ANALYSIS OF ENERGY, EXERGY AND GHG EMISSIONS APPLIED TO VAV SYSTEMS IN AN OFFICE BUILDING

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ABSTRACT

This paper presents a combined analysis of energy, exergy and greenhouse gas (GHG) emissions applied to the evaluation of the operation of two variable air volume (VAV) systems in an office building in Montreal. Energy and exergy related indices such as energy consumption, energy efficiency, exergy destruction, entropy generation and exergy efficiency are estimated at both system and power plant levels.

The energy and exergy analyses reveal that the electric boilers contribute to more than 60% of the total energy consumption and exergy destruction at the system level while the entire VAV systems consume more than 66% of the total exergy input at the power plant level. More appropriate selections of heating sources can significantly improve the energy and exergy performance of heating, ventilating and air-conditioning (HVAC) systems applied to office buildings. Exergy flow charts identify exergy interactions between components. Finally, some improvements to the operation of HVAC systems used in office buildings are recommended.

INTRODUCTION

Since the energy crisis in the early 1970s, scientists have paid more attention to the efficiency of energy use by exploiting renewable energy resources and reducing the impacts of energy use on the environment. The use of fossil fuels and the release of corresponding GHG emissions appear to have a significant impact on the global climate change. One key-point of the sustainable development concept applied to the building industry is the development of safe, healthy, energy-efficient and comfortable buildings and the reduction or mitigation of their impacts on natural resources and the environment. In 2003, commercial buildings in Canada accounted for 14% of the annual national end-use energy consumption and 13% of the annual GHG emissions (NRCan 2005). Meanwhile, in the commercial building sector, the energy consumption by space heating and cooling accounted for more than 60% of the total energy consumption and caused more than 58% of the total GHG emissions (NRCan 2005). Therefore, the effectiveness of energy use for space heating and cooling is essential for the successful implementation of the sustainable development concept in the building industry.

The application of the second law of thermodynamics analysis, or exergy analysis, in thermal systems is not a new approach. However, only a few publications discussed the applications of exergy analysis to HVAC systems for commercial buildings. Currently, energy analysis programs such as DOE-2, TRNSYS and EnergyPlus are only based on the first law analysis. Therefore, this paper addresses building simulation using another dimension, which normally neglected.

Wepfer et al. (1979) developed formulas to calculate the exergy of different fluids commonly encounted in HVAC applications and examined the exergy efficiency of some air-conditioning processes by utilizing the outdoor environment state as the dead or reference state. Since then, research about the exergy analysis of HVAC systems has been widely developed, but most of them were focused only on components or subsystems such as heat pumps or air distribution systems. For example, Tsaros et al. (1987) analyzed a residential heat pump and presented exergy flow diagrams to identify the exergy consumption and transportation between components of the equipment. Krakow (1991) used effective reservoir temperatures, instead of environment temperatures, as dead-state temperatures for the exergy evaluation of refrigeration machines or heat engines. He concluded that the use of the effective reservoir temperatures, rather than the environment temperatures, leads to less than 1% deviation in the calculation of exergy destruction and system exergy efficiency, while the use of a fixed temperature (e.g., 21.1ºC) instead of the effective temperatures leads to 5-10% deviation in the calculations of exergy destruction, and 15-35% deviation in the estimates of system exergy efficiency. Franconi et al. (1999) compared the exergy performance of two types of air distribution systems, Constant Air Volume (CAV) and Variable Air Volume (VAV) systems. They outlined that negative exergy loads might occur in some building zones that require cooling when their indoor air temperatures are higher than the instant outdoor temperature, or in some zones that require

heating when their indoor air temperatures are lower than the instant outdoor temperature. Bridges et al. (2001) analyzed the performance of domestic refrigerators and air conditioners by using the exergy analysis and located the most inefficient sub-components.

Only few papers presented the exergy analysis of HVAC systems at the system level. For instance, Ren et al. (2002) utilized exergy analysis to evaluate the performance of evaporative cooling. They applied a novel selection of the dead state to the exergy analysis of HVAC systems; they used outdoor temperatures with 100% relative humidity as the dead-state. The use of this dead-state was regarded as avoiding the underestimation of the exergy efficiency of HVAC systems. In addition, the new selection of the dead-state also satisfies the principle that no work can be extracted from the dead-state. By using experimental data, Asada and Takeda (2002) performed exergy analysis for a ceiling radiant cooling system using well water as the cooling source and concluded that the well water pump consumed a major portion of exergy in this cooling system.

