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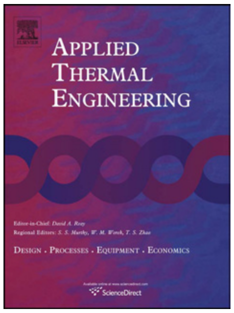

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Integrating heat recovery from milk powder spray dryer exhausts in the dairy industry

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Keywords: process integration; heat recovery; pinch analysis; spray drying.

Abstract

Heat recovery from milk powder spray dryer exhausts has proven challenging due to both economic and thermodynamic constraints. Integrating the dryer with the rest of the process (e.g. evaporation stages) can increase the viability of exhaust recovery. Several potential integration schemes for a milk powder plant have been investigated. Indirect heat transfer via a coupled loop between the spray dryer exhaust and various heat sinks were modeled and the practical heat recovery potential determined. Hot utility use was reduced by as much as 21% if suitable heat sinks are selected. Due to high particle loading and operating temperatures in the particle sticky regime, powder deposition in the exhaust heat exchanger is perhaps the greatest obstacle for implementing heat recovery schemes on spray dryers. Adequate cleaning systems are needed to ensure continuous dyer operation.

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Nomenclature

ε	effectiveness
ε_o	overall effectiveness
Δp	pressure drop (Pa)
ΔT_{cold}	temperature change in cold stream (°C)
ΔT_{min}	minimum approach temperature (°C)
c_p	specific heat capacity (kJ/kg°C)
m	mass flow rate (kg/s)
Nu	Nusselt number
Q	amount of heat exchange (kW)
Q_{cold}	cold utility (kW)
Q_{hot}	hot utility (kW)
Q_{max}	maximum possible heat transfer (kW)
Re	Reynolds number
RH	relative humidity (%)
T_{amb}	ambient temperature (°C)
T_{DP}	dew point temperature (°C)
T_{exh}	exhaust temperature (°C)
$T_{h.in}$	hot stream inlet temperature (°C)
$T_{c.in}$	cold stream inlet temperature (°C)
T_{in}	inlet temperature (°C)
T_{out}	outlet temperature (°C)
T_s	supply temperature (°C)
T_t	target temperature (°C)
U	overall heat transfer coefficient
v_{inlet}	average inlet heat exchanger face velocity (m/s)

v_{exh} average exhaust heat exchanger face velocity (m/s)

1. Introduction

Spray drying is an energy intensive operation and comprises a significant portion of final industrial energy use worldwide and is especially important in the drying of food products such as milk powder [1-3]. Most spray dryers have little or no heat integration and there remains a significant opportunity to reduce energy consumption by applying well established process integration principles [4, 5]. Kemp [6] discusses the application of these principles to drying in general and concludes that the potential for heat recovery from a typical dryer is particularly constrained both thermodynamically and economically. Kemp also discusses seven methods for energy reduction, which can be divided into three broad categories: a) reduce the heat required for drying; b) reduce the net heat supplied by hot utility (i.e. via heat recovery); and c) reduce energy cost by fuel switching, Combined Heat and Power (CHP), heat pumps, etc. There are constraints with the feasibility of some of these methods primarily due to the nature of product being produced and equipment limitations. For example, a common method to reduce drying load is to increase the solids concentration of the dryer feed; however there is a practical limit that a spray dryer can be fed due to the rheology of the feed material.

There has been limited success with heat recovery from milk spray dryer exhausts with heat exchangers or recuperators [7-10]. As a result typical milk powder spray dryers are not integrated whatsoever due to three main factors: a) economics; b) particle loading and fouling; and c) relatively low exhaust temperatures. The obvious place for recovered heat is to pre-heat the incoming dryer air, either directly or indirectly [10, 11]. From an integration perspective there are limited heat sinks if the dryer is considered in isolation to the entire process and the amount of feasible heat recovery may be severely constrained. The average inlet air temperature prior to the air heater may be as high as 40 °C and the exhaust as low as 65 °C, which restricts the amount of heat that can be recovered. In order to improve the opportunity for heat recovery the entire production process should be considered [12, 13].

Heat recovery opportunities from the exhaust of a typical milk powder spray dryers will be investigated in this paper. Several possible integration schemes using indirect liquid-coupled heat exchange from the exhaust of the spray drying with different heat sinks will be examined. The main impediment of particulate fouling of the exhaust heat exchanger surface will also be discussed.

2. Model Powder Plant and Process Description

A schematic of a typical milk powder production process is shown in Fig. 1 and the stream data for the process are summarised in Table 1. A coupled loop heat exchanger between the dryer exhaust and inlet air is shown as an example of a potential exhaust heat recovery scheme. The plant considered here has a nominal production capacity of 23 t/h and the total evaporative load for the entire plant is 63.6 kg/s with around 57.9 kg/s removed by the evaporators. A multi-effect evaporator train concentrates the milk from around 9% solids to 52% solids using both mechanical vapour recompression (MVR) and thermal vapour recompression (TVR). The milk concentrate is heated further to the dryer feed temperature before it is sprayed into the main drying chamber. The water removed in the evaporators is called cow water (CW) and is used to partially pre-heat the incoming evaporator feed before being discharged. The air for the dryer is taken from within the building and has a supply temperature (T_{amb}) of 25 °C and humidity of 0.0065 $kg_{water}/kg_{dry\ air}$ before being heated to the dryer inlet temperature air (T_{in}) of 200 °C, as illustrated by the line $T_{amb} – T_{in}$ on the Psychrometric and Mollier Charts in Fig. 2. The dryer air increases in humidity and decreases in temperature as the milk is dried in the main drying chamber, as shown by the line $T_{in} – T_{exh}$ in Fig. 2. The exhaust air from the main drying chamber and the fluidised beds are combine before exiting the baghouse at 75 °C. The relative humidity (*RH*) of the exhaust is low, around 18%, and there is a 36 °C approach temperature to the dew point (T_{DP}).

