Industrial refrigeration – energy saving opportunities

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Abstract

Refrigeration plants can contribute significantly to site electricity bills, but the key role these systems play in process and space cooling is often not reflected in the attention they receive during design and operation when considering energy usage. Throughout the worldwide pharmaceutical, food and beverage industries, there are many opportunities for efficiency gains that have frequently been overlooked, yet could so easily have been implemented. This paper outlines some of the more common opportunities, looking into their cost and logistical implications.

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1. Introduction

Energy efficiency has become a focus for pharmaceutical, food and beverage processors in recent times and as a result the associated industrial refrigeration plants have come under close scrutiny. The opportunities for savings start at the design concept stage and are present through construction, control, operation and maintenance. The opportunities arise for various reasons.

- Systems are modified or expanded in unforeseen ways throughout their lives, such that they are often asked to provide cooling in a manner greatly different from that envisaged by the original designers
- Operators change and have differing opinions of priorities
- New technology
- Our own expectations regarding efficiency change over time
- Energy prices rise making what were marginally justifiable energy-savings ideas viable

The opportunities described below for saving energy would apply to any industrial refrigeration system using any refrigerant, although it is assumed that nowadays ammonia or CO_2 would be the refrigerants of choice.

2. Load management

2.1. Air cooler fans

Fans within cold rooms and blast cells contribute significantly to the overall heat load. Fans not only consume power directly, but the energy they impart to the refrigerated space may require half as much again to remove via the refrigeration system.

Fan use should be minimised during design and operation. The Fan Affinity Laws shown below show that if a fan is slowed down, the reduction in energy used reduces as a third power – i.e. an 8kW fan running at 50% speed will consume 1kW.

Volume (V) varies directly with fan speed (N)

$$\frac{V2}{V1} = \frac{N2}{N1}$$

Pressure (dP) varies as the square of the fan speed (N)

$$\frac{dP2}{dP1} = \left(\frac{N2}{N1}\right)^2$$

Power (P) varies as the cube of the fan speed (N)

$$\frac{P2}{P1} = \left(\frac{N2}{N1}\right)^3$$

In blast cells, the resistance to cooling of a product consists of two parts – heat transfer through the product material and a surface heat transfer between the product and the air. At the start of a chill process, while the product is still warm, most heat will be departing from the surface or near to it; in which case, the main resistance to heat flow is the surface heat transfer element. This resistance can be reduced by using high velocity air, although a point of diminishing returns will be reached around 5 to 6 m/s beyond which the heat transfer coefficient will not reduce substantially with increased air velocity (see figure 1). Once the material near the surface has chilled and heat starts to be removed from deep within the product, the major resistance to heat flow is no longer the surface element, but the resistance of the material itself. At this point, fans should be slowed down. No

