Heat recovery-heat pump system with a thermal storage: A case study from an industrial application

Ehab Foda, Kai Sirén, Janne Hirvonen

Department of Energy Technology, School of Engineering, Aalto University, Espoo, Finland, Phone: +358947023598, Fax: +358947023611, e-mail: ehab.foda@aalto.fi

1. Introduction

The main objective of this study was to computationally analyze and assist the design of an energy-efficient HVAC system for a new industrial hall. The hall is planned to be in Kokkola city at the north-west of Finland. The production in this hall of fitness equipments will involve the use of laser tube-cutting machines. The electrical efficiency of such machines is around 0.1 and therefore, they require special cooling that is necessary for their functioning. The idea to utilize the extracted energy from the cooling of the laser machines in the HVAC system was the core of this work. According to the manufacturer's specifications, the temperature level of the cooling water, outflow from the laser machines, is not allowed to be higher than 25°C. Then, the machine's cooling water cannot be directly used even in a low-temperature space heating system. A heat pump system was then thought to be in-lined with the machines' cooling circuit to provide the supply water for the space heating system. The conceptual design included the use of the hall's fire-fighting water tank (447m³) to serve as a thermal capacitance between the machines' cooling circuit and the heat pump at the evaporator side. Furthermore, an outdoor water tank or pool (6000m³) was also adopted in the conceptual design to serve as an alternative heat source and sink during winter peaks and summer operation respectively. The use of ground boreholes, instead of the outdoor tank, was excluded for financial reasons.

2. Material and Methods

2.1. Case study

A new industrial hall belongs to a known company in the field of fitness equipments production is planned to be located at the north-west of Finland in Kokkola city. The design work of the hall started in 2009 with the involvement of our department to develop and analyze a conceptual design for the energy production of the space heating system. This was based on utilizing the heat wasted from industrial processes, particularly related to the cooling of laser tube-cutting machines. The total floor area of the hall is 7380m² with height up to 8m. The hall consists of 8 main sections: laser cutting (including 4 machines); welding, storage, assembly; main entrance and general space; dining and meeting; offices and shipping. Total number of occupants is expected to be around 150 workers. The production in the hall is planned to be performed with two working-shifts (i.e. 16h).

2.2. Conceptual design of the space heating system

The design of HVAC system for such an industrial application aimed at utilizing the extracted energy from cooling the laser machines along with possibilities for thermal storage. The extracted heat from the 4 laser machines is about 90% of its rated power (i.e. totally 305kW) from which 10% is transferred directly to the hall air. Then, the total recovered heat through the cooling water circuit is around 252kW. Figure 1 shows a schematic of the proposed system. The fire-fighting (F.F) tank (447m³) is used as the primary heat source/sink in the system. In accordance to the manufacturer's specifications, a limitation on the temperature of the cooling water (inflow/outflow) was adopted to be 20°C/25°C. A secondary heat source/sink in the system was initially proposed to be a configuration of ground boreholes to allow seasonal storage. However, this option was replaced for financial reasons with a 6000 m³ outdoor pool or tank. Then, a cooling tower was included in the circuit of the space heating system to enable the use of the heat pump in the cooling of the laser machines when needed under peak summer conditions.