The cyclones and baghouse filter attempt to remove any entrained powder from the air stream. The Site Hot Water (SHW) is used throughout the site for Cleaning in Place operations (CIP) etc. The composite and grand composite curves for the plant are illustrated in Fig. 3, using a minimum approach temperature (ΔT_{min}) of 20 °C. This temperature was chosen as a relative measure to avoid using a range of stream specific ΔT_{min} approach temperatures. The evaporators themselves have not been included in the composite curves although the condensing load of the final TVR effect has been included. The pinch temperature is 49 °C and the minimum hot and cold utility requirement is 25,241 kW and 15,871 kW respectively. These minimum utility targets should be treated with some caution as they represent an ideal case based on a ΔT_{min} of 20°C and the heat exchanger network that meets these targets may not be the most practical or ideal from an operational and structural viewpoint. The targets should be considered as a guide based on a reasonable ΔT_{min}. A large portion of the cold utility target is the exhaust air, which is a non-essential cooling load and is supplied by ambient air. Hot utility for the plant is supplied by steam. The cow water exit temperature does not necessarily need to be above 20 °C but should be low enough to comply with the maximum discharge temperature of 30 °C.

3. Coupled Loop and Heat Exchanger Methodology

Indirect heat recovery from the drier exhaust can be achieved using conventional finned-tube heat exchanger technology combined with a liquid coupled loop. The upper and lower temperature limits of the exhaust and heat sink streams set the constraints of the coupled loop and standard heat exchanger sizing methods apply.

To determine the performance of the heat recovery heat exchangers on the coupled loop the ε-NTU (Effectiveness – Number of Heat Transfer Units) approach and the methodology outlined by Kays and London [14] for finned-tube exchangers was used. For the plate heat exchangers (PHX) the ε-NTU approach and the method outlined in Shah and Sekulić [15] was used. The dimensions of both heat exchangers were fixed depending on the desired average face velocity of the air contacting the exchanger and fixed heat exchanger volume. The total volume of the inlet heat exchanger section was 26 m^3 for an average face velocity of 2.5 m/s. The total exhaust heat exchanger volume at an average face velocity of 4 m/s was 76.4 m^3. From the dimensions and the geometric data for the various tubes or plates considered the number of tubes or plates was calculated. The flow rate of the coupled loop was set and the hot and cold loop temperatures were arbitrarily set initially. The film heat transfer coefficients for the streams were calculated using the appropriate heat transfer and friction factor correlations. The heat transfer and friction factor design data for the several tubes considered here were taken from Kays and London [14].

Both the exhaust and inlet air heat exchangers were modelled as single pass cross-flow heat exchangers with both fluids unmixed. The flow inside the finned-tubes was treated as internal flow inside a smooth circular pipe, with the Nu = 4.37 approximation used in the laminar regime ($Re \geq 2300$), the Petukhov-Popov correlation [15] used for turbulent regime ($Re \geq 4000$), and a linear interpolation between the two correlations used in the transition regime. Water was selected as the working fluid for the loop. Where a PHX was used as a pre-heat exchanger the heat transfer and friction correlations were taken from Shah and Sekulić [15] and pure counter-current flow was assumed. The water side pressure drop (Δp) through the tubes and/or plates was also calculated based on the heat exchanger specifications and loop flow rate.

Based on the film heat transfer coefficients the overall heat transfer coefficient (U) was then calculated followed by the NTU value and effectiveness (ε). Once the effectiveness of the individual heat exchangers is known the duty of the exchanger was then calculated. From the duty of the heat exchangers the outlet temperatures of the hot and cold streams were calculated. The duties of the two exchangers need to be equal for the system to be solved. An iterative approach was used where the hot and cold temperature of the loop were varied until the duties of the two coupled heat exchangers were equal. Overall effectiveness (ε_o) was calculated based on the exhaust and cold stream as in Equation 1 where Q is the actual amount of heat exchange, Q_{max} is the maximum amount of heat exchange possible due to thermodynamic constraints, $mc_{p.min}$ is the minimum heat capacity rate, and $T_{h.in}$ and $T_{c.in}$ are in inlet temperatures of the hot and cold streams respectively.

$$\varepsilon_O = \frac{Q}{Q_{max}} = \frac{Q}{mc_{p.min}(T_{h.in} - T_{c.in})} \qquad (1)$$

4. Heat Recovery Schemes

Six indirect heat recovery schemes were considered matching the exhaust stream to different heat sinks including: a) inlet air; b) SHW; and c) milk. The obvious selection is to pre-heat the inlet air with heat recovered from the exhaust as illustrated by the schematic in Fig. 1. A summary of utility savings for each scheme is shown in Table 2.

A major heating requirement in the powder plant is the dryer inlet air and this heat sink is usually the first candidate to be considered for pre-heating using recovered heat from the exhaust. The overall effectiveness and temperature increase (ΔT_{cold}) of the inlet air is shown in Fig. 4. There is a trade-off between maximum heat recovery and heat exchanger size because there is a slight reduction in ε_o with using a much smaller heat exchanger (i.e. high face velocity). The volume of the heat exchanger with an average face velocity of 4 m/s is four times smaller than for 1 m/s with only a slight recovery penalty.

The optimum loop mass flow rate is 47 kg/s and the ε_o reaches 70%, which corresponds to 4,169 kW of heat recovery and a 35 °C increase in the inlet air or a 12.8% decrease in hot utility use. If a multi-pass exchanger is used then this amount will increase as more passes are used, although there is a diminishing benefit as more passes are added. Although the optimising the number of passes is not considered here, other work undertaken by the authors has demonstrated that around 4 to 5 passes is optimum [16].

The effect of the type of finned-tube on the amount of heat recovery for a set volume of heat exchanger is shown in Fig. 5. It is clear from the figure that more heat transfer is obtained for the small diameter tubes and decreases as the tube diameter is increased. There is only a slight increase in heat recovery with increasing the fin pitch, although increasing the fin pitch will have a detrimental affect on powder deposition, which will be discussed later. The increased heat transfer with the smaller diameter tubes is due to the increased surface area per unit volume although there is an increase in the water side pressure drop. The water side pressure drop across both the exhaust and inlet exchangers and the air side pressure drop for all of the finned-tubes considered here is not prohibitive at the range of flow rates.

