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RUNAROUND COILS AS A WAY TO IMPROVE THE EFFICIENCY OF HEAT PUMPS FOR THE DEHUMIDIFICATION OF NORDIC GREENHOUSES

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***Abstract*—Greenhouses generate greenhouse gas from the combustion of fossil fuel for heating and dehumidification. Heat pumps can use electricity generated from renewable to do both, but the heat pump performance might be improved by optimizing the cycle for the specific requirements of a greenhouse. In this work, runaround coils are added to a single-speed R-134a heat pump. A Dymola heat pump example model is used with the addition of an air cycle and additional water coils. Results demonstrated that runaround coils are an efficient way to increase simultaneously and significantly the heating Coefficient of Performance, the condensate flowrate and the exergy efficiency of the system, by 8.5%, 16.3% and 13.1%, respectively, for interior conditions typical of Nordic greenhouses.

***Keywords-component*—Nordic greenhouse; heat pump; dehumidification; runaround coils;**

I. INTRODUCTION

In Nordic countries like Canada, greenhouses could provide freshly grown fruits and vegetables all year. In Quebec, 50% of greenhouses' vegetables consumed are produced in the province. The government seeks to increase the self-production level of the province up to 80% before 2025 [1].

Even if greenhouse gas (GHG) emissions due to transport are reduced by producing fruit and vegetables locally, greenhouses have a large environmental impact because of their intensive use of fossil fuel to provide the required energy for heating and dehumidification. In a meta-analysis of 742 food production system, Clark and Tilman [2] show that greenhouses emit between two and four times more GHG than open fields to grow the same amount of vegetables. They explain this difference by the large amount of energy required to heat the greenhouse. In another meta-analysis of 1731 database entries, Clune *et al.* [3] show that kilograms of carbon dioxide (CO_2) produces per kilogram of vegetable range from a mean value of 0.47 for field-grown vegetables to a mean value of 2.81 for vegetables produce in a heated greenhouse when considering life cycle emissions. In Canada, a life cycle analysis for greenhouses typical of Leamington in southern Ontario shows that fuel account for 70% of energy and between 50 and 85% of total impact for ozone depletion, global warming, smog, acidification and respiratory effect [4]. Finally, fuel represents 10.1% of all expenses for greenhouses in Canada [5].

Ventilation is currently the most common method to control the humidity into greenhouses [6]. In this system, outside air with a low humidity ratio is introduced into the greenhouse to reduce the humidity. However, the air from outside should be reheated, increasing the heating load of the greenhouse. In addition, ventilation dehumidification also cause loss of CO. Finally, ventilation could create cold spots inside, reducing crop yield.

An alternative to reduce significantly energy use and GHG emissions is to use heat pumps for heating, cooling and dehumidification. It is especially true in Quebec, where 99.7% of the electricity is produced from renewable energy sources [7]. Vadiee and Martin [8] compared conventional closed and semi-closed greenhouses including heat pumps using the TRNSYS software. They showed that the closed greenhouse concept could reduce significantly the heating load with a good payback period for a specific case in Sweden. They latter concluded that closed greenhouses with heat pumps is one of the most promising options for Nordic countries [9]. Recently, Yildiz [10] compared three types of conventional greenhouses using heat pumps across Canada: open, semi-closed and closed. They demonstrated that heat pumps can

