EXERGY EVALUATION OF MECHANICAL VENTILATION SYSTEMS

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ABSTRACT

Energy performance of mechanical ventilation systems in modern low energy and passsive buildings is a crucial factor influencing overall energy performance of building. Energy balance is commonly used tool in evaluation of mechanical ventilation systems. In the case of low energy and passive buildings that tool might be insufficient and should be replaced by exergy analysis taking into account the first and the second Law of Thermodynamics. The paper presents principles of exergy evaluation of mechanical ventilation systems and case study calculations for an office building.

KEYWORDS

Exergy analysis, mechanical ventilation systems, demand controlled ventilation (DCV)

1 INTRODUCTION

Reduction of energy consumption is one of the biggest challenges in the modern world. Buildings consume about 40% of final energy in European Union. Heating and cooling is more than 50% of the annual energy demand of buildings in the operational phase, thus building sector requires significant energy efficiency improvement.

One of the most promising strategies in the improvement of energy performance of buildings is optimal control algorithm or control strategy of ventilation systems. Several control strategies are proposed for ventilation systems. The CO_2-based demand controlled ventilation (DCV) is one of the strategies that allow for the energy use reduction.

The performance evaluation of ventilation systems is usually based on an energy analysis (first law of thermodynamics). The exergy analysis, which takes into account both first and second law of thermodynamics, is much better evaluation tool, which can give better and more accurate indication of system inefficiencies. The results from exergy analysis can be used to assess and optimize the performance of HVAC including ventilation systems [1]. Exergy analysis is not so commonly used, the examples of exergy balance calculations for different buildings and technical systems of buildings can be found in [2,3,4,7,10,14].

The paper present the application of exergy analysis of DCV system installed in office buildings.

2 THE CONCEPT OF EXERGY ANALYSIS

All real thermodynamic processes taking place in nature are irreversible, which means that they are a source of entropy. It is described by the Gouy-Stodoli Law, whose differential form is given by equation 1 [8]:

$$\delta \dot{B} = T_0 \cdot \dot{S}_{gen} \tag{1}$$

where:

$\delta \dot{B}$ – flux of the internal exergy loss of a thermodynamic system in an infinitely short time $d\tau$, W,

$\dot{S}_{gen}$ – flux of total increase of entropy of a thermodynamic system in an infinitely short time $d\tau$, W K^{-1}.

The flux of increase of entropy is very often referred to as the source of entropy. There are six causes of irreversibility of real thermodynamic processes - six potential entropy sources [7, 8]:

- heat exchange caused by a finite temperature difference,
- electric current flow caused by a finite potential difference,
- chemical reactions,
- mixing of fluids,
- mechanical friction
- hydraulic friction.

The exergy balance of the thermodynamically open system can be presented in a mathematical form – equations 2, or a graphical form (Grassmann-Szargut graph) – figure 1. [8]:

$$\dot{B}_{in} = \delta\dot{B} + \dot{B}_{out}^{use} + \delta\dot{B}_{ext} \pm \Delta\dot{B}_{HS} \tag{2}$$

where:

$\dot{B}_{in}$ – flux of exergy entering the control volume of a thermodynamically open system, W,

$\dot{B}_{out}^{use}$ – flux of useable exergy leaving the control volume of a thermodynamically open system, W.

$\delta\dot{B}$ – flux of the internal exergy losses of a thermodynamically open system, W,

$\delta\dot{B}_{ext}$ flux of external exergy losses of a thermodynamically open system, W,

$\Delta\dot{B}_{HS}$ – flux of change of exergy of an external heat source being in contact with a thermodynamically open system, W.

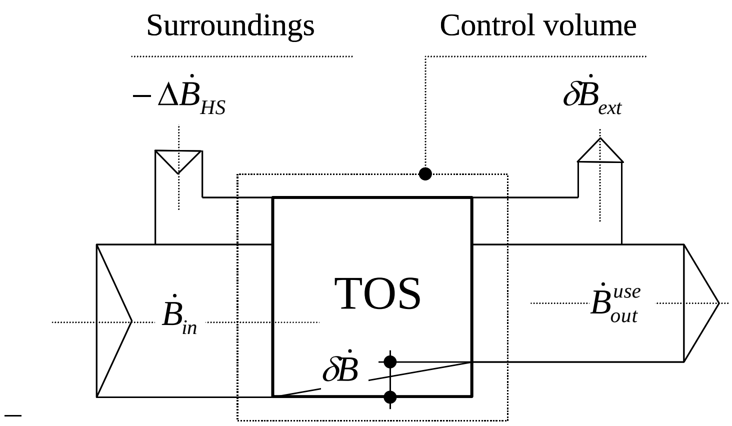

Figure 1. Grassmann-Szargut graph of exergy balance of a thermodynamically – open system (TOS) [8]

Mechanical ventilation systems are dissipative systems what means, that driving exergy introduced to that systems is fully utilized for the coverage of total exergy loss.