One of the disadvantages of matching the inlet air stream with the exhaust is that the maximum temperature difference is limited to 50 °C, which limits the amount of heat recovery that is possible. If the exhaust is matched to another sink with a lower temperature, more heat is able to be recovered. If that stream is a liquid stream a PHX can be used to heat the size, which involves much less capital than a finned-tube exchanger. The next sink to be considered is the SHW stream as this stream provides several advantages over using the inlet air. Firstly, the supply temperature of the SHW is 10 °C lower than the inlet air, and the mc_p is slightly higher than the inlet air. In both cases the sink streams are the limiting factor for heat recovery. The second advantage is that a PHX can be used for exchange between the SHW and coupled loop stream. The ε_o as a function of loop flow rate is illustrated in Fig. 6 for three average channel velocities in the PHX.

The coldest stream is the incoming milk stream although heating this stream with heat from the exhaust would involve re-examining how the CW is used for heating. Raw Milk A involves substituting CW heating of the milk with recovered heat from the exhaust and uses CW to pre-heat the incoming inlet air followed by SHW. After the inlet air pre-heat the CW has an outlet temperature of around 47 °C, which can then be used to pre-heat the SHW. There is the need for additional cooling on the CW to meet a

30 °C discharge requirement. The amount recovered from the exhaust in this case is 7,527 kW; however the load on the raw milk heater is increased by around 3,300 kW and so the overall benefit is only marginally better than for the inlet air case.

Raw Milk B, is similar to Raw Milk A, but uses the CW as the first pre-heat to the raw milk after the inlet air pre-heat. The amount of heat recovered from the exhaust is less than before because of the elevated temperature of the raw milk due to the CW pre-heat and the overall benefit is less because of this reduced amount of recovery from the exhaust is decreased and because there is no heat recovery from the evaporator condenser.

Raw Milk C involves using the CW to first pre-heat inlet air and raw milk as before, which lowers the temperature sufficiently to allow the CW to be used as cooling utility to the condenser. Some of the heat from the condenser can be recovered from the CW to further heat the raw milk followed by the SHW. The final discharge temperature of the CW is around 26 °C and therefore no further cooling is required before discharge. There is a reduction of hot utility of 21.3% and a complete elimination of the need for a cooling tower.

If the exhaust stream is split into two and is matched to the SHW and the inlet air then the amount of heat recovery increases and the hot utility savings are 20.8%. This scheme is much simpler than Milk Option C and has many operational advantages. The capital requirement is greater as it involves two exhaust exchangers and piping loops but the benefits could far outweigh any additional capital requirements. From a simplicity and capital cost point of view this scheme appears to be the most attractive.

The economic feasibility of dryer exhaust heat recovery is highly dependent on the site energy cost and the primary fuel source and there can be a large variation of energy costs between plants. However for sites with moderate to high thermal energy costs, exhaust heat recovery can be economically attractive with simple pay backs in the order of two to five years.

5. Exhaust Heat Exchanger Fouling

One of the greatest obstacles to the viability of heat recovery from the spray dryer exhaust is the problem of fouling due to powder deposition. Baghouses are used to recover as much powder as possible before the air is exhausted and have collection efficiencies of between 99.5 and 99.98% [17]. The particle or fines loading on the baghouse is highly dependent on the milk powder product being produced and can range anywhere from 40 to 90% of the produced powder [18]. Therefore the actual particle load on the exhaust heat exchanger is unknown; however for a 50% fines loading the loading through the exchanger could range between 57.5 to 2.3 kg/h depending on the collection efficiency. Stickiness of milk powder is determined by the composition, temperature, *RH*, particle size, and velocity. The sticky curve for pure lactose and an adjusted sticky curve for velocity are shown in Fig. 7. Lactose is a major component in milk powder and represents the worst case scenario for stickiness. The final temperature of the exhaust stream is also indicated for the various scenarios. It is clear that all of the schemes will have a portion of the exhaust heat exchanger in the sticky region and therefore a high percentage of the powder is expected to deposit on the tubes and fins of the exchanger. The affect of this deposition on heat transfer and pressure drop is uncertain and more research is needed to quantify these factors so that optimal cleaning regimes and scheduling can be developed.

6. Conclusions

Heat recovery from the exhaust of a milk powder spray dryer can reduce hot utility by between 12 and 20% depending on the heat sinks that are selected. Modelling the actual heat exchangers rather than just using an assumed minimum approach temperature is vital to assess the recovery potential using liquid coupled heat exchangers between the hot exhaust and the heat sink. Perhaps the greatest challenge for implementation of exhaust heat recovery systems in milk powder plants is particulate fouling in the exhaust heat exchanger. A large portion of the heat exchanger will be in the particle sticky region and as a consequence adequate cleaning systems and procedures are required.

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Figure 1. Schematic of a typical milk powder plant including an example of a couple loop heat exchanger from the exhaust to the dryer inlet air.

Figure 2. Psychrometric chart (left) and Mollier chart (right) showing the path for dryer air in the spray dryer.

Figure 3. Hot and cold composite curves (left) and grand composite curve (right) for the evaporator and dryer plant (evaporators are excluded).

Figure 4. Overall effectiveness and cold stream temperature increase for a range of loop mass flow rates.

Figure 5. Overall effectiveness and cold stream temperature increase for a range of loop mass flow rates for several different finned-tubes.

Figure 6. Overall effectiveness for a range of loop mass flow rates at several different PHX channel velocities.

Figure 7. Powder sticky curves for lactose and exhaust stream temperature/relative humidity profile.

