Accepted Manuscript

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PII: S0140-7007(17)30482-6
DOI: 10.1016/j.ijrefrig.2017.11.028
Reference: JIJR 3835

To appear in: *International Journal of Refrigeration*

Received date: 27 September 2017
Revised date: 21 November 2017
Accepted date: 25 November 2017

Please cite this article as: J.F. Wang , C. Brown , D.J. Cleland , HEAT PUMP HEAT RECOVERY OPTIONS FOR FOOD INDUSTRY DRYERS, *International Journal of Refrigeration* (2017), doi: 10.1016/j.ijrefrig.2017.11.028

Highlights

- air-to-air heat exchange (AHX) likely to be economic but limited to less than 20% of air heating
- AHX plus heat pump (HP) could provide up to 40% of air heating with 20% lower energy cost
- refrigerants with critical temperatures between 70-110^{o}C & evaporating 25-30^{o}C are most promising
- HP options attractive if capital costs are less than NZ\$300 kW^{-1} of heating

HEAT PUMP HEAT RECOVERY OPTIONS FOR FOOD INDUSTRY DRYERS

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ABSTRACT

Air heating by heat recovery using air-to-air heat exchangers (AHX) and heat pumps (HP) was analysed for a typical spray dryer with inlet ambient air heated to 200°C and exhaust air with dry bulb of 76°C and dewpoint of 38.5°C. The HP design is a tradeoff between greater heat recovery and lower COP as evaporation temperature decreases. With an evaporation temperature of 25-30°C, it was possible to provide up to about 40% of the air heating load with more than 20% lower energy cost. The transcritical cycle, with dehumidification of exhaust air at a constant temperature and inlet air heating in the supercritical region, is thermodynamically well-matched to the drying process. The ideal critical temperature is between 70°C and 110°C, so a trade-off between efficiency, cost and safety may be needed with respect to refrigerant choice. Refrigerants R134a looks most promising while R32 and R290 also have good energy efficiency but are flammable. Economically for the typical spray dryer in New Zealand, only use of an AHX is attractive but is limited to about 20% of the air heating. Combining with an R134a HP to get higher heat recovery than an AHX alone would only be attractive if HP capital costs could be reduced significantly to less than about NZ$300 kW^{-1} heating capacity.

Keywords: heat recovery, dryers, air-to-air heat exchangers, heat pumps, technical and economic feasibility

1. INTRODUCTION

Dryers commonly used in the food industry are characterized by quite high inlet air temperatures, no air recycle and minimal heat recovery so the thermal efficiency is often less than 50% (Baker, 1997). This is particularly true for spray dryers where typically inlet ambient air is heated to 200°C or higher by a steam heat exchanger. The exhaust air is often at 60-80°C with dew-point temperatures of 35-40°C (Hahveci & Cihan, 2007).

The simplest form of dryer heat recovery is direct partial recycling of exhaust air but has the disadvantage of slower drying rates due to the higher humidity of the mixed dryer air (Baker, 1997). Conventional heat recovery is achieved by preheating dryer inlet air using the dryer exhaust air via a heat exchanger (Colak & Hepbasli, 2009a, 2009b). Such air-to-air heat exchangers (AHXs) principally recover sensible heat, are constrained by temperature differences between the exhaust and inlet air, and in some cases the distance between the inlet and exhaust air stream. Use of an intermediate fluid in a run-around loop can avoid co-locating the air streams, but incurs fluid pumping costs, requires an extra heat exchanger, and the approach of the inlet air to the exhaust air temperature tends to be wider due to the extra heat exchange step. To be economic, the reduction in energy use must justify the extra capital cost of the heat exchanger and any other components. For a typical milk powder spray dryer in New Zealand, Walmsley *et al.* (2015) showed that exhaust air heat recovery using a liquid run-around loop could have internal rates of return up to 71% but was limited to about 15% of the total air heating.

Heat pumps (HPs) can recover latent heat from moisture condensation from the exhaust air, in addition to sensible heat, if the evaporation temperature is low enough, and can operate against a temperature gradient so higher amounts of inlet air preheating are possible (Perera & Rahman, 1997). If mechanical vapour compression HPs are used, they are generally driven by electricity so to be economically feasible the Coefficient of Performance (*COP*) must be significantly greater than the electricity to fuel cost ratio (typically about 3:1 in many countries). Use of AHXs is generally simpler and less costly than HPs, so they are often the first heat recovery option with HPs being used in series. Wang *et al.* (2010) showed that combining AHXs and HP systems would give greater heat recovery and higher overall energy efficiency than heat recovery by AHXs or HPs alone. While closed circuit HP dehumidifier dryers are available for drying at lower temperatures (e.g.

Carrington, 2007), they are not yet available for the higher temperatures currently the norm for food industry spray drying.

This paper evaluates heat recovery using AHX and HPs in a typical food spray dryer in New Zealand in terms of energy efficiency, and the technical and economic feasibility.

2. HEAT RECOVERY ANALYSIS

2.1 Heat Recovery Configuration

The combined AHX-HP configuration is shown in Figure 1. The inlet air is initially heated in a countercurrent AHX by the cooling exhaust air. The HP evaporator then further cools and dehumidifies the exhaust air and the HP condenser/gas cooler further heats the inlet air. The HP is a simple single stage mechanical vapour recompression system without an internal heat exchanger (recuperator). The HP has a sub-cooler in which ambient air sub-cools the refrigerant.

2.2 Heat Pump (HP) Characteristics

Calculations of heat recovery system performance, including determination of the refrigerant thermodynamic properties, were performed in the Engineering Equation Solver (EES) software (Klein, 2009). The following assumptions were made for the HP system: only sensible heat transfer in the AHX; negligible heat losses from the HP circuit; negligible refrigerant pressure drop in pipelines and heat exchangers; constant refrigerant superheat of 5°C at the evaporator exit; liquid sub-cooling as defined by the minimum temperature approach in the ambient sub-cooler; negligible extra fan power for the evaporator, sub-cooler, condenser or gas cooler than for the normal air heating heat exchanger; negligible compressor cooling, compressor motor efficiency of 90%; and the following typical relationship between compressor isentropic efficiency (η_{is}) and pressure ratio (PR):

$$\eta_{is} = 0.65 + 0.015PR - 0.0015PR^2 \qquad (1)$$

For the HP, the possibility of some latent heat recovery due to dehumidification of the exhaust air, rather than just sensible cooling, is an important potential advantage relative to an AHX. Therefore the evaporator performance including dehumidification was estimated assuming a straight line approach of the inlet air condition to saturated air at the HP evaporation temperature as shown in Figure 2 (Stoecker and Jones, 1982). The following equation gives the humidity of the outlet exhaust air (H_{ae2}) knowing the exhaust air outlet temperature (T_{ae2}) from the specified approach to the refrigerant evaporation temperature (T_{eva}):

$$H_{ae2} = H_{s\,eva} + \frac{(T_{ae2} - T_{eva})}{(T_{aem} - T_{eva})}(H_{aem} - H_{s\,eva}) \qquad (2)$$

The change in enthalpy of the exhaust air was evaluated using the psychrometric relationships in EES. For a fixed HP evaporation temperature, T_{eva}, the HP discharge pressure and the HP capacity (mass flowrate of refrigerant) were iteratively adjusted to allow both the condenser/gas cooler and ambient sub-cooler minimum temperature approach (TA) criteria to be met, and to balance the HP heat of rejection with the air preheating as shown in Figures 3a and 3b. For example, a lower T_{eva} meant there was higher heat removal from the exhaust air so the discharge pressure and refrigerant flowrate both had to be higher to allow all the heat of rejection to heat inlet air whilst maintaining the desired temperature approach in the air heater.

The HP cycle was calculated using standard vapour compression cycle and heat exchanger mass and energy balance equations (e.g. Love & Cleland, 2007). Outputs from the model included the HP cycle conditions (pressure and temperature), compressor size and power use, heat flows for each of the heat exchangers and inlet and exhaust air temperatures and humidities. Most refrigerants available in EES were assessed. Both subcritical and transcritical HP cycles resulted, depending on the refrigerant and the compressor discharge conditions necessary for an energy balance.