Badescu (2002) studied a heat pump heating system with PV panels for a residential building, using both energy and exergy analyses. He concluded that solar energy can properly drive the compressor during the heating mode if an appropriate electrical storage system is provided. The exergy analysis revealed that the compression and condensation are the most inefficient processes during the operation of the heat pump.

Based on the energy and exergy analyses of a ground source heat pump (GSHP) system that serves a class room in Turkey, using field measured data, Hepbasli et al. (2004) calculated the exergy efficiency of the GSHP to be about 3%, while its Coefficient of Performance (COP) is around 1.7. They concluded that the compressor is the most inefficient component. Ozgener et al. (2005) studied a geothermal district heating system in Turkey and indicated that exergy losses are mainly due to pumps, heat exchangers, thermal water injection, and leaks in the hot water distribution. The exergy efficiency of the entire district heating system is 46% while the energy efficiency is 41.9%.

Only a few published papers have discussed the energy efficiency of whole air-conditioning systems. For instance, Dunn et al. (2005) found good agreement between the measured system efficiency of air-conditioning systems in office buildings and the simulation results estimated by the TAS and DOE-2 programs. The seasonal energy efficiency of VAV systems in office buildings was estimated as 0.81 (TAS) and 1.19 (DOE-2).

The literature review did not reveal any publication about the application of exergy analysis, concerned about the whole system that includes heating and cooling plants and power plant and delivery systems such as transmission lines for electricity, to HVAC systems for commercial buildings.

The objective of this paper is the evaluation of the performance of two VAV systems used in an office building by combining thermodynamic analysis with the estimation of GHG emissions due to the operation of the VAV systems.

SIMULATION

Description of the building

The description of the existing office building used for the case study is presented in (Zmeureanu et al. 1995). The sample building has five identical floors of about 1,000 m^2 each, and each floor contains five thermal zones: one core zone and four perimeter zones. The hourly space heating and cooling loads were estimated by the DOE-2E program and imported to the EES program (Klein 2004) that is used in this study. The temperature set points are 20ºC for heating and 23ºC for cooling.

Description of studied HVAC systems

Two types of VAV systems are simulated and compared. System no. 1 uses a conventional constant supply air temperature, while system no. 2 uses a discriminatory control, which controls the supply air temperature in terms of the highest cooling loads of all zones. Design parameters are introduced in the following sections.

The schematic of the air distribution system is presented in Figure 1. Free cooling is used on the air side in order to reduce the needs of mechanical cooling.

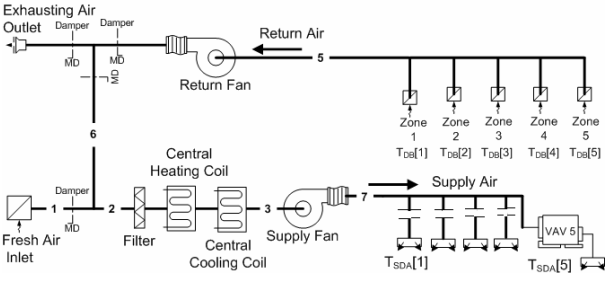

Figure 1. Schematic of a single-duct multi-zone VAV system with reheat coils.

The design supply air temperature is set to 43ºC for heating and 13ºC for cooling (Canadian Commission on Building and Fire Codes 1998). The ventilation and circulation requirements are set to 0.4 l/s·m^2 and 2.0 l/s·m^2, respectively (Canadian Commission on Building and Fire Codes 1998). Each VAV box can vary the supply air flow rate between 100% and 30%

of the design value, when the cooling loads decreases in each zone.

The following components of the hot-water circuit (Figure 2) are considered: a boiler, a hot-water circulating pump, a by-pass mixing pipe, and reheat coils.