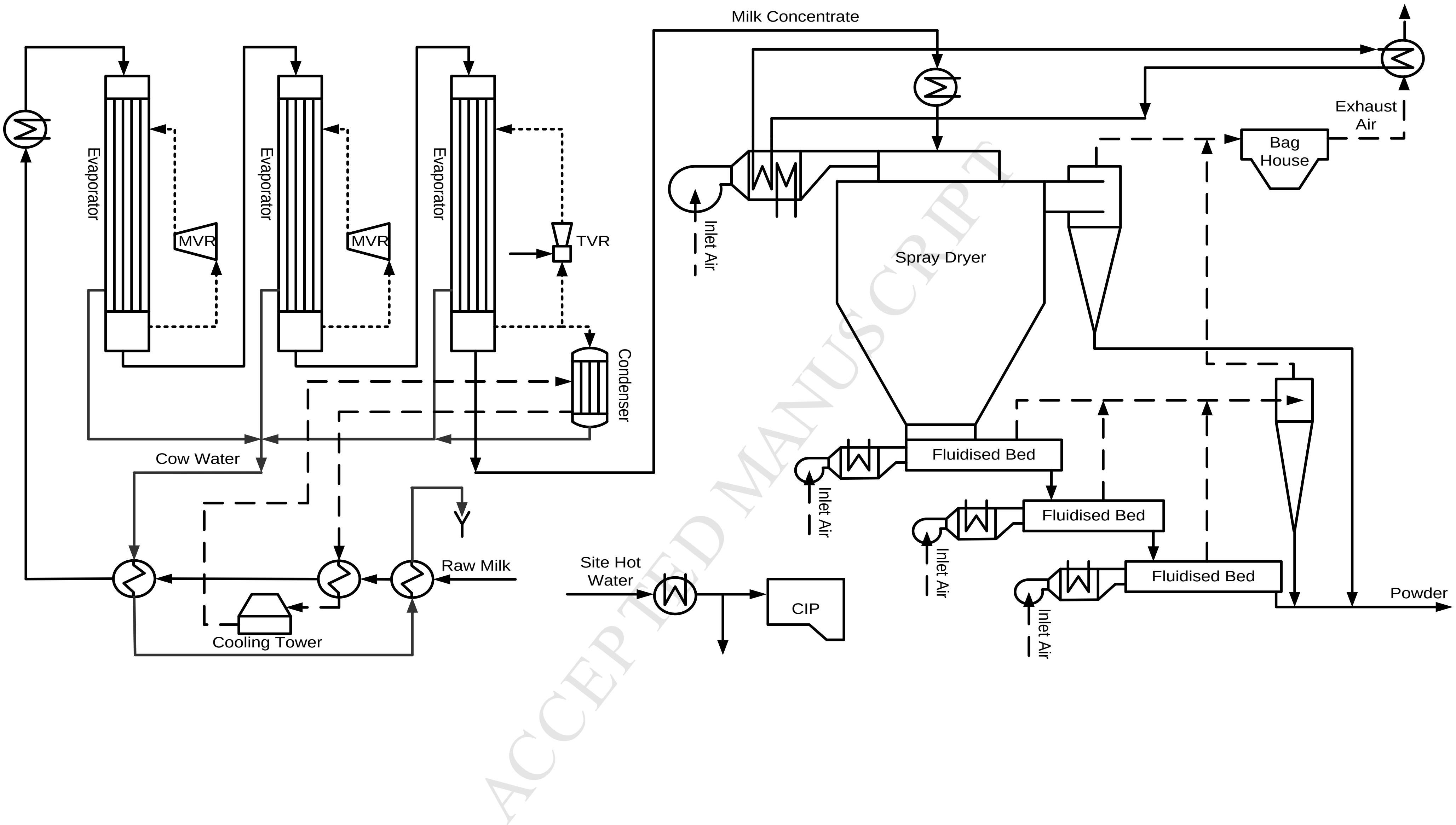
Table 1. Stream data for a typical milk powder plant (excluding evaporators).

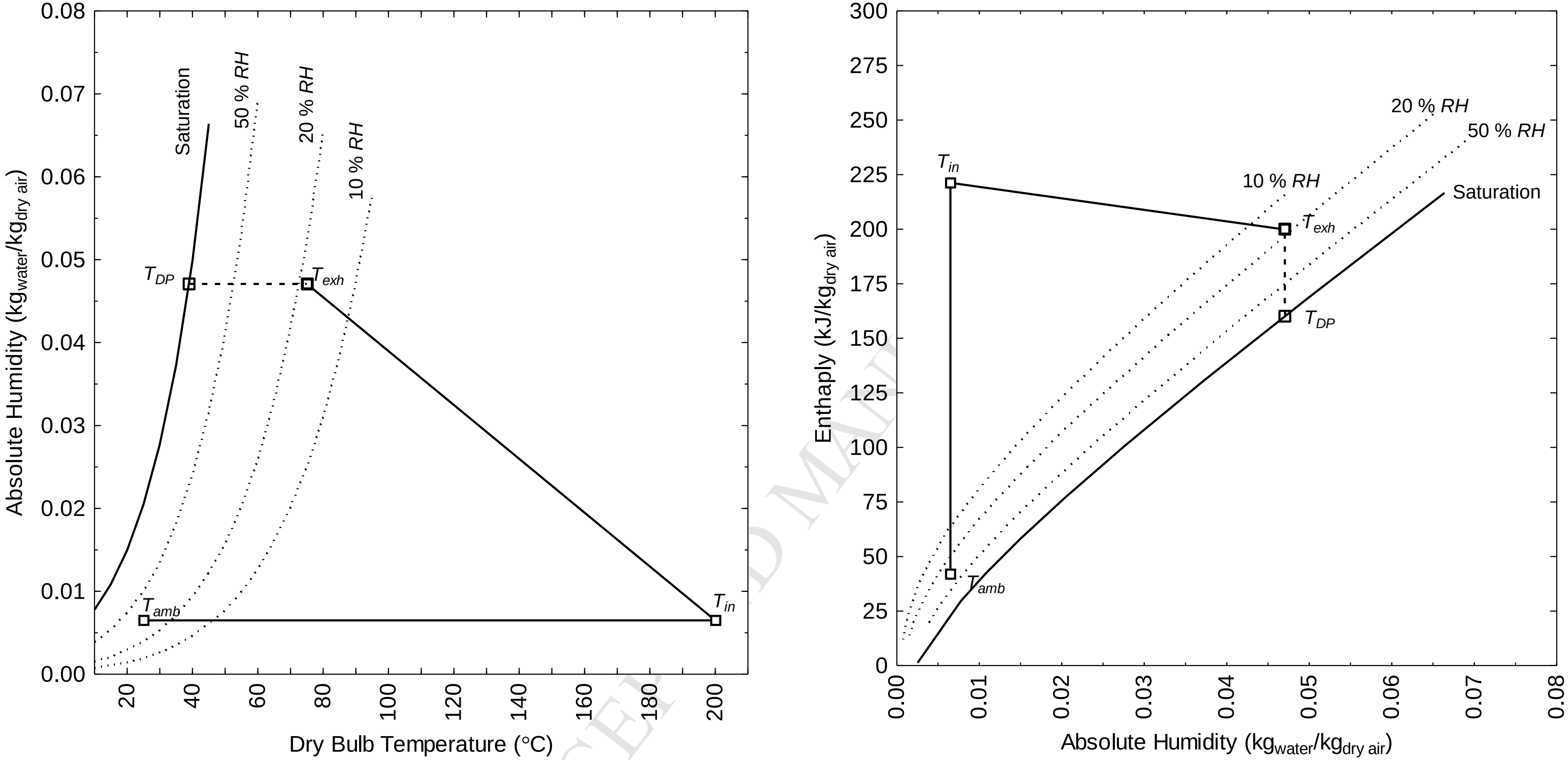
Table 2. Summary of utility savings for the several heat recovery schemes.

