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Integration Potential of Milk Powder Plants using Conventional Heat Recovery Options

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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 are investigated. Indirect heat transfer via a coupled loop between the spray dryer exhaust and various heat sinks are modeled and the practical heat recovery potential determined. Hot utility use can be 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.

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 (Baker and McKenzie, 2005). 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. Kemp (2005) 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 (2005) 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, 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. 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 (Miller, 1987; Donhowe et al., 1989; Krokida and Bisharat, 2004). As a result typical milk powder spray dryers are not integrated whatsoever due to three main factors: a)

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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. There are limited heat sinks if the dryer is considered in isolation and the feasible heat recovery may be severely constrained. The average inlet air temperature 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. Several possible integration opportunities using indirect liquid-coupled heat exchange from the exhaust of a milk powder spray drying will be investigated in this paper.

2. Model Powder Plant and Process Description

A schematic of the process, including inlet air pre-heat option, is shown in Figure 1 and the stream data for the process are summarised in Table 1. 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. 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. The exhaust is the combined air from the main drying chamber and the fluidised beds and exits 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. The cyclones and baghouse filter attempt to remove any entrained powder. The Site Hot Water (SHW) is used throughout the site for Cleaning In Place operations (CIP) etc. The pinch temperature is 49 °C for 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. Most of the cooling demand is actually cooling the exhaust therefore most of the cooling is supplied by the ambient air and is not an essential cooling demand. Steam is the hot utility.

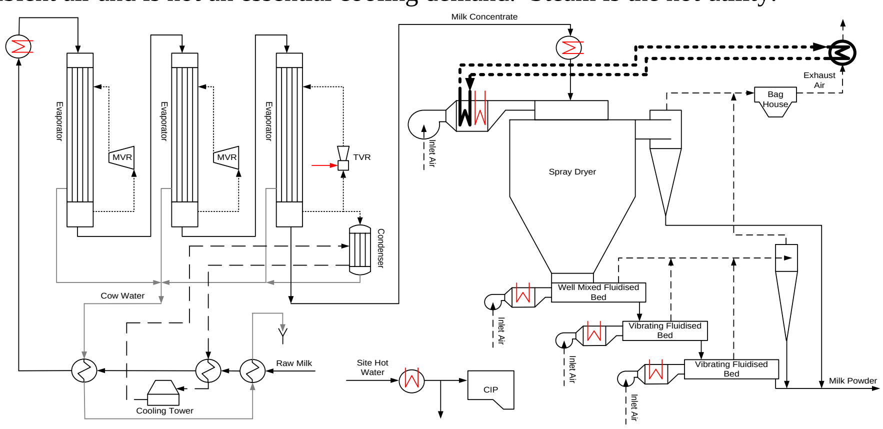

Figure 1: Milk powder plant schematic with indirect inlet air pre-heat (highlighted).

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

Stream Name	*Type*	T_s *(°C)*	T_t *(°C)*	mc_p *(kW/°C)*	*Absolute Humidity* $(kg_{water}/kg_{dry\ air})$
Raw Milk	Cold	10	75	280.0	-
Cow Water	Hot	64	20	245.9	-
TVR Condenser	Hot	54	53	2388.2	-
Milk Concentrate	Cold	54	65	37.6	-
Main Air Inlet	Cold	25	200	119.2	0.0065
Well Mixed Air Inlet	Cold	25	50	10.2	0.0065
VF1 Air Inlet	Cold	25	45	14.9	0.0065
VF2 Air Inlet	Cold	25	32	11.2	0.0065
Air Exhaust	Hot	75	20	174.7	0.0471
Site Hot Water	Cold	15	55	125.4	-

2.1 Coupled Loop and Heat Exchanger Methodology

To determine the performance of the heat recovery heat exchangers on the coupled loop the ε-NTU approach and the methodology outlined in Kays and London (1998) for finned-tube exchangers was used. For the plate heat exchangers (PHX) the ε-NTU approach and the method outlined in Shah & Sekulić (2003) was used. The dimensions of both heat exchangers were fixed depending on the desired average face velocity and volume. 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 (1998). 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 (Shah and Sekulić, 2003) 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 (2003) and pure counter-current flow was assumed. 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 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 was varied until the duties were equal. Overall effectiveness (ε_o) was calculated based on the exhaust and cold stream.

