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Project 2783-11
Final Report
October 1988

Industrial Heat Pump Manual

Technical and Applications Resource Guide for Electric Utilities

Prepared by
Linnhoff March, Inc.
Leesburg, Virginia

REPORT SUMMARY

SUBJECT	Industrial electric technologies
TOPICS	End use Electrotechnology Heat pumps Technology utilization Electrification Industry
AUDIENCE	Customer service engineers / Marketing managers

Industrial Heat Pump Manual

Technical and Applications Resource Guide for Electric Utilities

Heat pumps can significantly improve efficiency and reduce energy costs in industrial processing applications. This resource guide presents the basic technical features of heat pumps, describes the differences between industrial and commercial units, and reviews and analyzes the domestic and worldwide industrial market.

BACKGROUND Heat pumps can improve the efficiency of industrial processes. However, the heat pump's unique features make it preferred for some applications and unsuitable for others. A good technical and economic reputation depends on choosing the right heat pump for each situation.

OBJECTIVE To explain the use of industrial process heat pump systems, including their benefits and proper application.

APPROACH Researchers compiled data on industrial heat pump applications, as well as experience from government and private sources including DOE reports, publications from the Heat Pump Technology Center of Japan, the Commission of the European Communities, and the International Union of Electroheat. In addition, they reviewed recent EPRI and DOE reports on pinch technology and its role in evaluating energy cost reduction alternatives. A comprehensive industrial heat pump design methodology provided input for a manual for customer service and marketing personnel.

RESULTS This resource guide presents the basic technical features of heat pumps and describes the differences between industrial and commercial units, such as the choice of sources and sinks. It reviews and analyzes the current state of the domestic and worldwide industrial heat pump market. The manual also discusses in detail how heat pump technology fits into the complex puzzle of industrial energy cost reduction options. Just as coal, gas, and oil compete as energy sources, industrial heat pumps compete with a range of energy cost reduction options and technologies, including passive heat recovery or process modification.

The manual presents a comprehensive industrial heat pump design methodology, incorporating developments from the past decade. It identifies heat pump application opportunities, as well as design and operation standards.

EPRI PERSPECTIVE

The *Industrial Heat Pump Manual* can constitute an important tool for people and agencies promoting energy-saving technology—utilities, government agencies, independent consultants, industry associations, and those who advise companies on cost reduction options. Industrial companies can also use the guide to verify heat pump evaluation before installation. The manual, together with the HPSCAN code, a forthcoming heat pump screening analysis, and the APLUS code, a plant utility system analysis software (EPRI project RP2783-5), will form a complete package on placing and using industrial heat pumps for the benefit of utilities and their industrial customers.

PROJECT

RP2783-11
EPRI Project Manager: I. Leslie Harry
Energy Management and Utilization Division
Contractor: Linnhoff March, Inc.

For further information on EPRI research programs, call EPRI Technical Information Specialists (415) 855-2411.

Industrial Heat Pump Manual
Technical and Applications Resource Guide for Electric Utilities

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Research Project 2783-11

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Prepared by
Linnhoff March, Inc.
Leesburg, Virginia

ABSTRACT

When properly applied, industrial process heat pumps offer process owners the opportunity to achieve significant energy cost and efficiency improvements.

This Resource Guide presents the basic technical features of heat pumps and describes the differences between industrial process (or more simply, industrial) heat pumps and commercial units. The current state of the industrial heat pump market is reviewed and analyzed.

Industrial heat pumps are not the only energy saving technology available; there are many 'competitors' including passive heat recovery, process modifications, and cogeneration. The way in which the industrial heat pump interacts and competes with these other alternatives is discussed in detail. Pinch Technology is introduced as the best method for finding and assessing all industrial heat and power options.

A comprehensive industrial heat pump design methodology is presented. It can be used in industrial plants to identify heat pump opportunities and to ensure good design and operation. This methodology is based on practical experience and will help those interested in promoting and using industrial heat pumps benefit from the last decade of heat pump development.

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ABOUT THIS GUIDE

Heat pumps have an important role to play in improving the efficiency of industrial processes. The purpose of this Resource Guide is to promote greater understanding of industrial heat pump systems including their benefits and their proper application. Improved understanding will lead naturally to more installations, better engineered installations, and a more competitive industry.

This Resource Guide is aimed specifically at people or agencies who promote the use of energy saving technology to potential industrial users. This includes, utilities, government agencies, independent consultants, industry associations, and others who can act as 'facilitators' to advise companies on their cost reduction options. The Guide can also be used by industrial companies themselves to make certain that a proper evaluation has been performed before a heat pump is installed.

It should be stressed that the Resource Guide is not a textbook or design guide; it is not intended that the reader be able to build a heat pump having read it! The Resource Guide provides enough information to enable a reader with a technical background to act as an intermediary between a potential purchaser and a heat pump manufacturer through all the stages of the evaluation, design, and installation of the heat pump system.

The Resource Guide presents the basic technical features of heat pumps and describes the differences between industrial and commercial units. The current state of the industrial heat pump market is reviewed and analyzed. Another very important issue that is discussed in some detail, is the way in which the industrial heat pump fits into the 'complex puzzle' of industrial energy cost reduction options. Just as coal, gas and oil compete as energy sources, the industrial heat pump competes with a whole range of energy cost reduction options and technologies. These competitors are shown by the 'Option Tree' in Figure A1. The 'Option Tree' is discussed in greater detail in Chapter 5.

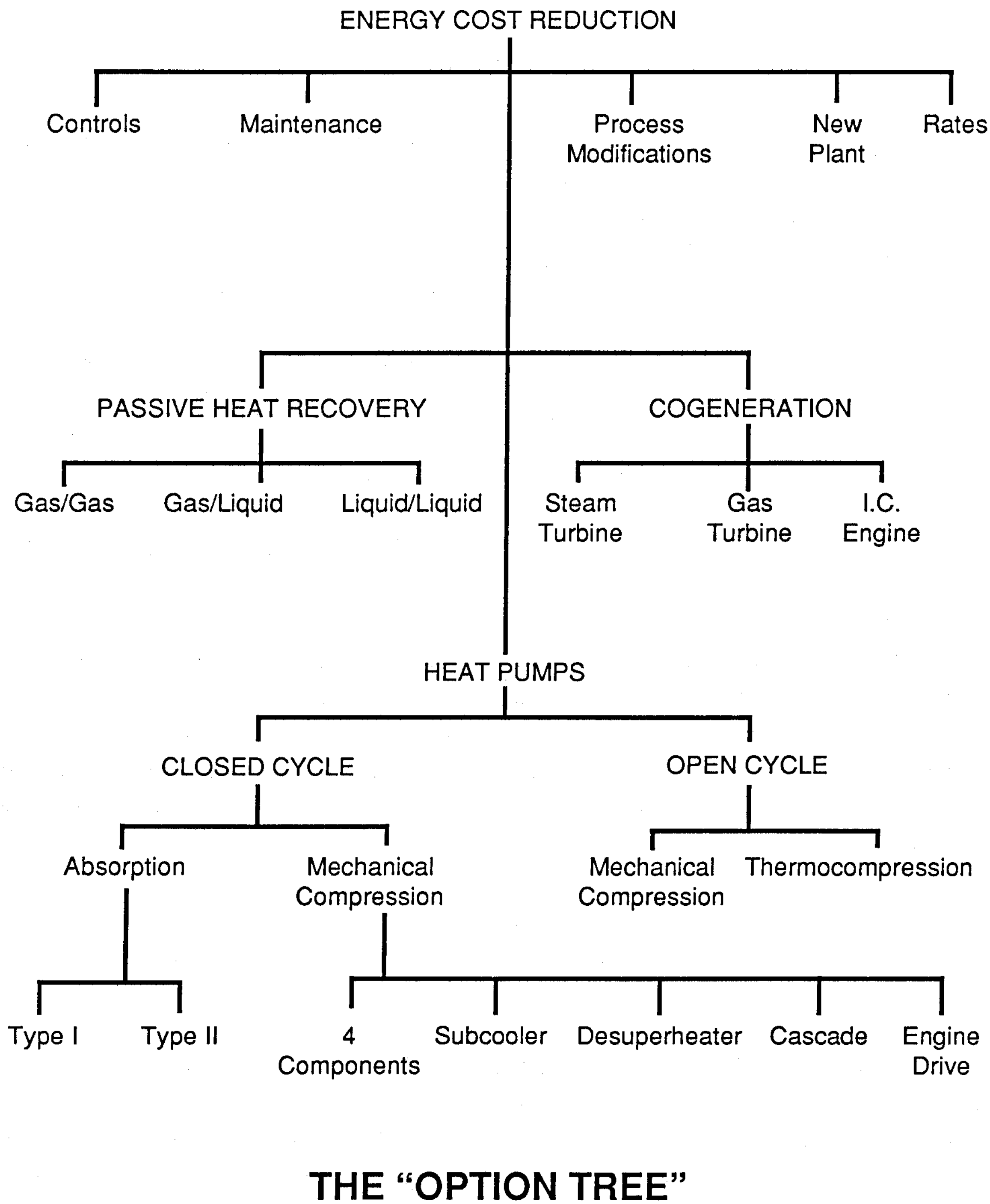


Figure S-1. The Option Tree

The heat pump has some unique features which make it preferred for some applications, but unsuitable in others. To establish a good technical and economic reputation it is important to assure that the right heat pumps are applied in the right situations. How these good applications can be found is addressed in this Resource Guide also. One thing is certain, rushing out to find a waste heat stream as a heat source and another process stream to accept the heat delivered by the heat pump is not good enough. This 'cherry picking' approach may produce some projects which are acceptable, but installation of a heat pump is usually more capital intensive than other heat recovery options. This leads to some important questions; could a different heat recovery project have saved as much or more money at a lower investment? Was there a better heat pump project? Was heat pumping really the right thing to do? A consistent, sound approach to assessing energy cost reduction alternatives is essential. Such an approach must be thermodynamically based and allow alternatives to be identified and compared in a systematic fashion. Pinch Technology is introduced as the best method for finding and assessing heat recovery, heat pumping, and, indeed, all heat and power options. If the market is to grow, it is particularly important that heat pumps are compared to the other options in an unbiased fashion. The engineer must generate a number of possible projects and then evaluate them for technical and economic feasibility. Encouraging the installation of heat pumps when they are not the best technical or economic solution will eventually be detrimental to their use where they are the best choice.