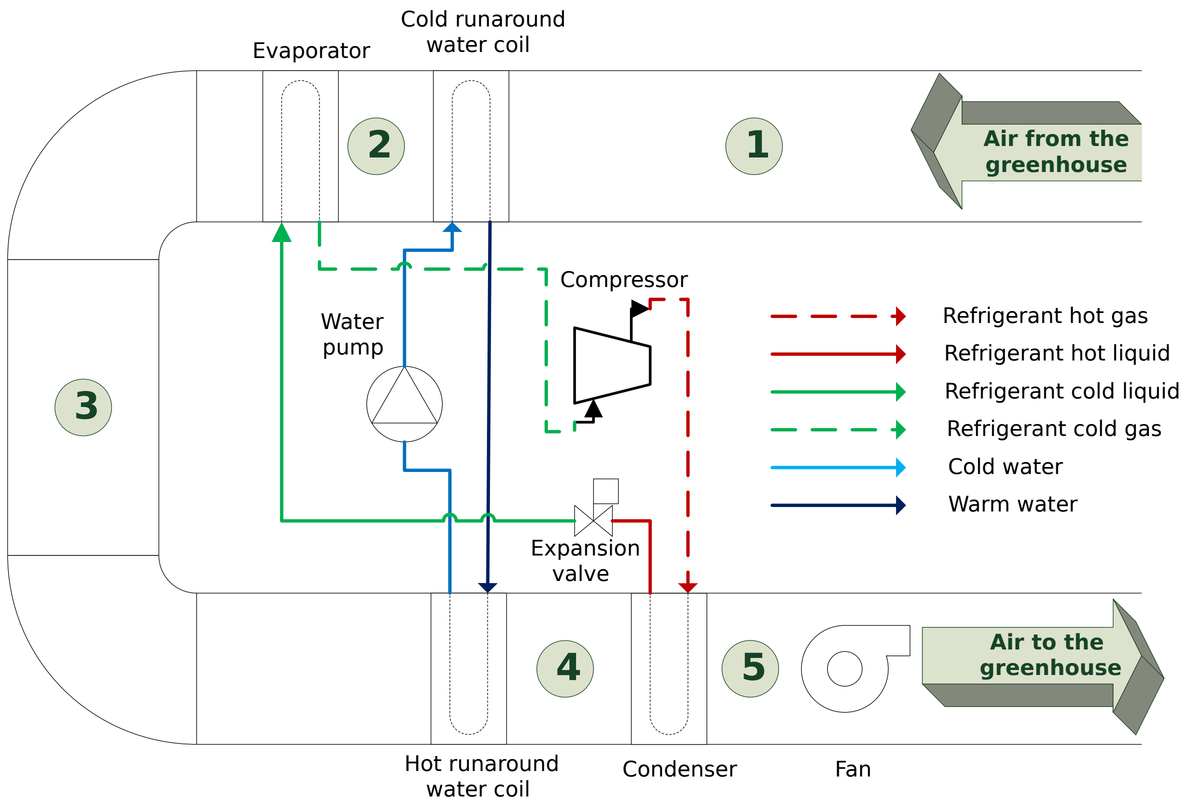

Figure. 1. Airflow in the modeled heat pump used for dehumidification. The heat pump is modeled with and without the two run-around coils.

reduce significantly the energy consumption in all scenarios. The closed greenhouse concept consumes less energy in colder months, but the semi-closed concept has the lowest annual energy consumption because of the lower cooling load. It confirmed the work of Van't Ooster *et al.* [11], stating that a semi-closed greenhouse combined with heat pumps is the most efficient concept for a zero-energy greenhouse in the Netherlands.

Some heat pumps are designed specifically for dehumidification. As shown in Figure 1, the heat pump cools air to reduce its water content. Then, the heat from the condenser is used to reheat air, providing both dehumidification and heating at the same time. Amani *et al.* [12] reviewed various dehumidification strategies in greenhouses and listed 15 works on the use of heat pumps in greenhouses. They observed that 50% of the energy supplied to the heat pump is for dehumidification. The other 50% is for cooling, which is useless most of the time. In Canada, Han [13] compared a commercial mechanical refrigeration dehumidification system with ventilation and air-to-air heat exchangers for a tomato greenhouse located in Saskatchewan. She found that mechanical refrigeration is the only efficient method to dehumidify at night during the mild season and in summer. In addition, the mechanical refrigeration system uses less total energy than the two other methods.

