

C09

Performance Evaluation of Two-Stage Air-Source Heat Pump for Cold Climates considering the effects of Frosting

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Abstract: Air-source heat pumps (ASHPs) are energy-efficient alternatives to conventional natural gas heating systems. At very low ambient temperatures, the coefficient of performance (COP) of an ASHP drops drastically with rising refrigerant discharge temperatures at the compressor exit. Two-stage ASHPs have a staged compression process, which improves the COP and reduces discharge temperatures. The current study investigates the performance of a two-stage ASHP used to supply heat to an office building in Saskatoon, considering the effects of frosting on the evaporator. The average annual COP of two-stage ASHP is 2.04, while frosting and defrosting cycles increase energy consumption by 12.5%. The GHG emissions from two-stage ASHP operation are compared with natural gas heating systems indicating that two-stage ASHPs can reduce emission by 8% compared with natural gas heating systems in Saskatoon.

Abbreviations/ Symbols

<i>ASHP</i>	<i>Air-source heat pump</i>
<i>c</i>	<i>Specific heat (kJ/kg.K, BTU/lb°F)</i>
<i>COP</i>	<i>Coefficient of performance</i>
<i>h</i>	<i>Specific enthalpy (kJ/kg, BTU/lb)</i>
<i>m</i>	<i>Mass (kg,lb)</i>
<i>P</i>	<i>Pressure (kPa, psi)</i>
<i>Q</i>	<i>Heat transferred (W, BTU/hr)</i>
<i>W</i>	<i>Work consumed (W, BTU/hr)</i>

Subscripts

<i>a</i>	<i>Air</i>
<i>cond</i>	<i>condenser</i>
<i>ev</i>	<i>evaporator</i>

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i	<i>Intermediate stage</i>
r	<i>refrigerant</i>
Greek letters	
ε	<i>Effectiveness of the heat exchanger</i>
ϵ	<i>Sub-cooling parameter of the economizer</i>
u	<i>Refrigerant mass fraction</i>

INTRODUCTION

Building heating systems are responsible for 12.5% of GHG emissions worldwide (Krishnan et al. 2021; IEA 2019). Electrification of building heat is a practical approach to mitigate emissions from buildings and comply with the Paris Climate Accord (“The Paris Agreement”). Air-source heat pumps (ASHPs) are the electric alternative to fossil fuel-based systems and have proven to be effective in mitigating emissions and reducing energy consumption (Valancius et al. 2019; Staffell et al. 2012).

The cold climate operation of ASHPs (ambient temperatures below -10°C (14°F)) is a cause of concern. The two main problems that affect the ASHP operation are frosting and high discharge temperatures (Liu et al. 2017). Frosting occurs on the evaporator when the moisture in the air condenses and freezes. Frosting increases heat transfer resistance, deteriorating performance. As the ambient air temperature falls, the temperature difference between the evaporator and the condenser increases, increasing the compression ratios and leading to high discharge temperatures. The discharge temperatures of an ASHP must be maintained below 115°C (239°F) for safe operation (Luo et al. 2022). If the discharge temperatures go beyond the limit, the compressor experience lubrication issues causing eventual failure.

Techniques like cascade cycles (Arpagaus et al. 2016), two-stage cycles with economizer (Yan, Jia, and Bai 2016; Jiang et al. 2015), intercoolers (Arpagaus et al. 2016), thermal energy storage (Gill 2021; Alkhwildi, Elhashmi, and Chiasson 2020) etc., are studied in the literature to limit discharge temperature and improve ASHPs in cold climates. From all the techniques mentioned above, two-stage ASHP with economizer is easy to implement and showcases relatively high performance in cold climates. Furthermore, it is easy to switch from heating to cooling mode in a two-stage ASHP with an economizer than a cascade cycle which requires an extra heat exchanger (Bertsch and Groll 2008).

Very few studies discuss the effects of frosting on the performance of two-stage ASHP on an annual basis. The frosting is predominant on ASHP when operating from -15°C (5°F) to 6°C (42.8°F) (Y. Wu et al. 2021; Shao et al. 2021). The influence of frosting on performance starts to subside as the temperature drops below -15°C (5°F), and the amount of moisture in the air is very low at such temperatures (Shao et al. 2021; Zhu et al. 2015). As an ASHP will operate for a significant amount of time in frosting conditions in cold climates like Saskatoon, where the temperature is below 0°C (32°F) for 37% of the time, it becomes imperative to consider the effects of frosting while evaluating annual performance and choosing retrofit strategies. In general, frost formed on the evaporator is removed by the reverse cycle defrost method. In this method, the ASHP is switched from heating mode to cooling mode, which transfers heat from the indoor space to melt the frost formed on the evaporator.

