Thermal Coupling of Cooling and Heating Systems

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INTRODUCTION

The economics justifying heat pump installations are brutally honest. If the system is too expensive or not efficient enough to compete with conventional heating methods, fancy sales arguments will not convince an owner to buy the system. The additional expense of the heat pump must be weighed against the reduction in operating cost. When the heat source is the discharge from a cooling system, the economics of heat pumps are immediately more attractive.

Refrigeration systems extract heat at temperatures ranging from about 41°F (5°C) in chillers down to –58°F (–50°C) in freezers. In contrast to this range, the heat rejected from the system is in a relatively narrow band. For large systems using evaporative cooling in temperate climates, the design condensing temperature typically will be about 95°F (35°C); in smaller air-cooled systems in hotter climates, this could be as high as 122°F (60°C).

The output from the system is called “waste heat” because at these temperatures there is not much use for it, even in an industrial food factory or facilities such as a hotel, apartment block, hospital or prison. In all of these applications, a large demand exists for heat at slightly higher temperatures, typically ranging from 122°F to 203°F (60°C to 95°C). These heating demands have traditionally been served by burning fossil fuel on site for water heating or for steam raising, or by electrical heating where the electricity has been generated at much higher temperatures.

It does not take complex exergy analysis to realise that a process that burns fossil fuel at about 3,200°F (about 1750°C) to heat something to 203°F (95°C) is capable of improvement. It also does not make sense to install an air source or geothermal heat pump in a facility that is rejecting plentiful amounts of heat to atmosphere at a higher temperature than the air or geothermal source for the heat pump installation.

Harnessing the heat output from cooling systems can be done in several ways. The simplest is to run the plant at a higher heat rejection temperature so that the heat output is at a useful temperature. It is usually necessary to include a means of heat rejection to atmosphere to cover for the times when there is a need for cooling but no demand for heating. The additional complexity required to harness the heat includes an extra condenser, usually heating water or some other heat transfer fluid, and typically a shell and tube vessel or plate heat exchanger.

If the temperature lift in the cooling system is wider than acceptable limits for the compressor, then an additional stage of compression is required. This is more complex because an extra compressor is in the system, but discharge gas from the main plant sometimes can be taken directly to the suction of the high-stage compressor so the heat recovery condenser is the only additional heat exchanger. If this is not feasible (because the cooling and heating circuits need to be kept separate, or the discharge gas from the cooling compressors is too hot to be fed directly to the suction of the high-stage compressor), an additional heat exchanger, either a cascade condenser or a desuperheater, also is required.

Recent developments in ammonia compressors (first in reciprocating machines designed for 580 psig [40 bar] and then in single and twin screw compressors) have enabled heat recovery plants to be designed for all three configurations. Single screw compressors now are available with operating pressures sufficient to heat water to 203°F (95°C), so the full range of sub-steam temperature requirements can be met.

If the required water temperature is above 187°F (75°C) then a steel bodied compressor, rated for 1,125 psig (75 bar) is required, but for lower temperatures a body of spheroidal graphite (SG) iron, rated for 725 psig (50 bar), is sufficient. The steel body is significantly more expensive. In a two-stage heat pump, the segregation of stages should be arranged so that steel bodies are only required for the high-stage compressors.

Significant operational and environmental benefits have been achieved through the installation of a combined heating and cooling system at a confectionery factory in England that manufactures candy bars and chocolate. As with many such facilities, the heating and cooling requirements were previously
handled separately. Cooling was done by a bank of 12 R-22 chillers with a total installed capacity of 1,535 tons (5400 kW) and heating was provided by coal-fired steam boilers delivering 60,000 lb/h (7.5 kg/s) of low pressure steam.

The client was required to replace the equipment because R-22 use was becoming severely restricted, and the coal-fired boilers incurred a significant operating penalty in carbon credits. However, the need to install a new plant was viewed as a rare opportunity to make a significant improvement in site operating cost. The new system was required to provide efficient operation across a wide range of cooling and heating duties and to be flexible in operation, allowing for future changes in site utilisation.

