Heat Pump for Low Temperature Condensing Heat Utilization in a Hockey Ice Arena

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Abstract

There is an effort to decrease energy consumption in ice arenas as much as possible. The single largest energy consumer in ice arenas is the refrigeration system, therefore, it is the natural target for efficiency improvements. The heat which arises as a side product of the cooling system can be reused to cover the heat requirements in the arena or in neighborhood buildings. The refrigeration heat has different temperature levels. Technically, the biggest part is represented by low temperature condensing heat. The utilization of the heat is done by the contribution of the open cycle type of heat pump, equivalent to a second compression stage of the main refrigeration cycle. A case study of the system in an ice arena in the Czech Republic will be presented in the paper. The energy consumptions for heating, domestic hot water and technological water heating were measured as well as the working hours of the heat pump and the chiller. Nowadays, the system only uses part of the energy to cover the requirements in the ice arena. Unfortunately, the utilization of low temperature condensing heat is usually maximum 30 % of the total amount. The low percentage is caused by a time mismatch between the delivery and consumption. To increase the amount of condensing heat usage, the possibilities of its utilization in the neighborhood will be analyzed on a model.

Selection and/or peer-review under responsibility of the organizers of the12th IEA Heat Pump Conference 2017.

Keywords: Ammonia Heat Pump; Condensing Heat; Superheat; Ice Arena

1. Introduction

The main purpose of this paper is a project of gathering and using thermal energy that would normally be rejected to the ambient environment. In other words, the potential for recovering thermal energy from the refrigeration system is explored. The paper reports on the development and the case study application of the ammonia heat pump that uses a heat source such as a condenser heat sink from the existing refrigeration plant to produce hot water. Technically, the availability of the waste heat stream is not coincident with the demand. This time mismatch is, therefore, minimized by storage tanks. The existing system is analyzed and, considering the gained results, the improvement of the project is suggested.

2. Ice Arena

The described ice arena is located in a city with 20000 inhabitants, the Czech Republic. The main hockey team is in the district division and youth league and it has trainings during the week and competitions during weekends. The load bearing walls and the roof of the ice arena were modernized in 2002 and the ice sheet was

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reconstructed in 2008. The size of the ice sheet is 29x59 m. There are locker rooms with showers and management offices, their total area being 880 m^2. There are no places for seating the audience. Originally, there was an indirect cooling system with brine. According to its disadvantages like non-uniform quality of ice and slow reaction when heat loads were applied, the original solution was replaced by a direct system with ammonia. Nowadays, the volume of ammonia in the cooling system is 1650 kg. There is not any dehumidification system or air handling system.

The cooling season starts last week in August and finishes on March 31st. During weekdays, the program on the ice sheet usually starts at 3 pm and finishes around 9 pm. During weekends, the program is busier and starts at 8 am. The ice resurfacing machine is in operation 10 to 12 times per day on weekdays and approximately 20 times per day at weekends. The ice resurfacing machine water consumption in 2014 is shown in figure 1, together with energy consumption for snow melting in the snow pit in 2013 and 2014. The snow melting pit is a tank for melting wasted snow and ice removed from the ice sheet by resurfacing machine. The water consumption was read from the water-meter every week; the energy consumption was deducted every month.

