c_{pw} \cdot (T_{w,in,hDHW} - T_{w,out,hDHW}) \quad \text{(kW)} \tag{3-63}$$

where

$\dot{Q}_{DHW}$ is the heat exchange rate in the DHW tank, kW;

$\dot{m}_{w,hDHW}$ is the mass flow rate of the water through the warmer side of the DHW tank, kg/s;

$T_{w,in,hDHW}$ is the temperature of the water entering the warmer side of the DHW tank, °C;

$T_{w,out,hDHW}$ is the temperature of the water leaving the warmer side of the DHW tank, °C;

$T_{w,in,DHW}$ is the temperature of the water entering the cooler side of the DHW tank, °C;

$T_{w,out,DHW}$ is the temperature of the water leaving the cooler side of the DHW tank, °C.

Entropy generation

$$\dot{S}_{gen,DHW} = \dot{m}_{w,DHW} \cdot (s_{w,out,DHW} - s_{w,in,DHW}) + \dot{m}_{w,hDHW} \cdot (s_{w,out,hDHW} - s_{w,in,hDHW}) \quad \text{(kW/K)} \qquad (3\text{-}64)$$

where:

$\dot{S}_{gen,DHW}$ is the entropy generation rate within the hot water DHW tank, kW/K;

$s_{w,in,DHW}$ is the specific entropy of the water entering the cooler side of the DHW tank, kJ/kg·K, at $T = T_{w,in,DHW}$, $P = 101$ KPa;

$s_{w,out,DHW}$ is the specific entropy of water leaving the cooler side of the DHW tank, kJ/kg·K, at $T = T_{w,out,DHW}$, $P = 101$ KPa.

$s_{w,in,hDHW}$ is the specific entropy of the water entering the warmer side of the DHW tank, kJ/kg·K, at $T = T_{w,in,hDHW}$, $P = 101$ KPa;

$s_{w,out,hDHW}$ is the specific entropy of the water leaving the warming side of the DHW tank, kJ/kg·K, at $T = T_{w,out,hDHW}$, $P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,DHW} = TK_o \cdot \dot{S}_{gen,DHW} \qquad \text{(kW)} \qquad (3\text{-}65)$$

where $\dot{X}_{de,DHW}$ is the exergy destruction rate within the hot water DHW tank, kW.

3.2.12 Radiant heating floor

A radiant heating floor works like any hydraulic heating system, but instead of distributing heat through a convective baseboard or wall-mounted radiator, a pump circulates hot water through the tubes in the floor. Radiant heating floor uses lower water temperatures than hydraulic baseboards. This makes the radiant heating floor a good match with a low-temperature heat source like an air- or ground-source heat pump, or even an active solar system. Figure 3.13 shows a radiant heating floor with its inputs and outputs.

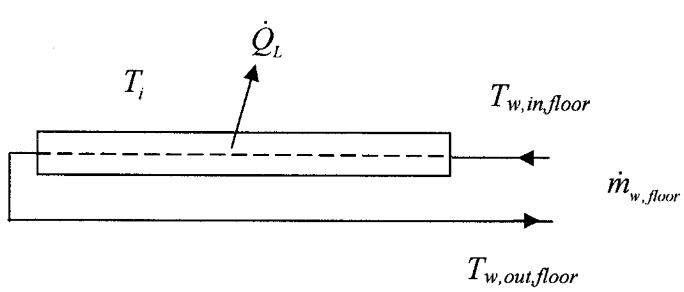

Figure 3.13 Radiant heating floor

Energy balance

$$\dot{Q}_L = \dot{m}_{w,floor} \cdot c_{pw} \cdot (T_{w,in,floor} - T_{w,out,floor}) \qquad \text{(kW)} \qquad (3\text{-}66)$$

where

$\dot{Q}_L$ is the heating load of the space, kW;

$\dot{m}_{w,floor}$ is the mass flow rate of the water through the heating floor, kg/s;

$T_{w,in,floor}$ is the temperature of the water entering the heating floor, ^oC;

$T_{w,out,floor}$ is the temperature of the water leaving the heating floor, ^oC.

Entropy generation

$$\dot{S}_{gen,floor} = \dot{m}_{w,floor} \cdot (s_{w,out,floor} - s_{w,in,floor}) + \dot{Q}_L / TK_i \qquad \text{(kW/K)} \tag{3-67}$$

where:

$\dot{S}_{gen,floor}$ is the entropy generation rate within the radiant heating floor, kW/K;

$s_{w,in,floor}$ is the specific entropy of the water entering the heating floor kJ/kg·K, at $T = T_{w,in,floor}, P = 101$ KPa;

$s_{w,out,floor}$ is the specific entropy of the water leaving the heating floor kJ/kg·K, at $T = T_{w,out,floor}, P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,floor} = TK_o \cdot \dot{S}_{gen,floor} \qquad \text{(kW)} \tag{3-68}$$

where $\dot{X}_{de,floor}$ is the exergy destruction rate within the radiant heating floor, kW.

3.2.13 Ground source heat pump (GSHP)

Because the temperature of the deep soil remains relatively constant year-round, in this research, a ground source heat pump was selected to make use of the geothermal energy to heat the space and preheat the domestic hot water.

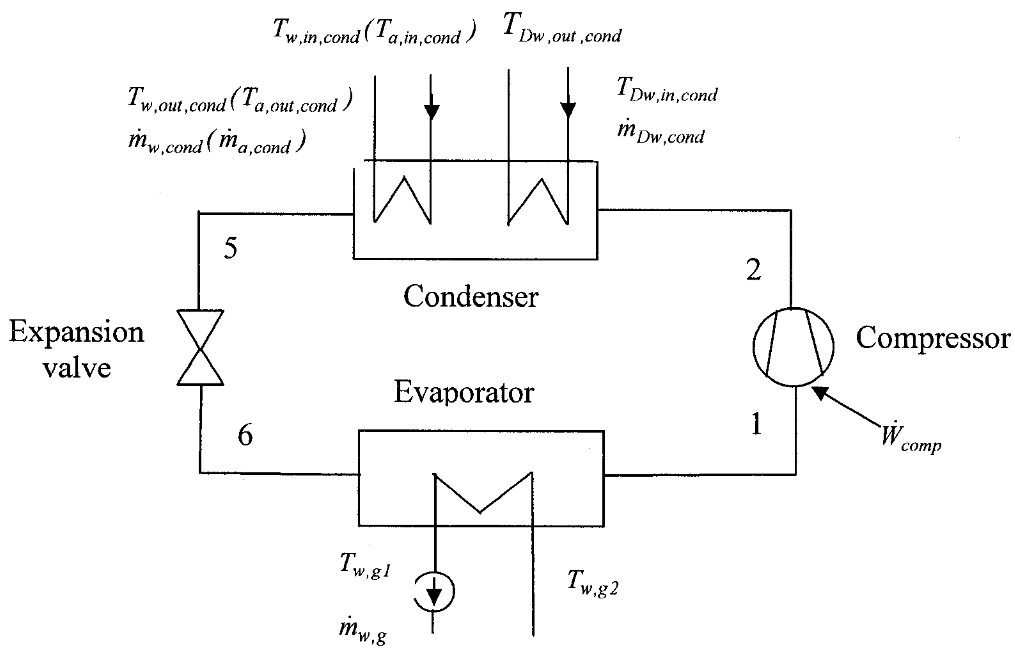

Figure 3.14 Schematic diagram of a ground source heat pump

The heat-pump operation cycle is presented in the temperature-specific entropy (T-s) diagram in Figure 3.15. The refrigerant R-134a at state 6 is heated in the evaporator at constant temperature and pressure (T_{evap} and P_{evap}, respectively). It becomes saturated vapor at evaporator outlet (state 1). Then a vapor compressor is used to increase the refrigerant pressure before entering the condenser. If the compressing process was isentropic ($s = constant$), the state before entering the condenser would be state 3. However, due to the irreversibilities in the compressor, there is entropy generation, and the actual process in the compressor is 1 to 2. Inside the condenser, the refrigerant is cooled at constant pressure P_{cond}. The refrigerant is first cooled to saturated vapor (State 4), and then at the condenser outlet the refrigerant becomes saturated liquid (state 5). Finally, an expansion valve is used to decrease the pressure of the refrigerant to P_{evap} before the evaporator inlet (state 6).

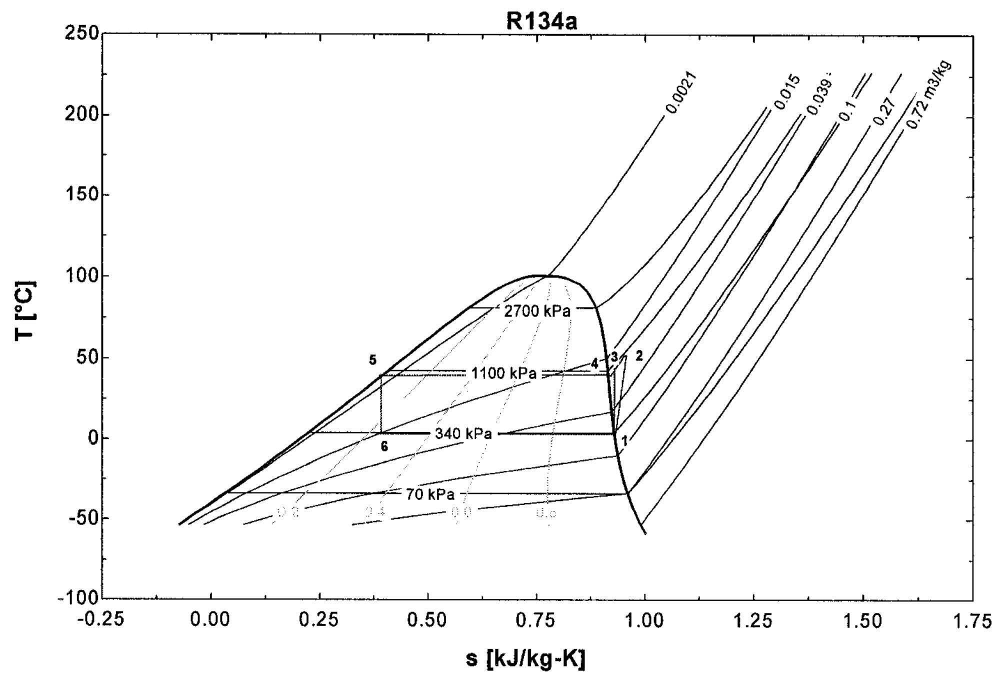

Figure 3.15 Representation of the thermodynamic cycle of GSHP in temperature-specific entropy diagram

The refrigerant state parameters s_{ri} and h_{ri} (i=1 to 6) are estimated in terms of temperature, pressure, or quality in the case of saturated state. In the following equations, the refrigerant parameters have the subscript "r", the water parameters have the subscript "w", and the air parameters have the subscript "a". The following assumptions are used:

(1) The temperature difference between the two fluids (i.e. water and refrigerant) within evaporator and condenser is $\Delta T_{cond} = \Delta T_{evap} = 5$ °C (Badescu, 2002);

(2) The temperature difference between the water entering and leaving the evaporator is $\Delta T_{w,g} = 4$ °C (Badescu, 2002).

These two assumptions lead to the following equation:

$$T_{r4} = T_{w,out,cond} + \Delta T_{cond} = T_{a,out,cond} + \Delta T_{cond} \quad (^{o}C) \tag{3-69}$$

$$T_{r1} = T_{g1} - \Delta T_{evap} \quad (^{o}C) \tag{3-70}$$

$$T_{w,g1} = T_{w,g2} - \Delta T_{w,g} \quad (^{o}C) \tag{3-71}$$

<u>Energy balance</u>

The heat released by the condenser is used for space heating and DHW preheating. The load of the condenser $\dot{Q}_{load}$ can be expressed as:

$$\dot{Q}_{load} = \dot{Q}_{sp,cond} + \dot{Q}_{DHW,cond} = \dot{m}_r \cdot (h_{r2} - h_{r5}) \quad \text{(kW)} \tag{3-72}$$

where

$\dot{Q}_{sp,cond}$ is the load of the GSHP for space heating, kW. When the GSHP is integrated with radiant heating floor:

$$\dot{Q}_{sp,cond} = \dot{m}_{w,cond} \cdot c_{pw} \cdot (T_{w,out,cond} - T_{w,in,cond}) \quad \text{(kW)} \tag{3-73}$$

When the GSHP is integrated with forced air system:

$$\dot{Q}_{sp,cond} = \dot{m}_{a,cond} \cdot c_{pa} \cdot (T_{a,out,cond} - T_{a,in,cond}) \quad \text{(kW)} \tag{3-74}$$

$\dot{Q}_{DHW,cond}$ is the load of the GSHP for DHW preheating, kW.

$$\dot{Q}_{DHW,cond} = \dot{m}_{Dw,cond} \cdot c_{pw} \cdot (T_{Dw,out,cond} - T_{Dw,in,cond}) \quad \text{(kW)} \tag{3-75}$$

$\dot{m}_{w,cond}$ is the mass flow rate of the water through the condenser for space heating, kg/s;

$\dot{m}_{a,cond}$ is the mass flow rate of the air through the condenser, kg/s;

$\dot{m}_{Dw,cond}$ is the mass flow rate of the DHW through the condenser, kg/s;

$\dot{m}_r$ is the mass flow rate of the refrigerant, kg/s;

$T_{a,in,cond}$ is the temperature of the air entering the condenser, °C;

$T_{a,out,cond}$ is the temperature of the air leaving the condenser, °C;

$T_{w,in,cond}$ is the temperature of the water entering the condenser, °C;

$T_{w,out,cond}$ is the temperature of the water leaving the condenser, °C;

$T_{Dw,in,cond}$ is the temperature of the DHW entering the condenser, °C;

$T_{Dw,out,cond}$ is the temperature of the DHW leaving the condenser, °C;

h_{r2} is the specific enthalpy of R-134a leaving the compressor, kJ/kg;

h_{r5} is the specific enthalpy of R-134a leaving the condenser and entering the expansion valve, kJ/kg.

For the evaporator, the heat balance is:

$$\dot{m}_{w,g} \cdot c_{pw} \cdot (T_{w,g2} - T_{w,g1}) = \dot{m}_r \cdot (h_{r1} - h_{r6}) \quad \text{(kW)} \tag{3-76}$$

where

$\dot{m}_{w,g}$ is the mass flow rate of the water through the evaporator, kg/s;

$T_{w,g1}$ is the temperature of the water leaving the evaporator, °C;

$T_{w,g2}$ is the temperature of the water entering the evaporator, °C;

h_{r1} is the specific enthalpy of R-134a leaving the evaporator, kJ/kg;

h_{r6} is the specific enthalpy of R-134a leaving the expansion valve and entering the evaporator, kJ/kg.

The electric power input to the compressor is (DOE, 1982):

$$\dot{W}_{comp} = \dot{Q}_{cap} \cdot EIR \cdot FRAC \qquad \text{(kW)} \tag{3-77}$$

The heating capacity of the GSHP at design conditions is expressed by the following correlation-based model, developed from published product data (Water Furnace, 1986):

$$\dot{Q}_{cap} = 3.103 + 0.428 \cdot T_{w,g1} + 3.651 \cdot \dot{m}_{w,dg} \qquad \text{(kW)} \tag{3-78}$$

where

$\dot{m}_{w,dg}$ is the mass flow rate of the water through the evaporator, at the design conditions of the GSHP, kg/s;

$T_{w,g1}$ is the temperature of the water entering the evaporator, ^{o}C.

Electric input ratio is (DOE, 1982): $EIR = (0.11 + 0.89 \cdot PLR) / COP_{cap}$ (3-79)

Part load ratio is (DOE, 1982): $PLR = \dot{Q}_{load} / \dot{Q}_{cap}$ (3-80)

COP_{cap} is the coefficient of performance of the GSHP , at the design conditions expressed by the following correlation-based model, developed from published product data (Water Furnace, 1986):

$$COP_{cap} = 2.94 + 0.031 \cdot T_{w,g1} + 0.191 \cdot \dot{m}_{w,dg} \tag{3-81}$$

The fraction of the hour the GSHP is running is: $FRAC = PLR / RMIN$ (3-82)

The minimum part load ratio is assumed $RMIN = 0.1$

If PLR is less than $RMIN$, the GSHP is cycling on and off; otherwise, the GSHP is running, that is $FRAC=1$.

The isentropic efficiency of the compressor can be calculated as follows:

$$\eta_{is} = \dot{w}_{is} / \dot{w}_a = (h_{r3} - h_{r1})/(\dot{W}_{comp} / \dot{m}_r) = (h_{r3} - h_{r1})/(h_{r2} - h_{r1}) \qquad (3\text{-}83)$$

where

$\dot{w}_{is}$ is the work required by the isentropic compression process, between 1 and 3, kW;

$\dot{w}_a$ is the work required by the actual compression process, between 1 and 2, kW;

h_{r3} is the enthalpy of R-134a leaving the compressor in the isentropic compression process, kJ/kg;

Entropy generation

Entropy generation in the condenser:

$$\dot{S}_{gengcond} = \dot{m}_r \cdot (s_{r5} - s_{r2}) + \dot{m}_{w,cond} \cdot (s_{w,out,cond} - s_{w,in,cond}) + \dot{m}_{Dw,cond} \cdot (s_{Dw,out,cond} - s_{Dw,in,cond}) \text{ (kW/K)} \qquad (3\text{-}84)$$

or $\dot{S}_{gengcond} = \dot{m}_r \cdot (s_{r5} - s_{r2}) + \dot{m}_{a,cond} \cdot (s_{a,out,cond} - s_{a,in,cond}) + \dot{m}_{Dw,cond} \cdot (s_{Dw,out,cond} - s_{Dw,in,cond})$ (kW/K) (3-85)

where

equation (3-84) is for the integration of GSHP and radiant floor, and equation (3-85) is for the integration of GSHP and forced air system;

s_{r2} is the specific entropy of refrigerant R-134a at point 2 in the *T-s* diagram, kJ/kg·K;

s_{r5} is the specific entropy of refrigerant R-134a at point 5 in the *T-s* diagram, kJ/kg·K;

$s_{a,in,cond}$ is the specific entropy of the air entering the condenser, (kJ/kg·K), at $T = T_{a,in,cond}$, $P = 101$ KPa;

$s_{a,out,cond}$ is the specific entropy of the air leaving the condenser, (kJ/kg·K), at $T = T_{a,out,cond}$, $P = 101$ KPa;

$s_{w,in,cond}$ is the specific entropy of the water for space heating entering the condenser, (kJ/kg·K), at $T = T_{w,in,cond}$, $P = 101$ KPa;

$s_{w,out,cond}$ is the specific entropy of the water for space heating leaving the condenser, (kJ/kg·K), at $T = T_{w,out,cond}$, $P = 101$ KPa;

$s_{Dw,in,cond}$ is the specific entropy of the DHW entering the condenser, (kJ/kg·K), at $T = T_{Dw,in,cond}$, $P = 101$ KPa;

$s_{Dw,out,cond}$ is the specific entropy of the DHW leaving the condenser, (kJ/kg·K), at $T = T_{Dw,out,cond}$, $P = 101$ KPa.

Entropy generation in the expansion valve:

$$\dot{S}_{gen,valve} = \dot{m}_r \cdot (s_{r6} - s_{r5}) \quad \text{(kW/K)} \tag{3-86}$$

where

s_{r6} is the specific entropy of refrigerant R-134a leaving the expansion valve and entering the evaporator, kJ/kg·K.

Entropy generation in the evaporator:

$$\dot{S}_{gen,evap} = \dot{m}_r \cdot (s_{r1} - s_{r6}) + \dot{m}_{w,g} \cdot (s_{w,g1} - s_{w,g2}) \quad \text{(kW/K)} \tag{3-87}$$

s_{r1} is the specific entropy of refrigerant R-134a leaving the evaporator, kJ/kg·K;

$s_{w,g1}$ is the specific entropy of the water leaving the evaporator kJ/kg·K, at $T = T_{w,g1}, P = 101$ KPa;

$s_{w,g2}$ is the specific entropy of the water entering the evaporator, kJ/kg·K, at $T = T_{w,g2}, P = 101$ KPa.

Entropy generation in the compressor:

$$\dot{S}_{gen,comp} = \dot{m}_r \cdot (s_{r2} - s_{r1}) \quad \text{(kW/K)} \tag{3-88}$$

Total entropy generation within the GSHP is:

$$\dot{S}_{gen,gshp} = \dot{S}_{gen,cond} + \dot{S}_{gen,valve} + \dot{S}_{gen,evap} + \dot{S}_{gen,comp} \quad \text{(kW/K)} \tag{3-89}$$

Exergy destruction

Exergy destruction within the GSHP is: $\dot{X}_{de,gshp} = TK_o \cdot \dot{S}_{gen,gshp}$ (kW) (3-90)

3.2.14 Underground heat exchanger for GSHP

Underground closed loop heat exchanger is used to exchange heat with the ground. Figure 3.16 presents the inputs and outputs of an underground heat exchanger for analysis.

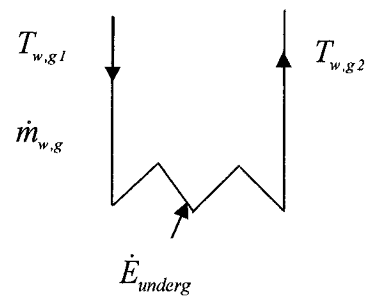

Figure 3.16 Underground heat exchanger for GSHP

Energy balance

$$\dot{E}_{underg} = \dot{m}_{w,g} \cdot c_{pw} \cdot (T_{w,g2} - T_{w,g1}) \quad \text{(kW)} \tag{3-91}$$

where

$\dot{E}_{underg}$ is the heat exchange rate between the ground and the underground heat exchanger, kW.

Entropy generation

$$\dot{S}_{gen,underg} = \dot{m}_{w,g} \cdot (s_{w,g2} - s_{w,g1}) - \dot{E}_{underg} / TK_{w,ag} \quad \text{(kW/K)} \tag{3-92}$$

where

$$TK_{w,ag} = 273.15 + (T_{w,g1} + T_{w,g2})/2 \quad \text{(K)} \tag{3-93}$$

Exergy destruction

Exergy destruction within the underground heat exchanger is:

$$\dot{X}_{de,underg} = TK_o \cdot \dot{S}_{gen,underg} \quad \text{(kW)} \tag{3-94}$$

3.2.15 Air source heat pump (ASHP)

The air source heat pump (ASHP) is used to extract heat from outdoor air to preheat the air in the forced air system and preheat the domestic hot water. Figure 3.17 shows the schematic diagram of the ASHP, and Figure 3.18 presents its operation cycle in the temperature-specific entropy (T-s) diagram.

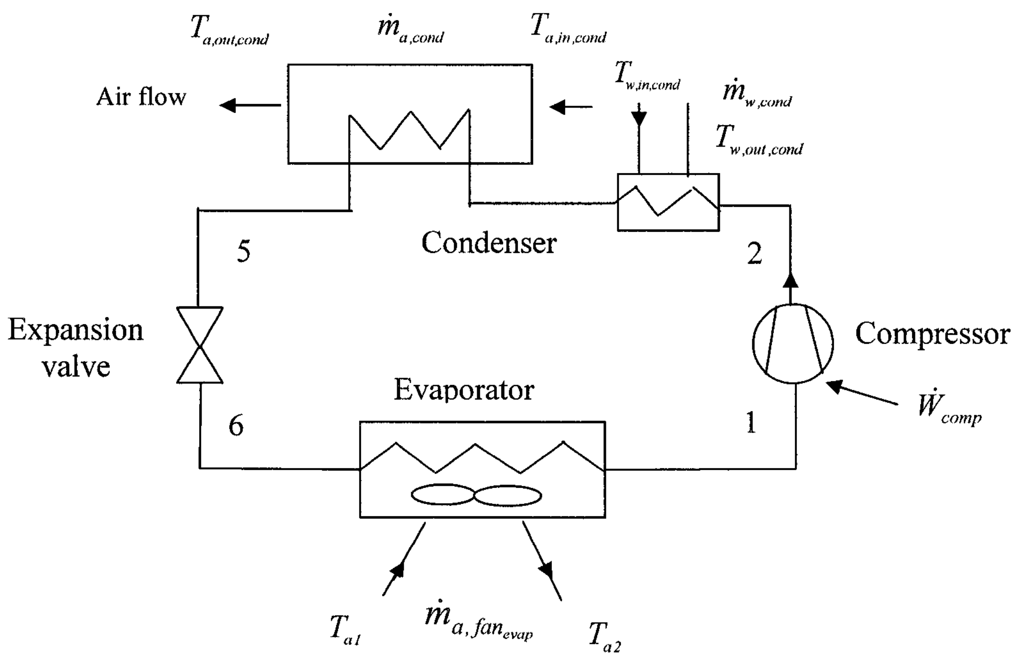

Figure 3.17 Schematic diagram of an air source heat pump

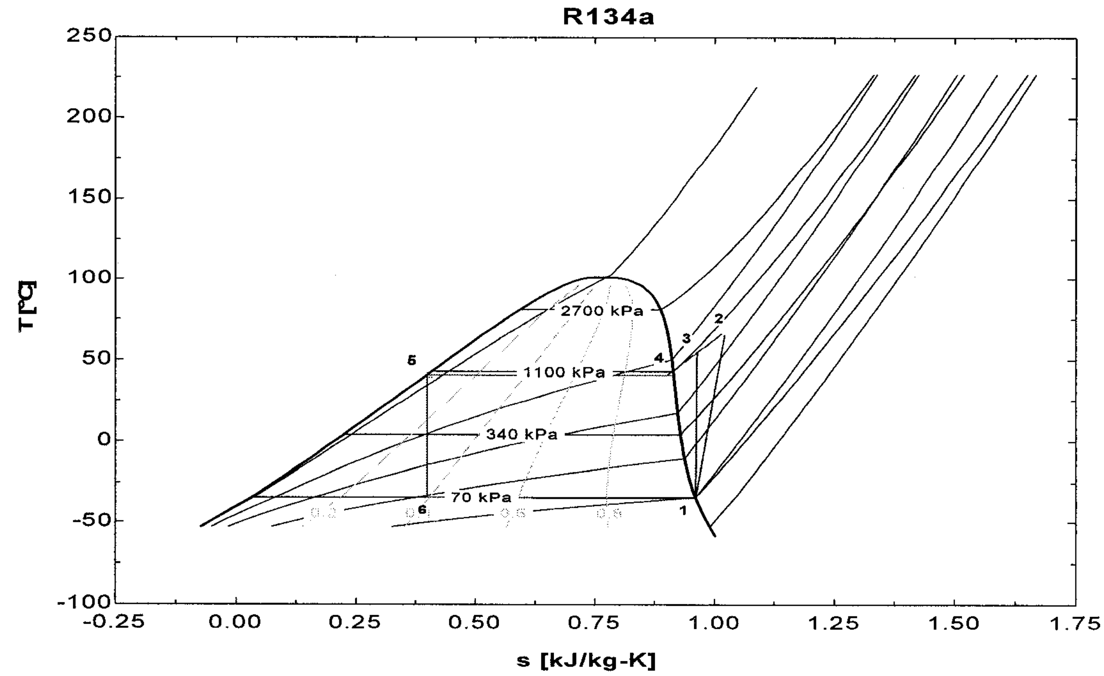

Figure 3.18 Representation of the thermodynamic cycle of ASHP in temperature-specific entropy diagram

The refrigerant state parameters s_{ri} and h_{ri} (i=1 to 6) are estimated in terms of temperature, pressure, or quality in the case of saturated refrigerant. In the following equations, the refrigerant parameters have the subscript "*r*", the water parameters have the subscript "*w*", and the air parameters have the subscript "*a*". The following assumptions are used:

(1) The temperature difference between the two fluids (i.e. water and refrigerant) within evaporator and condenser is $\Delta T_{cond} = \Delta T_{evap} = 5$ °C (Badescu, 2002);.

(2) The temperature difference between the air entering and leaving the evaporator is $\Delta T_a = 6$ °C.

These assumptions lead to the following equations:

$$T_{r4} = T_{w,out,cond} + \Delta T_{cond} = T_{a,out,cond} + \Delta T_{cond} \quad (^{o}C) \tag{3-95}$$

$$T_{r1} = T_{a2} - \Delta T_{evap} \quad (^{o}C) \tag{3-96}$$

$$T_{a1} = T_{a2} + \Delta T_{a} \quad (^{o}C) \tag{3-97}$$

where

T_{r1} is the temperature of the refrigerant at point 1 in the T-s diagram, °C;

T_{r4} is the temperature of the refrigerant at point 4 in the T-s diagram, °C;

T_{a1} is the temperature of the air entering the evaporator, °C;

T_{a2} is the temperature of the air leaving the evaporator, °C;

$T_{w,out,cond}$ is the temperature of the water leaving the condenser, °C;

$T_{a,out,cond}$ is the temperature of the air leaving the condenser, °C;

<u>Energy balance</u>

The heat released by the condenser is used for space heating and DHW preheating. The load of the condenser $\dot{Q}_{load}$ can be expressed as:

$$\dot{Q}_{load} = \dot{m}_{r} \cdot (h_{r2} - h_{r5}) = \dot{Q}_{air} + \dot{Q}_{water} \quad (kW)$$

where

$\dot{Q}_{air}$ is the load of the ASHP for space heating, kW;

$$\dot{Q}_{air} = \dot{m}_{a,cond} \cdot c_{pa} \cdot (T_{a,out,cond} - T_{a,in,cond}) \quad (kW)$$

$\dot{Q}_{water}$ is the load of the ASHP for DHW heating, kW.

$$\dot{Q}_{water} = \dot{m}_{w,cond} \cdot c_{pw} \cdot (T_{w,out,cond} - T_{w,in,cond}) \quad (kW)$$

$\dot{m}_{a,cond}$ is the mass flow rate of the air through the condenser, kg/s;

$\dot{m}_{w,cond}$ is the mass flow rate of the water through the condenser, kg/s;

$\dot{m}_r$ is the mass flow rate of the refrigerant, kg/s;

$T_{a,in,cond}$ is the temperature of the air entering the condenser, ^{o}C;

$T_{a,out,cond}$ is the temperature of the air leaving the condenser, ^{o}C;

$T_{w,in,cond}$ is the temperature of the water entering the condenser, ^{o}C;

$T_{w,out,cond}$ is the temperature of the water leaving the condenser, ^{o}C;

h_{r2} is the specific enthalpy of R-134a leaving the compressor, kJ/kg;

h_{r5} is the specific enthalpy of R-134a leaving the condenser and entering the expansion valve, kJ/kg.

For the evaporator the heat balance is:

$$\dot{m}_a \cdot c_{pa} \cdot (T_{a1} - T_{a2}) = \dot{m}_r \cdot (h_{r1} - h_{r6}) \quad \text{(kW)} \tag{3-98}$$

where

$\dot{m}_a$ is the mass flow rate of the air through the evaporator, kg/s;

T_{a1} is the temperature of the air entering the evaporator, ^{o}C;

T_{a2} is the temperature of the air leaving the evaporator, ^{o}C;

h_{r1} is the specific enthalpy of R-134a leaving the evaporator and entering the compressor, kJ/kg;

h_{r6} is the specific enthalpy of R-134a entering evaporator, kJ/kg.

The electric power input to the compressor is (Henderson et al, 1999):

$$\dot{W}_{comp} = \dot{Q}_{cap} \cdot EIR \cdot FRAC \quad \text{(kW)} \tag{3-99}$$

where

the heating capacity of the ASHP at design conditions is (Henderson et al, 1999):

$$\dot{Q}_{cap} = 20.152 \cdot \dot{m}_{a,da} + 0.3572 \cdot T_o - 3.8946 \;\; \text{(kW)} \tag{3-100}$$

$\dot{m}_{a,da}$ is the mass flow rate of the air through the evaporator, at the design conditions of the ASHP, kg/s.

Electric input ratio is (Henderson et al, 1999):

$$EIR = (0.0000625583 + 1.17517 \cdot PLR - 0.201513 \cdot PLR^2 + 0.0263344 \, PLR^3) / COP_{cap} \tag{3-101}$$

COP_{cap} is the coefficient of performance of the ASHP, at the design conditions (Henderson et al, 1999):

$$COP_{cap} = -0.5957 \dot{m}_{da} + 0.0748 T_o + 4.087 \tag{3-102}$$

PLR is Part load ratio of ASHP is: $PLR = \dot{Q}_{load} / \dot{Q}_{cap}$ (3-103)

The fraction of the hour the ASHP is running is: $FRAC = PLR / RMIN$ (3-104)

The minimum part load ratio is assumed: $RMIN = 0.1$

If PLR is less than $RMIN$, the ASHP is cycling on and off; otherwise, the ASHP is running, that is $FRAC=1$.