Even if these studies indicate that heat pumps are a promising alternative, few authors have tried to optimize the heat pump cycle the specific needs of a greenhouse, especially for air source heat pump. Campen and Bot [14] proposed a dehumidification system using natural convection to dehumidify a greenhouse using a water source heat pump. It includes a heat recovery device between the cold side and the hot side to increase the system efficiency. Yang and Rhee [15] proposed a heat pump system with a water heat storage tank to recover excess heat in the greenhouse during the day for a greenhouse located in Korea. They measured daily average COP between 2.32 and 3.55 from January to March. They suggested using this technique in a greenhouse with lower heating setpoints to maximize the economy generated by this system. Other authors used underground air from a volcanic site as a source for an air to water heat pump used for greenhouse heating [16].

In all these papers, authors used commercially available heat pumps with some external component. However, none of them tried to optimize the heat pump cycle for this specific application. In the present project, the objective is to optimize a heat pump for deshumidification purpose of a 0.74 hectare greenhouse at *Les Serres Royales* (see Figure 2) using the dynamic simulation software Dymola. Authors investigate if the addition of runaround coils on a heat pump could increase significantly its performance with indoor temperatures and relative humidity typical of a greenhouse for tomato production.

Figure. 2. Picture of the greenhouse at *Les Serres Royales*, located in Saint-Jérôme (QC), Canada.

II. Thermodynamic model

This section includes eight spans of 122 per 7.6 meters each (400 per 25 feets each) and is used to grow organic tomatoes.

Figure 3 presents the schematic of the reference heat pump using the modelling software Dymola (version 2021). The SimpleACCycle R134a example from the TIL Library is used as the reference heat pump. All components are unmodified in both the reference and the improved system. In addition, the air mass flowrate ($\dot{m}_a$=0.174 kg/s) is kept at the same level as in the example. The compressor run at a constant speed of 25 Hz. The compressor has a displacement of $150{\cdot}10^{-6}$ m^3 and isentropic and volumetric efficiencies of 0.7. As for the expansion valve, the effective flow area is fixed at $6.82{\cdot}10^{-6}$ m^2, maintaining an almost constant refrigerant mass flowrate (35.5±0.5 g/s).

Secondly, an air-moving system is added to model the air going from the greenhouse to the evaporator, to the condenser then back to the greenhouse. For this preliminary work, the greenhouse is not modelled. Instead, the fan is fed at 100% with air at constant properties at the inlet (21°C and 70% relative humidity (RH)). The fan moves the air through the system, but the power consumed is set to zero (ideal system without pressure drop).

In the improved system (see Fig. 1), a runaround heat recovery coil system is added. One coil is located before the evaporator and the other one is located between the evaporator and the condenser. In a dehumidification application, it is important to cool the air down to the desire dew point, but the air is often overcooled during the process. The coil located in front of the evaporator reduces the entering air temperature in the evaporator by transferring some heat to the air after the evaporator using water, which reduces the work done by the compressor. The water volumetric flowrate is set to 0.167 l/s. A perfect water loop without any pressure drop is implemented in this preliminary work. Consequently, the power consumed by the pump is zero in the Dymola model.

The simulation is run for five minutes with 3000 time steps. The simulation time has been chosen to reach a steady-state for the compressor discharge pressure. The simulation is initialized with arbitrary starting conditions.

The performance of each cycle is compared using a combined energy and exergy analysis. For the energy analysis, the work done by the compressor ($\dot{W}_c$) in kW is defined as:

$$\dot{W}_c = \dot{m}_r \left(h_{r,out} - h_{r,in} \right), \tag{1}$$

with $\dot{m}_r$ the refrigerant mass flowrate in kg/s and h_r the specific enthalpy of the refrigerant at the compressor's inlet (in) or outlet (out) in kJ/kg.

