DESIGN AND OPERATION METHODS FOR BETTER PERFORMING HEAT RECOVERY LOOPS

M. R. W. Walmsley*, T. G. Walmsley, M. J. Atkins & J. R. Neale
Energy Research Centre, School of Engineering, University of Waikato, Hamilton, New Zealand
* m.walmsley@waikato.ac.nz

ABSTRACT

Inter-plant integration via a heat recovery loop (HRL) is an economic method for increasing total site process energy efficiency of semi-continuous processes. Results show that both the constant storage temperature approach and variable storage temperature approach have merit. Depending on the mix of source and sink streams attached, it may be advantageous to change the operation of an existing HRL from a constant temperature storage to a variable temperature storage. To realise the full benefits of this change in operation, a redistribution of the existing heat exchanger area may be needed.

INTRODUCTION

The effective process integration of independent semi-continuous plants clustered on a single site is a challenging unsteady-state design problem. Overall site pinch analysis gives energy targets that are rarely met in practice. Other analysis methods involving a combination of direct zonal or intra-plant integration and indirect inter-plant integration are needed to understand what heat recovery is feasible and achievable. Direct intra-plant integration is relatively easy since streams to be cooled (sources) and streams to be heated (sinks) tend to be available at the same time and standard steady-state pinch analysis can be applied. Inter-plant integration is complicated by the stop/start nature of semi-continuous processes and the potential distance between the streams. In this case feasible source and sink matches from different plants may be viable from a thermodynamic point of view, but from a practical point of view matches may not be economic due to limited levels of stream availability, or because of distance.

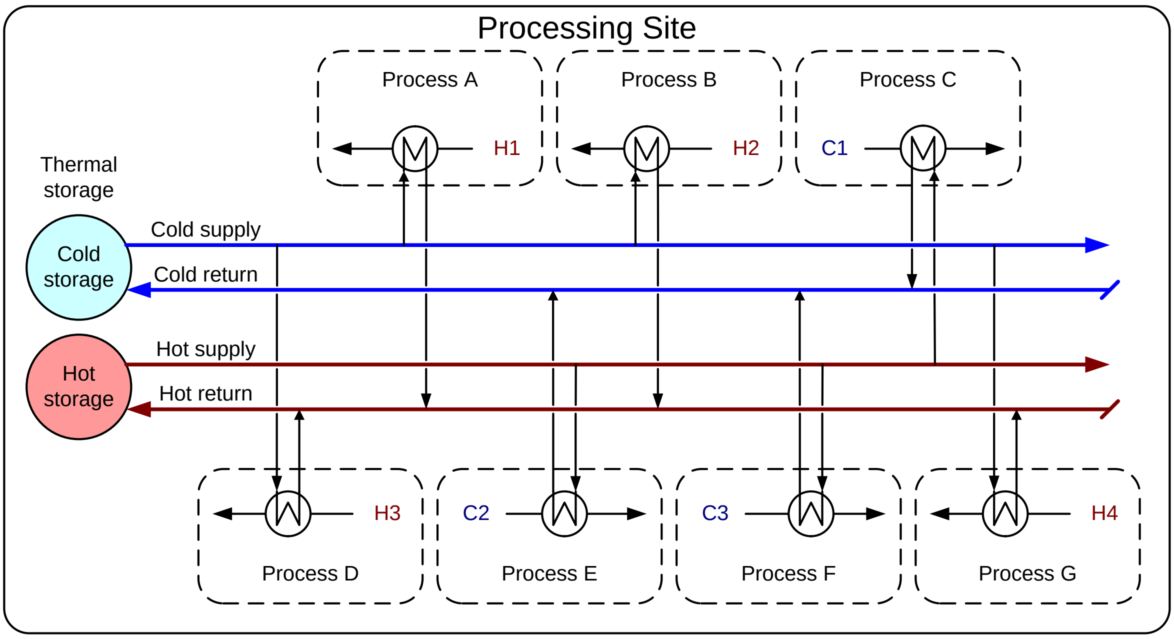


Fig. 1: Heat recovery loop network schematic.

For a site with clusters of low temperature semi-continuous processes, as in many food and beverage factories, an effective indirect heat integration approach is a Heat Recovery Loop (HRL). Excess available heat from one plant may be transferred to another plant using an intermediate fluid, usually water, and additional heat exchangers in a HRL system (Fig 1). The intermediate fluid is stored to successfully meet the time dependent nature of the source and sink streams. Typically hot and cold storage temperatures are fixed and the source and sink streams heat and cool the intermediate fluid between two storage temperature levels.

Several recent papers have considered the application of a HRL to large dairy processing sites. Atkins *et al.* (2010) demonstrated the importance of selecting the HRL storage temperatures when targeting for a particular ΔT_{min} to maximise heating and cooling utility savings. The optimal operational temperatures of the storage tanks were shown to vary significantly at different times of the year depending on what processes are in production. Production schedules in the dairy industry are strongly linked to the milk supply throughout the year. During peak milk supply all plants are running, and as milk supply decreases plants come off line depending on the mix of products needed. Plants also come off line for regular cleaning and for product grade changes, which all add to the variability of the HRL operation.

To maximise indirect heat recovery in the face of plant disruptions and capital constraints thermal storage is an essential variable to optimise. The sizing of the storage tanks is best determined using stream histories, so the trade-off between storage capacity and heat recovery is economically optimised (Atkins *et al.* 2012). An approach to minimising the total HRL heat exchanger area, while maintaining maximum indirect heat recovery, has also been recently demonstrated (Walmsley *et al.*, 2012).