The isentropic efficiency of the compressor can be calculated as follows:

$$\eta_{is} = \dot{w}_{is} / \dot{w}_a = (h_{r3} - h_{r1}) / (\dot{W}_{comp} / \dot{m}_r) = (h_{r3} - h_{r1}) / (h_{r2} - h_{r1}) \tag{3-105}$$

where

$\dot{w}_{is}$ is the work required by the isentropic compression process, between 1 and 3, kW;

$\dot{w}_a$ is the work required by the actual compression process, between 1 and 2, kW;

h_{r3} is the specific enthalpy of R-134a leaving the compressor in the isentropic compression process, kJ/kg;

<u>Entropy generation</u>

Entropy generation in the condenser:

$$\dot{S}_{gengcond}=\dot{m}_r\cdot(s_{r5}-s_{r2})+\dot{m}_{a,cond}\cdot(s_{a,out,cond}-s_{a,in,cond})+\dot{m}_{w,cond}\cdot(s_{w,out,cond}-s_{w,in,cond})\quad \text{(kW/K)} \tag{3-106}$$

where

s_{r2} is the specific entropy of refrigerant R-134a leaving the compressor, kJ/kg·K;

s_{r5} is the specific entropy of refrigerant R-134a leaving the condenser, kJ/kg·K;

$s_{a,in,cond}$ is the specific entropy of the air entering the condenser, (kJ/kg·K), at $T=T_{a,in,cond}$, $P=101$ KPa;

$s_{a,out,cond}$ is the specific entropy of the air leaving the condenser, kJ/kg·K, at $T=T_{a,out,cond}$, $P=101$ KPa;

$s_{w,in,cond}$ is the specific entropy of the DHW entering the condenser, kJ/kg·K, at $T=T_{w,in,cond}$, $P=101$ KPa;

$s_{w,out,cond}$ is the specific entropy of the DHW leaving the condenser, kJ/kg·K, at $T=T_{w,out,cond1}$, $P=101$ kPa.

Entropy generation in the expansion value:

$$\dot{S}_{gen,valve}=\dot{m}_r\cdot(s_{r6}-s_{r5})\quad \text{(kW/K)} \tag{3-107}$$

where

s_{r6} is the specific entropy of refrigerant R-134a entering the evaporator, kJ/kg·K.

Entropy generation in the evaporator:

$$\dot{S}_{gen,evap} = \dot{m}_r \cdot (s_{r1} - s_{r6}) + \dot{m}_a \cdot (s_{a2} - s_{a1}) \quad \text{(kW/K)} \tag{3-108}$$

s_{r1} is the specific entropy of refrigerant R-134a entering the compressor, kJ/kg·K;

s_{a1} is the specific entropy of the air entering the evaporator, kJ/kg·K, at $T = T_{a1}, P = 101$ KPa;

s_{a2} is the specific entropy of the air leaving the evaporator kJ/kg·K, at $T = T_{a2}, P = 101$ KPa.

Entropy generation in the compressor:

$$\dot{S}_{gen,comp} = \dot{m}_r \cdot (s_{r2} - s_{r1}) \quad \text{(kW/K)} \tag{3-109}$$

Total entropy generation within the ASHP is:

$$\dot{S}_{gen,ashp} = \dot{S}_{gen,cond} + \dot{S}_{gen,valve} + \dot{S}_{gen,evap} + \dot{S}_{gen,comp} \quad \text{(kW/K)} \tag{3-110}$$

Exergy destruction within the ASHP is:

$$\dot{X}_{de,ashp} = TK_o \cdot \dot{S}_{gen,ashp} \quad \text{(kW)} \tag{3-111}$$

3.2.16 Mixing box

The mixing box is used to mix the outdoor ventilation air with the return air in the forced air system. Figure 3.19 shows the analysis diagram of the mixing box.

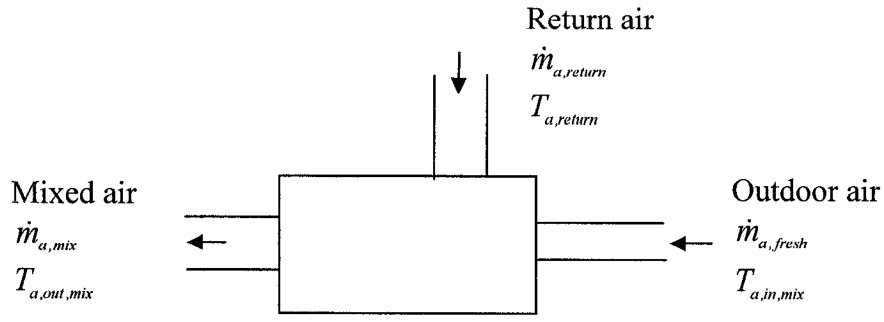

Figure 3.19 Mixing box

<u>Energy balance</u>

$$\dot{m}_{a,return} \cdot c_{pa} \cdot T_{a,return} + \dot{m}_{a,fresh} \cdot c_{pa} \cdot T_{a,in,mix} = \dot{m}_{a,mix} \cdot c_{pa} \cdot T_{a,out,mix} \quad \text{(kW)} \qquad (3\text{-}112)$$

where

$\dot{m}_{a,return}$ is the mass flow rate of return air, kg/s;

$\dot{m}_{a,fresh}$ is the mass flow rate of outdoor ventilation air, kg/s;

$\dot{m}_{a,mix}$ is the mass flow rate of mixed air, kg/s;

$T_{a,return}$ is the temperature of return air, °C;

$T_{a,in,mix}$ is the temperature of outdoor ventilation air entering the mixing box, °C;

$T_{a,out,mix}$ is the temperature of mixed air leaving the mixing box, °C.

Entropy generation

$$\dot{S}_{gen,mix} = \dot{m}_{a,mix} \cdot s_{a,out,mix} - \dot{m}_{a,fresh} \cdot s_{a,in,mix} - \dot{m}_{a,return} \cdot s_{a,return} \quad \text{(kW/K)} \qquad (3\text{-}113)$$

where

$s_{a,out,mix}$ is the specific entropy of the air leaving the mixing box, kJ/kg·K, at $T = T_{a,out,mix}$, $P = 101$ KPa;

$s_{a,in,mix}$ is the specific entropy of the outdoor ventilation air entering the mixing box, kJ/kg·K, at $T = T_{a,in,mix}$, $P = 101$ KPa;

$s_{a,return}$ is the specific entropy of the return air, kJ/kg·K, at $T = T_{a,return}$, $P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,mix} = TK_o \cdot \dot{S}_{gen,mix} \quad \text{(kW)} \qquad (3\text{-}114)$$

3.2.17 Electric air reheater (EARH)

An electric air reheater is used to reheat the air from the condenser of GSHP or ASHP using electric power, in the forced air system (Figure 3.20).

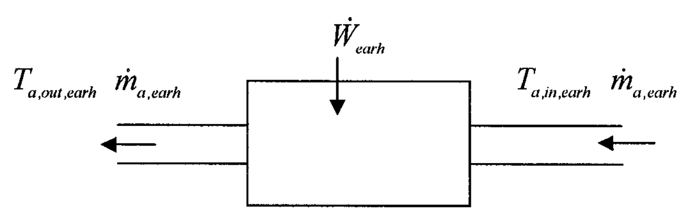

Figure 3.20 Electric air reheater

Energy balance

$$\dot{W}_{earh} = \dot{m}_{a,earh} \cdot c_{pa} \cdot (T_{a,out,earh} - T_{a,in,earh}) \quad \text{(kW)} \tag{3-115}$$

where

$\dot{W}_{earh}$ is the electric power input of the EARH, kW;

$\dot{m}_{a,earh}$ is the mass flow rate of the air through the EARH, kg/s;

$T_{a,in,earh}$ is the temperature of the air entering the EARH, °C;

$T_{a,out,earh}$ is the temperature of the air leaving the EARH, °C.

Entropy generation

$$\dot{S}_{gen,earh} = \dot{m}_{a,earh} \cdot (s_{a,out,earh} - s_{a,in,earh}) \quad \text{(kW/K)} \tag{3-116}$$

where

$s_{a,in,earh}$ is the specific entropy of the air entering the EARH kJ/kg·K, at $T = T_{a,in,earh}, P = 101$ kPa;

$s_{a,out,earh}$ is the specific entropy of the air leaving the EARH, kJ/kg·K, at $T = T_{a,out,earh}, P = 101$ kPa.

Exergy destruction

$$\dot{X}_{de,earh} = TK_o \cdot \dot{S}_{gen,earh} \quad \text{(kW)} \tag{3-117}$$

3.2.18 Hot water air reheater (WARH)

A hot water air reheater is used in the forced air system, to reheat the air from the condenser of GSHP or ASHP using hot water. Figure 3.21 presents the analysis diagram of the hot water air reheater.

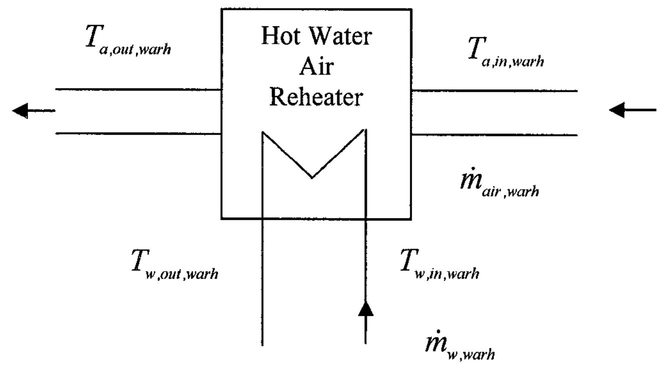

Figure 3.21 Hot water air reheater

Energy balance

$$\dot{Q}_{warh} = \dot{m}_{a,warh} \cdot c_{pa} \cdot (T_{a,out,warh} - T_{a,in,warh}) = \dot{m}_{w,warh} \cdot c_{pw} \cdot (T_{w,in,warh} - T_{w,out,warh}) \text{ (kW)} \qquad (3\text{-}118)$$

where

$\dot{m}_{a,warh}$ is the mass flow rate of the air through the WARH, kg/s;

$\dot{m}_{w,warh}$ is the mass flow rate of the water through the WARH, kg/s;

$T_{a,in,warh}$ is the temperature of the air entering the WARH, °C;

$T_{a,out,warh}$ is the temperature of the air leaving the WARH, °C;

$T_{w,in,warh}$ is the temperature of the water entering the WARH, °C;

$T_{w,out,warh}$ is the temperature of the water leaving the WARH, °C;

Entropy generation

$$\dot{S}_{gen,warh} = \dot{m}_{air,warh} \cdot (s_{a,out,warh} - s_{a,in,warh}) + \dot{m}_{w,warh} \cdot (s_{w,out.warh} - s_{w,in,warh}) \quad \text{(kW/K)} \qquad (3\text{-}119)$$

where

$s_{a,in,warh}$ is the specific entropy of the air entering the WARH kJ/kg·K, at $T = T_{a,in,warh}, P = 101$ kPa;

$s_{a,out,warh}$ is the specific entropy of the air leaving the WARH kJ/kg·K, at $T = T_{a,out,warh}, P = 101$ kPa;

$s_{w,in,warh}$ is the specific entropy of the water entering the WARH kJ/kg·K, at $T = T_{w,in,warh}, P = 101$ kPa;

$s_{w,out,warh}$ is the specific entropy of the water leaving the WARH kJ/kg·K, at $T = T_{w,out,warh}, P = 101$ kPa.

Exergy destruction

$$\dot{X}_{de,warh} = TK_o \cdot \dot{S}_{gen,warh} \qquad \text{(kW)} \qquad (3\text{-}120)$$

3.2.19 Fan

Energy balance

The electric power input of a variable speed fan can be correlated as a function of the air volume flow rate (Broan-Nutone, 1988):

$$\dot{F} = 2.9 \cdot \dot{m}_{a,fan} / \rho_a \qquad \text{(kW)} \qquad (3\text{-}121)$$

where

$\dot{m}_{a,fan}$ is the mass flow rate of the air circulated by the fan, kg/s;

ρ_a is the density of air at $T = (T_i + T_o)/2$, $P=101$ kPa.

The calculated values are, however, constrained by the minimum electric power input of fan, which is assumed to be 0.1 kW. If a calculated $\dot{F} < 0.1$ kW, $\dot{F}$ is determined as 0.1 kW.

Entropy generation

$$\dot{S}_{gen,fan} = \dot{F} / TK_o \qquad \text{(kW/K)} \tag{3-122}$$

Exergy destruction

$$\dot{X}_{de,fan} = TK_o \cdot \dot{S}_{gen,fan} \qquad \text{(kW)} \tag{3-123}$$

3.2.20 Pump

Energy balance

The electric power input of a variable speed pump can be correlated as a function of the thermal load transferred by the pump:

$$\dot{P} = 0.0058 \cdot \dot{Q}_{pump} \qquad \text{(kW)} \tag{3-124}$$

where $\dot{Q}_{pump}$ is the hot water heating capacity transported by the pump, kW.

The calculated values are, however, constrained by the minimum electric input of pump, which is assumed to be 0.1 kW. If a calculated $\dot{P} < 0.1$ kW, $\dot{P}$ is determined as 0.1 kW.

Entropy generation

$$\dot{S}_{gen,pump} = \dot{P} / TK_o \quad \text{(kW/K)} \tag{3-125}$$

Exergy destruction

$$\dot{X}_{de,pump} = TK_o \cdot \dot{S}_{gen,pump} \quad \text{(kW)} \tag{3-126}$$

3.2.21 Water-to-water heat exchanger (WWHEx)

The temperature of the water directly from the boiler is higher than the requirement of radiant heating floor. Therefore, a water-to-water heat exchanger is used to exchange heat between the water from boiler and the water supplied to the radiant floor.

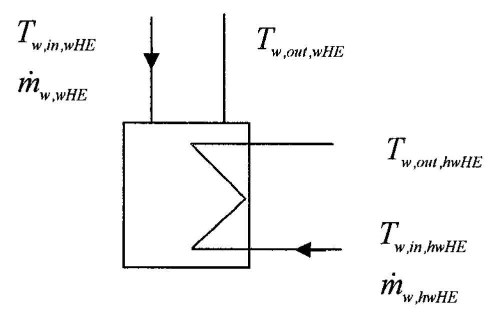

Figure 3.22 Water-to-water heat exchanger

Heat balance

$$\dot{m}_{w,wHE} \cdot c_{pw} \cdot (T_{w,out,wHE} - T_{w,in,wHE}) = \dot{m}_{w,hwHE} \cdot c_{pw} \cdot (T_{w,in,hwHE} - T_{w,out,hwHE}) \quad \text{(kW)} \tag{3-127}$$

where

$\dot{m}_{w,wHE}$ is the mass flow rate of the water through the cooler side of WWHEx, kg/s;

$\dot{m}_{w,hwHE}$ is the mass flow rate of the water through the warmer side of WWHEx, kg/s;

$T_{w,in,wHE}$ is the temperature of the water entering the cooler side of WWHEx, ^{o}C;

$T_{w,out,wHE}$ is the temperature of the water leaving the cooler side of WWHEx, ^{o}C;

$T_{w,in,hwHE}$ is the temperature of the water entering the warmer side of WWHEx, ^{o}C;

$T_{w,out,hwHE}$ is the temperature of the water leaving the warmer side of WWHEx, ^{o}C.

Entropy generation

$$\dot{S}_{gen,wHE} = \dot{m}_{w,hwHE} \cdot (s_{w,out,hwHE} - s_{w,in,hwHE}) + \dot{m}_{w,wHE} \cdot (s_{w,out,hwHE} - s_{w,in,hwHE}) \text{ (kW/K)} \qquad (3\text{-}128)$$

where

$s_{w,in,hwHE}$ is the specific entropy of the water entering the warmer side of WWHEx, kJ/kg·K, at $T = T_{w,in,hwHE}$, $P = 101$ kPa;

$s_{w,out,hwHE}$ is the specific entropy of the water leaving the warmer side of WWHEx kJ/kg·K, at $T = T_{w,out,hwHE}$, $P = 101$ kPa;

$s_{w,in,wHE}$ is the specific entropy of the water entering the cooler side of WWHEx, kJ/kg·K, at $T = T_{w,in,wHE}$, $P = 101$ kPa;

$s_{w,out,wHE}$ is the specific entropy of the water leaving the cooler side of WWHEx kJ/kg·K, at $T = T_{w,out,wHE}$, $P = 101$ kPa.

Exergy destruction

$$\dot{X}_{de,wHE} = TK_o \cdot \dot{S}_{gen,wHE} \qquad \text{(kW)} \qquad (3\text{-}129)$$

3.2.22 Power transmission lines

Power transmission lines are used to simulate the exergy losses due to the transmission of electricity from the power plant to the house.

Energy balance

The electricity output of the power plant corresponding to the electricity use in the HVAC system is:

$$\dot{W}_{pp} = \sum(\dot{W} + \dot{F} + \dot{P}) / \eta_{trans} \quad \text{(kW)} \tag{3-130}$$

where

$\sum(\dot{W} + \dot{F} + \dot{P})$ is the total on-site electric power required by the HVAC-DHW system, kW;

η_{trans} is the efficiency of electricity transmission.

Entropy generation

$$\dot{S}_{gen,trans} = \dot{W}_{pp} \cdot (1 - \eta_{trans}) / TK_o \quad \text{(kW/K)} \tag{3-131}$$

Exergy destruction:

$$\dot{X}_{de,trans} = TK_o \cdot \dot{S}_{gen,trans} \quad \text{(kW)} \tag{3-132}$$

3.2.23 Power plant

This component is used to simulate the exergy losses in the power generating plant, using a combination of energy sources.

Energy balance

In the power plant, the total primary energy supply is

$$\dot{E}_{pp,supplied} = \dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal} + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} \quad \text{(kW)} \tag{3-133}$$

where

$$\dot{E}_{pp,gas} = \alpha_{gas} \cdot \dot{W}_{pp} / \eta_{pp,gas} \quad \text{(kW)} \tag{3-134}$$

$$\dot{E}_{pp,oil} = \alpha_{oil} \cdot \dot{W}_{pp} / \eta_{pp,oil} \quad \text{(kW)} \tag{3-135}$$

$$\dot{E}_{pp,coal} = \alpha_{coal} \cdot \dot{W}_{pp} / \eta_{pp,coal} \quad \text{(kW)} \tag{3-136}$$

$$\dot{E}_{pp,nuclear} = \alpha_{nuclear} \cdot \dot{W}_{pp} / \eta_{pp,nuclear} \quad \text{(kW)} \tag{3-137}$$

$$\dot{E}_{pp,hydro} = \alpha_{hydro} \cdot \dot{W}_{pp} / \eta_{pp,hydro} \quad \text{(kW)} \tag{3-138}$$

α with its subscripts indicates the contribution of different energy sources to the off-site generation of electricity;

η_{pp} is the energy efficiency of the power plant for different energy sources;

$\dot{W}_{pp}$ is the total electricity demand including transmission loss, kW.

Entropy generation

In the power plant, the total entropy generation is

$$\dot{S}_{gen,pp} = \dot{S}_{gen,gas} + \dot{S}_{gen,oil} + \dot{S}_{gen,coal} + \dot{S}_{gen,nuclear} + \dot{S}_{gen,hydro} \quad \text{(kW/K)} \tag{3-139}$$

where

$$\dot{S}_{gengas} = \dot{E}_{pp,gas} \cdot (1-\eta_{pp,gas}) / TK_o - \dot{E}_{pp,gas} / TK_{flame} \text{ (kW/K)} \tag{3-140}$$

$$\dot{S}_{gen,oil} = \dot{E}_{pp,oil} \cdot (1-\eta_{pp,oil}) / TK_o - \dot{E}_{pp,oil} / TK_{flame} \text{ (kW/K)} \tag{3-141}$$

$$\dot{S}_{gen,coal} = \dot{E}_{pp,coal} \cdot (1-\eta_{pp,coal})/TK_o - \dot{E}_{pp,coal}/TK_{flame} \quad \text{(kW/K)} \tag{3-142}$$

$$\dot{S}_{gen,nuclear} = \dot{E}_{pp,nuclear} \cdot (1-\eta_{pp,nuclear})/TK_o \quad \text{(kW/K)} \tag{3-143}$$

$$\dot{S}_{gen,hydro} = \dot{E}_{pp,hydro} \cdot (1-\eta_{pp,hydro})/TK_o \quad \text{(kW/K)} \tag{3-144}$$

TK_{flame} is the flame temperature, K.

Exergy destruction

$$X_{de,pp} = TK_o \cdot \dot{S}_{gen,pp} \quad \text{(kW)} \tag{3-145}$$

3.2.24 "Inside house"

"Inside house" is a component used to simulate the exergy losses in the house, when the house is heated by forced air system. In this case, to maintain the indoor air temperature at the prescribed value T_i, the air is supplied at higher temperature.

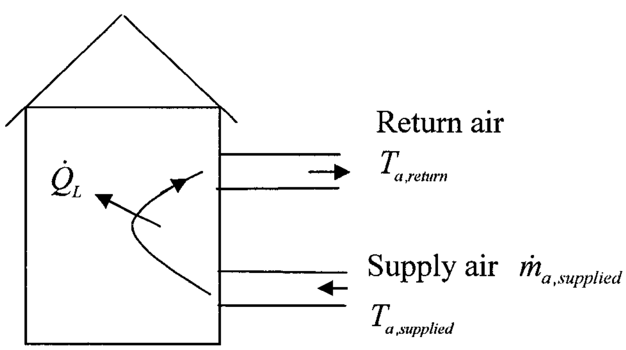

Figure 3.23 Analysis diagram of "inside house"

Energy balance

$$\dot{Q}_L = \dot{m}_{a,supplied} \cdot c_{pa} \cdot (T_{a,supplied} - T_{a,return}) \quad \text{(kW)} \tag{3-146}$$

where

$\dot{m}_{a,supplied}$ is the mass flow rate of the air supplied to heat the house by the forced air system, kg/s;

$T_{a,supplied}$ is the temperature of the air entering the house in forced air system, ^{o}C;

$T_{a,return}$ is the temperature of the air leaving the house in forced air system, ^{o}C;

Entropy generation

$$\dot{S}_{gen,inside} = \dot{Q}_L / TK_i + \dot{m}_{a,supplied} \cdot (s_{a,return} - s_{a,supplied}) \quad \text{(kW/K)} \tag{3-147}$$

where

$s_{a,supplied}$ is the specific entropy of the air entering the house in forced air system, kJ/kg·K, at $T = T_{a,supplied}$, $P = 101$ kPa;

$s_{a,return}$ is the specific entropy of the air leaving the house in forced air system, kJ/kg·K, at $T = T_{a,return}$, $P = 101$ kPa.

Exergy destruction

$$\dot{X}_{de,inside} = TK_o \cdot \dot{S}_{gen,inside} \quad \text{(kW)} \tag{3-148}$$

3.2.25 Exhaust air

When air is exhausted by the HVAC system into the outdoor environment, the energy contained in the exhaust air is discharged to the environment with it.

Energy balance

$$\dot{Q}_{exhaust} = \dot{m}_{ea,out} \cdot c_{pa} \cdot (T_{ea,out} - T_o) \quad \text{(kW)} \tag{3-149}$$

where

$\dot{m}_{ea,out}$ is the mass flow rate of the exhaust air, kg/s;

$T_{ea,out}$ is the temperature of the exhaust air leaving the HVAC system, °C;

Entropy generation

$$\dot{S}_{gen,exhaust} = \dot{m}_{ea,out} \cdot (s_o - s_{ea,out}) + \dot{Q}_{exhaust} / TK_o \qquad \text{(kW/K)} \qquad (3\text{-}150)$$

where

s_o is the specific entropy of the outdoor air, kJ/kg·K, at $T = T_o$, $P = 101$ KPa;

$s_{ea,out}$ is the specific entropy of the exhaust air leaving the HVAC system, kJ/kg·K, at $T = T_{ea,out}$, $P = 101$ KPa.

Exergy destruction

$$\dot{X}_{de,exhaust} = TK_o \cdot \dot{S}_{gen,exhaust} \qquad \text{(kW/K)} \qquad (3\text{-}151)$$

CHAPTER 4

DEVELOPMENT OF MODELS OF HVAC SYSTEMS

The component models presented in Chapter 3 can be assembled according to each proposed integrated HVAC-DHW system, and the corresponding performance can be estimated. The following performance parameters are used in this study: energy and exergy efficiency, energy and exergy demand, entropy generation, and exergy destruction. Exergy analysis is conducted under peak design conditions and annual operating conditions. The results of exergy analysis can locate the inefficient areas of the system where it is necessary to apply some measures to reduce exergy consumption.

4.1 Selected HVAC-DHW systems for a house

For comparison purposes, some non-integrated systems are presented in this study. In addition, design alternatives No.1, No.2, and No.3 do not provide ventilation air. Twenty different design alternatives are selected. Table 4.1 shows the description of these design alternatives. The design alternative No.1 is the base case which is commonly used in Quebec, and the design alternatives No.2 to No.20, are generated with one or more improvements over the base case.

Table 4.1 Description of the design alternative No.1 to No.20

Alt. No.	Heating	Ventilation	DHW
1	electric baseboard heaters	none	electric water heater
2	hot water baseboard heaters with gas-fired boiler	none	heat exchanger with gas-fired boiler
3	radiant heating floor with GSHP	none	GSHP and electric water heater
4	electric baseboard heaters	electric air heater	electric water heater
5	electric baseboard heaters	electric air heater and air-to-air heat exchanger	electric water heater
6	electric baseboard heaters	electric air heater, air-to-air heat exchanger and earth tube heat exchanger	electric water heater
7	hot water baseboard heaters with gas-fired boiler	hot water air heater, air-to-air heat exchanger, and earth tube heat exchanger	heat exchanger with gas-fired boiler
8	hot water baseboard heaters with gas-fired boiler with economizer	hot water air heater, air-to-air heat exchanger and earth tube heat exchanger	heat exchanger with gas-fired boiler with economizer
9	forced air system with electric air heater	air-to-air heat exchanger and earth tube heat exchanger	electric water heater
10	forced air system with hot water heating coil and gas-fired boiler	air-to-air heat exchanger and earth tube heat exchanger	heat exchanger with gas-fired boiler
11	forced air system with hot water heating coil and gas-fired boiler with economizer	air-to-air heat exchanger and earth tube heat exchanger	heat exchanger with gas-fired boiler with economizer
12	forced air system with electric air heater and GSHP	hot water air heater, electric air heater, air-to-air heat exchanger and earth tube heat exchanger	GSHP and electric water heater
13	forced air system with hot water heating coil, gas-fired boiler and GSHP	air-to-air heat exchanger and earth tube heat exchanger	GSHP and gas-fired boiler
14	forced air system with hot water heating coil, ASHP and electric air heater	air-to-air heat exchanger and earth tube heat exchanger	ASHP and electric water heater
15	forced air system with hot water heating coil, gas-fired boiler and ASHP	air-to-air heat exchanger and earth tube heat exchanger	ASHP and gas-fired boiler
16	radiant floor with electric boiler	electric air heater, air-to-air heat exchanger and earth tube heat exchanger	electric water heater
17	radiant floor with gas-fired boiler	hot water air heater, air-to-air heat exchanger and earth tube heat exchanger	heat exchanger with gas-fired boiler
18	radiant heating floor with gas-fired boiler with economizer	hot water air heater, air-to-air heat exchanger and earth tube heat exchanger	heat exchanger with gas-fired boiler with economizer
19	radiant heating floor with GSHP	hot water air heater, air-to-air heat exchanger and earth tube heat exchanger	GSHP and electric water heater
20	radiant heating floor with GSHP	hot water air heater, air-to-air heat exchanger and earth tube heat changer	GSHP and gas-fired water heater

4.2 System models

4.2.1 Design alternative No.1

The configuration of design alternative No.1 is shown in Figure 4.1. The house is heated by electric baseboard heater, and domestic hot water is heated by electricity. There is no ventilation system.

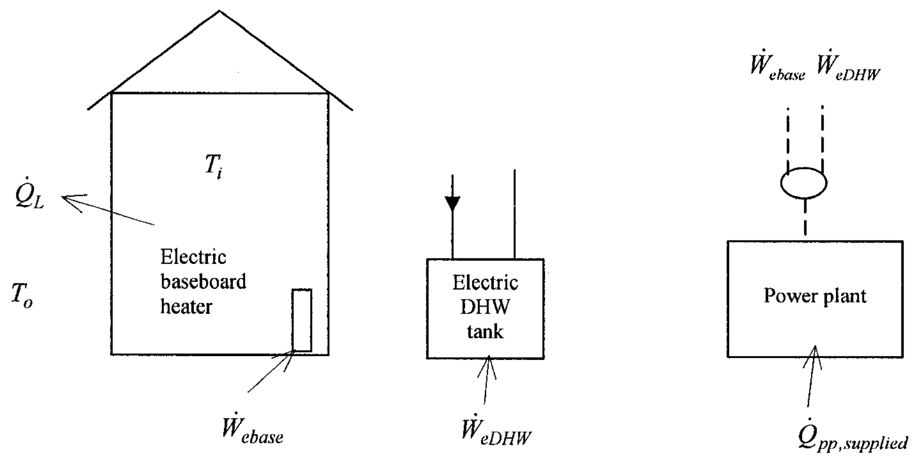

Figure 4.1 Configuration of design alternative No.1

<u>Energy analysis</u>

Total useful energy $\dot{E}_{useful} = \dot{W}_{ebase} + \dot{W}_{eDHW}$ (kW) (4-1)

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{W}_{ebase} + \dot{W}_{eDHW})/\eta_{trans} \quad \text{(kW)} \tag{4-2}$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied}$ (kW) (4-3)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-4)

<u>Entropy analysis</u>

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,ebase} + \dot{S}_{eDHW} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \tag{4-5}$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-6)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} \quad \text{(kW)} \tag{4-7}$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-8)

4.2.2 Design alternative No.2

This design alternative uses hot water baseboard heaters with gas-fired boiler for space heating. There is no ventilation system. Domestic hot water is heated in the hot water DHW tank by hot water from gas-fired boiler.

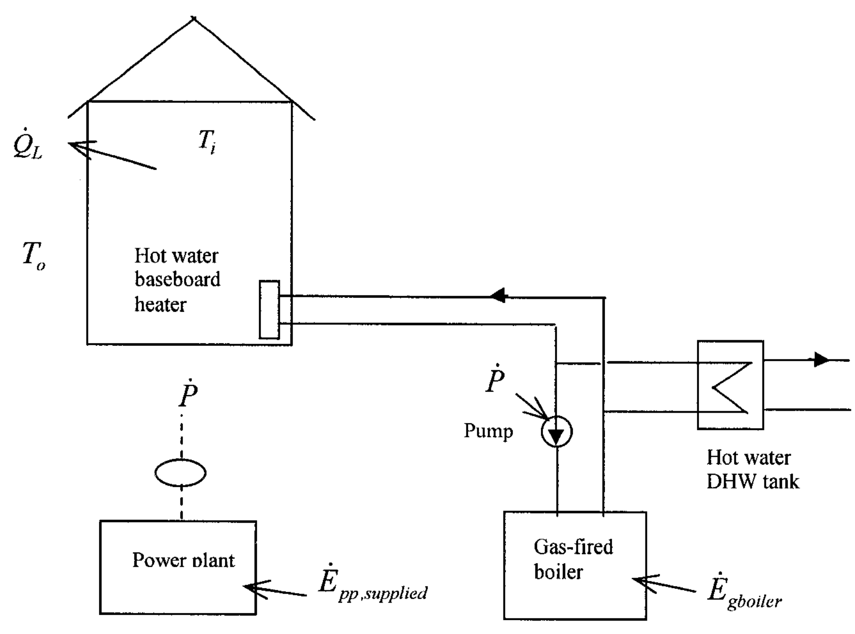

Figure 4.2 Configuration of design alternative No.2

In this case, the following conditions apply:

$\dot{m}_{w,gboiler} = \dot{m}_{w,wbase} + \dot{m}_{w,hDHW}$ (kg/s)

$T_{w,in,wbase} = T_{w,in,hDHW} = T_{w,out,gboiler}$ (^oC)

$T_{w,out,wbase} = T_{w,out,hDHW} = T_{w,in,gboiler}$ (^oC)

$\dot{Q}_{pump} = \dot{Q}_L + \dot{Q}_{DHW}$ (kW)

Energy analysis

Total useful energy $\dot{E}_{useful} = \dot{Q}_L + \dot{Q}_{DHW}$ (kW) (4-9)

Total electricity demand including electricity transmission loss

$\dot{W}_{pp} = \dot{P} / \eta_{trans}$ (kW) (4-10)

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler}$ (kW) (4-11)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-12)

Entropy analysis

Total entropy generation:

$\dot{S}_{gen,total} = \dot{S}_{gen,wbase} + \dot{S}_{DHW} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp}$ (kW/K) (4-13)

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-14)

Total exergy supply from various energy sources

$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{ppgas} + \dot{E}_{ppoil} + \dot{E}_{ppcoal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{ppnuclear} + \dot{E}_{pphydr}$ (kW) (4-15)

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-16)

4.2.3 Design alternative No.3

This design alternative uses radiant heating floor with GSHP for space heating. There is no ventilation system. Domestic hot water is first preheated by GSHP and then reheated by electricity. Figure 4.3 shows the schematic of design alternative No.3 for analysis.