2.3 Minimum Temperature Approach (TA)

Optimal heat exchanger design involves a complicated tradeoff between capital costs of the heat exchanger network and on-going operating costs including maintenance and energy to run the system. Minimum temperature approach (*TA*) is often used as a de facto method to approximate this optimization. If *TA* is lowered then capital costs increase but energy costs tend to decrease and vice versa. The optimum *TA* generally depends on the type of heat transfer fluids and other factors such as operating temperature and pressure.

Figures 3(a) to (d) show the temperature distributions in the HP condenser (or gas cooler), sub-cooler and evaporator, and in the AHX. For sub-critical HP systems, the condenser heat transfer performance was analysed by dividing the condenser into de-superheating, condensing and sub-cooling sections as shown in Figure 3a. For transcritical HP systems, the gas-cooler was broken into 11 sections with equal heat flow in order to determine the position where minimum *TA* occurs. The minimum *TA* is just as likely to occur internally in the condenser as shown in Figure 3a or gas cooler as shown in Figure 3b, as at the ends of the heat exchanger as for the evaporator and AHX shown in Figures 3c and 3d. The following three *TA*s were defined, reflecting high pressure refrigerant-to-air, air-to-low pressure refrigerant and air-to-air heat exchange pinch points respectively:

$$TA_1 = T_3 - T_{ai@3} = T_5 - T_{aim} = T_6 - T_{amb} \tag{3}$$

$$TA_2 = T_{ae2} - T_{eva} \tag{4}$$

$$TA_3 = T_{aem} - T_{ai1} \tag{5}$$

The combination of the heat and mass flows for each heat exchanger and the temperature approach defines the sensible heat transfer rating, *UA* (W/K), and other characteristics of the heat exchangers. In particular:

$$UA = \frac{\phi}{\Delta T_{lm}} \tag{6}$$

2.4 Reference Conditions

The assessment of heat recovery options was performed for a typically 10 tonnes per hour spray dryer in the New Zealand dairy industry with the following characteristics: ambient temperature (T_{amb}) of 14°C; air heater inlet temperature (T_{ai1}) of 28°C (assuming that the air is passively preheated by 14°C on average by being drawn through the dryer enclosure) and 42% *RH* (humidity of 0.01 kg/kg, dewpoint of 14.0°C); dryer inlet air temperature (T_d) of 200°C; exhaust air temperature (T_{ae1}) of 76°C and 17% *RH* (humidity of 0.045 kg/kg, dewpoint of 38.5°C); and equal mass flow rates (dry air basis) for the inlet and exhaust air ($m_{ai} = m_{ae}$) of 54.7 kg dry air s^{-1}. From a product drying perspective, these conditions represent a dryer with large ambient heat losses or significant sensible heating of the inlet liquid product. For these exhaust air conditions, if heat losses or sensible heating were lower, then the dryer inlet temperature would normally be less than 200°C. To estimate the energy operating costs relative to traditional steam air heaters, it was assumed that the boiler efficiency was 80%, and that the ratio of electricity to boiler fuel costs was 3:1.

Calculations were made for a reference heat recovery system with TA_1 = 10°C, TA_2 = 10°C, TA_3 = 20°C and T_{eva} = 30°C for a HP only configuration ($T_{aim} = T_{ai1}$ and $T_{aem} = T_{ae1}$) to screen a wide range of refrigerants for those best suited to the drying application. For the most promising refrigerants, calculations were made for either HP alone or a combined AHX-HP system across a practical range of *TA* and HP evaporation temperatures (T_{eva}) values. The higher value of TA_3 for the AHX, than for the other heat exchangers, reflects the likely need to use an indirect AHX with a liquid run-around loop rather than a direct AHX (Walmsley *et al*., 2015).

2.5 Performance Metrics

The performance of the heat recovery system was quantified in terms of the heat recovery ratio (*HRR*) being the fraction of air preheating that could be achieved, the HP and overall heat recovery system COPs (COP_{HP} and COP_t) and, *RCost*, the fractional energy costs relative to a dryer without heat recovery (a fraction *HRR* of the air heating by the HP and a fraction 1-*HRR* by steam heating relative to 100% of air heating by steam only):

$$HRR = \frac{\text{heat recovered into inlet air}}{\text{total load to heat air from 28 to 200}^\circ\text{C}} \tag{7}$$

$$COP_{HP} = \frac{\text{heat recovered into inlet air by the heat pump}}{\text{heat pump electrical power input}} \tag{8}$$

$$COP_t = \frac{\text{heat recovered into inlet air}}{\text{heat pump electrical power input}} \tag{9}$$

$$RCost = \frac{HRR \times \text{electricity cost} \times \eta_b}{COP_t \times \text{fuel cost}} + (1 - HRR) \tag{10}$$

3. ECONOMIC ANALYSIS

The economic analysis considered retrofitting of a heat recovery system to an existing spray dryer facility so the only changes to the existing steam heating heat exchanger was reduced heating duty and steam use.

3.1 Refrigerant Choice

The economic analysis focussed on HP systems using R134a as the refrigerant. The rationale included:

1. R134a is a common refrigerant so there are plentiful equipment options and significant design and operating experience for R134a systems;
2. R134a is a relatively low pressure refrigerant so high pressure compressor and heat exchange equipment will be less likely;
3. R134a had one of the higher energy efficiencies of the refrigerants considered in the heat recovery analysis;
4. R134a is relatively low cost and is non-flammable and non-toxic;
5. When R134a is compressed, high superheats do not occur thereby simplifying lubricant selection and/or potentially avoiding a need for compressor cooling;
6. While R134a has relatively high GWP, there are low GWP alternatives entering the market that have similar performance (e.g. HFO-1234yf) so future conversion to these is a distinct possibility if GWP reduction is important;
7. Many of the other refrigerants with high energy efficiency are flammable and/or toxic (e.g. R32, R290 and R717).

3.2 Methodology

The analysis methodology was broadly similar to that used by Krokida and Bisharat (2004) but the dryer inlet temperature was higher (200^{o}C rather than 150^{o}C), the costing information was updated, and the HP model was much more realistic as described in Section 2.2 (i.e. real compressor efficiencies rather than Carnot efficiency, detailed analysis of temperature profiles in the condenser/gas cooler, inclusion of exhaust air dehumidification in the evaporator model). All costs were indexed to NZ$ in 2011.

The main equipment items are the AHX, HP condenser/gas cooler, HP evaporator, and HP compressor and motor. Capital costs were estimated using the methods outlined by Peters *et al*. (2003) and Ulrich & Vasudevan (2004). Installed capital costs for the HP and AHX where assumed to be 415% and 348% of the main equipment item costs respectively to account for installation, instrumentation and controls, piping and electrical systems, buildings and services plus general design and construction expenses. The equipment costs for all of the AHX, HP evaporator and HP condenser were related to the required heat transfer area using data for industrial finned tube heat exchangers installations in New Zealand which were a precursor to the costs reported by Walmsley et al. (2015). The curve-fit relationship was:

$$EC_{HX} = 341A^{0.8} \tag{11}$$

The overall heat transfer coefficients (*U* values) were assumed to be 25, 40 and 45 W m^{-2} K^{-1} for the AHX, HP condenser/gas cooler and HP evaporator respectively. HP compressor and motor costs were related to compressor power using the relationships given by Bouman *et al*. (2005) yielding:

$$EC_{comp} = 4990P_{comp}^{0.676} + 179 + 87P_{comp} \tag{12}$$

The marginal operating costs relative to the existing plant were estimated using:

a) Utilities – reduced steam use at \$15.9 GJ^{-1} and extra electricity at \$27.8 GJ^{-1}.

b) Labour – no change for AHX but an extra 0.5 full-time-equivalent (FTE) per shift at $60k per FTE p.a. for the HP plus 5% for supervision and support.
c) Maintenance and Insurance – 3% of installed capital cost.

A simple discounted cash flow analysis was performed to estimate the internal rate of return on the extra capital (*IRR*) assuming the plant operated 4000 hours per year over a 10 year life. The most significant changes from the heat recovery analysis were that the actual electricity to steam energy price ratio was less than 2 rather than 2.4, and compressor motor efficiency was 95% rather than 90%. The analysis also assumed that the AHX could be costed as a single direct AHX rather than two indirect AHX with a liquid run-around loop which will make the AHX option appear more favourable. More details of the methodology and data sources are described by Brown & Cleland (2012).