An alternate way of running a HRL is to allow the temperature of the intermediate fluid to vary around the loop and in the storage tanks. With this approach, the intermediate fluid flow rate is controlled to give an outlet temperature that is ΔT_{min} from the supply temperature of each source or sink stream on the loop. The hot storage temperature is the mixed temperature of all the hot return streams, and the cold storage temperature is the mixed temperature of all the cold return streams (Fig 1). Over time the storage tank temperature and volume both vary depending on the thermal loads on the loop and the variability of the streams.

The variable temperature approach has not been widely applied to HRLs, even though the possibility exists for improvements in indirect heat recovery from a simple operational change. A comparison of the constant and variable temperature storage approach is made using a hypothetical set of stream data. A spreadsheet tool is used to simulate HRL performance for given heat exchanger areas and storage temperature operations. Focus in the model is given to the amount of utility consumed when a stream falls short of its target temperature. The estimated annual utility and capital costs for various HRL design and operation methodologies are reported.

HEAT RECOVERY LOOP NETWORK DESIGN AND MODELLING

Steady-state HRL design

A graphical composite curve approach lays the foundation for an insight based method for designing a HRL under steady-state conditions. After zonal or intra-plant integration streams that still require hot or cold utility are potentially suitable for inter-plant integration. Daily time averaged stream data can be used to draw hot and cold composite curves that show the long term average heating and cooling enthalpy deficits in each temperature range. These curves can then be shifted together until a pinch occurs at a limiting stream and HRL storage temperature, to identify the indirect heat recovery potential and minimum utility (Fig. 2).

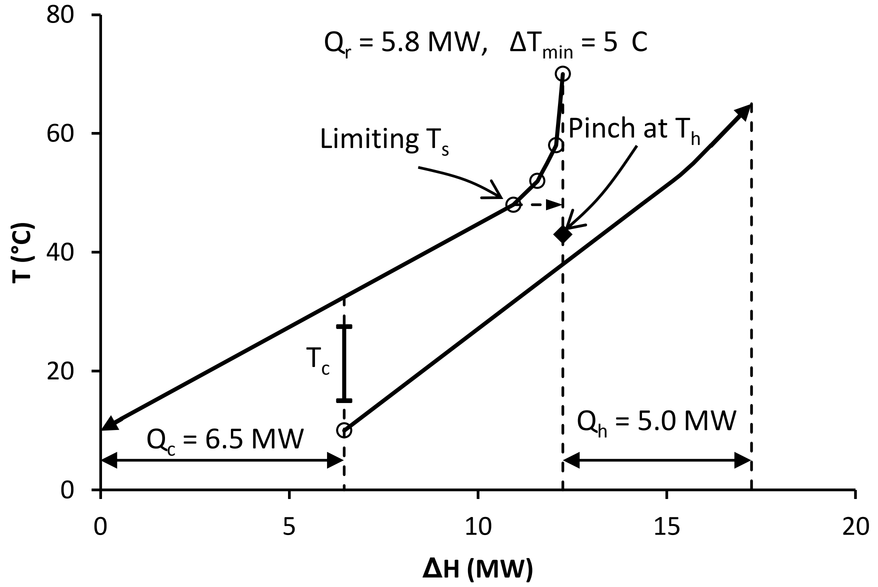


Fig. 2: Inter-plant composite curves for indirect heat recovery using a HRL.

The hot and cold storage temperatures can also be determined directly from the pinched composite curves, and a sloped line drawn to span the overlapping heat recovery region represents the recommended heat capacity flow rate of the HRL. A full minimum ΔT for both the hot and cold curves to the HRL line is used. The traditional concept of a pinch between process composite curves is adapted for the HRL idea; where a pinch usually occurs between the limiting supply temperature and one of the composite curves. At the HRL pinch point, the storage temperature, (T_h) is fixed and the other storage temperature (T_c) can varied within a small range.

Targets obtained from composite curves based on time averaged stream data represent the long-term average heat recovery. The targets assume intermediate fluid storage is always available, which is not always the case in practice. Composites curves based on typical plant operating values, (i.e. design values) may also be useful in understanding the real time balance between sources and sinks. Time averaged data is typically lower than the design values, therefore determining heat recovery targets from design flows will over predict what can be recovered.

After identifying the best HRL storage temperatures for maximising heat recovery, heat exchanger area targets are calculated and optimised using the steady state design flow stream data. As shown one storage temperature level at the pinch point is fixed, while the other may be slightly varied without affecting overall heat recovery. This one degree of freedom can be used to the designer's advantage to minimise the amount of heat exchanger area.

It is recommended that area sizing of the HRL loop exchangers is based on the design flow rates of each of the process streams. To achieve the maximum heat recovery, exchangers are required to deliver the design point duty and, therefore, must have sufficient area to accommodate the design flow rate. For flow rates below the design point, closer approach temperatures are likely to occur that extract or replace more heat.

Operation and control of thermal storage

HRL storage operation and control strategies are of two general types: (1) constant temperature storage (conventional) and (2) variable temperature storage. Whether the storage temperature is constant or variable, is dependent on the heat exchanger control. Take for example a simple feedback control loop that measures the outlet temperature of the loop stream and adjusts its flow rate so that the measured value and set point are the same. In the case of a constant storage temperature, it would require the set point of the control loops to be the same as the storage temperature. The storage fluid and loop fluid are mixed isothermally. Whereas loop fluid entering the storage tanks in the variable temperature method purposely have different loop temperature set point values for the various streams.