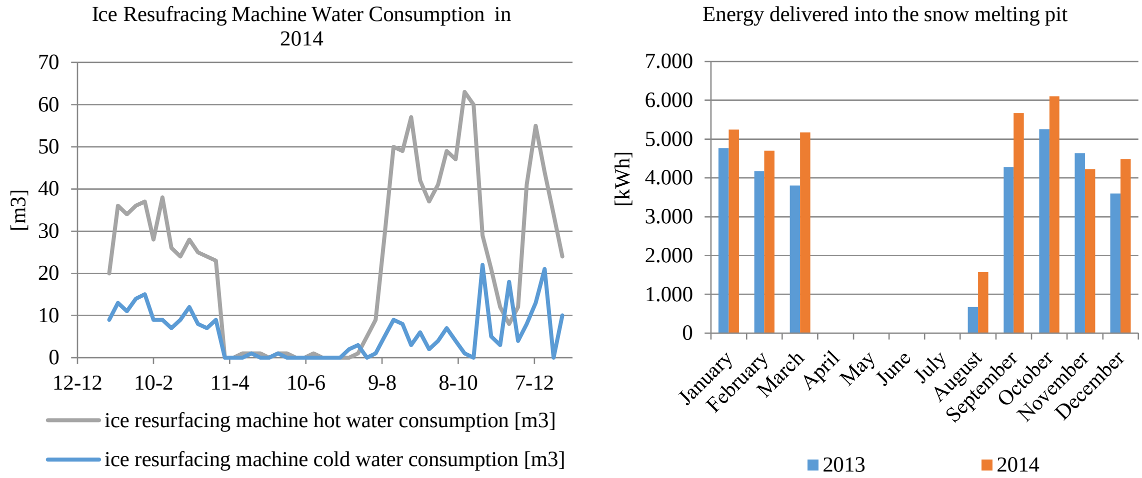

Fig. 1. Water consumption and ice melting energy consumption in the ice arena

To estimate the energy consumptions which are not measured separately, a model of the hall with the ice sheet was created in the Trnsys program [7]. The model is composed of a multizone building, a radiant slab with embedded pipes and ground storage effects which interface with the multizone building, a water cooled chiller, a closed circuit cooling tower and other necessary types to make a model matching the existing ice arena.

3. Cooling and Heating System in the Ice Arena

The cooling system includes two reciprocating refrigeration compressors which have been in operation since 2007. Both compressors have four cylinders and their refrigeration capacity is 252 kW at temperatures - 12/35 °C, the shaft power is 70 kW. The control system enables the operation of the cooling unit at 33 %, 66 % or 99 % of the refrigeration capacity. The isentropic efficiency is 84 %, COP is 3.58 [4]. The cooling unit's parameters are shown in table 1. The temperature of the ice sheet is approximately – 8 °C and is measured by four sensors.

Table 1. Cooling unit's parameters at evaporation temperature of -12 °C [4]

Condensing temperature [°C]	Discharge temperature [°C]	Power input [kW]	Cooling power [kW]	COP [-]
35 °C	127.6	70.49	252.34	3.58
30 °C	114.7	66.13	266.52	4.03
25 °C	101.8	61.15	280.09	4.58
20 °C	88.1	55.22	293.06	5.31
15 °C	74.2	48.76	305.43	6.26

There are three gas boilers in the technical room in a cascade. The boilers are used when the required amount of refrigeration heat is not available. The total heat power of the boilers is 100 kW. The water temperature in the space heating system is 60/50 °C. The heating system also has to be used by fits and starts outside the heating season when there is a lot of wet hockey clothes in the cloakrooms and, therefore, the air humidity is too high. The hot domestic water design temperature is 60 °C.

4. Ammonia Properties

The convenience of a fluid as a refrigerant can be determined from four properties:

- The vapor pressure at the evaporating temperature,
- The critical pressure,
- The critical temperature,
- The molar mass.

The main disadvantage of ammonia as a refrigerant (R-717) is toxicity, smell and flammability and, for these reasons, many ammonia refrigeration systems are of an indirect type using secondary refrigerants. Nevertheless, these disadvantages are paid back by several advantages named below.

The large evaporation latent heat of R-717 does not lead to smaller compressors and gas pipes lines because the reduced mass flow is compensated by reduced vapor density due to the low molecular weight. The majority of the heat is recovered from the phase change and, therefore, is available at a temperature slightly below the condensing temperature [2]. The high discharge temperature can be used as a small addition to the main heat recovery or it can be used to heat a smaller mass flow to a much higher temperature. The high thermal conductivity and low viscosity provide high coefficients of the heat transfer in the evaporator without suffering the penalty of an undue pressure drop. As written in [1] the ammonia offers higher heat transfer coefficients than any other common refrigerating fluid.

5. Refrigeration Heat

The waste refrigeration heat has thermal energy in the form of both sensible (superheat) and latent (condensing) heat. The portion of superheat and condensing heat was calculated in the Coolpack program [8] according to the parameters of chillers mentioned in table 1. The running hours of the chillers were taken into account. The chillers are working 2140 hours per season; the months with the longest chiller operation time are September with 529 hours and October with 390 hours.