4. RESULTS AND ANALYSIS

4.1 Heat Pump Refrigerants

Figure 4 shows the performance (COP_{hp}, *HRR* and *RCost*) of 21 refrigerants for a HP alone without an AHX at the reference condition as a function of their critical temperature. Table 1 summarises the properties and performance of a selection of these refrigerants. Of the refrigerants, those with a critical temperature less than about 105°C operated transcritical and those above 105°C operated sub-critical. Some of the discharge pressures were close to the critical pressure where thermodynamic properties change rapidly and are less reliably estimated, so the EES calculations were often very sensitive to small changes in operating conditions.

Figure 4 and Table 1 suggest that the better HP refrigerants operate with a transcritical cycle and have a critical temperature between 70°C and 105°C. Thermodynamically this makes sense because the cooling and dehumidification of the exhaust air is close to an isothermal process, which is well matched to evaporation of refrigerant well below the critical point, while the sensible heating of the inlet air is well-matched to the counter-current cooling of supercritical refrigerant (at pressures significantly higher than the critical point so that the internal "pinch" in the gas cooler is minimised but not so high as to reduce *COP* significantly). For example, using R134a at the reference conditions, the refrigerant exits the compressor and enters the gas cooler at about 125°C (and 45 bar) and the dryer inlet air is heated to about 113°C.

In terms of COP_{HP} and *RCost*, the most promising refrigerants are R32, R134a, R22 and R290. Of these R32 is a HFC with moderate GWP and is flammable, R134a has moderate GWP, R22 is an ozone depleting HCFC and R290 is highly flammable so R134a is probably the best option unless flammability can be cheaply mitigated. Therefore, the more detailed analysis was limited to R134a although some spot checks suggested the trends were similar for the other refrigerants.

4.2 Heat Pump Cycle Variations

The ambient sub-cooler provided mixed benefits. For the better matched refrigerants, use of an ambient sub-cooler, reduced *HRR* by 11-13% because some of the heat rejection was to ambient rather than the inlet air but increased COP_{HP} by 5-8% because lower discharge pressures could be used to reject the reduced *HRR*. The net effect was that *RCost* changed by up to 3% (increased for transcritical; decreased for sub-critical), but the big effect was that HP refrigerant mass flowrate decreased by more than 10% so compressor capital costs and the overall economic feasibility were likely to improve (although offset by the capital cost for the sub-cooler).

Further analysis for the most promising refrigerants showed that an internal heat exchanger in the HP with an effectiveness of 0.3 changed the heat recovery system *COP* by less than 2% and resulted in less than 5% change in *HRR*. Also, the compressor discharge temperature increased which may exacerbate lubricant selection problems. Therefore the rest of the analysis was for a HP configuration without an internal heat exchanger. Similarly, a Brayton cycle using air was also investigated. To be competitive with the most common sub-critical or transcritical cycle HPs, the expander and compressor efficiencies both had to be at least 10 to 15% higher than the compressor efficiency for the common cycles. The main advantage of the Brayton cycle would be availability of large scale machinery.

4.3 Effect of Refrigerant Evaporation Temperature

Figure 5 shows the performance (COP_{HP}, *HRR* and *RCost*) of the combined AHX-HP heat recovery system with a R134a HP as a function of the refrigerant evaporation temperature (T_{eva}). At the reference condition, 16.2% of the air heating is performed by the AHX (i.e. the minimum *HRR* is 0.162). Above a T_{eva} of about 35°C the HP does negligible heating so most of the heat recovery is by the AHX. Relative to the HP alone configuration, COP_{HP} and *HRR* are both lower because the temperature lift for the HP is higher after inlet air preheating and exhaust air cooling by the AHX. The likely lower cost for an AHX than a HP, mean that complete omission of an AHX is unlikely.

RCost is lower than for a HP alone and there appears to be an optimum at about 25-30°C. As anticipated, there is a complex tradeoff between *HRR* increasing at lower HP T_{eva}, COP_{HP} decreasing at lower T_{eva}, and capital cost probably increasing at lower T_{eva} as the capacity of the HP increases. This latter fact means that the true economic optimum including capital cost is probably at a HP T_{eva} greater than 25-30°C. It is possible that the marginal savings of the HP over and above the AHX do not justify the capital cost even although COP_{HP} can be significantly higher than 3.

4.4 Effect of Heat Exchanger Temperature Approach

Figure 6 shows *HRR*, COP_{HP} and *RCost* as TA_1 and TA_2 vary from the reference conditions for a R134a HP alone system. As expected as TA_1 decreases, COP_{HP} increases and *RCost* decreases because a lower discharge pressure can be used to reject all the heat into the inlet air. However, potentially counter-intuitively, *HRR* decreases slightly with decrease in TA_1 because the compressor discharge temperature reduces so there is slightly less heat to reject to the inlet air.

As TA_2 decreases a greater fraction of the exhaust air heat is recovered so *HRR* increases but to reject all this heat to the inlet air requires a much higher discharge pressure and therefore a lower COP_{HP} results (Figure 6). The net effect on *RCost* is a slight decrease so capital cost implications are likely to be more important than the effect of TA_2 on energy use. At higher *HRR*, the capital of all HP components will increase due to the greater capacity required, while at lower TA_2 the capital cost of the gas cooler will increase. The net effect is that the overall economic optimum may not be at lower TA_2.

Figure 7 shows *HRR*, COP_{HP} and *RCost* for R134a as TA_3 varies from the reference conditions for a combined AHX-HP system. As expected as TA_3 decreases, the AHX does more heat recovery so the HP does less and must operate with a higher temperature lift. Consequently, total *HRR* decreases, COP_{HP} decreases and overall *RCost* decreases slightly. However, the overall economics may improve because the capital cost of the HP will be reduced relative to a HP alone.

4.5 Economic Feasibility

The fact that the actual ratio of electricity to steam price used in the economic analysis was about 27% lower than in the heat recovery analysis means that use of HPs is likely to be more economically attractive.

4.5.1 AHX Alone

Figure 8 shows the *HRR*, operating cost savings and *IRR* for an AHX alone as a function of temperature approach (TA_3). For the reference case, an AHX is economically attractive and can provide over 20% of the air heating with an *IRR* greater than 40%. The *IRR* declines as the *HRR* increases so the size of AHX system to maximize heat recovery would probably be one that just meets the minimum *IRR* for the business. Walmsley *et al*. (2015) explore the AHX alone option in more detail for a similar spray dryer.

4.5.2 HP Alone

Figure 9 shows shows the *HRR*, *RCost*, operating cost savings and *IRR* for a HP alone as a function of evaporation temperature (T_{eva}). There are significant cost savings of up to $390,000 p.a., *RCost* gets to as low as 0.68 when T_{eva} is about 27°C, and the HP can provide most of the air heating if T_{eva} is 25°C or less. However, the capital cost is so high that the *IRR* is about -11% when operating cost savings are maximized and *IRR* is never better than -7% even if less than 20% of the heating duty is satisfied. If an *IRR* of 30% was required, then for the reference case the capital cost would need to be less than NZ$1.7M (about NZ$300 kW^{-1} of heating) which is about four times lower than the current estimate of NZ$7.0M. To operate in the region with higher costs savings, the compressor discharge pressure would need to be greater than 60 bar and possibly at high as 120 bar. This may be possible given recent developments in carbon dioxide compressors.

4.5.3 Combined AHX-HP System

Figure 10 shows the *HRR*, *RCost*, operating cost savings and *IRR* for a combined AHX-HP system as a function of HP evaporation temperature (T_{eva}). The *IRR* of 56% for an AHX alone at the reference conditions (Figure 8) is reduced by being combined with a HP which is clearly not economic given the current estimates of capital costs.

Figure 11 shows the marginal *HRR*, operating cost savings and *IRR* for a combined AHX-HP system as a function of HP evaporation temperature (T_{eva}) relative to a AHX alone system at the reference conditions. Comparing Figures 9 and 11, shows that as part of the combined system, the HP is less viable than as a HP alone, because the exhaust air temperature entering the HP is cooler which lowers the HP COP (lower T_{eva}) for the same *HRR* or reduces the capacity (*HRR*) for the same T_{eva} even although the discharge pressures are also slightly lower for the same T_{eva} due to the reduced capacity. For the combined system to be economically feasible, then the capital cost of the HP system would need to significantly reduced as discussed for the HP alone system. Reduced temperature approach (*TA*) slightly improves the *IRR* for the combined system but the overall economics are controlled by the HP capital cost.