The amount of heat recovered is often constrained by the storage temperatures. It is normally advantageous to increase the difference in temperature between the hot and cold storage. This allows hot streams to cool to lower temperatures and cold streams to reach higher temperatures, and as a result allows for greater heat recovery. Conventional wisdom would indicate that moving from a constant storage temperature to variable storage temperature results in a large area penalty due to lower temperature driving forces and the downgrading of higher temperature quality fluid through non-isothermal mixing. However these effects may be offset by a potential for greater difference in average storage temperatures and overall heat recovery. This potential arises from the constant storage temperature methodology being limited by the highest cold process stream supply temperature and lowest hot process stream supply temperature.

When hot or cold storage is running out, process streams may bypass HRL exchangers to recovery or use less heat, effectively shifting the duty to the subsequent utility exchangers. A second option is to use utility to transfer mass between the hot and cold storage tanks. For both methods, the increase in utility consumption is the same.

Modelling heat recovery loop performance

An Excel^{TM} based spreadsheet tool has been developed to simulate the performance of a HRL. The tool uses the storage temperatures and heat exchanger areas targeted from a steady state design to step-wise calculate the level and temperature of the hot and cold storage tanks. The model is an extension of the method presented by Atkins *et al.* (2012). With representative stream data, the model may be applied to estimate actual heat recovery potential. When a stream falls short of its target temperature, utility is consumed.

The model calculates thousands of simple counter-current heat exchanger problems. Each problem has an unknown loop heat flow rate (CP_L), process stream outlet temperature ($T_{P,2}$) and heat duty (Q). Loop temperatures, T_{L1} and T_{L2}, are defined by the average storage temperature from the previous time step and the storage temperature operation mode. Given a heat exchanger area (A) and overall heat transfer coefficient (U), the heat exchanger problems become fully defined. However to calculate the unknowns neither the Log-Mean-Temperature-Difference (LMTD) or the effectiveness-Number of Transfer Units (ε-NTU) method may be directly applied. The LMTD method requires the temperatures in and out of the heat exchanger to be defined; whereas the ε-NTU method needs both heat capacity flow rates (CP) to be known. Hence an iterative approach was implemented and a generalised solutions table (2000 x 2000) was generated using the simple heat exchanger model. Looking up the solution on the table then enabled the model to solve quickly (<10 s) avoiding the need to iteratively solving over 7000 heat exchanger problems (>1 h).

Fluctuations in process stream flow rates (and temperature), which are characteristic of semi-continuous processes, are successfully accounted for in the model. Heat exchanger areas are sized according to the design point flow rate, which is typically the maximum flow rate of the process stream. When the flow rate of a stream falls below its maximum, U and Q are reduced. U is calculated from individual film coefficients (h) for the process and loop streams, which is a function of Reynolds number (Re). Assuming the fluids have a constant viscosity, density, and heat capacity, the ratio of the instantaneous h to the design h_{dp} is related to the ratio of CP through the Reynolds number, where A and B are constants specific to a heat exchangers type and design,

$$h = A \cdot \mathrm{Re}^{B} \Rightarrow \frac{h}{h_{dp}} = \left(\frac{\mathrm{Re}}{\mathrm{Re}_{dp}}\right)^{B} \cong \left(\frac{CP}{CP_{dp}}\right)^{B} \tag{1}$$

The model uses a value of 0.8 for B (Kakaç and Liu, 2002). Again, to avoid an iterative solution, a value for h of the loop side of the heat exchangers was required without first knowing the loop CP. As a result, the loop side flow rate was approximated by,

$$\frac{CP_{l}}{CP_{P}} \cong \frac{CP_{l,dp}}{CP_{P,dp}} \tag{2}$$

The difference between the estimated and calculated loop CP values was found to be at most 3 %. Changes in the stream temperature have not been included in the analysis, although the model has the capability to allow for such changes to occur.

Capital cost estimation

The annualised total cost function of a HRL, C_T, is composed of a utility cost C_U, heat exchanger cost C_{HE}, tank cost C_{Tank}, piping cost C_{pipe} and pumping cost,

$$C_{T} = C_{U} + C_{HE} + C_{Tank} + C_{Piping} + C_{Pumping} \tag{3}$$

Capital costs are amortized on a yearly basis using a life expectancy of 10 years and discount rate of 10 %. In the analysis the tank capacity is set at 500 m^3. It is assumed that the costs of piping and pumping are similar for networks with the same number of heat exchangers. C_{Tank} is also constant for all networks because its capacity is fixed. As a result the minimised C_T is dependent on the sum of C_U and C_{HE} also being a minimum.

A stainless steel gasket plate heat exchanger cost function adapted from Bouman *et al.* (2005) is presented as Eq.4. A Lang factor of 2.5 has been already included in Eq. 4.

$$C_{HE} = 984A + 9255 \tag{4}$$

HEAT RECOVERY LOOP EXAMPLE PROBLEM

Stream data and utility demand

Stream and utility data of a large low temperature food processing site with multiple independent processes similar to Fig 1 are presented in Tables 1 and 2. Heat flow rate (CP) and stream duty (Q) data are given as daily time averaged values and design point values. The time average values are calculated from the stream history over a normal days production. Over the year the plant is in operation for 5000 hours. Some plants and streams are not available continuously throughout a production day and where this occurs it is seen as a large difference between the average and design point values. Stream variability and stream availability therefore cause heating and cooling duties to vary considerably within each plant and across the combined site. This is demonstrated in Fig 3 for the modelled data over a three day period.

Tab. 1: Process stream data.

Stream	Type	T_s (°C)	T_t (°C)	CP_{ave} (kW/°C)	Q_{ave} (kW)	CP_{dp} (kW/°C)	Q_{dp} (kW)
H1	Hot	52	10	78	3263	96	4032
H2	Hot	48	10	129	4893	189	7165
H3	Hot	58	10	67	3222	77	3686
H4	Hot	70	12	15	871	19	1079
C1	Cold	10	65	154	8448	219	12070
C2	Cold	10	55	22	988	27	1224
C3	Cold	10	53	31	1351	92	3959

Tab. 2: Utility data.