5.1. Superheat

The amount of superheat depends on the coolant's condensation temperature and its thermodynamic properties, on the level of vapor superheating and on the type of compressor. The high discharge temperature of ammonia compressors allows the condensing condition to be held at 35 °C during summer, with desuperheating of the discharge gas from 127 °C providing additional heating to raise the hot water temperature usually up to 55 °C. There is a heat exchanger added into the cooling cycle shown in figure 4. The total amount of available superheat is 235.4 MWh.year^{-1} at a maximum usable temperature of 50 °C. A considerable part of the superheat arises in September; it is 28 % of the total amount. An inconsiderable part arises in October (18 %), April (15 %), November and March (10 %). During winter months, the total amount is 50.4 MWh.

5.2. Condensing Heat

The coolant goes through an intercooler to the ammonia heat pump. When there is no utilization demand then the heat pump is not in operation and the standard condenser cooling system is used. The total amount of condensing heat is 1216.7 MWh.year^{-1}, which is 5 times more than the superheat. The condensing temperature was calculated considering the outside air temperature and wet-bulb temperature in the city. The monthly refrigeration heat balance during one year and the daily based refrigeration heat balance for two months are displayed in figure 2.

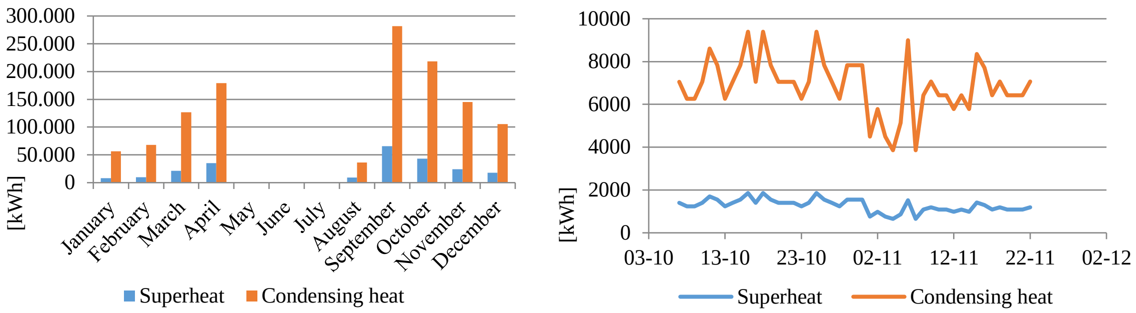

Fig. 2. Superheat and condensing heat calculation results

6. Heat Pump

The chiller's condensing heat temperature is from 25 to 33 °C [5]; unfortunately, its temperature level is not high enough to be used without heating up. Therefore, the ammonia heat pump was added into the cycle. The heat pump's characteristics are shown in table 2. The heat pump was installed in 2011. The temperature leaving the heat pump's condenser is 60 °C.

Table 2. Heat pump's parameters [5]

Condensing temperature	evaporation temperature		15 °C	25 °C	35 °C	45 °C
65 °C	heating capacity	[kW]	252.3	357.8	484.5	633.3
	power input	[kW]	66.1	72.2	74.1	70.4
	COP	-	3.82	4.96	6.54	9.00
70 °C	heating capacity	[kW]	241.2	345.4	470.2	616.5
	power input	[kW]	69.6	77.7	81.8	80.4
	COP	-	3.47	4.45	5.75	7.67

The amount of heat produced by the heat pump is measured every day. The data from October 7th until November 22nd 2015 are presented in figure 3. The measured data in this period were used to verify the Trnsys model of the cooling system. The heat pump was in operation for 2670 hours in 2015.