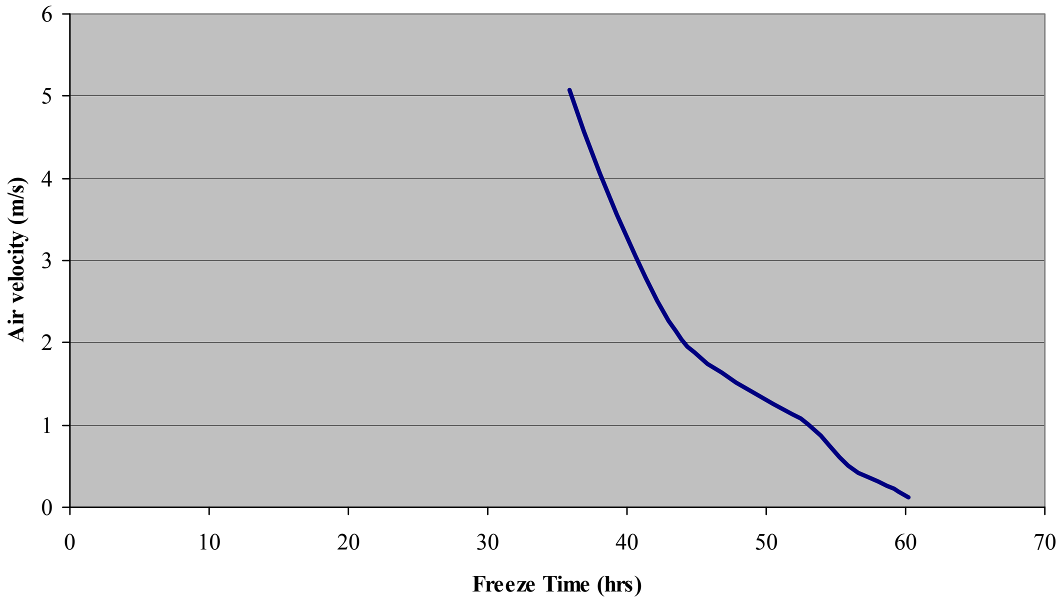


Figure 1
Effect of increased air velocity on freeze times of boxed meat

benefit will be gained from high air velocities; they will only add to the room load and may increase weight loss in open product. On a similar note, airflow in blast cells is employed as a heat transfer medium: its role is to remove heat from the product and deliver it to the evaporator coil. Any air that bypasses the product is useless and the fan power absorbed to move it has been wasted. Airflow control in blast cells is vital not only to the performance of the cell, but also to the energy usage of the cell.

Similarly, VSD control of fans in cold stores is another valuable source of electrical power savings, although controlling fan VSD speeds independently may cause unstable air distribution with one evaporator's operation at 100% affecting another nearby operating at 50%. A unified fan speed control (with preference given to those areas of higher load) should be considered, although overall control strategy may become complex. Fan speed, suction temperature and zone temperature control should all be modulated together. It is likely that for the fan speed control to yield energy-saving results, priority to should be given to suction pressure and only then should fan speed control be used, followed by liquid feed valve cycling.

Load shedding during peak electrical rate periods can also help reduce power costs, however savings will depend on the strategy used. Simply restricting compressor operation can cause suction pressure fluctuations and reductions in evaporator efficiency. By shutting down evaporators that are nearest their set points and leaving running those that are furthest above theirs, the system will suffer the least disruption.

Slow down or stop the fans and save!

2.2. Average temperature control

Evaporators are usually thermostatically controlled, either individually or in groups that are independent of each other. There will be times when none are calling for refrigeration and there will be times when they are all calling. This will cause unstable operation. By weight-averaging room temperatures across all temperature sensors, the room as a whole can be controlled in a much more even manner. Priority can be given to some areas more than others and some sensors can be ignored during a local defrosting periods.

2.3. Defrosting


Figure 2
Steam trap used for ammonia condensate

Defrosting evaporators consumes energy. Obviously the preferred approach here is to avoid frost build-up in the first place. Frosted surfaces are inefficient heat transfer surfaces and removing the frost consumes energy. Various methods exist for removing frost including those using electricity, water and hot gas. Electric defrost obviously consumes power to generate heat but the energy costs are compounded by the fact that some of the heat is lost to the space. Water defrost is often used when coils are washed while they are offline. This arrangement is frequently used in spiral freezers and, in some cases, cold stores. Hot gas defrost is most commonly used, but is often misapplied. Hot gas at 60C to 80C from the compressor discharge is passed through the evaporator coils, heating them from within, thereby melting the ice on the outside. (High hot gas temperatures may create steam, which must be re-condensed later.) Defrosting using heat within the coil in this manner minimises heat loss to the space, but frequently uses more hot gas and energy than is necessary. Since pressure regulators are used at the coil outlets during defrost, a significant portion of the gas entering the coil simply cools slightly without condensing and leaves the coil through the regulator. This cool gas now enters the refrigeration system and must be recompressed. By installing small traps on the coil outlets and outlet pressure regulators on the coil inlets, the pressure in the coil can be set at a lower temperature and only liquid may leave the coil. The temperature of the outer surface of the evaporator coil only needs to be raised slightly above 0C to dislodge the ice. The coil pressure need only be set to approximately 6 or 7 bara (10 – 15C); there is no need for 60C+ hot gas to be fed into cold room air coolers. Condensation of lower temperature ammonia in the coil will mean that the much more abundant latent heat is used. The liquid/steel heat transfer coefficient will be much higher than the vapour/steel heat transfer coefficient experienced in the more common defrost method leading to markedly faster defrost periods. So much faster in fact that a longer defrost cycle for a separate defrost circuit in the pan may be required. The gains offered here include:

- Much faster defrost due to liquid/steel heat transfer coefficient
- Smaller temperature swings in space
- Less hot gas used due to use of latent heat
- Negligible hot gas flow directed to compressor suction
- The condensed liquid passing through the trap is fed to the low side as high pressure liquid – just as the system does anyway
- Less extreme pressure and temperature swings – less stress, less leaks
- No need to maintain high condensing pressures for defrosting purposes

Note that this also works well when using ammonia coils to provide heat, e.g. a reheat coil for dehumidifying. If this type of hot gas piping and control arrangement is used, artificially maintaining an elevated condensing pressure simply to provide high pressure hot gas is unnecessary. In large systems where hot gas must be fed long distances, small hot gas compressors may be employed to keep the hot gas header pressure elevated, but only in extreme circumstances.

Whether with the traditional defrost arrangement or with this enhanced method, the defrost condensate should be returned to the highest suction level available. For example if there is a high temperature suction line in the vicinity of a cold store, the hot gas defrost condensate should be directed to it. This will minimise the energy used to recompress uncondensed hot gas.

Finally, a common source of false-loading is leaking hot gas valves. If not the main hot gas valve itself, then the pilot valves at hot gas powered suction valves have been known to leak high pressure gas into the suction line.

Defrosting consumes energy – be efficient!

2.4. Dirty evaporator coils

Most food or beverage facilities are relatively clean or are cleaned regularly and dirty evaporator coils are not an issue. There are areas is some facilities, however, where this can become problematic. Airborne fats and oils, dust from boxes or soil from dirty root vegetables for example can all accumulate on coil surfaces.

2.5. Free cooling from air handlers

A number of production areas use high spec air handlers to provide process room conditioning. These air handlers provide:

- Cooling
- Pressurisation
- Air filtration
- Fresh air make-up

During cleaning cycles, these air handling units often have the capability to provide 100% outside air for drying the room. The outside air is highly filtered to ensure room cleanliness at all times. This capability presents the possibility for “free” cooling during low ambient temperature conditions. Since the outside air flow can be modulated from 100% for drying down to 5% – 10% for pressurisation during cooling, a setting anywhere in between could be used during low ambient temperatures to provide outside air for cooling, pressurisation and fresh air. The refrigeration coil could be switched off or cut back to a minimum level.

2.6. Others

Other load management opportunities include the more obvious ones:

- Door management – infiltration is a major contributor to refrigeration loads. It will also affect plant pressurisation, air quality and humidity levels.
- Placing warm, steaming product into cold rooms or blast freezers. This is not just a heat issue, but it also involves product weight loss and ice build-up on coils.
- Excessive ice build-up – a sure sign of moist air ingress. Can result in badly sealed doors and openings

3. Compressors

3.1. Single stage vs. two stage

The major power consumers in any refrigeration systems are the compressors. Their energy consumption is directly related to their compression ratios; however other factors can affect efficiency also.

At large compression ratios, compressors lose their efficiency due to inherent design complications and also due to thermodynamic properties of the refrigerant. Multistage compression addresses this problem and has been utilized in a number of ways using compound machines, dual compressor packages and fully built-up two stage systems.