From the exergy point of view the choice of mechanical system elements and their operating parameters has to be done in the way that allows for the minimization of driving exergy required for its exploitation [7]:

$$\dot{B}_{drv} \rightarrow \min \tag{3}$$

The following section describes case study calculations of exergy analysis of DCV system installed in the office building located in the city of Poznań, Poland.

3 CASE STUDY ANALYSIS

3.1. Description of office building

The evaluated office building is located in the city of Poznań, Poland. It consists of three stories, each of it is divided into six different zones. The basic data of evaluated building are presented in Table 1.

Table 1: The basic data of building

Parameter	**Unit**	
I storey	-	-
Area	[m^2]	393,3
Volume	[m^3]	1410
II storey	-	-
Area	[m^2]	441,8
Volume	[m^3]	1628
III storey	-	-
Area	[m^2]	441,4
Volume	[m^3]	1626,5
Indoor design temperature	-	-
Winter	[K]	293
Summer	[K]	295
U-value	-	-
Exterior wall	[$W/m^2.K$]	0,111
Roof	[$W/m^2.K$]]	0,095
Floor	[$W/m^2.K$]	0,150

3.2. Description of mechanical ventilation system

The schematic drawing of the mechanical ventilation system is shown in Figure 2. The system consists of three main items: air handling unit, air distribution system and rooms (control zones). The exergy analysis for the main part of air handling unit: rotating heat/mass regenerator, air heater, air cooler, steam humidifier, fans, air filters and air dampers has been performed. Rotating heat/mass regenerator works as long as outside air temperature is lower than ($T_{in,4}$-1K). This study calculates also air heater thermal capacity, cooling capacity, electrical power of inlet and exhaust fans in one example week in winter and summer. Typical meteorological data were taken for the city of Poznań (winter period between 1st and 7th

February and summer period between 1st and 7th July) [13]. All calculations have been conducted using Matlab R2008 simulation tool.

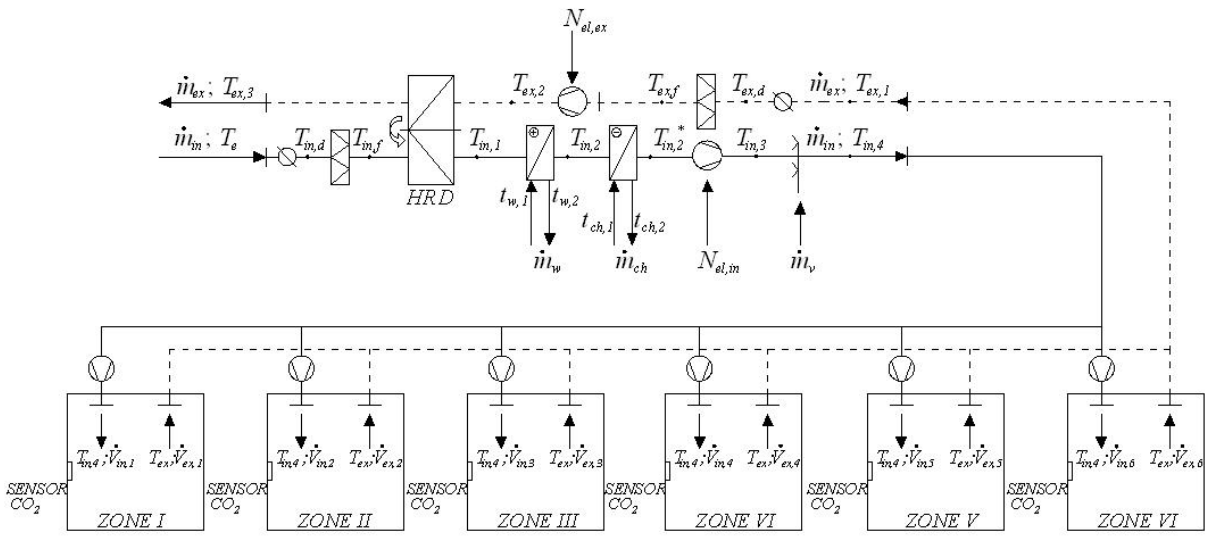

Figure 2: The schematic of mechanical ventilation system.

