INDUSTRIAL HEAT PUMPS: CASE STUDIES AND LESSONS LEARNED

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Abstract: Over the last five years a new style of industrial heat pump, using high pressure ammonia to deliver heat efficiently at temperatures up to 90 °C, has been introduced in Europe, with installations in Norway, England, France, Belgium and Switzerland. This was a protracted and difficult birth, with many technical and commercial challenges overcome along the way. The benefits gained from the use of these systems are substantial and it is hoped that the perseverence that has been required in order to deliver efficient and reliable systems will pay off through significant volumes of repeat business now that the concept has been successfully demonstrated over several heating seasons. This paper describes three case study applications, gives details of some of the technical and commercial challenges that have been faced, and considers ways in which the lessons learned can be applied in other market sectors.

Key Words: heat pumps, industrial, efficiency

1 INTRODUCTION

Ammonia has been used as a refrigerant in industrial sized systems for over 140 years and considerable experience has been gained in its safe and efficient use over that long period. It is the only working fluid to have been in continuous use in refrigerating systems since its first introduction in a practial Perkins cycle refrigerator in 1872. Although the concept of using the Perkins cycle to provide useful heat to a process is as old as ammonia refrigeration, and despite the benefits of using ammonia in this application, heat pumps with ammonia have not been very common throughout that period. Prior to the phase out of CFCs mandated by the Montreal Protocol in the mid 1980s R-12 was widely used in industral heat pumps, and for very large systems employing centrifugal compressors R-11 gave excellent efficiency at relatively low cost. The fluids which replaced these CFCs do not offer such attractive properties and so the recent interest in large heat pumps which has been triggered by the combination of increasing energy costs and increasing uncertainty about energy supply has led to the installation of several large systems (greater than 1000 kW capacity) using ammonia as the working fluid. Three of these systems are presented in this paper.

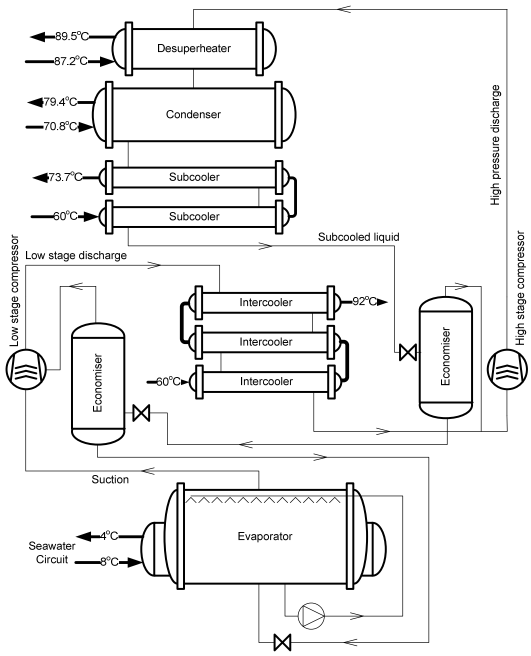
2 THE BENEFITS OF AMMONIA

Ammonia is a highly unusual chemical which is often taken for granted and sometimes considered to be unattractive, unsuitable or otherwise unwanted. For most people the smell of ammonia evokes unpleasant associations with putrefaction, soiled diapers, manure and decay (Lindborg 1999). It's excellent properties as a refrigerant are detailed in a paper presented to the 2012 ASHRAE conference on refrigerants (Pearson 2012) so are not repeated here, other than to emphasise that the high latent heat (seven times higher than R-134a at 50 °C) and high critical temperature (31 K higher than R-134a) make it particularly suitable for use in high temperature heat pumps. In an industrial environment ammonia is a wholly appropriate choice of working fluid: it is cheap, readily avaliable, easy to work with, well understood, unlikely to be subject to more stringent environment regulations and not subject to any future phase down or phase out. It has previously been described as “future-proof” (Pearson 2008).

3 CASE STUDY 1 – RIVER SOURCE DISTRICT HEATING SYSTEM

An ammonia heat pump system was added to an existing district heating system in Norway in 2011. The heat pump has a flow temperature of 90 °C and a return temperature of 60 °C, and delivers a maximum heating load of 13MW. This is achieved with three two-stage modules connected in series on the heating side. The modules each comprise two single screw compressors, low stage and high stage. The low stage machines are cast iron bodies, rated for 36.9 bar gauge allowable pressure. The high stage machines however are cast steel bodies, rated for 69 bar gauge.