Two stage systems are efficient, but come at a price: two compressors are required in place of one single stage machine. The gains in efficiency come from many sources when using two stage systems. Assuming inter-cooling is employed, two staging the compressor discharges ("two-stage compression") will ensure that compressor discharge temperatures are not excessively high. External oil cooling of the low stage machines allows that heat to pass directly to the condensers or heat reclaim systems without passing through the high stage machines. Although injected oil should provide sealing between rotors and housings, screw compressors suffer a certain amount of refrigerant vapour "blow back" from discharge to suction, the amount of which is dependent on the compressor ratio. The lower compression ratios in a two stage system will minimise this back-flow. The greatest gain to be had from a two stage system however is the cascading drop in pressure of the liquid flow – or "two staging the liquid". High pressure liquid from the receiver is fed first to the high temperature system (-10C for example). Here it drops in pressure, flashes off a small portion of liquid into vapour and reduces its temperature, ready to be fed finally to the low temperature system. Without this cascading of liquid, the vapour that flashes off in the high temperature system would otherwise have formed in the low temperature vessel from where it must be compressed.

For a typical -30C ammonia system designed to condense between +25C (10 Bara) and +35C (13.5 Bara), table 1 below illustrates the increase in Coefficient of Performance (COP) that can be seen by successively two-staging the liquid then by adding two-staged discharge. As can be seen in the table, the greatest contribution to efficiency gain is realised by two-staging the liquid.

As suction temperatures drop, efficiency gains from two-stage compression begin to increase in significance until approximately -40C, where it becomes almost unquestionably justifiable. Between -30C and -40C, it would be advisable to run energy models based on expected load and ambient condition profiles to decide whether the extra capital outlay for full two -stage compression can be warranted.

	System configuration	**C.O.P.**
1	Single stage	2.00 – 2.60
2	Single stage discharge with two stage liquid	2.15 – 2.75
3	Two stage discharge and two stage liquid	2.25 – 2.80

Table 1 Effects of two-staging

Another inexpensive means of achieving two-stage liquid feed is by using screw compressor side-ports. Referred to as “economising”, liquid is flashed down to an intermediate temperature using the intermediate pressure at a compressor side port. The flash gas created enters the compressor at this midway point in the compression process and the cooled liquid carries on to the low temperature load.

Spend the money, multi-stage and save the money!

3.2. Liquid injection oil cooling

Cooling compressors (which compress vapour) by injecting liquid (which is incompressible) into them is not a good idea. Although the compressor and its oil flow will be cooled, the vapour resulting from the evaporated liquid must be recompressed, which requires energy. Depending upon where on the block the injection occurs, this vapour may displace other vapour that could be providing refrigeration, thereby reducing compressor capacity. Malfunction of the liquid cooling feed valves could result in overfeeding of liquid refrigerant into the compressor. Since the liquid is incompressible, it would displace the oil, causing lubrication and sealing problems and ultimately damaging the machine. Finally, some of the ports in the compressor housing into which the liquid is fed are close to discharge pressure. Condensing pressure must be maintained high enough to ensure that cooling liquid can still be fed into the compressor. Liquid injection oil cooling is inexpensive to install meaning that some contractors are happy to provide it to reduce their costs. It may be retained for use in an emergency, but external oil cooling should be the first choice, every time. It will pay for itself within a short time, either directly (through efficiency gains) or indirectly (by making waste heat available for reclaim).

Avoid the use of liquid injection oil cooling where possible!

3.3. Oil management

Some compressors rely on differential pressure across the compressor to generate oil flow. Slight modifications or the addition of cycling full lube oil pumps would allow the condensing pressure to drop while still having the capability to maintain adequate oil flow. Another compressor issue is oil separation. Lower condensing pressures will produce high specific volume discharge vapour which in turn will cause high oil separator velocities. If the compressors are already in place, this may become the limiting factor. New systems, however, may be specified with oversized oil separators.

3.4. Compressor volume ratio

Some older compressors have fixed volume ratios (Vi), designed for specific compression ratios. Operators may try to maintain a constant compression ratio to attain greatest efficiency from the machine. This is wrong. Although operating at a Vi different than the design will absorb more power, it will still be less than operating at the right Vi when a lower condensing pressure could be used.

Some modern screw compressors have automatic variable Vi control which will manage volume ratio as conditions vary. Others have manually-variable Vi. This must be set for the "general" operating condition to ensure that the machine is working as efficiently as possible and not under- or over-compressing.