The present study investigates the annual performance of a two-stage ASHP with an economizer to supply heat to a single-story office building in Saskatoon. The energy consumption and the GHG emissions from the yearly operation are compared with that of a natural gas heating system to estimate the energy and emission reduction capacity of a two-stage ASHP with an economizer.

METHODOLOGY

The working of a two-stage ASHP is demonstrated in Figure 1. The refrigerant exiting the condenser (point 6) is branched into two refrigerant streams (the evaporator stream and the intermediate stream). The intermediate refrigerant stream is throttled to an intermediate pressure, after which it extracts heat from the evaporator stream in the economizer. The evaporator stream exiting the first stage of compression (point 3) is cooled by the intermediate refrigerant stream, after which the refrigerant enters the final stage of compression (point 4). Cooling the refrigerant at the intermediate stage decreases discharge temperature compared to single-stage ASHPs.

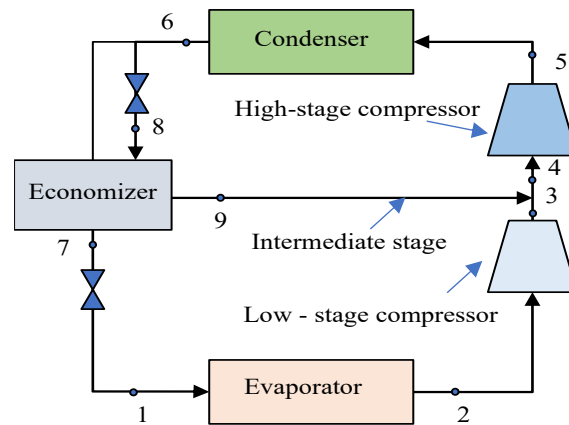


Figure 1: The schematic of a two-stage ASHP with an economizer

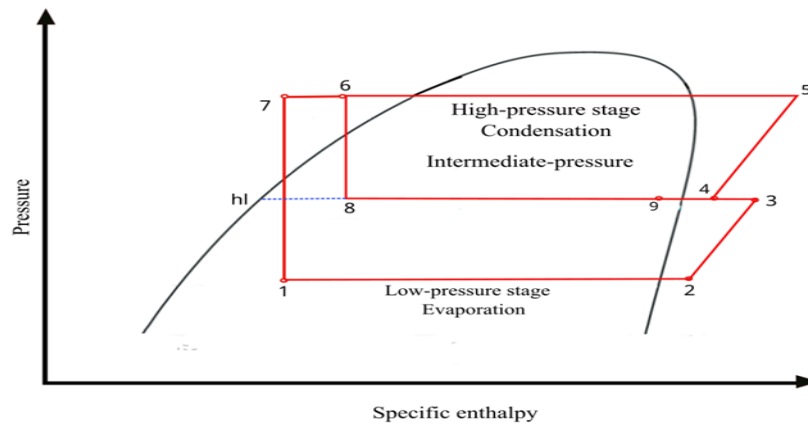


Figure 2: Thermodynamic cycle of a two-stage ASHP with an economizer

As shown in Figure 2, the thermodynamic cycle is mathematically modelled in MATLAB using coolprop to determine refrigerant properties. The assumptions of the model are (Zehnder 2004; Torrella, Llopis, and Cabello 2009; Bertsch and Groll 2008; Li and Yu 2016):

1. Heat losses and pressure drop in the refrigerant flow lines are neglected.
2. The refrigerant exits the evaporator with a superheat of 5K (9°F) and leaves the condenser with a subcooling of 3K (5.4°F).
3. The effectiveness of the heat exchangers is considered to be 75%.

At temperatures above 0°C (32°F) the compression ratios for each stage become very small, and two-stage ASHP is switched to single-stage operation.