The work done by the pump (W_p) and the fan (W_f) in kW is not modeled in Dymola. However, both could be approximate for an ideal case using the Bernouilli equation, on the inlet and the outlet, which simplifies to:

$$\dot{W}_{p,f} = \dot{V}_{w,a} \Delta P, \tag{2}$$

with $\dot{V}_{w,a}$ the volumetric flowrate of water or air in m^3/s and ΔP the pressure drop in kPa in the assembly. In this case, a pressure drop of 60 kPa is assumed for each coil on the water side. On the air side, the assumed pressure drop is 0.2 kPa for each coil.

The sensible heating power $\dot{Q}_h$ (in kW) of the heat pump is calculated with the ambient air dry-bulb temperature in the greenhouse ($T_{amb} = T_1$) and the dry-bulb temperature at the outlet of the condenser (T_5) using:

$$\dot{Q}_h = \dot{m}_a C_p \left(T_5 - T_1 \right), \tag{3}$$

with C_p being the specific heat at constant pressure for air in kJ/(kg.K). C_p is considered constant at 1.005 in this case. The index for temperature is defined in Figure 1.

For the evaporator, the sensible cooling capacity $\dot{Q}_s$ can be calculated using Eq. 3 by using $T_3 - T_2$ as the temperature differential. As for the dehumidification capacity, it could be obtained by calculating how much kilogram of condensate per hour is removed by the heat pump ($\dot{V}_c$) as:

$$\dot{V}_c = \dot{m}_a \left(w_5 - w_1 \right), \tag{4}$$

with w the humidity ratio of the air (in kilograms of water per kilogram of dry air) at the outlet and at the inlet of the heat pump. The humidity ratio at the inlet is equal to the humidity ratio inside the greenhouse (w_{amb}). The latent heat capacity ($\dot{Q}_l$) in kW is defined as:

$$\dot{Q}_l = h_v \dot{V}_c, \tag{5}$$

with h_v=2500 kJ/kg the vaporization enthalpy of water.

Both latent cooling (L), heating capacities (h) and works could be combined to form coefficients of performance (COP) as:

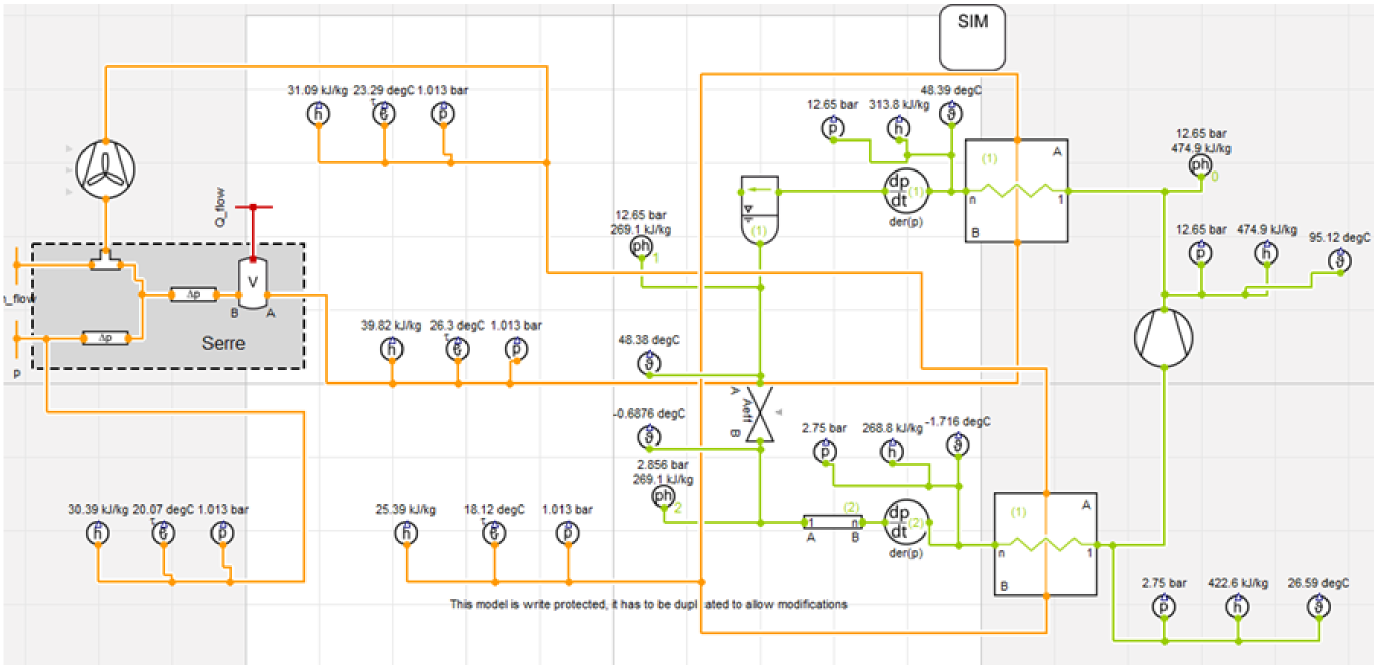

Figure. 3. Schematic of the basic cycle using Dymola.

$$\mathrm{COP}_h = \frac{\dot{Q}_h}{\dot{W}_c + \dot{W}_f + \dot{W}_p}, \tag{6}$$

$$\mathrm{COP}_L = \frac{\dot{Q}_l}{\dot{W}_c + \dot{W}_f + \dot{W}_p}. \tag{7}$$

Finally, the specific exergy e in kJ/kg at any point in the system could be calculated using the inlet as the reference state as:

$$e_5 = \left(h_5 - h_1\right) - T_{amb} \times \left(s_5 - s_1\right), \tag{8}$$

with h_{air} and s_{air} being the specific enthalpy in kJ/kg and the specific entropy of moist air in kJ/(kg.K) calculated using the Coolprop library [17] from the dry bulb temperature and the humidity ratio obtained by the Dymola model. Using this definition, the exergy at the inlet is zero. If a perfect motor is considered, the exergy efficiency η on the air side of the system could be calculated using:

$$\eta = \frac{e_5 \dot{m}_a}{\dot{W}_c + \dot{W}_f + \dot{W}_p}. \tag{9}$$

III. Results and Discussion

Table I presents the state of moist air defined by its dry-bulb temperature and humidity ratio and Table II displays the performance parameters obtained for both configurations. One could observe in Table I that the air is significantly colder and dryer after the evaporator (node 3) because of the precooling from the runaround coil. The condensation on the runaround coil is minimal (w from 10.8 to 10.4 grams of water per kilogram of air), but it brings the air near saturation.

In Table II, one could observe that the runaround coils increase the evaporator latent to sensible heat ratio from 0.81 to 1.13. It means that the heat pump spends a greater portion of its cooling power for dehumidification and less for sensible cooling, which is not useful in this application.

For the performance in heating, runaround coils increase both $\dot{Q}_h$r and COP$_h$. It is a modest improvement, but it is still significant when using the heat pump in heating mode. However, the most important gain in efficiency is the 16% increase in water removal rate with the improved cycle. As shown in Table II, The precooling of the air before the evaporator makes it possible to cool the air to a colder temperature after the evaporator. The amount of water removed from the air increases in the process. This modification could reduce the amount of heat pumps to be installed in a greenhouse for dehumidification purposes.

For the performance in heating, runaround coils increase both $\dot{Q}_h$r and COP$_h$ because T_5 is higher. It is a modest improvement, but it is still significant when the greenhouse needs simultaneous dehumidification and heating. However, the most important gain in efficiency is the 16% increase in water removal rate with the improved cycle. As shown in Table I, the precooling of the air before the evaporator makes it possible to cool the air to a colder temperature after the evaporator. The amount of water removed from the air increases in the process. This modification could reduce the amount of heat pumps to be installed in a greenhouse for dehumidification purposes and compensate for the increase in cost of each heat pump.