3.5. Compressor unloading

A more obvious source of inefficiencies is compressor loading and unloading. Reciprocating machines unload fairly efficiently in the limited, stepped-nature of their cylinder unloading mechanisms. Screw compressors are much preferred in large system for their relatively small footprint, small number of moving parts and their high compression ratio capability. Their unloading characteristics, however, are not so desirable. Below about 90% load, screw compressor efficiencies show a marked decrease. In multi-compressor installations, there may be many parallel compressors running at various stages of unloading. Sophisticated control systems can help by running only those compressors required at full load and leaving only one unloaded, preferable a reciprocating machine. More recently, the lower costs and increased reliability of Variable Speed Drives has seen the application of these to refrigeration compressors. Most screw compressors can be driven fully-loaded from 3,600 down to 1,800rpm with little appreciable loss of efficiency.

Don't run screw compressors unloaded!

3.6. Suction pressure control

The suction pressure of the system directly affects compressor power and capacity. Lower suction pressure results in lower capacity, higher specific power and, therefore, lower efficiency. Since pressure and temperature are inextricably linked, lower room temperature also means lower efficiency. Vapour pressure drop in piping and equipment must be minimised. Running high temperature rooms on low temperature systems is highly inefficient. If this must happen, the evaporators in the low temperature spaces should be over-sized to allow the low temperature system to run as high as possible.

Piping pressure drop must always be present, since pressure difference is the sole means for driving ammonia from the evaporator to the compressor. However, excessive pressure drop caused by undersized piping should be avoided. Wet suction return piping carrying over-fed, un-evaporated liquid refrigerant entrained in the vapour is especially prone to high pressure drops. Mixed-phase flow such as this inherently creates higher pressure drops in piping, so sizing is crucial here. Pipe should flow downhill toward the suction separator, avoiding risers. Risers will generate columns of liquid and resultant pressure drops.

Remote low temperature pumping vessels should receive low temperature liquid to reduce flash at the vessel and reduce the flash gas returning down the wet suction line. This in turn will reduce pressure drop.

Run suction pressures as high as possible!

4. Condensers

4.1. Condenser head pressure control.

The other end of the compression ratio equation is the condensing pressure. This value must remain above the saturated temperature of the condensing medium. In other words, if an evaporative condenser is used, the saturated condensing temperature must be above the ambient wet bulb temperature; if an air-cooled condenser is used, the saturated condensing temperature must be above the ambient dry bulb temperature. How much above these temperatures depends on the load, but, importantly, there should be no other factors artificially maintaining elevated condensing temperatures and pressures. In practice however, other factors present themselves either by design or via operating procedures. These should be rigorously challenged and negated. The condensing pressure (sometimes referred to as the "head pressure" or "discharge pressure") must be allowed to float with the ambient conditions; falling as low as possible during low load or low ambient temperature times, without being artificially held high to overcome other system shortcomings. To do this a number of things must take place. Firstly a large condenser should be installed. The larger the condenser, the closer the approach can be between the ambient and the condensing temperatures. The optimum operating condition for a condenser is with its design temperature differential, (usually 10 – 14K) between the condensing temperature and the local wet/dry bulb temperature. A head pressure control system would maintain this differential by varying fan speed. The control objective is, of course, to minimise energy usage by both the condensers and the compressors. Reducing the head pressure will reduce compressor power, but running more condenser fans to do so may be counter-productive. There is standard control software available that will control condenser fan speed to achieve this balance.

Don't artificially elevate condensing pressure!

4.2. Air purging

Non-condensable gases in the system will act to raise condensing pressure. These gases are primarily air, but other gases may also be present. These gases may enter the system from a variety of sources.