4.5.4 Maximising Heat Recovery

From an environmental perspective, maximizing heat recovery is likely to minimize fossil fuel use. Figure 12 shows the *IRR* as a function of *HRR* for the various systems at the reference conditions. The AHX alone is clearly the most economic option but is limited to a *HRR* less than 25%. Addition of a HP allows higher *HRR*, but it is not economically attractive unless capital costs can be significantly reduced. The combined AHX-HP system is more attractive than a HP alone for the same *HRR* because the AHX is so cost-effective.

5. CONCLUSIONS

For practical temperature differences an AHX provides about 15-20% of the air heating load. Combined with a HP, it is practical to provide up to about 40% of the air heating load with a HP *COP* greater than 3 such that the overall energy costs are reduced by more than 20% relative to steam heating. The transcritical cycle with R134a evaporating at about 25-30^{o}C is well-matched to the process.

Economically, only use of an AHX is attractive but is limited to about 20% of the heating load. Adding an R134a HP to get higher heat recovery than an AHX alone would only be attractive if HP capital costs could be reduced significantly to less than about NZ$300 kW^{-1} heating capacity.

6. ACKNOWLEDGMENTS

Funding: This work was supported by the New Zealand Foundation for Research, Science & Technology via a sub-contract from the University of Waikato as part of the programme – Engineering Energy Efficiency [Contract UoWX0709].

7. NOMENCLATURE

A	heat transfer area (m^2)
COP	coefficient of performance
EC	equipment cost (2011 NZ$)
H	absolute humidity (kg kg^{-1} dry air)
HRR	heat recovery ratio
IRR	internal rate of return (% year^{-1})
m	mass flow rate of dry air (kg s^{-1})
P	pressure (Pa) or power (kW)
PR	pressure ratio of heat pump
RCost	fractional operating costs relative to steam air heating
RH	air relative humidity (%)

T temperature (°C)
TA_1 temperature approach in condenser (gas cooler) and liquid sub-cooler (°C)
TA_2 temperature approach in evaporator (°C)
TA_3 temperature approach in AHX (°C)
U overall heat transfer coefficient ($Wm^{-2}K^{-1}$)
ΔT_{lm} log-mean temperature difference (°C)
ϕ sensible heat transfer rate (W)
η_b boiler thermal efficiency
η_{is} compressor isentropic efficiency

Subscripts
ae1 initial state of dryer exhaust air
ae2 outlet state of dryer exhaust air
ai1 initial state of inlet air
ai2 outlet state of inlet air
amb ambient air
con condensation condition
comp compressor and motor
crit refrigerant critical point
d dryer inlet air
dew dewpoint of moist air
dis discharge condition
HP heat pump
HX heat exchanger
eva refrigerant evaporation
m intermediate state
s saturation
t total/overall

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Table 1. Performance of heat pump (HP) alone heat recovery systems for a range of refrigerants with T_{eva}=30^{o}C, TA_1=TA_2=10^{o}C, TA_3=20^{o}C.

Refrigerant	COP_{hp}	*HRR*	*RCost*	Cycle Type	T_{con} or P_{dis}	T_{crit} (^{o}C)	P_{crit} (MPa)
R32	3.907	0.499	0.807	Transcritical	7.22 MPa	78.1	5.78
R134a	3.849	0.494	0.814	Transcritical	4.47 MPa	101.0	4.06
R143a	3.828	0.481	0.821	Transcritical	6.59 MPa	72.7	3.76
R22	3.796	0.5046	0.814	Transcritical	5.21 MPa	96.1	4.99
R290	3.755	0.497	0.821	Transcritical	5.04 MPa	96.7	4.25
R152a	3.743	0.509	0.817	Subcritical	106^{o}C	113.3	4.52
R124	3.676	0.503	0.825	Subcritical	116^{o}C	122.3	3.62
R125	3.637	0.476	0.838	Transcritical	8.50 MPa	66.0	3.62
R600a	3.480	0.510	0.842	Subcritical	124^{o}C	135.0	3.65
R717	3.419	0.543	0.838	Subcritical	103^{o}C	132.3	11.33
R114	3.377	0.5153	0.851	Subcritical	131^{o}C	145.7	3.29
R123	2.997	0.556	0.889	Subcritical	128^{o}C	183.7	3.67
R744	1.527	0.835	1.477	Transcritical	64.50 MPa	31.0	7.38

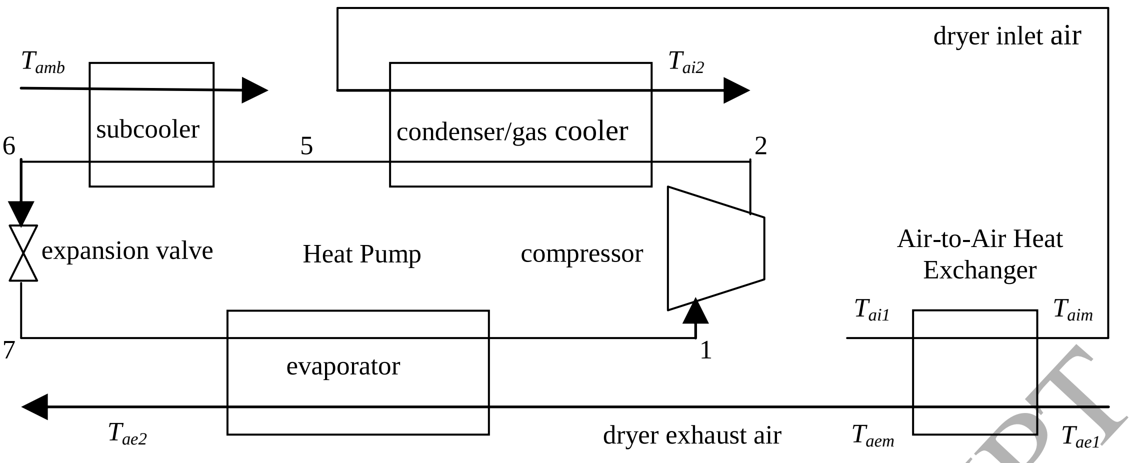

Figure 1: Schematic of the dryer heat recovery using a combined air-to-air heat exchanger (AHX) and heat pump (HP) in series.

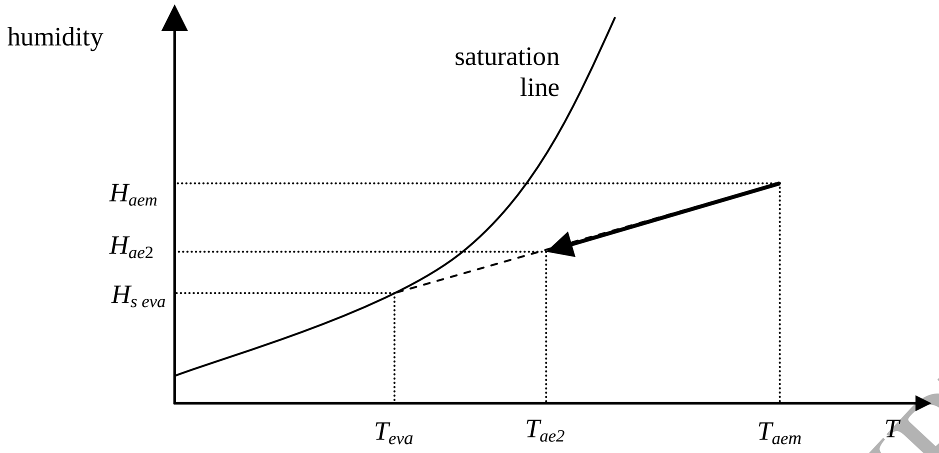

Figure 2: Psychrometric chart showing cooling and dehumidification of exhaust air in the heat pump evaporator.