The second half of the manual sets out a comprehensive design methodology that can be used in industrial plants to identify heat pump opportunities and ensure good design and operation.

The first step in this methodology is the Feasibility Stage. At this point, many options need comparison, and the technical and economic feasibility of each must be assessed. Assuming that a heat pump option is identified as the best investment the next step is the Design Stage. Detailed operating data is required to ensure a good design. The methodology encourages the purchaser to carefully specify the heat pump, not just to rely on equipment manufacturers to "know best". Finally, the important points to consider during commissioning and operation of the heat pump are highlighted.

This Technical and Applications Resource Guide is quite unique. For the first time in a document of this type, the industrial heat pump is treated quite separately from space conditioning systems. This is vital because many design criteria are quite different. Also for the first time, the practical experiences gained during the last decade of vigorous heat pump activity have been included. Use of this Guide allows the reader to benefit from previous successes and failures and help to establish a strong industrial heat pump market.

Chapter 1

INTRODUCTION

CHAPTER SUMMARY

This first chapter sets the scene for the rest of the Resource Guide. The technique of heat pumping is outlined and compared with other heat recovery options. The industrial heat pump is then defined and typical applications described. A clear distinction is drawn between industrial heat pumps and the more commonly known space heating and cooling units. The industrial market and its importance are examined briefly and the concept of first and second generation heat pumps is introduced. Finally, the aims and structure of the Resource Guide are discussed.

WHAT IS A HEAT PUMP?

Heat pumps are one of a family of systems that are able to recover waste heat. They have a unique feature that makes them different from all other heat recovery devices: a heat pump is able to deliver useful heat at a temperature higher than the waste heat source. All other heat recovery systems are "passive" and must pass the waste heat downhill along the temperature gradient. The heat pump is an "active" device that is able to lift heat energy up the temperature gradient. Of course, you cannot get something for nothing - the temperature lift must be paid for by supplying the heat pump with prime energy (such as electricity) as well as with waste heat. A useful mechanical analogy to the heat pump is a pulley system to lift weights. The amount of work required to lift a weight depends on how high you wish to raise it; similarly, the prime energy input to a heat pump is related to the temperature lift. Figure 1-1 illustrates this analogy.

There are many uses for heat pumps and equally many different designs and systems. All heat pumps consist of three essential parts:

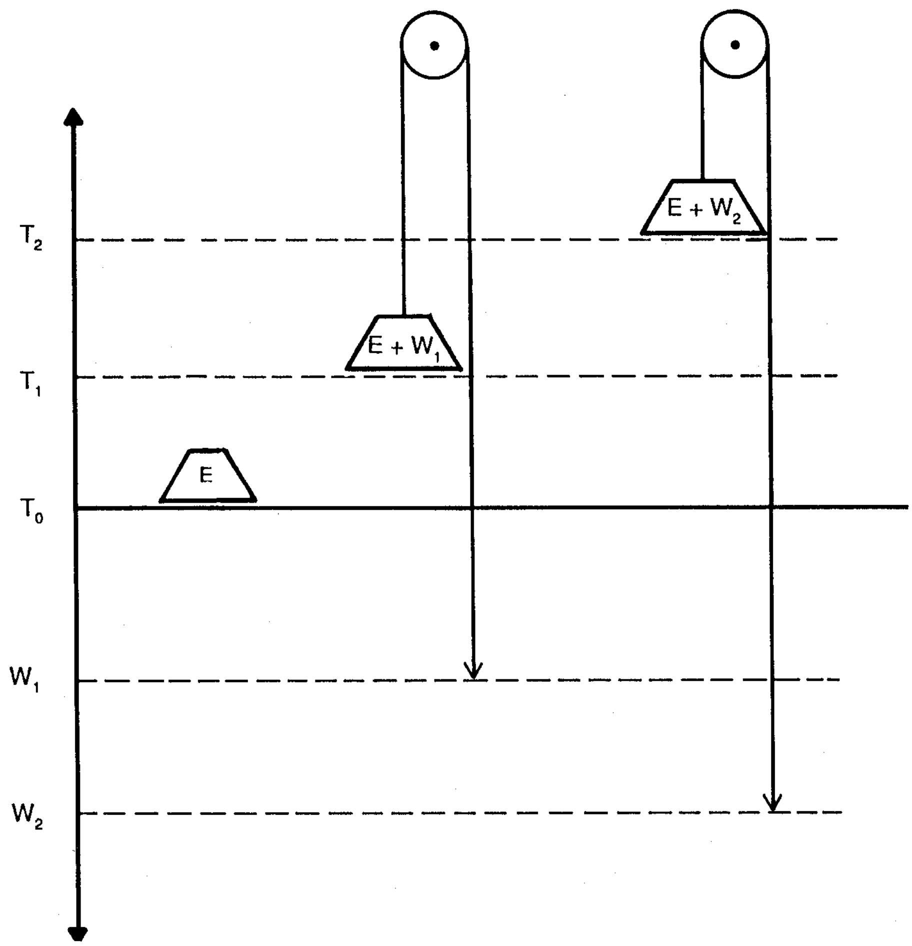


Figure 1-1. Heat Pulley Analogy

- A device to receive waste heat from the source
- A process that raises the temperature of the waste heat
- A device to deliver useful heat at a higher temperature

The types of systems that carry out these three steps are examined in Chapter 2. However, it is worth remembering here that these steps are the same as those required for refrigeration. The only difference is that the "low" temperature level is usually above ambient temperature for a heat pump while it is below ambient for refrigeration. Hence, the main distinction between refrigeration and heat pumping is one of temperature level. For this reason, many types of heat pumps operate in exactly the same way as refrigeration plants. Indeed in some cases the same machine can simultaneously provide refrigeration at "the cold end" and act as a heat pump at "the hot end." Many space heating heat pumps take advantage of the similarity between refrigeration and heat pumping to provide heat in the winter and cooling in the summer. Such dual purpose units can be extremely cost effective.

THE INDUSTRIAL HEAT PUMP

The majority of heat pumps are used for space heating or cooling; there are several million systems in use in domestic and commercial buildings. This Resource Guide is devoted to industrial systems which differ in many ways from the better known space heating/cooling devices.

It is difficult to define rigorously what is meant by an industrial heat pump. Inevitably there are many gray areas in which industrial and other types of systems overlap. However, in general terms, industrial heat pumps should satisfy two basic criteria:

- The heat pump must involve an industrial process stream as either the heat source or as the heat user (or heat sink). In many cases, both the source and user are process related.
- Also, the heat pump must be large enough to be "of industrial scale." A reasonable size to choose as an indicator of scale is a minimum heat output of 250,000 Btu/hr.

As explained later in the Guide, there is an enormous range of system types in the industrial field. This is partly because of the large variation in sizes (from 250,000 Btu/hr to at least 50 million Btu/hr heat output) and also because of the variety of temperature levels and fluids encountered in industry. The "low" temperature heat source could be near ambient temperature in one application while another will use source heat at 250°F. Similarly, the heat delivered by heat pumps can vary in temperature between 100°F and 300°F. The fluids that represent the heat source and heat user may be familiar ones such as air or water; however, many others are encountered including dirty or corrosive streams, condensing vapors and boiling liquids. The potential differences in application mean that the industrial heat pump designer must be much more flexible in approach than the equivalent space heating specialist. To illustrate the variety of systems, Table 1-1 shows some common applications.

THE IMPORTANCE OF THE HEAT PUMP

The importance of industrial heat pump technology must be examined from the viewpoint of both the user and the utilities.

Clearly, the user only has one thing in mind - to save money! In most industrial situations the benefits of a heat pump relate to energy savings, or more importantly energy cost savings. The heat pump has the potential to supply heat to a process at a lower cost than by conventional means. If this can be done with an adequate return on investment, then heat pumps will be considered important by the end user. Sometimes extra benefits can be identified to improve the economic case for a heat pump e.g., better product quality or improved raw materials recovery. Equally, there may be certain disadvantages (such as extra maintenance, less control or reduced operability) which the end user will wish to consider and should be made aware of.

From the perspective of the electric power utility the heat pump must also be seen as a way to sell additional electricity. Space conditioning heat pumps offer the possibility of providing an improved base load in winter, which may be important in those areas with a summer peak demand. This opportunity does not apply to industrial heat pumps because they will operate throughout the year. Additionally, in some circumstances, industrial heat pumps may be considered as an alternative to cogeneration; as such, they offer an important option for utilities.

Table 1-1

COMMON HEAT PUMP APPLICATIONS

Industrial Sector	Typical Applications
Lumber	Lumber Drying
Pulp and Paper	Evaporation of Pulping Liquors
Petrochemicals/Refining	Distillation
Food	Evaporation/Concentration of Dairy Products/Juices
	Wash/Process Water Heating
Chemicals	Evaporation/Concentration of Product and Waste Streams
	Process/Boiler Feed Water Heating

DIFFERENCES BETWEEN INDUSTRIAL PROCESS AND SPACE CONDITIONING APPLICATIONS

As stated above, the industrial heat pump does not produce the same load modifying opportunities as the residential/commercial heat pump. This is just one of many important differences between the two markets. It is important to recognize these differences as they affect design and marketing philosophies to be used.

The most important difference is related to "the air conditioning benefit." In general, heat pumps used in buildings are used as air conditioners in summer and space heaters in winter. The alternative to the combined heat pump/air conditioner is an air conditioner plus a separate furnace. The financial return on the small additional cost involved in converting an air conditioner into a dual purpose heat pump is extremely good. (It is worth noting that in most parts of Europe, where air conditioning is not required, space conditioning heat pumps are relatively unsuccessful.) In the industrial field the same benefits are rarely available and the heat pump must pay back its <u>full</u> capital cost by providing energy savings and other benefits.

Another major difference relates to scale of production. Because space heating applications are so similar, it is possible to offer a range of "packaged" units that can be mass produced. This is not true in the industrial field, mainly because each application requires some degree of customization.

In terms of technology, the large variety of potential applications means that industrial heat pumps utilize a far wider range of equipment types than space conditioning equivalents. There are also a number of more subtle differences. For example, the purchaser of an industrial system will often be an engineer; although he or she may not have direct heat pump experience, the purchaser will have many expectations regarding the specification of a heat pump system concerning factors such as reliability, safety, maintainability, etc.