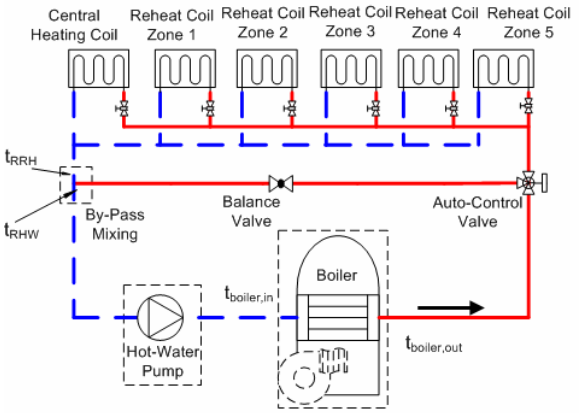

Figure 2. Schematic of the hot-water loop.

All components are well insulated, so heat losses to their surroundings can be neglected. The pump operates at a constant speed. The water temperature of the boiler's outlet is set to 65 ºC. The hot water temperature differences through heating and reheating coils are set to 16°C. The water flow rate through the by-pass line varies according to the thermal loads of heating and reheating coils.

The nominal capacity of the electric boiler is equal to the peak hourly heating load required by all heating and reheating coils. The efficiency of the boiler under the design condition is 80%, and its energy consumption under part-load conditions is estimated as follows (Canadian Commission on Building and Fire Codes 1998):

$$\dot{E}_{in,part} = \frac{\dot{Q}_{rated,boiler}}{\eta_{I,boiler}} \begin{pmatrix} 0.082597 + \\ 0.996764 PLR_B \\ -0.079361 PLR_B{}^2 \end{pmatrix} \quad (1)$$

where the numerator, $\dot{Q}_{rated,boiler}$, of the first term denotes the nominal capacity of the boiler (kW); $\eta_{I,boiler}$ denotes the energy efficiency of the boiler; PLR_B denotes part-load ratios of the boiler in use.

The electricity consumption of the hot water pump is based on the basic pump laws from ASHRAE (2004). The overall energy efficiency of the hot-water circulating pump is 60% (Canadian Commission on Building and Fire Codes 1998) and the total pressure drop of the hot-water circuit is estimated as 250 kPa.

The cooling central plant is composed of a chiller, a chilled water circulating pump, a condensing water circulating pump and a cooling tower (Figure 3). The pumps operate at a constant speed and water flow rate.

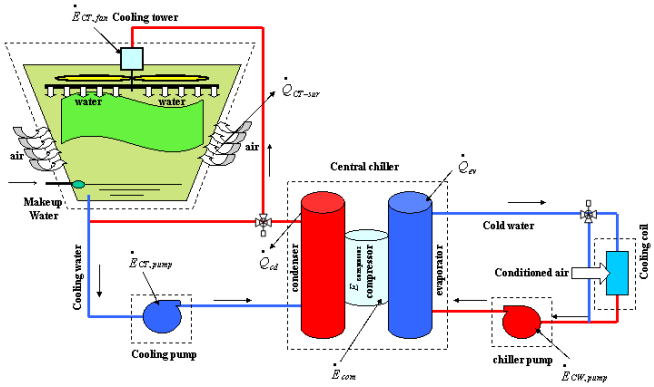

Figure 3. Schematic of central cooling plant.

The temperature of water leaving the evaporator is set at 7ºC and the temperature of water entering the condenser is set at 29ºC (Canadian Commission on Building and Fire Codes 1998). The chilled water temperature difference through the cooling coil is set to 5°C, and the water temperature difference through the cooling tower is set to 6°C.

The nominal capacity of the chiller is equal to the peak hourly load of the cooling coil installed in the air handling unit. The COP of the chiller is a function of the temperatures of chilled water leaving the chiller and condenser water returning from the cooling tower. The following correlation-based model is used to describe the variation of the chiller's COP at full-load and part-load conditions (Solati et al. 2003):

$$\begin{aligned} COP_{FL} &= 8.02711 + 0.00268889 T_{CT,out}^2 \\ &\quad - 0.25483 T_{CT,out} + 0.000367989 T_{EV,out}^2 \\ &\quad + 0.203902 T_{EV,out} + 0.203902 T_{EV,out} \\ &\quad - 0.00396354 T_{EV,out} T_{CT,out} \end{aligned} \quad (2)$$

where subscript "*CT*" denotes the cooling tower and subscript "*EV*" represents the evaporator;

$$\begin{aligned} COP_{PL} &= 7.81922 + 0.00530412 T_{CT,out}^2 \\ &\quad - 0.455848 T_{CT,out} - 0.00926693 T_{EV,out}^2 \\ &\quad - 0.0045712 T_{EV,out} T_{CT,out} + 0.450384 T_{EV,out} \\ &\quad - 7.96205 PLR_C^2 + 11.8861 PLR_C \end{aligned} \quad (3)$$

where "*PLR*" denotes part load ratios, which are equal to the hourly load of the chiller divided by its design load, and subscript "*C*" denotes the chiller.