As the calculation experiment the simulation of exergy performance of one builing's storey divided into six separate CO_2 control zones has been performed. One zone is a conference room – the maximum occupancy is 14 peoples, the rest zones are office rooms – the maximum occupancy density is form 5 to 10 people/zone. The office works from Monday to Friday between 8 a.m. – 8 p.m. Occupancy profiles of all six zones are presented in Figure 3. The ventilation system operates continuously from 8 a.m. – 8 p.m during work day. In all the zones CO_2 sensor were installed, which measure concentration of carbon dioxide. The ASHARE 62 recommended a limit of 1000 ppm CO_2 to satisfy comfort criteria for indoor space. The outside air CO_2 concentration in the location of analyzed building has been 400 ppm.

The air flow per person was determined using equation (4) [9]:

$$\dot{V} = \frac{N}{C_S - C_O} \tag{4}$$

where:

N – CO_2 generation per person, based on activity level; N=0,02 m^3/h,

C_S – CO_2 concentration inside zone, ppm,

C_o – CO_2 concentration outside air, ppm.

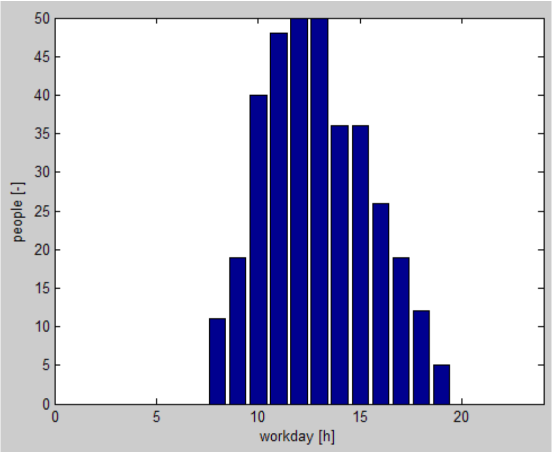

Figure 3: The daily occupancy profile of all six zones

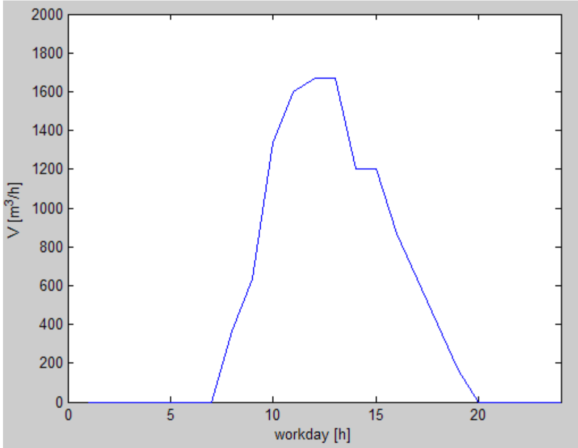

Figure 4: Modelled hourly air flow for all zones during work day

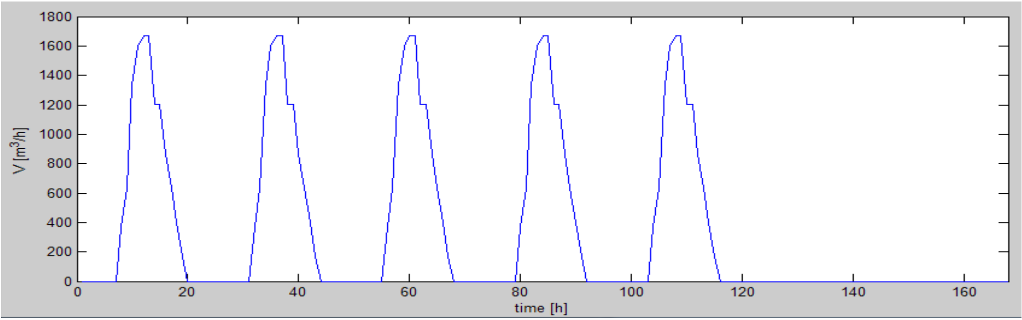

Figure 5: Modelled hourly air flow for all zones during a week

The climatic condition for winter and summer weeks taken as input data are presented in Figures 6 and 7.