Although there are a number of heat pump text books and resource guides, they all address the space conditioning system as the main topic. Little or no detail is given about industrial applications. Recently, however, both DOE and EPRI have sponsored work aimed specifically at industrial systems. One of the main aims of this Resource Guide is to examine properly all the technical and marketing issues which currently prevent the wider use of heat pumps in industrial applications.

THE INDUSTRIAL HEAT PUMP MARKET

In Chapter 3 of this guide the state of the industrial heat pump market is examined in some detail. Research has identified close to 1400 industrial heat pumps with a total heating capacity of 2.7 x 10^{10} Btu/h and a corresponding installed driver load of 439 MW. This compares with approximately 3 million space heating heat pumps in the domestic and commercial fields with a total installed load of about 10,000 MW.

It is useful to examine why the market penetration has been so low. The main reasons relate to lack of widespread knowledge about industrial heat pump systems and applications, high capital costs, the risks involved with unfamiliar technology, and, of course, economics. The financial benefit of heat pumps are related to the value of the energy they save. Clearly this is a function of both fuel and electricity pries. In general, high fuel prices will tend to favor heat pumps. Of course, high energy prices will also tend to favor all types of energy recovery schemes.

This economic effect is not the only reason for poor market penetration. In technical terms, many of the existing installations have been disappointing. In many cases, heat pumps have failed to reach design values of heating capacity or efficiency. This can be translated back into economic terms and generally means the heat pump failed to achieve the design payback period. Poor reliability has exacerbated the problem - many heat pumps achieve poor utilization factors due to frequent breakdowns, particularly in the first year of operation.

The key question is whether or not this poor technical reputation is a valid worry for the potential purchasers of new systems. The reputation represents the historical performance of existing systems, which can be thought of as "first generation" heat pumps. It is worth noting that a similarly disappointing reputation was established in Europe during the period 1975-85 when about 850 industrial systems were installed.

With the present state of knowledge about heat pumps there is no need for this poor technical reputation to continue. It is possible to build good "second generation" heat pumps that will meet design performance figures and operate reliably, providing that the technology is not oversold or wrongly applied. Many first generation heat pumps failed because too little effort was made at the feasibility and design stages. This is probably because systems were installed before they were sufficiently developed for operation in real industrial processes. Inevitably, mistakes were made and the poor reputation resulted. With hindsight, we know what many of these problems are and there is no need to repeat them.

The industrial heat pump market is at an interesting and somewhat precarious watershed. More than 10 years of first generation experience has been accumulated. Around 2,000 industrial systems have been installed worldwide not including several thousand 'lumber drying', and similar, units. A significant body of experience and knowledge is available. Second generation industrial heat pumps can be successful providing they fully benefit from the experiences gained in the last 10 years. However, it would be just as easy to see previous mistakes repeated, particularly by people who fail to understand properly the difference between the residential/commercial and industrial heat pump markets. The electric power utilities have an important role to play to ensure that second generation heat pumps are a success. This will be of benefit to the end user, the heat pump industry and the utilities themselves. By acting as an informed advisor, the utility sales engineer can help the industrial purchaser select a good system from a reputable company.

An unbiased view is particularly important. It is vital that the heat pump is not oversold. There are often better energy saving alternatives than heat pumps; in these situations, the industrial advisor must point the client toward the best alternative rather than the best heat pump! In the long term, a biased attitude in favor of electric heat pumps could have a severe negative effect on the potential market.

Recently Pinch Technology has been receiving a lot of interest from industrial energy users. As will be described in Chapter 4, it is a methodology that helps to find good heat pumping and other energy recovery projects. It assists in the comparison of alternatives, and can ensure unbiased advice.

RESOURCE GUIDE STRUCTURE

It is clear that great care must be taken to develop the industrial heat pump market effectively. Obviously, the specialist heat pumps designers and manufacturers will play an important role. However, people with a good knowledge of heat pumps, but also with a more independent position, will have an equally important role. They will act as "facilitators" who will advise industrial companies on the best energy saving options. As mentioned earlier, it is hoped that this Resource Guide will assist in educating those people who are in a position to help promote the technology. It may also help owners of existing heat pumps by suggesting ways in which the performance of these units can be improved. The first half of the guide gives background to the whole technology. In the second half, the steps in the feasibility and design stages of a potential heat pump application are addressed in detail.

Chapter 2, Technology Basics, gives a description of all the major types of heat pumps and their components. Important terms like Coefficient of Performance (COP) are defined.

Chapter 3, Market Basics, looks at existing U.S. and World-wide industrial heat pump usage and describes potential applications in different industrial sectors.

Chapter 4, The Industrial Heat Pump in Perspective, is aimed at examining how the heat pump fits into a set of competing technologies, and where it is likely to be the best solution. Pinch Technology is introduced as a powerful tool for identifying good heat pump applications.

Chapter 5, The Feasibility Stage, describes how to carry out an analysis of an industrial plant to identify heat pump opportunities and to prove their economic feasibility.

Chapter 6, The Design Stage, assumes a cost effective opportunity has been identified and then elaborates on the design procedure that is required to ensure the heat pump is well specified.

Chapter 7, Commissioning, Operation and Maintenance, discusses the important issues that arise after a heat pump has been installed.

Finally, Chapter 8, How to Make it Happen, examines how the information presented in this Resource Guide can be used to help turn opportunities into real installations.

Chapter 2

TECHNOLOGY BASICS

• CHAPTER SUMMARY

This chapter is divided into four principal parts which introduce the reader to industrial heat pump technology and the factors which influence the heat pump's performance.

A wide variety of systems and components are discussed. Absorption and thermocompression heat pumps are described as well as the more common vapor compression cycle and its open and closed cycle derivatives.

Concentrating on the latter type, the system components are described in some detail, and selection criteria given. The performance of a heat pump may be expressed in several ways, and these are defined and discussed. The operational limitations, both thermodynamic and practical, are described, and a concluding section gives information on trends in heat pump development which may significantly influence the market in the future.

INDUSTRIAL HEAT PUMP SYSTEMS

In the introductory chapter of this Resource Guide the heat pumping process was defined in general terms. The process of upgrading (i.e., raising the temperature of) waste heat was described in terms of the pulley analogy (see Figure 1-1). In more practical terms it was shown that heat pumps all perform three essential steps. These are:

- receiving low temperature waste heat
- raising the temperature of the waste heat
- delivering useful heat at a higher temperature

Figure 2-1 illustrates these steps and shows the three distinct fluid streams in many heat pumps [i.e., the source, the refrigerant and the user (or sink)]. The heat pump cycle itself receives a quantity of waste heat, Q_{in}, by passive heat exchange with the source. Energy is added to raise the temperature of the recovered energy. This is most commonly shaft power, W. The heat user then eceives the total energy input, $Q_{out} = Q_{in} + W$, again by passive heat exchange. As long as the temperature lift ($T_2 - T_1$) is not too high, then Q_{out} is significantly greater than W and good energy efficiency can be achieved. The performance of a heat pump is often defined by the ratio Q_{out}/W which is known as the Coefficient of Performance (COP). Industrial heat pumps generally have COP's in the range of 3 to 30 (although values outside this range are possible). COP and other heat pump performance parameters are discussed in more detail later in this chapter.

Now it is necessary to describe the types of systems that can actually perform these steps and to understand their advantages and disadvantages. There are four principal heat pump systems currently competing in the market. These are the closed and open cycle compression systems, the absorption cycle and thermocompression systems. Each of these basic systems is described below. They all have many design variants, some of which are described later in this chapter.

A number of other basic heat pump types (e.g., Brayton cycles) are sometimes quoted in the literature. It must be stressed that these are only at the R&D stage and are not in commercial production. Such systems are discussed briefly at the end of this section. The closed and open cycle compression systems are the most important industrial heat pumps for the readers of this guide because (a) they dominate the market place (more than 90% of industrial heat pumps are in this category) and (b) they can be driven by electricity (unlike the other cycles which are essentially heat driven).

Before describing the basic heat pump types, it is interesting to identify how they are able to raise the temperature of waste heat. In all four cases, the heat pumps rely on the fact that the boiling point of a fluid changes with pressure. For example, water boils at 212°F at atmospheric pressure but at higher temperatures when the pressure is raised (e.g., 250°F at 30 psia). Hence, a heat pump can absorb heat by boiling a fluid at a low pressure and then deliver heat at a higher temperature, by condensing the same fluid at a higher pressure. The