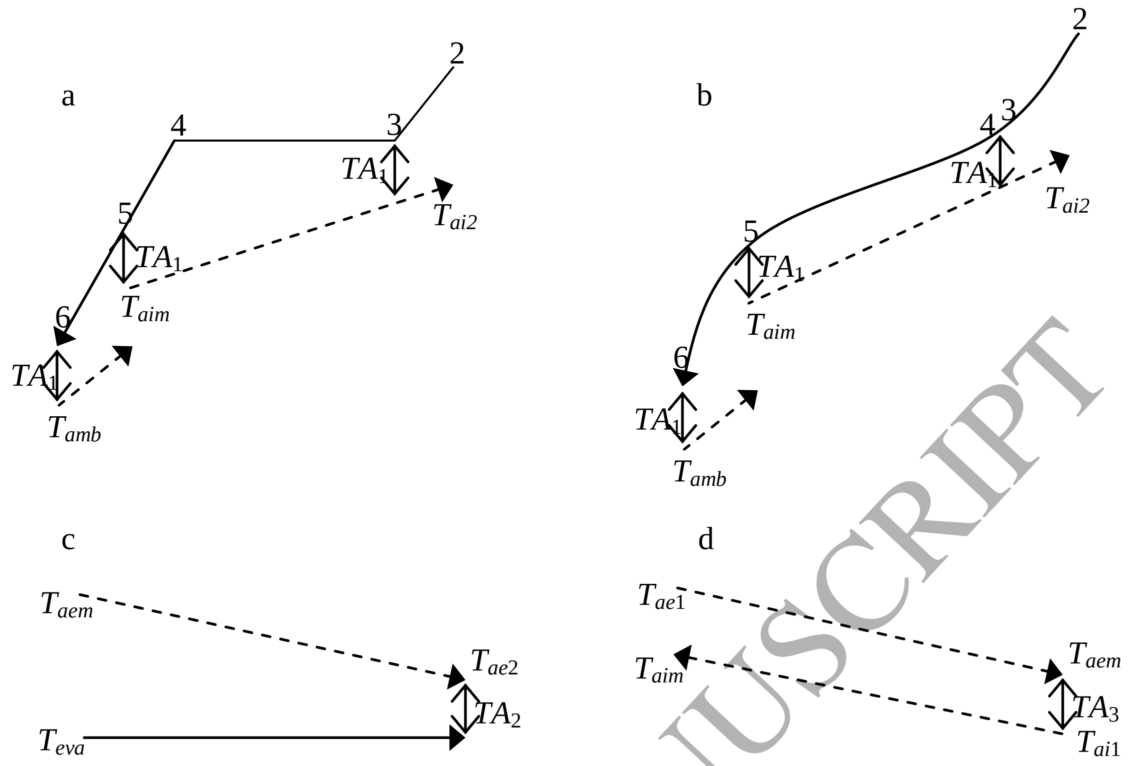

Figure 3: Schematic plots of temperature (vertical axis) versus heat flow (horizontal axis) showing temperature approaches (*TA*) in (a) subcritical cycle condenser and ambient sub-cooler (b) transcritical cycle gas cooler and ambient sub-cooler (c) heat pump evaporator (d) exhaust to inlet air-to-air heat exchanger.

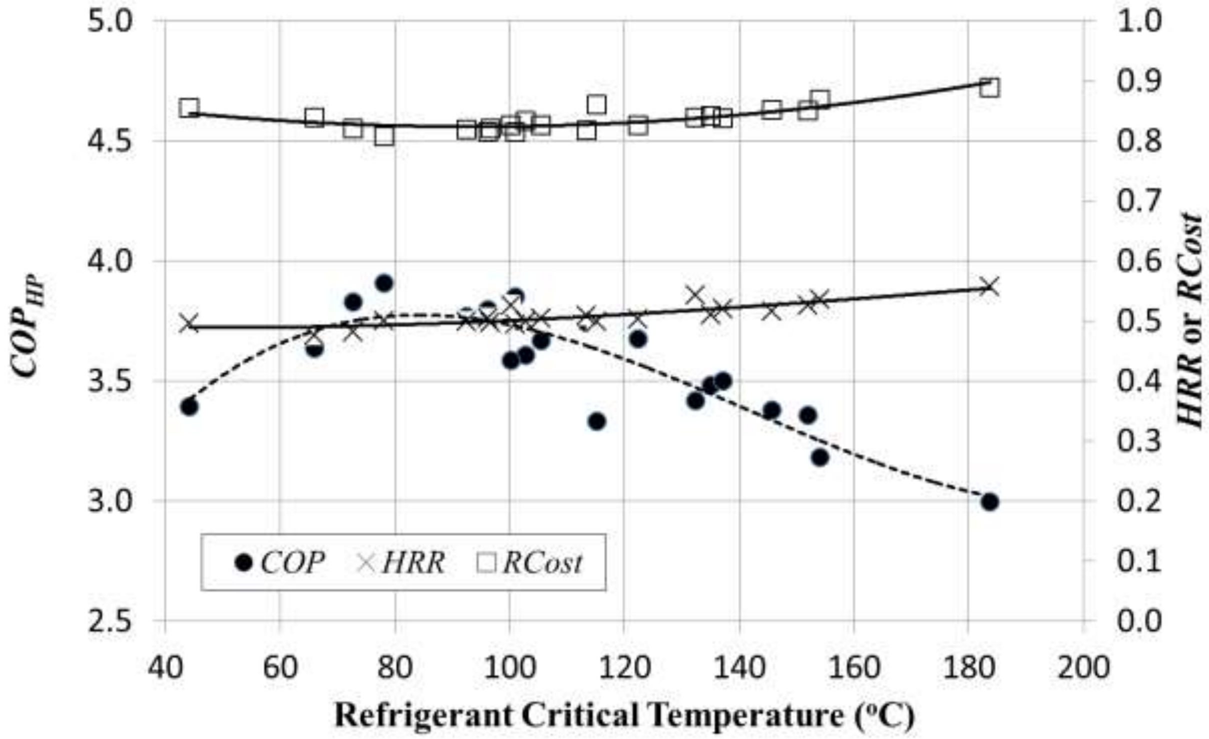

Figure 4: COP_{HP} (--●--), *HRR* (×) and *RCost* (□) as a function of refrigerant critical temperature for a heat pump (HP) alone heat recovery system at the reference conditions.

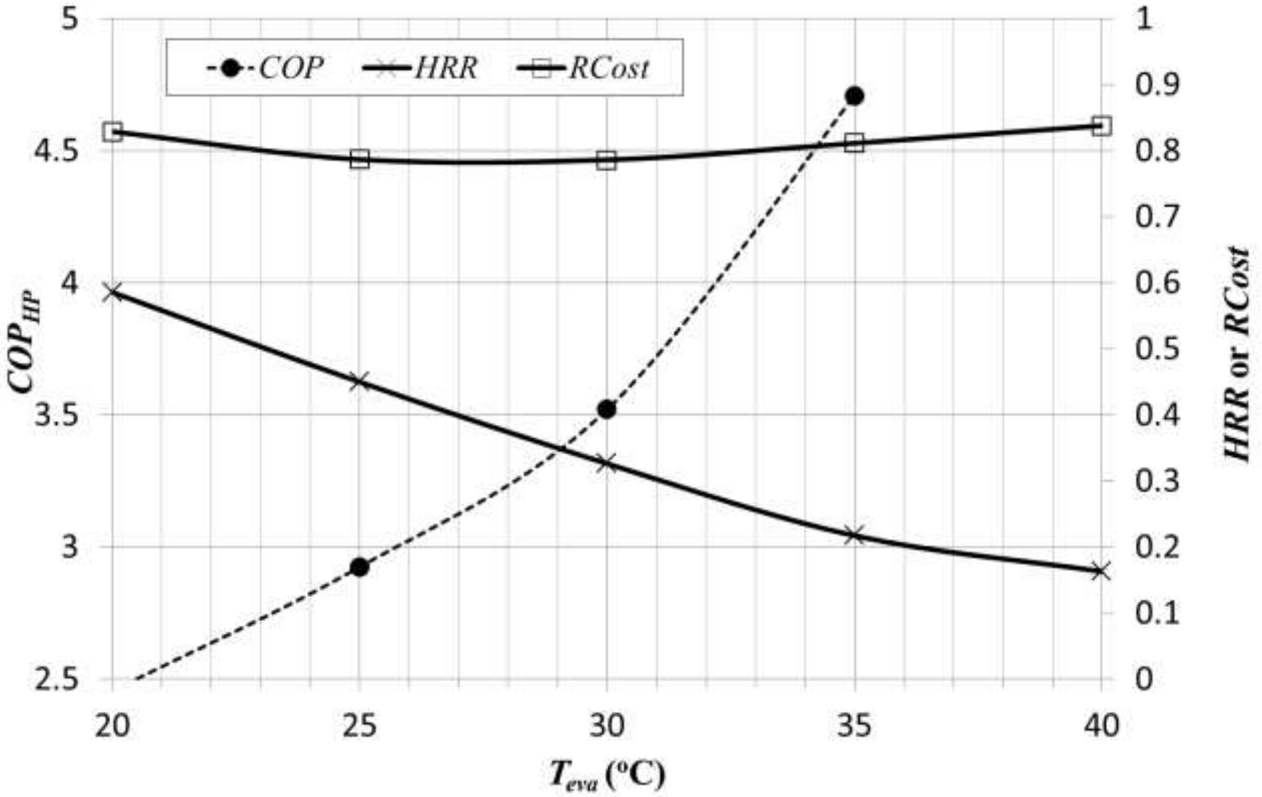

Figure 5: COP_{HP} (--●--), *HRR* (×) and *RCost* (□) as a function of heat pump (HP) evaporation temperature, T_{eva}, for a combined AHX-HP heat recovery system using R134a.