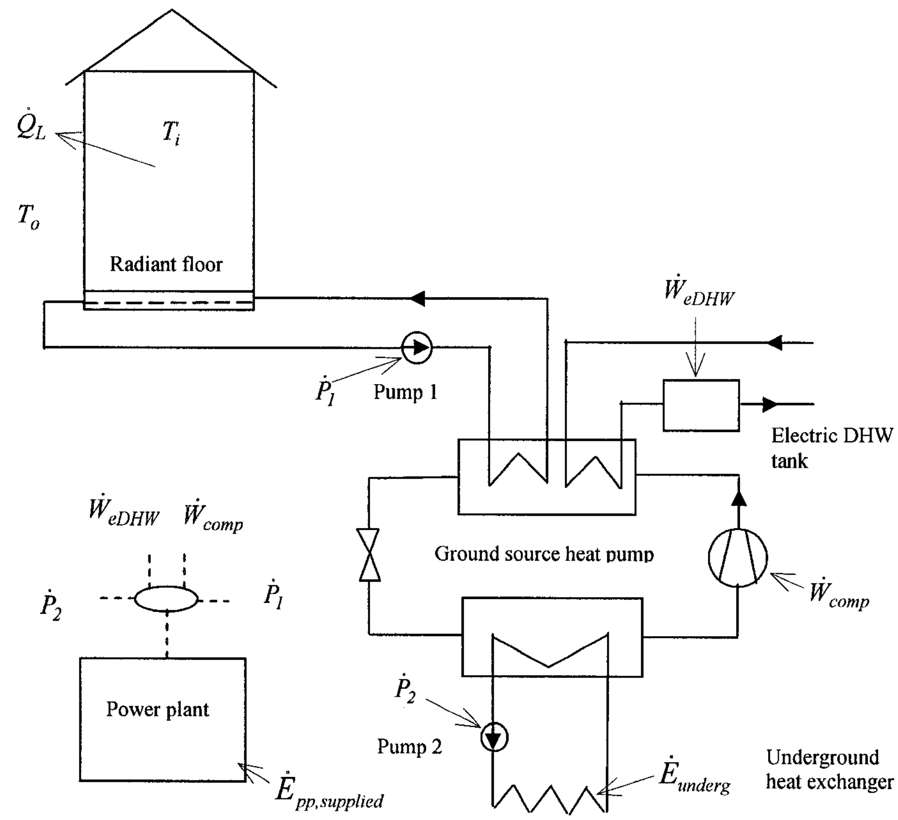

Figure 4.3 Configuration of design alternative No.3

In this case, the following conditions apply:

$T_{Dw,out,cond} = T_{w,in,eDHW}$ (°C); $T_{w,in,cond} = T_{w,out,floor}$ (°C); $T_{w,out,cond} = T_{w,in,floor}$ (°C)

$\dot{Q}_{pump1} = \dot{Q}_L$ (kW); $\dot{Q}_{pump2} = \dot{E}_{underg}$ (kW)

$\dot{m}_{w,floor} = \dot{m}_{w,cond}$ (kg/s); $\dot{m}_{w,DHW} = \dot{m}_{Dw,cond}$ (kg/s)

Energy analysis

Total useful energy $\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,cond1})$ (kW) (4-17)

Total electricity demand including electricity transmission loss

$\dot{W}_{pp} = (\dot{P}_1 + \dot{P}_2 + \dot{W}_{comp} + \dot{W}_{eDHW}) / \eta_{trans}$ (kW) (4-18)

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{underg}$ (kW) (4-19)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-20)

Entropy analysis

Total entropy generation

$\dot{S}_{gen,total} = \dot{S}_{gen,eDHW} + \dot{S}_{gen,underg} + \dot{S}_{gen,pump} + \dot{S}_{gen,gshp} + \dot{S}_{gen,floor} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp}$ (kW/K) (4-21)

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-22)

Total exergy supply from various energy sources

$\dot{X}_{supplied} = (\dot{E}_{ppgas} + \dot{E}_{ppoil} + \dot{E}_{ppcoal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{ppnuclear} + \dot{E}_{pphydro} + \left| \dot{Q}_{underg} \cdot (1 - TK_o / TK_{w,ag}) \right|$ (kW) (4-23)

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-24)

4.2.4 Design alternative No.4

The house is heated by electric baseboard heater. The ventilation air is heated by an electric air heater up to the value of indoor air temperature. Domestic hot water is heated by electricity. The model of design alternative No.4 is generated by assembling the corresponding component models together shown in Figure 4.4.

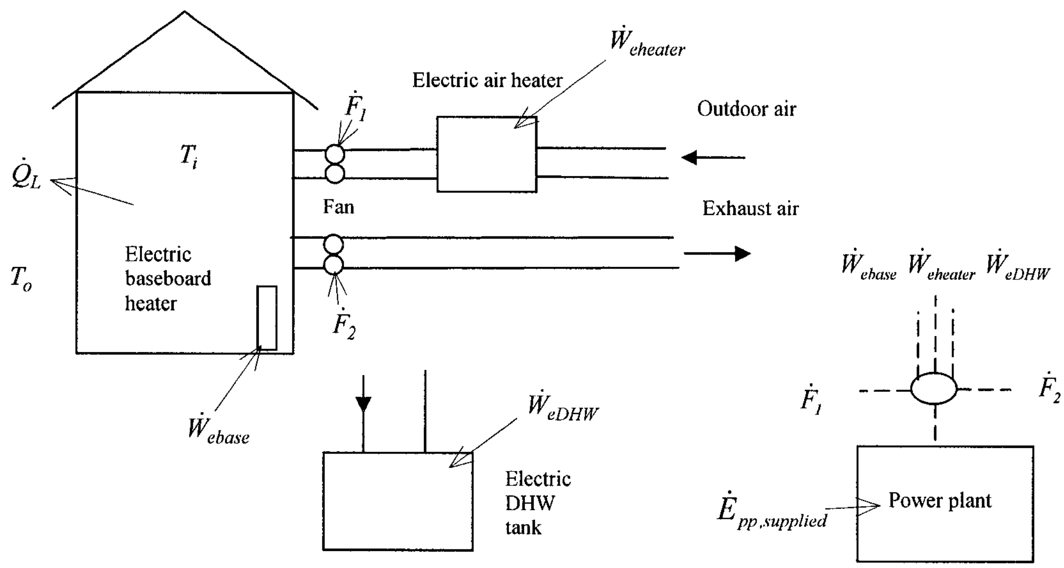

Figure 4.4 Configuration of design alternative No.4

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,eheater} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{ea,out} = T_i \text{ (°C)}; \; T_{a,out,eheater} = T_i \text{ (°C)}; \; T_{a,in,eheater} = T_o \text{ (°C)}$$

where

$\dot{m}_{a,fan1}$ is the mass flow rate of the air transported by fan 1, kg/s;

$\dot{m}_{a,fan2}$ is the mass flow rate of the air transported by fan 2, kg/s.

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,eDHW} - T_{w,in,eDHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}25)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{W}_{ebase} + \dot{W}_{eheater} + \dot{W}_{eDHW} + \dot{F}_1 + \dot{F}_2) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}26)$$

Total primary energy supply

$$\dot{E}_{supplied} = \dot{E}_{pp,supplied} \quad \text{(kW)} \qquad (4\text{-}27)$$

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-28)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,eheater} + \dot{S}_{gen,ebase} + \dot{S}_{eDHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}29)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-30)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} \quad \text{(kW)} \qquad (4\text{-}31)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-32)

4.2.5 Design alternative No.5

The only difference between design alternative No.4 and No.5 is the use of air-to-air heat exchanger in design alternative No.5 to recover heat from the exhaust air. The model is shown in Figure 4.5.

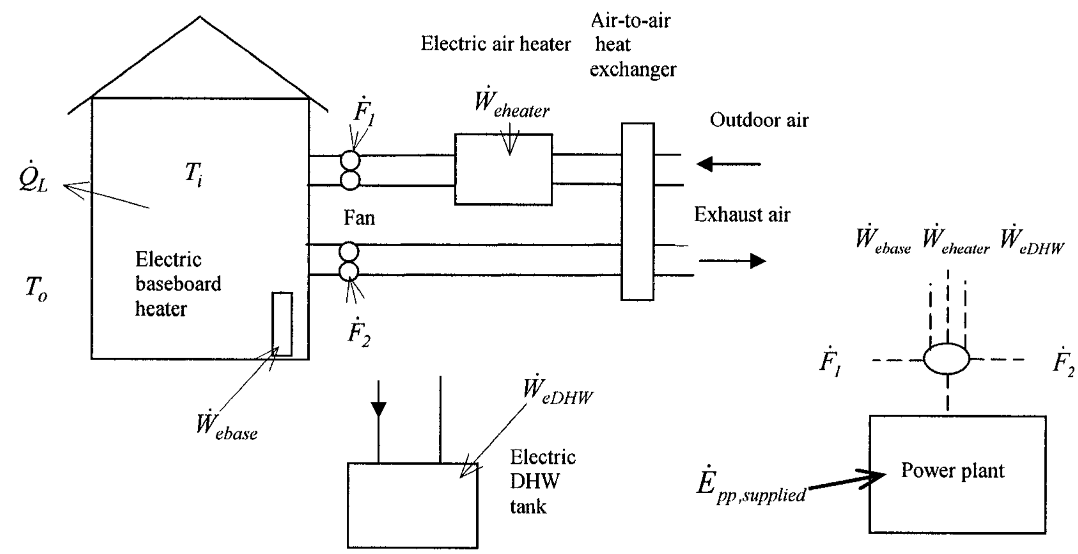

Figure 4.5 Configuration of design alternative No.5

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,eheater} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,HE} = T_o \text{ (°C)};\ T_{a,out,HE} = T_{a,in,eheater} \text{ (°C)};\ T_{a,out,eheater} = T_i \text{ (°C)}$$

$$T_{ea,in,HE} = T_i \text{ (°C)};\ T_{ea,out,HE} = T_{ea,out} \text{ (°C)}$$

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,eDHW} - T_{w,in,eDHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}33)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{W}_{ebase} + \dot{W}_{eheater} + \dot{W}_{eDHW} + \dot{F}_1 + \dot{F}_2) / \eta_{trans} \quad \text{(kW)} \tag{4-34}$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied}$ (kW) (4-35)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-36)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,HE} + \dot{S}_{gen,eheater} + \dot{S}_{gen,ebaseboard} + \dot{S}_{eDHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \text{(kW/K)} \tag{4-37}$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-38)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} \quad \text{(kW)} \tag{4-39}$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-40)

4.2.6 Design alternative No.6

The improvement of design alternative No.6 over No.5 is the adoption of an earth tube heat exchanger to preheat outdoor ventilation air. The model of this alternative is established by adding the component earth tube heat exchanger to the model of alternative No.3.

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,eheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o \,(^oC);\; T_{a,in,HE} = T_{a,out,tube} \,(^oC);\; T_{a,out,HE} = T_{a,in,eheater} \,(^oC);\; T_{a,out,eheater} = T_i \,(^oC)$$

$$T_{ea,in,HE} = T_i \,(^oC);\; T_{ea,out,HE} = T_{ea,out} \,(^oC)$$

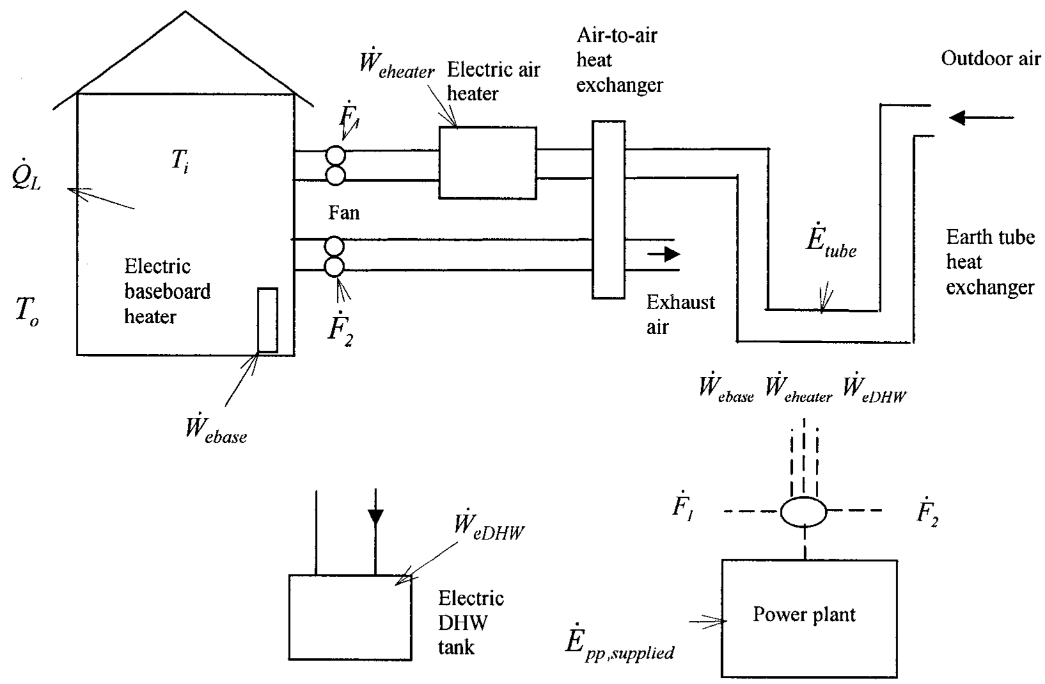

Figure 4.6 Configuration of design alternative No.6

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,eDHW} - T_{w,in,eDHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}41)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{W}_{ebase} + \dot{W}_{eheater} + \dot{W}_{eDHW} + \dot{F}_1 + \dot{F}_2)/\eta_{trans} \quad \text{(kW)} \qquad (4\text{-}42)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{tube}$ (kW) (4-43)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-44)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,eheater} + \dot{S}_{gen,ebaseboard} + \dot{S}_{eDHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \text{ (kW/K)} \quad (4\text{-}45)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-46)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) \text{ (kW)} \quad (4\text{-}47)$$

where $TK_{ground} = 273.15 + T_{ground}$ (K) (4-48)

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-49)

4.2.7 Design alternative No.7

Based on the design alternative No.2, the design alternative No.6 uses a ventilation system, in which the outdoor air is first preheated in the earth tube heat exchanger, and then in the air-to-air heat exchanger before enters the hot water air heater. The ventilation air is finally supplied to the room at the indoor air temperature. Figure 4.7 presents its configuration.

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$\dot{m}_{w,gboiler} = \dot{m}_{w,wbase} + \dot{m}_{w,wheater} + \dot{m}_{w,hDHW} \quad \text{(kg/s)}$$

$$T_{a,in,tube} = T_o \,(^\circ\text{C});\; T_{a,in,HE} = T_{a,out,tube} \,(^\circ\text{C});\; T_{a,out,HE} = T_{a,in,wheater} \,(^\circ\text{C});\; T_{a,out,wheater} = T_i \,(^\circ\text{C})$$

$$T_{ea,in,HE} = T_i \,(^\circ\text{C});\; T_{ea,out} = T_{ea,out,HE} \;(^\circ\text{C})$$

$$T_{w,in,wbase} = T_{w,in,wheater} = T_{w,in,hDHW} = T_{w,out,gboiler} \;(^\circ\text{C})$$

$$T_{w,out,wbase} = T_{w,out,wheater} = T_{w,out,hDHW} = T_{w,in,gboiler} \;(^\circ\text{C})$$

$$\dot{Q}_{pump} = \dot{Q}_L + \dot{Q}_{wheater} + \dot{Q}_{DHW} \quad \text{(kW)}$$

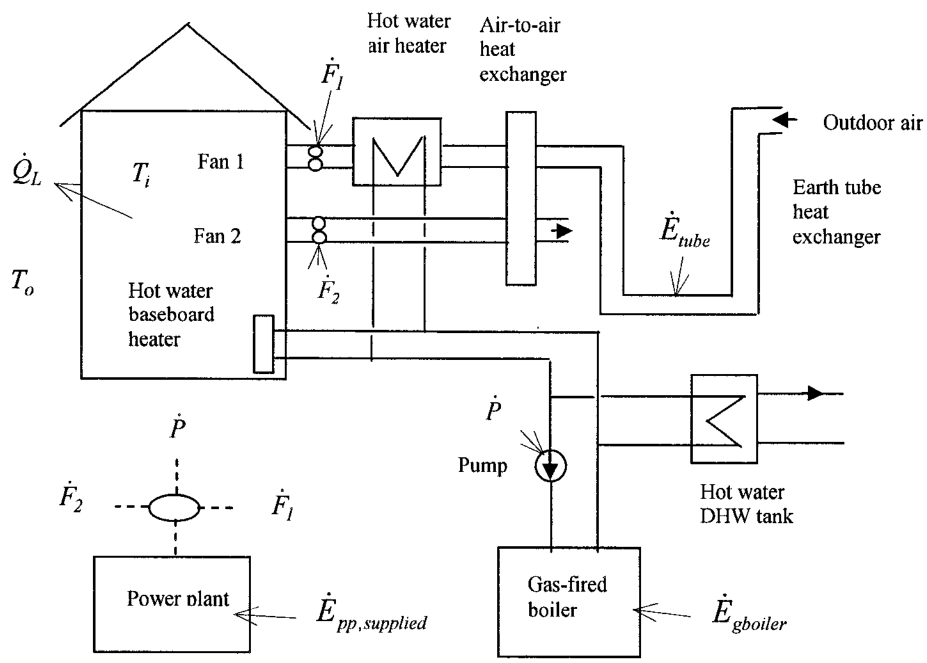

Figure 4.7 Configuration of design alternative No.7

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}50)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P}) / \eta_{trans} \quad \text{(kW)} \tag{4-51}$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{tube}$ (kW) (4-52)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-53)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,wbase} + \dot{S}_{DHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \tag{4-54}$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-55)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) \quad \text{(kW)} \tag{4-56}$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-57)

4.2.8 Design alternative No.8

In order to improve the performance of gas-fired boiler in design alternative No.7, an economizer is used to recover the heat from flue gases to preheat the domestic hot water. Figure 4.8 shows the schematic of design alternative No.8.

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o\,(^{o}\mathrm{C});\ T_{a,in,HE} = T_{a,out,tube}\,(^{o}\mathrm{C});\ T_{a,out,HE} = T_{a,in,wheater}\,(^{o}\mathrm{C});\ T_{a,out,wheater} = T_i\,(^{o}\mathrm{C})$$

$$T_{ea,in,HE} = T_i\,(^{o}\mathrm{C});\ T_{ea,out,HE} = T_{ea,out}\ (^{o}\mathrm{C})$$

$$T_{w,in,wbase} = T_{w,in,wheater} = T_{w,in,hDHW} = T_{w,out,gboiler}\,(^{o}\mathrm{C});$$

$$T_{w,out,wbase} = T_{w,out,wheater} = T_{w,out,hDHW} = T_{w,in,gboiler};\ T_{w,out,econ} = T_{w,in,DHW}\ (^{o}\mathrm{C})$$

$$T_{flue,in,gbolier} = T_o\ (^{o}\mathrm{C});\ T_{flue,out,gboiler} = T_{flue,in,econ}\,(^{o}\mathrm{C})$$

$$\dot{Q}_{pump} = \dot{Q}_L + \dot{Q}_{wheater} + \dot{Q}_{DHW}\ \ \text{(kW)}$$

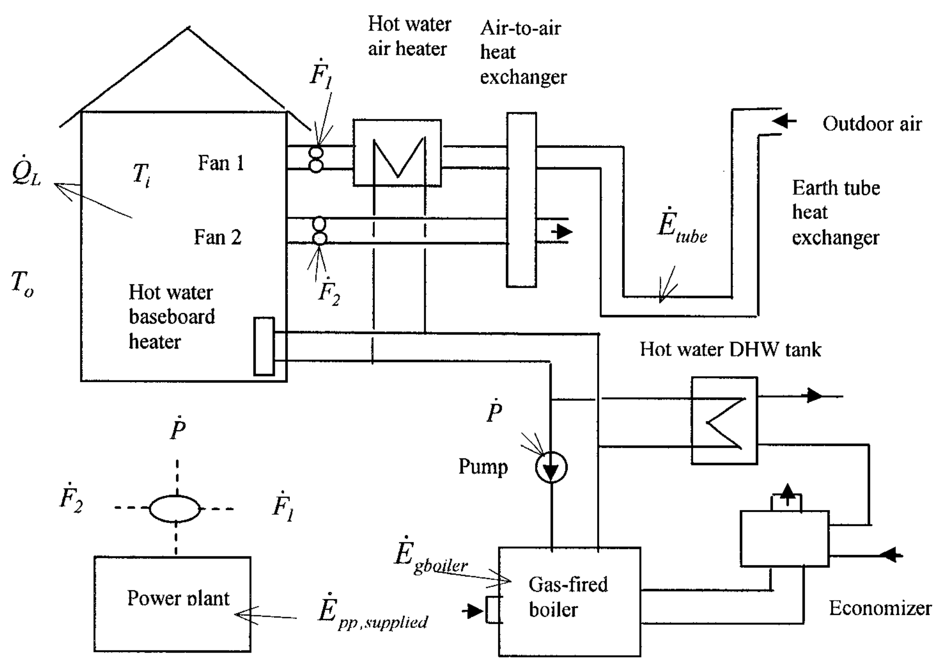

Figure 4.8 Configuration of design alternative No.8

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,econ}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \text{ (kW)} \qquad (4\text{-}58)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}59)$$

Total primary energy supply $\dot{E}_{\text{supplied}} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{tube}$ (kW) (4-60)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-61)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,wbase} + \dot{S}_{DHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,econ} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}62)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-63)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) \quad \text{(kW)} \qquad (4\text{-}64)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-65)

4.2.9 Design alternative No.9

This design alternative uses forced air system to heat the house and bring in fresh outdoor air, and electricity is used to heat the forced air and domestic hot water. Figure 4.9 shows the schematic of design alternative No.9.

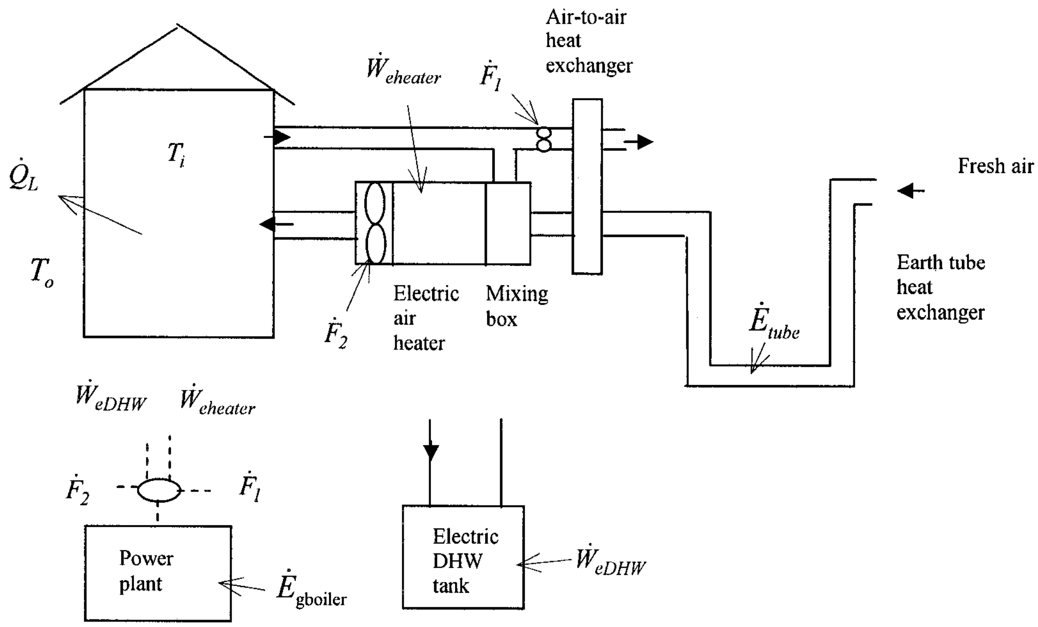

Figure 4.9 Configuration of design alternative No.9

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan,1} \text{ (kg/s)}$$

$$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,eheater} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o\,(^oC); T_{a,in,HE} = T_{a,out,tube}\,(^oC); T_{a,out,HE} = T_{a,in,mix}\,(^oC)$$

$$T_{a,out,mix} = T_{a,in,eheater}\,(^oC);\ T_{a,out,eheater} = T_{a,supplied}\,(^oC); T_{ea,in,HE} = T_{a,return} = T_i\,(^oC)$$

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \text{ (kW)} \qquad (4\text{-}66)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{W}_{eheater} + \dot{W}_{eDHW}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}67)$$

Total primary energy supply $\dot{E}_{\mathrm{supplied}} = \dot{E}_{pp,supplied} + \dot{E}_{tube}$ (kW) (4-68)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-69)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,eheater} + \dot{S}_{gen,mix} + \dot{S}_{gen,eDHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,inside} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}70)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-71)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{ppgas} + \dot{E}_{ppoil} + \dot{E}_{ppcoal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{ppnuclear} + \dot{E}_{pphydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) \text{ (kW)} \qquad (4\text{-}72)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-73)

4.2.10 Design alternative No.10

This alternative uses forced air system to heat the house and bring in outdoor air, and gas-fired boiler is adopted to heat the forced air and domestic hot water. Figure 4.10 shows the schematic of design alternative No.10 for analysis.

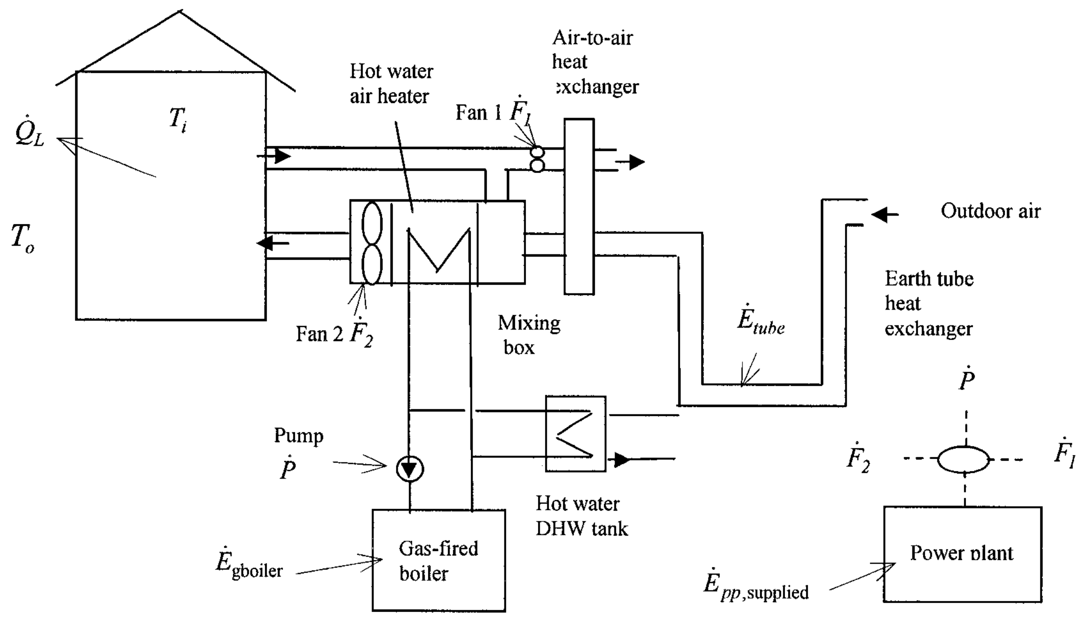

Figure 4.10 Configuration of design alternative No.10

In the forced air system, the supply air mass flow rate must be greater than or equal to the minimum ventilation air. The following calculation sequence is modeled.

The supply air mass flow rate is:

$$\dot{m}_{a,supplied} = \dot{Q}_L / (c_{pa} \cdot (T_{a,supplied} - T_i)) \quad \text{(kg/s)} \tag{4-74}$$

where

$T_{a,supplied}$ is the supply air temperature (e.g., 50 °C), °C.

If $\dot{m}_{a,supplied} < \dot{m}_{a,fresh}$, then $\dot{m}_{a,supplied} = \dot{m}_{a,fresh}$

thus $T_{a,supplied} = \dot{Q}_L / (\dot{m}_{a,supplied} \cdot c_{pa}) + T_i$ (4-75)

This strategy is applicable to all the forced air systems in this study.

In this case, the following conditions apply:

$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1}$ (kg/s)

$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,fan2} = \dot{m}_{wheater}$ (kg/s)

$T_{a,in,tube} = T_o$ (°C); $T_{a,in,HE} = T_{a,out,tube}$ (°C); $T_{a,out,HE} = T_{a,in,mix}$ (°C); $T_{a,out,mix} = T_{a,in,wheater}$ (°C);

$T_{a,out,wheater} = T_{a,supplied}$ (°C); $T_{ea,in,HE} = T_i$ (°C); $T_{ea,out,HE} = T_{ea,out}$ (°C)

$T_{w,in,wheater} = T_{w,in,hDHW} = T_{w,out,gboiler}$ (°C); $T_{w,out,wheater} = T_{w,out,hDHW} = T_{w,in,gboiler}$ (°C)

$\dot{Q}_{pump} = \dot{Q}_{wheater} + \dot{Q}_{DHW}$ (kW)

Energy analysis

Total useful energy

$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o)$ (kW) (4-76)

Total electricity demand including electricity transmission loss

$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P}) / \eta_{trans}$ (kW) (4-77)

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{tube}$ (kW) (4-78)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-79)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,mix} + \dot{S}_{gen,DHW} + \dot{S}_{gen,exhaust}$$

$$+ \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,inside} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \quad (4\text{-}80)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-81)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame})$$

$$+ \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) \quad \text{(kW)} \quad (4\text{-}82)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-83)

4.2.11 Design alternative No.11

Design alternative No.11 is derived from design alternative No.10 by adding an economizer to the gas-fired boiler. The recovered heat from flue gases is used to preheat domestic hot water. Figure 4.11 shows the schematic of design alternative No.11.

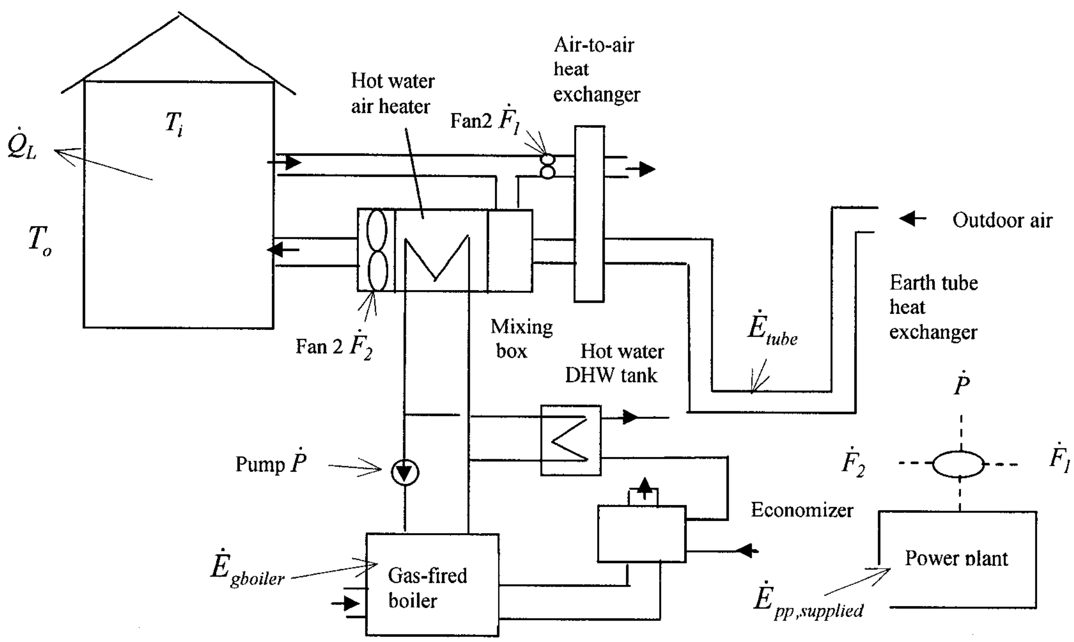

Figure 4.11 Configuration of design alternative No.11

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} \text{ (kg/s)}$$

$$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,fan2} = \dot{m}_{a,wheater} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o\ (^{\circ}\mathrm{C}); T_{a,in,HE} = T_{a,out,tube}\ (^{\circ}\mathrm{C}); T_{a,out,HE} = T_{a,in,mix}\ (^{\circ}\mathrm{C}); T_{a,out,mix} = T_{a,in,wheater}\ (^{\circ}\mathrm{C})$$

$$T_{a,out,wheater} = T_{a,supplied}\ (^{\circ}\mathrm{C}); T_{ea,in,HE} = T_i\ (^{\circ}\mathrm{C});\ T_{ea,out,HE} = T_{ea,out}\ (^{\circ}\mathrm{C})$$

$$T_{w,in,wheater} = T_{w,in,hDHW} = T_{w,out,gboiler}\ (^{\circ}\mathrm{C});\ T_{w,out,wheater} = T_{w,out,hDHW} = T_{w,in,gboiler}\ (^{\circ}\mathrm{C})$$

$$T_{w,out,econ} = T_{w,in,DHW}\ (^{\circ}\mathrm{C})$$

$$\dot{Q}_{pump} = \dot{Q}_{wheater} + \dot{Q}_{DHW}\ \text{(kW)}$$

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot c_{pw} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \text{ (kW)} \qquad (4\text{-}84)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}85)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{tube}$ (kW) (4-86)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-87)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,mix} + \dot{S}_{gen,DHW} + \dot{S}_{gen,exhaust}$$
$$+ \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,inside} + S_{gen,econ} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}88)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-89)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame})$$
$$+ \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) \quad \text{(kW)} \qquad (4\text{-}90)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-91)

4.2.12 Design alternative No.12

This design alternative integrates GSHP and forced air system in the HVAC system of the house with domestic hot water preheated and heated by GSHP condenser and electricity respectively. Outdoor air is brought in through earth tube heat exchanger, air-to-air heat exchanger; Figure 4.12 shows the integration of design alternative No.12.