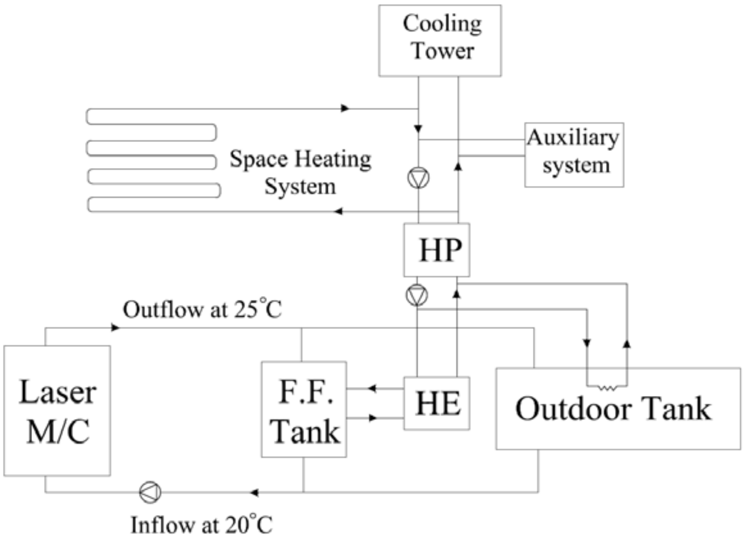

Figure 1 Schematic of the system

2.3. Building energy dynamic simulator

A model for the hall was created on IDA-Indoor Climate and Energy (ICE) 3.0 simulation software to hourly simulate the building energy. IDA-ICE 3.0 is a dynamic multi-zone simulation application for accurate study of thermal indoor climate of individual zones as well as energy consumption of entire building [1]. The model construction was based on preliminary architectural design and the Finnish code [2,3]. Table 1 shows the U-values of the construction.

Table 1. Construction U-values

Item	U-value (W/m²K)
External wall	0.2
External roof	0.16
Floor in contact with ground	0.24
Windows and glass façade [a]	1.1

[a] This value was assumed by the authors based on preliminary information.

A time schedule with two working-shifts was implemented into IDA-ICE 3.0 to schedule the internal gains due to machines, people, lighting and other appliances over working days. The internal gains were entered according to the data provided by the hall designer for the different zones. Losses due to thermal bridges and air infiltration were estimated for the construction at each zone. In the simulation, a floor heating system was used at the different zones as the main space heating system. The ventilation air handling units (AHU) were implemented into IDA-ICE 3.0 with constant air volume (CAV) supply scheme and according to the initial sizing by the involved HVAC designer. Table 2 shows the characteristics of the different AHU's. The AHU type on IDA-ICE 3.0 was equipped with a heat recovery battery. The efficiency of the heat recovery was assumed to be 0.7 as an annual average value. The inlet air temperature and the power of the AHU's were then calculated by the simulation software. The hourly simulation of the building energy was performed based on weather data of the Finnish Zone III-climate. The output of the simulation was based on 2010 calendar.

Table 2. Ventilation air handling units description

Item	Description	Air volume flow rate (l/s)	Supply temperature (°C)
AHU 1	Assembly, shipping and storage space	3000	17
AHU 2	Laser machines section	6000	17
AHU 3	Welding section	6000	17
AHU 4	Entrance and general spaces	1500	20
AHU 5	Offices	2500	20
AHU 6	Dinning and meeting room	1000	20

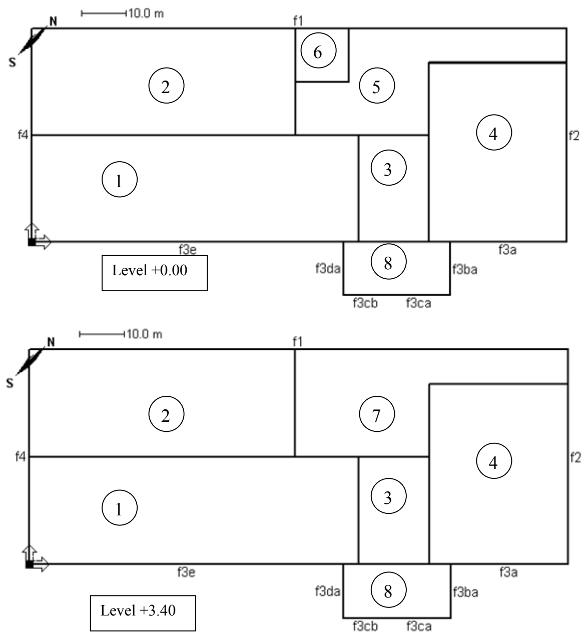

Figure 2 *IDA-ICE 3.0 model (1.Laser section, 2.Welding section, 3.Storage, 4.Assembly section, 5.Main entrance and general space, 6.Dining and meeting, 7.Upper level offices, 8. Shipping)*

2.4. System modelling

The space heating hourly demand, output from IDA-ICE 3.0, was the input to a heat pump/tank model that was created to analyze the hourly performance of the system. The recovered heat from the laser machines was entered to the model based on full hours of the two working-shifts. The model then estimates the hourly water temperatures in the fire-fighting and the outdoor tanks, the heat pump hourly/average annual coefficient of performance *COP*, the compressor hourly power and the tanks' hourly charged/ discharged energies.