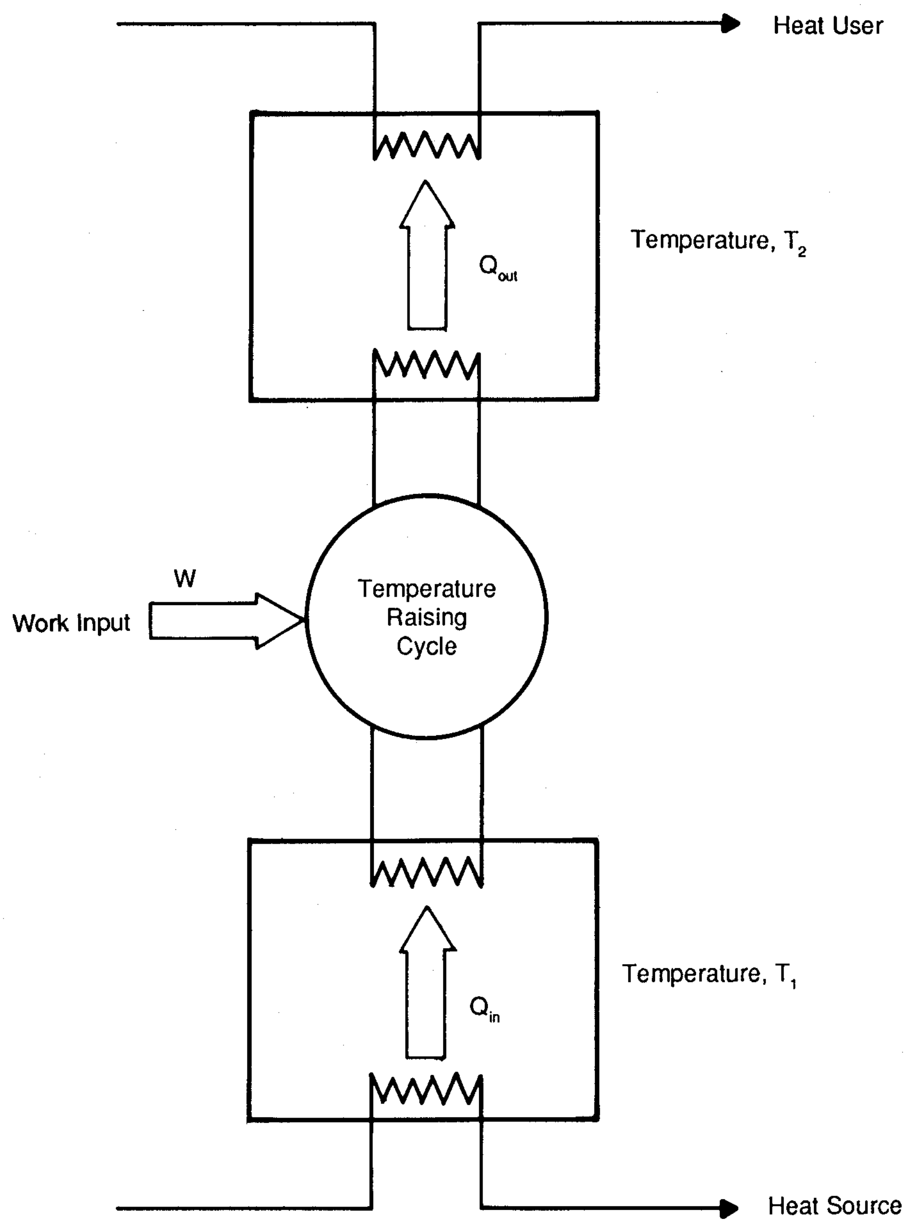


Figure 2-1. The Three Elements of a Heat Pump

heat pumps described below all utilize this principle. The differences between them are:

(a) Whether a mechanical drive (requiring shaft power) or a thermal drive (requiring heat) is used; and

(b) whether a refrigerant fluid is reused or not.

It should be noted that the definition of COP given earlier in this section only applies to mechanically driven heat pumps. A more appropriate measure of performance for thermally driven systems (i.e. absorption, thermo compression, and engine driven) is Primary Energy Ratio (PER). This is also defined later in this chapter.

<u>The Closed Cycle Vapor Compression Heat Pump</u>

The basic closed cycle vapor compression heat pump in is shown in Figure 2-2. This heat pump type is very similar to most refrigeration cycles and the majority of space heating heat pumps. Compression of the refrigerant is achieved with a mechanical compressor. The phrase 'closed cycle' means that the refrigerant circuit is physically separated from the source and user streams, and that the refrigerant fluid is reused in a cyclical fashion.

The four main components are (i) the evaporator, where heat is extracted from the heat source (e.g., a low temperature waste heat stream) to boil a refrigerant; (ii) the compressor and its drive, where the refrigerant is compressed, thus raising its pressure and temperature; (iii) the condenser, where the refrigerant delivers the heat taken in at the evaporator (plus the heat equivalent of the work of compression), and (iv) the throttling valve where the refrigerant pressure is reduced back to the evaporator condition. The refrigerant itself is also an important element of the system. All of the components are described in more detail later in this chapter.

<u>The Open Cycle Vapor Compression Heat Pump</u>

The open cycle vapor compression heat pump is illustrated in Figure 2-3. This type also uses a mechanical compressor; however, instead of employing a separate refrigerant, the process vapor itself is compressed and raised in temperature directly. In other words, the process vapor acts as both the waste heat stream and the heat pump working fluid. The high pressure vapor can then be reused

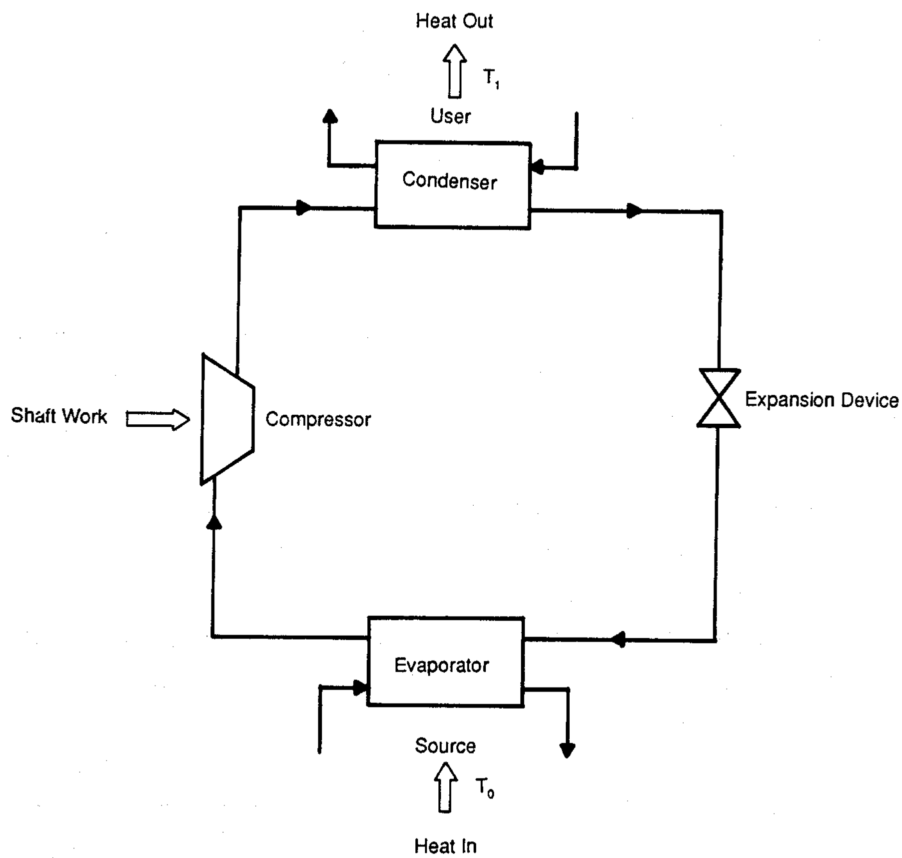


Figure 2-2. Closed Cycle Vapor Compression Cycle

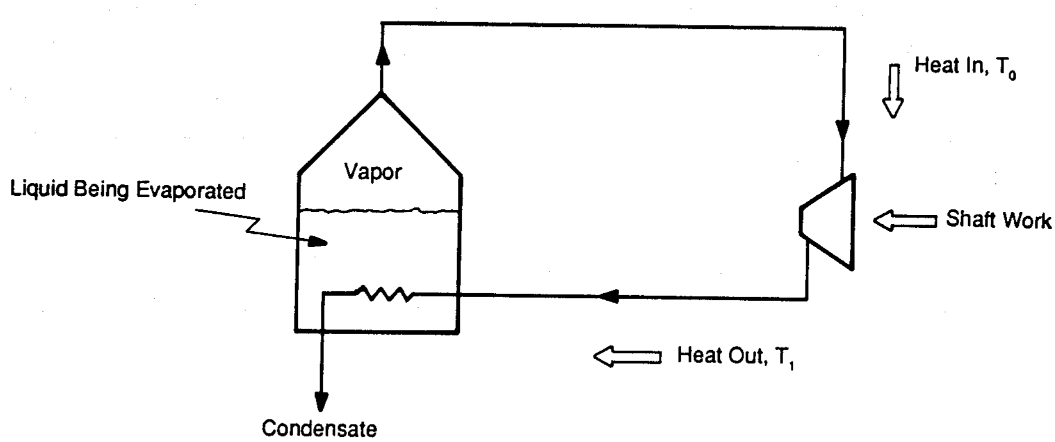


Figure 2-3. Open Cycle Vapor Compression Cycle

directly or, more commonly, condensed in an exchanger to supply heat to another process stream, as shown in Figure 2-3. This configuration is often known as semi-open. These open cycle variants are discussed later in this section. The term 'mechanical vapor recompression' (MVR) is frequently used to describe open cycle heat pumps, particularly in the context of evaporation and distillation plants.

Absorption Cycle Heat Pumps

Absorption cycle heat pumps differ from mechanical vapor compression types in that the mechanical energy input is normally very small, and the principal external energy supply is in the form of high temperature heat. The major components of the absorption cycle are shown in Figure 2-4. The absorption cycle is a closed cycle requiring two working fluids, a refrigerant and an absorbent. The mechanical compressor is replaced by two heat exchangers (the absorber and the generator) and a liquid pump. In operation, waste heat is supplied to the evaporator. This evaporates the refrigerant in the same way as in a closed cycle mechanical compression heat pump. The refrigerant vapor is then absorbed at low pressure by the absorbent, generating useful medium temperature heat. The absorbent, now diluted with refrigerant, is raised in pressure using a liquid pump. High pressure refrigerant vapor is then produced by heating the mixture in the generator with high temperature heat. This refrigerant vapor is condensed in the condenser, also providing useful medium temperature heat, before passing back to the evaporator. Meanwhile, the reconcentrated absorbent is returned to the absorber.

The generator high temperature heat input is the major prime energy consumer, unlike the mechanical vapor compression cycle where the compressor power requirement is the major prime energy consumer. In the absorption cycle, the pump power is very small because the liquid mixture leaving the absorber is virtually incompressible. It is also important to note that there are two medium temperature level heat outputs, the absorber and the condenser.

Various refrigerant/absorbent pairs are used in absorption heat pumps. A water/lithium bromide combination is most commonly found in industrial process applications, water being the refrigerant.