Stream Name	Type	T_s (°C)	T_t (°C)	m (kg/s)	mc_p (kW/°C)	Absolute Humidity ($kg_{water}/kg_{dry\ air}$)
Raw Milk	Cold	10	75	70.00	280.0	-
Cow Water	Hot	64	20	58.83	245.9	-
TVR Condenser	Hot	54	53	0.95	2388.2	-
Milk Concentrate	Cold	54	65	12.12	37.6	-
Dryer Inlet Air	Cold	25	200	117.00	119.2	0.0065
Fluid Bed A Inlet Air	Cold	25	50	10.00	10.2	0.0065
Fluid Bed B Inlet Air	Cold	25	45	14.60	14.9	0.0065
Fluid Bed C Inlet Air	Cold	25	32	11.00	11.2	0.0065
Air Exhaust	Hot	75	20	159.03	174.7	0.0471
Site Hot Water	Cold	15	55	30.00	125.4	-

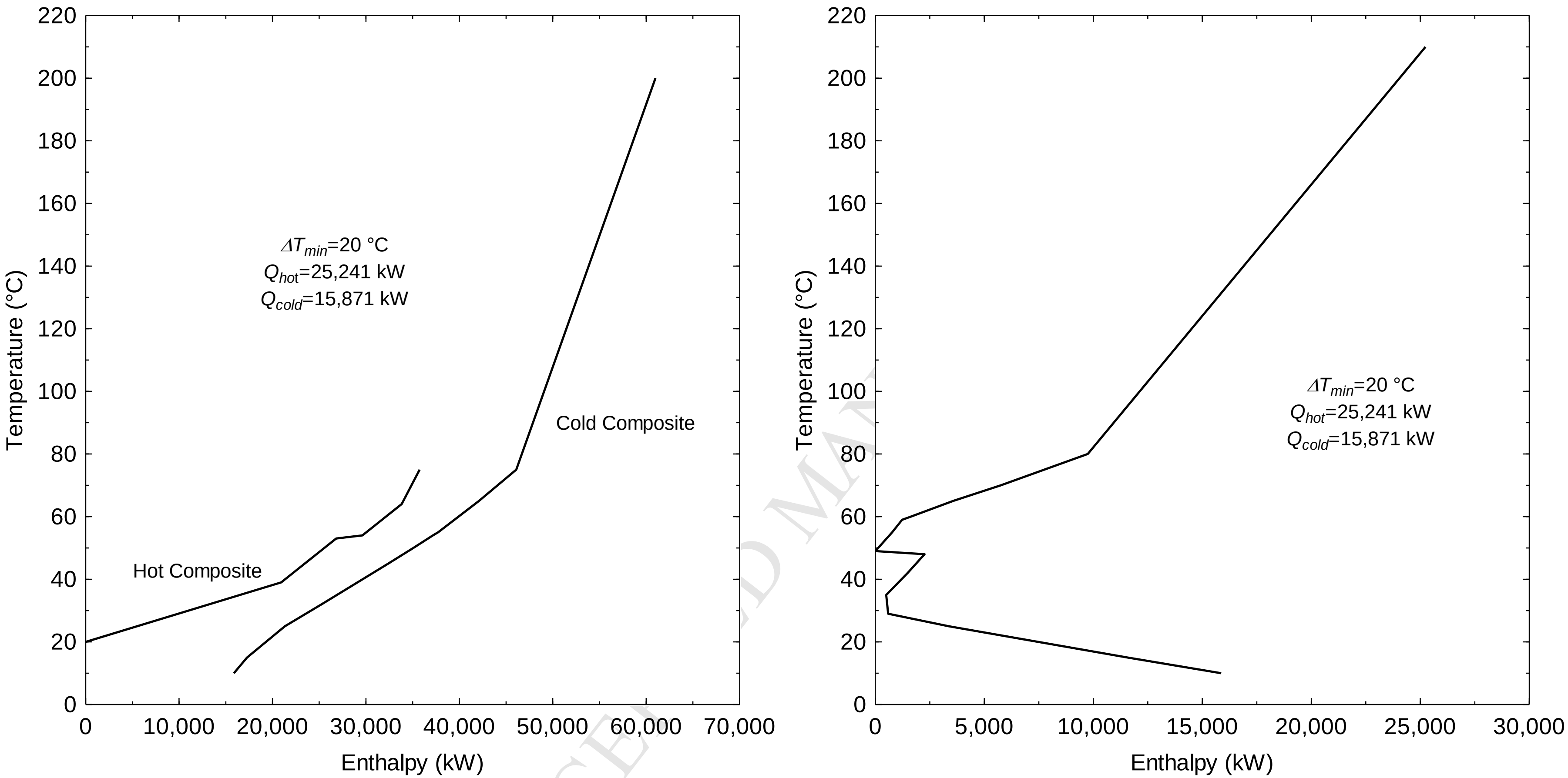
Scheme	Q_{hot} (kW)	Savings Q_{hot} (kW)	Savings Q_{hot} (%)	Q_{cold} (kW)[a]	Savings Q_{cold} (kW)	Savings Q_{cold} (%)
Base Case	32,693	-	-	788	-	-
Inlet Air	28,524	4,169	12.8	788	0	0
SHW	27,677	5,016	15.3	788	0	0
Raw Milk A	28,167	4,526	13.8	1,329	-541	-69
Raw Milk B	29,521	3,172	9.7	2,388	-1,600	-203
Raw Milk C	25,733	6,960	21.3	0	788	100
Split Exhaust	25,893	6,800	20.8	788	0	0