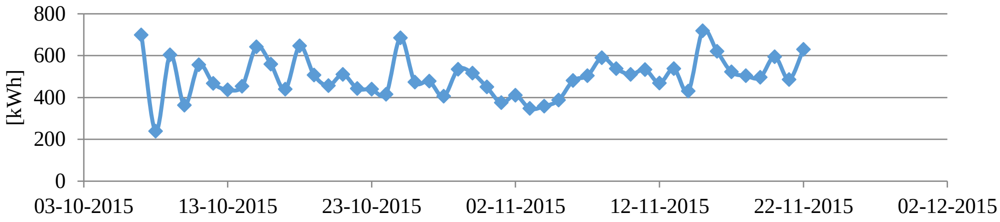

Fig. 3. Measured amount of heat produced by the heat pump

For a better understanding, the principal scheme of the cooling system with the ammonia heat pump is shown in figure 4. The ammonia pipes are presented by black lines. In the cooling cycle, there is an ice sheet as an evaporator, chiller's compressors, a condenser cooling system, a low pressure tank and an expansion valve. The superheat heat exchanger is marked by number 1. To implement the condensing heat utilization, the coolant goes through an intercooler to the heat pump's compressor. The heat pump cycle is represented by the intercooler, compressor, condenser; the low pressure tank is conjoint with the cooling cycle. The refrigeration heat usage system is presented by red lines. The water heated up by superheat is accumulated in a 4 m^3 tank. The water may be used directly or it may be heated up by the heat pump or by the gas boiler. There is also a 1 m^3 accumulation tank heated up by the heat pump and a 0.8 m^3 tank heated up by the boiler. The possibilities of the refrigeration heat usage are written in purple and they will be described more in the following chapter.

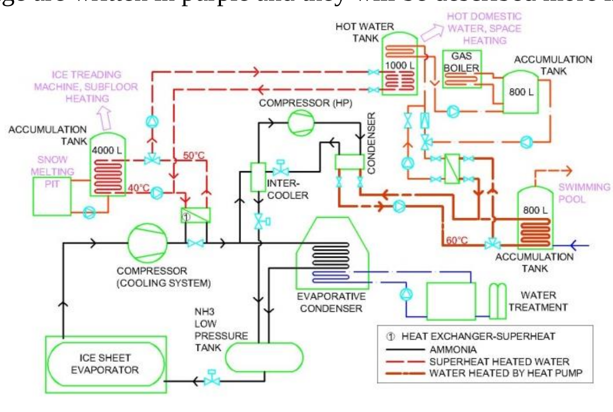

Fig. 4. Principal scheme of the cooling system and the ammonia heat pump

7. Refrigeration Heat Utilization

The possibilities of the refrigeration heat utilization are described according to the place of usage and the temperature level. The waste heat may be used inside the ice arena or in a nearby swimming pool center. To make a proper design, first, the analysis of related energy consumptions must be done. Second, the time mismatch between delivery and consumption must be evaluated.

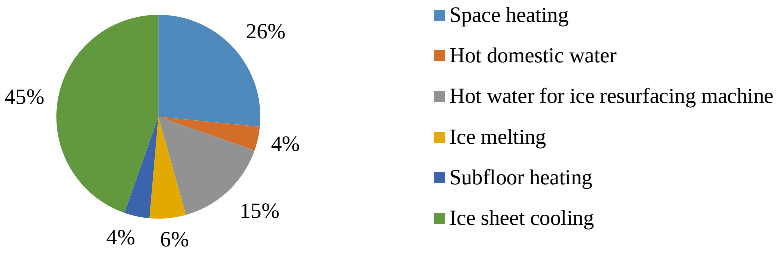

Fig. 5. Ice arena energy consumption

The energy consumption of the cooling system and heat consumers in the ice arena which may be supplied by the refrigeration heat are shown in figure 5. The highest energy consumer is the cooling of the ice sheet. The second is space heating of adjacent spaces such as showers, cloakrooms and offices. A less important energy consumer is also hot water for the ice resurfacing machine preparation. The technological water temperature is between 45 and 35 °C. Other negligible energy consumers are subfloor heating under the ice sheet, hot domestic water preparation and snow melting in the snow pit. As the subfloor heating energy consumption is not measured separately the Trnsys results were considered in the following calculations. The characteristic temperature profiles in the hall with the ice sheet calculated in Trnsys for one week in January are shown in figure 6.