In the model, the hourly supply water temperature (T_s) to the space heating system is calculated from a linear correlation with the outdoor air temperature T_o as:

$$T_s = -0.326 T_o + 26.52 \text{, (°C)} \tag{1}$$

An estimation of the conductance of the space heating system (G_r) was initially carried out based on the space heating peak power. Then, the return temperature (T_r) is calculated in the model iteratively with the water mass flow rate (m_c) from the following two equations:

$$T_r = T_{room} + \exp\left[\ln(T_s - T_{room}) - \frac{G_r}{m_c c_p}\right], \text{ (°C)} \tag{2}$$

$$m_c = \frac{Q_c}{c_p}(T_s - T_r), \text{ (kg/s)} \tag{3}$$

where

T_{room} is the room air temperature (°C)

c_p is the water heat capacity (J/kg)

Q_c is the condenser's produced heat (W)

In the iterations, the mass flow rate is calculated based on the covered demand (produced heat) upon the calculation of the compressor power. The compressor was sized to cover only 80% of the peak power. The condenser conductance (G_c) was estimated at the space heating peak power by assuming that the condenser temperature (T_c) is 5 K higher than the water supply temperature (T_s). The condenser temperature is then calculated in the model from:

$$T_c = \frac{\exp\left(\frac{G_C}{m_c c_p}\right) T_s - T_r}{\exp\left(\frac{G_C}{m_c c_p}\right) - 1}, \text{ (°C)} \tag{4}$$

The water hourly temperature in the fire-fighting and outdoor tanks was explicitly calculated using the following equation:

$$T_t^{n+1} = T_t^{n} + \frac{(Q_{laser} - Q_{ev} - U_{gr} A(T_t^{n} - T_{gr}) + U_{air} A(T_{room} - T_t^{n}))}{(\rho c_p V / \Delta\tau)}, \text{ (°C)} \tag{5}$$

where,

T_t^{n+1} is the tank temperature at the new time step (°C)

T_t^{n} is the current tank temperature (°C)

Q_{lase} is the laser recovered heat (W)

Q_{ev} is the evaporator power (W)

U_{gr} is the ground/tank U-value (W/m²°C)

A is the tank top or bottom area (m²)

T_{gr} is the ground temperature (°C)

U_{air} is the room overall heat transfer coefficient (W/m²°C)

T_{room} is the room temperature (°C)

ρ is the air density (kg/m³)

V is the tank volume (m^3)
$\Delta\tau$ is the time step (h)

In the model, the connection of the heat pump's evaporator was switched over the outdoor tank when the water temperature in the F.F. tank approaches 7°C. The water temperature in the outdoor tank was not allowed to drop below -5°C as a rough modeling for the freezing process in winter operation.
The evaporator temperature (T_e) is calculated iteratively with other parameters as the *COP*, compressor power (*P*) and the brine mass flow rate (m_e) from:

$$T_e = \frac{\exp\left(\frac{G_e}{m_e c_{pb}}\right) T_{b2} - T_{b1}}{\exp\left(\frac{G_e}{m_e c_{pb}}\right) - 1}, \text{ (°C)} \tag{6}$$

where

c_{pb} is the brine heat capacity (J/kg)
G_e is the evaporator conductance that was taken equal to G_c (W/K)
T_{b2} is the brine return temperature (°C)
T_{b1} is the brine supply temperature (°C)
The brine temperature difference was assumed to be constant (=5K) in these calculations.