[a]Cold utility values do not include nonessential ambient cooling of the exhaust air and therefore are well below the target.

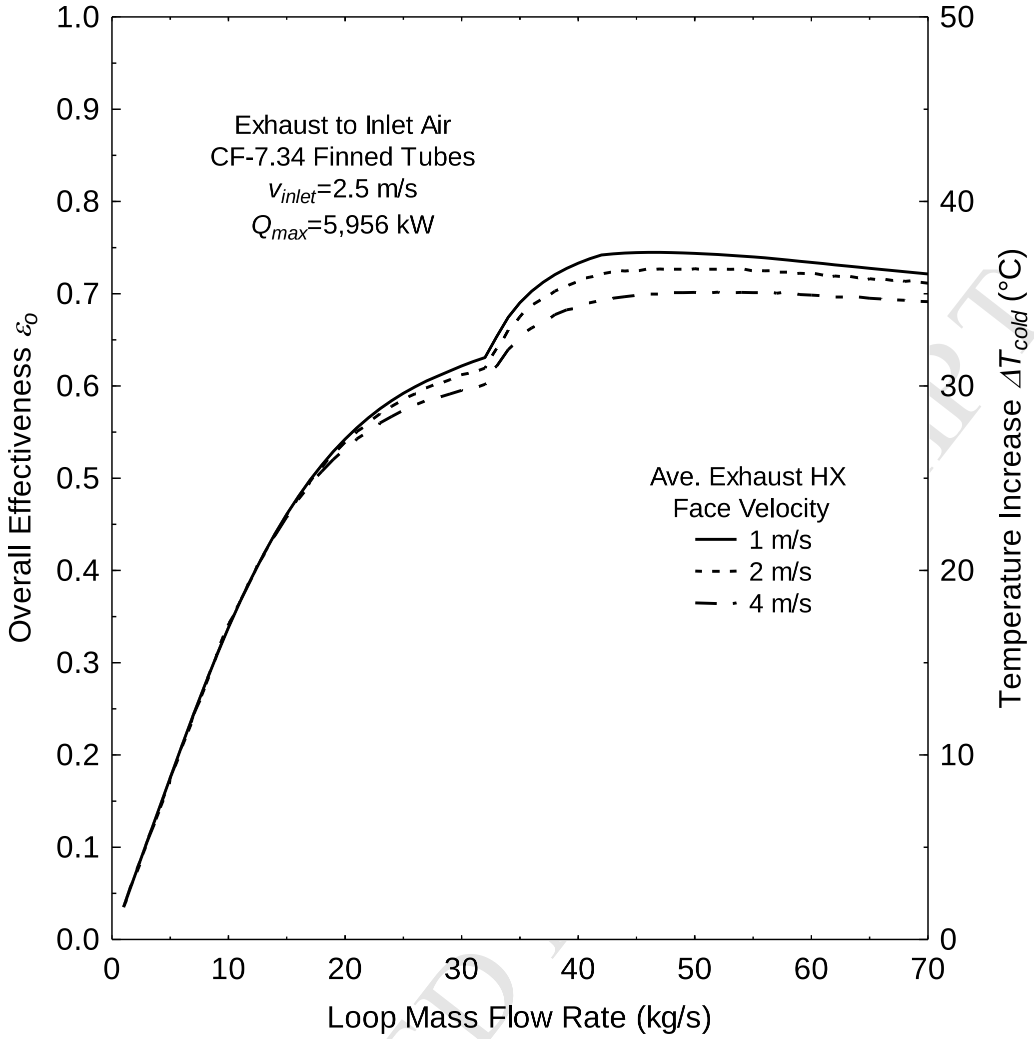

Milk Concentrate
Evaporator
Evaporator
Evaporator
MVR
MVR
TVR
Condenser
Cow Water
Raw Milk
Cooling Tower
Site Hot Water
CIP
Inlet Air
Spray Dryer
Exhaust Air
Bag House
Fluidised Bed
Fluidised Bed
Fluidised Bed
Inlet Air
Inlet Air
Inlet Air
Powder