- Leaks into systems operating in a vacuum
- Gaskets, shaft seals
- Compressed air from pneumatic actuators
- Equipment opened for maintenance, charging
- Degassing of oils

Air flows with the ammonia throughout the system, through the compressors to the condensers. The outlets of the condensers are usually trapped, so the air remains there, driven by the flow of the ammonia toward the liquid outlets. Depending on piping arrangements, it may also accumulate in the high pressure receiver. Continuous purging from these points can easily remove it. Alternatively, shutting down a condenser and isolating it will stop all through-flow and allow the air to rise to the top, from where it can be removed manually. To determine whether a system has air in it, the condensing pressure and the liquid draining temperature should be known. When fully loaded, a condenser will provide some subcooling of the liquid draining from it. The liquid will approach the wet bulb temperature and usually around 5K of subcooling can be expected. Adding 5K to the temperature measured should give the saturated condensing temperature as derived from the condensing pressure. If these two do not match (i.e. if it appears that there is much more liquid subcooling taking place), then non-condensables are present.

If an automatic purger is used, it will continuously remove air. Some automatic purgers will dump their water when it becomes saturated with ammonia. By monitoring how often this takes place, a measure can be made of the rate of air ingress. Any sudden change should give rise to concern.

It has been estimated that 1bar extra pressure due to the presence of air in a 5,000 kW, -30C system may result in the consumption of an extra 1,000,000 kWh per year [4]. Since installation of a purger may only cost on the region of €15,000 to €20,000, payback time is short.

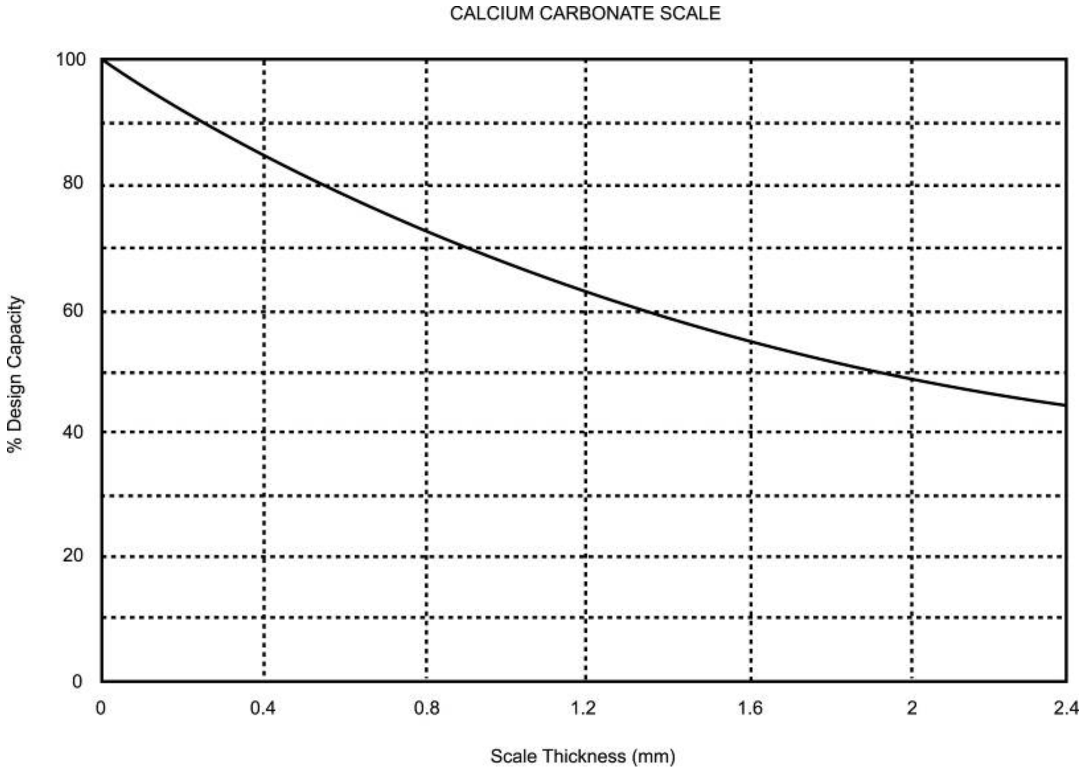
4.3. Water nozzle performance

The water distribution system in an evaporative condenser is vital to the efficient performance of the heat exchange coil. Clogged or missing nozzles can result in dry portions of the coil. An under-performing or cavitating pump will also lead to poor water flow.