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} \text{ (kg/s)}$$

$$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,cond} = \dot{m}_{a.earh} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o\ (^{o}\text{C}); T_{a,in,HE} = T_{a,out,tube}\ (^{o}\text{C}); T_{a,out,HE} = T_{a,in,mix}\ (^{o}\text{C}); T_{a,out,mix} = T_{a,in,cond}\ (^{o}\text{C})$$

$$T_{a,out,cond} = T_{a,in,earh}\ (^{o}\text{C})\ T_{a,out,earh} = T_{a,supplied}\ (^{o}\text{C}); T_{ea,in,HE} = T_i\ (^{o}\text{C});\ T_{ea,out,HE} = T_{ea,out}\ (^{o}\text{C})$$

$$T_{w,out,cond} = T_{w,in,eDHW}\ (^{o}\text{C})$$

$$\dot{Q}_{pump} = \dot{E}_{underg} \text{ (kW)}$$

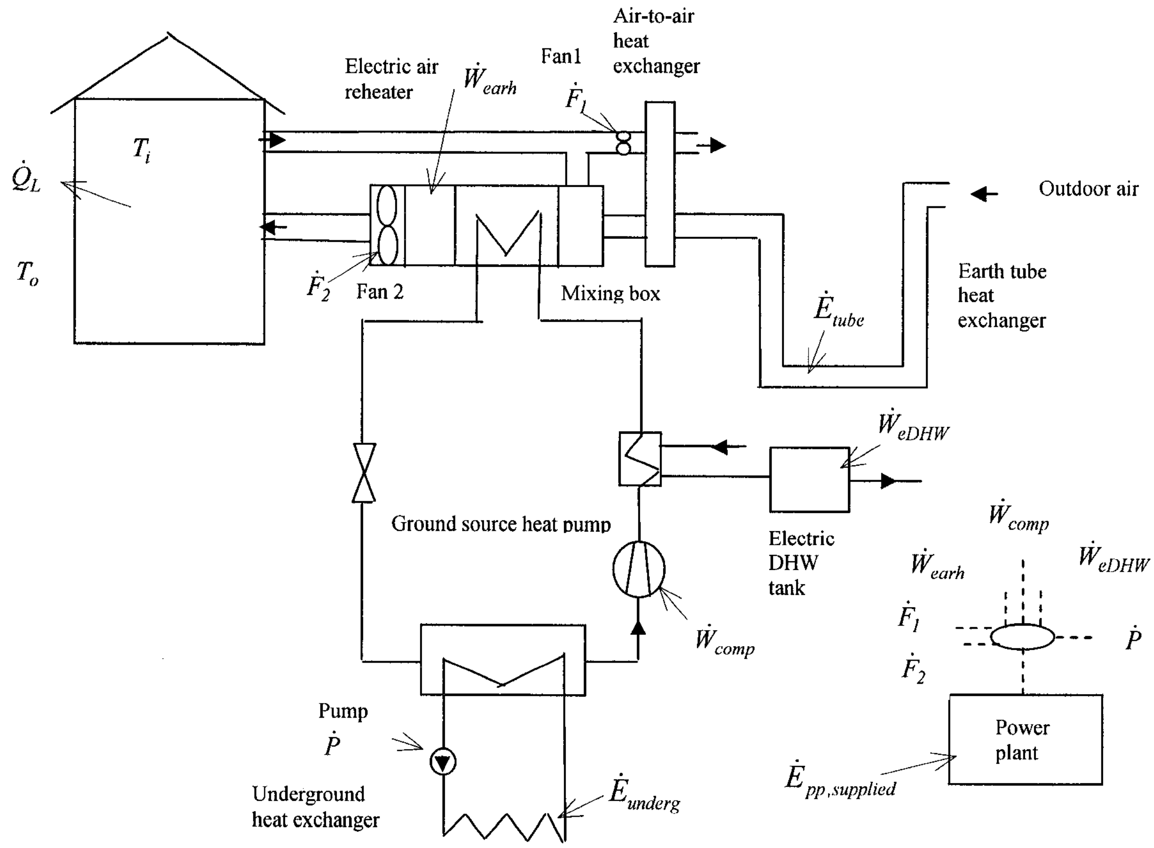

Figure 4.12 Configuration of design alternative No.12

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,eDHW} - T_{w,in,cond}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}92)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{P} + \dot{F}_1 + \dot{F}_2 + \dot{W}_{comp} + \dot{W}_{eDHW} + \dot{W}_{earh}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}93)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{underg} + \dot{E}_{tube}$ (kW) (4-94)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-95)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,earh} + \dot{S}_{gen,mix} + \dot{S}_{gen,eDHW} + \dot{S}_{gen,exhaust}$$
$$+ \dot{S}_{gen,fan} + \dot{S}_{pump} + \dot{S}_{gen,inside} + \dot{S}_{gshp} + \dot{S}_{underg} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}96)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-97)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear}$$
$$+ \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \left| \dot{E}_{underg} \cdot (1 - TK_o / TK_{w,ag}) \right| \quad \text{(kW)} \qquad (4\text{-}98)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-99)

4.2.13 Design alternative No.13

Design alternative No.13 is generated from the design alternative No.12, by replacing the electric air reheater and electric DHW tank with the hot water air reheater and hot water DHW tank respectively (Figure 4.13). The hot water is produced by a gas-fired boiler.

In this case, the following conditions apply:

$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1}$ (kg/s)

$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,wheater} = \dot{m}_{a,warh} = \dot{m}_{a,fan2} = \dot{m}_{a,cond}$ (kg/s)

$T_{a,in,tube} = T_o$ (°C); $T_{a,in,HE} = T_{a,out,tube}$ (°C); $T_{a,out,HE} = T_{a,in,mix}$ (°C); $T_{a,out,mix} = T_{a,in,wheater}$ (°C);

$T_{a,out,wheater} = T_{a,in,warh}$ (°C); $T_{a,out,warh} = T_{a,supplied}$ (°C); $T_{ea,in,HE} = T_i$ (°C); $T_{ea,out,HE} = T_{ea,out}$ (°C)

$T_{w,in,warh} = T_{w,in,hDHW} = T_{w,out,gboiler}$ (°C); $T_{w,out,warh} = T_{w,out,hDHW} = T_{w,in,gboiler}$ (°C)

$T_{w,out,cond} = T_{w,in,DHW}$ (°C)

$\dot{Q}_{pump1} = \dot{E}_{underg}$ (kW); $\dot{Q}_{pump2} = \dot{Q}_{warh} + \dot{Q}_{DHW}$ (kW)

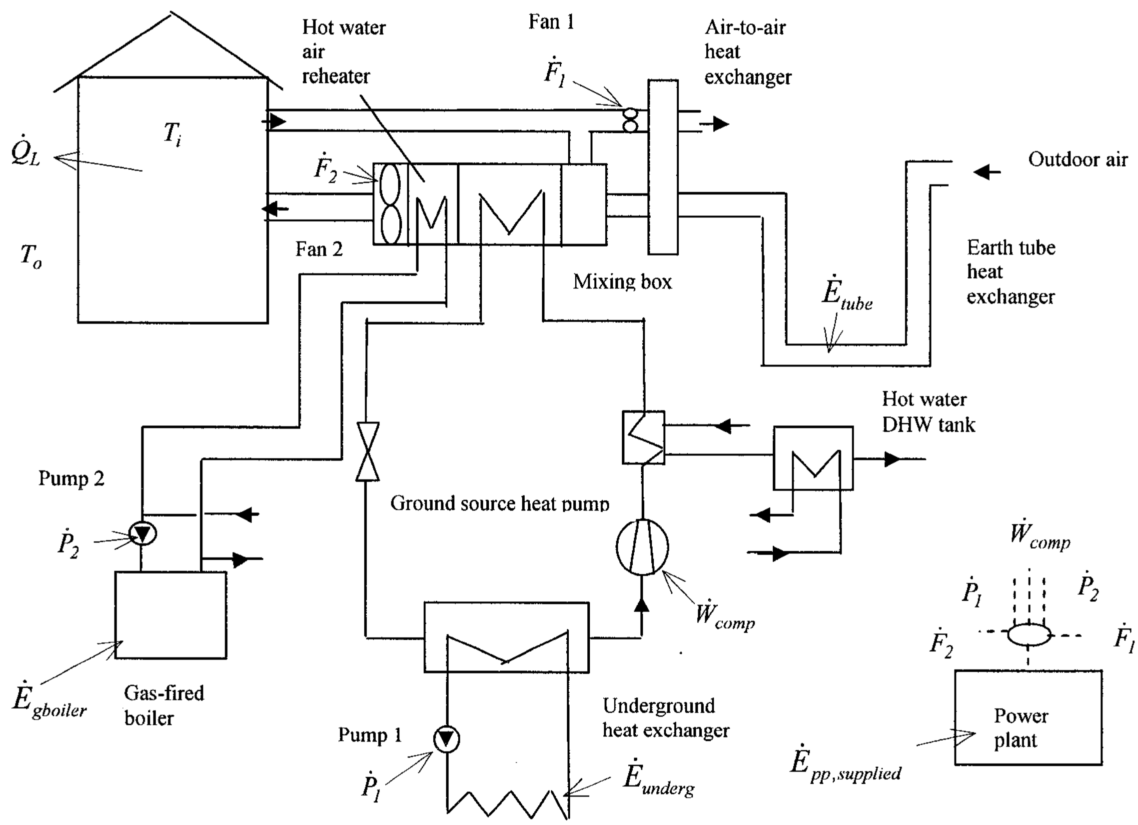

Figure 4.13 Configuration of design alternative No.13

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,cond}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}100)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{P}_1 + \dot{P}_2 + \dot{F}_1 + \dot{F}_2 + \dot{W}_{comp}) / \eta_{trans} \quad \text{(kW)} \tag{4-101}$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{underg} + \dot{E}_{tube}$ (kW) 4-102)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-103)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,warh} + \dot{S}_{gen,mix} + \dot{S}_{gen,DHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan}$$
$$+ \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,inside} + \dot{S}_{gen,underg} + \dot{S}_{gen,gshp} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \tag{4-104}$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-105)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame})$$
$$+ \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \left| \dot{E}_{underg} \cdot (1 - TK_o / TK_{w,ag}) \right| \quad \text{(kW)} \tag{4-106}$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-107)

4.2.14 Design alternative No.14

This design alternative integrates an air source heat pump and forced air system for space heating. Outdoor fresh air is brought in through the earth tube heat exchanger, air-to-air heat exchanger. Domestic hot water is first preheated by the condenser of ASHP and then reheated by electricity. Figure 4.14 shows the integration of design alternative No.14.

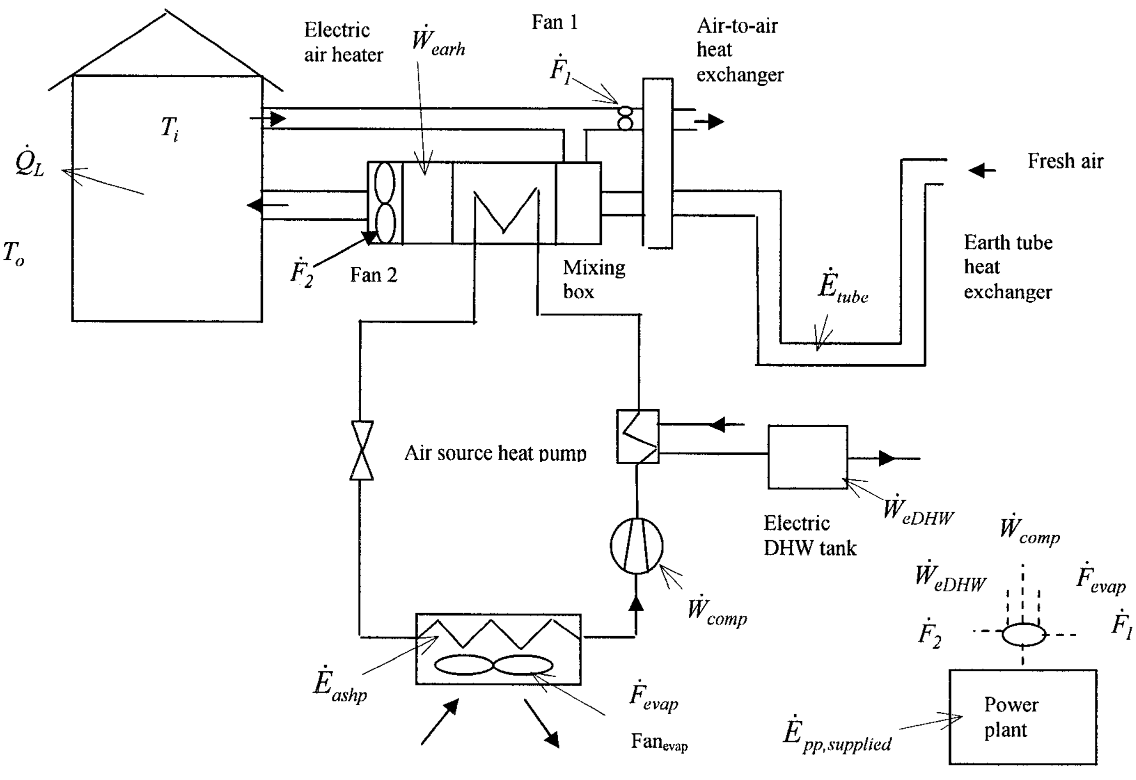

Figure 4.14 Configuration of design alternative No.14

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} \text{ (kg/s)}$$

$$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,cond} = \dot{m}_{a,earh} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o\,(^{\circ}\mathrm{C});\; T_{a,in,HE} = T_{a,out,tube}\,(^{\circ}\mathrm{C});\; T_{a,out,HE} = T_{a,in,mix}\,(^{\circ}\mathrm{C});\; T_{a,out,mix} = T_{a,in,cond}\,(^{\circ}\mathrm{C});$$

$$T_{a,out,cond} = T_{a,in,earh}\,(^{\circ}\mathrm{C}); T_{a,out,earh} = T_{a,supplied}\,(^{\circ}\mathrm{C});\; T_{ea,in,HE} = T_i\,(^{\circ}\mathrm{C});\; T_{ea,out,HE} = T_{ea,out}\,(^{\circ}\mathrm{C});$$

$$T_{w,out,cond} = T_{w,in,DHW}\,(^{\circ}\mathrm{C})$$

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,cond}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \tag{4-108}$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{W}_{comp} + \dot{W}_{earh} + \dot{W}_{eDHW}) / \eta_{trans} \quad \text{(kW)} \tag{4-109}$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{ashp} + \dot{E}_{tube}$ (kW) (4-110)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-111)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,earh} + \dot{S}_{gen,mix} + \dot{S}_{gen,eDHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,inside} + \dot{S}_{gen,ashp} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \tag{4-112}$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \left| \dot{E}_{ashp} \cdot (1 - TK_o / TK_{a,a}) \right| \quad \text{(kW)} \tag{4-113}$$

where $TK_{a,a} = 273.15 + (T_{a1} + T_{a2}) / 2$ (K) (4-114)

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-115)

4.2.15 Design alternative No.15

Design alternative No.15 is generated from the design alternative No. 14 by replacing the electric air reheater and electric DHW tank by hot water air reheater and hot water DHW tank respectively (shown in Figure 4.15), is generated. The hot water is produced by a gas-fired boiler.

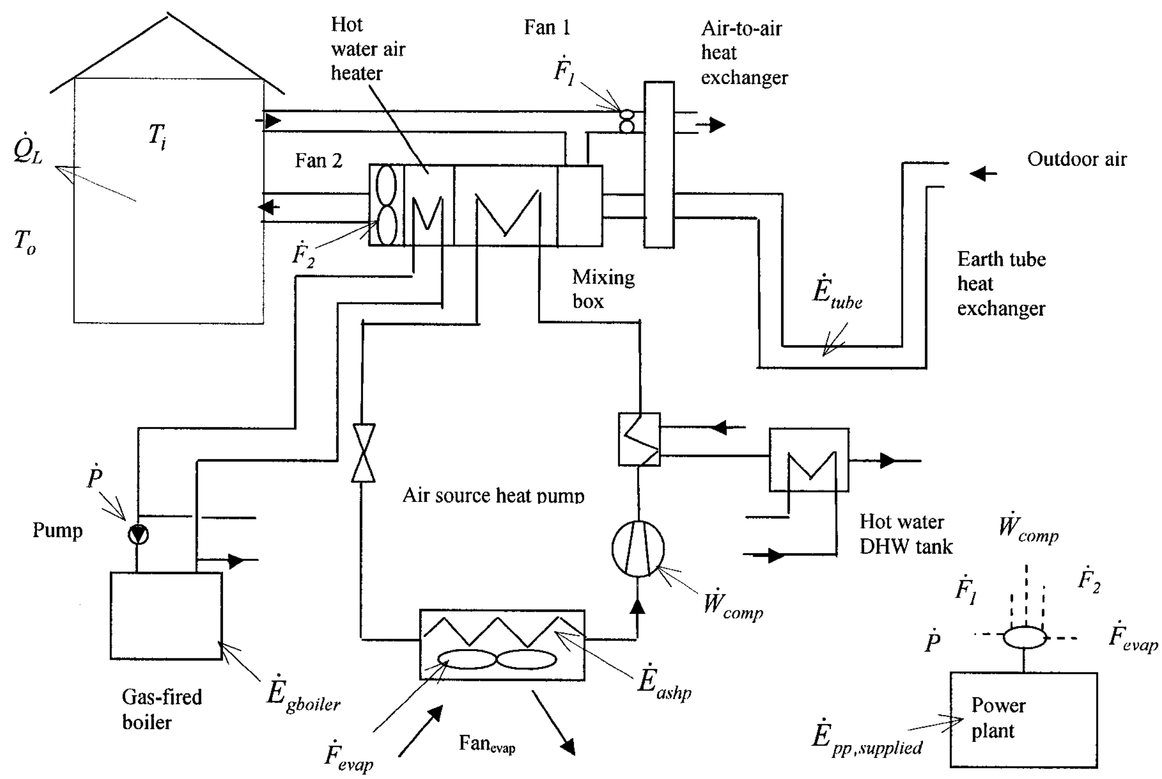

Figure 4.15 Configuration of design alternative No.15

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} \text{ (kg/s)}$$

$$\dot{m}_{a,return} + \dot{m}_{a,HE} = \dot{m}_{a,mix} = \dot{m}_{a,cond} = \dot{m}_{a,warh} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o\,(^{\circ}\mathrm{C}); T_{a,in,HE} = T_{a,out,tube}\,(^{\circ}\mathrm{C}); T_{a,out,HE} = T_{a,in,mix}\,(^{\circ}\mathrm{C}); T_{a,out,mix} = T_{a,in,cond}\,(^{\circ}\mathrm{C};$$

$$T_{a,out,cond} = T_{a,in,warh}\,(^{\circ}\mathrm{C}); T_{a,out,warh} = T_{a,supplied}\,(^{\circ}\mathrm{C}); T_{ea,in,HE} = T_i\,(^{\circ}\mathrm{C});\; T_{ea,out,HE} = T_{ea,out}\,(^{\circ}\mathrm{C});$$

$T_{w,in,warh} = T_{w,in,hDHW} = T_{w,out,gboiler}\,(^{o}C);\; T_{w,out,warh} = T_{w,out,hDHW} = T_{w,in,gboiler}\,(^{o}C)$

$T_{w,out,cond} = T_{w,in,DHW}$ (^{o}C)

$\dot{Q}_{pump} = \dot{Q}_{warh} + \dot{Q}_{DHW}$ (kW)

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,cond}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}116)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{F}_{evap} + \dot{P} + \dot{W}_{comp}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}117)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{ashp} + \dot{E}_{tube} + \dot{E}_{gboiler}$ (kW) (4-118)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-119)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,warh} + \dot{S}_{gen,mix} + \dot{S}_{gen,DHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,inside} + \dot{S}_{gen,ashp} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}120)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-121)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame})$$

(kW) (4-122)

$$+ \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \left| \dot{E}_{ashp} \cdot (1 - TK_o / TK_{a,a}) \right|$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-123)

4.2.16 Design alternative No.16

This design alternative integrates radiant heating floor and electric boiler for space heating; earth tube heat exchanger, air-to-air heat exchanger, and electric air heater for ventilation; and electric DHW tank for DHW heating. Figure 4.16 shows the integration of design alternative No.16.

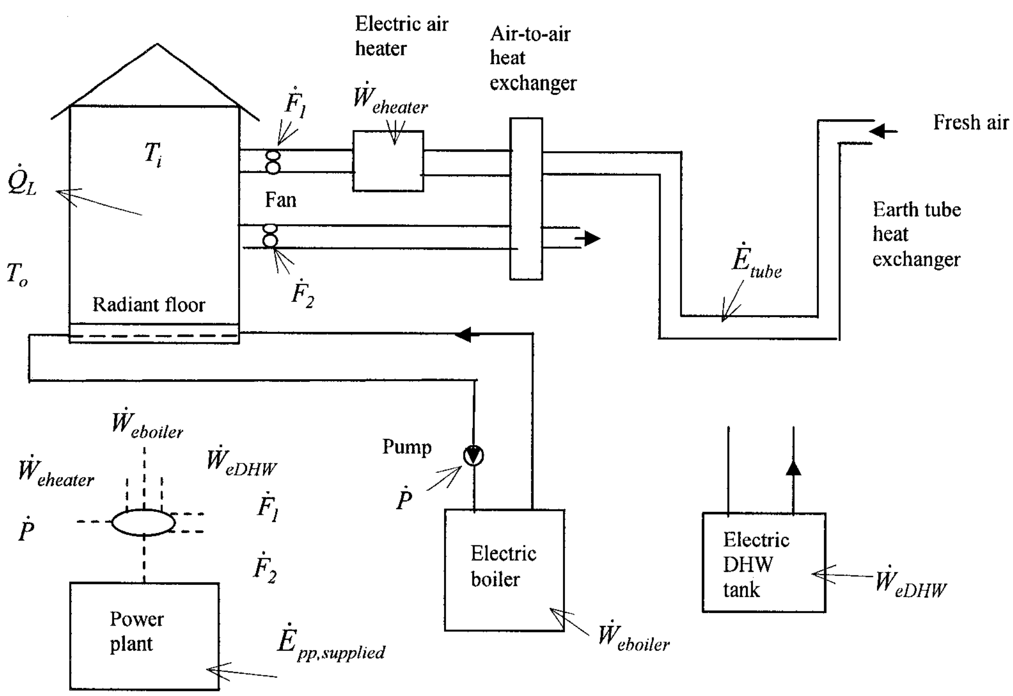

Figure 4.16 Configuration of design alternative No.16

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2}\,(\text{kg/s})$$

$$\dot{m}_{w,floor} = \dot{m}_{w,eboiler}\,(\text{kg/s})$$

$$T_{a,in,tube} = T_o\,(^{o}\text{C});\ T_{a,in,HE} = T_{a,out,tube}\ (^{o}\text{C});\ T_{a,out,HE} = T_{a,in,eheater}\ (^{o}\text{C});\ T_{a,out,eheater} = T_i\ (^{o}\text{C})$$

$$T_{ea,in,HE} = T_i\,(^{o}\text{C});\ T_{ea,out,HE} = T_{ea,out}\,(^{o}\text{C})$$

$$T_{w,in,floor} = T_{w,out,eboiler}\,(^{o}\text{C});\ T_{w,out,floor} = T_{w,in,eboiler}\,(^{o}\text{C})$$

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o)\ \ (\text{kW}) \qquad (4\text{-}124)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P} + \dot{W}_{comp} + \dot{W}_{eDHW} + \dot{W}_{eheater}) / \eta_{trans} \quad (\text{kW}) \qquad (4\text{-}125)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{tube}$ (kW) (4-126)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-127)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,eheater} + \dot{S}_{gen,eDHW} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,floor} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad (\text{kW/K}) \qquad (4\text{-}128)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-129)

Total exergy supply from various energy sources

$$\dot{X}_{supplied}=(\dot{E}_{pp,gas}+\dot{E}_{pp,oil}+\dot{E}_{pp,coal})\cdot(1-TK_o/TK_{flame})$$

$$+\dot{E}_{pp,nuclear}+\dot{E}_{pp,hydro}+\dot{E}_{tube}\cdot(1-TK_o/TK_{ground}) \qquad \text{(kW)} \qquad (4\text{-}130)$$

Exergy efficiency $\eta_2 = 1-\dot{X}_{de,total}/\dot{X}_{supplied}$ (4-131)

4.2.17 Design alternative No.17

This design alternative integrates radiant heating floor and gas-fired boiler with water-to-water heat exchanger for space heating; earth tube heat exchanger, air-to-air heat exchanger, and electric air heater for ventilation; and hot water DHW tank for DHW heating. Figure 4.17 shows the integration of design alternative No.17.

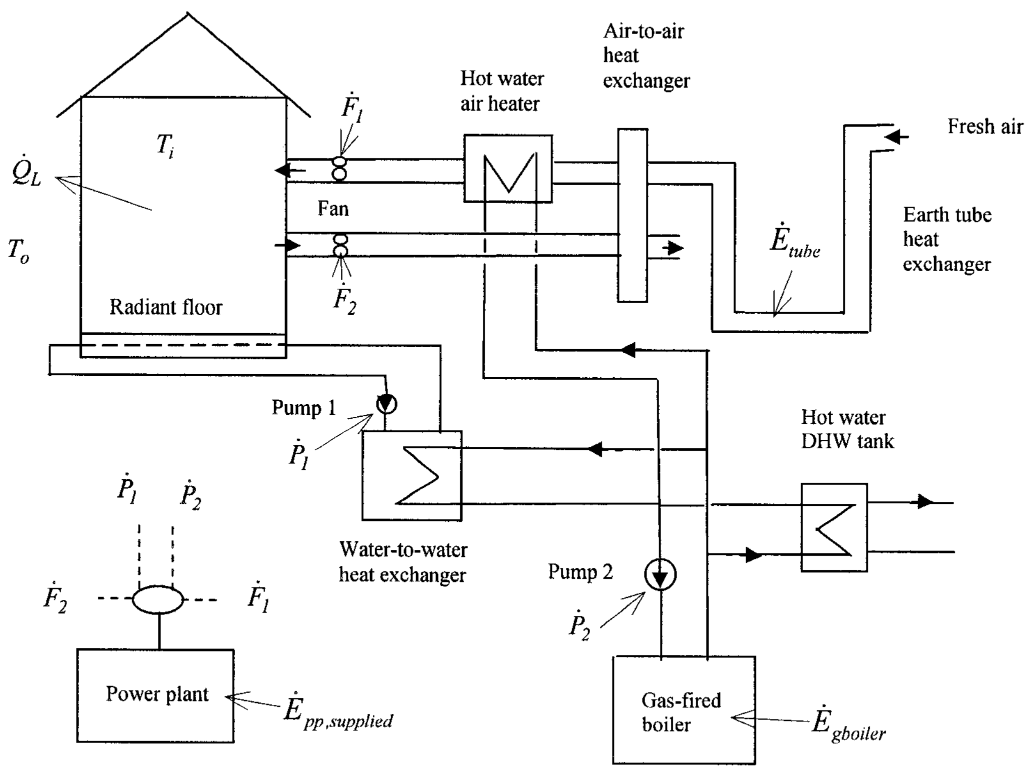

Figure 4.17 Configuration of design alternative No.17

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \quad \text{(kg/s)}$$

$$\dot{m}_{w,floor} = \dot{m}_{w,wHE} \quad \text{(kg/s)}$$

$$T_{a,in,tube} = T_o \ (^oC); \ T_{a,in,HE} = T_{a,out,tube} \ (^oC); \ T_{a,out,HE} = T_{a,in,wheater} \ (^oC);$$

$$T_{a,out,wheater} = T_i \ (^oC); \ T_{ea,in,HE} = T_i \ (^oC); \ T_{ea,out,HE} = T_{ea,out} \ (^oC)$$

$$T_{w,in,floor} = T_{w,out,wHE} \ (^oC); \ T_{w,out,floor} = T_{w,in,wHE} \ (^oC)$$

$$T_{w,in,hwHE} = T_{w,in,wheater} = T_{w,in,hDHW} = T_{w,out,gboiler} \ (^oC)$$

$$T_{w,out,hwHE} = T_{w,out,wheater} = T_{w,out,hDHW} = T_{w,in,gboiler} \ (^oC)$$

$$\dot{Q}_{pump1} = \dot{Q}_L \ \text{(kW)}; \ \dot{Q}_{pump2} = \dot{Q}_L + \dot{Q}_{wheater} + \dot{Q}_{DHW} \ \text{(kW)}$$

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,DHW}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \quad \text{(kW)} \qquad (4\text{-}132)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P}_1 + \dot{P}_2) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}133)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{tube}$ (kW) (4-134)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-135)

<u>Entropy analysis</u>

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,wHE} + \dot{S}_{gen,floor} + \dot{S}_{gen,DHW}$$

(kW/K) (4-136)

$$+ \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp}$$

<u>Exergy analysis</u>

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-137)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear}$$

(kW) (4-138)

$$+ \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame})$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-139)

4.2.18 Design alternative No.18

Design alternative No.18 is generated by adding an economizer to the gas-fired boiler of design alternative No.17. This economizer is used to recover the heat from flue gases to preheat the domestic hot water.

In this case, the following conditions apply:

$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan,2}$ (kg/s)

$\dot{m}_{w,floor} = \dot{m}_{w,wHE}$ (kg/s)

$T_{a,in,tube} = T_o$ (^oC); $T_{a,in,HE} = T_{a,out,tube}$ (^oC); $T_{a,out,HE} = T_{a,in,wheater}$ (^oC);

$T_{a,out,wheater} = T_i$ (^oC); $T_{ea,in,HE} = T_i$ (^oC); $T_{ea,out,HE} = T_{ea,out}$ (^oC)

$T_{flueg,in,econ} = T_{flueg,out,gboiler}$ (^oC); $T_{flueg,in,gboiler} = T_o$ (^oC)

$$T_{w,in,floor} = T_{w,out,wHE}\ (^{o}C); T_{w,out,floor} = T_{w,in,wHE}\ (^{o}C)$$

$$T_{w,in,hwHE} = T_{w,in,wheater} = T_{w,in,hDHW} = T_{w,out,gboiler}\ (^{o}C)$$

$$T_{w,out,hwHE} = T_{w,out,wheater} = T_{w,out,hDHW} = T_{w,in,gboiler}\ (^{o}C)$$

$$T_{w,out,econ} = T_{w,in,DHW}\ (^{o}C)$$

$$\dot{Q}_{pump1} = \dot{Q}_L \text{ (kW)}; \quad \dot{Q}_{pump2} = \dot{Q}_L + \dot{Q}_{wheater} + \dot{Q}_{DHW} \text{ (kW)}$$

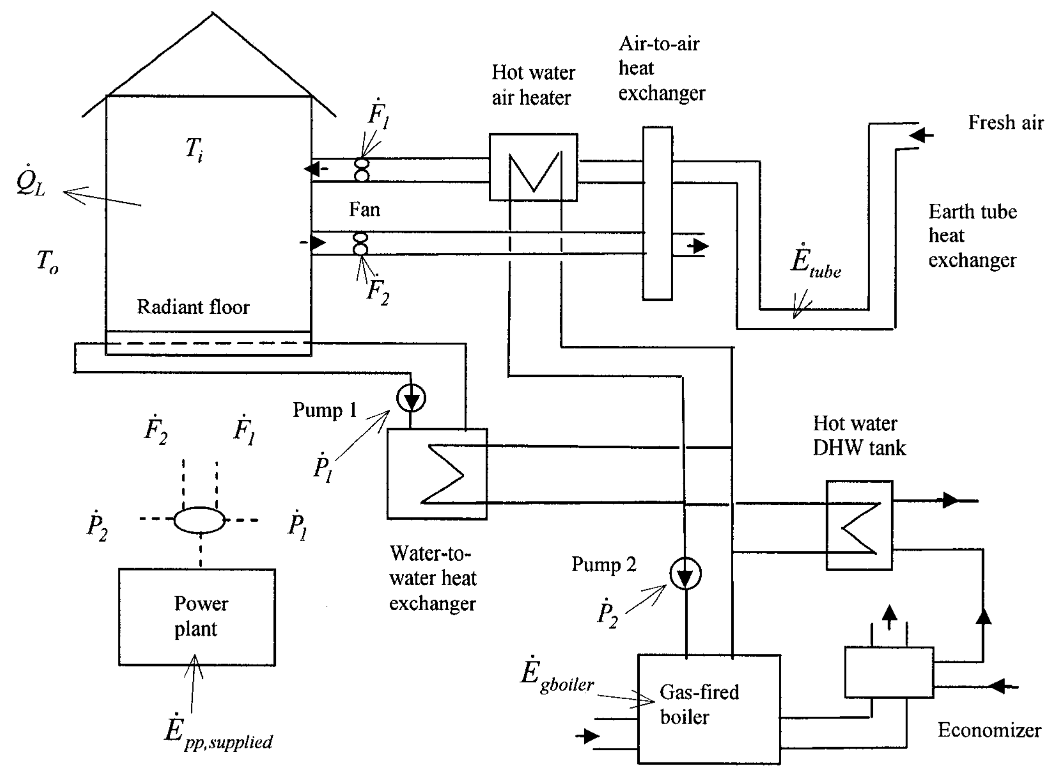

Figure 4.18 Configuration of design alternative No.18

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,econ}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \text{ (kW)} \qquad (4\text{-}140)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{F}_1 + \dot{F}_2 + \dot{P}_1 + \dot{P}_2) / \eta_{trans} \quad \text{(kW)} \tag{4-141}$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{tube}$ (kW) (4-142)

Energy efficiency $\eta_1 = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-143)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,wHE} + \dot{S}_{gen,floor} + \dot{S}_{gen,DHW}$$
$$+ \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,econ} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \tag{4-144}$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-145)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear}$$
$$+ \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) \quad \text{(kW)} \tag{4-146}$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-147)

4.2.19 Design alternative No.19

Figure 4.19 presents the integration of design alternative No.19. The space heating is provided by the radiant heating floor with a ground source heat pump. Outdoor ventilation air is processed through an earth tube heat exchanger, an air-to-air heat exchanger and a hot water air heater. Domestic hot water is preheated and heated by GSHP condenser and electricity respectively.