The energy and mass balance equations used to model the parts of the two-stage ASHP are illustrated in Equations (1-14):

Evaporator

$$\dot{Q}_{ev} = \dot{m}_{ev}(h_{ev_{out}} - h_{ev_{in}}) \quad (1)$$

$$\dot{Q}_{ev_{max}} = \dot{m}_a c_a (T_a - T_{ev}) \quad (2)$$

$$\dot{Q}_{ev} = \varepsilon \dot{Q}_{ev_{max}} \quad (3)$$

Condenser

$$\dot{Q}_{con} = \dot{m}_r (h_{con_{out}} - h_{con_{in}}) \quad (4)$$

Economizer

$$\dot{m}_r = \dot{m}_{ev} + \dot{m}_i \quad (5)$$

$$\frac{\dot{m}_{ev}}{\dot{m}_r} = u \quad (6)$$

$$u(h_6 - h_7) = (1 - u)(h_9 - h_6) \quad (7)$$

The enthalpy at point 7 is dictated by the subcooling parameter Eqn. 8. where h_l is the saturation enthalpy of liquid at an intermediate pressure. The subcooling parameter can be approximated to the effectiveness of the heat exchanger, which is 75% in the current study.

$$\varepsilon = \frac{h_6 - h_7}{h_6 - h_l} \quad (8)$$

Refrigerant mixing

$$\dot{m}_r h_4 = u \dot{m}_r h_3 + (1 - u) \dot{m}_r h_9 \quad (9)$$

Compressor

$$W_{stage\ 1} = \frac{\dot{m}_{ev}(h_3 - h_2)}{\eta_{isentropic\ stage\ 1}} \quad (10)$$

$$W_{stage\ 2} = \frac{\dot{m}_r(h_5 - h_4)}{\eta_{isentropic\ stage\ 2}} \quad (11)$$

$$\eta_{isentropic\ stage\ 1} = f(P_i, P_{ev}) \quad (12)$$

$$\eta_{isentropic\ stage\ 2} = f(P_{cond}, P_i) \quad (13)$$

$$W_{total} = \frac{W_{stage 2} + W_{stage 1}}{\eta_{system}} + W_{fan} \quad (14)$$

The intermediate pressure is the square root of the product of condenser and evaporator pressures (Nag 2002). Electrical and mechanical efficiencies account for η_{system} and is set as 0.8. The isentropic efficiency of the compressor is a function of the condenser and evaporator temperature. In the current study, the condenser temperature is 50°C (122°F), and the working fluid is R-410A.

FROST MODELLING

Accounting for the effects of frosting while performing annual energy simulations becomes vital as the ASHPs operate for a significant amount of time in frosting conditions. The frosting phenomenon occurs when moisture in the ambient freezes on the evaporator, causing a drop in performance. The effects of frosting become less significant at temperatures below -15°C (5°F) because of the decrease in the moisture-carrying capacity of air at lower temperatures (Shao et al. 2021; Zhu et al. 2015; C. Wu et al. 2022). Furthermore, Shao et al. (Shao et al. 2021) have used loss coefficients based on the severity of frosting as mapped by Zhu et al. (Zhu et al. 2015), as shown in Figure 3. The frosting zones 'A' to 'E' are classified based on the temperature and humidity, which dictate the severity of frosting. 'A' is the most severe zone, 'E' being the least (Shao et al. 2021). The severity decreases as the ambient air humidity decreases. Therefore, the energy loss is lower in region E (less severe frost region) compared to region A (the most severe frost region). The paper does not include detailed modelling of the defrost method. However, to account for the energy loss to defrost, COP correction factors are used depending on the severity of frost as mentioned in Table 1 (Shao et al. 2021).

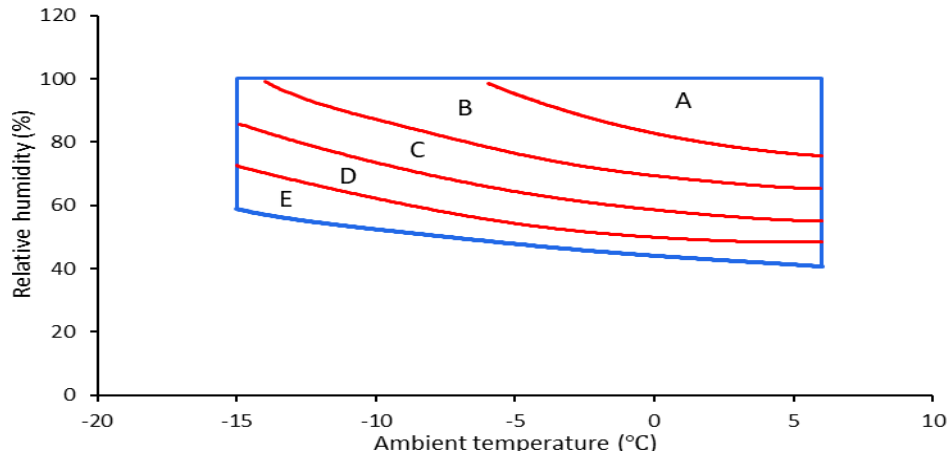


Figure 3: Frosting regions categorized based on the severity of frosting (adapted from (Zhu et al. 2015))