Figure 3
Water spray nozzles

4.4. Scale

Evaporation of water from an evaporative condenser will naturally leave behind mineral deposits on the coil. Water treatment and hardness monitoring should take care of this and minimise scale. However, it only takes a very small amount of scale for a condenser to loose a considerable amount of capacity. The graph shown in figure 4 confirms that 1 mm of scale on the coil can result in 30%+ loss in capacity.



Keep condensers clean

5. Miscellaneous

5.1. Underfloor heating

There are cases where de-superheaters are used to provide energy to heat glycol for underfloor heating. In an argument similar to that dispelling the myth that “hot” hot gas is required for defrosting, a case can be made for similarly dispelling this underfloor heating issue. The ground beneath a cold store only has to be held slightly above freezing to prevent frost heave. Any higher and you risk adding unnecessary heat to the cold store itself. +10C to +15C glycol is all that is required and using +60C to +80C superheat to produce it is a waste of good high grade heat. A plate heat exchanger piped as a condenser will give plenty of low grade (+20C - +35C) heat and the superheat can be harnessed for better uses (see section 5.2)

5.2. Heat reclaim

Refrigeration systems remove low grade heat from one area, convert it high grade heat and reject it to another area. The rejected high grade heat can be captured and reused, however it must be stressed that the primary use for a refrigeration system is to provide refrigeration. Jeopardising refrigeration performance, reliability, efficiency or safety by also running it as a heating plant is not recommended. Ill-conceived de-superheaters that add pressure restrictions to the discharge piping and accumulate liquid during low load/ambient conditions are frequently a problem. Head pressure variations and liquid feed unreliability often result.

The primary sources for heat reclaim from refrigeration systems are compressor oil coolers and compressor discharge superheat. Plant rooms may be adjacent to air compressors, in which case these too are an excellent source of high grade waste heat.

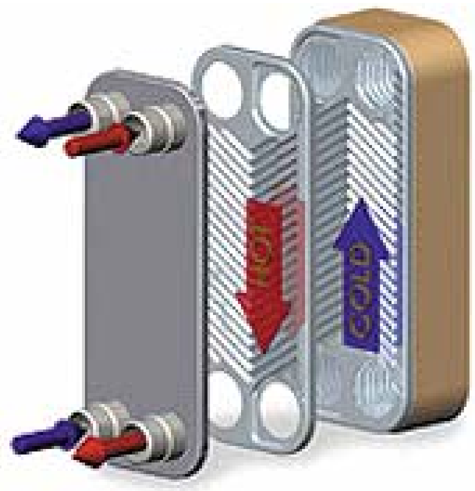


Figure 5
Water heating plate heat exchanger

Pipe routes and configurations will depend on the ultimate use for the reclaimed energy. Large amounts of warm water for boiler feed or general site water usage can be accumulated and stored in tanks, topped up continuously. Higher temperatures can be reached by feeding directly from the heat exchangers. De-superheaters can produce +50C or +60C water and air compressors even higher. Only 4% of the electricity used by an air compressor is actually used to create compressed air. The remaining 96% of the electrical energy is converted into the by-product of compression: heat It may not be possible to recapture all this heat, but by recapturing up to 80% of it, air compressors can become a ready source of hot water up to +70C.[5]

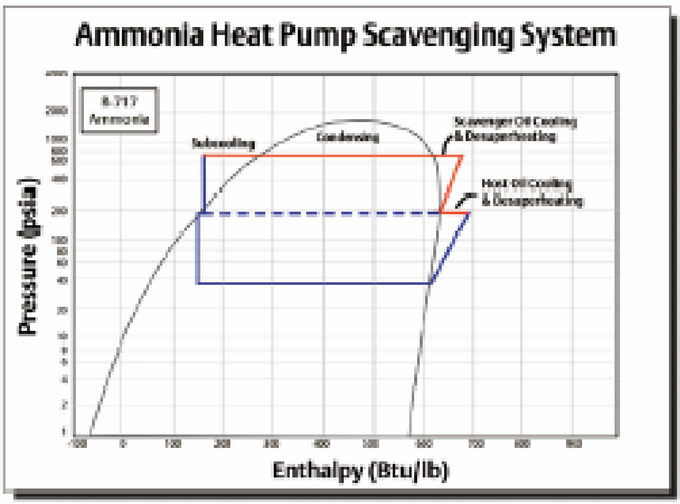


Figure 6
Heat Potential of a Scavenging Heat Pump System[6]