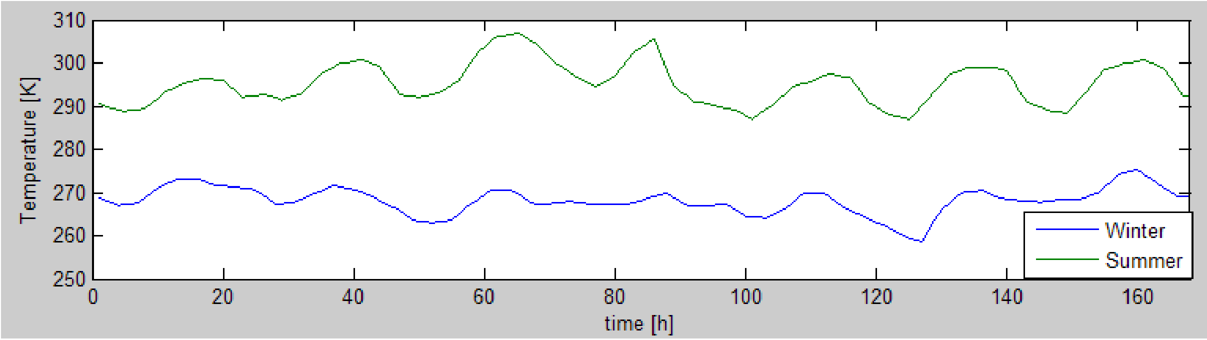

Figure 6: The ambient temperature for reference winter and summer week (data for Poznań from [13]).

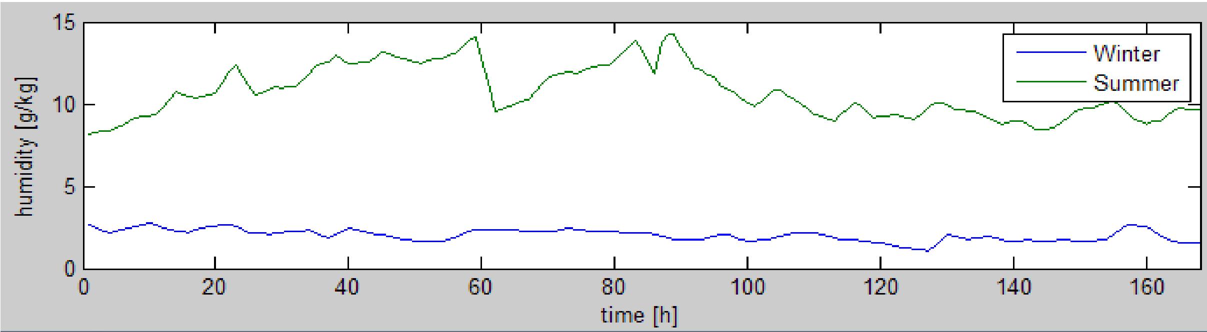

Figure 7: The ambient humidity for the reference winter and summer weeks (data for Poznań from [13]).

The other input data used in exergy simulations are given in Table 2.

Table 2: The input data

Parameter	**Unit**	**Winter**	**Summer**
$T_{in,4}$	[K]	291	293
$X_{in,4}$	[g/kg]	8	Resulting
Δp_{in}	[Pa]	650	650
Δp_{ex}	[Pa]	500	500
$\eta_{HR,t}$	[%]	65	65
$\eta_{HR,x}$	[%]	65	65
$\eta_{v,in,i}= \eta_{v,ex,i}$	[%]	70	70
$\eta_{v,in,em}= \eta_{v,ex,em}$	[%]	90	90
T_{w1}/T_{w2}	[K]	348,15/333,15	-
T_{ch1}/T_{ch2}	[K]	-	280,15/285,15

3.3 Theoretical model

Exergy balance equations have been constructed for each of the items of energy chain creating analyzed ventilation system. They are represented by following equations:

- The flux of the internal exergy loss in the air damper:

- Inlet side:

$$\delta \dot{B}_{D,in} = \dot{m}_{in} \cdot \left(b_e - b_{in,d} \right) \tag{5}$$