3. Heat Recovery Schemes

Six indirect heat recovery schemes were modeled including pre-heating: a) inlet air; b) SHW; and c) milk. The obvious scheme is to pre-heat the inlet air (Figure 1). The ε_o, temperature increase (ΔT_{cold}) of the inlet air and water side pressure drop (ΔP) is shown in Figure 2. The ΔT_{cold} using different circular finned-tubes is also shown in Figure 2.

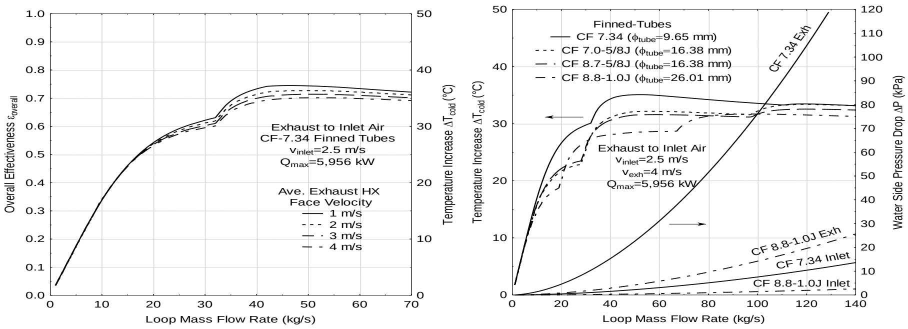

Figure 2: ε_o and ΔT_{cold} (left) ΔT_{cold} and ΔP (right) for a range of loop mass flow rates.

There is a trade-off between maximum heat recovery and heat exchanger size because a high ε_o is achieved with a much smaller heat exchanger (i.e. high face velocity). The water side pressure drop across both exchangers and the air side pressure drop for all of the finned-tubes considered here is not prohibitive at the flow rate. It is clear from Figure 2 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 with increasing the fin pitch, although increasing the fin pitch will have a detrimental affect powder deposition. The increased heat transfer with the smaller diameter tubes is due to the increased surface area per unit volume.

One of the disadvantages of matching the inlet air stream with the exhaust is that the maximum temperature difference is limited to 50 °C. If the hot exhaust is matched to the SHW stream as the heat sink 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 Figure 3 for three average channel velocities. The SHW temperature increase and pressure drop through the loop side of the PHX and the water side of the exhaust exchanger as the flow rate is increased is shown on the right of Figure 3.

The heat recovered from the exhaust could also be used to pre-heat the incoming milk stream, although this would involve re-examining how the CW is used for heating. Milk Option A is to substitute CW heating with the coupled loop and CW to pre-heat the incoming inlet air and 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 7,526.5 kW; however the load on the raw milk heater is increased by around 3,300 kW.

Milk Option B, is similar to Option A, but uses the CW as the first pre-heat to the raw milk after the inlet air pre-heat. The situation is worse because the amount recovered from the exhaust is decreased and there is no heat recovery from the condenser.