The fan installed on the cooling tower has an ON/OFF control, and its electric demand is a function of the part-load ratio of the cooling tower and estimated by the following set of equations (Canadian Commission on Building and Fire Codes 1998):

$$\dot{Q}_{CT} = \dot{Q}_{cd} \tag{4}$$

$$PR_{CT} = \frac{\dot{Q}_{CT}}{\dot{Q}_{rated,chiller} * \left(1 + \frac{1}{COP_{FL}}\right)} \tag{5}$$

$$\dot{E}_{CT} = 0.015 * \dot{Q}_{rated,chiller} * \left(1 + \frac{1}{COP_{FL}}\right) * PR_{CT} \tag{6}$$

where the left side of Equation 4 denotes the heat rejected from the cooling tower, in kW; PR_{CT} denotes the part-load ratio of the cooling tower; and the left side of Equation 6 denotes the electric demand of the cooling tower fan, in kW.

In this study, the flow rate of makeup water is about 1% of the volumetric flow rate of condenser water at design conditions (ASHRAE 2004).

The overall energy efficiency of the chilled water circulating pump and the cooling tower water pump is 60% and 70%, respectively (Canadian Commission on Building and Fire Codes 1998). The pressure drops in two water systems, which must be covered by those pumps, are 250 kPa and 182 kPa, respectively.

By taking into consideration the efficiency of power plant and delivery systems, the total primary off-site energy input can be calculated as follows (Wu 2004):

$$\dot{E}_{power,plant} = \frac{\dot{E}_{total}}{\eta_{power,delivery}} * \left(\frac{0.967}{0.8} + \frac{0.011}{0.3} + \frac{0.011}{0.33} + \frac{0.011}{0.431}\right) \tag{7}$$

where $\eta_{power,delivery}$ denotes the efficiency of transmission lines, assumed to be equal to 0.86; 0.967 is the contribution of the electricity generated by hydro power plants, and 0.011 is the contribution of the electricity generated by nuclear power plants, gas fired power plants and oil fired power plants in Quebec (Baouendi 2003); 0.8 is the overall efficiency of hydro-power plants (Ileri et al. 1998); 0.3 is the overall efficiency of nuclear power plants (Rosen 2001); 0.33 is the overall efficiency of oil-fired plants (Kannan et. al 2004); 0.431 is the overall efficiency of natural gas fired power plants (AIE 1998).

The on-site electricity input to the VAV systems, which is the summation of electricity input to all components, is calculated as follows:

$$\begin{aligned}\dot{E}_{total} = {} & \dot{E}_{\sup ply,fan} + \dot{E}_{return,fan} + \dot{E}_{CW,pump} + \dot{E}_{HW,pump} \\ & + \dot{E}_{CT,pump} + \dot{E}_{com} + \dot{E}_{CT,fan} + \dot{E}_{boiler}\end{aligned} \tag{8}$$

Energy and exergy analysis

The COP of a VAV system is defined as the overall thermal output, including the thermal load required by the ventilation air, divided by the total energy input to sustain the system operation, which can be the total on-site electricity input (with respect to the index of energy performance at the system level) or the total off-site primary energy input (with respect to the index of energy performance at the power plant level).

In this study, the dead-state for exergy analysis is taken as the outdoor environment state, defined with the variable outdoor air temperatures and 100% relative humidity. The exergy efficiency is defined as the useful output to the input. For instance, the exergy efficiency of the chiller is the rate of the exergy gained by chilled water divided by the exergy provided to the chiller.