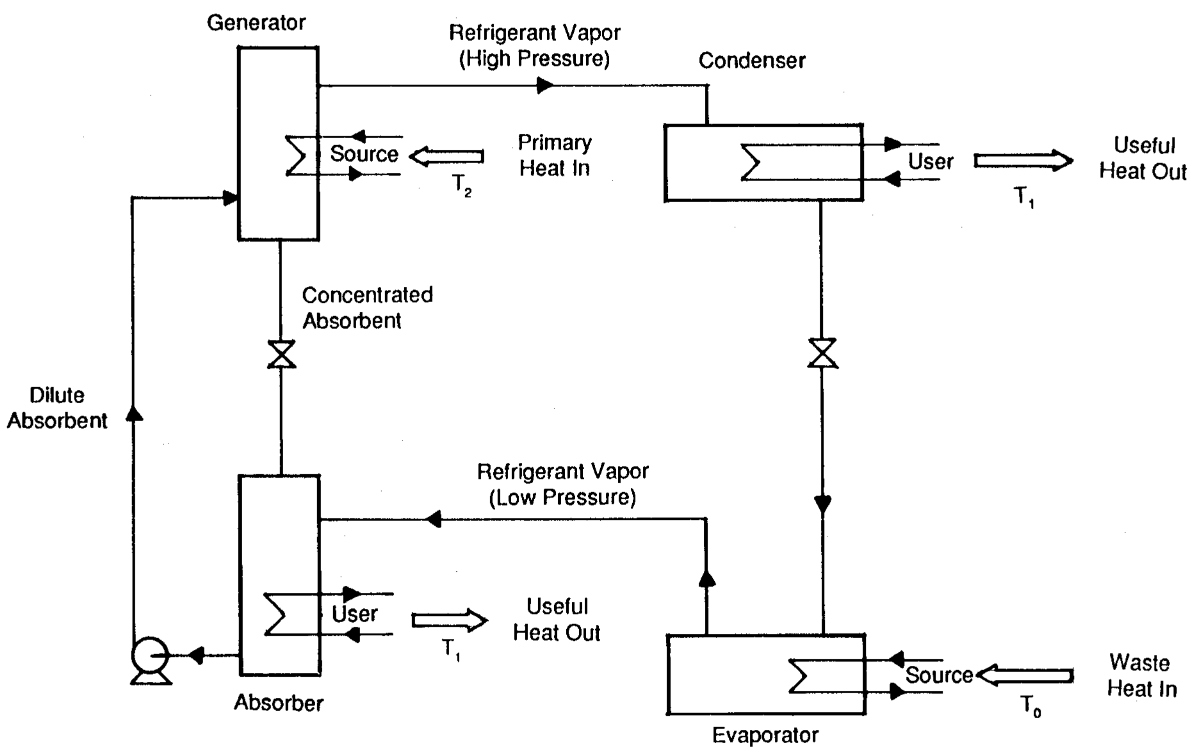


Figure 2-4. Absorption Cycle

Thermocompression Heat Pumps

The thermocompression system is similar to open cycle vapor compression except that no mechanical energy input is required. Thermocompressors, sometimes known as jet compressors, vapor jet heat pumps, or steam ejectors, use the kinetic energy of a vapor jet, normally steam, to compress a lower pressure vapor stream. No mechanical energy is needed to effect compression. The way in which the pressure of the refrigerant is raised is shown in Figure 2-5. In principle this type of compressor is very practical for heat pump applications because it is so simple and reliable. However, in most applications the quantity of steam required to drive the compressor is much greater than that being compressed. This restricts thermocompression heat pumps to systems with a small heat source and a large heat user. Also, the performance of a thermocompressor can fall dramatically at off design conditions making these systems more suitable for processes with steady operating conditions.

IMPORTANT SYSTEM VARIANTS

The four heat pump systems described above were outlined in their most basic forms. There are many variations that can be engineered into real systems to take advantage of site specific circumstances. Such system variants are very important when considering the potential for industrial systems because it is often possible to improve the heat pump performance by more than 25% by using a non-standard system design.

Use of Refrigerant Subcooling

One on the simplest and most effective variants to the closed cycle vapor compression system is the addition of a refrigerant liquid subcooler as shown in Figure 2-6. This extra heat exchanger cools the liquid refrigerant after it has left the condenser. The benefits are best illustrated with an example. If we use a heat source at 50°F to heat water from 60°F to 140°F the coefficient of performance of a basic closed cycle vapor compression cycle is approximately 3. By preheating the water from 60°F to 79°F in a subcooler, the COP improves by 33%. This is achieved because we obtain 'free' heat from the liquid refrigerant coming out of the condenser (note, without a subcooler the hot condensed liquid is cooled by internal flashing in a wasteful manner). Although the heat exchanger cost goes

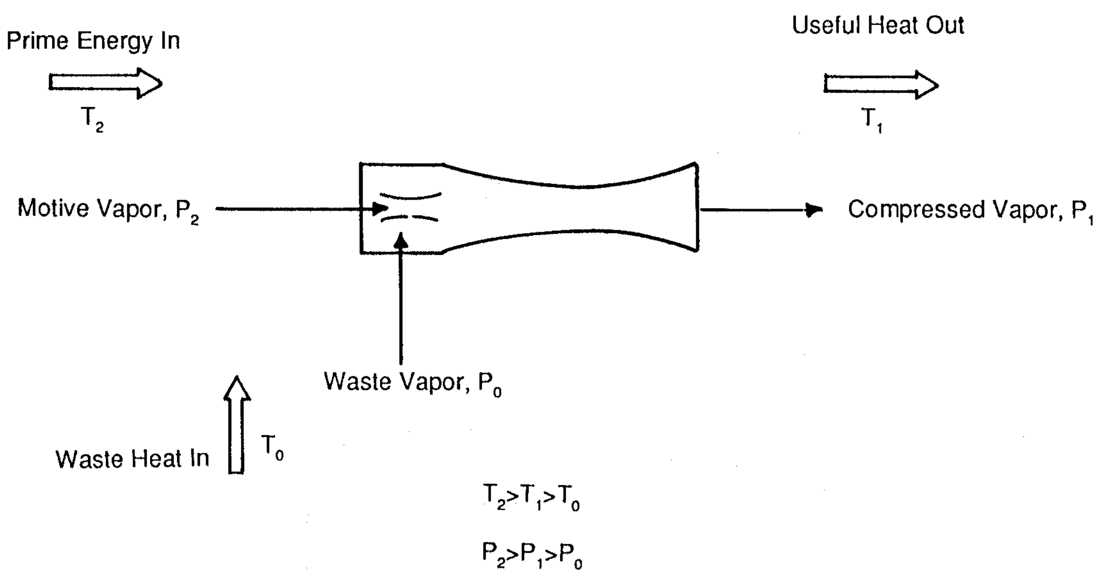


Figure 2-5. Thermocompression Cycle

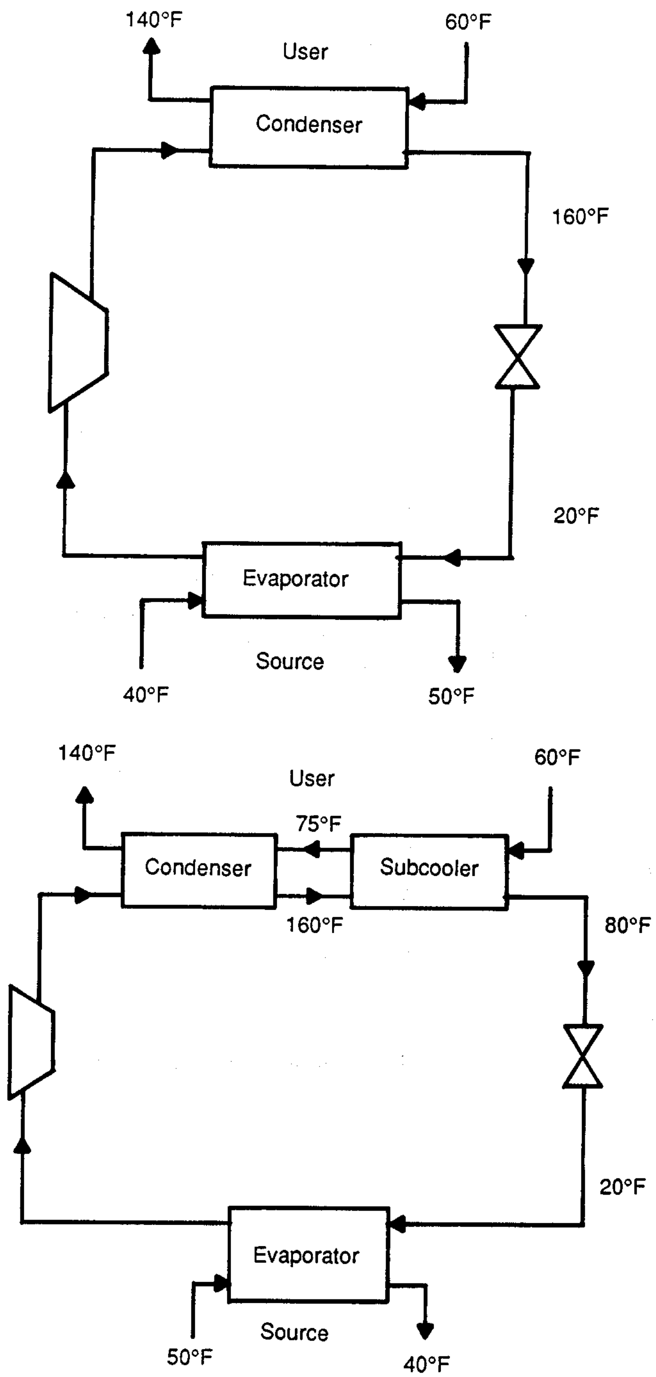


Figure 2-6. Refrigerant Subcooling Cycle

up because of the addition of a subcooler, the compressor size will fall by about 30% in this case.

The use of refrigerant subcooling is particularly relevant in two situations:

(a) when the heat user is heated through a large temperature range, as in the example above. (See Chapter 5 for more details.)

(b) if a lower temperature secondary heat user is available (e.g., a heat pump heating water from 130°F to 140°F could not use subcooling unless a separate stream needs heating from, say, 60°F to 80°F).

These observations are special cases of a more general set of rules for determining when refrigerant subcooling is appropriate. These rules are derived from Pinch Technology which is described in Chapter 4.