The heat pump is water-source, using brackish water drawn from the mouth of a river where it feeds into a fjord. The water is drawn from a depth of 35 m at a distance of 1000 m from the shore and has a reasonably consistent year-round temperature of 8°C. This is caused by a temperature inversion where the fresh river water meets the salty fjord water and the colder river water remains closer to the surface. The refrigeration system uses spray chillers evaporating ammonia at 2°C using on each module, with an average cooling capacity of 3,500 kW. The average heating capacity of the three modules is 4,700 kW, with a power input of 1,490kW, giving a heating coefficient of performance of 3.15. To raise the heating circuit from 60 °C to 90 °C the modules condense at 72.4 °C, 81.3 °C and 89.0 °C respectively. A flow diagram for a single unit is shown in Figure 1 and a block diagram showing how the three are connected together is shown in Figure 2.



Note: temperatures shown are for the second stage of three

Figure 1 – High Pressure Ammonia Heat Pump (from Pearson 2010)

The condensers of the three units are piped in series to produce as low a pressure lift on the first unit as possible. The desuperheaters and subcoolers are piped in parallel, to enable optimum performance to be achieved and to reduce the flow rates through these smaller heat exchangers. Putting the subcoolers in parallel means that all units are able to cool the liquid ammonia as much as possible – likewise having the desuperheaters in parallel enables maximum temperature benefit to be derived from the compressor discharge superheat.

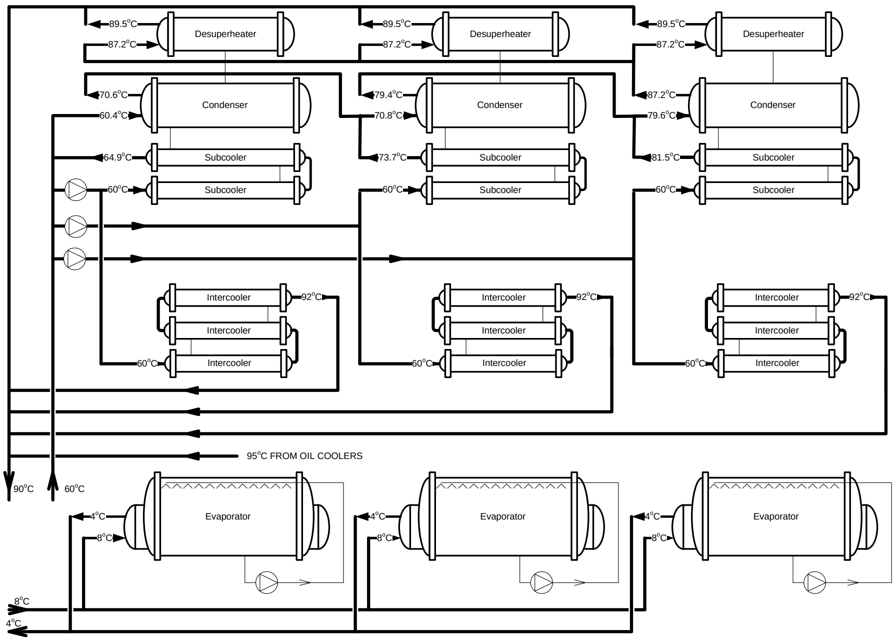


Figure 2 – Schematic of the interconnecting hot and cold water systems for district heating (from Pearson 2010)

On the chiller side of the heat pump the evaporators are piped in parallel. Automatic isolating valves, not shown in Figure 2, prevent flow through chillers when the compressors on that unit are not running.

When the system is required to operate on part load the user can configure the mass flows on the hot and cold water circuits to suit the part load requirement. For example if the part load is two-thirds flow across the full temperature difference of 60 ^{o}C heated to 90 ^{o}C then two units can run on full capacity to meet the demand. If the load is the full flow heated from 60 ^{o}C to 80 ^{o}C then running all three units at part capacity might be more appropriate. Operating experience has shown that maximum efficiency is achieved in summer time by running one of the units on full load and raising the distribution loop return temperature, then switching off the unit and allowing the store of heat to be gradually depleted. In winter time, when the boilers have to run anyway to meet the peak load, efficiency can be further improved by raising the water flow rate and lowering the flow temperature by as much as possible. This can give up to a 10% improvement in the heating CoP with all three units running on full load, reducing the electrical consumption by over 400kW.