Utility	Type	T_s (°C)	T_t (°C)	Cost ($/MW)
Steam	Hot	220	219	45
Cooling water	Cold	15	25	5
Chilled water	Cold	1	6	40

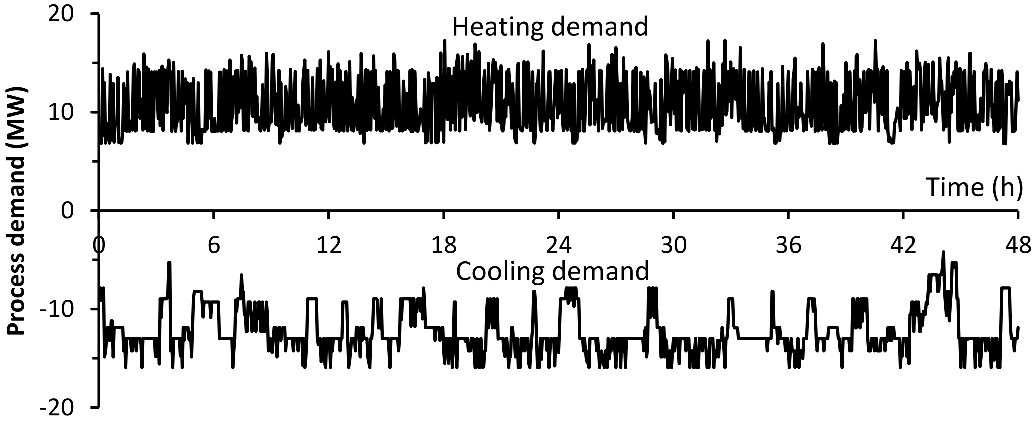


Fig. 3: Total site process heating and cooling demand.

Fig. 4a plots the time averaged composite curve showing the average utility targets; whereas Fig. 4b plots the design point composite curves showing the utility targets when all streams are running at the design operating conditions.

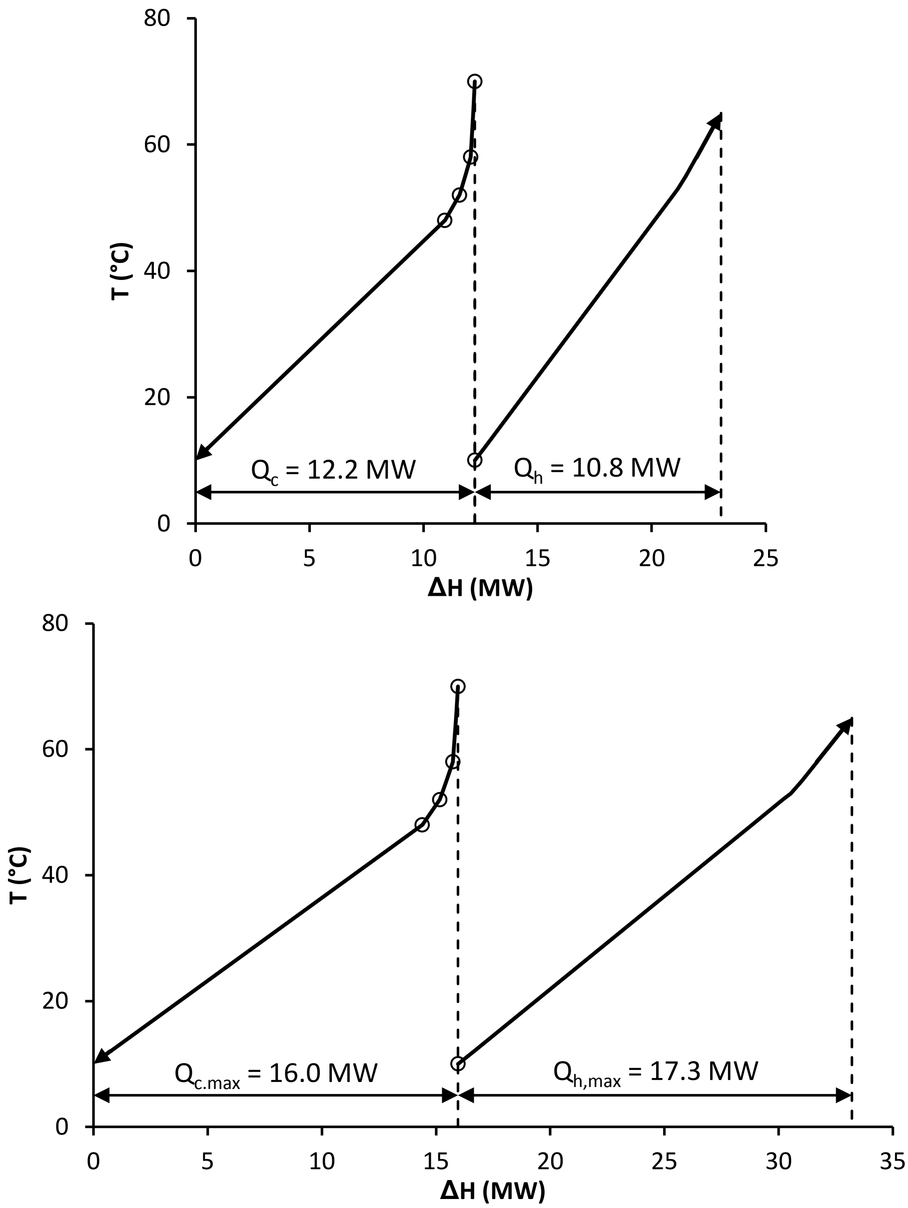


Fig. 4: Composite curves showing stream supply temperatures, (a) time averaged and (b) design point.