Flow rates at these elevated temperatures may be limited by the available energy; however the use of heat pumps to augment this is becoming more common in applications such as this. Although small stand-alone, self contained heat pumps have been available for a long time, recent emerging technologies have allowed “scavenging-type” heat pumps to harness the rejected heat of industrial refrigeration systems. Heat pumps of this type can be built-up on site or (probably preferably) purchased as a skid mounted, pre-packaged item. Constant, high flow rates of 70C and above are possible. Note that energy used in the heat pump will benefit the overall refrigeration system by also removing load from the condensers. Heat reclaim system of this type are now used throughout the world with capacities reaching as high as 14mW.

Heat reclaim systems should be carefully designed to reuse heat as efficiently as possible whilst ensuring that performance of the refrigeration system itself is not adversely affected. Retrofitting heat reclaim systems may cost upwards of €100,000, nonetheless, the savings offered will allow this to be recovered in a short time.

Install heat reclaim, but remember – it's a refrigeration system first and foremost.

5.3. Liquid flow.

Frequently high pressure liquid is fed to distant users throughout a facility. Since the liquid is saturated, pressure drop in the lines will cause flashing. Poor design, undersized piping or increased loads often lead to this. Elevated condensing temperatures go some way to addressing this, but fixing the problem (pumps, pipe insulation, liquid subcooling, etc) would be far more energy-efficient than running the compressors harder.

5.4. Set point analysis

During the commissioning phase of any system, various set points are chosen as starting points. These may include:

- Suction pressure
- Discharge pressure
- Coil pressures / pressure regulator settings
- Room / glycol / chilled water temperatures

There are other values that are selected and set within control systems during the life of the system, including:

- Compressor set points – timers, dead-bands, VFD settings
- Condenser set points – fan and pump staging, head pressure control criteria, heaters
- Evaporator fan cycling
- Hot gas defrost schedule – initiation, pump down, pressurisation, defrost time, depressurisation, coil freeze, fan delay, termination, regulator set points

None of these are set in stone. Systems change over their lifetimes; new loads, modified loads, utility prices, new equipment, changed priorities. All set points should be reviewed periodically and adjusted to be certain that good performance, energy efficiency, system reliability and overall safety are all fully addressed.

Don't be afraid to re-set set points!

6. Conclusion

Refrigeration systems can be one of the primary energy users in food, beverage & pharmaceutical production. Too often, however, short-term profitability and first cost are allowed to overshadow sound engineering principles. Small, independent, relatively inefficient air-cooled refrigeration systems are frequently installed piece-meal into a facility. Cheap to buy and easy to install, these systems have attractive short-term characteristics. Life-cycle costing, however, reveals their true nature and it is worth spending more money buying industrial central ammonia refrigeration systems. Economies of scale, large and more efficient motors lead to lower overall energy bills. Small, scattered systems are very hard to harvest waste energy from while the waste heat producers in centralised systems are gathered in one location, making them easier to work with.

Having made the right decision and opted for a centralised ammonia refrigeration system, there will be further opportunities for efficiency improvements. Although the components, the manufacturers, the refrigerants and the overall refrigeration cycles are common everywhere, each system is unique and presents its own unique opportunities.

The common failures to address energy efficiency outlined in this paper are coupled with the observation that each system should be treated as one-of-a-kind. It may have problems in common with other systems and, hopefully, the preceding pages have brought some of these to light. However, a detailed energy appraisal may uncover other issues, issues specific to your system. Audits should be regular activities, but successful energy conservation will not be achieved through one-off, “snap-shot” audits. There should be an ongoing programme of periodic appraisals, modified daily routines and regular monitoring. Common or unique, all energy-related issues should be addressed.

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