4 CASE STUDY 2 – R22 PHASE-OUT IN A CHOCOLATE FACTORY

An ammonia heat pump was installed in 2009 in a chocolate factory in England. This system is more fully described in an article in the ASHRAE Journal (Pearson 2011). The system is designed to provide useful cooling of process glycol (30% monopropylene glycol) to 0^{o}C while at the same time heating a closed loop heating system to 60^{o}C. The system comprises four compressors on a central plant, with two of them rejecting heat to the water heating circuit when required. All four machines are also capable of rejecting heat to atmosphere through a bank of air-cooled condensers. A schematic of the system is shown in Figure 3. Up to 1.25MW of heat can be recovered, with a combined cycle coefficient of performance of 5.46.

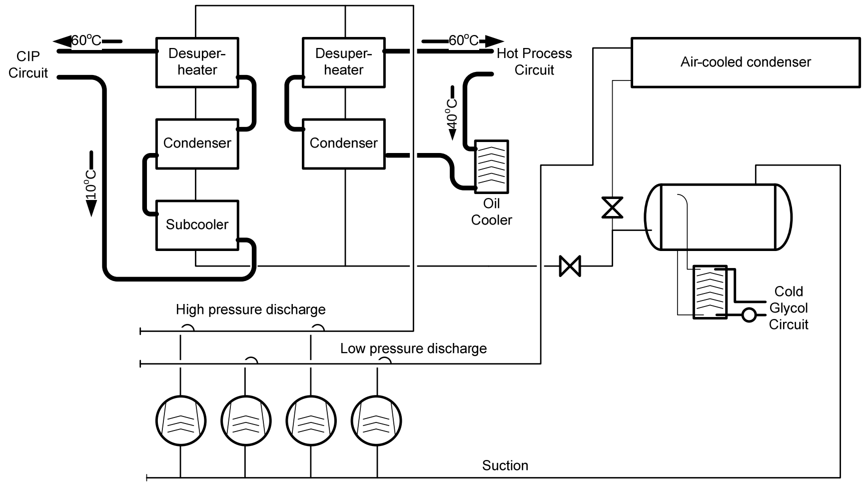


Figure 3 – Schematic diagram of combined factory cooling and heating system (from Pearson 2010)

The high discharge temperature of the ammonia compressors allows the condensing condition to be held at 59^{o}C, with desuperheating of the discharge gas from 100^{o}C to 59^{o}C providing the additional heating to raise the hot water to 60^{o}C. The heating loop is split into two requirements: clean in place (CIP) and closed loop (product heating).

The total glycol cooling capacity of the plant is 3,300kW. About one third of this is provided by the heat pump compressors, which are configured to run as the lead cooling requirement. Since a large proportion of the closed loop cooling duty is related to product heating the two duties are coincident and so a significant saving can be made by serving them both from the same refrigerating system. The CIP load is much less frequent but requires a significantly higher instantaneous load. This is reduced by using a hot water buffer system which is charged and discharged during normal operation. This enables the heating requirement to be better matched to the cooling load. Use of a buffer system like this also allows the rate of heat supply to the process to be far higher than the rate of heat recovery from the refrigeration system, provided the total volume supplied does not exceed the buffer capacity and the recovery time between discharges is sufficiently long.

The installation formed part of a larger site refurbishment which included the replacement of coal-fired boilers with gas burners and the removal of a bank of air-cooled R-22 chillers. The project therefore delivered several environmental goals, including reduction of particulate emissions and removal of HCFC refrigerant as well as providing a significant increase in cooling and heating efficiency. The effect of these changes is shown in Table 1, which indicates that the energy cost was reduced by more than 50% and the total savings made exceeded £1.38 million. As the capital cost of the project was just under £4.0 million it can be seen that the simple payback for this essential work was less than three years. More significantly the capital cost for a project which simply replaced like for like would have cost around £3.0 million so the payback on the incremental cost of designing and installing the combined heating and cooling system was less than twelve months.

Table 1 – summary of operating costs and savings.