HRL design and operation conditions assessed

Four methodologies for operating and designing a HRL have been considered. Two methods (A and B) follow the conventional constant temperature storage control strategy; whereas the other two methods (C and D) use a variable temperature storage control idea. Distinguishing design features of each method are described.

HRL design and operation methods:

A. Storage temperatures are constant. Heat exchangers are sized based on vertical integration using a time averaged composite curve. Storage temperatures are selected to maximise heat recovery for a given ΔT_{min} while minimising the area.
B. Same as A, except heat exchangers on the non-limiting (hot) side of the HRL are sized to exchange as much heat as possible without violating the ΔT_{min} constraint.
C. Storage temperatures are variable and are set by a long-term average. Heat exchanger areas are sized so that the difference between process supply temperatures and the storage temperature is a constant ΔT_{min} apart.

D. Same as C, except the ΔT_{min} applied to the hot and cold supply temperatures is different.

Methods A and B are conventional design methodologies. The advantage of Method A is heat exchangers are not oversized to provide a surplus of heating or cooling. The expected result, therefore, is a long term balanced load on a HRL. But a significant assumption of this method is that storage is always available, which may not be practical if imbalances are sustained for long periods of time and the required storage is very large. If storage runs out at any time, less heat will be recovered.

Method B allows for heat exchangers on one side of a HRL (the non-limiting composite curve) to be over designed without violating the ΔT_{min} condition. Over designed exchangers may be able to deliver additional heating or cooling loads to satisfy momentary imbalances and prevent storage from draining. In the long-run, a natural imbalance occurs because the over-designed side of the HRL will deliver more load on average than the over side of the HRL. An advantage to this approach is the amount of storage can be much less than in Method A, while still achieving peak heat recovery. The trade-off is between adding heat exchanger area for reducing storage capacity and capital cost.

Methods C and D both use a variable storage temperature approach. In method C, heat exchangers are sized and controlled to have the loop fluid exit one ΔT_{min} away from the process streams supply temperature. As a result, loop fluid entering the storage tanks may be of different temperature. The long-term average temperature of the storage is estimated from the time averaged stream data. Method D differs from C by having a different ΔT_{min} around the hot and cold storage temperatures. Results are reported for a cold side ΔT_{min} double the hot side ΔT_{min}. Several combinations of hot and cold ΔT_{min} were calculated with the best ratio of hot to cold ΔT_{min} being reported.

The effect of changing the tank storage capacity is not considered. Results are based on using hot and cold tanks of 500 m^3 each. The intermediate fluid considered is water.

RESULTS AND DISCUSSION

Heat recovery loop performance

Using the Excel^{TM} based spreadsheet tool and the variable stream data (Table 1 and Fig. 3) the performance of the HRL can be modelled. A sample of two modelling results using design methods A and D are presented as Fig. 5a and b, respectively.

The average heat recovery and the annual utility cost were determined for each of the four HRL design and operation methods (A, B, C and D), and results are summarised in Figs 6 and 7. Each curve is drawn through 34 data points calculated by the computer model by inputting different ΔT_{min} values. After entering a ΔT_{min} into the computer model, heat exchangers are sized and the total heat exchanger network area is summed. The heat recovery and utility costs curves follow a law of diminishing returns to an asymptotic value. Method A (constant temperature storage; vertical integration) provides the highest heat recovery for the majority of the range of network area considered. However there is a significant range (2500 – 4000 m^2) in which D (variable temperature storage; different hot and cold ΔT_{min}) performs best.