Table 1: Frosting correction factors (C_f) for different zones presented by (Zhu et al. 2015; Shao et al. 2021)

Frosting region	C_f value
E	0.9
D	0.86
C	0.8
B	0.74
A	0.68

$$COP_{frosting} = COP_{no\ frosting} * C_f \quad (15)$$

A two-stage ASHP with economizer is modelled according to the methodology described above to meet the building heating loads of an office building of 510 m² (5500 ft²) area. The building geometry and envelope are adopted from (Torcellini

et al. 2008; Eldeeb, 2012). The annual heating demand of the building is found to be 70,150 kWh (239.36 MMBTU) with a peak heating demand of 64 kW (218,377 BTU/hr) and is determined by using TRNSYS. The thermodynamic model uses the loads generated by TRNSYS to evaluate the performance of an ASHP. The cooling loads are not considered in the present study as the focus is on heating in cold climates.

The GHG emission intensity of the electricity grid in Saskatoon is 530 g/kWh (342.4 lb/MMBTU) of electricity produced and distributed compared to 240 g/kWh (155 lb/MMBTU) from natural gas (City of Regina 2020; Asaee, Ugursal, and Beausoleil-Morrison 2017). These values are used to calculate and compare the annual GHG emissions from two-stage ASHP and natural gas heating systems.

RESULTS AND DISCUSSION

The current section discusses the performance of a two-stage ASHP modelled to operate in Saskatoon as described in the above paragraphs. Figure 4 depicts the variation in COP of a two-stage ASHP with ambient air temperature. As observed in several studies, the COP increases with ambient air temperature. The improved performance is because the compressor work required decreases at higher ambient temperatures while the compressor's efficiency increases. However, a drop in COP is observed at -15°C (5°F) because of the effects of frosting. Figure 4 also depicts the COP trend for both frosting and a hypothetical no-frosting condition, this comparison is done to understand the negative impact of frosting. The COP would only differ between -15°C (5°F) to 6°C (42.8°F) for frosting and no-frost conditions as the frosting is only observed in this region. In Saskatoon, according to the frosting model described above, the ASHP operates in frosting conditions 67% of the time, increasing energy consumption by 12.4% indicates that frost suppression and removal techniques are essential for ASHPs working in colder climates. Table 2 depicts the energy consumption of Two-stage ASHP during frost, no-frost, and natural gas heating systems.

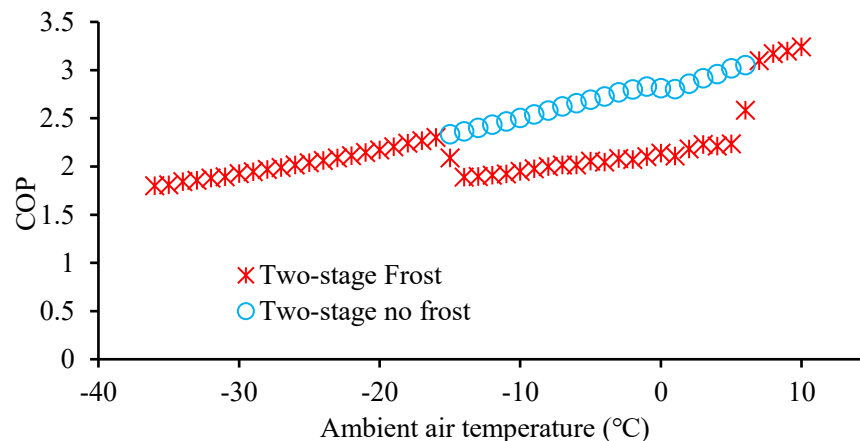


Figure 4: COP variation of two-stage ASHP (frost and no-frost conditions) for different ambient temperatures ($^{\circ}\text{C}$)