	2008Actual	**2011Projected**	**Savings**
Energy Cost	£1,748,000	£835,000	£913,000
Operating Cost	£369,000	£78,000	£291,000
ClimateChange Levy	£321,000	£160,000	£161,000
Costs	£70,000	£46,000	£24,000
Total	£2,508,000	£1,119,000	£1,389,000

These figures are based upon the same production output in the two comparison years. Subsequent plant performance has suggested that the operating cost estimates for the new system were conservative and the actual savings achieved are slightly higher than those given here, but accurate production information was not available for publication.

5 CASE STUDY 3 – HEAT BOOSTING IN A DAIRY

A heat booster system was installed as part of a new dairy installation in France in 2011. The dairy has a central ammonia plant with three compressors chilling glycol to about 0 °C and rejecting heat through air cooled condensers to atmosphere. An auxiliary system of two high pressure ammonia compressors was incorporated in order to enable the discharge gas from the low stage compressors to be raised to 31 bar gauge, enabling it to condense at about 68 °C in order to provide hot water to the dairy for pasteurising and washdown. The low stage compressors discharge at a design condensing condition of 35 °C, providing superheated suction gas to the high stage machines. This arrangement gives 1288 kW of heat per compressor, including 70 kW from the oil cooler, with a coefficient of heating performance of 6.28. In hot weather the low stage discharge pressure rises above the normal design condition, with a peak of 17 bar gauge, condensing at 45 °C. This increases the high stage CoP to more than 8.0.

This system is able to supply some of the heat to the hot water circuit and at the same time reject the remainder, if any, to atmosphere. There are therefore three modes of operation.

- Cooling glycol with no heat demand – all discharge gas to the air-cooled condenser
- Cooling glycol with low heat demand – some gas goes to the high pressure machines
- Cooling glycol with high heat demand – all low stage discharge gas is fed to the high pressure machines.

In this configuration, because there is no alternative heat source, it is not possible to provide heating when there is no demand for cooling. However since the main heat load, the pasteuriser, always provides a demand for cooling the two requirements are usually simultaneous. Wash down water is heated during the production run and stored in insulated buffer tanks for end-of-shift cleaning. Once the hot water storage tank is fully charged the excess heat not required for the pasteurising process is rejected to atmosphere.

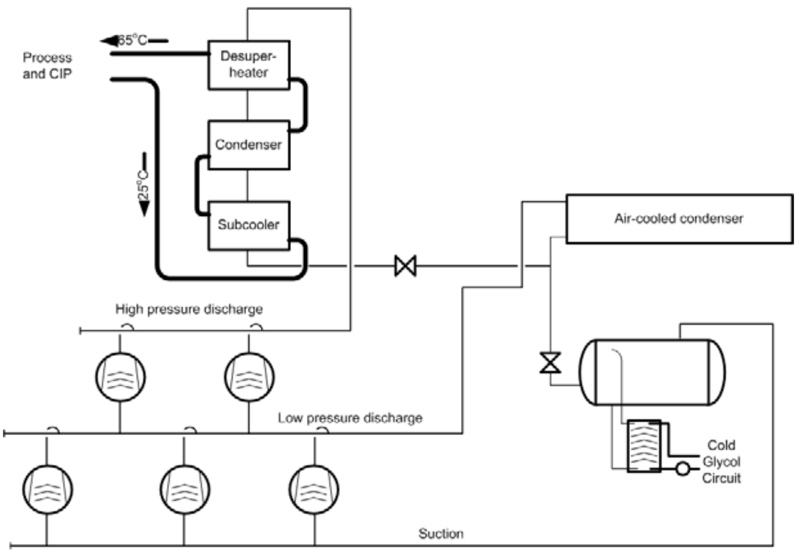


Figure 4 – Schematic diagram of a dairy cooling and heating system

This type of system, where the heat extracted from a cooling process is fed directly into a heating process could be described as an “open” heat pump since there is no intermediate loop between the cooling plant and the heating circuit. This gives higher efficiency than systems where one or two intermediate heat exchangers are placed between the cooling loop and the heating loop. If a break heat exchanger of this type is required then it is more efficient to use a cascade heat exchanger acting as the condenser of the low temperature circuit and the evaporator of the high temperature circuit rather than rejecting the heat into a cooling water loop and then using the cooling water as the heat source for a conventional water source system. The latter arrangement however may be more convenient, particularly if the cooling requirement and the heating demand are not as closely linked as in the dairy.