Four alternatives were analysed. The first proposal was to install a modern gas-fired boiler plant with packaged ammonia chillers. The second scheme also used packaged ammonia chillers for cooling, but with a combined heat and power plant providing the electrical power required to run the chillers and delivering the heating requirement. In the third scheme a comprehensive geothermal network provided the heating, again with packaged ammonia chillers. The fourth scheme was the only one to integrate the two requirements thermodynamically in what was called a “thermally coupled” system. This was based on a central plant cooling system, but with additional high-pressure compressors raising the condensing condition sufficiently high enough to meet the heating requirement.

The alternatives were rated according to the energy required, the present value of the system and qualitative factors (items that could not be assessed in terms of energy or value, but still had a bearing on the utility of the proposed solution). The results of the analysis for the project are shown in Table 1.

Table 1: Assessment of alternative schemes

<table>
<thead>
<tr>
<th>Technology</th>
<th>Energy Rating</th>
<th>Life-Cycle Cost</th>
<th>Qualitative Score</th>
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<tbody>
<tr>
<td>Gas Boiler/Ammonia Chillers</td>
<td>5.26</td>
<td>$36.37 Million</td>
<td>15</td>
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<tr>
<td>CHP/Ammonia Chillers</td>
<td>7.87</td>
<td>$34.50 Million</td>
<td>19</td>
</tr>
<tr>
<td>Geothermal/Ammonia Chillers</td>
<td>2.13</td>
<td>$26.73 Million</td>
<td>23</td>
</tr>
<tr>
<td>Thermally Coupled Heating and Cooling</td>
<td>3.93</td>
<td>$31.39 Million</td>
<td>13</td>
</tr>
</tbody>
</table>

The qualitative assessment included consideration of factors such as future flexibility, disruption to site during installation, initial capital cost, environmental impact and operational complexity. A low score indicates that the scheme is attractive.

Although the geothermal system was cost effective over the 15-year life of the plant and consumed the least energy, it was not favoured in the qualitative analysis for several reasons. The required size of an underground collector would have covered most of the site, and it was perceived as a significant restriction on future site use. For example, the collector would have made it difficult to sell a portion of the land for development in the future.

With the geothermal solution ruled out, the thermally coupled heating and cooling system was the most attractive option. It used the least energy of the three alternatives, it had the lowest life-cycle cost, and it scored best in the qualitative analysis. The system was installed early in 2010, and has been fully operational since March. The cooling loads are summarised in Table 2 (Page 20), and are all serviced by a propylene glycol loop that delivers glycol at 32°F (0°C).

The heating loads can be split into three main streams: cooking, cleaning and “closed loop” loads. The cooking requires steam at 260°F (125°C) and so cannot be fed by the heat pump system, but it only accounts for about one third of the total site demand for heating. The cleaning load takes fresh water at 50°F (10°C) and heats it to 196°F (90°C). In the closed loop heating circuits, for example stopping chocolate from solidifying in the machines, the water circulates between 104°F and 140°F (40°C and 60°C).

When the cooling and heating loads were compared it was concluded that the most economic installation would only provide a portion of the heating by coupling with the cooling system. The cooling system was designed with an installed capacity of 853 tons (3000 kW) but with pumps, vessels and pipework sized for a maximum capacity of 1,023 tons (3600 kW). The installed
capacity is less than the total schedule because there is some diversity in the timing of the loads. For example, the air-conditioning load is more significant in the summer and the forming and molding loads are not continuous.

The heating system was designed to accept water at 140°F (60°C) from the heat recovery circuit. The steam for cookers is generated by gas boilers and the top-up heating for the other circuits, from 140°F to 194°F (60°C to 90°C) is provided by the boilers. This reduces the heat demand from the heat recovery system to a total of 1,230 kW, with 40% going to the once-through circuit and 60% going to the closed loop.

With this configuration, half of the heating demand is provided by waste heat from the cooling process, and it is only necessary to boost the condensing condition of the chillers from the standard design of 118°F to 138°F (43°C to 59°C). Heat rejection to atmosphere is through air-cooled condensers designed for a summer ambient of 95°F (35°C).