Figure 5 demonstrates the annual GHG emissions for the heating scenarios mentioned in Table 2. Using a two-stage ASHP, 1.61 tonnes a year in GHG emission can be saved compared to natural gas heating systems, which is about 3.15 kg/m^2 (0.64 lb/ft^2), saving in GHG emissions. The effects of frosting increase GHG emissions by 12.4 % compared to no frost case. Figure 4 also demonstrates the project emissions from different heating scenarios for the expected GHG emissions in 2035. The GHG intensity of electricity is expected to drop to 221 g/kWh (142.7 lb/MMBTU) in 2035 (SaskPOWER, n.d.). The emissions from two-stage ASHP will be lower by 61.6% in 2035 compared to the natural gas heating system. This indicates that two-stage ASHP has excellent potential to decarbonize HVAC systems.

Table 2: Annual energy consumption with different heating scenarios to meet the annual heating demand of

Heating system	COP	Energy consumed in kWh (MMBTU)
Two-stage ASHP without frosting	2.3	30,541 (104)
Two-stage ASHP with frosting	2.04	34,352 (117)

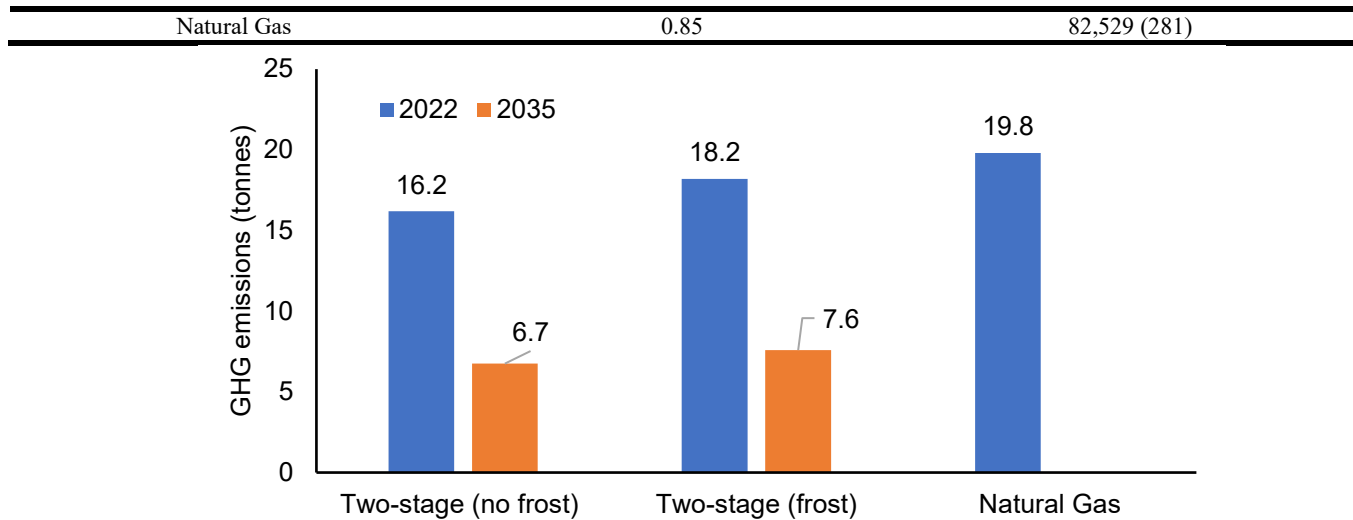


Figure 5: GHG emissions two two-stage ASHP with frosting and hypothetical no-frost scenario compared with natural gas heating

CONCLUSIONS

Two-stage air-source heat pumps can operate in extremely cold climates experienced in Saskatoon with an average annual COP of 2.04. The frosting has a crucial effect on the performance and the GHG emissions of TW-ASHP. Frosting increases energy consumption and GHG emissions by 12.4% compared to a no-frost case. Despite the Saskatoon electricity grid's relatively high GHG emission intensity, two-stage ASHPs effectively minimize GHG emissions. As the electricity grid gets cleaner in the coming future, the emissions from two-stage ASHP will further reduce. The economics of two-stage ASHP needs to be compared with that of single-stage and natural gas heating systems to determine the true efficacy of two-stage ASHPs. Furthermore, measures to reduce or eliminate frosting need to be investigated, given that two-stage ASHP operates 67% of the time in frosting conditions.

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