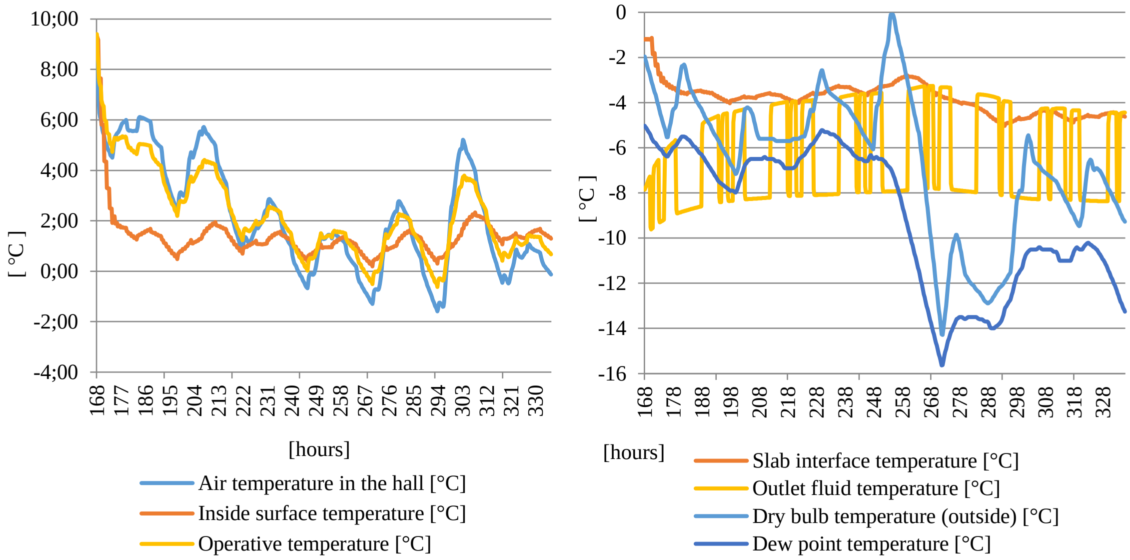

Fig. 6. Temperature profile calculated in Trnsys

7.1. Utilization in the Ice Arena

The options for the superheat utilization in the ice arena are snow pit melting, ice resurfacing machine technological water heating and subfloor heating. Hot water for the ice resurfacing machine is supplied by superheat from 91 % of the total consumption. Because of a time mismatch, in January and February the heat delivered by the heat pump must be used partly. A similar solution is in the case of melting the snow pit and subsoil heating. The snow melting pit energy consumption is supplied from 68 % by superheat, the subsoil heating energy consumption from 57 %. All the above named energy consumers can be supplied first by the superheat, second by the heat pump; no natural gas has to be used. The system controlled as described consumes 55 % of available superheat and 2.6 % of available condensing heat.

The principal options for the condensing heat utilization in the ice arena are space heating and hot domestic water heating. The space heating energy consumption is supplied by the heat pump from 98 %. The rest of the energy demand has to be supplied by gas boilers. The natural gas requirement is 3.1 MWh in May, because the cooling system is not in operation. The overall hot domestic water preparation energy consumption is supplied by the heat pump.

As a result of the monthly energy balance it can be summarized that only 19 % of the total amount of condensing heat can be used in the ice arena. The daily schedule is in the range of a district, therefore, a low ratio of the condensing heat utilization is obtained.

The computational evaluation of the heat pump operation was carried out using the bin method. This method is applied in standard [6]. The bins at the heat source side (chiller's condensing heat temperature) used in the calculation were divided by 1 K. The calculation is also based on the performance characteristic of the heat pump and on other characteristics of the products as included in the system. The total power input of circulation pumps was calculated as 1 kW. The heat pump is in operation for 1570 hours.year^{-1}. The resulting total electricity consumption of the heat pump and related circulation pumps is 35.5 MWh.year^{-1}, the COP being 6.35. The project's simple payback period was 2.24 years [5].

7.2. Analysis of Utilization in the Neighborhood Swimming Pool

Since considerable amounts of condensing heat from the cooling system are available, new options for its usage are investigated. There is one outdoor and one indoor swimming pool in the sports complex next to the ice arena. The refrigeration heat could be used in both pools.

In the indoor swimming pool, the refrigeration heat may be used for space heating of cloakrooms, for domestic water heating and for pool-water heating. The total area of the water surface is 438 m^2, the required water temperature is 29 °C, and the air temperature 30 °C. Specific heat loss from the water surface is 150 W.m^{-2}, the calculated heating energy delivery is thus 55 MWh.year^{-1}. In the cloakrooms, the required air temperature is 22 °C and according to the size of the hall the amount of space heating energy delivery was calculated. The calculated space heating energy demand is 60 MWh.year^{-1}. The hot domestic water consumption was calculated considering standardized consumptions in swimming pools, the resulting amount being 35 MWh.year^{-1}.