The exergy input to the system is equal to the total electric exergy plus the thermal exergy due to the outdoor air and the makeup water induced to the VAV system, and it is expressed as:

$$\dot{\Phi}_{\sup ply,sys} = \dot{E}_{total} + \dot{m}_{MW} * \psi_{MW} + \dot{m}_{FR} * \psi_1 \tag{9}$$

where ψ_{MW} denotes the specific exergy of the makeup water of the cooling tower (kJ/kg), calculated by Equation 10 (Wark 1995); $\dot{m}_{MW}$ denotes the mass flow rate of makeup water (kg/s); the subscript "1" refers to the air state listed in Figure 1;

$$\begin{aligned}\psi_{MW} = {} & h_{i,MW} - h_{@t_0} - T_0 * (s_{i,MW} - s_{@t_0}) \\ & - 0.416 * T_0 * \left[\ln\left(\frac{P_0}{P_{g@t_0}}\right) + \ln\left(\frac{1.6078 * \omega_0}{1 + 1.6078 * \omega_0}\right)\right]\end{aligned} \tag{10}$$

where $P_{g@t_0}$ denotes the saturated pressure of water at oudoor temperature (kPa); $h_{@t_0}$ denotes the specific enthalpy of the saturated liquid water at the outdoor temperatures (kJ/kg).

The task exergy refers to the exergy needed by the building system and includes the exergy of the thermal energy from or to zones, and the exergy difference between the indoor and outdoor air. In this research, the positive task exergy refers to the exergy needed to be provided by the HVAC system to the building spaces while the negative task exergy refers to the exergy needed to be removed by the HVAC system. The task exergy of the VAV system due to different components is presented below.

The exergy of sensible load is calculated as (Franconi et al. 1999):

$$\dot{\Phi}_{sensible} = \left(\frac{T_o}{T_{DB}[j] + 273.15} - 1\right)\dot{Q}_{sen}[j] \tag{11}$$

where subscript "*o*" denotes the dead state; subscript "*sen*" represents sensible; subscript "*DB*"denotes dry-bulb; term "*j*" denotes the number of zones.

The exergy of latent load is calculated as (Ren et al. 2002):

$$\psi_{MW} = -1.6078 R_a T_o \ln\left(\frac{(1+1.6078\omega_o)\omega}{(1+1.6078\omega)\omega_o}\right) \qquad (12)$$

$$\frac{\dot{Q}_{lat}[j]}{2501+1.87*T_{DB}[j]}$$

where ω denotes the humidity ratio of indoor air; subscript "*LAT*" denotes latent, and subscript "*a*" represents dry-air; term "R" denotes the universal gas constant.

The exergy due to ventilation load is calculated as:

$$\dot{\Phi}_{ventialtion} = \dot{m}_{\min outair}(\psi_5 - \psi_1) \qquad (13)$$

where subscript "1" or "5" refers to the air state listed in Figure 1.

The total exergy input corresponding to the off-site primary energy is calculated as follows:

$$\dot{\Phi}_{power,plants} = \frac{\dot{E}_{total}}{0.86} * \left[\left(\frac{0.967}{0.8}+\frac{0.011}{0.3}\right)+\left(\frac{0.011}{0.431}+\frac{0.011}{0.33}\right)\left(1-\frac{T_o}{T_{flame}}\right)\right] \qquad (14)$$

where T_{flame} denotes the flame temperature (K), estimated as 2200 K for gas and oil-fired power plants; T_o denotes variable outdoor temperatures.

The net exergy flowing through electricity grids is calculated as follows:

$$\dot{\Phi}_{power,delivery} = \frac{\dot{E}_{total}}{0.86} \qquad (15)$$

Estimates of green house gas (GHG) emissions

The equivalent CO_2 emissions, in kg/year, due to the off-site primary energy use are estimated as follows:

$$E_{CO_2} = \frac{(\alpha_1 E_{hydro} + \alpha_2 E_{nuclear} + \alpha_3 E_{gas} + \alpha_4 E_{oil})}{1000} \qquad (16)$$

where E_{hydro}, $E_{nuclear}$, E_{gas}, and E_{oil} are the annual off-site primary energy input to different types of power plants, respectively, in kWh/year; α is the specific equivalent CO_2 emissions (kton/TWh) for different types of power generation plants (Gagnon et al. 2002); α_1 is 15 kton/TWh for the hydro power plant with reservoirs; α_2 is 15 kton/TWh for the nuclear power plant; α_3 is 443 kton/TWh for the gas-fired power plant; α_4 is 778 kton/TWh for the heavy oil-fired power plant.

Simulation

Mathematical models of the two VAV systems are developed and implemented with the Engineering Equation Solver (EES) program, developed by Klein (2004). The simulation time covers 4176 hours, referring to the total time from 8:00 am to 11:00 pm of every day during a year excluding all weekends.