In cooler weather the discharge pressure reduces to match the ambient conditions, with a minimum condensing condition of 68°F (20°C). When heating is not required by the process, and there is no further storage capacity available, the discharge pressure on the heat recovery part of the system drops back to the lower figure, making the cooling duty as efficient as possible.

The system configuration is shown in Figure 1. Two of the compressors provide the base cooling load, with heat rejection to ambient. When heat recovery is required the other two compressors run at higher discharge pressure, drawing ammonia from the common evaporator on the glycol cooling circuit. Therefore, it is possible, with the two independent expansion valves feeding the gravity-fed receiver, to run the heat recovery circuit at a high discharge pressure and the heat rejection circuit at a much lower discharge pressure. If the cooling demand is high, but there is no requirement for recovered heat, then the heat recovery compressors are able to deliver gas to the air cooled condenser through the bypass valves on the discharge header, and will then operate more efficiently at reduced discharge pressure.

To maximise the efficiency of the heat recovery system the oil separators on the heat recovery compressors are insulated. The compressor units can be seen in Figure 2. The glycol cooling circuit is served by a gravity fed plate heat exchanger, which uses a low ammonia charge to achieve the full cooling and heating requirement. The total ammonia content of the system is 2,600 lbs (1,200 kg), which is equivalent to a charge of 2.9 lb/tons (0.375 kg/kW). The additional charge due to the heat recovery is not significant because the heat recovery part of the circuit uses plate and shell heat exchangers. The main proportion of the ammonia charge is held in the air-cooled condensers and in the gravity feed surge drum to the glycol chiller. In this configuration adding the heat recovery increased the charge by approximately 5%.

The glycol cooling circuit was redesigned from the previous configuration. This reduced the total number of pumps from 20 to four, and introduced variable speed drives to all four pumps. The connected pump load is reduced from 221.5 kW to 155 kW, but the real saving comes from using the variable speed pumps in
Table 4: Projected running cost savings for the project.

<table>
<thead>
<tr>
<th></th>
<th>2008 Actual</th>
<th>2011 Projected</th>
<th>Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy cost</td>
<td>$2,797,470</td>
<td>$1,336,569</td>
<td>$1,460,901</td>
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<tr>
<td>Operating cost</td>
<td>$590,577</td>
<td>$124,332</td>
<td>$466,245</td>
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<tr>
<td>Climate change levy</td>
<td>$512,870</td>
<td>$256,435</td>
<td>$256,435</td>
</tr>
<tr>
<td>Water costs</td>
<td>$111,900</td>
<td>$73,045</td>
<td>$38,854</td>
</tr>
<tr>
<td>Total</td>
<td>$3,900,917</td>
<td>$1,717,336</td>
<td>$2,222,435</td>
</tr>
</tbody>
</table>

The key factor in discovering and exploiting the savings is to look for places where fossil fuel is being burned to provide heat, but the end delivery of the heat is in the 100°F to 200°F (38°C to 93°C) range. In many cases, heat recovery is not considered because the heat distribution is provided by a low pressure steam system and the refrigeration system does not deliver recovered heat at high enough temperatures to raise steam.

However, the losses in the steam delivery system were a significant proportion of the total heating bill. Using steam to distribute heat required at lower temperatures is convenient if the temperatures required to make steam are available from a combustion plant, but it is ultimately an inefficient method of providing low to medium grade heat. Significant operating cost reductions can be achieved by taking some of the heat loads off the steam circuit.

The balance between capital and operating costs was struck at a point that took a significant proportion of the heating requirement as recovered heat from the cooling circuit, but blended this with heat from boilers rather than seeking to meet the total demand from one source. This approach (thermal coupling) is suited to a range of applications, and could be scaled down from the system described here to suit installation in a wide range of industrial and commercial buildings.

CONCLUSIONS

The cost savings in the confectionery factory project were achieved by:

- Improving the coefficient of performance of the chillers;
- Improving the energy efficiency of the cold glycol distribution circuit;
- Exchanging inefficient heating and steam distribution system; and
- Coupling the heating and cooling demands of the building to make use of extracted heat.

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