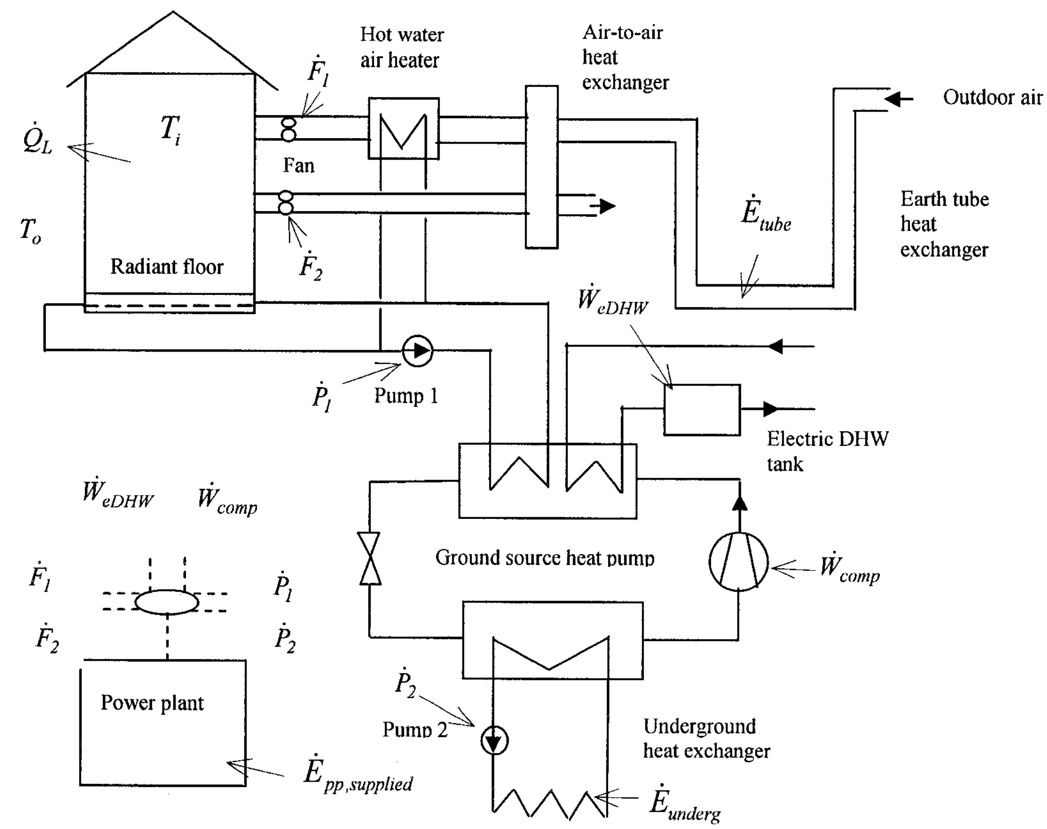

Figure 4.19 Configuration of design alternative No.19

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{(kg/s)}$$

$$\dot{m}_{w,floor} + \dot{m}_{w,wheater} = \dot{m}_{w,cond} \text{(kg/s)}$$

$$\dot{m}_{w,DHW} = m_{Dw,cond} \text{(kg/s)}$$

$$T_{a,in,tube} = T_o (^{o}\text{C});\ T_{a,in,HE} = T_{a,out,tube}\ (^{o}\text{C});\ T_{a,out,HE} = T_{a,in,wheater}\ (^{o}\text{C});\ T_{a,out,wheater} = T_i\ (^{o}\text{C})$$

$$T_{ea,in,HE} = T_i (^{o}\text{C});\ T_{ea,out,HE} = T_{ea,out} (^{o}\text{C})$$

$T_{Dw,out,cond} = T_{w,in,eDHW}$ (°C)

$T_{w,in,cond} = T_{w,out,floor}$ (°C); $T_{w,out,cond} = T_{w,in,floor}$ (°C)

$\dot{Q}_{pump1} = \dot{Q}_L + \dot{Q}_{wheater}$ (kW); $\dot{Q}_{pump2} = \dot{E}_{underg}$ (kW)

Energy analysis

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,cond1}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \text{ (kW)} \qquad (4\text{-}148)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{P}_1 + \dot{P}_2 + \dot{F}_1 + \dot{F}_2 + \dot{W}_{comp} + \dot{W}_{eDHW}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}149)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{underg} + \dot{E}_{tube}$ (kW) (4-150)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-151)

Entropy analysis

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,eDHW} + \dot{S}_{gen,underg} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gshp} + \dot{S}_{gen,floor} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \qquad (4\text{-}152)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-153)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear}$$

$$+ \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \left| \dot{E}_{underg} \cdot (1 - TK_o / TK_{w,ag}) \right| \quad \text{(kW)} \qquad (4\text{-}154)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-155)

4.2.20 Design alternative No.20

Design alternative No.20, which is shown in Figure 4.20, is generated based on the following consideration: the electric DHW tank used in the design alternative No.19 is replaced by a hot water DHW tank with gas-fired boiler.

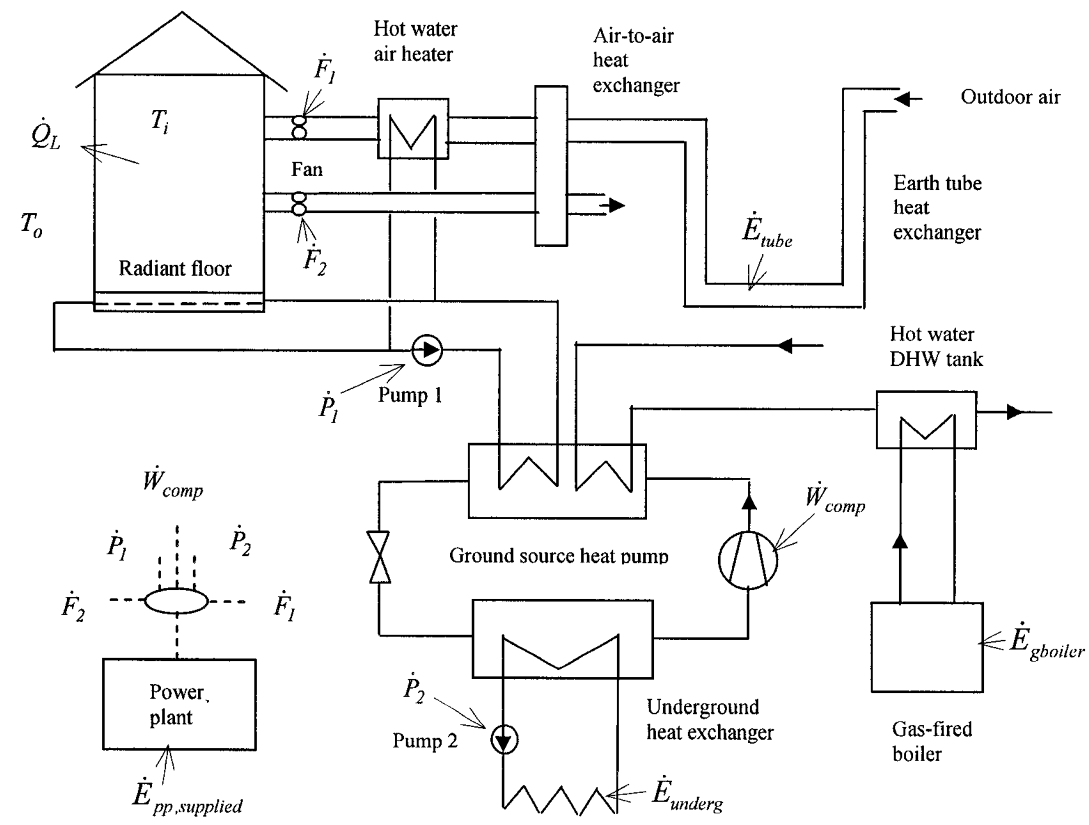

Figure 4.20 Configuration of design alternative No.20

In this case, the following conditions apply:

$$\dot{m}_{a,fresh} = \dot{m}_{a,tube} = \dot{m}_{a,HE} = \dot{m}_{a,wheater} = \dot{m}_{ea,HE} = \dot{m}_{ea,out} = \dot{m}_{a,fan1} = \dot{m}_{a,fan2} \text{ (kg/s)}$$

$$\dot{m}_{w,floor} + \dot{m}_{w,wheater} = m_{w,cond} \text{ (kg/s)}; \ \dot{m}_{w,DHW} = m_{Dw,cond} \text{ (kg/s)}$$

$$T_{a,in,tube} = T_o \ (^{o}\mathrm{C}); \ T_{a,in,HE} = T_{a,out,tube} \ (^{o}\mathrm{C}); \ T_{a,out,HE} = T_{a,in,wheater} \ (^{o}\mathrm{C}); \ T_{a,out,wheater} = T_i \ (^{o}\mathrm{C})$$

$$T_{ea,in,HE} = T_i \ (^{o}\mathrm{C}); \ T_{ea,out,HE} = T_{ea,out} \ (^{o}\mathrm{C})$$

$$T_{Dw,out,cond} = T_{w,in,DHW} \ (^{o}\mathrm{C})$$

$$T_{w,in,cond} = T_{w,out,floor} = T_{w,out,wheater} (^{o}\mathrm{C}); \ T_{w,out,cond} = T_{w,in,floor} = T_{w,in,wheater} (^{o}\mathrm{C})$$

$$T_{w,in,gboiler} = T_{w,out,hDHW} \ (^{o}\mathrm{C}); \ T_{w,out,gboiler} = T_{w,in,hDHW} \ (^{o}\mathrm{C})$$

$$\dot{Q}_{pump1} = \dot{Q}_L + \dot{Q}_{wheater} \text{ (kW)}; \ \dot{Q}_{pump2} = \dot{E}_{underg} \text{ (kW)}$$

<u>Energy analysis</u>

Total useful energy

$$\dot{E}_{useful} = \dot{Q}_L + \dot{m}_{w,DHW} \cdot (T_{w,out,DHW} - T_{w,in,cond1}) + \dot{m}_{a,fresh} \cdot c_{pa} \cdot (T_i - T_o) \text{ (kW)} \qquad (4\text{-}156)$$

Total electricity demand including electricity transmission loss

$$\dot{W}_{pp} = (\dot{P}_1 + \dot{P}_2 + \dot{F}_1 + \dot{F}_2 + \dot{W}_{comp}) / \eta_{trans} \quad \text{(kW)} \qquad (4\text{-}157)$$

Total primary energy supply $\dot{E}_{supplied} = \dot{E}_{pp,supplied} + \dot{E}_{gboiler} + \dot{E}_{underg} + \dot{E}_{tube}$ (kW) (4-158)

Energy efficiency $\eta_I = \dot{E}_{useful} / \dot{E}_{supplied}$ (4-159)

<u>Entropy analysis</u>

Total entropy generation

$$\dot{S}_{gen,total} = \dot{S}_{gen,tube} + \dot{S}_{gen,HE} + \dot{S}_{gen,wheater} + \dot{S}_{gen,DHW} + \dot{S}_{gen,underg} + \dot{S}_{gen,exhaust} + \dot{S}_{gen,fan} + \dot{S}_{gen,pump} + \dot{S}_{gen,gshp} + \dot{S}_{gen,floor} + \dot{S}_{gen,gboiler} + \dot{S}_{gen,trans} + \dot{S}_{gen,pp} \quad \text{(kW/K)} \quad (4\text{-}160)$$

Exergy analysis

Total exergy destruction $\dot{X}_{de,total} = TK_o \cdot \dot{S}_{gen,total}$ (kW) (4-161)

Total exergy supply from various energy sources

$$\dot{X}_{supplied} = \dot{E}_{gboiler} \cdot (1 - TK_o / TK_{flame}) + (\dot{E}_{pp,gas} + \dot{E}_{pp,oil} + \dot{E}_{pp,coal}) \cdot (1 - TK_o / TK_{flame}) + \dot{E}_{pp,nuclear} + \dot{E}_{pp,hydro} + \dot{E}_{tube} \cdot (1 - TK_o / TK_{ground}) + \left| \dot{E}_{underg} \cdot (1 - TK_o / TK_{w,ag}) \right| \quad \text{(kW)} \quad (4\text{-}162)$$

Exergy efficiency $\eta_2 = 1 - \dot{X}_{de,total} / \dot{X}_{supplied}$ (4-163)

CHAPTER 5

SIMULATION RESULTS AND DISCUSSIONS

A house located in Montreal with the total floor area of 310 m^2 is used as a case study. The peak and hourly heating loads used in this study were obtained by using the BLAST program (Kassab et al, 2003). A series of simulation programs were developed on the EES (engineering equation solver) (Klein, 2003) platform to perform the second law analysis for the models of the HVAC-DHW components and systems presented in Chapter 3 and 4. These programs were used to evaluate the performance of all the design alternatives mentioned in Chapter 4 under both winter peak design and annual operating conditions in Montreal. The results are presented in the following sections. Sample programs are presented in Appendix A, B and C for the design alternative No.1, No.8, and No.20.

5.1 Overall performance of the HVAC-DHW systems

The overall performance of the selected HVAC-DHW systems will be compared using the following criteria: energy efficiency, exergy efficiency, entropy generation, exergy destruction, energy demand, and exergy demand at both winter peak design and annual operating conditions. The best HVAC-DHW systems are those with the highest overall exergy efficiency, that is, with minimum entropy generation and exergy destruction. In this study, both on-site use of energy resources and off-site use of energy resources for the electricity generation are evaluated. Table 5.1 shows the input values of the parameters used in the simulations.

Table 5.1 Input values of the parameters in the simulations

Items	Parameters	Input values
House	Indoor design air temperature	T_i=21°C
	outdoor design air temperature	T_o=-23°C
	Heating load	$\dot{Q}_L$=11.1 kW
	Area of the house	A=310m^2
	Height of the room	H=2.8m
Ventilation systems	Ventilation air change rate	β=0.35 ach
	Sensible heat recovery efficiency of air-to-air heat exchanger	η_{HE}=60%
	Dimensions of the earth tube heat exchanger	L=10m,
		bxb=0.25m x 0.25m
Heating systems	Energy efficiency of gas-fired boiler	$\eta_{gboiler}$=75%
	Temperature of flue gases at the outlet of gas-fired boiler	$T_{flueg,out,boiler}$=230°C
	Flame temperature	TK_{flame}=2200 K
	Heat recovery efficiency of boiler economizer	η_{econ}=45%
	Temperature of water entering the boiler	$T_{w,in,boiler}$=70°C
	Temperature of water leaving the boiler	$T_{w,out,boiler}$=90°C
	Ground temperature	T_{groung}=8°C
DHW systems	Mass flow rate of domestic hot water	$\dot{m}_{w,DHW}$=0.0105 kg/s
	Temperature of domestic hot water leaving DHW tank	$T_{w,out,DHW}$=60°C
	Temperature of the supply water from main	T_w=8°C
Electricity generation	Contribution of different energy sources to the off-site generation of electricity in Quebec	α_{hydro}=96.7%
		$\alpha_{nuclear}$=1.1%
		α_{gas}=1.1%
		α_{oil}=1.1%
		α_{coal}=0%
	Energy efficiencies of power plant for different energy sources	η_{hydro}=80%
		$\eta_{nuclear}$=30%
		η_{gas}=43.1%
		η_{oil}=33%
		η_{coal}=37%
Electricity transmission lines	Efficiency of electricity transmission	η_{trans}=86%

Table 5.2 and table 5.3 present the overall simulation results of the second law analysis for design alternatives No.1 to 20 at winter peak design conditions and annual heating operating conditions, respectively.

Table 5.2 Overall results of the second law analysis at peak design conditions

Alt.	η_1 %	η_2 %	$\dot{S}_{gen,total}$ kW/K	$\dot{Q}_{useful}$ kW	$\dot{Q}_{\text{supplied}}$ kW	$\dot{X}_{\text{supplied}}$ kW	$\dot{X}_{\text{destroyed}}$ kW
1	65.9	10.3	0.0724	13.38	20.30	20.19	18.11
2	74.4	13.0	0.0555	13.38	18.00	15.97	13.89
3	80.9	23.7	0.0267	13.38	16.54	8.77	6.69
4	64.2	7.4	0.1045	18.22	28.37	28.22	26.14
5	76.0	8.7	0.0870	18.22	23.97	23.85	21.77
6	73.1	9.9	0.0817	18.22	24.91	22.68	20.43
7	81.2	12.4	0.0635	18.22	22.43	18.13	15.88
8	**90.2**	**15.3**	**0.0547**	**18.22**	**20.20**	**16.15**	**13.68**
9	70.5	9.5	0.0854	18.22	25.83	23.57	21.35
10	78.0	11.8	0.0672	18.22	23.35	19.05	16.80
11	86.2	14.5	0.0583	18.22	21.12	17.07	14.59
12	77.6	13.3	0.0584	18.22	23.46	16.85	14.60
13	81.5	15.2	0.0500	18.22	22.36	14.76	12.52
14	76.0	13.1	0.0620	18.22	23.98	17.84	15.51
15	79.6	14.8	0.0537	18.22	22.88	15.76	13.42
16	72.7	9.8	0.0823	18.22	25.06	22.83	20.58
17	80.5	12.2	0.0643	18.22	22.63	18.33	16.09
18	89.5	15.2	0.0553	18.22	20.35	16.30	13.83
19	88.6	22.4	0.0312	18.22	20.57	10.04	7.80
20	**88.5**	**23.2**	**0.0304**	**18.22**	**20.58**	**9.89**	**7.60**

Table 5.3 Overall results of the second law analysis at annual operating conditions

Alt.	η_1 %	η_2 %	$S_{gen,total}$ kWh/K	Q_{useful} kWh	$Q_{supplied}$ kWh	$X_{supplied}$ kWh	$X_{destroyed}$ kWh
1	65.9	6.8	102.6	19655	29809	29644	27630
2	73.2	8.5	80.3	19655	26869	23655	21641
3	74.8	12.0	55.3	19655	26268	16969	14930
4	61.6	4.1	174.7	30343	49260	48989	46974
5	76.7	5.1	138.6	30343	39535	39317	37302
6	74.2	5.7	130.9	30343	40908	37415	35272
7	80.3	7.0	106.1	30343	37796	30762	28622
8	**90.1**	**8.4**	**92.3**	**30343**	**33689**	**27161**	**24873**
9	73.6	5.7	132.2	30343	41246	37751	35608
10	80.3	7.0	106.3	30343	37779	30789	28645
11	89.2	8.3	93.5	30343	34027	27497	25210
12	79.4	7.7	95.7	30343	38236	27968	25807
13	81.8	8.5	85.8	30343	37107	25318	23158
14	80.7	9.0	87.3	30343	37587	25847	23519
15	83.2	10.0	77.5	30343	36458	23197	20869
16	73.0	5.6	133.3	30343	41571	38074	35931
17	79.2	6.8	108.3	30343	38326	31333	29190
18	88.3	8.2	94.7	30343	34352	27820	25532
19	83.7	10.0	72.0	30343	36244	21631	19459
20	**83.9**	**10.9**	**68.5**	**30343**	**36154**	**20788**	**18515**

Design alternative No.1 using electric baseboard heaters for space heating and electric domestic water heater is used as the base case, because it is the most used/installed HVAC-DHW system in Quebec. There is no mechanical ventilation system. The energy efficiencies of the base case design alternative are 65.9% at both peak design conditions and annual operating conditions. The exergy efficiencies are only 10.3% and 6.8%, respectively.

The following comparison is conducted between design alternative No.4 to No.20, where there are mechanical ventilation systems. Design alternative No.1 to No.3 with no mechanical ventilation systems are used to evaluate the ventilation effect on the overall performance. Design alternative No.8 has the best energy performance, with the highest energy efficiencies of 90.2% (at peak design conditions) and 90.1% (at annual operating conditions). It also has the lowest overall power demand of 20.2 kW and the lowest annual energy use of 33689 kWh. However, exergy and entropy analysis tells a different story: the design alternative No.20 has the best exergy performance. Alternative No.20 has the highest exergy efficiency of 23.2% at winter peak design conditions and 10.9% at annual operating conditions. It has the lowest exergy demand of 9.89 kW and 20788 kWh, and lowest entropy generation of 0.03038 kW/K and 68.54 kWh/K at peak design conditions and annual operating conditions, respectively.

Table 5.4 presents the overall comparison between alternative No.8 and alternative No.20. It can be seen that at the peak design conditions, the energy efficiency of alternative No.8 is higher than No.20 by 1.7%, but the exergy efficiency of alternative No.8 is less than No.20 by 7.9%. At the annual operating conditions, the energy efficiency of No.8 exceeds No.20 by 6.3%, but the exergy efficiency of alternative No.8 is 2.5% less than alternative No.20. Except its higher energy demand, the entropy generation, exergy destruction, and exergy demand of alternative No.20 are all less than those of No.8. The reason should be that alternative No.20 integrates the use of geothermal energy, in which the content of exergy is relatively small. In addition, it is also noted that the difference of exergy efficiency between the two alternatives at annual operating conditions (2.5%) is much less than that at the peak design conditions (7.9%). This is due to the part load operation of the ground source heat pump most of the time, which affects the exergy efficiency of alternative No.20.

Table 5.4 Overall comparisons between the design alternatives No. 8 and No. 20

Alternative No.	Conditions	η_1 %	η_2 %	Energy supplied	Entropy generation	Exergy destruction	Exergy supplied
No.8	Peak design conditions	90.2	15.3	20.20 kW	0.05467 kW/K	13.68 kW	16.15 kW
	Annual operating conditions	90.1	8.4	33689 kWh	92.27 kWh/K	24873 kWh	27161 kWh
No.20	Peak design conditions	88.5	23.2	20.58 kW	0.03038 kW/K	7.60 kW	9.89 kW
	Annual operating conditions	83.9	10.9	36154 kWh	68.54 kWh/K	18515 kWh	20788 kWh

The comparison between alternative No.8 and No.20 also demonstrates that exergy related parameters (exergy efficiency, entropy generation, exergy destruction, exergy demand) are more meaningful and comparable than energy related parameters (energy efficiency, energy demand), when the supplied energy are from different sources.

5.2 Performance of individual components within the selected HVAC-DHW systems

Table 5.5 show the entropy generation in each component of the design alternatives No. 1 to 20 at winter peak design conditions, while Table 5.6 shows the corresponding exergy destruction. Table 5.7 presents the contribution (in percentage) of each component to the total entropy generation for each design alternative. The performance of each component is analyzed at winter peak design conditions based on these results.

Table 5.5a Entropy analysis results at winter peak design conditions

	Entropy generation (kW/K)									
	Heating systems									
Alt. No.	Gas boiler	Econ	Electric boiler	GSHP	Underg HEx	ASHP	Pumps	Baseboard	RADF	WWHEx
1	--	--	--	--	--	--	--	0.03774	--	--
2	0.04773	--	--	--	--	--	0.00040	0.00622	--	--
3	--	--	--	0.00965	0.00010	--	0.00183	--	0.00112	--
4	--	--	--	--	--	--	--	0.03774	--	--
5	--	--	--	--	--	--	--	0.03774	--	--
6	--	--	--	--	--	--	--	0.03774	--	--
7	0.05123	--	--	--	--	--	0.00040	0.00622	--	--
8	0.04010	0.00313	--	--	--	--	0.00040	0.00622	--	--
9	--	--	--	--	--	--	--	--	--	--
10	0.05123	--	--	--	--	--	0.00040	--	--	--
11	0.04010	0.00313	--	--	--	--	0.00040	--	--	--
12	--	--	--	0.00749	0.00005	--	0.00046	--	--	--
13	0.02439	--	--	0.00749	0.00005	--	0.00086	--	--	--
14	--	--	--	--	--	0.00910	--	--	--	--
15	0.02439	--	--	--	--	0.00910	0.00040	--	--	--
16	--	--	0.03661	--	--	--	0.00040	--	0.00112	--
17	0.05123	--	--	--	--	--	0.00080	--	0.00112	0.00509
18	0.04010	0.00313	--	--	--	--	0.00080	--	0.00112	0.00509
19	--	--	--	0.01033	0.00010	--	0.00127	--	0.00112	--
20	0.00392	--	--	0.01033	0.00010	--	0.00167	--	0.00112	--

Table 5.5b Entropy analysis results at winter peak design conditions

Entropy generation (kW/K)											
Alt. No.	Ventilation components								DHW	Power generation and transmission	
	ETHEx	AAHEx	Air heater	Fans	Exhaust air	Mixing box	Inside house	Air reheater	DHW tank	Power trans.	Power plant
1	--	--	--	--	--	--	--	--	0.00745	0.00871	0.01851
2	--	--	--	--	--	--	--	--	0.00097	0.00007	0.00014
3	--	--	--	--	--	--	--	--	0.00343	0.00340	0.00722
4	--	--	0.01779	0.00196	0.00153	--	--	--	0.00745	0.01217	0.02587
5	--	0.00068	0.00678	0.00196	0.00027	--	--	--	0.00745	0.01029	0.02186
6	0	0.00016	0.00339	0.00196	0.00077	--	--	--	0.00745	0.00967	0.02055
7	0	0.00016	0.00060	0.00196	0.00077	--	--	--	0.00097	0.00038	0.00082
8	0	0.00016	0.00060	0.00196	0.00077	--	--	--	0.00014	0.00038	0.00082
9	0	0.00016	0.03936	0.00439	0.00077	0.00004	0.00173	--	0.00745	0.01006	0.02139
10	0	0.00016	0.00506	0.00439	0.00077	0.00004	0.00173	--	0.00097	0.00078	0.00166
11	0	0.00016	0.00506	0.00439	0.00077	0.00004	0.00173	--	0.00014	0.00078	0.00166
12	0	0.00016	--	0.00439	0.00077	0.00004	0.00173	0.01820	0.00343	0.00693	0.01474
13	0	0.00016	--	0.00439	0.00077	0.00004	0.00173	0.00190	0.00031	0.00255	0.00541
14	0	0.00016	--	0.00494	0.00077	0.00004	0.00173	0.01820	0.00343	0.00765	0.01607
15	0	0.00016	--	0.00494	0.00077	0.00004	0.00173	0.00190	0.00031	0.00318	0.00675
16	0	0.00016	0.00339	0.00196	0.00077	--	--	--	0.00745	0.00973	0.02069
17	0	0.00016	0.00060	0.00196	0.00077	--	--	--	0.00097	0.00061	0.00100
18	0	0.00016	0.00060	0.00196	0.00077	--	--	--	0.00014	0.00045	0.00095
19	0	0.00016	0.00015	0.00196	0.00077	--	--	--	0.00343	0.00380	0.00807
20	0	0.00016	0.00015	0.00196	0.00077	--	--	--	0.00031	0.00315	0.00674

Table 5.6a Exergy analyses at winter peak design conditions

Exergy destruction (kW)										
Alt. No.	Heating components									
	Gas boiler	Econ	Electric boiler	GSHP	Underg HEx	ASHP	Pumps	Baseboard	RADF	WWHEx
1	--	--	--	--	--	--	--	9.44	--	--
2	11.94	--	--	--	--	--	0.10	1.56	--	--
3	--	--	--	2.41	0.02	--	0.46	--	0.28	--
4	--	--	--	--	--	--	--	9.44	--	--
5	--	--	--	--	--	--	--	9.44	--	--
6	--	--	--	--	--	--	--	9.44	--	--
7	12.81	--	--	--	--	--	0.10	1.56	--	--
8	10.03	0.78	--	--	--	--	0.10	1.56	--	--
9	--	--	--	--	--	--	--	--	--	--
10	12.81	--	--	--	--	--	0.10	--	--	--
11	10.03	0.78	--	--	--	--	0.10	--	--	--
12	--	--	--	1.87	0.01	--	0.11	--	--	--
13	6.10	--	--	1.87	0.01	--	0.21	--	--	--
14	--	--	--	--	--	2.28	--	--	--	--
15	6.10	--	--	--	--	2.28	0.10	--	--	--
16	--	--	9.16	--	--	--	0.10	--	0.28	--
17	12.81	--	--	--	--	--	0.20	--	0.28	1.27
18	10.03	0.78	--	--	--	--	0.20	--	0.28	1.27
19	--	--	--	2.59	0.03	--	0.32	--	0.28	--
20	0.98	--	--	2.59	0.03	--	0.42	--	0.28	--

Table 5.6b Exergy analyses at winter peak design conditions

Exergy destruction (kW)											
Alt. No.	Ventilation components								DHW	Power generation and transmission	
	ETHEx	AAHEx	Air heater	Fans	Exhaust air	Mixing box	Inside house	Air reheater	DHW tank	Power trans.	Power plant
1	--	--	--	--	--	--	--	--	1.87	2.18	4.63
2	--	--	--	--	--	--	--	--	0.24	0.02	0.03
3	--	--	--	--	--	--	--	--	0.86	0.85	1.81
4	--	--	4.45	0.49	0.38	--	--	--	1.87	3.04	6.47
5	--	0.17	1.70	0.49	0.07	--	--	--	1.87	2.57	5.47
6	0	0.04	0.85	0.49	0.19	--	--	--	1.87	2.42	5.14
7	0	0.04	0.15	0.49	0.19	--	--	--	0.24	0.10	0.20
8	0	0.04	0.15	0.49	0.19	--	--	--	0.03	0.10	0.20
9	0	0.04	9.85	1.10	0.19	0.01	0.43	--	1.87	2.52	5.35
10	0	0.04	1.27	1.10	0.19	0.01	0.43	--	0.24	0.20	0.41
11	0	0.04	1.27	1.10	0.19	0.01	0.43	--	0.03	0.20	0.41
12	0	0.04	--	1.10	0.19	0.01	0.43	4.55	0.86	1.73	3.69
13	0	0.04	--	1.10	0.19	0.01	0.43	0.48	0.08	0.64	1.35
14	0	0.04	--	1.24	0.19	0.01	0.43	4.55	0.86	1.89	4.02
15	0	0.04	--	1.24	0.19	0.01	0.43	0.48	0.08	0.79	1.69
16	0	0.04	0.85	0.49	0.19	--	--	--	1.87	2.43	5.18
17	0	0.04	0.15	0.49	0.19	--	--	--	0.24	0.15	0.25
18	0	0.04	0.15	0.49	0.19	--	--	--	0.03	0.11	0.24
19	0	0.04	0.04	0.49	0.19	--	--	--	0.86	0.95	2.02
20	0	0.04	0.04	0.49	0.19	--	--	--	0.08	0.79	1.69

Table 5.7a Contribution (%) of each component to the overall entropy generation for each design alternative

Alt. No.	Total entropy generation kW/K	Distribution of entropy generation (%)									
		Heating components									
		Gas boiler	Econ	Electric boiler	GSHP	Underg HEx	ASHP	Pumps	Base.	RADF	WWHEx
1	0.07241	--	--	--	--	--	--	--	52.1	--	--
2	0.05552	86.0	--	--	--	--	--	0.7	11.2	--	--
3	0.02674	--	--	--	36.1	0.4	--	6.8	--	4.2	--
4	0.10450	--	--	--	--	--	--	--	36.1	--	--
5	0.08702	--	--	--	--	--	--	--	43.4	--	--
6	0.08168	--	--	--	--	--	--	--	46.3	--	--
7	0.06350	80.7	--	--	--	--	--	0.6	9.8	--	--
8	0.05467	73.3	5.7	--	--	--	--	0.7	11.4	--	--
9	0.08535	--	--	--	--	--	--	--	--	--	--
10	0.06717	76.3	--	--	--	--	--	0.6	--	--	--
11	0.05834	68.7	5.4	--	--	--	--	0.7	--	--	--
12	0.05838	--	--	--	12.8	0.1	--	0.8	--	--	--
13	0.05004	48.8	--	--	15.1	0.1	--	1.7	--	--	--
14	0.06200	--	--	--	--	--	14.6	--	--	--	--
15	0.05366	45.5	--	--	--	--	17.0	0.7	--	--	--
16	0.08228	--	--	44.5	--	--	--	0.5	--	1.4	--
17	0.06430	79.8	--	--	--	--	--	1.2	--	1.7	7.9
18	0.05527	72.6	5.7	--	--	--	--	1.4	--	2.0	9.2
19	0.03116	--	--	--	33.1	0.3	--	4.1	--	3.6	--
20	0.03038	12.9	--	--	34.0	0.3	--	5.5	--	3.7	--

Table 5.7b Contribution (%) of each component to the overall entropy generation for each design

Alt. No.	Total entropy generation kW/K	Distribution of entropy generation (%)										
		Ventilation components								DHW	Power generation and transmission	
		ETHEx	AAHEx	Air heater	Fans	Exhaust air	Mixing box	Inside house	Air reheater	DHW tank	Power trans.	Power plant
1	0.07241	--	--	--	--	--	--	--	--	10.3	12.0	25.6
2	0.05552	--	--	--	--	--	--	--	--	1.7	0.1	0.3
3	0.02674	--	--	--	--	--	--	--	--	12.8	12.7	27.0
4	0.10450	--	--	17.0	1.9	1.5	--	--	--	7.1	11.6	24.8
5	0.08702	--	0.8	7.8	2.2	0.3	--	--	--	8.6	11.8	25.1
6	0.08168	0	0.2	4.1	2.4	0.9	--	--	--	9.1	11.8	25.2
7	0.06350	0	0.3	0.9	3.1	1.2	--	--	--	1.5	0.6	1.3
8	0.05467	0	0.3	1.1	3.6	1.4	--	--	--	0.3	0.7	1.5
9	0.08535	0	0.2	46.1	5.1	0.9	0.1	2.0	--	8.7	11.8	25.1
10	0.06717	0	0.2	7.5	6.5	1.1	0.1	2.6	--	1.4	1.2	2.5
11	0.05834	0	0.3	8.7	7.5	1.3	0.1	3.0	--	0.2	1.3	2.8
12	0.05838	0	0.3	--	7.5	1.3	0.1	3.0	31.2	5.9	11.9	25.1
13	0.05004	0	0.3	--	8.8	1.5	0.1	3.4	3.8	0.6	5.0	10.8
14	0.06200	0	0.3	--	8.0	1.2	0.1	2.8	29.4	5.5	12.3	25.8
15	0.05366	0	0.3	--	9.2	1.4	0.1	3.2	3.5	0.6	5.9	12.6
16	0.08288	0	0.2	4.1	2.4	0.9	--	--	--	9.1	11.8	25.1
17	0.06430	0	0.3	0.9	3.0	1.2	--	--	--	1.5	0.9	1.6
18	0.05527	0	0.3	1.1	3.5	1.4	--	--	--	0.3	0.8	1.7
19	0.03116	0	0.5	0.5	6.3	2.5	--	--	--	11.0	12.2	25.9
20	0.03038	0	0.5	0.5	6.4	2.5	--	--	--	1.0	10.4	22.3

5.2.1 Design alternative No.4

As discussed above, the following comparison is conducted between design alternatives No.4 to No.20, where there are mechanical ventilation systems. Design alternatives No.1 to No.3 with no mechanical ventilation systems were used to evaluate the ventilation effect on the overall performance. For design alternative No.4, at winter peak design conditions, the energy and exergy efficiencies are 64.2% and 7.4% respectively. The total exergy destruction is 26.14 kW. Figure 5.1 presents the breakdown of exergy destruction in the design alternative No.4 at peak design conditions. The power generation and transmission account for 36.4% of the total exergy destruction, and space heating, ventilation and DHW heating account for 36.1%, 20.4%, 7.1% respectively. Therefore, in the case of the design alternative No.4, emphasis should be put on improving the performance in the four sectors in the above sequence: power generation and transmission, space heating, ventilation, and DHW heating. For the power generation and transmission sector, proper selection of the on-site energy sources, such as geothermal energy and natural gas instead of electricity, could decrease the exergy destruction, while for the other three sectors, energy efficient technologies appear more important.