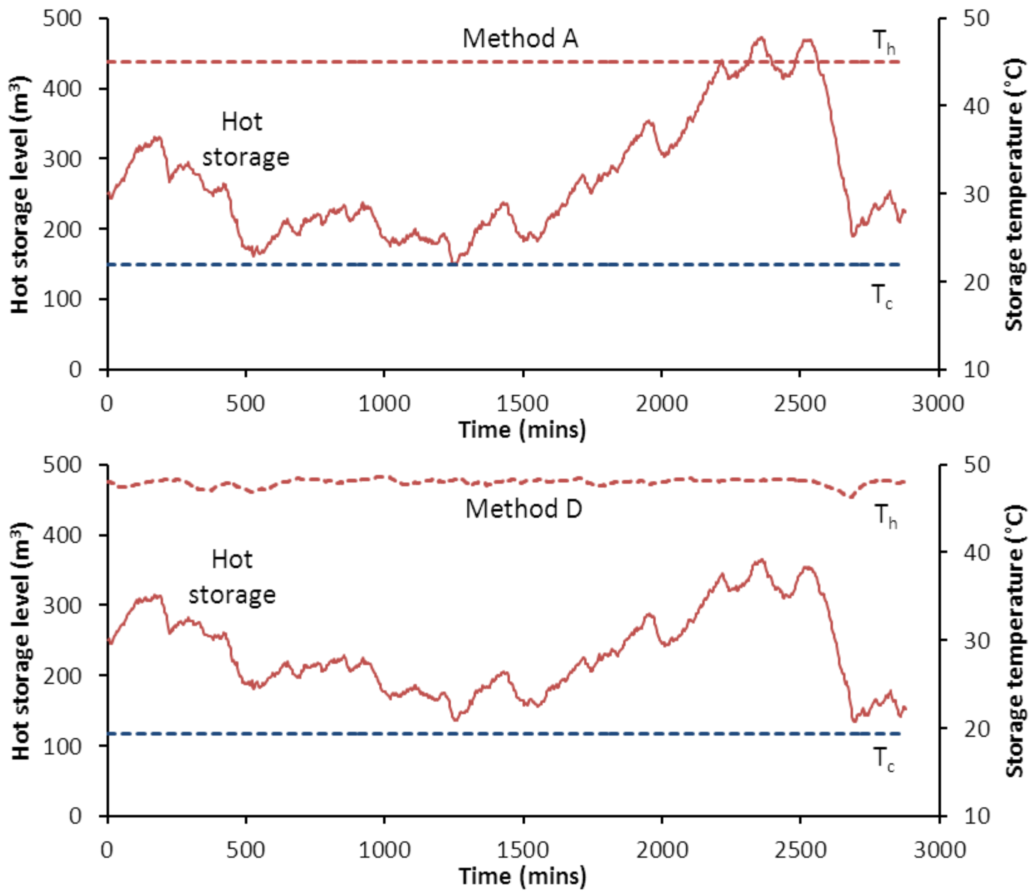


Fig. 5: Predicted HRL performance over the three day period, (a) method A using ΔT_{min} = 3 °C, and (b) method D using ΔT_{min} = 3 °C. A and D have total areas of 2950 m^2.

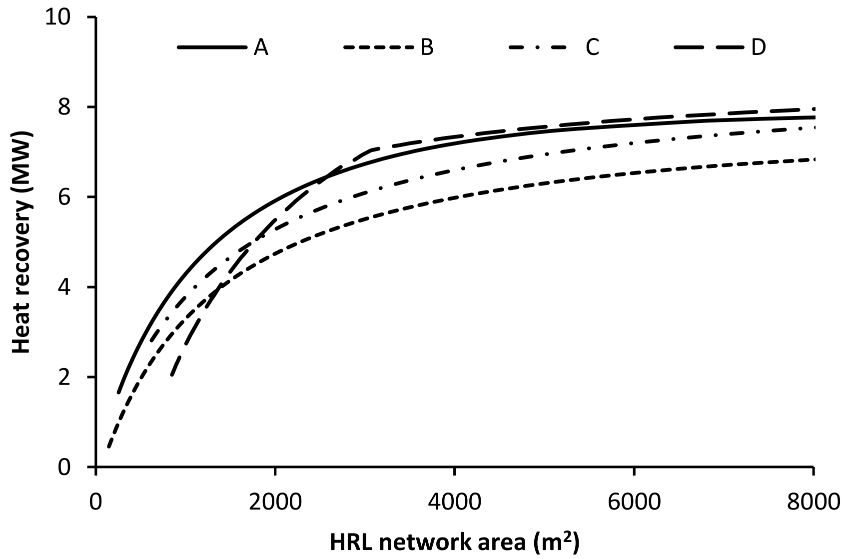


Fig. 6: Effect of HRL network area and HRL control method on average heat recovery.

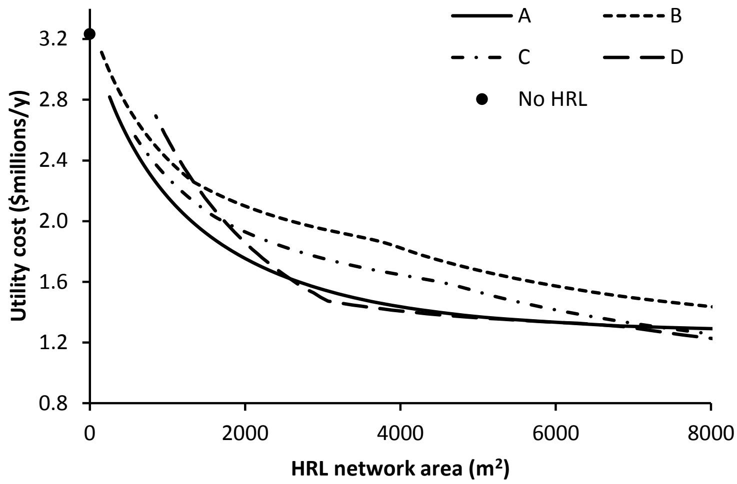


Fig. 7: Effect of HRL network area and HRL control method on total utility costs.