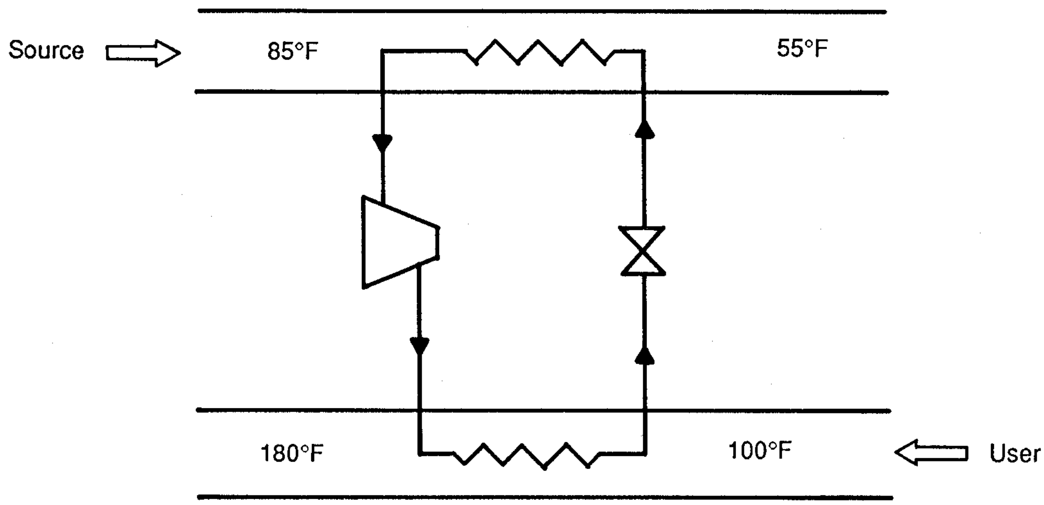
Cascaded Heat Pumps

Another way of taking advantage of large heating temperature ranges is to use more than one heat pump in series or in "cascade." This is illustrated in Figure 2-7. With a single heat pump the temperature lift is 145°F (allowing for 10°F temperature differences in the evaporator and condenser). By installing 2 smaller units in cascade, the temperature lift is only 130°F for unit 1 and 105°F for unit 2. In both cases, this will give better performance than for a single unit.

In many large installations, customers will require more than one unit for control and reliability purposes. By arranging the units in cascade as illustrated here substantial benefits can be gained.

Refrigerant Desuperheaters

A small proportion of the energy delivered by a closed cycle heat pump (usually about 10%) is available at temperatures well above the condensing temperature. The energy is available in the form of gas superheat at the compressor exit. If a desuperheater is used before the condenser it may be possible to slightly reduce the heat pump temperature lift and hence improve the COP. It must be noted, however, that this superheat energy is recovered under normal circumstances in the condenser itself. The benefit is simply to reduce the condensing temperature. This is quite different to the subcooler described above, which recovers extra energy that would otherwise be wasted in the expansion valve. Again, Pinch



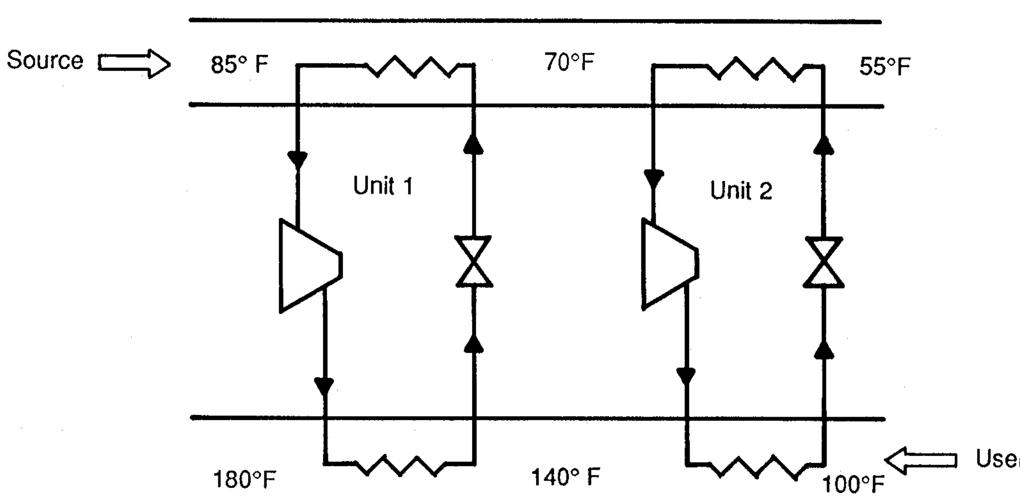


Figure 2-7. Horizontal Cascade Cycle

Technology provides rules for determining where the desuperheating heat should be used. (This is also described in Chapter 4.)

Open Cycle Variants

There is some confusion over terminology regarding different open cycle configurations. Figure 2-8 shows the three main types of open cycle variant. Type (a) is the truly open cycle. It simply consists of a compressor operating directly on a waste vapor and delivering high pressure vapor into a process. This configuration is actually quite unusual; most open cycles involve at least one heat exchanger.

Type (b) shows the most usual variant. Waste vapor is compressed and then condensed in a heat exchanger. This type of arrangement can be considered a semi-open cycle although many people think of it simply as an open cycle.

Type (c) shows an alternative semi-open cycle. In this case the heat source is used to boil a liquid in a heat exchanger. This vapor is then compressed and supplies heat to the user either directly or indirectly. In the latter case, the condensed liquid is not reused (if it were supplied back to the evaporator the cycle would become closed). This variant is useful when the heat source is contaminated.

Absorption Cycle Variants

The absorption cycle can operate in two quite different ways. These are illustrated in Figure 2-9. Type I operation is known as a Heat Amplifier. In this mode, there are two heat inputs; a low temperature heat source and a high temperature prime energy source (e.g., steam or direct gas firing). The useful heat output is at a temperature between the source and prime energy levels.

Type II operation is known as a Temperature Amplifier (or sometimes a Heat Transformer). In this mode, there is only one heat input which is always waste heat at a medium temperature level. There are two output levels, one at a temperature higher than the waste heat source and the other rejected at a low temperature into cooling water. Hence, in this mode, the heat pump requires no primary energy input (i.e., it costs nothing to run). Approximately 40% of the waste heat can be upgraded in temperature by about 100°F.

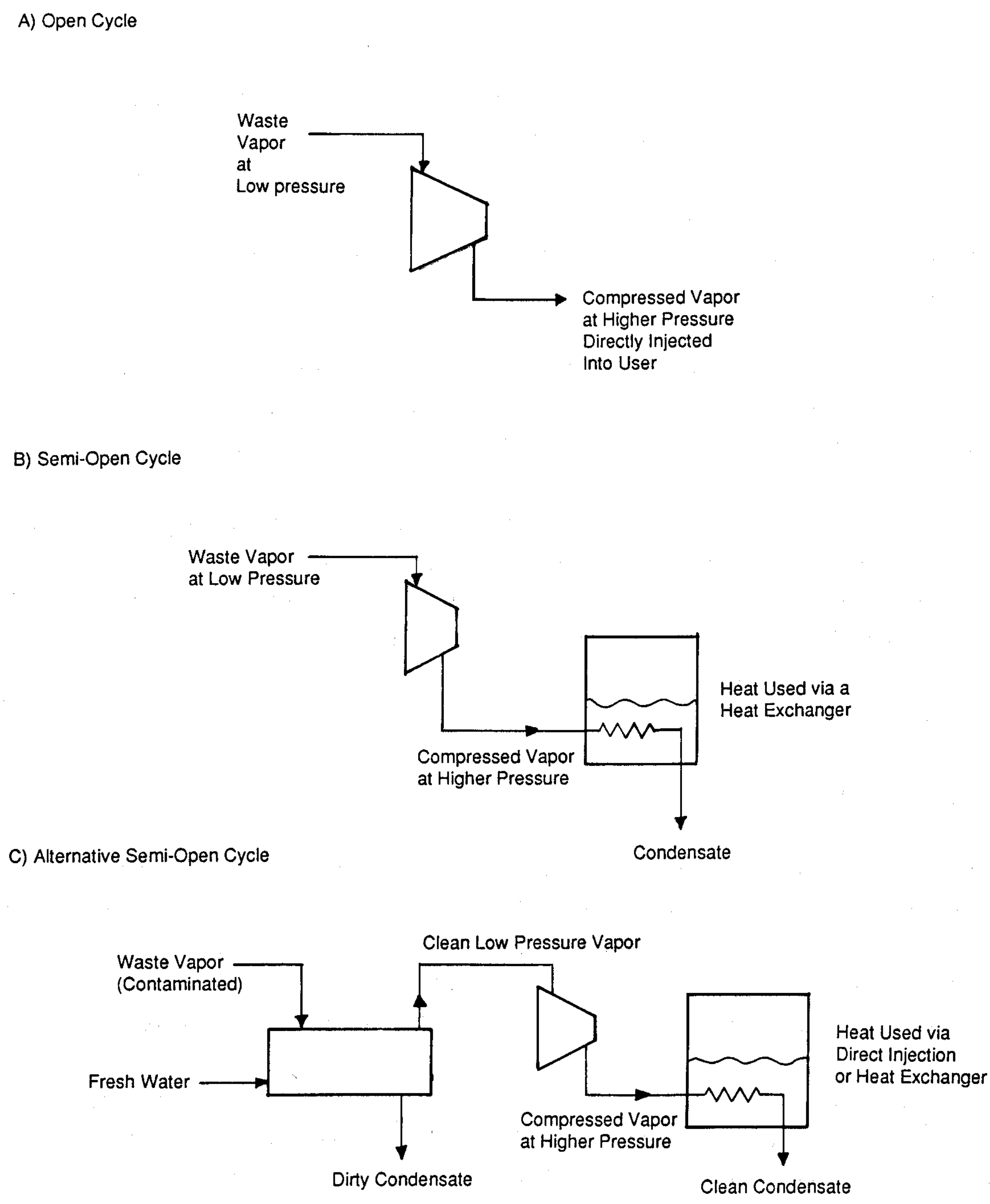
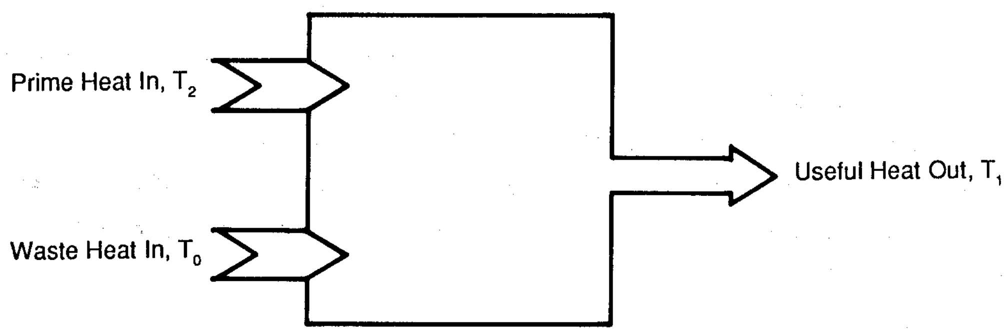