6 LESSONS LEARNED

The introduction of these new ammonia systems has presented several challenges, both technical and commercial. A few of the more important ones are detailed here. It is important to note that none of these hurdles was insurmountable and with a significant team effort from all the stakeholders in the projects, including the client end-users, the installing contractors and the equipment suppliers, successful results have ultimately been achieved in all cases.

6.1 Technical Lessons

The operation of ammonia plant at such high pressures created several difficulties. Equipment was more heavily loaded mechanically and in some cases manufacturing flaws that would probably have gone unnoticed in a standard refrigeration system caused a catastrophic equipment failure.

6.1.1 Condenser tube leakage

In one case a lamination in a stainless steel condenser tube leaked, allowing ammonia to be absorbed into the heating loop water. In order to confirm that there were no other faulty tubes in the plant all the other stainless steel tubes were eddy current tested: this involved testing 21 heat exchangers of various sizes, with over 8,000 tubes in total. Many other anomalies were found although in the majority of cases the tubes were perfectly serviceable. A few of the tubes with the most severe eddy current traces were removed and replaced. On further investigation it was found that the batch of tubes used in the condenser that leaked had been purchased separately from the rest of the tubes due to delivery time pressure, and the only laminated tubes were in this batch. The condenser was removed from site and completely retubed. The other heat exchangers, having been eddy current tested and with a few tubes removed for more detailed examination were passed as fit for service and have now been in service for three years with no further incidence of leakage. However the consequence of a single tube leak was a significant delay to the whole project, with one of the systems out of service for nearly a year, and an extremely expensive repair. It is possible that the tube would have been sufficiently good for lower pressure service, but in the condenser of the district heating system it was pressurised to 50 bar g.

6.1.2 Compressor thrust system failure

At the same time as the tube failure, but on one of the other systems in the installation the low stage compressor suffered a catastrophic failure. It turned out that the five bolts which held the thrust bearing in place on the end of the main rotor had been over-torqued during assembly and one by one they had suffered fatigue failure with the final three failing in quick succession after about 400 hours of operation. As a result the rotor was displaced axially and contacted the gate rotor supports, resulting in a total write-off of the compressor. The lack of attention to detail in the assembly of the compressor would probably have gone unnoticed in a lower pressure system but because the whole assembly was operating under higher loads it was enough to stress the bolts beyond their fatigue limit.

These two incidents, completely independent but occurring on the same project within two weeks of each other, illustrated how the forces related to high pressure operation could create problems. However there were also more subtle effects which took longer to appear but were equally problematic.

6.1.3 Lubricant dilution

For all these systems the lubricant chosen was a polyalfaolefin, with a seal additive to improve o-ring swell. In fact even with the seal additive it was found that standard o-rings made of HNBR tended to shrink, harden and extrude so replacements using either Aflas, FFKM or Kalrez were sourced. One effect that was foreseen was that at high pressure some ammonia would be dissolved in the lubricant. Laboratory testing had shown that for the PAO with seal additive this could be up to 4% dilution, so a higher viscosity grade lubricant was used. However the interaction between the lubricant and the oil had several additional unexpected consequences.

6.1.3.1 Oil filter damage

The oil filters showed a tendency for the end plate to become detached from the main filter element, which could allow unfiltered oil to bypass the element. When this was discovered no damage had been done to the compressors but the decision was taken to replace all the bearings and fit a secondary filter with a fine metal mesh element in the bearing feed line, so that if the main filter failed again the bearings would be protected. Ultimately the filter failure was attributed to a faulty degreasing process in the manufacturing plant. Like the tube and thrust bearing bolt failures it had not been severe enough to be detected under normal conditions but the higher pressures and temperatures caused the end plate joint to fail.

6.1.3.2 Coalescer element failure

The ends of the coalescer elements in the oil separator also became detached from the main element but the failure mechanism was different. In this case the end cover was bonded to the main coalescer element by a resin. The combination of high pressure and temperature caused the resin to exceed its glass transition temperature (it is suspected that the oil/ammonia mixture lowered the glass transition temperature of the resin) and so it became brittle and broke into many small pieces, some the size of sugar granules, which caked the inside of the oil separator and choked the oil filters. The solution in this case was to change to an all-metal coalescer.

6.1.3.3 Oil pump seal and bearing failures

The oil pumps showed a tendency towards premature bearing failure and the shaft seal would then leak. It was not clear whether this effect was due to vibration created in the pump itself or transmitted from the compressor, either through the baseframe of the unit or back along the oil pipe from the oil injection ports. The pump mounting arrangement was changed to make it independent of the baseframe (freestanding on the plant room floor) and the bearings were mounted external to the oil flow. This combination of changes effected a significant improvement in seal life.