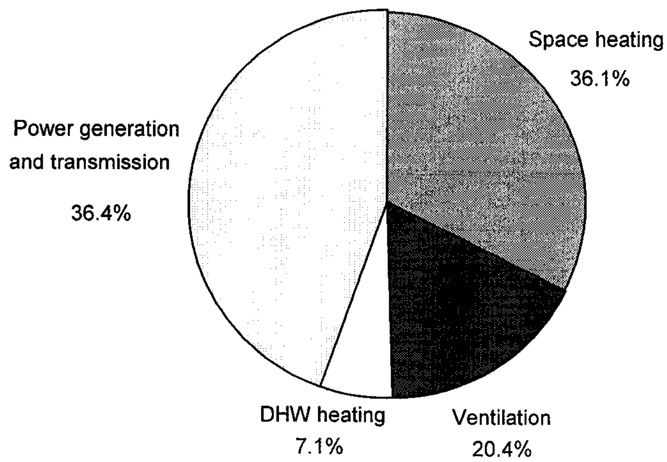

Figure 5.1 Breakdown of exergy destruction in alternative No.4 at peak design conditions (Total exergy destruction is 28.22 kW)

5.2.2 Air-to-air heat exchanger

Compared to design alternative No.4, design alternative No. 5 is equipped with an air-to-air heat exchanger, which contributes to the improvement of both energy and exergy efficiencies to 76.0% and 8.7% respectively. Due to the air-to-air heat exchanger, the exergy destruction in electric air heater, exhaust air and power generation and transmission is decreased by 2.75kW, 0.32W and 1.48 kW respectively, while the increase of exergy destruction in air-to-air heat exchanger is only 0.1709 kW. The total exergy destruction is reduced by 4.37 kW.

5.2.3 Earth tube heat exchanger

The only difference between alternative No.5 and alternative No.6 is the use of an earth tube heat exchanger in alternative No.6. Compared to alternative No.5, the energy efficiency of alternative No.6 (73.1%) is lower, but its exergy efficiency (9.9%) is higher.

This is because earth tube in alternative No.6 makes use of geothermal energy in which the amount of exergy is negligible. The exergy destruction in electric air heater, air-to-air heater exchanger and power generation and transmission is decreased by 0.85 kW, 0.13 kW and 0.48 kW respectively, and in the meantime, exergy destruction in exhaust air is increased by 0.13kW. The total exergy destruction is reduced to 23.31 kW which corresponds to 6.2% less than that in alternative No.5. In addition, the earth tube is helpful in preventing the air-to-air heat exchanger from freezing in very cold conditions (Gieseler, et al, 2002).

5.2.4 Natural gas-fired boiler

In alternative No.7, instead of electricity, hot water with gas-fired boiler is used as the heating energy source. Compared to alternative No.6, the energy and exergy performances of alternative No.6 are all improved (Table 5.8). The major exergy saving occurs in the baseboard heater and air heater. The hot water baseboard heaters consume 1.56 kW exergy which corresponds to 83.5% less exergy than that in the electric baseboard heaters. The exergy destruction in hot water air heater is only 0.15 kW which is 82.2% less than that in electric air heater. The total exergy destruction is reduced to 15.88 kW which corresponds to 22.3% less than that in alternative No.6. Therefore, space heating with a gas-fired boiler is superior to electric heating, from the point of exergy analysis.

Table 5.8 Performance comparison between alternative No. 6 and No. 7

Alternative No.	η_1 %	η_2 %	$\dot{S}_{gen,total}$ kW/K	$\dot{Q}_{supplied}$ kW	$\dot{X}_{supplied}$ kW	$\dot{X}_{destroyed}$ kW
No.6	73.1	9.9	0.08168	24.19	22.68	20.43
No.7	81.2	12.4	0.06350	22.43	18.13	15.88

5.2.5 Boiler economizer

By adding an economizer to the gas-fired boiler of alternative No.7, the energy and exergy efficiencies of alternative No.8 are improved to 90.2% and 15.3%. In alternative No.8, the total exergy destruction of boiler and economizer (10.81kW) is less than that of the boiler without economizer in alternative No.7 (12.81 kW). In addition, the exergy destruction in DHW tank is also decreased from 0.24 kW of alternative No. 7 to 0.03 kW of No.7. The total exergy destruction of the system is reduced by 13.9% compared to alternative No. 7.

Table 5.9 Performance comparison between alternative No. 7 and No. 8

Alternative No.	η_1 %	η_2 %	$\dot{S}_{gen,total}$ kW/K	$\dot{Q}_{supplied}$ kW	$\dot{X}_{supplied}$ kW	$\dot{X}_{destroyed}$ kW
No.7	81.2	12.4	0.06350	22.43	18.13	15.88
No.8	90.2	15.3	0.05467	20.20	16.15	13.68

5.2.6 Forced air system

Comparison between alternative No.6 and No.9 in Table 5.10 shows that the forced air system is less efficient than the baseboard heating system using electricity as the heating energy source. In alternative No.9, the exergy consumption in fans is 1.10 kW which is 1.24 times more than that in alternative No. 6. The fan section contributes to the major extra exergy destruction. The same conclusion can be obtained by comparing alternative No.7 to No.10 and alternative No.8 to No.11.

Table 5.10 Performance comparison between alternative No. 6 and No. 9

Alternative No.	η_1 %	η_2 %	$\dot{S}_{gen,total}$ kW/K	$\dot{Q}_{supplied}$ kW	$\dot{X}_{supplied}$ kW	$\dot{X}_{destroyed}$ kW
No.6	73.1	9.9	0.08168	24.91	22.68	20.43
No.9	70.5	9.5	0.08535	25.83	23.57	21.35

5.2.7 Radiant heating floor

A low temperature radiant heating floor is adopted in design alternatives No.16, No.17, No.18, No.19 and No.20. Compared to the exergy destruction in electric baseboard heater (9.44 kW) and hot water baseboard heater (1.56kW), the exergy destruction in radiant heating floor is only 0.28 kW. This is because of its lower heating temperature. Radiant floor heating is very exergy efficient for space heating. Table 5.11 shows the effect of different energy sources on the radiant heating floor system. The ground source heat pump presents as the most exergy efficient means to match the radiant heating floor. The electric boiler is the last choice.

Table 5.11 Comparison results of on-site energy sources for the radiant heating floor systems

Alternative No.	Radiant floor heating system	η_2
No.16	Electric boiler	9.8%
No.17	Gas-fired boiler	12.2%
No.18	Gas-fired boiler with economizer	15.2%
No.19	Ground source heat pump &electric DHW tank	22.4%
No.20	Ground source heat pump &hot water DHW tank	23.2%

5.2.8 Ground source heat pump

Design alternative No.19 integrates the ground source heat pump into its system. The total exergy destruction in GSHP and underground heat exchanger is 2.61 kW, which is much less than the exergy destruction (12.81 kW) in the gas-fired boiler of alternative No.7. Although there is extra electricity use in the GSHP, the total exergy destruction of alternative No.19 (7.80 kW) is still much less than that of alternative No.7 (13.68kW). That means a low quality geothermal energy is more suitable to the task of space heating than natural gas or electricity. The energy and exergy efficiencies of alternative No.19 are 88.6% and 22.4% respectively.

5.2.9 Air source heat pump (ASHP)

In Montreal, because the outdoor air temperature is not stable and very cold in the winter, the application of ASHP is limited by its capacity. When it is used in the HVAC system, it needs a backup electricity or hot water air reheater. Compared to GSHP, the energy and exergy efficiencies of ASHP are not good. For instance, in alternative No. 14, the energy and exergy efficiencies are 76.0% and 13.1%, respectively.

5.2.10 Domestic hot water (DHW) tank

Table 5.12 presents the exergy consumed by DHW tank which is heated by different on-site energy sources or their combination. From this table, it is found that the DHW tank with the gas-fired boiler with economizer has the best performance, while the electric water heater is the worst. When GSHP is used in the system, GSHP combined with gas-fired water heater is also preferred, which consumes 0.08 kW of exergy.

Table 5.12 Comparison of exergy destruction in DHW tank

Alternative No.	DHW heating	Exergy destruction (kW)
No.16	Electric water heater	1.87
No.17	Gas-fired boiler	0.24
No.18	Gas-fired boiler with economizer	0.03
No.19	GSHP and electric water heater	0.86
No.20	GSHP and gas-fired water heater	0.08

5.2.11 Ventilation system

A ventilation system is used to deliver outdoor air to the rooms to satisfy the indoor air quality requirement. Heating ventilation outdoor air to the indoor air temperature consumes large amount of exergy. Comparison between design alternative No.1 (without ventilation) and No.4 (with ventilation) shows that ventilation air heating accounts for 8.03 kW exergy destruction, which corresponds to 30.4% of the total exergy destruction in alternative No.4 (26.14 kW). Thus, it can be seen that great potential exists in the improvement of the exergy performance of the ventilation system. With the use of an air-to-air heat exchanger and an earth tube heat exchanger, in design alternative No.6, the ventilation air heating accounts for 11.4% of the total exergy destruction (20.43 kW). Comparison between alternative No.2 and No.7 and between alternative No. 3 and No.19 shows that in alternative No. 7, the ventilation air heating accounts for 12.5% of the total exergy destruction (15.88 kW), and in alternative No. 19, the ventilation air heating accounts for 14.2% of the total exergy destruction (7.80 kW).

5.3 Evaluation of the HVAC system.

From the view point of exergy analysis, the objective of the HVAC-DHW system design is to maximize the exergy efficiency and, in the meantime, minimize the entropy generation and the exergy destruction. In the design of the HVAC-DHW system, first of all, the exergy efficient technologies should be applied in the system, and then the system is evaluated based on its exergy performance. From the results of exergy analysis and discussions in the above, air-to-air heat exchanger, earth tube heat exchanger, radiant heating floor, and ground source heat pump are recommended. Gas-fired boiler with economizer is more exergy efficient than without economizer. The integration of radiant heating floor and ground source heat pumps is an excellent way to make use of the low quality geothermal energy to match the low quality energy demand of space heating. An electric heating system is the system with the lowest exergy efficiency.

In the evaluation of the exergy performance of the HVAC system, annual exergy performance reflects the annual operating conditions and is more preferable to that at peak design conditions. The design alternative No.20, which integrates hot water radiant floor, ground source heat pump, air-to-air heat exchanger, earth tube heat exchanger, and gas-fired water heater, is the most exergy efficient among all these design alternatives. It consumes 18515 kW exergy annually, which is 60.6% less than the design alternative No.4. However, it is still not the perfect solution for this specific house, because its exergy efficiency is only 23.2% at peak design conditions and 10.9% at annual operating conditions. It still has some potential to improve its exergy performance. In Tables 5.5, 5.6 and 5.7, the relative inefficient areas can also be detected in each alternative where it is necessary to apply some measures to improve its exergy performance. Figure 5.2 presents the exergy generation in each component of the design alternative No. 20 at winter peak design conditions. Power generation and transmission account for 32.6% of the total exergy destruction. The exergy destruction is unavoidable when electricity is generated far from the residential areas and transmitted to the end users. However, it can be reduced by using the on-site generation of electricity using renewable sources such as solar energy (e.g., by using photovoltaic panels) or wind (e.g., by using wind mills). A ground source heat pump and a gas boiler account for 46.9%, and fans and pumps account for 11.9% of the total entropy generation. Selection of a highly efficient ground source heat pump, gas-fired boiler, fans and pumps is a feasible way to increase the exergy performance in this area.

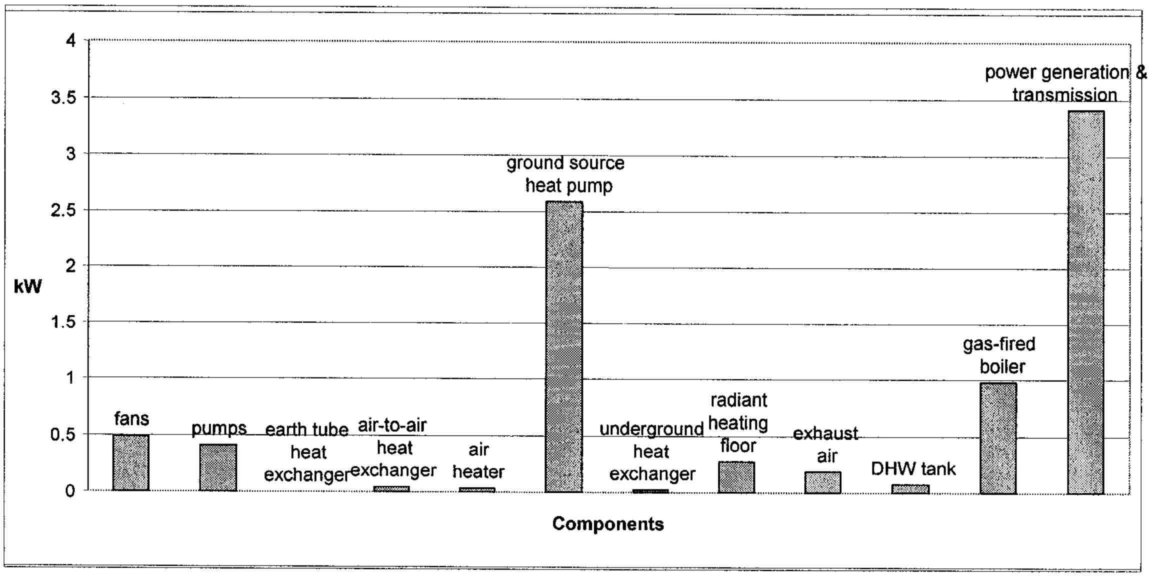

Figure 5.2 Exergy destruction within design alternative No.20 at peak design conditions (Total exergy destruction is 7.60 kW)

CHAPTER 6

EQUIVALENT CO_2 EMISSIONS CORRESPONDING TO THE ENERGY USE IN THE HVAC-DHW SYSTEM

Global emissions of greenhouse gases (GHG), including carbon dioxide (CO_2), methane (CH_4) and nitrous oxides, are increasing, which are believed to be the cause of the global warming. Most GHGs originate from the use of fossil fuels, such as oil, gas, and coal, in factories, vehicles, buildings, and electricity power generating plants. Except for water vapor, CO_2 from combustion of fossil fuels is the largest single source of anthropogenic GHG emissions, accounting for about 80% in the United States and 87% in Canada during the past 50 years (Gentzis 2000). Therefore, in order to mitigate the global warming effect, essential actions should be taken to reduce the equivalent CO_2 emissions. In Canada, the government ratified the Kyoto Protocol on April 29, 1998, with the equivalent CO_2 emissions reduction target being 6% below 1990 levels over the years 2008–2012 (Gentzis 2000).

In this research, two strategies are recommended to reduce energy-related equivalent CO_2 emissions:

- Improvement of the energy and exergy efficiency of the HVAC system;
- Use of fuels with lower CO_2 emissions per unit of useful energy produced.

Equivalent CO_2 emissions are used to be an important indicator to evaluate the environmental impact of a HVAC system. The estimation of equivalent CO_2 emissions due to the annual energy use by the HVAC-DHW systems is presented.

6.1 The annual equivalent CO_2 emissions due to the electricity generation

The annual equivalent CO_2 emissions related to electricity generation are estimated as follows (Gagnon et al, 2002):

$$ECO_2-1=(a_1 \cdot E_{pp,hydro}+a_2 \cdot E_{pp,oil}+a_3 \cdot E_{pp,coal}+a_4 \cdot E_{pp,gas}+a_5 \cdot E_{pp,nuclear})/1000 \text{ (kg } CO_2\text{)} \quad (6\text{-}1)$$

where

$E_{pp,hydro}$, $E_{pp,oil}$, $E_{pp,coal}$, $E_{pp,gas}$, and $E_{pp,nuclear}$ are the annual primary energy use of hydro power, heavy oil, coal, natural gas, and nuclear power, respectively in the generation of electricity, kWh/yr;

a_i (i=1-5) are the equivalent CO_2 emissions due to the use of hydro, heavy oil, coal, natural gas, and nuclear power, respectively, in the generation of electricity, in kt CO_2/TWh (Gagnon et al, 2002);

a_1 =2 kt CO_2/TWh, for run-on-river hydro power plant; 15 kt CO_2/TWh, for hydro power plant with reservoir. In this study, a_1 is selected as 15 kt CO_2/TWh;

a_2 =778 kt CO_2/TWh, for heavy oil power plant;

a_3 =1050 kt CO_2/TWh, for existing coal (1% S) power plant without SO_2scrubbing; 960 kt CO_2/TWh, for modern coal (2% S) power plant with SO_2 scrubbing. In this study, a_3 is selected as 1050 kt CO_2/TWh;

a_4=443 kt CO_2/TWh, for natural gas (+2000km delivery) power plant;

a_5=15 kt CO_2/TWh, for nuclear power plant.

6.2 The annual equivalent CO_2 emissions due to the on-site fossil fuel use

The annual equivalent CO_2 emissions related to on-site fossil fuel use are estimated as follows (Masters, 1998):

$$ECO_2\text{-}2=E_{on\text{-}site}\cdot(\text{pollutant coefficient for } CO_2+\text{pollutant coefficient for } NO_x\cdot GWP_{NOx}+\text{pollutant coefficient for } CH_4\cdot GWP_{CH4})/1000 \quad (\text{kg } CO_2) \qquad (6\text{-}2)$$

where

$E_{on\text{-}site}$ is the annual on-site primary energy use, MJ/yr;

Pollutant coefficients for CO_2, CH_4, and NO_x are presented in Table 6.1.

Global warming potential (GWP) is a weighting factor that enables comparisons to be made between the global warming impact of 1 kg of any greenhouse gas and 1 kg CO_2. GWP is dimensionless and includes a time horizon during which the impact will be felt. The global warming potentials for CH_4 and N_2O with 100-year time horizon are 310 and 21, respectively (Buhl, 1997).

Table 6.1 Pollutant coefficients for CO_2, CH_4, and N_2O (g/MJ) (Buhl, 1997)

	CO_2	CH_4	NO_X
On-site use of natural gas	73.05556	0.157678	0.003669
On-site use of oil	49.44444	0.058889	0.00025
On-site use of coal	86.1111	0.2511	0.0018

6.3 Example of equivalent CO_2 emission calculation

Design alternative No.20 is used as an example to estimate the equivalent CO_2 emissions due to the energy use in the HVAC-DHW system.

The annual equivalent CO_2 emissions due to the electricity generation (ECO_2-1)

In this case, the annual primary energy use of hydro, heavy oil, coal, natural gas, and nuclear power in the electricity generating plant can be obtained from the simulation results:

$E_{pp,hydro}=13535$ kWh/yr; $E_{pp,oil}=369.8$ kWh/yr; $E_{pp,coal}=0$ kWh/yr;

$E_{pp,gas}=283.2$ kWh/yr; and $E_{pp,nuclear}=406.8$ kWh/yr.

$$ECO_2-1=(a_1\cdot E_{pp,hydro}+a_2\cdot E_{pp,oil}+a_3\cdot E_{pp,coal}+a_4\cdot E_{pp,gas}+a_5\cdot E_{pp,nuclear})/1000$$

$$= (15\cdot 13535+778\cdot 369.8+1050\cdot 0+443\cdot 283.2+15\cdot 406.8)/1000$$

$$= 622\ (kg\ CO_2/yr)$$

The annual equivalent CO_2 emission due to the on-site fossil fuel use (ECO_2-2)

In this case, the on-site fossil fuel is natural gas. The annual on-site fossil fuel use is:

$E_{on\text{-}site}=6396\ kWh.$

$ECO_2\text{-}2=E_{on\text{-}site}\cdot$ (pollutant coefficient for CO_2+ pollutant coefficient for $NO_x \cdot GWP_{NOx}$+ pollutant coefficient for $CH_4 \cdot GWP_{CH4}$)/1000

$=6396\cdot(73.05556+0.157678\cdot310+0.003669\cdot21)/1000$

$=780$ (kg CO_2/yr)

<u>The total annual equivalent CO_2 emission due to the energy use in the HVAC-DHW system (ECO_2)</u>

$ECO_2= ECO_2\text{-}1+ ECO_2\text{-}2=622.3+780.4=1043$ (kg CO_2/yr)

Using the calculation method described above, the equivalent CO_2 emissions are estimated for all these design alternatives. Table 6.2 presents the annual equivalent CO_2 emission for each design alternative. Equivalent CO_2 emissions are an important environmental indicator in the evaluation of the HVAC systems. From the comparison of these results, it is found that:

(1) The use of energy efficient technologies, such as air-to-air heat exchanger and earth tube heat exchanger can reduce the annual equivalent CO_2 emissions (design alternative No.4, 5, and 6);

(2) the system with on-site fossil fuel combustion emits more equivalent CO_2 (design alternatives No.2, 7, 10, and 20);

(3) energy cascading use strategies, such as a boiler economizer, can be used to reduce the annual equivalent CO_2 emissions (design alternatives No.7 and 8, No.10 and 11, and No.17 and 18);

(4) GSHP and ASHP can reduce the annual equivalent CO_2 emissions significantly (design alternatives No.3, 12, 13,14 ,15, 19, 20);

(5) design alternative No.19 which emits 904 kg equivalent CO_2 annually has the best environmental performance for the conditions used in this study.

Table 6.2 Calculation results of annual equivalent CO_2 emissions

Alt.	E_{hydro}	E_{oil}	E_{coal}	E_{gas}	$E_{nuclear}$	$E_{on\text{-}site}$	$ECO_2\text{-}1$	$ECO_2\text{-}2$	ECO_2
No.	kWh	kWh	kWh	kWh	kWh	kWh	kg/yr	kg/yr	kg/yr
1	27626	762	0	583	838	0	1278	0	1278
2	613.9	17	0	13	19	26207	28	3198	3226
3	15459	426	0	326	469	0	715	0	715.2
4	45653	1259	0	964	1385	0	2112	0	2112
5	36640	1010	0	774	1111	0	1695	0	1695
6	34670	956	0	732.	1052	0	1604	0	1604
7	3619	100	0	76	110	30400	167	3709	3877
8	3619	100	0	76	110	26294	167	3208	3376
9	34990	965	0	739	1061	0	1619	0	1619
10	3932	108	0	83	119	30045	182	3666	3848
11	3932	108	0	83	119	26294	182	3208	3390
12	25611	706	0	541	777	0	1185	0	1185
13	12486	344	0	264	379	13033	578	1590	2168
14	23808	657	0	503	722	0	1101	0	1101
15	10684	295	0	226	324	13033	494	1590	2084
16	35291	973	0	745	107	0	1618	0	1618
17	4439	122	0	94	135	30045	205	3666	3871
18	4233	117	0	89	128	26294	196	3208	3404
19	19539	539	0	413	593	0	904	0	904
20	13535	370	0	283	407	6396	622	780	1403

CHAPTER 7

SENSITIVITY ANALYSIS

In this research, a sensitivity analysis was performed for three selected design alternatives No.1 (base case), No. 8 (most energy efficient), and No. 20 (most exergy efficient) to assess the impact of the key variables on the exergy performance of the HVAC systems under the winter peak design conditions. The results of sensitivity analysis indicate which parameter is expected to have a large influence on the simulation results. Thus, more effort should be put to ensure the accuracy of these more sensitive parameters. These results will also indicate where more effort should be given to increase the overall performance of the HVAC systems.

In this analysis, ten input variables were chosen to perform the sensitivity analysis for the three selected design alternatives: (1) the efficiency of hydro-electricity generation, (2) the efficiency of electricity transmission, (3) the flame temperature, (4) the efficiency of gas-fired boiler, (5) the heat recovery efficiency of boiler economizer, (6) the ventilation air change rate, (7) the air flow velocity in the earth tube heat exchanger, (8) the length of the earth tube heat exchanger, (9) the heat exchange efficiency of air-to-air heat exchanger, and (10) the ground temperature. The values of each input variable for each design alternative are modified in a reasonable range, while the values of other variables are hold constant. Based on this condition, simulations are performed. The results of these simulations are tabulated in terms of exergy efficiency (η_2). These results show the sensitivity of the overall exergy performance of the HVAC systems with respect to each variable in response to an incremental change in input value.

7.1 Efficiency of hydro-electricity generation

The sensitivity analysis results with respect to the efficiency of hydro-electricity generation (η_{hydro}) is presented in Table 7.1 and Figure 7.1. It is found that η_{hydro} has a significant impact on the overall exergy efficiencies of design alternatives No.1 and No.20, but little impact on the overall exergy efficiency of design alternative No.8. When η_{hydro} increases from 50% to 100%, exergy efficiency increases from 6.61% to 12.66%, from 14.88% to 15.49%, and from 16.37% to 26.93% for design alternative No. 1, No.8, and No.20, respectively. Since the hydro-electricity generation efficiency varies from country to country, and from place to place, in order to ensure the accuracy of the simulation results, for design alternative No.1 and No.20, a large emphasis should put on the estimation of this value. On the other hand, for design alternatives No.1 and No.20, this situation indicates that the improvement of the efficiencies of hydro-electricity generation can save exergy significantly.

Table 7.1 Sensitivity analysis of exergy efficiency (η_2) due to the variation of the efficiency of hydro-electricity generation (η_{hydro})

η_{hydro} (%)	50	60	70	80	90	100
η_2 (%) (No.1)	6.61	7.86	9.09	10.30	11.49	12.66
η_2 (%) (No. 8)	14.88	15.08	15.22	15.33	15.42	15.49
η_2 (%) (No.20)	16.37	18.83	21.10	23.19	25.13	26.93

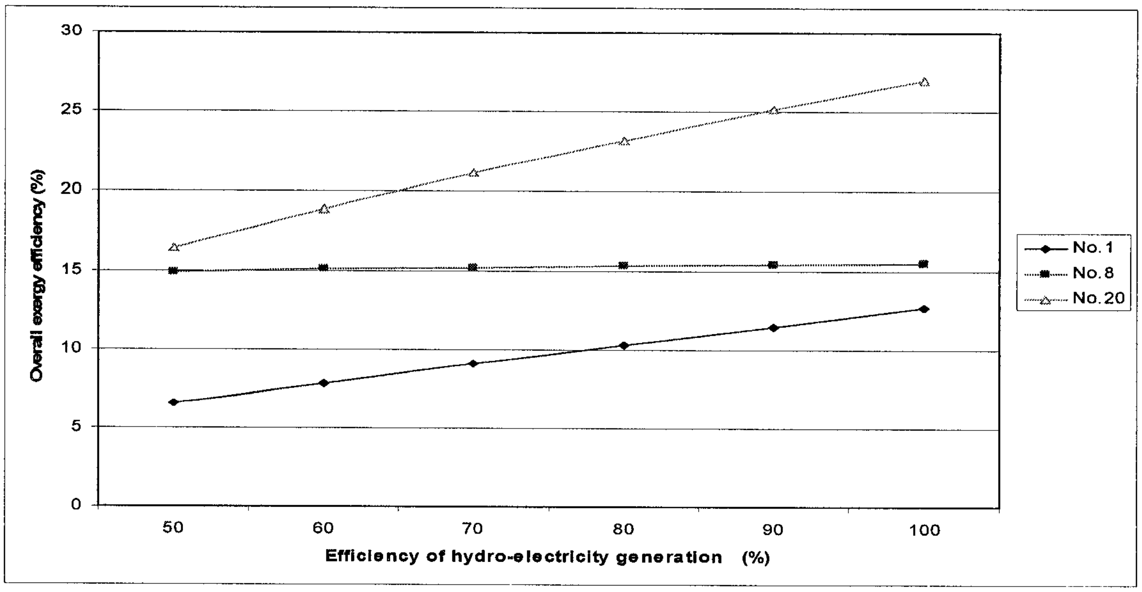

Figure 7.1 Overall exergy efficiency (η_2) versus efficiency of hydro-electricity generation (η_{hydro})

7.2 Efficiency of electricity transmission

A large amount of exergy is consumed during the transmission of electricity. Hence, the electricity transmission efficiency (η_{trans}) has a large influence on the simulation results when electricity is used as the principle energy source. Table 7.2 and Figure 7.2 present the sensitivity analysis results with respect to the electricity transmission efficiency (η_{trans}). It is found that the exergy performance of the HVAC systems is very sensitive to η_{trans} for design alternatives No.1 and No.20, but less sensitive to η_{trans} for design alternative No.8. When η_{trans} is increased from 40% to 100%, exergy efficiency increases from 4.79% to 11.98%, from 14.42% to 15.45%, and from 12.81% to 25.81% for design alternative No. 1, No.8, and No.20, respectively. These results indicate that, for design alternative No. 1, No.20, the accuracy of this input value is very important on a proper simulation results. On the other hand, the results show that the decentralization of electricity generation (to reduce the transmission losses) can significantly increase the exergy performance of the HVAC systems for design alternative No. 1, and No.20.