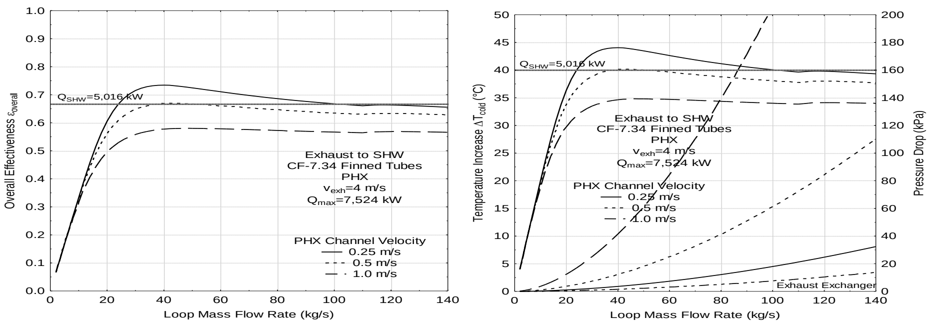

Figure 3: ε_o(left) ΔT_{cold} and PHX ΔP (right) for a range of loop mass flow rates.

Milk Option C involves mass integration of the CW to provide both heating and cooling before being discharged to drain. The advantage to this approach is that it eliminates the need for cooling tower and reduces the load on the raw milk heater and allows the all of the heat from the condenser to be used to pre-heat the milk and SHW.

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 around 20 %. 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. A summary of utility savings for each scheme is shown in Table 2.

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

Case Name	*Q_{hot} (kW)*	*Savings Q_{hot} (%)*	*Q_{cold} (kW)*	*Savings Q_{cold} (%)*
Base Case	32,692.8	-	788.2	-
Inlet Air	28,523.9	12.8	788.2	0
SHW	27,676.8	15.3	788.2	0
Raw Milk A	28,166.7	13.8	1329.2	-68.6
Raw Milk B	29,520.8	9.7	2,388.2	-203
Raw Milk C (Mass Int)	25,733.2	21.3	0.0	100
Split Exh, Inlet Air + SHW	25,892.6	20.8	788.2	0

4. Exhaust Heat Exchanger Fouling

One of the greatest obstacles to the viability of heat recovery from the exhaust is the problem of fouling due to powder deposition. Baghouse filters have collection efficiencies of between 99.5 – 99.98 % (Gabites et al., 2007). he particle or fines loading is highly dependent on the milk powder product being produced and can range anywhere from 40 – 90 %. 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, and velocity. The sticky curve for pure lactose and an adjusted sticky curve for velocity are shown on the left in Figure 4. 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 expected that for all the schemes, except for the inlet air scheme, powder will deposit somewhat on the tubes and fins. Adequate CIP systems are needed to ensure continuous dyer operation.

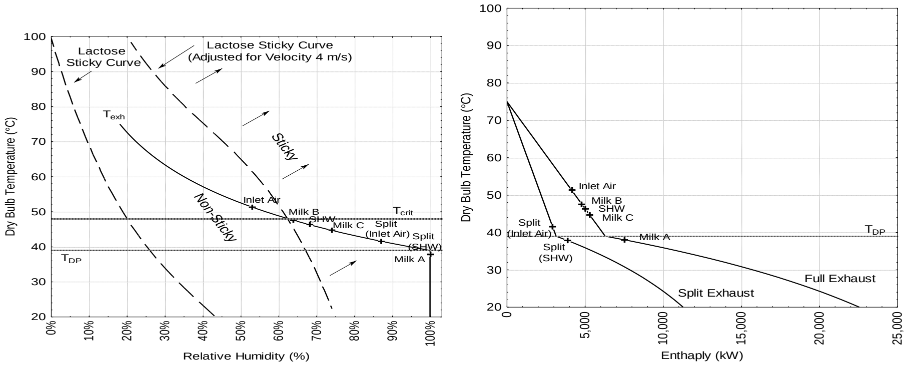

Figure 4: Sticky curve for Lactose (left) and exhaust outlet temperatures (right).

5. Conclusions

Heat recovery from a milk powder spray dryer can reduce hot utility usage by up to 21 % if the appropriate heat sinks are selected. Furthermore splitting the exhaust stream and matching it with two sinks increased the amount of heat recovery potentials. The problem of fouling in the exhaust heat exchanger is perhaps the greatest obstacle for implementation and adequate CIP systems are required.

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