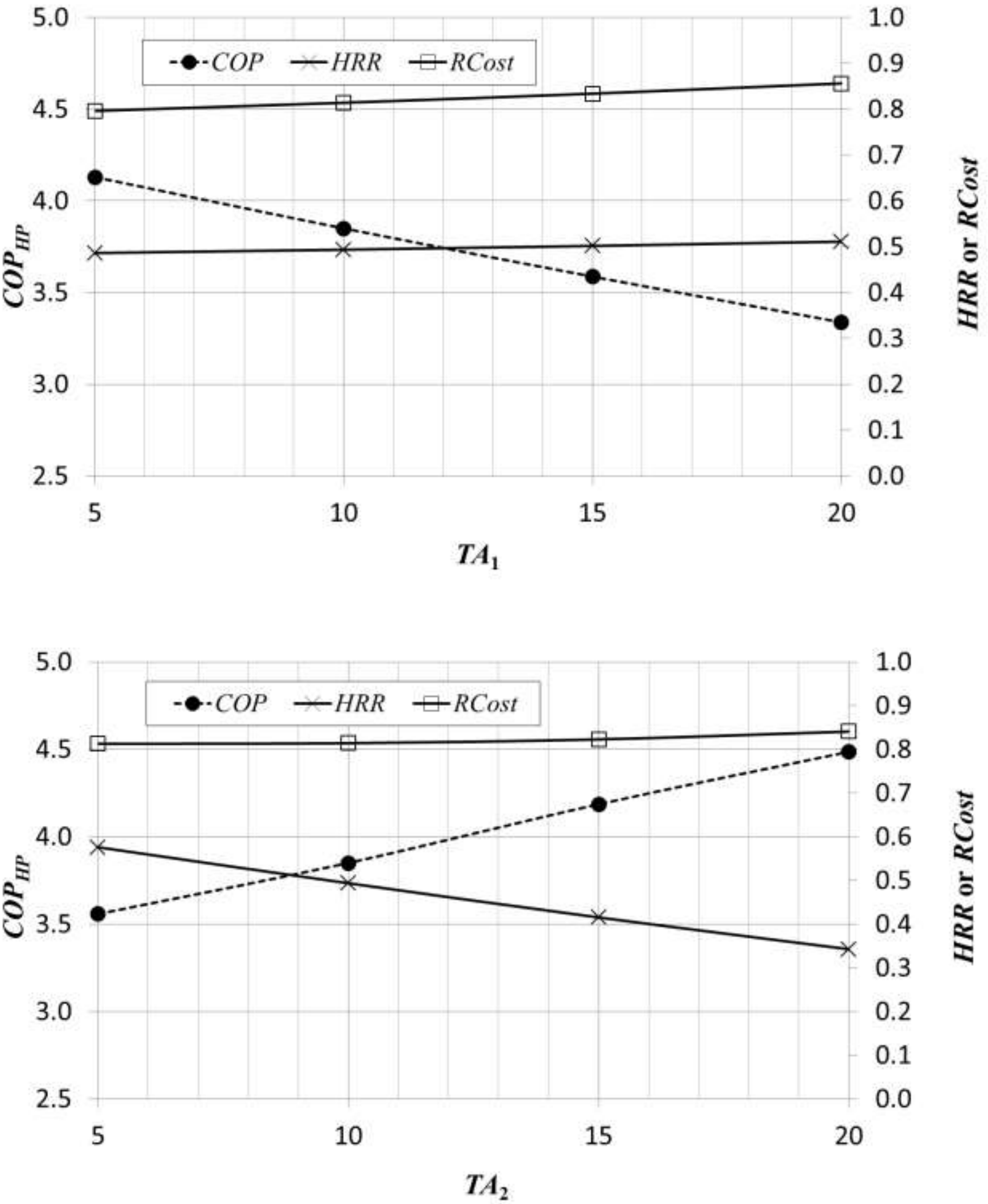

Figure 6: COP_{HP} (--●--) , HRR (×) and $RCost$ (□) for a R134a heat pump (HP) alone as temperature approach, TA_1 or TA_2, vary from reference conditions.

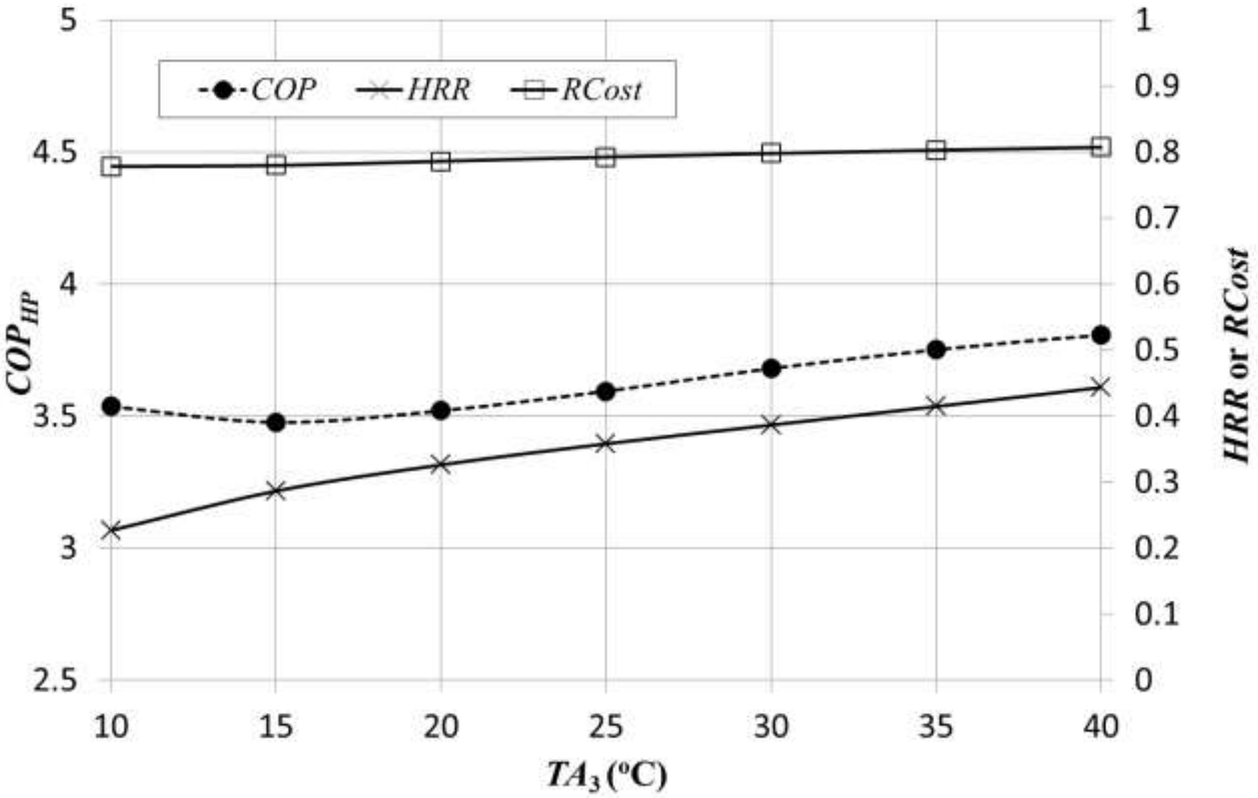

Figure 7: COP_{HP} (--●--) , *HRR* (×) and *RCost* (□) for a R134a heat pump (HP) in a combined AHX-HP system as temperature approach, TA_3, varies from reference conditions.