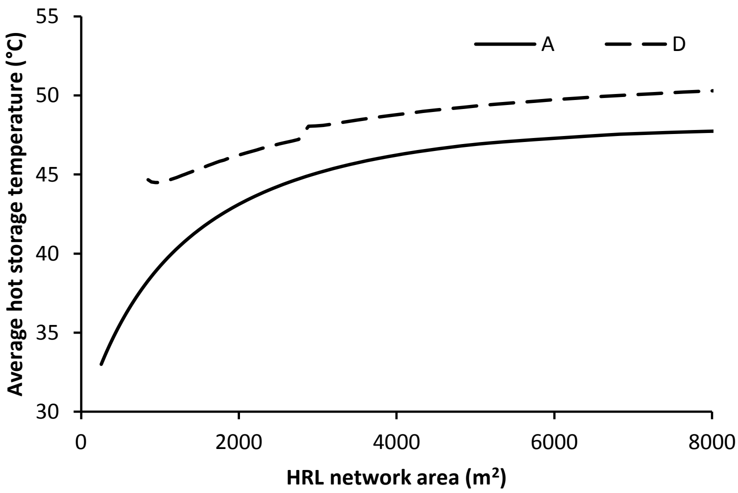


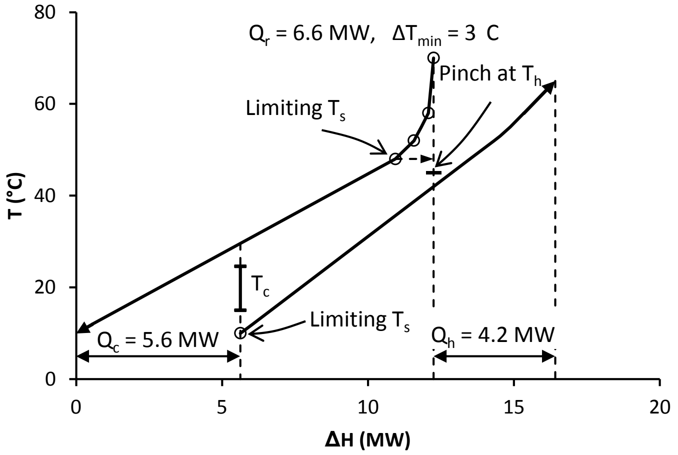
Fig. 8: Hot storage temperature of methods A and D.

Method D is disadvantaged due to the constant ratio of ΔT_{min}(Hot) to ΔT_{min}(Cold) of two. The optimal ratio is dependent on the value of ΔT_{min}. The best performing ratios have been found to occur when the hot and cold sides of the loop are balanced in the long-run. Unfortunately for a constant ratio of ΔT_{min}(Hot) to ΔT_{min}(Cold) the HRL is only well balanced when ΔT_{min}(Hot) ≈ 4.5 °C and ΔT_{min}(Cold) ≈ 9.0 °C. When the loop is imbalanced in the long-run, it is the result of non-optimal area distribution.

Results show method A significantly out-performs C. Method A at times delivers up to 22 % greater heat recovery, or a utility savings of $390 000 per year, than C for the same total exchanger area. In C, some exchangers are purposely over-designed without violating the ΔT_{min}. The result is a loop that will be imbalanced to the side of the loop with over designed exchangers. The true benefit of C is a smaller storage may be able to be installed, while maintaining the maximum heat recovery. Sizing of the storage is not considered in this paper.

Heat recovery is strongly connected to the storage temperature levels. In particular the hot storage temperature can vary to allow for increased heat recovery. The cold storage temperature is relatively constant, for the streams modelled, due to all streams having the same supply temperature. Results from Fig. 6 for methods A and D may also be plotted against the average hot storage temperature across the three day period. Fig. 8 shows method

D has an average hot storage temperature is on average 2 - 3 °C hotter than method A for the same total area. This is important when the hot storage temperature limits heat recovery.



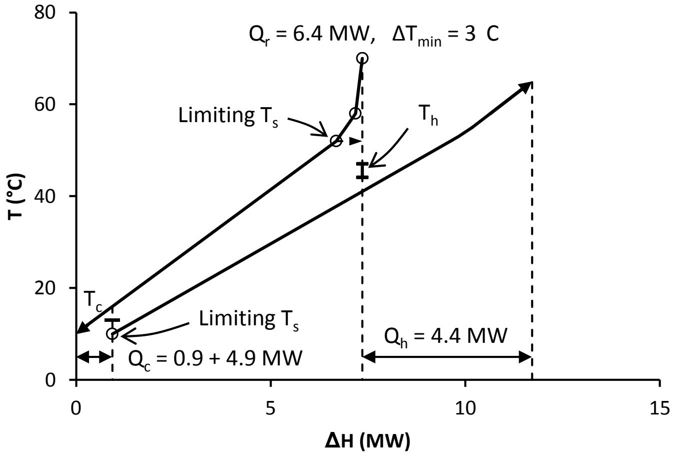


Fig. 9: Process stream composite curves (a) all streams and (b) without H2.

The daily time averaged composite curve in Fig. 9 can aid understanding of why D, a variable storage temperature approach, can recover more heat and normally has a higher average hot storage temperature than A, a constant storage temperature approach. Intuition suggests that mixing fluids of different temperature in a HRL would downgrade the heat and, therefore, lose potential for heat recovery; however this has been shown in Fig. 6 to not always be the case. Fig. 9a shows the composite curve plot of all the process streams recovering on average 6.6 MW for ΔT_{min} of 3 °C. The storage is on the hot side of the loop at 45 °C, with a limiting supply temperature of 48 °C. In this case, the quantity of cold streams limits the amount of heat recovery. This limiting supply temperature is caused by stream H2. If H2 is removed from consideration, the composite curve must be redrawn and re-shifted as in Fig. 9b. Now the storage pinch changes to the cold side of the loop, T_c is fixed, T_h can now vary within a small range, the sources limit heat recovery and the time average heat recovery is now 6.4 MW.

In Fig. 6 a cross over occurs between methods A and D indicating that the higher temperature hot storage in method D is not the only controlling factor of heat recovery. Other factors include the distribution of area, the long-term balance of sources and sinks, and storage capacity.