Figure 2-8. Open and Semi-Open Cycles

Type 1 - Heat Amplifier



Type 2 - Temperature Amplifier

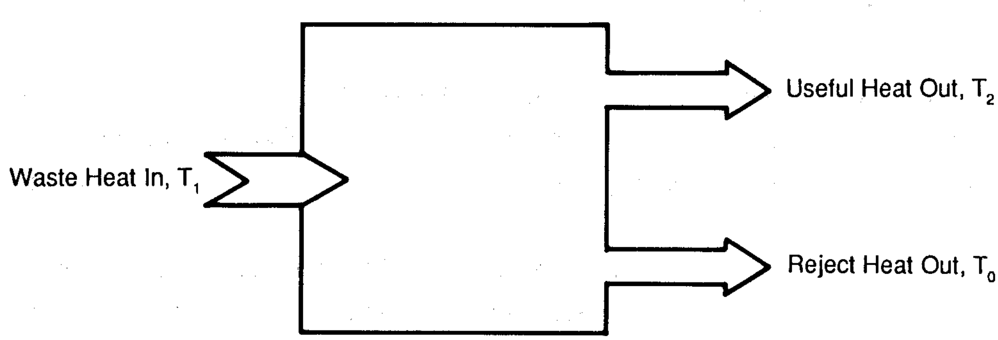


Figure 2-9. Absorption Cycle Types 1 and 2

Drive Variants

The power required to drive the compressor of closed or open cycle vapor compression heat pumps can come from a number of sources. The most common drive is the electric motor, but particularly in industrial applications, other methods are being used. These include Diesel and natural gas internal combustion engines and steam turbines.

Electric motors have relatively low capital and maintenance costs, are simple and compact. Some heat pumps use compressors which contain the electric motor within the same casing - such hermetic systems utilize refrigerant cooling of the electric motor. Alternatively, an open drive may be used, via belts, a gearbox or direct coupling. The applicability of electric motors covers the complete compressor type and size range.

After electric motors, the most commonly used prime movers are gas or Diesel engines. In addition to producing shaft power, they reject most of the balance of the fossil fuel input in the form of exhaust gases and jacket cooling water. The exhaust gases are at a relatively high temperature, while the engine cooling circuit produces useful heat in the range 160°F to 240°F. Single phase water cooling or ebullient (two-phase) cooling can be employed.

Effective use of these heat sources and, in larger engines, heat from the oil cooler means that up to 80% of the primary energy input can be used for process or space heating applications. A typical energy balance is shown in Figure 2-10 for an engine fueled with natural gas.

However, this obvious advantage is countered by the higher cost of such engines, particularly maintenance costs. The relative price of electricity and Diesel oil or natural gas also has a strong influence on the attractiveness of such systems.

The use of a steam turbine as a prime mover is often more practical than internal combustion engines as maintenance costs are much lower. However the use of steam turbines assumes a high pressure steam source is available and that the relatively large amount of low pressure pass out steam can be used effectively. As with thermocompressors this limits applicability. In general non-electric drives are only worth considering as heat pumps with relatively low COP (less than about 8). If COP's are higher than this the benefit is relatively small and it cannot offset

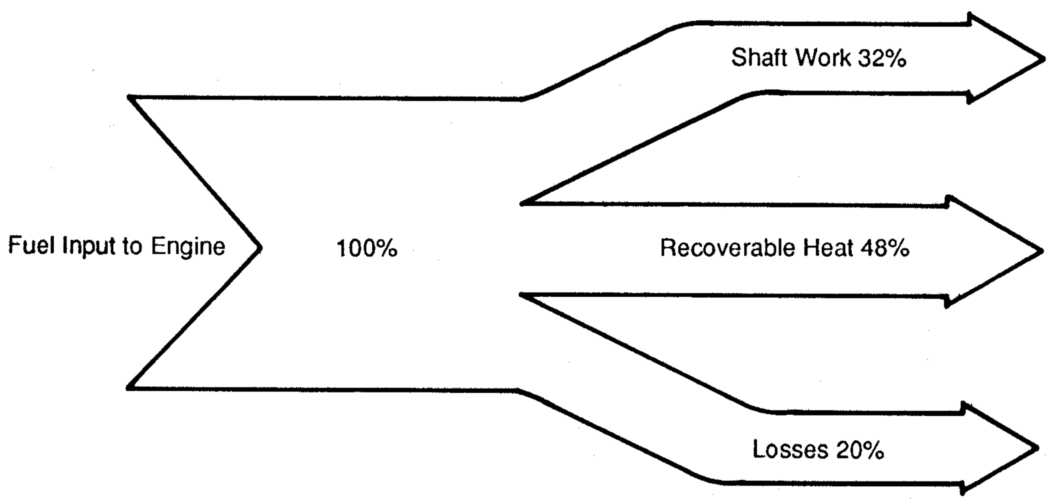


Figure 2-10. Energy Output from Gas Engine

the extra maintenance cost. This means that most MVR systems will be electrically driven.

HEAT PUMP PERFORMANCE PARAMETERS

There are a number of parameters which are used to compare the performance of heat pumps or their components. The most commonly used is the Coefficient of Performance, (COP). Others include the Primary Energy Ratio (PER), and the capacity. The efficiency of compression is critical in ensuring high values of these overall performance criteria, and may also be regarded as a basis for comparison.

Coefficient of Performance, COP

The COP of a mechanical compression heat pump is defined as the ratio of the heat output to the power input. A heat pump delivering 4 MMBtu/h and driven by a 1 MMBtu/h (293 kW) electric motor, has a COP of 4. It is important to make a distinction here between the COP used by heat pump designers and that used by refrigeration engineers. Referring back to Figure 2-1 the COP of the heat pump may be defined as:

$$COP = Q_{out}/W$$

However, when considering the performance of a refrigeration system, the heat output is not usually of prime interest. It is the heat input (i.e. the cooling duty) that is of relevance. Hence, refrigeration COP is defined as the ratio of heat input to power input, i.e.:

$$COP_R = Q_{in}/W$$

By applying a simple energy balance it is clear that Q_{out} is the sum of Q_{in} + W. Using this fact in conjunction with the two above equations one can show:

$$COP = COP_R + 1$$

COP is often found to be a difficult concept to understand. Most engineers are familiar with efficiency measurements on plants such as boilers and they know that values over 100% are impossible. A COP of 4 implies a 400% efficiency which seems to defy the laws of thermodynamics! Of course, this is not the case; the delivered heat is made up of some "free" heat from the heat source and some purchased energy (to drive the compressor). As COP rises the proportion of free heat is increasing. With very low temperature lifts (e.g., 10°F in an open cycle evaporator) COP's in the region of 50 are possible!

Care must be taken in using COP as a comparative parameter. It usually does not include auxiliary power for things like circulating pumps and fans; these often add significantly to the running cost. Also it makes no allowance for the fact that the heat being supplied and the power being used are worth different amounts. Finally, it is not possible to deduce anything about capital cost from the COP.

Primary Energy Ratio, PER

The COP, while being a popular expression of heat pump effectiveness, does not take into account the fact that energy available as work is normally more valuable than energy available as heat. The PER recognizes not only the heat pump COP, but also the efficiency of conversion of the primary fuel (e.g. oil, gas or coal). It is defined as:

$$PER = \frac{\text{Useful heat delivered}}{\text{Primary energy consumed}}$$

When a heat engine with a thermal efficiency of η is used to drive a heat pump compressor,

$$PER = \eta \times COP.$$

Taking the example quoted above if the electric motor was replaced by a reciprocating natural gas engine with 35% efficiency, the PER is:

$$PER = 0.35 \times 4.0 = 1.4.$$

This means that the heat pump would deliver 40% more heat than by direct combustion of the fuel at 100% efficiency.

The use of PER also enables one to account for heat inputs to the system from the engine cooling jacket and exhaust. The engine described above rejects 65% of the fuel input as waste heat. It is not practical to recover all this waste heat, but 45% of fuel input is recoverable. This can supplement the heat pump output and PER becomes:

$$PER = 1.4 + 0.45 = 1.85$$

It can be seen that in such a case the heat pump would supply just over twice as much heat as a high efficiency boiler, which would have a PER of about 0.9.

This definition is also used to define the performance of absorption and thermocompression heat pumps. A typical Type I absorption system PER is 1.7. This implies that for every 0.7 units of waste heat available, 1.0 units of prime energy are consumed. This results in 1.7 units of useful heat being supplied.

Cost of Heat Supplied

It is important to understand the COP and the PER as these parameters are often encountered. However, when carrying out a heat pump feasibility analysis it is much simpler to express the performance simply as an operating cost in $/MMBtu. In this way all auxiliary power costs can be included and different fuel costs are properly accounted for.

Capacity

The capacity of a heat pump system is conventionally defined as the amount of heat delivered to the user. This is normally expressed in kW, MW or Btu/h, not, as is common in refrigeration, as 'tons of refrigeration'.

Compression Efficiency

The design of a heat pump, be it open or closed cycle, is dominated by the compressor. Compressor types are discussed later in this section, but it is appropriate to define compression efficiency here. More correctly, three different efficiencies need to be defined, and two of these have to be taken into account when calculating the COP of a heat pump.

Isentropic Efficiency. The isentropic efficiency represents the difference between a real compressor and one which operates under ideal or isentropic conditions. The deviation from ideal is principally due to irreversibilities, such as heat transfer between the refrigerant and the compressor components, in the fluid flow through the compressor.

Isentropic efficiency is a complex function of machine design, refrigerant, and operating conditions (suction and discharge pressure, compressor loading, etc.). Large variations are common between different machines and even the same machine operating under different conditions.

A good value of isentropic efficiency is above 75%. Many machines are on the market with a best efficiency point of only 50%. These will cost one third more to operate than good machines. Part load efficiency of most compressors is poor. Some machines operate efficiently over a wide range of suction and delivery pressures while others have a high peak efficiency that quickly falls at off-design conditions.

Selection of a compressor with good isentropic efficiency characteristics is essential if a heat pump is to have good economics. Relatively little attention is given to compressor efficiency by refrigeration plant users. Bearing in mind that most heat pump compressors are designed for use in chilling or freezing, there is a good chance of choosing an inefficient machine! Figure 2-11 illustrates a good way of comparing compressor efficiency. Efficiency contours are plotted on axes representing evaporating temperature and condensing temperature (roughly equivalent to source and user temperatures). Compressor (a) has the highest efficiency of either machine, at 80%. However this value falls at off design conditions. Compressor (b) would be a better choice if the source and user temperatures are lower.