RESULTS AND DISCUSSION

Energy analysis

Figure 4 shows the hourly COP of both VAV systems versus outdoor dry-bulb temperatures. The trend-line of hourly COP shows that the energy performance of VAV system no. 2 is better than that of VAV system no. 1 when outdoor temperatures are in the range of -25ºC to 20ºC. When outdoor temperatures are higher than 20ºC, the two systems have about the same COP.

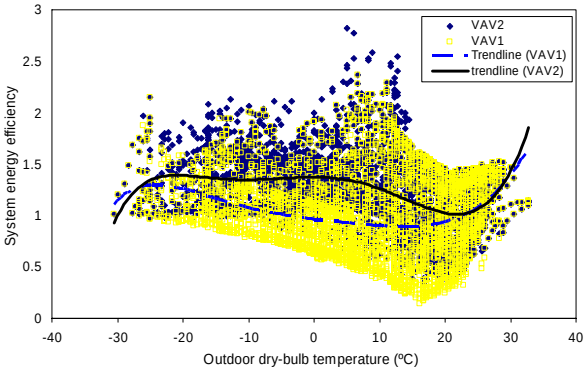

Figure 4. System COP versus outdoor dry-bulb temperature

Figure 5 presents the monthly average COP of both VAV systems. The difference between the simulated COPs of VAV systems is more significant in winter and swing months. The difference is about 44% in October and November but only 4% in July and August. The annual COP of VAV system no. 2, without considering power plants is 1.24, compared with the COP of system no. 1, of about 1.0. The estimated COP of both VAV systems agrees with those predicted by (Dunn et al. 2005): about 1.19 obtained from the DOE-2 program, and 0.81-0.91 obtained from the TAS program. When the efficiency of power plants and delivery systems is included in the evaluation, the annual energy efficiency drops to 0.65 for system no. 1 and 0.82 for system no. 2.

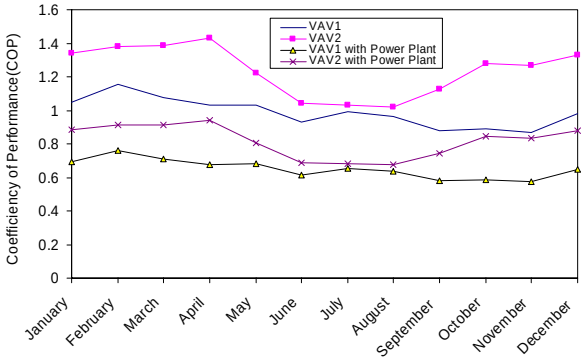

Figure 5. Monthly average COP of both VAV systems.

There is no significant difference between the monthly average COPs of chillers used by both VAV systems (Figure 6). The annual average COP of the chiller is about 4.0 for VAV system no. 1, and 4.1 for VAV system no. 2.

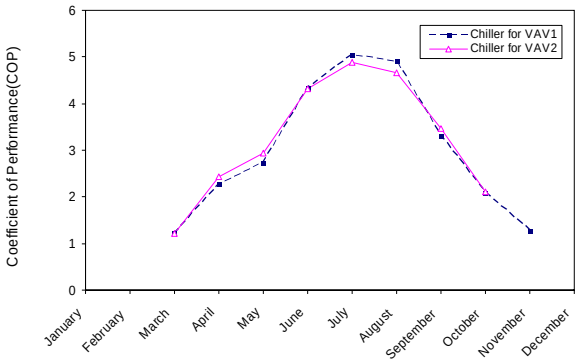

Figure 6. Monthly average COP of chillers.

The annual energy consumption of the boiler for system no. 2 is by 28% less than that for system no. 1, because of the reduced thermal loads of reheating coils. The maximum energy savings reach 39% in April. However, the monthly average energy efficiency of the boiler (Figure 7) used by system no. 1 is higher than that for system no. 2 because the boiler of system no. 1 operates at a higher part-load ratio than the boiler of system no. 2.