- Exhaust side:

$$\delta \dot{B}_{D,ex} = \dot{m}_{ex} \cdot \left(b_{ex,1} - b_{ex,d} \right) \tag{6}$$

- The flux of the internal exergy loss in the air filter:

- Inlet side:

$$\delta \dot{B}_{F,in} = \dot{m}_{in} \cdot \left(b_{in,d} - b_{in,f} \right) \tag{7}$$

- - Exhaust side:

$$\delta \dot{B}_{F,ex} = \dot{m}_{ex} \cdot \left(b_{ex,d} - b_{ex,f} \right) \tag{8}$$

- The flux of the internal energy loss in the heat recovery device:

$$\delta \dot{B}_{HR} = \dot{m}_{in} \cdot \left(b_{in,f} - b_{in,1} \right) + \dot{m}_{ex} \cdot \left(b_{ex,2} - b_{ex,3} \right) \tag{9}$$

- The flux of the internal exergy loss in the air heater:

$$\delta \dot{B}_{AH} = \dot{m}_{in} \cdot \left(b_{in,1} - b_{in,2} \right) + \dot{m}_{w} \cdot \left(b_{w,1} - b_{w,2} \right) \tag{10}$$

- The flux of the internal exergy loss in the cooler:

$$\delta \dot{B}_{CH} = \dot{m}_{in} \cdot \left(b_{in,2} - b^{*}_{in,2} \right) + \dot{m}_{ch} \cdot \left(b_{ch,1} - b_{ch,2} \right) \tag{11}$$

- The flux of the internal exergy loss in the fan:

- Inlet side:

$$\delta\dot{B}_{V,in} = \dot{m}_{in} \cdot \left(b^{*}_{in,2} - b_{in,3}\right) + N_{el,in} \tag{12}$$

- Exhaust side

$$\delta\dot{B}_{V,ex} = \dot{m}_{ex} \cdot \left(b_{ex,f} - b_{ex,2}\right) + N_{el,ex} \tag{13}$$

- Electrical power of the fan motor $N_{el,in}$ and $N_{el,ex}$ depends on the volumetric flow of humid air, its static pressure growth and total efficiency of the fan.
-

$$N_{el,in} = \dot{m}_{in} \cdot v_{in} \cdot \frac{p_{in,3} - p^{*}_{in,2}}{\eta_{V,in}} \tag{14}$$

$$N_{el,ex} = \dot{m}_{ex} \cdot v_{ex} \cdot \frac{p_{ex,f} - p_{ex,2}}{\eta_{V,in}} \tag{15}$$

- The flux of the internal exergy loss in the steam humidifier:

$$\delta\dot{B}_{SH} = \dot{m}_{in} \cdot \left(b_{in,3} - b_{in,4}\right) + \dot{m}_{v} \cdot b_{v} \tag{16}$$

Specific thermal exergy of humid air can be calculated using the equation:

$$b_{in,ex} = c_a \cdot \left[\left(T - T_0\right) - T_0 \cdot \ln\frac{T}{T_0}\right] + T_0 \cdot R_a \cdot \ln\frac{p_a}{p_{a,0}} + x \cdot \left[c_v \cdot \left(T - T_0\right) - T_0 \cdot \ln\frac{T}{T_0}\right] + T_0 \cdot R_v \cdot \ln\frac{p_v}{p_{v,0}} \tag{17}$$

Specific thermal exergy of water can be calculated using the formula:

$$b_w = c_w \cdot \left[\left(T - T_0\right) - T_0 \cdot \ln\frac{T}{T_0}\right] \tag{18}$$

Specific thermal exergy of saturated water vapor (x=1) can be calculated using the equation:

$$b_v = c_v \cdot \left[\left(T - T_0\right) - T_0 \cdot \ln \frac{T}{T_0} \right] + T_0 \cdot R_v \cdot \ln \frac{p_v}{p_{v,0}} \qquad (19)$$

Heating capacity of the air heater can be calculated:

$$\dot{Q}_{AH} = \dot{m}_{in} \cdot c_a \cdot (T_{in,2} - T_{in,1}) \qquad (20)$$

Cooler capacity of the air heater can be calculated:

$$\dot{Q}_{CH:} = \dot{m}_{in} \cdot c_a \cdot (T_{in,2} - T^{*}_{in,2}) \qquad (21)$$

4 CALCULATION RESULTS AND DISCUSSION

The results of the exergy balance calculations, electrical power of the fan motor, air heater capacity and air cooler capacity for one week in winter and one week in summer are presented below.