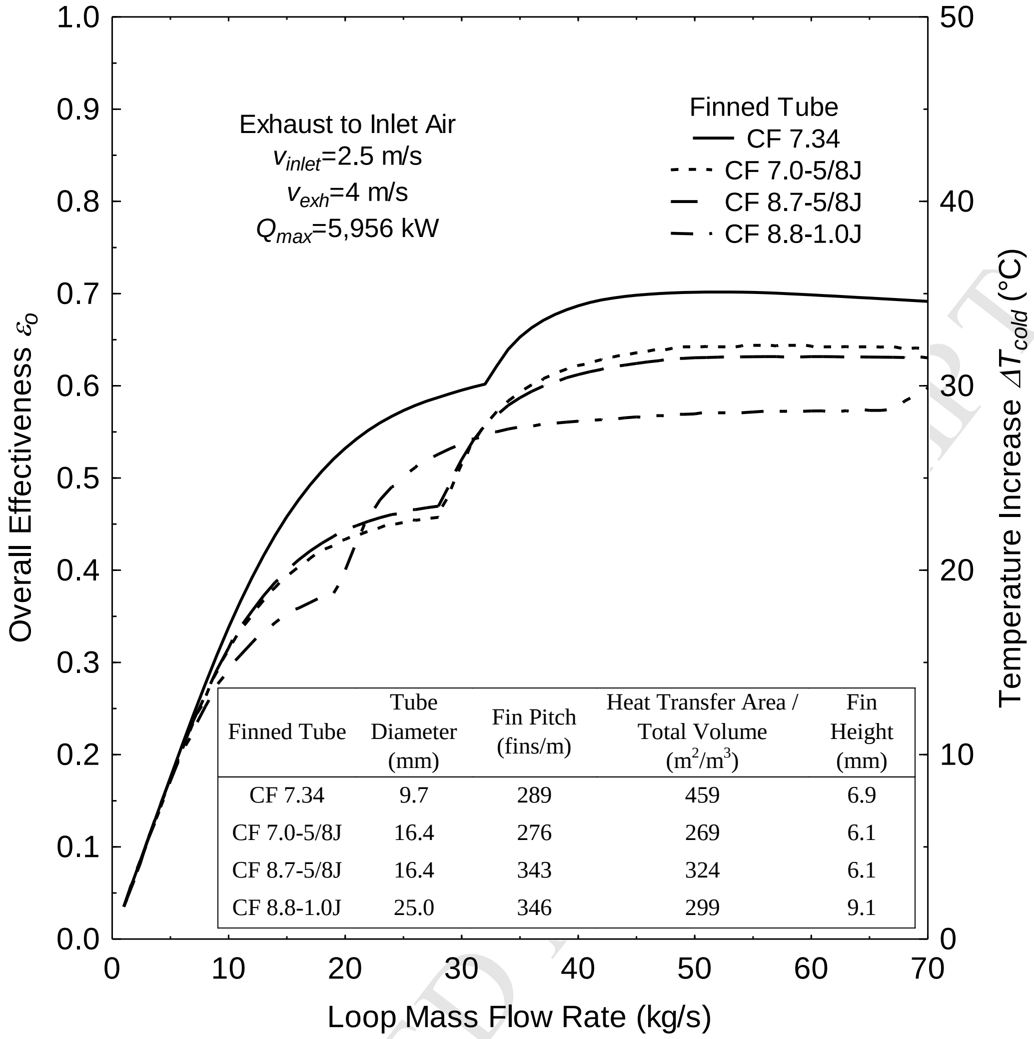


Saturation
50 % RH
20 % RH
10 % RH
T_{DP}
T_{exh}
T_{amb}
T_{in}
Absolute Humidity (kg$_{water}$/kg$_{dry\ air}$)
Dry Bulb Temperature (°C)
0.00
0.01
0.02
0.03
0.04
0.05
0.06
0.07
0.08
20
40
60
80
100
120
140
160
180
200
Enthaply (kJ/kg$_{dry\ air}$)
0
25
50
75
100
125
150
175
200
225
250
275
300

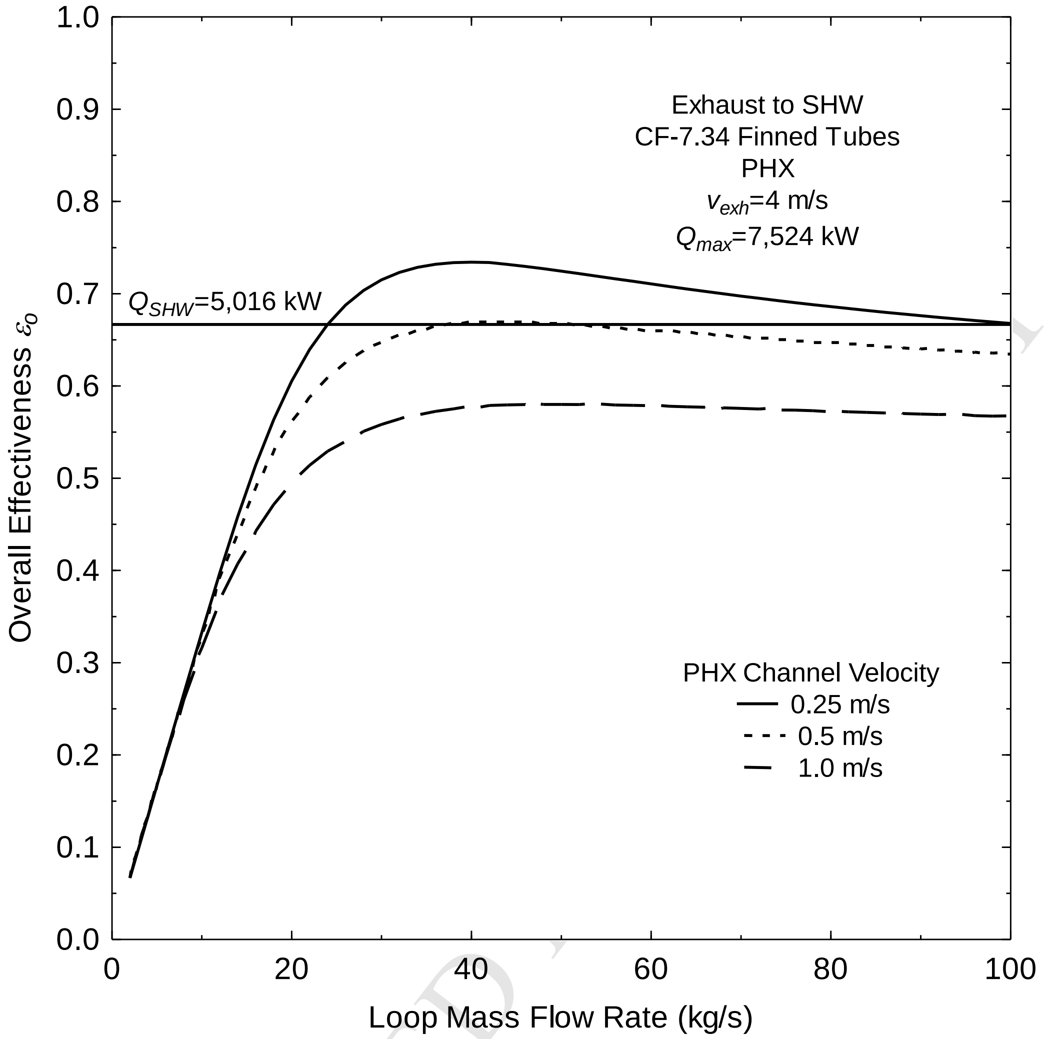


220
200
180
160
140
120
100
80
60
40
20
0
Temperature (°C)
ΔTmin=20 °C
Qhot=25,241 kW
Qcold=15,871 kW
Cold Composite
Hot Composite
0
10,000
20,000
30,000
40,000
50,000
60,000
70,000
Enthalpy (kW)
5,000
15,000
25,000