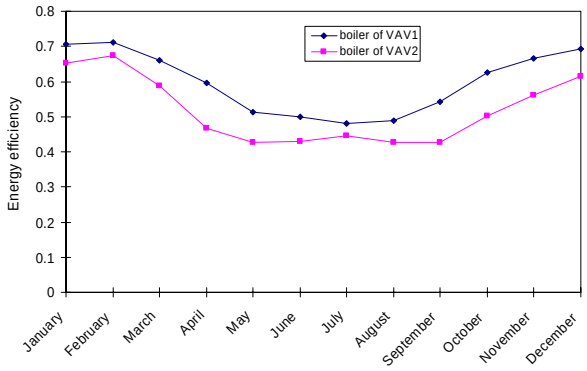

Figure 7. Monthly average energy efficiency of boilers.

The annual energy use intensity is 132.5kW/m^2 for system no. 1 and 103.4kW/m^2 for system no. 2, indicating a reduction of 22%.

Exergy analysis

The difference of exergy efficiency between the two VAV systems is relatively large when outdoor dry-bulb temperatures are in the range of -20ºC to 15ºC, corresponding to the operation of the VAV systems with 100% outdoor air (air-side economizer). The difference is negligible when outdoor dry-bulb temperatures are higher than 20ºC corresponding to the operation with the minimum amount of outdoor air (Figure 8). The highest hourly exergy efficiency of both VAV systems is less than 16%. The low value of exergy efficiency indicates a large potential for the performance improvement of VAV systems through the design and operation. When the efficiency of power plants and delivery grids is not considered in calculations, the annual exergy efficiency of system no. 2 is 2.5%, while it is only 2.0% for system no. 1.

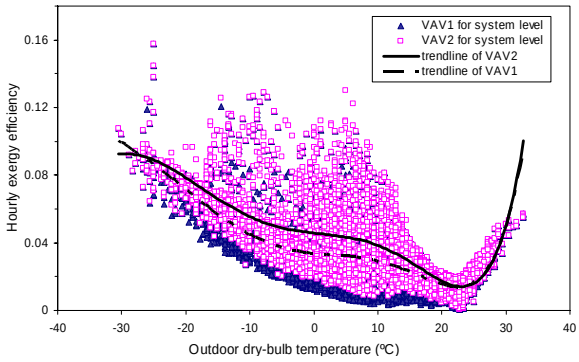

Figure 8. Hourly exergy efficiency of both VAV systems versus dry-bulb outdoor temperature.

The total annual entropy generation within VAV system no. 1 is 2.41 MWh/K, while in the case of VAV system no. 2 is 1.88 MWh/K. Figure 9 shows the distribution of annual entropy generation within major components of VAV system no. 1 when the power plant and transmission lines are not included in calculations. About 65% of the total entropy generation occurs in the boiler.

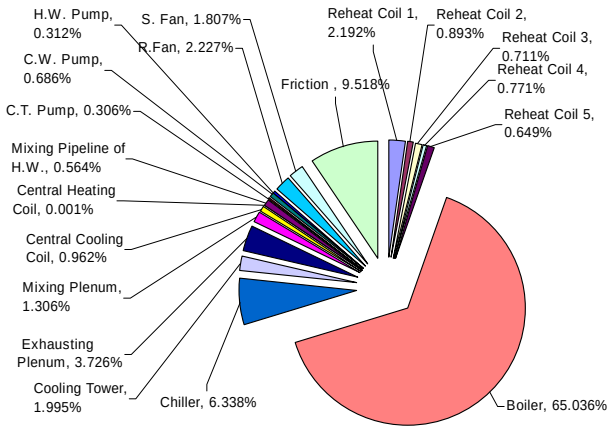

Figure 9. Distribution of total annual entropy generation in VAV system no.1.

Figure 10 details the exergy flows through the following major components: power generation plants, power delivery grids, the VAV system and the building. For VAV system no. 1, the annual exergy supply through the primary off-site energy use is 998.07 MWh/year, and the major exergy destruction (668.284 MWh/year) occurs in the VAV system.