The results of internal exergy loss of all energy chain in air handing unit for reference winter and summer weeks are presented in Table 3 and Figures 8 and 9.

Table 3: Case study – exergy calculation result

Parameter	**Winter**		**Summer**	
	[W/week]	**[%]**	**[W/week]**	**[%]**
$\delta\dot{B}_{D,in}$	451,62	0,63	351,9	1,97
$\delta\dot{B}_{F,in}$	753,0	1,05	856,4	4,8
$\delta\dot{B}_{HR}$	20165,2	28,2	687,97	3,86
$\delta\dot{B}_{AH}$	19261,8	26,94	0	0
$\delta\dot{B}_{CH}$	0	0	4177,8	23,39
$\delta\dot{B}_{V,in}$	6884,1	9,63	5928,6	33,2
$\delta\dot{B}_{SH}$	17386,8	24,31	0	0

$\delta\dot{B}_{D,ex}$	464,5	0,65	352,2	1,97
$\delta\dot{B}_{F,ex}$	753	1,05	856,4	4,8
$\delta\dot{B}_{V,ex}$	5379,0	7,52	4644,35	26,01
$\boldsymbol{\delta\dot{B}_{drv}}$	**39759,57**	**100**	**9928,96**	**100**

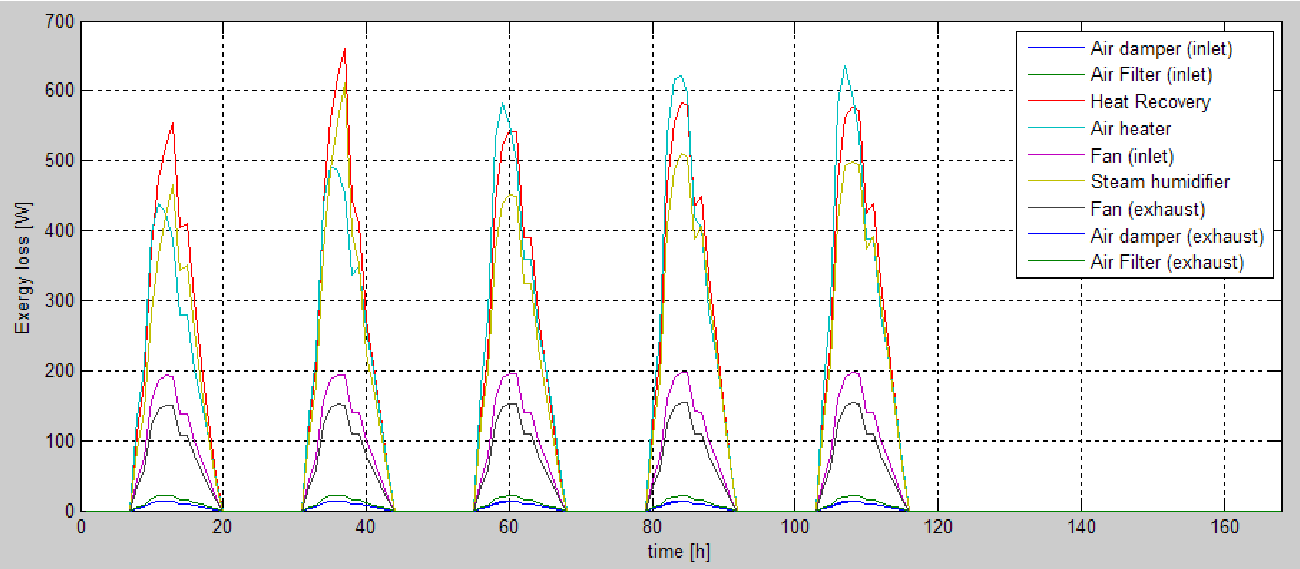

Figure 8: Calculation results – exergy loss in all analysed energy items during reference winter week.