6.1.3.4 Balance piston corrosion

The compressor uses small aluminum pistons to balance the gas forces on the slide valve in order to reduce the forces needed to move the slides. In the original compressors these were found to be severely pitted in all machines (when the compressors were opened to change all the bearings). It is thought that the level of vibration caused cavitation when a thin film of lubricant and ammonia was trapped in a narrow gap between two surfaces. The cavitation caused erosion of the protective layer of Al_2O_3 on the piston surface, exposing the aluminum beneath which then quickly corroded. This looked really unpleasant but it seemed that once the surface was pock-marked the cavitation reduced and so the effect was self-limiting. However several alternative materials were tried and the long term solution is to use all-plastic balance pistons made from PEEK (polyether ether ketone).

6.1.3.5 Slide valve pitting

The combination of high vibration levels and soluble oil also caused some pitting and wear of the slide valves in both the high stage and low stage compressors. To try to overcome this issue one of the district heating systems was changed from PAO to a hydrocracked mineral oil, also using a 100-grade lubricant. However it was found that the ammonia solubility in the hydrocracked oil was also 4% and there was no significant difference noted in the slide valve condition. The slide pitting has not yet caused any long-term problems for the compressors but it continues to be monitored during annual maintenance inspections.

6.1.3.6 Noise and vibration

The low stage compressors on the district heating system are economised and they were found to emit much higher noise levels than predicted. By experiment it was found that part of this noise was related to the economiser ports – with the economiser isolated the noise dropped by almost 30 dB from over 125 dB(A) at 1m (far higher than expected, and completely intolerable) to about 98 dB(A) at 1m (much more in line with what was expected). The economiser could not be isolated permanently because it was used to recover motor heat and feed it back into the system so the penalty on efficiency and capacity was even greater than just the economiser effect. It was concluded that the single screw compressor gave rise to constructive interference between the two economiser ports and that this effect was worse at 3,000rpm (the speed of a fixed speed machine in Europe) than it is at the United States speed on 3,600rpm which is more usual for these compressors. Again this effect is not normally noticeable but at the higher operating pressure for the heat pump, even in the low stage machine, it was unacceptable. To cancel out the vibrations from the economiser ports a Helmholtz resonator was sized and fitted to each of the economiser pipes as close to the port as possible. This was very successful and reduced the noise level to about 100 dB(A) when the economiser was in use, a reduction of over 25 dB(A). There was also a significant pulsation in the main lubrication line. This was thought to be due to liquid hammer because the oil flow is intermittently stopped by the flutes of the rotor passing the oil injection point. As with the economiser port there is scope for constructive interference between the pulsations from the two compression paths, but unlike the economiser the oil galleries are internal to the compressor so it is more difficult to modify them. Nitrogen-filled hydraulic dampers were added to the lubricant pipes, initially to good effect but it was found that the rubber diaphragm inside the damper was not compatible with the PAO and tended to shrink and tear. An alternative design has been difficult to source, but several options are now under trial.

6.1.3.7 Summary of technical lessons

Most of these adverse effects, including the oil solubility, are directly related to the operating pressure. Raising the required temperature from 80 °C to 90 °C increases the operating pressure from 40 bar g to 50 bar g. That is 1 bar for every degree of temperature rise and a 25% increase in pressure for the 10 degree rise. At more familiar operating temperatures the pressure rise is not so extreme. For example a rise of 1 bar from a saturated temperature of -30 °C takes the saturation temperature to -16.7 °C, and from 0 °C to 5.7 °C. Designing for the stresses caused by the high pressure was not difficult, but coping with the other, less obvious consequences of the high pressure was much more challenging.