Table 7.2 Sensitivity analysis of exergy efficiency (η_2) due to the variation of the efficiency of electricity transmission (η_{trans})

η_{trans} (%)	40	50	60	70	80	86	90	100
η_2 (%) (No.1)	4.79	5.99	7.19	8.38	9.58	10.30	10.78	11.98
η_2 (%) (No. 8)	14.42	14.75	14.98	15.14	15.27	15.33	15.37	15.45
η_2 (%) (No.20)	12.81	15.34	17.70	19.92	22.00	23.19	23.96	25.81

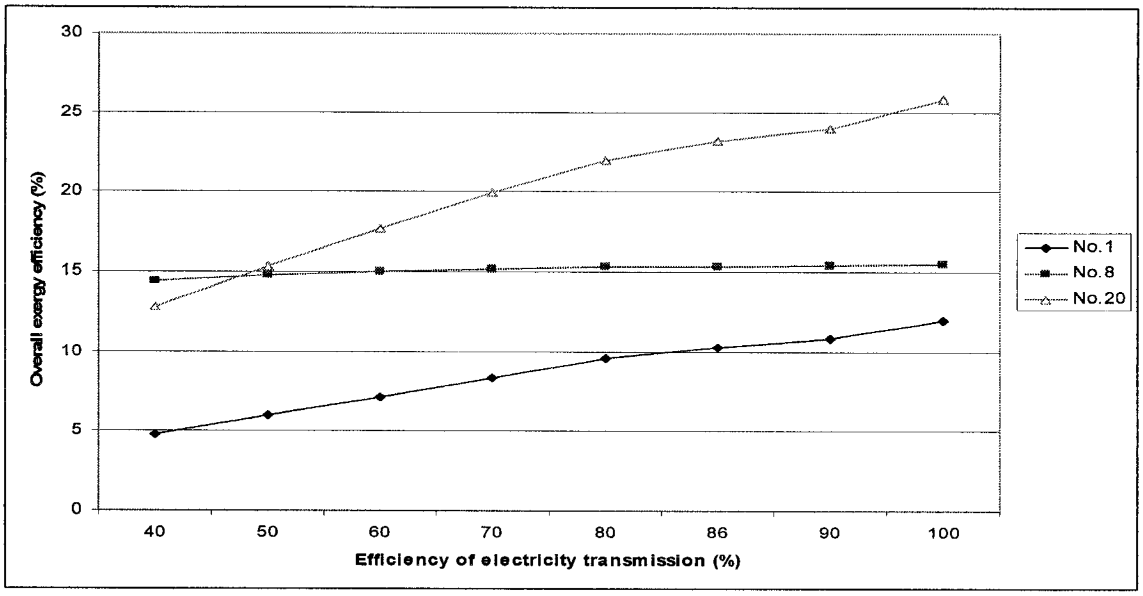

Figure 7.2 Overall exergy efficiency (η_2) versus efficiency of electricity transmission (η_{trans})

7.3 Flame temperature

In Quebec, the hydropower plant accounts for 96.7% of the off-site electricity generation and the thermal power plants contribute to only 3.3% of the off-site electricity generation (Baouendi, 2003). Therefore, the impact of the flame temperature on the overall exergy efficiency is very low. However, if the electricity is produced mostly by thermal power plants, for instance in the case of Alberta where coal power plants account for 84%, oil power plant for 8%, and natural gas power plant for 8% of the off-site electricity generation (Baouendi, 2003), the impact of the flame temperature is obvious. Table 7.3 shows the sensitivity analysis results with respect to the flame temperature (TK_{flame}). When TK_{flame} is increased from 1700 K to 2500 K, with an increment of 100K, the exergy efficiency decreases from 5.81% to 5.50%, from 15.22% to 14.44%, and from 14.54% to 13.84% for design alternative No.1, No.8, and No.20 respectively.

Table 7.3 Sensitivity analysis of exergy efficiency (η_2) due to the variation of the flame temperature (TK_{flame})

TK_{flame} (K)	1700	1800	1900	2000	2100	2200	2300	2400	2500
η_2 (%) (No. 1)	5.81	5.75	5.70	5.66	5.62	5.59	5.56	5.53	5.50
η_2 (%) (No. 8)	15.22	15.08	14.95	14.84	14.74	14.65	14.57	14.50	14.44
η_2 (%) (No. 20)	14.54	14.41	14.30	14.20	14.12	14.04	13.96	13.90	13.84

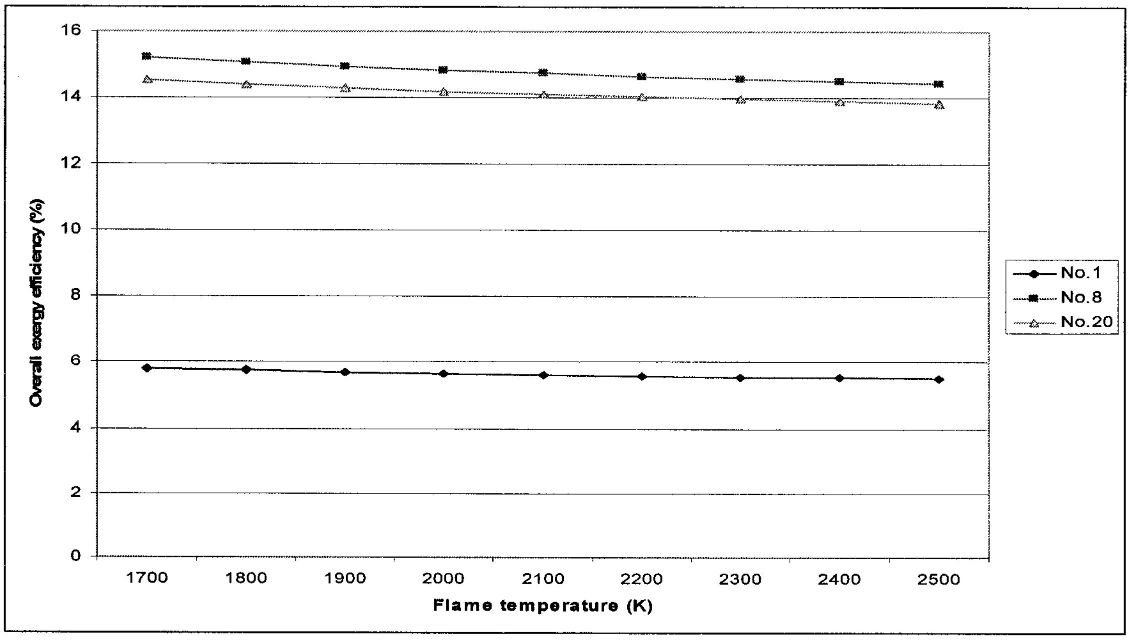

Figure 7.3 Overall exergy efficiency (η_2) versus flame temperature (TK_{flame})

7.4 Efficiency of gas-fired boiler

Table 7.4 and Figure 7.4 present the sensitivity analysis results with respect to the efficiency of the gas-fired boiler ($\eta_{gboiler}$). Varying $\eta_{gboiler}$ from 50% to 90% with an increment of 5%, causes η_2 to increase from 14.49% to 15.83% and from 21.76% to 23.71% for design alternative No.8 and No.20, respectively.

Table 7.4 Sensitivity analysis of exergy efficiency (η_2) due to the variation of the efficiency of the gas-fired boiler ($\eta_{gboiler}$)

$\eta_{gboiler}$ (%)	50	55	60	65	70	75	80	85	90
η_2 (%) (No. 8)	14.49	14.66	14.83	15.00	15.17	15.33	15.50	15.67	15.83
η_2 (%) (No. 20)	21.76	22.13	22.45	22.73	22.97	23.19	23.38	23.55	23.71

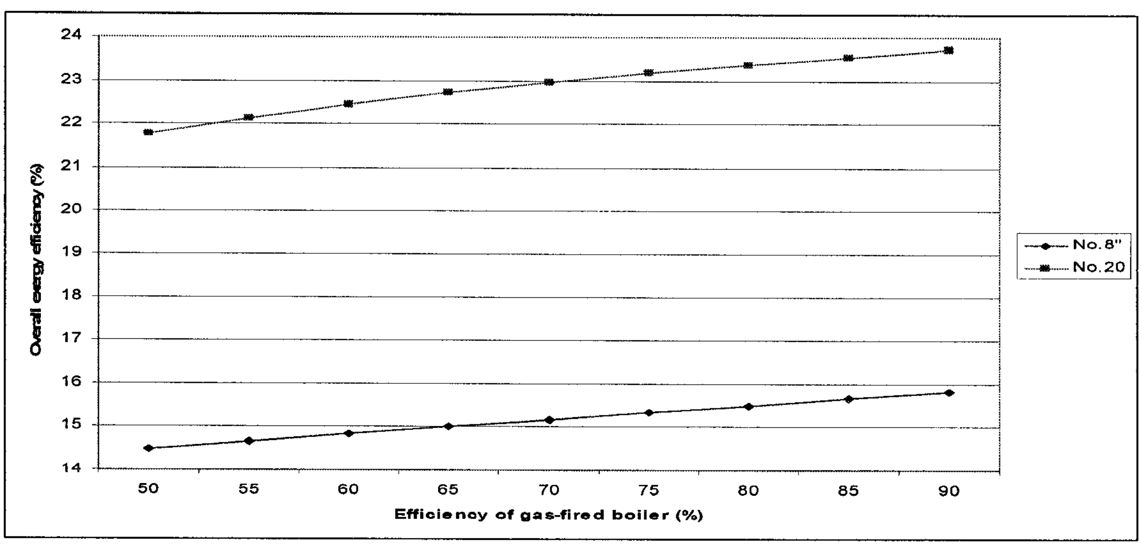

Figure 7.4 Overall exergy efficiency (η_2) versus efficiency of gas-fired boiler ($\eta_{gboiler}$)

7.5 Heat recovery efficiency of boiler economizer

Table 7.5 presents the sensitivity analysis results with respect to the heat recovery efficiency of boiler economizer for design alternative No.8. It is noted that when η_{econ} is modified from 20% to 70% with an increment of 5%, η_2 increases from 13.70% to 16.96%. This indicates that higher performance boiler economizer contributes much to the total exergy performance of the HVAC system.

Table 7.5 Sensitivity analysis of exergy efficiency (η_2) due to the variation of heat recovery efficiency of boiler economizer

η_{econ} (%)	20	25	30	35	40	45	50	55	60	65	70
η_2 (%)(No.8)	13.70	14.03	14.35	14.68	15.01	15.33	15.66	15.99	16.31	16.63	16.96

7.6 Ventilation air change rate

Ventilation air change rate (β) is determined by the designer at the beginning of the design to satisfy the indoor air quality. It is noted from Table 7.6 that the selection of β has a great influence on the simulation results. When β is increased from 0.2 ach to 0.5 ach with an increment of 0.1 ach, η_2 decreases from 15.83% to 14.85% and from 24.17% to 22.27% for design alternative No.8 and No.20, respectively. This indicates that the selection of ventilation air change rate is very important in the design stage to satisfy both indoor air quality and the HVAC system exergy performance requirement.

Table 7.6 Sensitivity analysis of exergy efficiency (η_2) due to the variation of the ventilation air change rate (β)

β (ach)	0.2	0.3	0.35	0.4	0.5
η_2 (%) (No.8)	15.83	15.50	15.33	15.17	14.85
η_2 (%) (No.20)	24.17	23.51	23.19	22.87	22.27

7.7 Air flow velocity in the earth tube heat exchanger

The cross section dimension (b) of the earth tube heat exchanger determines the air flow velocity (v), if the ventilation air change rate (β) is hold constant. Table 7.7 shows the sensitivity analysis results with respect to the air flow velocity in the earth tube heat exchanger for design alternative No.8 and No.20. It is noted that if b is modified from 0.1m to 0.5m, v changes from 8.44 m/s to 0.34 m/s, and as a result, η_2 varies in a range from 15.54% to 15.33% and from 23.42% to 23.19% for design alternative No.8 and No.20, respectively. Therefore the impact of b or v is relatively low. However, since the cross section dimension (b) of the earth tube heat exchanger and the air flow velocity (v) can be controlled by the designer, they should be chosen based on the sensitivity analysis. Figure 7.5 shows that v=1.35 m/s or b=0.25m is the worst condition for design alternative No.20 with the minimum exergy efficiency of 23.19%.

Table 7.7 Sensitivity analysis of exergy efficiency (η_2) due to the variation of the air flow velocity (v) in the earth tube heat exchanger

b (m)	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.45	0.50
v (m/s)	8.44	3.75	2.11	1.35	0.94	0.69	0.53	0.42	0.34
η_2 (%) (No.8)	15.54	15.40	15.34	15.33	15.35	15.38	15.42	15.45	15.49
η_2 (%)(No.20)	23.42	23.27	23.20	23.19	23.21	23.24	23.28	23.33	23.37

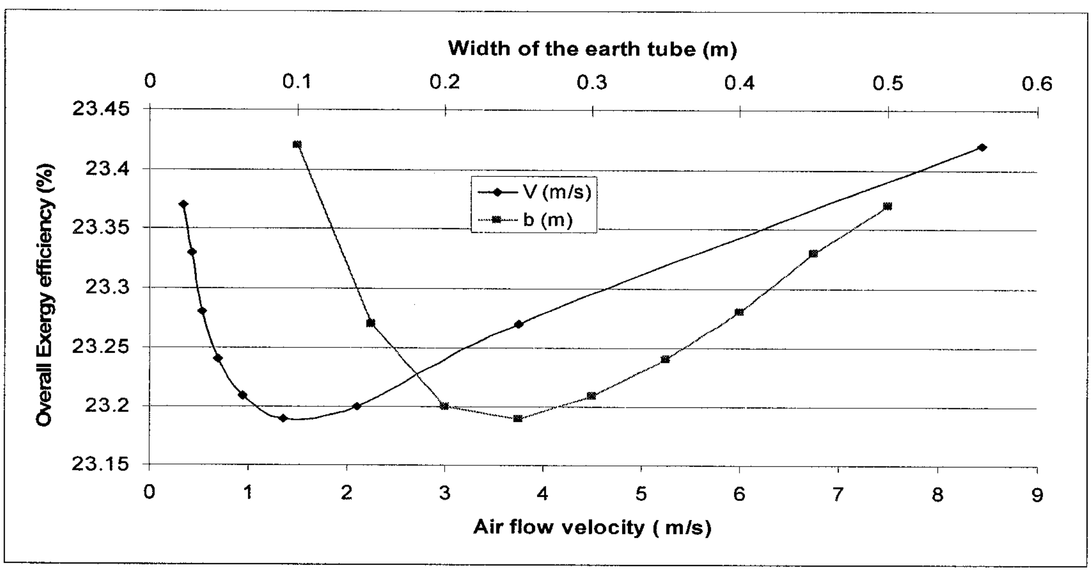

Figure 7.5 overall exergy efficiency (η_2) versus air flow velocity (v) in the earth tube heat exchanger (Design alternative No.20)

7.8 Length of the earth tube heat exchanger

Table 7.8 presents the sensitivity analysis results with respect to the length of the earth tube heat exchanger (L), which is also set by the designer. Extending L from 5 m to 25 m,

η_2 increases from 14.85% to 15.78% and from 22.57% to 23.61% for design alternative No.8 and No.20, respectively. It is also found that increasing the length of the earth tube heat exchanger beyond 25 m brings little increase in the exergy efficiency (Figure 7.6). It is important to note that the increase of fan power needs due to the increase of pressure loss in the earth tube heat exchanger is not considered.

Table 7.8 Sensitivity analysis of the exergy efficiency (η_2) due to the variation of the length of the earth tube heat exchanger (L)

L (m)	5	10	15	20	25	30
η_2 (%) (No.8)	14.85	15.33	15.58	15.71	15.78	15.82
η_2 (%) (No.20)	22.57	23.19	23.46	23.57	23.61	23.61

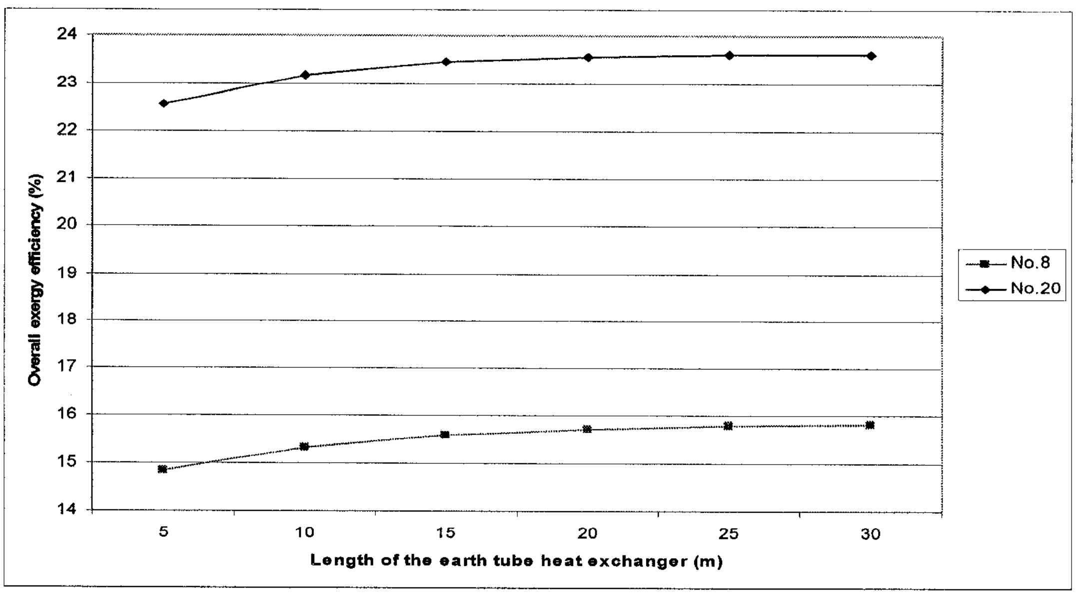

Figure 7.6 Overall exergy efficiency (η_2) versus length of the earth tube heat exchanger (L)

7.9 Heat exchange efficiency of air-to-air heat exchanger

Table 7.9 and Figure 7.7 present the sensitivity analysis results with respect to the heat exchange efficiency of the air-to-air heat exchanger (η_{HE}). A 10% increment in η_{HE} (from 20% to 80%) causes an increase in exergy efficiency by 0.21% and 0.29% for design alternative No.8 and No. 20, respectively.

Table 7.9 Sensitivity analysis of the exergy efficiency (η_2) due to the variation of the heat exchange efficiency of the air-to-air heat exchanger (η_{HE})

η_{HE} (%)	20	30	40	50	60	70	80
η_2 (%)(No.8)	14.51	14.71	14.91	15.12	15.33	15.56	15.79
η_2 (%) (No.20)	22.08	22.35	22.62	22.90	23.19	23.48	23.79

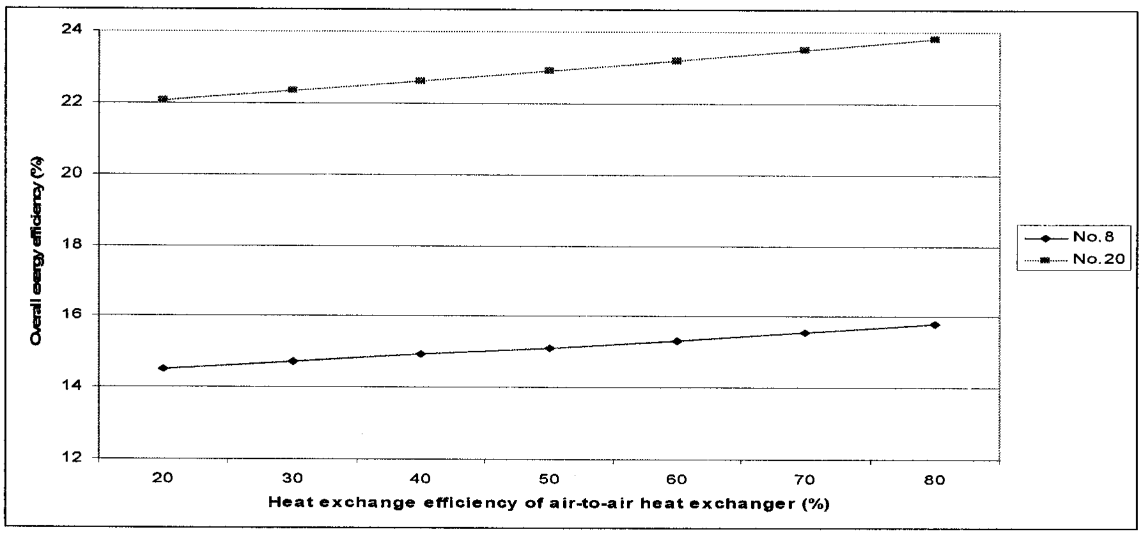

Figure 7.7 Overall exergy efficiency (η_2) versus heat exchange efficiency of air-to-air heat exchanger (η_{HE})

7.10 Ground temperature

Geothermal energy is utilized through ground source heat pump and/or earth tube heat exchanger, to provide for space heating, domestic hot water preheating and/or ventilation air preheating. The ground temperature (T_{ground}) is critical in the use of geothermal energy and it is usually determined by the buried depth of the underground heat exchanger. Table 7.10 shows the sensitivity analysis with respect to the ground temperature (T_{ground}). When T_{ground} is increased from 2°C to 12°C, with an increment of 2°C, the exergy efficiency increases from 14.61% to 15.83% and from 22.54% to 23.60% for design alternative No.8 and No.20, respectively.

Table 7.10 Sensitivity analysis of the exergy efficiency (η_2) due to the variation of the ground temperature (T_{ground})

T_{ground} (°C)	2	4	6	8	10	12
η_2 (%)(No.8)	14.61	14.85	15.09	15.33	15.58	15.83
η_2 (%)(No.20)	22.54	22.76	22.98	23.19	23.40	23.60

CHAPTER 8

CONCLUSIONS AND FUTURE WORK

8.1 Conclusions

The research presented here focused on the analysis and evaluation of the HVAC systems and their components by the exergy performance. A number of representative design alternatives of the HVAC system for a house located in Montreal have been analyzed based on the second law analysis. The results of the present path of analysis show the following:

- The energy performance of HVAC systems is usually evaluated based on the first law of thermodynamics. However, compared to energy analysis, the exergy analysis can better and accurately show the location of inefficiencies and point out the areas with great potential for improvement. In addition, integrated with energy analysis and entropy analysis, exergy analysis gives the whole picture and forms the basis for the performance evaluation of the HVAC system.
- Some HVAC systems adopted energy efficient technologies, and renewable energies, but failed to meet the expected improvement. The reason is that these technologies are not correctly integrated into the system. In order to achieve sustainable design in HVAC system, renewable energies and energy efficient technologies should be integrated in a manner where entropy generation and exergy destruction are minimized.

- Based on the second law analysis results and discussions in the above, in the Montreal area, air-to-air heat exchanger, earth tube heat exchanger, radiant heating floor, and ground source heat pump are exergy efficient and recommended. The integration of radiant heating floor and ground source heat pumps is an excellent way to make use of the low quality geothermal energy to match the low quality energy demand of space heating.
- The design alternative, which integrates hot water radiant floor, ground source heat pump, an air-to-air heat exchanger, an earth tube heat exchanger, and a gas-fired water heater, is the most exergy efficient among all these design alternatives. However, it still has some potential to improve its exergy performance. To improve the exergy performance in the power generation and transmission sector, more renewable energy sources should be used, such as, solar energy and wind power. Selection of high efficient ground source heat pump, gas-fired boiler, fans and pumps is a feasible way to increase the exergy performance in this area.
- The integration of a radiant heating floor and a GSHP can reduce the annual equivalent CO_2 emissions significantly. The use of energy efficient technologies, such as an air-to-air heat exchanger and an earth tube heat exchanger, and energy cascading use strategies, such as a boiler economizer, can reduce the annual equivalent CO_2 emissions.
- The sensitivity analysis indicated that the following parameters have great influence on the simulation results. When electricity is used as the principle energy source, the important parameters are: (1) the efficiency of hydro-electricity generation; (2) the efficiency of electricity transmission; (3) the ventilation air change rate; (4) the

efficiency of the air-to-air heat exchanger. When hot water from gas-fired boiler is used as the principle energy source, they are: (1) the efficiency of gas-fired boiler; (2) the ventilation air change rate; (3) the heat exchange efficiency of the air-to-air heat exchanger; (4) the heat recovery efficiency of the boiler economizer (if it is used). Thus, more attention should be given to ensure the accuracy of these sensitive parameters.

8.2 Future work

The present research work focused on performance analysis of the HVAC systems at heating conditions in Montreal. Therefore, future work should include:

- In addition to Montreal, the analysis should be extended to other regions of Canada, such as Toronto, Vancouver and Calgary;
- Cooling conditions should be integrated into the analysis to represent the overall annual performance of the HVAC systems;
- In order to make full use the advantage of exergy analysis, exergy analysis should be applied to some more complex systems, such as, in commercial and institutional buildings;
- Detailed information related to the life cycle of the HVAC systems will be needed in the analysis to reflect the whole picture of the HVAC systems;
- In the modeling of fans and pumps in this study, the pressure loss in the duct or piping system is not considered. Therefore, the future work should also include the more accurate models of fans and pumps where volumetric flow rate and pressure loss are taken into account.

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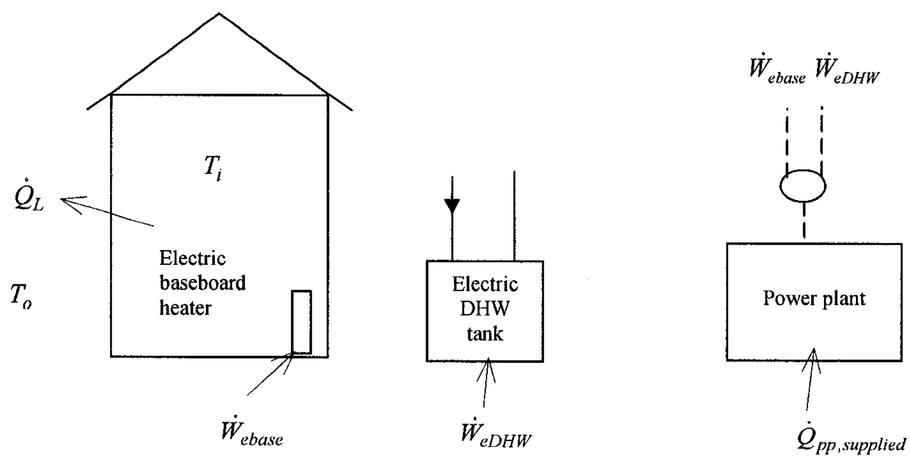
APPENDIX A

Second law analysis program for design alternative No.1



Configuration of design alternative No.1

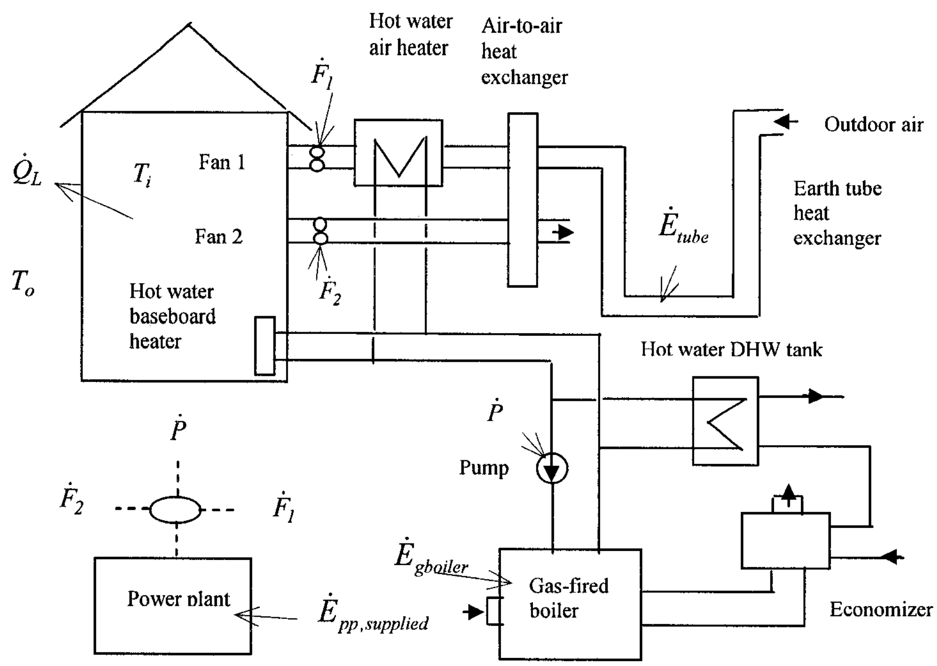
```
"! The second law analysis of HVAC-DHW system"

"Design alternative No.1

Heating: electric baseboard heaters.
Ventilation: none
DHW: electric water heater.

Design conditions;
1. Indoor air temperature T_i=21C, outdoor air temperature T_o=-23C.
2. Area of the house A=310m^2
3. Height of the room H=2.8m
4. Peak heating load Q_dot_L=11.1 kW."

"known information"

A=310"[m^2]";H=2.8"[m]";
rho_a=DENSITY(Air,T=T_av,P=101)
c_pa=cp(air,T=T_av)
c_pw=cp(water,T=15,P=101)
T_av=(T_i+T_o)/2
TK_i=273.15+T_i  "[K]"
TK_o=273.15+T_o


"! Electric baseboard heater"

W_dot_ebase=Q_dot_L  "[KW]"
```

```
S_dot_gen_ebase=Q_dot_L/TK_i  "[KW/k]"
X_dot_de_ebase=TK_o*S_dot_gen_ebase  "[KW]"

"! DHW tank"

w_dot_eDHW=m_dot_w_DHW*c_pw*(T_w_out_eDHW-T_w_in_eDHW)"[KW]"
S_dot_gen_eDHW=m_dot_w_DHW*(s_w_out_eDHW-s_w_in_eDHW) "[KW/k]"
s_w_in_eDHW=entropy(water,T=T_w_in_eDHW,P=101)  "[KJ/kgk]"
s_w_out_eDHW=entropy(water,T=T_w_out_eDHW,P=101)  "[KJ/kgk]"
X_dot_de_eDHW=TK_o*S_dot_gen_eDHW  "[KW]"

"! Power transmission"

S_dot_gen_trans=W_dot_pp*(1-eta_trans)/TK_o
X_dot_de_trans=TK_o*S_dot_gen_trans

"! Power plant"

S_dot_gen_pp=S_dot_gen_gas+S_dot_gen_oil+S_dot_gen_coal+S_dot_gen_nuclear+S_dot_ge
n_hydro
S_dot_gen_gas=Q_dot_pp_gas*(1-eta_pp_gas)/TK_o-Q_dot_pp_gas/TK_flame  "[KW/k]"
Q_dot_pp_gas=alpha_gas*W_dot_pp/eta_pp_gas
S_dot_gen_oil=Q_dot_pp_oil*(1-eta_pp_oil)/TK_o-Q_dot_pp_oil/TK_flame  "[KW/k]"
Q_dot_pp_oil=alpha_oil*W_dot_pp/eta_pp_oil
S_dot_gen_coal=Q_dot_pp_coal*(1-eta_pp_coal)/TK_o-Q_dot_pp_coal/TK_flame  "[KW/k]"
Q_dot_pp_coal=alpha_coal*W_dot_pp/eta_pp_coal
S_dot_gen_nuclear=Q_dot_pp_nuclear*(1-eta_pp_nuclear)/TK_o "[KW/k]"
Q_dot_pp_nuclear=alpha_nuclear*W_dot_pp/eta_pp_nuclear
S_dot_gen_hydro=Q_dot_pp_hydro*(1-eta_pp_hydro)/TK_o "[KW/K]"
Q_dot_pp_hydro=alpha_hydro*W_dot_pp/eta_pp_hydro
W_dot_pp=(W_dot_ebase+W_dot_eDHW)/eta_trans  "[KW]"
Q_dot_pp_supplied=Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal+Q_dot_pp_nuclear+Q_dot_p
p_hydro
X_dot_de_pp=TK_o*S_dot_gen_pp  "[KW]"

"! Whole system+power plant"

S_dot_gen_total=S_dot_gen_ebase+S_dot_gen_eDHW+S_dot_gen_pp
+S_dot_gen_trans"[KW/k]"
X_dot_de_total=TK_o*S_dot_gen_total  "[KW]"

"First law efficiency"

eta_1=Q_dot_useful/Q_dot_supplied
Q_dot_useful=w_dot_ebase+w_dot_eDHW  "[KW]"
Q_dot_supplied=Q_dot_pp_supplied "[KW]"

"Second law efficiency"

eta_2=1-X_dot_de_total/X_dot_supplied
X_dot_supplied=(Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal)*(1-TK_o/TK_flame)+
Q_dot_pp_nuclear+Q_dot_pp_hydro " [KW]"

"Energy from different sources"

Q_dot_gas=Q_dot_pp_gas
```

```
Q_dot_oil=Q_dot_pp_oil
Q_dot_coal=Q_dot_pp_coal
Q_dot_nuclear=Q_dot_pp_nuclear
Q_dot_hydro=Q_dot_pp_hydro
$SumRow on
```

APPENDIX B

Second law analysis program for design alternative No.8



Configuration of design alternative No.8

"! The second law analysis of HVAC system"

"Design alternative No.8

Heating: hot water baseboard heaters with gas-fired boiler with economizer.
Ventilation: hot water air heater, air-to-air heat exchanger and earth tube heat exchanger.
DHW: heat exchanger with gas-fired boiler with economizer.