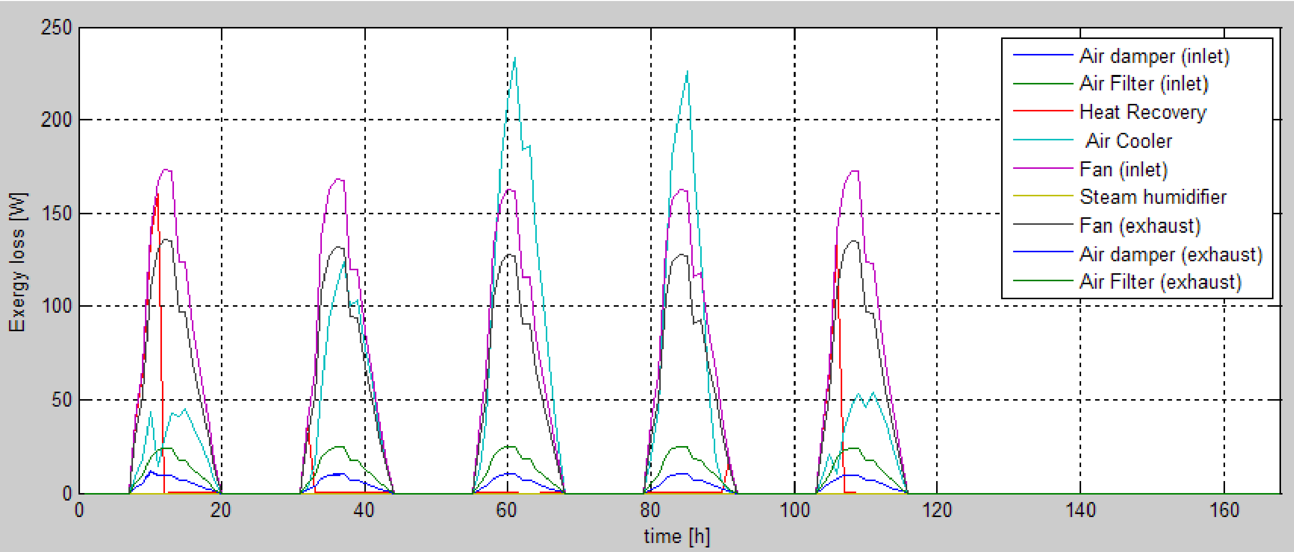

Figure 9: Calculation results – exergy loss in all analysed energy items in reference summer week.

The results of air heater thermal capacity, cooler capacity, electrical power of inlet and exhaust fans for reference winter and summer weeks are presented in Table 4 and Figures 10 and 11.

Table 4: Case study – calculation results

Parameter	Unit	Winter	Summer
Q_{AH}	[kW/week]	89,26	0
Q_{CH}	[kW/week]	0	126,37
$N_{el,in}$	[kW/week]	16,8	17,03
$N_{el,ex}$	[kW/week]	13,11	13,1

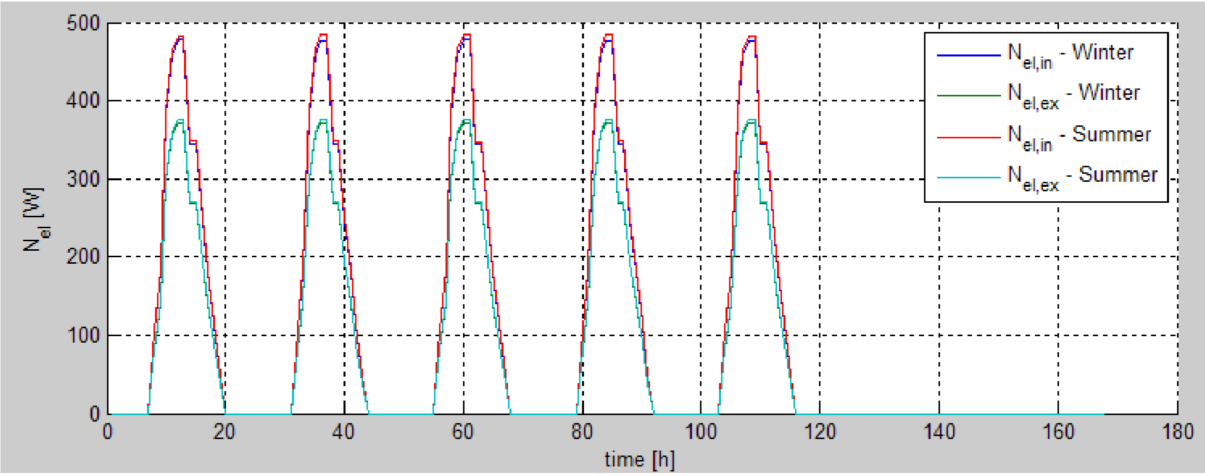

Figure 10: Electrical power of inlet and exhaust fans for reference week in winter and summer.