The hourly *COP* is calculated in this iteration loop from:

$$COP = \eta_c \frac{T_c + 273.15}{T_c - T_e} \tag{7}$$

where, η_c is the compressor power factor or cycle efficiency and it was taken as a constant value equals to 0.6 based on technical data given for a scroll compressor. The following two equations are used in the model to calculate the compressor and the evaporator powers.

$$P = \frac{Q_c}{COP}, \text{ (W)} \tag{8}$$

$$Q_e = Q_c - P, \text{ (W)} \tag{9}$$

The model calculations were carried out using VBA programming. The iterative calculations were carried out with the absolute error value ≤0.01.
The seasonal performance factor (SPF) based on whole year simulation is then calculated by dividing the useful energy for the space heating by the energy consumption of the compressor and pumps as:

$$SPF = \frac{E_{condenser}}{E_{compressor} + E_{pump}}, \tag{10}$$

where the pumping energies are roughly estimated in the model based on assumed circuit resistance and mass flow rates in the different circuits.

3. Results and discussion

In this work, hourly simulation of the proposed heat recovery-heat pump/tanks system linked to the building load was carried out to evaluate the system's dynamic performance. Figure 3 shows the estimated heating and cooling demand on a duration diagram from the simulation by IDA-ICE 3.0. The total annual heating energy demand was 920MWh/a while the cooling demand was only 61MWh/a. Based on the sizing of the heat pump, the maximum heating power covered by the system was 600kW. An auxiliary system (100kW) is then needed to cover the peak hours for the space heating and possibly the domestic hot water (DHW) demand. The cooling peak power was around 140kW. The minor cooling demand was thought to be covered using floor cooling along with the ventilation system. Figure 4 shows the compressor power on a duration diagram. The compressor was sized to cover 80% of the peak heating demand. The maximum power of the compressor is 110kW. The annual compressor energy was estimated to be 131MWh/a.