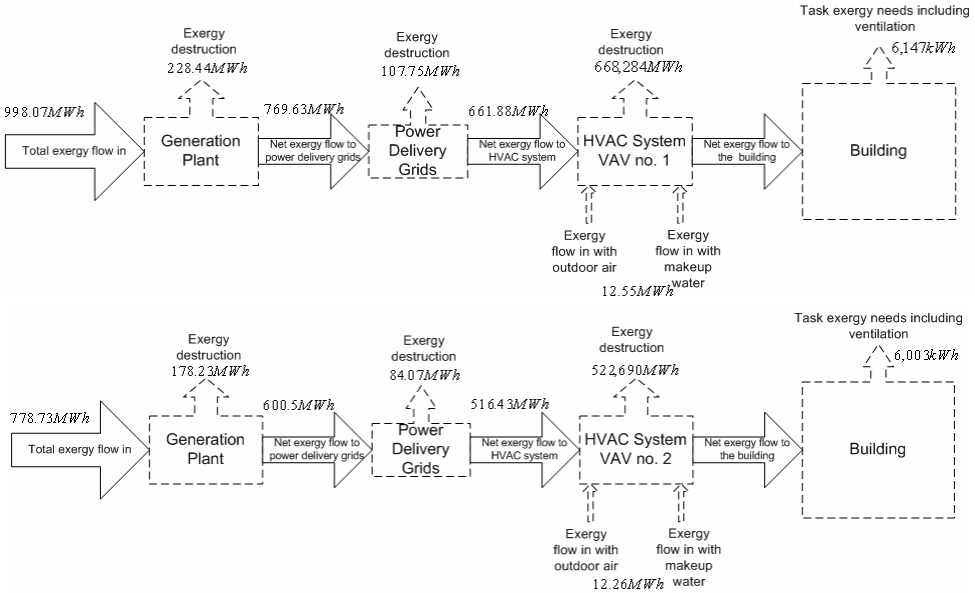

Figure 10. Exergy flow diagrams for both VAV systems including the power plant and delivery systems.

When the efficiency of power plant and delivery systems is included in calculations, the annual average exergy efficiency is 0.6% for VAV system no. 1, and 0.8% for VAV system no. 2. In other words, both VAV systems use less than 1% of the potential useful work that can be obtained through the off-site primary energy use, while more than 99% is lost. Hence, the exergy efficiency can be used as an indicator of the depletion of natural resources.

The analysis of irreversibilities of system no. 1 at the component-level shows that the major exergy destruction is due to the boiler (430.2 MWh/year) and friction losses in the piping, duct systems and equipment (65 MWh/year). The exergy destruction in the cooling plant is relatively small due to the use of free cooling. Potential exergy recovery from the air stream in cooling tower is negligible, if it is compared with the potential exergy recovery from the exhausting air and boiler.

The reduction of friction losses should be an area of concerns for improving the exergy efficiency.

The exergy destruction due to the friction losses can be reduced by selecting low velocity of working fluids in pipes and ducts, avoiding the sudden change of section areas or directions, and choosing materials with smoother surfaces.

The annual largest exergy destruction in the two VAV systems is due to the electric boilers (e.g., 430.24 MWh/year or 65% of the total exergy losses for VAV sytem no. 1). This result reveals the significance of choosing a suitable heating source. The electric boilers can be replaced by other heating sources such as solar or geothermal energy.

The exhaust process is the fourth largest cause of exergy destruction (e.g., 24.7 MWh/year for VAV system no. 1). An air-to-air heat exchanger installed on the exhaust stream and the use of the recovered heat may reduce the exergy destruction in this process.

Since the exergy loss due to the condensate water from the cooling coil is negligible (about 15 kWh/year), no efforts are required for improving the exergy efficiency of this process.

Annual equivalent CO_2 emissions are estimated as 43 tons/yr for VAV system no. 1 and 33.5 tons/yr for VAV system no. 2. This amount of CO_2 can fill a seven-floor office building that is about 23m high with 1000 m^2 of floor area. This is also equal to the annual emissions from more than 10 energy-efficient cars running about 20,000 km/year.

The annual GHG emissions of VAV system no. 2 have reduced by about 22%, compared with those of VAV system no. 1, and can be further reduced by using renewable energy sources instead of fossil fuels or electric energy.

CONCLUSION

In this study, the energy and exergy analyses are utilized for evaluating two VAV systems operating in a large office building in Montreal. The exergy analysis indicates where the largest exergy destruction takes place, and gives an insight into the potential of energy and exergy savings. The largest improvement of exergy efficiency can be obtained by changing the heating source from the electric boiler to renewable energy sources such as solar or geothermal energy.

ACKNOWLEDGEMENTS

The authors acknowledge the financial support from Natural Sciences and Engineering Research Council of Canada and from Faculty of Engineering and Computer Science through the SRT fund.

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