In the outdoor swimming pool, the refrigeration heat could be used for keeping the water temperature above zero during winter so that it does not get frozen. If the water in the outdoor pool was frozen, the structure of the pool would be damaged. Otherwise, if the water was drained, the structure would be destroyed, too. Therefore, the water has to be tempered at above zero temperatures. The considered water temperature in the calculation was 5 °C. The area of the water surface is 625 m^2. The designed amount of tempering water energy delivery is 41.3 MWh.year^{-1}.

The principal scheme in figure 4 has already considered this project. The existing scheme is enhanced with an accumulation tank for the purpose of this project. The pipes connecting the ice arena with a technical room for the swimming pools would be 400 m long (200 m supply, 200 m return piping). Nowadays, existing gas boilers are used to supply all heat requirements. The boilers are placed in a technical room in the area of the swimming pools.

The energy balance was made with monthly calculated energy consumptions. First, the available superheat, second the available condensing heat was balanced. The results of the balance present the superheat utilization for the outdoor pool tempering from 17 % and the condensing heat utilization from 78 %. Outside swimming pool water tempering natural gas consumption in January is 1.9 MWh. In the indoor swimming pool, the available refrigeration heat is not satisfactory either because of a time mismatch. Indoor pool-water and hot domestic water heating are supplied from 77 %, but for the remaining 23 % (20.5 MWh.year^{-1}) the existing gas boilers must be used. Space heating is satisfied from 80 % by refrigeration heat, while the remaining 12 MWh.year^{-1} must be supplied by gas boilers, too.

The bin method evaluation was extended with the swimming pools' data. The additional power input of circulation pumps is 0.5 kW. Then, the heat pump is in operation for 2679 hours.year^{-1}. The electricity consumption of the heat pump and related pumps increased at 52.2 MWh.year^{-1}, the COP being 6.4. If the project was implemented, the simple payback period would decrease by 7 months.

8. Energy Savings and Future Potential

To save primary energy sources superheat and condensing heat utilization systems are introduced in the ice arena. When all the previously mentioned energy savings measures are implemented, the total maximum energy saving assumption is 355 MWh.year^{-1} in the ice arena and 156 MWh.year^{-1} together in both swimming pools. This is equivalent to 48500 m^3 of natural gas. Since the calculated total amount of natural gas delivery is 37.5 MWh.year^{-1} (3570 m^3) significant energy savings could be reached. The heat pump using waste heat which would otherwise be released into the atmosphere provides a decrease in the natural gas consumption. Technically, high coefficients of performance are achieved. The simple payback period is shorter than the service life of the heat pump. Moreover, an ecological coolant with zero QWP and ODP is used for the heat pump operation.

Since 24 % of the total available superheat and 73 % of condensing heat is released to atmosphere there are possibilities of the presented system extending. To increase the heat pump's future potential a more sophisticated refrigeration heat accumulation solution or other possibilities of utilization would have to be introduced. Appropriate examples are long-term heat storage in an underground tank, adding new energy consumers into ice arena to improve comfort of players or prolonging the outdoor swimming pool use. The microenvironmental conditions in cloakrooms could be improved by introducing AHU system together with heating the air by refrigeration heat. Another option is to adapt the outdoor swimming pool as thermal during the winter. The water in a thermal pool usually has a temperature of up to 38 °C.

9. Conclusion

Traditional heat recovery systems focus on recovering heat from the high stage compressor discharge gas stream, desuperheater. The described case study focuses on the ammonia heat pump recovering condensing heat. The analysis confirmed that the system is profitable. With an increasing utilization ratio in neighborhood buildings the simple payback period is decreasing. The principal scheme can be used in any industrial process where there is a lot of waste heat that can be reused as a heat source for heat pumps. Moreover, the ammonia heat pump which uses natural refrigerants will contribute to reducing the environmental impact by the improvement of efficiency. Finally, the economics of recovering condensing heat in a small scale ice arena needs to be evaluated on a case-by-case basis.

Acknowledgements

The paper has been written with support from the SGS grant of the Czech Technical University in Prague.

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