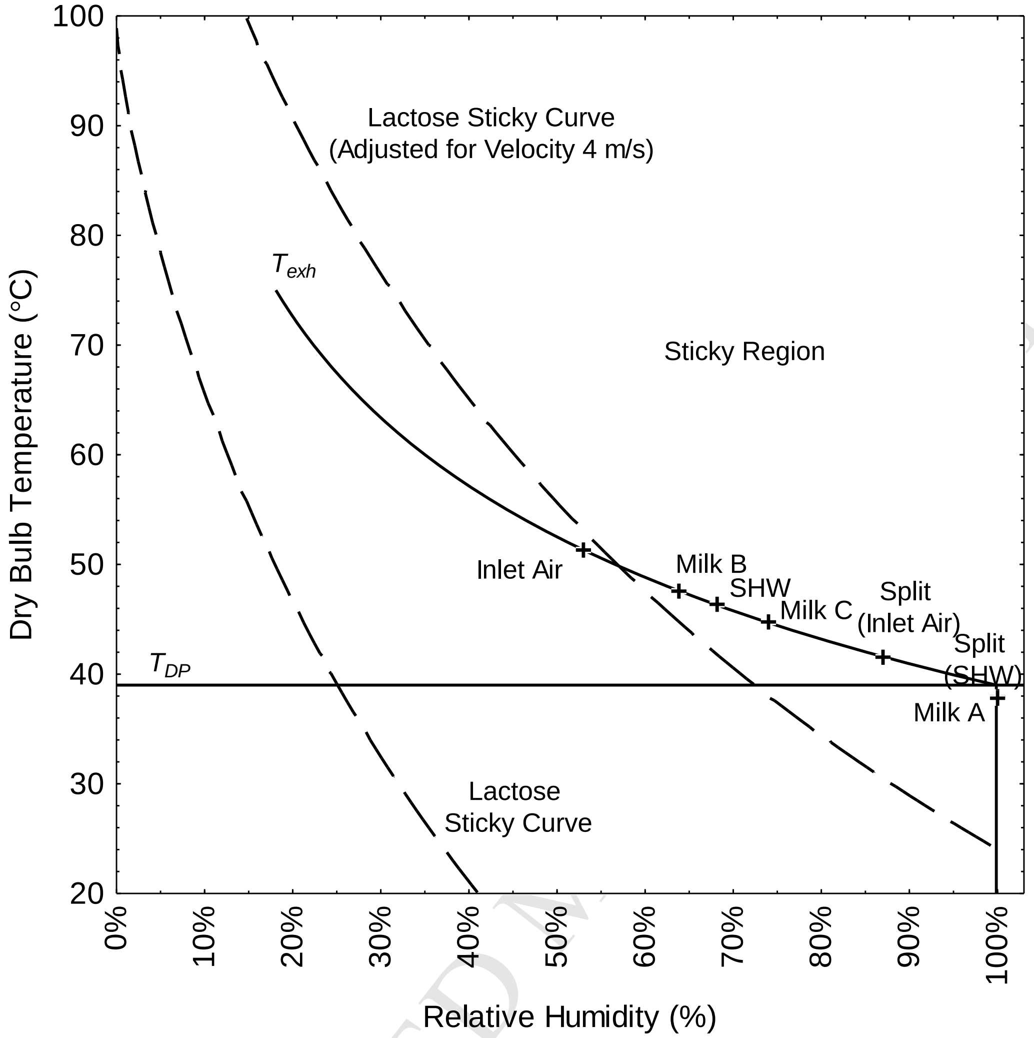

Exhaust to Inlet Air
CF-7.34 Finned Tubes
v_{inlet}=2.5 m/s
Q_{max}=5,956 kW
Ave. Exhaust HX
Face Velocity
1 m/s
2 m/s
4 m/s
Overall Effectiveness ε_o
Temperature Increase ΔT_{cold} (°C)
Loop Mass Flow Rate (kg/s)
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
0
10
20
30
40
50
60
70
0
10
20
30
40
50



Finned Tube	Tube Diameter (mm)	Fin Pitch (fins/m)	Heat Transfer Area / Total Volume (m^2/m^3)	Fin Height (mm)
CF 7.34	9.7	289	459	6.9
CF 7.0-5/8J	16.4	276	269	6.1
CF 8.7-5/8J	16.4	343	324	6.1
CF 8.8-1.0J	25.0	346	299	9.1



Exhaust to SHW
CF-7.34 Finned Tubes
PHX
v_{exh}=4 m/s
Q_{max}=7,524 kW
Q_{SHW}=5,016 kW
PHX Channel Velocity
0.25 m/s
0.5 m/s
1.0 m/s
Overall Effectiveness ε_o
Loop Mass Flow Rate (kg/s)
0.0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
0
20
40
60
80
100



Lactose Sticky Curve
(Adjusted for Velocity 4 m/s)
T_{exh}
Sticky Region
Inlet Air
Milk B
SHW
Milk C
Split
(Inlet Air)
Split
(SHW)
T_{DP}
Milk A
Lactose
Sticky Curve
Dry Bulb Temperature (°C)
Relative Humidity (%)
100
90
80
70
60
50
40
30
20
0%
10%
20%
30%
40%
50%
60%
70%
80%
90%
100%