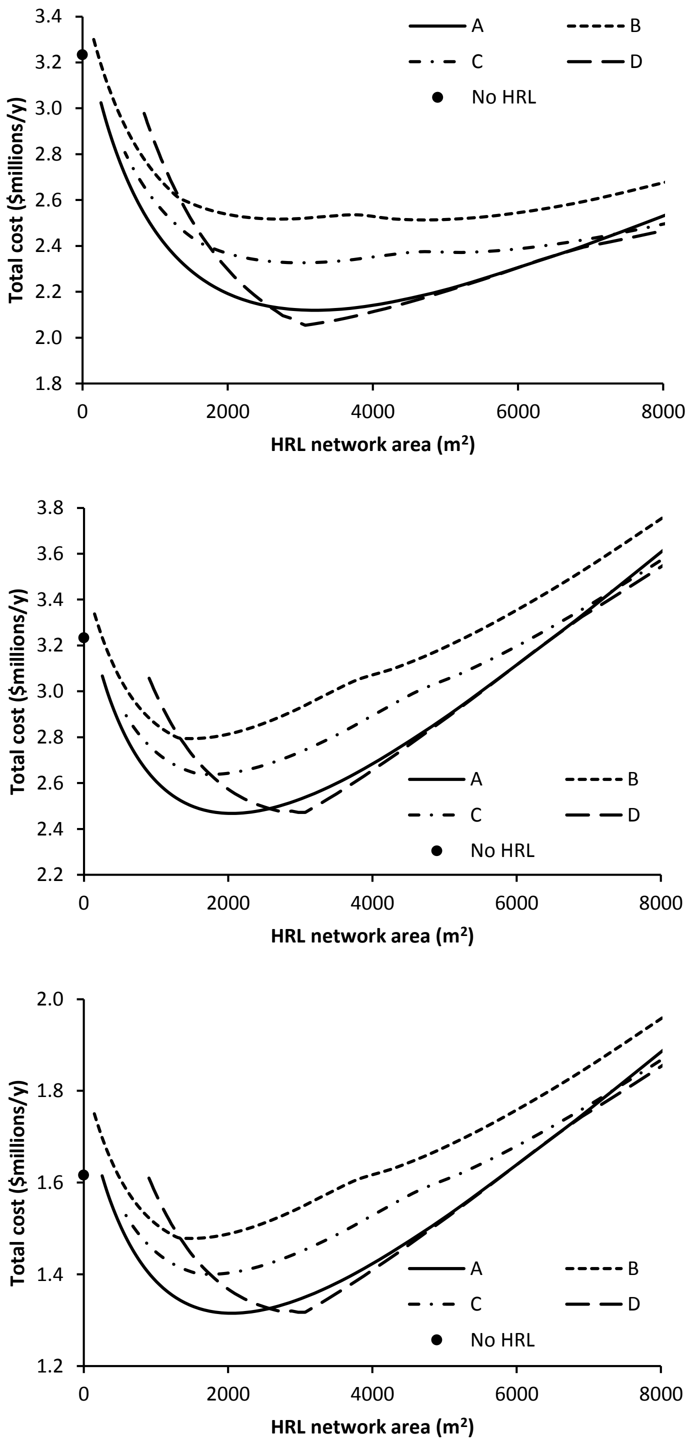


Fig. 10: Total cost versus total network area; (a) cost functions as given, (b) 50 % increase in heat exchanger costs, and (c) 50 % reduction in operating hours.

Heat recovery loop cost

Predicting accurate cost weightings within the total cost function is difficult. To address this issue some cost weighting analysis was carried out and results are presented for three different situations in Fig. 10, namely (a) cost functions as given, (b) 50% increase in heat exchanger costs, and (c) 50% reduction in operating hours. The cost of the storage tanks, piping and pumping were estimated to have a total installed capital cost of $1 million. Fig. 10a shows that for no adjustment to the cost function, the most cost effective method for operating a HRL is method D, the variable temperature storage case, with a minimum total cost of $2.05 million per year. However, method D is only marginally better than method A, the constant temperature storage case, and is significantly worse than A, and ultimately B and C as the network area decreases below 2400 m^2. A 50% increase in heat exchanger costs or a 50% reduction in annual operating hours gives a slight economic advantage to method A at network areas around 2000 m^2, but has little effect on B and C. Therefore, weighting the capital cost component of the total cost more heavily than the heat recovery savings, favours method A compared to method D.

Industrial application

At present, most HRL installed in industry operate using constant temperature storage. Results show that there may be benefit in changing HRL operation to the variable storage temperature approach and modifying the HRL network area to the optimal area range. The optimal area range will be specific to the industrial application and will need to be determined from modelling to ensure there is value in switch operation methods. An operational change could be achieved by changing the control set points of the current temperature control system within the HRL system. To further maximise heat recovery for method D, heat exchanger area may also need to be redistributed between existing heat exchangers. Redistribution of area is simple when plate heat exchangers are used and plates can be added and removed. The effect of redistributing area has not been investigated in this study.

CONCLUSION

Inter-plant indirect heat integration via a HRL is an economic method for increasing process energy efficiency in large processing sites with a low pinch temperature. Results show that both the constant and variable temperature storage approaches to operating a HRL have merit and can be economic. Under some circumstance, it may be advantageous to change the operation of an existing HRL from a constant temperature storage to a variable temperature storage. To realise the full benefits of this change in operation, a redistribution of the existing heat exchanger area may be needed.

REFERENCES

Atkins, M.J., Walmsley, M.R.W., Neale, J.R., 2010. The challenge of integrating non-continuous processes – milk powder plant case study. Journal of Cleaner Production 18, 927–934.

Atkins, M.J., Walmsley, M.R.W., Neale, J.R., 2012. Process integration between individual plants at a large dairy factory by the application of heat recovery loops and transient stream analysis. Journal of Cleaner Production.

Bouman, R.W., Jensen, S.B., Wake, M.L., Earl, W.B., 2005. Process Capital Cost Estimation for New Zealand 2004. Society of Chemical Engineers New Zealand.

Kakaç, S., Liu, H., 2002. Heat exchangers: selection, rating, and thermal design. CRC.

Walmsley, M.R.W., Walmsley, T.G., Atkins, M.J., Neale, J.R., 2012. Area targeting and storage temperature selection for heat recovery loops. Paper to be presented at 15th Conference PRES, Prague, CZ.