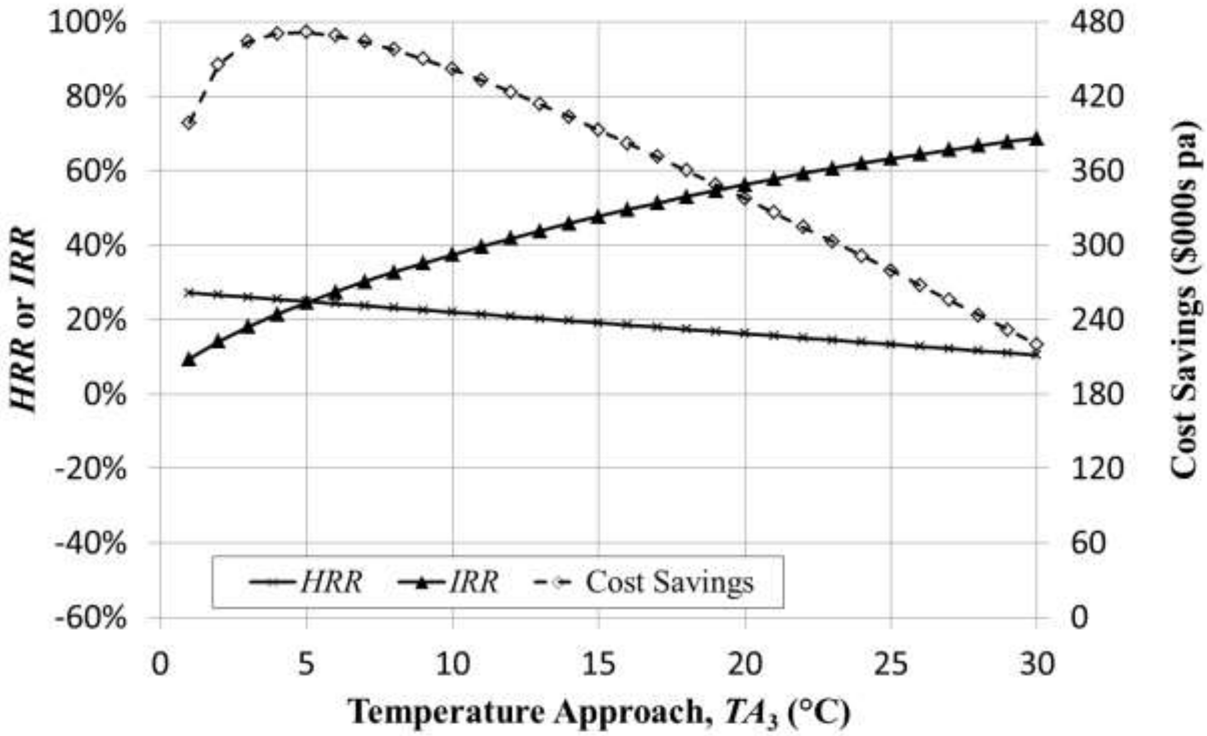

Figure 8: *HRR* (×),*IRR* (▲) and annual operational cost savings (--◊--) for an air-to-air heat exchanger (AHX) alone as temperature approach, TA_3, varies from reference conditions.

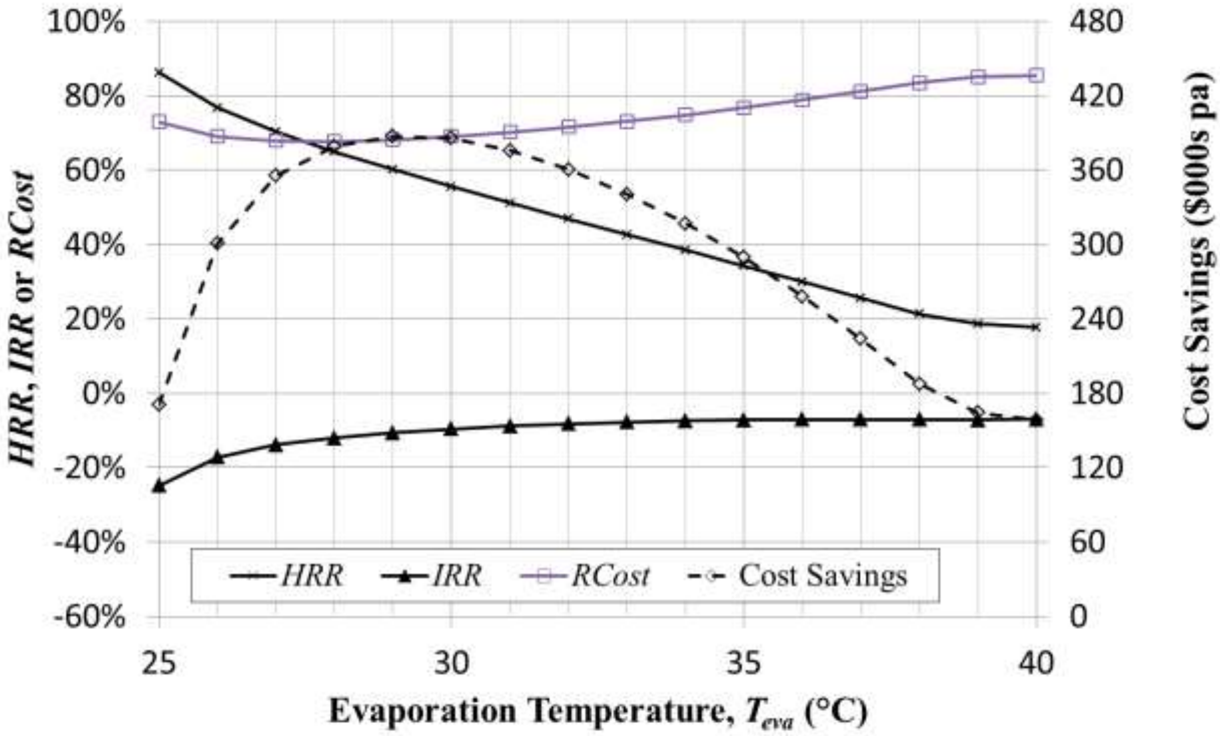

Figure 9: *HRR* (×), *RCost* (□), *IRR* (▲) and annual operational cost savings (--◊--) for a R134a heat pump (HP) alone as evaporation temperature, T_{eva}, varies from reference conditions.