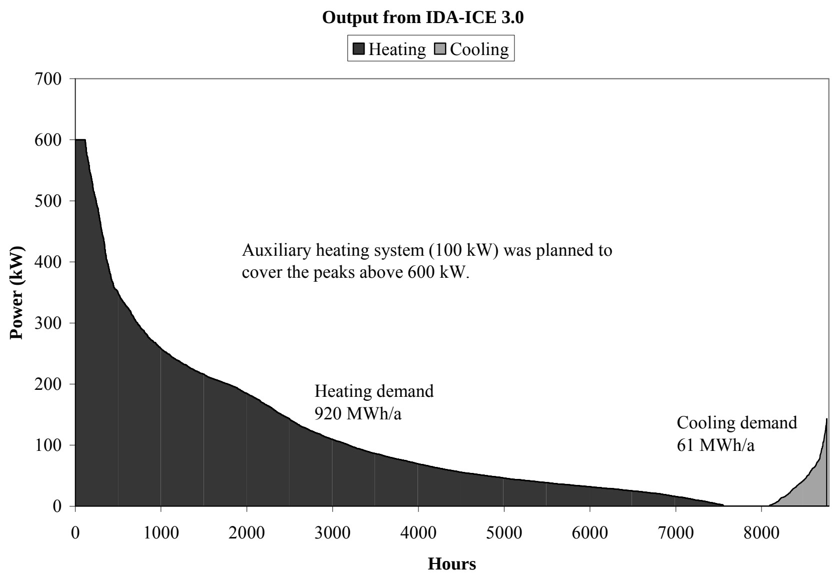

Figure 3. Heating and cooling demand on a duration diagram

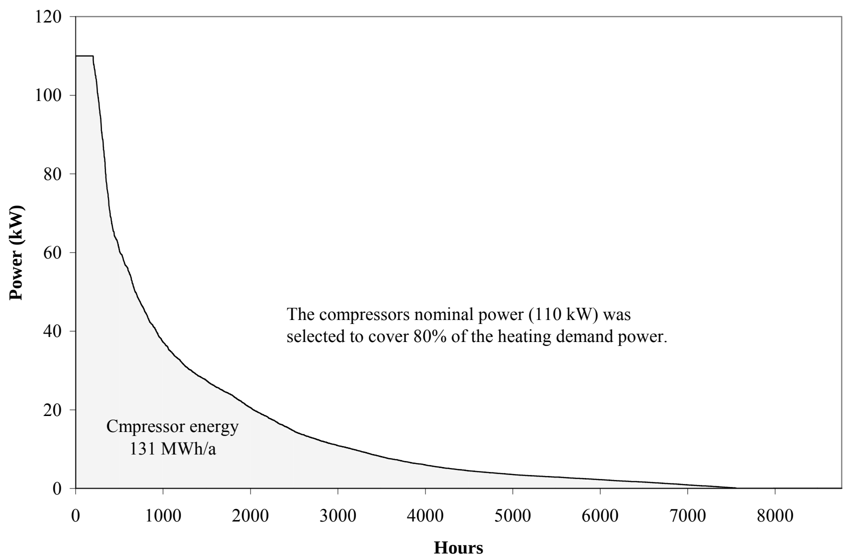

Figure 4. Compressor's energy on a duration diagram

In the simulations, the water in the fire-fighting tank was limited by a range from 7-20°C. The lower limit was used to avoid freezing and enable the tank's primary function with the sprinkler fire-fighting system in the emergency use. The upper limit was according to the permissible supply temperature for the cooling of the laser machines. The outdoor tank was used when these limits were reached. The hourly water temperature in the outdoor tank varied from the freezing point in the early wintertime to 24°C near the end of summertime.

Figure 5 shows the hourly temperatures of the water in the two tanks along with the outdoor air temperature (from the weather file of Zone III-climate). As can be seen, the water temperature of the tanks over a year duration dropped by 7°C and therefore, the tanks may need seasonal recharging before the next heating season. The temperature of the water in the outdoor tank (shown on Figure 5) was mainly affected by the outdoor conditions much more than by the extracted heat with the heat pump operation. The estimated water temperature of the two tanks in summertime indicates critical need for supplementary source of cooling for the laser machines. The use of the ground water with the outdoor tank was recommended. In addition, the use of the heat pump may also be necessary at this critical period and therefore a cooling tower was included in the conceptual design.

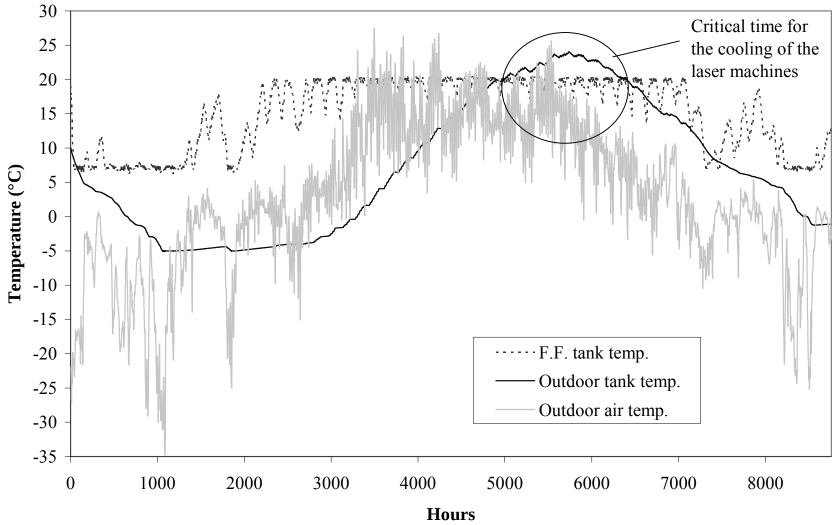

Figure 5. Hourly temperatures of the tanks' water and outdoor air

Figure 6 shows an example of the dynamics of the F.F tank with the heat demand and the recovered heat over a week period. As can be seen, with such average heating demand the tank's stored energy is totally discharged near the end of the weekend. This occurred even sooner under peak winter conditions. In these cases, the operation of the heat pump was switched over the outdoor tank; and hence, resulted in reducing the coefficient of performance (*COP*) of the system.