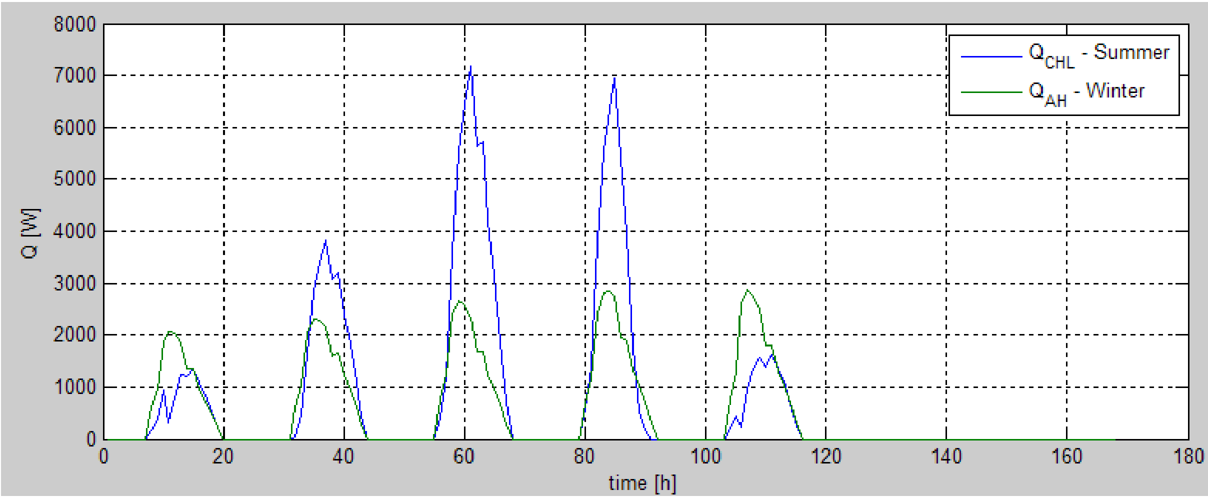

Figure 11: Air heater thermal capacity, cooling capacity for reference week in winter and summer.

Exergy analysis pinpoint the location, where the losses are the highest

In reference winter week three items (heat recovery, air heater, steam humidifier) generated more than 80% exergy losses. Exergy loss in heat recovery can not be perceived as a problem. Main part of heat is recovered from exhaust air, what can be treated in positive way, due to reduction of energy consumption of air heater and steam humidifier. The main sources of irreversibility of air heating process in cold days were: the air heater responsible for approximately 27% of the total flux of the exergy loss and air humidifier responsible for 24% of the total flux of the exergy loss.

In reference summer week the highest exergy losses were caused by fans and air cooler. Heat recovery works if temperature of outdoor air is below 292K, thus during summer season it is usually turned off, because of significant cost of work to expected profits. Supreme exergy loss during warm days is caused by air cooler, thus air handling unit needs improvements to limit exergy loss in this area. Exergy loss on fans are similar during whole year – both during summer and winter season.

During winter week exergy loss is more than four times higher than during summer season. In summer more devices are turned off, because of the high temperature and humidity and the fact that air cooler does not work as a dehumidifier – air cooler reduces the inlet air temperature only. If HVAC system controls also humidify, the exergy loss in summer would be much higher.

The above-mentioned three items of the energy chain - air heater, air humidifier in cold days and air cooler in warm days should be taken into detailed consideration in the optimization procedure.

5 CONCLUSIONS AND FUTURE WORK

Exergy analysis tool is the powerful tool for analysing, assessing, designing, improving and optimizing different systems and processes. Exergy analysis has a lot of benefits. This methods can assist for evaluation of the thermodynamic values of the energy products. Exergy losses clearly pinpoint the location, causes and sources of deviations from ideal conditions of system operation. Exergy efficienciy is the measure of the approach to ideal [11].

Subsequent steps of presented analysis should include: (i) exergy analysis for other main items in HVAC system: air distribution system and office rooms, (ii) chemical exergy analyses of the fossil fuels needed for the air handling unit operation and (iii) analysis of vetilation system for one typical year.

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