Concerning the exergy efficiency, it takes into consideration both the heating capacity and the removal of humidity. η is

TABLE. I

State of moist air at each node of the system. See Fig. 1 for nodes' location.

Node	1		2		3		4		5	
Properties	T (°C)	w (g/kg)	T (°C)	w (g/kg)	T (°C)	w (g/kg)	T (°C)	w (g/kg)	T (°C)	w (g/kg)
Reference	21.0	10.8	–	–	7.5	6.3	–	–	43.0	6.3
Improved	21.0	10.8	16.7	10.4	5.4	5.2	10.6	5.2	44.8	5.2

TABLE. II

Performance parameters for both configurations.

Parameter	Unit	Reference	Improved	Variation
$\dot{W}_c$	kW	1.87	1.87	0.0%
$\dot{W}_f$	kW	0.06	0.12	+50.0%
$\dot{W}_p$	kW	–	0.02	−−
$\dot{Q}_h$	kW	3.84	4.34	+13.0%
$\dot{V}_c$	$kg_w \cdot h^{-1}$	2.76	3.21	+16.3%
$\dot{Q}_s$	kW	2.36	1.98	−16.1%
$\dot{Q}_L$	kW	1.92	2.23	+16.1%
COP_h	–	1.99	2.16	+8.5%
COP_l	–	0.99	1.11	+12.1%
η	%	10.7	12.1	+13.1%

TABLE. III

State of refrigerant at different steps in the cycle.

Step	Suction		Discharge		Liquid
Variables	P (bar)	h (kJ/kg)	P (bar)	h (kJ/kg)	h (kJ/kg)
Reference	2.85	405	15.5	458	282
Improved	2.78	398	16.1	451	285

fairly low with this type of heat pump because some exergy is lost by heating the cold air after the evaporator to room temperature. Runaround coils recover part of this exergy. Consequently, these additional coils increase η by 13%, which is in between the percentage improvement of COP_l and of COP_h.

Finally, Table III presents the refrigerant state at the inlet of the compresor (suction), at the outlet (discharge) and between the evaporator and the condenser (liquid). One could observe that suction and discharge pressures change between the two options, even if $\dot{W}_c$ does not change (see Tab. II). The compressor runs at a constant speed and the total load at the evaporator ($Q_s + Q_l$) remains almost the same to fully evaporate the liquid. To achieve that with a lower air temperature, the suction pressure must be reduced. As for the discharge pressure, it needs to increase too to reject heat at a higher air inlet temperature at the condenser.

IV. Conclusion

This paper presented a way to improve the performance of a single-speed R-134a heat pump used for dehumidification in a Nordic greenhouse by the addition of runaround coils. Dymola was used to model the heat pump and the additional water coils. The model demonstrated that runaround coils could increase significantly the performance of this heat pump. As examples, the heating COP, the condensate flowrate and the exergy efficiency of the system were increased by 8.5%, 16.3% and 13.1%, respectively.

However, there are other options that could be considered to increase the heat pump efficiency for simultaneous heating and dehumidification. In future works, the runaround coil could be optimized by changing the air and the water mass flow rate. In addition, some alternatives to runaround coils will be considered: heat wheel, heat pipe or plate air-to-air heat exchangers. In addition, no optimization is done in the current work on the refrigeration cycle itself. Using runaround coils may reduce the optimal size of all components in the refrigeration cycle. A parametric study could provide the optimal heat pump for this application. In addition, variable speed and two-stage compressors could be implemented in the model to obtain additional gain in efficiency. Finally, the heat pump model could be coupled with a greenhouse model that calculates indoor conditions and plants' transpiration for a typical meteorological year. By coupling these two models, it will be possible to estimate the yearly energy consumption, which should include a lot of hours at part-load conditions.

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