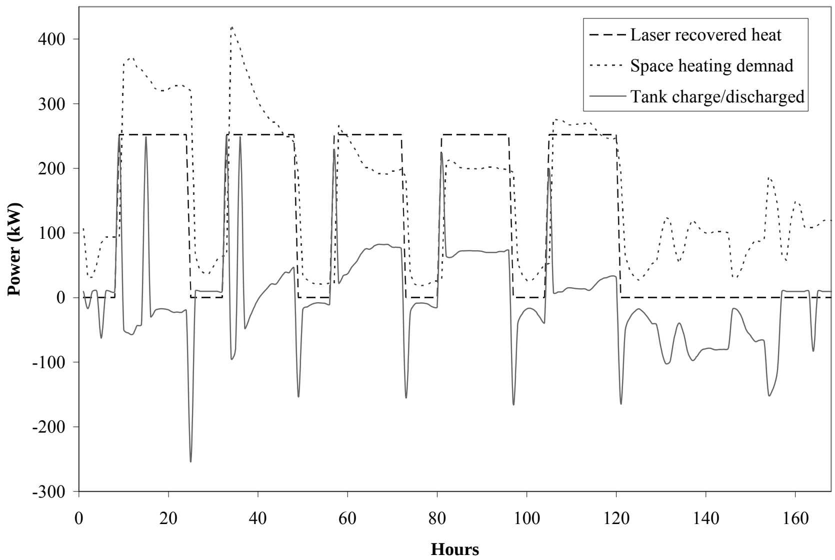

Figure 6. Dynamics of the F.F. water tank over a week period

Figure 7 shows the system energies on a pie chart. The estimated recovered energy from the laser machines was 1052MWh/a while the estimated annual heating demand was 920MWh/a. However, the recovered energy did not meet the space heating demand since the recovered heat in summer-time was mostly extracted without a beneficial use (with the lack of seasonal storage) else than the DHW demand. The current design aimed at maximizing the use of the recovered energy from the laser machines with the least initial cost. The annual average *COP* calculated from the annual produced/ consumed energies of the heat pump system was 7.1. The annual pumping energy was estimated to be around 6MWh/a; hence, the seasonal performance factor (excluding the auxiliary system) was 6.75.

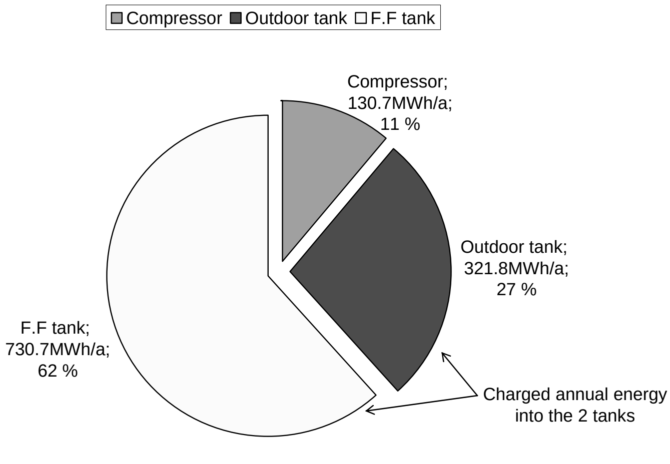

Figure 7 System's annual energies breakdown on a pie chart

4. Conclusions

In this work, the dynamic performance of a heat recovery-heat pump/tanks system was evaluated on hourly basis linked with the building load. The results showed that the system has a considerably high seasonal performance factor of 6.75. The estimated recovered energy from the laser machines was

1052MWh/a while the estimated annual heating demand was 920MWh/a. However, the recovered energy did not meet the space heating demand since the recovered heat in summer-time was mostly extracted without a beneficial use (with the lack of seasonal storage) else than the DHW demand. The recovered heat was mainly through the fire-fighting tank (i.e. 730MWh/a) while it was through the outdoor tank mainly during summer operation (322 MWh/a). The hourly water temperature in the outdoor tank varied from the freezing point in early wintertime to 24°C near the end of summertime. The tank temperature in summertime may cause a problem for the cooling of the laser machines. Therefore, the use of ground water as an emergency backup with the outdoor tank was recommended. In addition, the heat pump connected to a cooling tower (at the condenser side) may be also used in summertime operation for the cooling of the laser machines. From energy conservation perspective, the system's seasonal performance could be further improved by using seasonal storage with ground heat exchanger. However, the cost-efficiency of this solution was not investigated and it was excluded by the owners for financial reasons.

5. References

[1] IDA-ICE 3.0 (IDA Indoor Climate and Energy) http://www.equa.se/eng.ice.html
[2] Finnish code for thermal insulation C3–2007, Ministry of the Environment
[3] Finnish building code D5, Ministry of the Environment