6.2 Commercial Lessons

The commercial lessons learned through the projects are complex but can broadly be divided into two groups: pre-contract and post-handover

6.2.1 Pre-contract

Before the order is placed for a project there is an element of risk on both sides of the contract. When technology is perceived to be unfamiliar, perhaps also unproved, there is an understandable desire on the part of the purchaser to protect themselves against these risks. This is done by imposing harsh penalties on the supplier. Penalties themselves are not a bad thing: they can keep both parties focussed on dispute resolution and may help to maintain dialogue when times are difficult, but they can also work the other way. In the case studies described above the contractual penalties included

- Penalty for late delivery
- Penalty for failing to achieve capacity
- Penalty for failing to achieve efficiency
- Penalty for failing to complete a 640 hour reliability trial
- Penalty for failing to achieve a set level of availability for the first year of operation

These penalties were all backed by a bank guaranteed “on-demand” bond which means that the supplier’s bank will pay an agreed sum to the purchaser without question if requested to do so. It is understood that the bond will be called in if the supplier defaults on any of the contract terms, but in fact there does not need to be any breach of contract for the bond to be called.

These penalties are actually far tougher than typically applied in the food industry, where the worst case scenario is usually a 5% retention held for a period of 12 months from plant handover. The five penalties above could amount to 25% of the contract value, and the bond requirement means in effect that the supplier’s overdraft facility is likely to be offset by the same amount, so it significantly hampers the supplier’s ability to conduct the rest of his business. In the long term this is not actually in the purchaser’s best interests although it might look like a sensible risk management strategy. Making their supplier less financially sound is a dangerous tactic which could seriously backfire if problems arise and the contractor is put out of business just at the time that the purchaser needs them most.

The heatpump is an extremely attractive proposition, but the economics are brutal. If the capital cost is too high or the payback period is too long then the purchaser will turn to an alternative source of heat; perhaps a biomass boiler or a combined heat and power plant. Piling on heavy penalties requires the supplier to add cost in the form of bond fees, insurances and contingencies, and the performance and efficiency penalties naturally result in the system capability being somewhat understated at the pre-contract stage when the thresholds for penalties are being established. However if the cost is increased too much and the performance is too conservatively stated then the project, on paper, will not stack up and may not go ahead. Then the end user misses out on what was an excellent money making and money saving opportunity, all because the penalties were stacked too severely.

A more intelligent for of engagement between supplier and purchaser is required. This is one which provides for mutual benefit if the project is successful and focusses on the long term relationship between the two parties. A contract which offers performance bonusses or a share of operating profit is a much better motivator, particularly when things are not running smoothly. There is possibly also a role for a third party finance provider to carry some of the risk in return for some of the reward. We have already seen some contractors go out of business when projects fail to meet their promise, and in other cases projects have not gone ahead although they would probably have been highly profitable because the terms on offer were just too onerous.

6.2.2 Post-handover

One of the most important commercial lessons learned from the operational experience of the district heating system is that it is essential to run the heat pump as the base load in order to maximise running hours and minimise fuel costs. In 2013 the district heating system supplied 91 GWh to the town and 67 GWh (74%) came from their ammonia heatpump with a further 10 GWh from the biomass plant. The balance was provided by gas boilers running on either LNG or LPG. The cost of running the heat pump was about 11NOK (Norwegian kroner) whereas the biomass plant was about 35NOK. LNG and LPG cost 75NOK and 85NOK respectively, so the boilers are only run when the peak load needs to be met. The district heating company are required to sell kWh of heat at the same price as electric heating, so with a CoP of 3.0 it can be seen that they make a profit on the heat pump of about 22NOK per MWh, the biomass plant breaks even and running the gas boilers (which they are obliged to do to satisfy all of their contractual obligations during peak load conditions) loses them about 40NOK per MWh. The heat pump is only sized for one-third of the peak load but it provides three-quarters of the total heat supplied throughout the year. Improvements in efficiency are therefore also very valuable – in effect they provide more heat to the town for a given power input, raising the revenue without increasing the operating cost.

7 FUTURE PROSPECTS

The three projects have shown heat pumps to be a viable alternative to traditional heating in industrial systems. There are many more potential applications. Ideally a process, like the dairy should require simultaneous heating and cooling. If this is not the case then there may be opportunities to sell the cooling to a neighbouring business, for example a data centre, or to sell the heat from a factory cooling system into a district heating loop.

8 CONCLUSIONS

Many technical challenges have been addressed and overcome. To enable these lessons to be put into practice on other projects it is likely that a more relaxed and enlightened attitude to the commercial terms is required if end-users are to gain the benefit of these systems. It is perhaps fortunate that the most problematic project was in Norway where there is a rather more enlightened and co-operative attitude to contracts compared to the more litiginous United Kingdom or United States approaches.

9 ACKNOWLEDGEMENTS

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