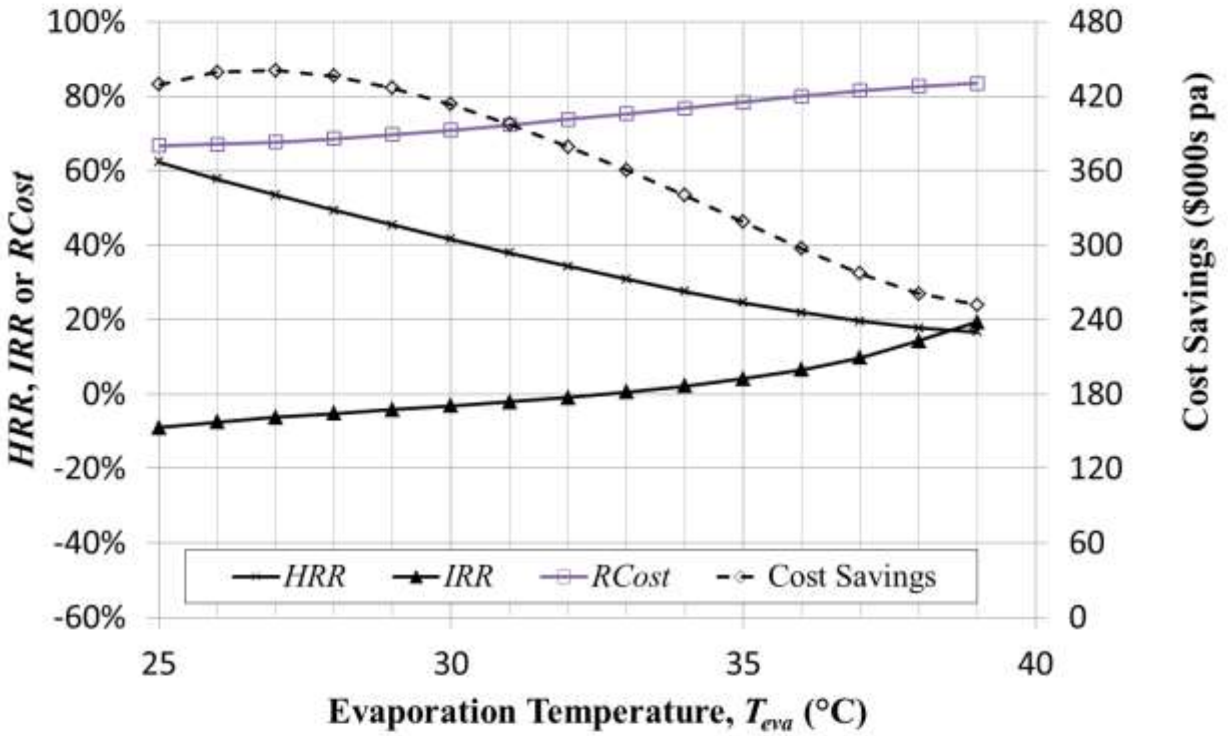

Figure 10: *HRR* (×), *RCost* (□), *IRR* (▲) and annual operational cost savings (--◊--) for a combined air-to-air heat exchanger and heat pump (AHX-HP) system as the R134a HP evaporation temperature, T_{eva}, varies from reference conditions.

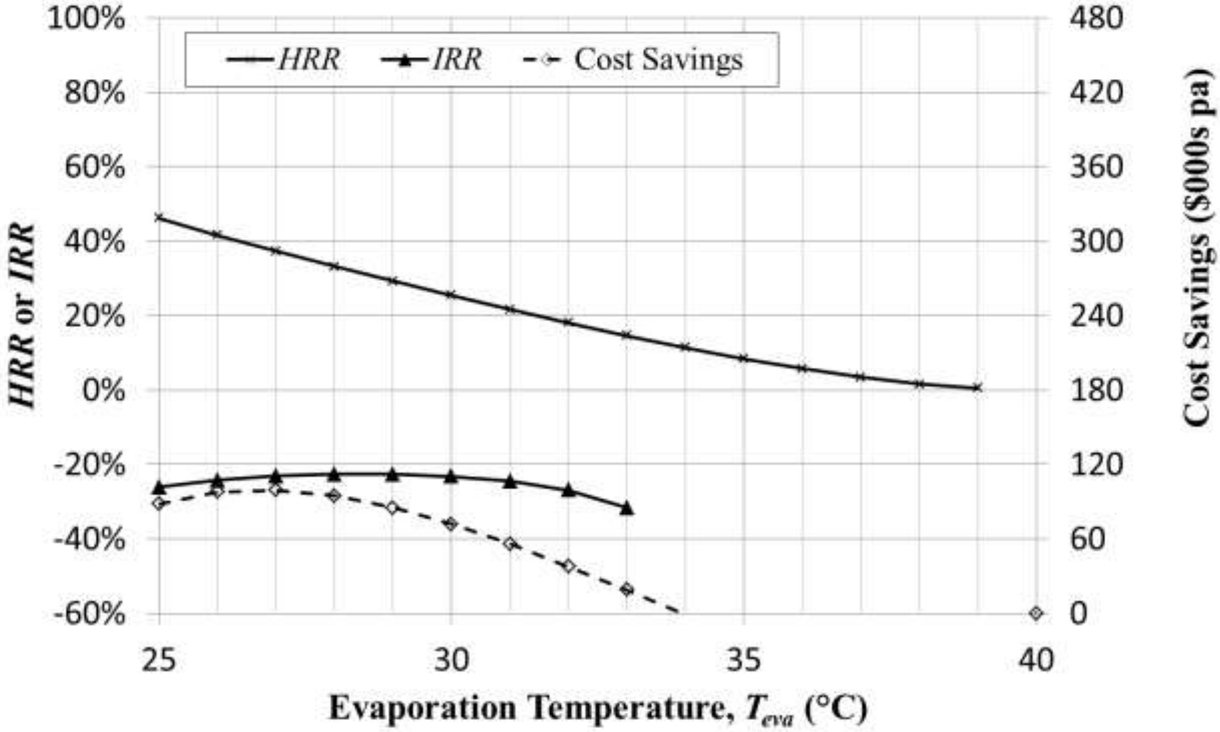

Figure 11: Relative to a air-to-air heat exchanger (AHX) alone, marginal *HRR* (×), *IRR* (▲) and annual operational cost savings (--◊--) for a combined AHX and heat pump (HP) system as the R134a HP evaporation temperature, T_{eva}, varies from reference conditions.

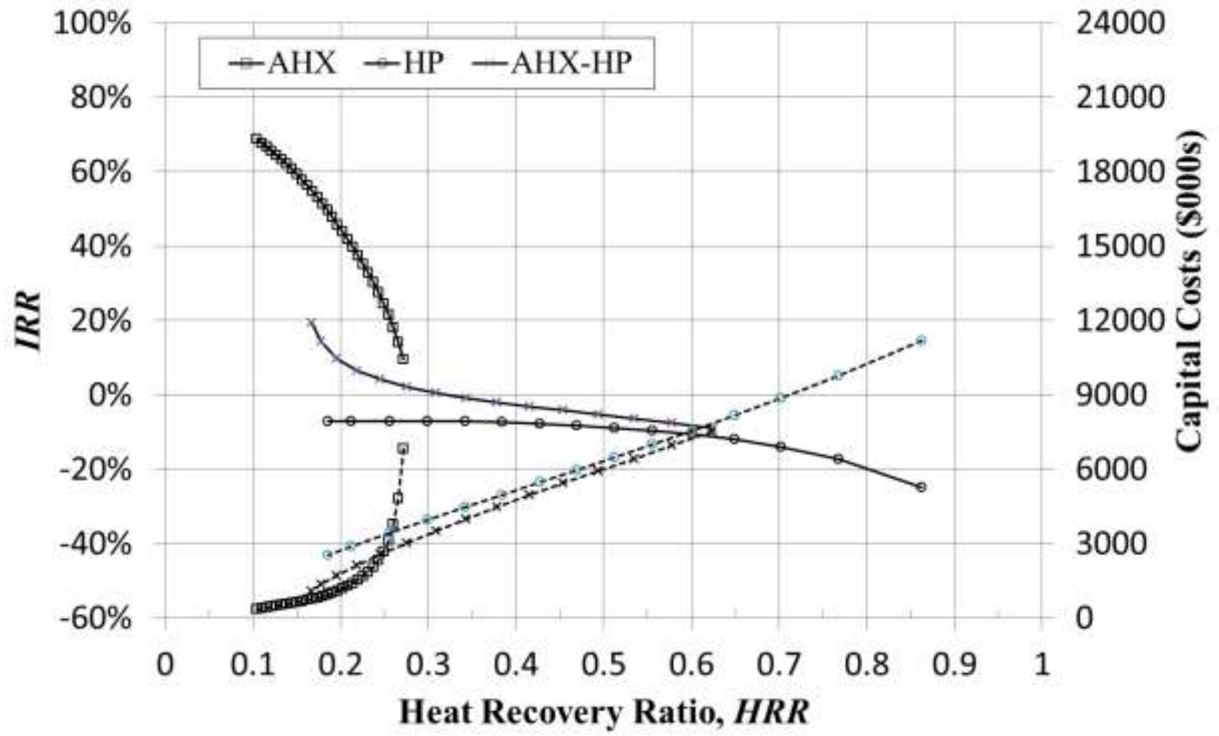

Figure 12: Internal rate of return, *IRR*, (□) and total capital costs (----) for an air-to-air heat exchanger (AHX) alone (□), heat pump (HP) alone (o) and a combined AHX-HP system (×) as a function of heat recovery ratio (*HRR*) for the reference conditions (different *HRR* by changing TA_3 for the AHX alone and T_{eva} for the HP alone and combined AHX-HP systems).