Design conditions;
1Ventilation air change beta=0.35ach;
2. Area of the house A=250m^2
3. Height of the room H=2.8m
4. Total heat loss of the house at winter design conditions Q_dot_L=12kw.
5. The dimensions of the earth tube bxb=250x250mm, length L=10m"

```
Procedure FP(a,b:X)
P=2.9*a/b
if (P<0.1) then
X=0.1  else
X=P
endif
end
```

```
Procedure PP(a,:X)
P=0.0058*a
if (P<0.1) then
X=0.1  else
X=P
endif
end
Procedure TL(T_o,T_ground:L)
if(T_o<T_ground) then
L=10
else
L=0
endif
end
call TL(T_o,T_ground:L)
call FP(m_dot_a_fan1,rho_a:F_dot_1)
call FP(m_dot_a_fan2,rho_a:F_dot_2)
call PP(Q_dot_pump:P_dot)

"!Know information"

rho_a=density(air,T=T_av,P=101)
c_pa=cp(air,T=T_av)
T_av=(T_i+T_o)/2
C_pw=cp(water,T=15,P=101)
beta=0.35;A=310[m^2];H=2.8[m];b=0.25[m]
m_dot_a_fresh=A*H*beta*rho_a*convert(kg/h,kg/s)
m_dot_a_fresh=m_dot_a_tube;m_dot_a_fresh=m_dot_a_HE;m_dot_a_fresh=m_dot_a_wheater;
m_dot_a_fresh=m_dot_ea_HE;m_dot_a_fresh=m_dot_ea_out
m_dot_a_fresh=m_dot_a_fan1;m_dot_ea_out=m_dot_a_fan2;
T_a_in_tube=T_o;T_a_out_tube=T_a_in_HE;T_a_out_HE=T_a_in_wheater;T_a_out_wheater=T
_i;T_ea_in_HE=T_i;T_ea_out_HE=T_ea_out

T_w_out_gboiler=T_w_in_wbase;T_w_out_gboiler=T_w_in_wheater;T_w_out_gboiler=T_w_in_h
DHW
T_w_in_gboiler=T_w_out_wbase;T_w_in_gboiler=T_w_out_wheater;T_w_in_gboiler=T_w_out_h
DHW
T_w_out_econ=T_w_in_DHW
T_flue_in_gboiler=T_o;T_flue_out_gboiler=T_flue_in_econ


"! Earth tube heat exchanger"

alpha=6.15+4.18*v          "[kw/m^2-k]"                    "heat tranfer coefficient"
v=(A*H*beta/(b*b))*convert(m/h,m/s)
Q_dot_tube=alpha*4*b*L*(T_ground-(T_a_in_tube+T_a_out_tube)/2)/1000
m_dot_a_tube*c_pa*(T_a_out_tube-T_a_in_tube)=alpha*4*b*L*(T_ground-
(T_a_in_tube+T_a_out_tube)/2)/1000  "[kw]"  "energy balance in earth tube"

S_dot_gen_tube=m_dot_a_tube*(s_a_out_tube-s_a_in_tube)-Q_dot_tube/TK_a_tube  "[kw/k]"
TK_a_tube=273.15+(T_a_in_tube+T_a_out_tube)/2  "[k]"
s_a_in_tube=entropy(air,T=T_a_in_tube,P=101)
s_a_out_tube=entropy(air,T=T_a_out_tube,P=101)
X_dot_de_tube=TK_o*S_dot_gen_tube  "[kw]"
TK_o=273.15+T_o  "[k]"
```

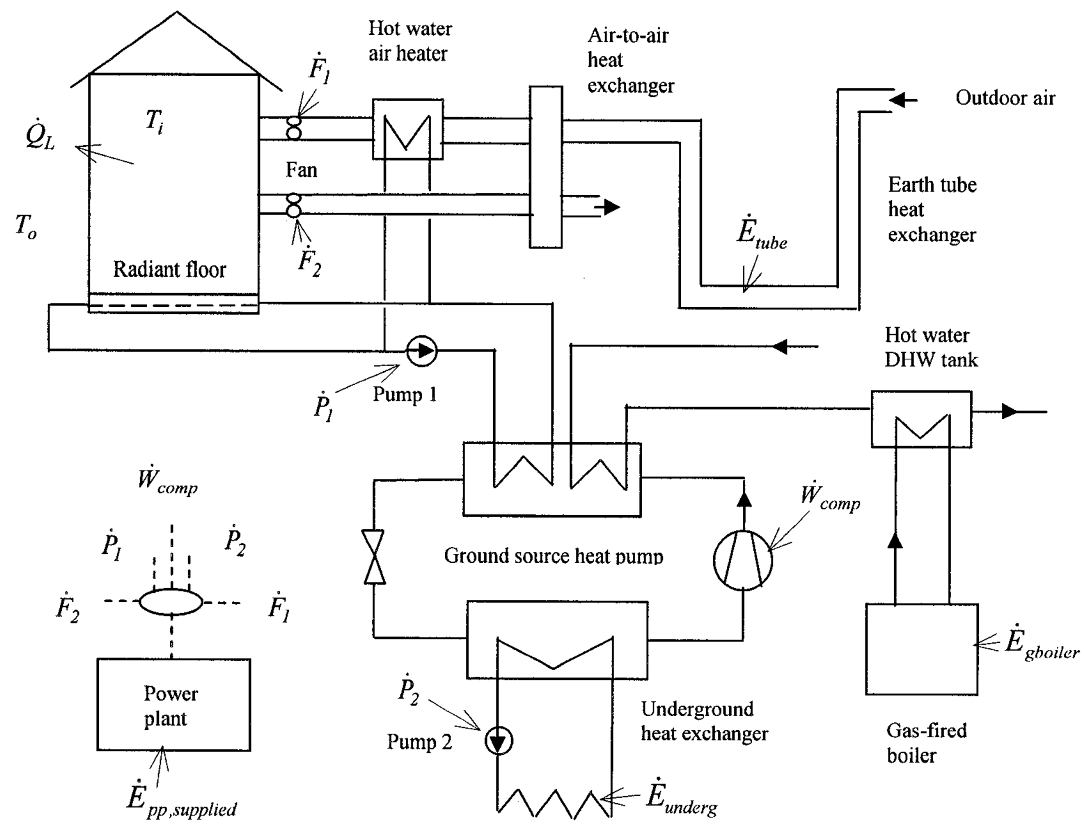
```
"! Air-to-air heat exchanger"

eta_HE=(T_a_out_HE-T_a_in_HE)/(T_ea_in_HE-T_a_in_HE)
eta_HE=(T_ea_in_HE-T_ea_out_HE)/(T_ea_in_HE-T_a_in_HE)
S_dot_gen_HE=m_dot_a_HE*(s_a_out_HE-s_a_in_HE)+m_dot_ea_HE*(s_ea_out_HE-
s_ea_in_HE)
s_a_in_HE=entropy(air,T=T_a_in_HE,P=101)
s_a_out_HE=entropy(air,T=T_a_out_HE,P=101)
s_ea_in_HE=entropy(air,T=T_ea_in_HE,P=101)
s_ea_out_HE=entropy(air,T=T_ea_out_HE,P=101)
X_dot_de_HE=TK_o*S_dot_gen_HE  "[kw]"

"! Exhaust air"

S_dot_gen_exhaust=m_dot_ea_out*(s_o-s_ea_out)+m_dot_ea_out*c_pa*(T_ea_out-T_o)/TK_o
s_o=entropy(air,T=T_o,P=101)
s_ea_out=entropy(air,T=T_ea_out,P=101)
X_dot_de_exhaust=TK_o*S_dot_gen_exhaust "[kw]"

"! Fans"

S_dot_gen_fan=(F_dot_1+F_dot_2)/TK_o
x_dot_de_fan=TK_o*S_dot_gen_fan

"! Pump"

Q_dot_pump=Q_dot_L+Q_dot_wheater+Q_dot_DHW
S_dot_gen_pump=P_dot/TK_o
X_dot_de_pump=TK_o*S_dot_gen_pump

"! Hot water air heater"

Q_dot_wheater=m_dot_a_wheater*c_pa*(T_a_out_wheater-T_a_in_wheater)   "[kw]"
Q_dot_wheater =m_dot_w_wheater*c_pw*(T_w_in_wheater-T_w_out_wheater)
S_dot_gen_wheater=m_dot_a_wheater*(s_a_out_wheater-
s_a_in_wheater)+m_dot_w_wheater*(s_w_out_wheater-s_w_in_wheater)  "[kw/k]"
s_a_in_wheater=entropy(air,T=T_a_in_wheater,P=101)
s_a_out_wheater=entropy(air,T=T_a_out_wheater,P=101)
s_w_in_wheater=entropy(water,T=T_w_in_wheater,P=101)
s_w_out_wheater=entropy(water,T=T_w_out_wheater,P=101)
X_dot_de_wheater=TK_o*S_dot_gen_wheater  "[kw]"

"! Hot water baseboard heater"

S_dot_gen_wbase=m_dot_w_wbase*(s_w_out_wbase-s_w_in_wbase)+Q_dot_L/TK_i  "[kw/k]"
s_w_in_wbase=entropy(water,T=T_w_in_wbase,P=101)
s_w_out_wbase=entropy(water,T=T_w_out_wbase,P=101)
TK_i=273.15+T_i  "[k]"
m_dot_w_wbase=Q_dot_L/(c_pw*(T_w_in_wbase-T_w_out_wbase))  "[kg/s]"
X_dot_de_wbase=TK_o*S_dot_gen_wbase  "[kw]"

"! Gas-fired boiler"

m_dot_w_gboiler=m_dot_w_wbase+m_dot_w_wheater+m_dot_w_hDHW
S_dot_gen_gboiler=m_dot_w_gboiler*(s_w_out_gboiler-s_w_in_gboiler)-
Q_dot_gboiler/TK_flame+m_dot_flue*(s_flue_out_gboiler-s_flue_in_gboiler)  "[kw/k]"
```

```
m_dot_flue=Q_dot_gboiler*(1-eta_gboiler)/(c_pa*(T_flue_out_gboiler-T_flue_in_gboiler))
s_w_in_gboiler=entropy(water,T=T_w_in_gboiler,P=101)
s_w_out_gboiler=entropy(water,T=T_w_out_gboiler,P=101)
s_flue_in_gboiler=entropy(air,T=T_flue_in_gboiler,P=101)
s_flue_out_gboiler=entropy(air,T=T_flue_out_gboiler,P=101)
Q_dot_gboiler=(Q_dot_L+Q_dot_wheater+Q_dot_DHW)/eta_gboiler   "[kw]"

X_dot_de_gboiler=TK_o*S_dot_gen_gboiler   "[kw]"

"! Boiler economizer"

eta_econ=(m_dot_flue*c_pa*(T_flue_in_econ-
T_flue_out_econ))/(m_dot_flue*c_pa*(T_flue_in_econ-T_w_in_econ))
eta_econ=(m_dot_w_DHW*c_pw*(T_w_out_econ-
T_w_in_econ))/(m_dot_flue*c_pa*(T_flue_in_econ-T_w_in_econ))
S_dot_gen_econ=m_dot_w_DHW*(s_w_out_econ-s_w_in_econ)+Q_dot_gboiler*(1-
eta_gboiler)*(1-eta_econ)/TK_o+m_dot_flue*(s_o-s_flue_in_econ)

s_flue_in_econ=entropy(air,T=T_flue_in_econ,P=101)
s_w_in_econ=entropy(water,T=T_w_in_econ,P=101)
s_w_out_econ=entropy(water,T=T_w_out_econ,P=101)
X_dot_de_econ=TK_o*S_dot_gen_econ  "[kw]"

"! Hot water DHW tank"

Q_dot_DHW=m_dot_w_hDHW*c_pw*(T_w_in_hDHW-T_w_out_hDHW)
Q_dot_DHW=m_dot_w_DHW*c_pw*(T_w_out_DHW-T_w_in_DHW)
S_dot_gen_DHW=m_dot_w_hDHW*(s_w_out_hDHW-
s_w_in_hDHW)+m_dot_w_DHW*(s_w_out_DHW-s_w_in_DHW)
s_w_in_DHW=entropy(water,T=T_w_in_DHW,P=101)
s_w_out_DHW=entropy(water,T=T_w_out_DHW,P=101)
s_w_in_hDHW=entropy(water,T=T_w_in_hDHW,P=101)
s_w_out_hDHW=entropy(water,T=T_w_out_hDHW,P=101)
X_dot_de_DHW=TK_o*S_dot_gen_DHW  "[kw]"

"! Power transmission"

S_dot_gen_trans=W_dot_pp*(1-eta_trans)/TK_o
X_dot_de_trans=TK_o*S_dot_gen_trans

"! Power plant"

S_dot_gen_pp=S_dot_gen_gas+S_dot_gen_oil+S_dot_gen_coal+S_dot_gen_nuclear+S_dot_ge
n_hydro
S_dot_gen_gas=Q_dot_pp_gas*(1-eta_pp_gas)/TK_o-Q_dot_pp_gas/TK_flame   "[KW/k]"
Q_dot_pp_gas=alpha_gas*W_dot_pp/eta_pp_gas
S_dot_gen_oil=Q_dot_pp_oil*(1-eta_pp_oil)/TK_o-Q_dot_pp_oil/TK_flame   "[KW/k]"
Q_dot_pp_oil=alpha_oil*W_dot_pp/eta_pp_oil
S_dot_gen_coal=Q_dot_pp_coal*(1-eta_pp_coal)/TK_o-Q_dot_pp_coal/TK_flame   "[KW/k]"
Q_dot_pp_coal=alpha_coal*W_dot_pp/eta_pp_coal
S_dot_gen_nuclear=Q_dot_pp_nuclear*(1-eta_pp_nuclear)/TK_o "[KW/k]"
Q_dot_pp_nuclear=alpha_nuclear*W_dot_pp/eta_pp_nuclear
S_dot_gen_hydro=Q_dot_pp_hydro*(1-eta_pp_hydro)/TK_o "[KW/K]"
Q_dot_pp_hydro=alpha_hydro*W_dot_pp/eta_pp_hydro
W_dot_pp=(P_dot+F_dot_1+F_dot_2)/eta_trans  "[KW]"
```

```
Q_dot_pp_supplied=Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal+Q_dot_pp_nuclear+Q_dot_p
p_hydro
X_dot_de_pp=TK_o*S_dot_gen_pp  "[KW]"

"! Ventilation system"

S_dot_gen_vent=S_dot_gen_tube+S_dot_gen_HE+S_dot_gen_wheater + S_dot_gen_exhaust
+S_dot_gen_fan"[kw/k]"
X_dot_de_vent=TK_o*S_dot_gen_vent  "[kw]"

"! Heating system"

S_dot_gen_heating=S_dot_gen_gboiler+S_dot_gen_wbase+S_dot_gen_pump  "[kw/k]"
X_dot_de_heating=TK_o*S_dot_gen_heating  "[kw]"

"! Whole system"

S_dot_gen_total=S_dot_gen_vent+S_dot_gen_heating+S_dot_gen_DHW+S_dot_gen_econ+S_
dot_gen_trans+S_dot_gen_pp  "[kw/k]"
X_dot_de_total=TK_o*S_dot_gen_total  "[kw]"

"First law efficiency"

eta_1=Q_dot_useful/Q_dot_supplied
Q_dot_useful=Q_dot_L+m_dot_w_DHW*c_pw*(T_w_out_DHW-
T_w_in_econ)+m_dot_a_fresh*c_pa*(T_i-T_o)  "[kw]"
Q_dot_supplied=(Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal)+Q_dot_pp_nuclear+Q_dot_pp_
hydro+Q_dot_gboiler+Q_dot_tube "[kw]"

"Second law efficiency"

eta_2=1-X_dot_de_total/X_dot_supplied
X_dot_supplied=(Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal)*(1-
TK_o/TK_flame)+Q_dot_gboiler*(1-TK_o/TK_flame)+Q_dot_pp_nuclear+Q_dot_pp_hydro
+abs(Q_dot_tube*(1-TK_o/TK_ground)) "[kw]"
TK_ground=273.15+T_ground "[K]"

"Primary energy  from different sources"

Q_dot_gas=Q_dot_pp_gas
Q_dot_oil=Q_dot_pp_oil
Q_dot_coal=Q_dot_pp_coal
Q_dot_nuclear=Q_dot_pp_nuclear
Q_dot_hydro=Q_dot_pp_hydro

"On-site primary energy"

Q_dot_onsite_gas=Q_dot_gboiler
$SumRow on
```

APPENDIX C

Second law analysis program for design alternative No.20



Configuration of design alternative No.20

"! The second law analysis of HVAC system"

"Design alternative No.20

Heating: radiant heating floor with GSHP.
Ventilation: hot water air heater, air-to-air heat exchanger and earth tube heat changer.
DHW: GSHP and gas-fired water heater.

Design conditions;
1.Earth temperature T_ground=8°C;
2.Indoor air temperature T_i=21C, outdoor air temperature T_o=-23C, ventilation air exchange rate 0.35 ach
3.Area of the house A=310m^2;
4.Height of the room H=2.8m.
5.Peak heating load Q_dot_L=11.1kW"
Procedure FP(a,b:X)

```
P=2.9*a/b
if (P<0.1) then
X=0.1  else
X=P
endif
end
Procedure PP1(a:X)
P=0.0058*a
if (P<0.1) then
X=0.1  else
X=P
endif
end
Procedure PP2(a:X)
P=0.0058*4*a
if (P<0.1)  then
X=0.1  else
X=P
endif
end
Function FRAC(a,b)
if (a<b) then
X=1
else
X=b/a
endif
FRAC=X
end
Function Length(a,b)
if (a<b) then
X=10
else
X=0
endif
length=X
end
L=Length(T_o,T_ground) "[m]"
call FP(m_dot_a_fan1,rho_a:F_dot_1)
call FP(m_dot_a_fan2,rho_a:F_dot_2)
call PP1(Q_dot_pump1:P_dot_1)
call PP2(Q_dot_pump2:P_dot_2)
call PP1(Q_dot_pump3:P_dot_3)
Q_dot_floor=Q_dot_L
FRAC=FRAC(PMIN,PLR)
PMIN=0.1

"! Known information"

rho_a=density(air,T=T_av,P=101)
rho_w=density(water,T=15,P=101)
c_pa=cp(air,T=T_av)
c_pw=cp(water,T=15,P=101)
T_av=(T_i+T_o)/2
 beta=0.35;A=310[m^2];H=2.8[m];b=0.25[m]
m_dot_a_fresh=A*H*beta*rho_a*convert(kg/h,kg/s)
m_dot_a_fresh=m_dot_a_tube;m_dot_a_fresh=m_dot_a_HE;m_dot_a_HE=m_dot_a_wheater
```

```
m_dot_a_fresh=m_dot_a_fan1;m_dot_a_fan2=m_dot_ea_out
m_dot_ea_HE=m_dot_a_fresh; m_dot_ea_HE=m_dot_ea_out
m_dot_w_floor=m_dot_w_cond1;m_dot_w_wheater=m_dot_w_cond2;m_dot_w_DHW=m_dot_w
_cond3;
m_dot_w_gboiler=m_dot_w_hDHW
T_a_in_tube=T_o;T_a_out_tube=T_a_in_HE;T_a_out_HE=T_a_in_wheater;T_a_out_wheater=T
_i
T_ea_in_HE=T_i;T_ea_out_HE=T_ea_out
T_w_in_cond1=T_w_in_cond2;T_w_out_cond1=T_w_in_floor;T_w_out_cond2=T_w_in_wheater;
T_w_out_floor=T_w_in_cond1;T_w_out_wheater=T_w_in_cond2;
T_w_in_DHW=T_w_out_cond3
T_w_in_gboiler=T_w_out_hDHW;T_w_out_gboiler=T_w_in_hDHW
TK_o=T_o+273.15   "[K]"
TK_i=273.15+T_i "[K]"


"! Earth tube heat exchanger"

alpha=6.15+4.18*v          "[kw/m^2-k]"                    "heat tranfer coefficient"
v=(A*H*beta/(b*b))*convert(m/h,m/s)
Q_dot_tube=alpha*4*b*L*(T_ground-(T_a_in_tube+T_a_out_tube)/2)/1000   "[kw]"
alpha*4*b*L*(T_ground-
(T_a_in_tube+T_a_out_tube)/2)/1000=m_dot_a_tube*c_pa*(T_a_out_tube-T_a_in_tube)
S_dot_gen_tube=m_dot_a_tube*(s_a_out_tube-s_a_in_tube)-Q_dot_tube/TK_tube   "[kw/k]"
TK_tube=273.15+(T_a_in_tube+T_a_out_tube)/2
s_a_in_tube=entropy(air,T=T_a_in_tube,P=101)
s_a_out_tube=entropy(air,T=T_a_out_tube,P=101)
X_dot_de_tube=TK_o*S_dot_gen_tube   "[kw]"

"! Air-to-air heat exchanger"

eta_HE=(T_a_out_HE-T_a_in_HE)/(T_ea_in_HE-T_a_in_HE)
eta_HE=(T_ea_in_HE-T_ea_out_HE)/(T_ea_in_HE-T_a_in_HE)
S_dot_gen_HE=m_dot_a_HE*(s_a_out_HE-s_a_in_HE)+m_dot_ea_HE*(s_ea_out_HE-
s_ea_in_HE)  "[kw/k]"
s_a_in_HE=entropy(air,T=T_a_in_HE,P=101)
s_a_out_HE=entropy(air,T=T_a_out_HE,P=101)
s_ea_in_HE=entropy(air,T=T_ea_in_HE,P=101)
s_ea_out_HE=entropy(air,T=T_ea_out_HE,P=101)
X_dot_de_HE=TK_o*S_dot_gen_HE"[kw]"

"! Exhaust air"

S_dot_gen_exhaust=m_dot_ea_out*(s_o-s_ea_out)+m_dot_ea_out*c_pa*(T_ea_out-T_o)/TK_o
"[kw/k]"
s_o=entropy(air,T=T_o,P=101)
s_ea_out=entropy(air,T=T_ea_out,P=101)
X_dot_de_exhaust=TK_o*S_dot_gen_exhaust"[kw]"

"! Hot water air heater"

Q_dot_wheater=m_dot_a_wheater*c_pa*(T_a_out_wheater-T_a_in_wheater)  "[kw]"
Q_dot_wheater=m_dot_w_wheater*c_pw*(T_w_in_wheater-T_w_out_wheater)
S_dot_gen_wheater=m_dot_a_wheater*(s_a_out_wheater-
s_a_in_wheater)+m_dot_w_wheater*(s_w_out_wheater-s_w_in_wheater)  "[kw/k]"
s_a_in_wheater=entropy(air,T=T_a_in_wheater,P=101)
```

```
s_a_out_wheater=entropy(air,T=T_a_out_wheater,P=101)
s_w_in_wheater=entropy(water,T=T_w_in_wheater,P=101)
s_w_out_wheater=entropy(water,T=T_w_out_wheater,P=101)
X_dot_de_wheater=TK_o*S_dot_gen_wheater"[kw]"

"! Heating floor"

Q_dot_floor=m_dot_w_floor*c_pw*(T_w_in_floor-T_w_out_floor)
S_dot_gen_floor=Q_dot_floor/TK_i+m_dot_w_floor*(s_w_out_floor-s_w_in_floor)   "[kw/k]"
s_w_in_floor=entropy(water,T=T_w_in_floor,P=101)
s_w_out_floor=entropy(water,T=T_w_out_floor,P=101)
X_dot_de_floor=TK_o*S_dot_gen_floor"[kw]"

"! Ground source heat pump"

"Condenser heating load"

Q_dot_load=m_dot_w_cond1*c_pw*(T_w_out_cond1-
T_w_in_cond1)+m_dot_w_cond2*c_pw*(T_w_out_cond2-
T_w_in_cond2)+m_dot_w_cond3*c_pw*(T_w_out_cond3-T_w_in_cond3)"[kw]"
T_r4=T_r5;
T_r4=T_w_out_cond1+dT_cond;T_w_out_cond2=T_w_out_cond1;T_w_out_cond3=T_w_out_co
nd1
T_r1=T_r6; T_r1=T_w_g2-dT_evap
dT_cond=5 "[C]"; dT_evap=5 "[C]"
T_w_g1=T_w_g2-dT_w_g; dT_w_g=4 "[C]"

"State 1"

s_r1=entropy(R$,T=T_r1, x=1)
h_r1=enthalpy(R$,T=T_r1, x=1)

"State 2"

P_r2=P_r4
eta_is=(h_r3-h_r1)/(w_dot_comp/m_dot_r)
eta_is=(h_r3-h_r1)/(h_r2-h_r1)
s_r2=entropy(R$,P=P_r2,h=h_r2)

"State 3"

s_r3=s_r1
P_r3=P_r4
h_r3=enthalpy(R$,s=s_r3,P=P_r3)

"State 4"

h_r4=enthalpy(R$,T=T_r4,x=1)
s_r4=entropy(R$,T=T_r4,x=1)
P_r4=pressure(R$,T=T_r4,x=1)

"State 5"

h_r5=enthalpy(R$,T=T_r5,x=0)
s_r5=entropy(R$,T=T_r5,x=0)
```

```
"State 6"

h_r6=h_r5
s_r6=entropy(R$,h=h_r6,T=T_r6)

"Refrigerant flow rate"

m_dot_r=Q_dot_load/(h_r2-h_r5)  "[kg/s]"

"Condenser"

S_dot_gen_cond=m_dot_r*(s_r5-s_r2)+m_dot_w_cond1*(s_w_out_cond1-
s_w_in_cond1)+m_dot_w_cond2*(s_w_out_cond2-
s_w_in_cond2)+m_dot_w_cond3*(s_w_out_cond3-s_w_in_cond3)  "[kw/k]"
s_w_in_cond1=entropy(water,T=T_w_in_cond1,P=101)
s_w_out_cond1=entropy(water,T=T_w_out_cond1,P=101)
s_w_in_cond2=entropy(water,T=T_w_in_cond2,P=101)
s_w_out_cond2=entropy(water,T=T_w_out_cond2,P=101)
s_w_in_cond3=entropy(water,T=T_w_in_cond3,P=101)
s_w_out_cond3=entropy(water,T=T_w_out_cond3,P=101)

"Expansion valve"

S_dot_gen_valve=m_dot_r*(s_r6-s_r5)  "[kw/k]"

"Evaporator"

S_dot_gen_evap=m_dot_r*(s_r1-s_r6)+m_dot_w_g*(s_w_g1-s_w_g2)  "[kw/k]"
m_dot_w_g=m_dot_r*(h_r1-h_r6)/(c_pw*(T_w_g2-T_w_g1))  "[kg/s]"
s_w_g1=entropy(water,T=T_w_g1,P=101)
s_w_g2=entropy(water,T=T_w_g2,P=101)

"Compressor"

S_dot_gen_comp=m_dot_r*(s_r2-s_r1)  "[kw/k]"

"Overall heat pump"

PLR=Q_dot_load/Q_dot_cap
Q_dot_cap=3.103+0.428*T_w_g1+3.651*m_dot_w_dg
cop_cap=2.94+0.031*T_w_g1+0.191*m_dot_w_dg
EIR=(0.11+0.89*PLR)/cop_cap
W_dot_comp=Q_dot_cap*EIR*FRAC
m_dot_w_dg=13*0.0631"[kg/s]"
S_dot_gen_gshp=S_dot_gen_cond+S_dot_gen_valve+S_dot_gen_evap+S_dot_gen_comp
"[kw/k]"
X_dot_de_gshp=TK_o*S_dot_gen_gshp  "[kw]"

"! Underground heat exchanger"

S_dot_gen_underg=m_dot_w_g*(s_w_g2-s_w_g1)-Q_dot_underg/TK_ag   "[kw/k]"
TK_ag=273.15+(T_w_g1+T_w_g2)/2  "[K]"
Q_dot_underg=m_dot_w_g*c_pw*(T_w_g2-T_w_g1)  "[kw]"
X_dot_de_underg=TK_o*S_dot_gen_underg  "[kw]"

"! Hot water DHW tank"
```

```
S_dot_gen_DHW=m_dot_w_hDHW*(s_w_out_hDHW-
s_w_in_hDHW)+m_dot_w_DHW*(s_w_out_DHW-s_w_in_DHW)  "[kw/k]"
Q_dot_DHW=m_dot_w_hDHW*c_pw*(T_w_in_hDHW-T_w_out_hDHW)
m_dot_w_DHW*c_pw*(T_w_out_DHW-T_w_in_DHW)=m_dot_w_hDHW*c_pw*(T_w_in_hDHW-
T_w_out_hDHW)
s_w_in_DHW=entropy(water,T=T_w_in_DHW,P=101)
s_w_out_DHW=entropy(water,T=T_w_out_DHW,P=101)
s_w_in_hDHW=entropy(water,T=T_w_in_hDHW,P=101)
s_w_out_hDHW=entropy(water,T=T_w_out_hDHW,P=101)
X_dot_de_DHW=TK_o*S_dot_gen_DHW  "[kw]"

"! Gas boiler"

Q_dot_gboiler=(m_dot_w_gboiler*c_pw*(T_w_out_gboiler-T_w_in_gboiler))/eta_gboiler
S_dot_gen_gboiler=m_dot_w_gboiler*(s_w_out_gboiler-s_w_in_gboiler)-
Q_dot_gboiler/TK_flame+Q_dot_gboiler*(1-eta_gboiler)/TK_o
s_w_in_gboiler=entropy(water,T=T_w_in_gboiler,P=101)
s_w_out_gboiler=entropy(water,T=T_w_out_gboiler,P=101)
X_dot_de_gboiler=TK_o*S_dot_gen_gboiler

"! Fans"

S_dot_gen_fan=(F_dot_1+F_dot_2)/TK_o
X_dot_de_fan=TK_o*S_dot_gen_fan

"! Pump"

Q_dot_pump1=Q_dot_L+Q_dot_wheater
Q_dot_pump2=Q_dot_underg
Q_dot_pump3=Q_dot_DHW
S_dot_gen_pump=(P_dot_1+P_dot_2+P_dot_3)/TK_o
x_dot_de_pump=TK_o*S_dot_gen_pump

"! Ventilation system"

S_dot_gen_vent=S_dot_gen_tube+S_dot_gen_HE+S_dot_gen_exhaust+S_dot_gen_wheater
+S_dot_gen_fan "[kw/k]"

"! Heating system"

S_dot_gen_heating=S_dot_gen_floor+S_dot_gen_gshp+S_dot_gen_underg
+S_dot_gen_pump+S_dot_gen_gboiler "[kw/k]"

"! Power transmission"

S_dot_gen_trans=W_dot_pp*(1-eta_trans)/TK_o
X_dot_de_trans=TK_o*S_dot_gen_trans

"! Power plant"

S_dot_gen_pp=S_dot_gen_gas+S_dot_gen_oil+S_dot_gen_coal+S_dot_gen_nuclear+S_dot_ge
n_hydro
S_dot_gen_gas=Q_dot_pp_gas*(1-eta_pp_gas)/TK_o-Q_dot_pp_gas/TK_flame  "[KW/k]"
Q_dot_pp_gas=alpha_gas*W_dot_pp/eta_pp_gas
S_dot_gen_oil=Q_dot_pp_oil*(1-eta_pp_oil)/TK_o-Q_dot_pp_oil/TK_flame  "[KW/k]"
```

```
Q_dot_pp_oil=alpha_oil*W_dot_pp/eta_pp_oil
S_dot_gen_coal=Q_dot_pp_coal*(1-eta_pp_coal)/TK_o-Q_dot_pp_coal/TK_flame    "[KW/k]"
Q_dot_pp_coal=alpha_coal*W_dot_pp/eta_pp_coal
S_dot_gen_nuclear=Q_dot_pp_nuclear*(1-eta_pp_nuclear)/TK_o "[KW/k]"
Q_dot_pp_nuclear=alpha_nuclear*W_dot_pp/eta_pp_nuclear
S_dot_gen_hydro=Q_dot_pp_hydro*(1-eta_pp_hydro)/TK_o "[KW/K]"
Q_dot_pp_hydro=alpha_hydro*W_dot_pp/eta_pp_hydro
W_dot_pp=(W_dot_comp+P_dot_3+P_dot_1+P_dot_2+F_dot_1+F_dot_2)/eta_trans  "[KW]"
Q_dot_pp_supplied=Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal+Q_dot_pp_nuclear+Q_dot_p
p_hydro
X_dot_de_pp=TK_o*S_dot_gen_pp   "[KW]"

"! Whole system"

S_dot_gen_total=S_dot_gen_vent+S_dot_gen_heating+S_dot_gen_pp+S_dot_gen_trans+S_dot
_gen_DHW   "[Kw/k]"
X_dot_de_total=TK_o*S_dot_gen_total   "[Kw]"

"First law efficiency"

eta_1=Q_dot_useful/Q_dot_supplied
Q_dot_useful=Q_dot_L+m_dot_w_DHW*c_pw*(T_w_out_DHW-
T_w_in_cond3)+m_dot_a_fresh*c_pa*(T_i-T_o)    "[Kw]"
Q_dot_supplied=Q_dot_pp_supplied+Q_dot_underg+Q_dot_tube+Q_dot_gboiler  "[Kw]"

"Second law efficiency"

eta_2=1-X_dot_de_total/X_dot_supplied

X_dot_supplied=(Q_dot_pp_gas+Q_dot_pp_oil+Q_dot_pp_coal)*(1-
TK_o/TK_flame)+abs(Q_dot_underg*(1-TK_o/TK_ag))+abs(Q_dot_tube*(1-
TK_o/TK_ground))+Q_dot_pp_hydro+Q_dot_pp_nuclear+Q_dot_gboiler*(1-TK_o/TK_flame)
TK_ground=273.15+T_ground

"Primary energy  from different sources"

Q_dot_gas=Q_dot_pp_gas
Q_dot_oil=Q_dot_pp_oil
Q_dot_coal=Q_dot_pp_coal
Q_dot_nuclear=Q_dot_pp_nuclear
Q_dot_hydro=Q_dot_pp_hydro

"On-site primary energy"

Q_dot_onsite_gas=Q_dot_gboiler
$SumRow on
```