Mechanical Efficiency. The mechanical efficiency of a compressor is a measure of the amount of work applied to the shaft which is eventually delivered to the fluid. This takes into account the mechanical losses of the drive, and other moving parts. The energy needed to circulate the lubricant is also taken into consideration in arriving at this figure. The mechanical efficiency is normally high, 95% is typical.

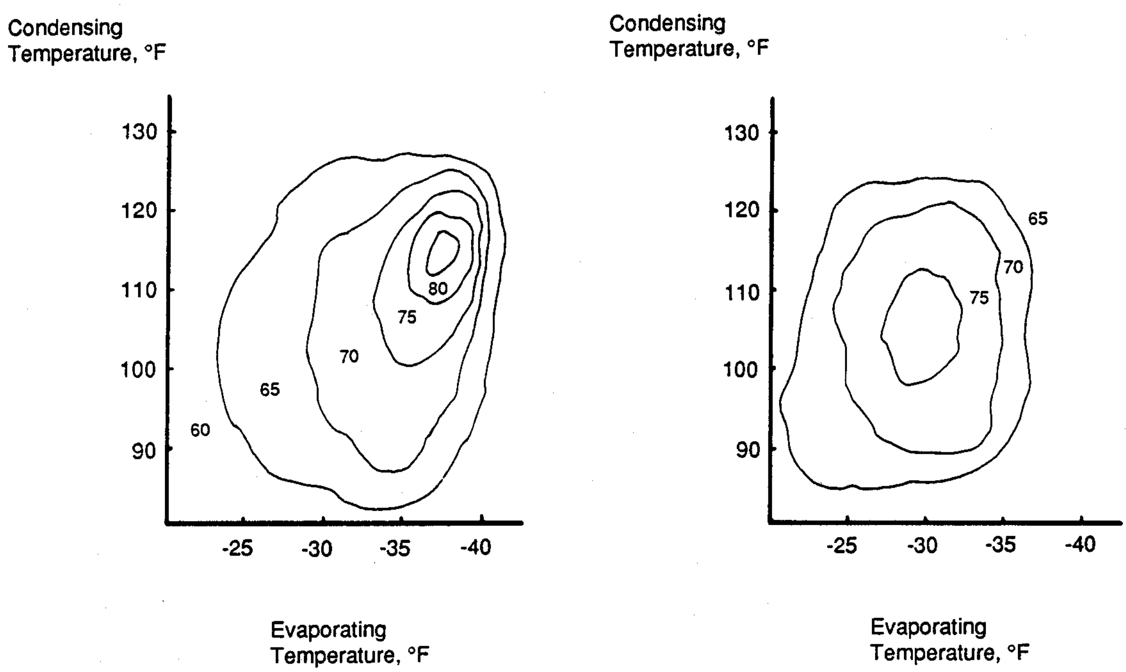


Figure 2-11. Compressor Efficiency Map

Volumetric Efficiency. The volumetric efficiency is defined as the ratio of the actual volume induced per cycle to the swept volume. Volumetric efficiencies vary, depending on the type of compressor and its design, but typically fall within the range of 85% to 95%.

INDUSTRIAL HEAT PUMP COMPONENTS

Heat pump systems were reviewed in the first half of this Chapter. Now it is useful to examine some of the individual components that make up such systems. These include the compressor and its drive, several heat exchangers (an evaporator, condenser, and subcooler would be typical), expansion valves and a refrigerant. Each of these components is discussed in the context of their use in industrial heat pump systems.

Compressors

The compressor may be regarded as the heart of a heat pump, and is critical to the determination of the performance, life, reliability and environmental acceptability of the system. While compressors for refrigeration duties are well proven, the direct use of refrigeration compressors for heat pump applications is not always advisable. At the outset it is useful to list some of the features of heat pump operation which set it apart from most refrigeration duties.

- First of all, heat pumps often operate at higher condensing temperatures and pressures than refrigeration systems. This puts more stress on both the refrigerant and the compressor.
- In the second place, heat pumps in industrial processes may operate for many more hours per year at their maximum design duty than refrigerators or air conditioning plants.
- Finally, a refrigeration compressor is designed to lose heat to the environment. A heat pump compressor preferably retains as much heat as possible for delivery at the condenser.

Compressor Categories. Compressors can be categorized in three ways; by type, by enclosure, and by oil usage. There are many basic types of compressor, although only four are frequently used for heat pumps. These are reciprocating, centrifugal, screw and rotary vane types. These are discussed in detail later.

Each type can have three types of enclosures known as hermetic, semi-hermetic or open. Hermetic compressors together with their motors are encased in a welded shell; this prevents any possible leakage from a rotating shaft seal. The electric motor is cooled by passing the refrigerant vapor over it prior to compression. Hermetic compressors are usually very small. Semi-hermetic machines are also fully enclosed but instead of a welded casing, a flanged casing is used allowing access to the compressor for servicing. The open machine is a bare shaft compressor with a rotating shaft seal. The electric motor is external to the compressor. Open machines generally have higher efficiency than hermetic machines, but are more liable to refrigerant leakage.

The third way of categorizing compressors relates to lubricating oil. Most compressors are "wet" in the sense that oil is allowed to come in contact with refrigerant (in fact some compressors utilize oil injected into the compression space for cooling and to improve efficiency). The other type is the oil free or "dry" compressor. Use of oil is restricted to items such as bearings which are sealed from the refrigerant.

Centrifugal Compressors. The centrifugal compressor is used by many refrigeration plant manufacturers and, unlike other types described here, relies not on positive displacement, but fluid acceleration in the rotor to raise the pressure. Single-stage (see Figure 2-12) or multi-stage centrifugal compressors of hermetic or open design are readily available. Such compressors are classed as dry since oil is of no benefit to the refrigerant side of the system.

Centrifugal machines handle pressure ratios up to about two in a single stage, and are more suited to large capacities and relatively small temperature lifts. Mechanical vapor recompression systems on evaporators, for example, routinely use single stage centrifugal compressors. They often have a high peak efficiency which drops rapidly at off design conditions as illustrated in Figure 2-11a. In many situations the compressor cannot operate too far away from design conditions because of compressor surge. This is caused by stalling inside the rotor and can have catastrophic effects such as motor burnout or compressor motor failure. Capacity control is normally carried out by varying the position of the inlet guide vanes, and hence the flow of the suction vapor. Variable speed drives are also used occasionally. Part load efficiency is often very poor.

Recently, high efficiency (80 + %), lower cost turbo fans have been developed for low pressure ratio steam compression duties up to 1.3. These are particularly suited evaporation systems.

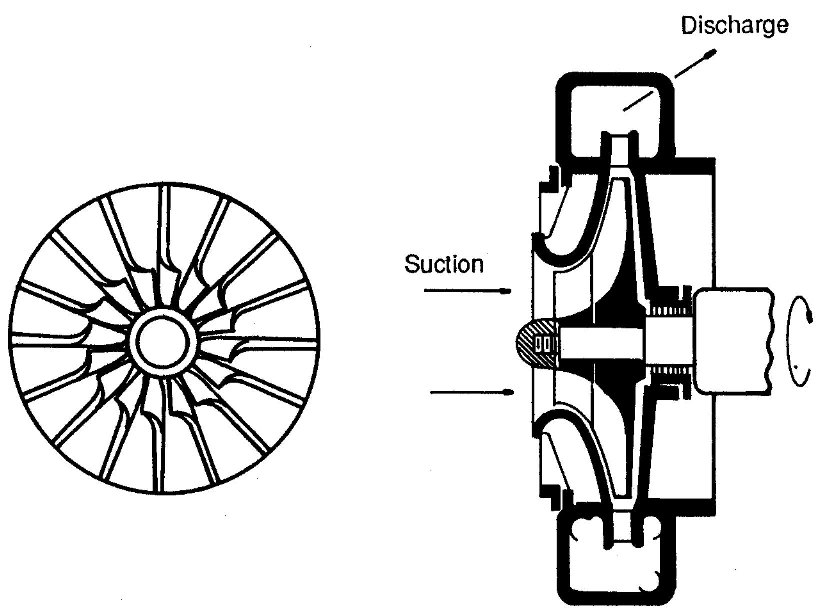


Figure 2-12. Centrifugal Compressor

Axial Compressors. Axial compressors can handle much larger vapor volumes than centrifugal machines. Unlike centrifugal machines, the vapor flow is parallel to the compressor shaft. A typical single stage compression ratio is 1.5, with efficiencies in the range of 80-85%. They tend to be smaller and lighter than comparable centrifugal units and have been extensively developed for use in gas turbines.

Reciprocating Compressors. Reciprocating compressors remain the most common type in the small to medium size range, being regarded as the 'work horse' of refrigeration systems. The basic design is simple and efficiencies are reasonable when operated at correct conditions. Reciprocating units are available in hermetic, semi-hermetic or open form. A typical open reciprocating compressor is illustrated in Figure 2-13. Size is not a constraint on reciprocating machinery operation, but it is normally found that centrifugal or screw compressors take over when drives reach 100 - 150kW. In the longer term, rotary compressor types may also dominate at lower powers.

In heat pump applications, reciprocating compressors are oil-lubricated, oil being fed from the pump to the crankshaft and cylinders. Substantial quantities of oil are brought into contact with the refrigerant, and effective separation of oil and refrigerant is needed.

Capacity control of reciprocating compressors is usually by unloading cylinders. This is effected by lifting suction valves from their seats on appropriate cylinders, using hydraulics or gas pressure. Thus, step load reductions can be achieved, e.g., 75%, 50% and 25% of full load.

Screw Compressors. Two types of screw compressor are competing in the market for open and closed cycle heat pumps. These are the single screw and twin screw types. The single screw machine is shown in Figure 2-14a, while the twin screw compressor is illustrated in Figure 2-14b.

Both types of screw compressor are positive displacement machines. To help understand their operation, it is useful to envisage the female rotor as a number of cylinders wrapped in the form of a helix. The vapor is drawn into the "cylinder" through the suction port and compressed by the "piston" which is a tooth of the star wheel in a single screw compressor or a lobe of the male rotor in the

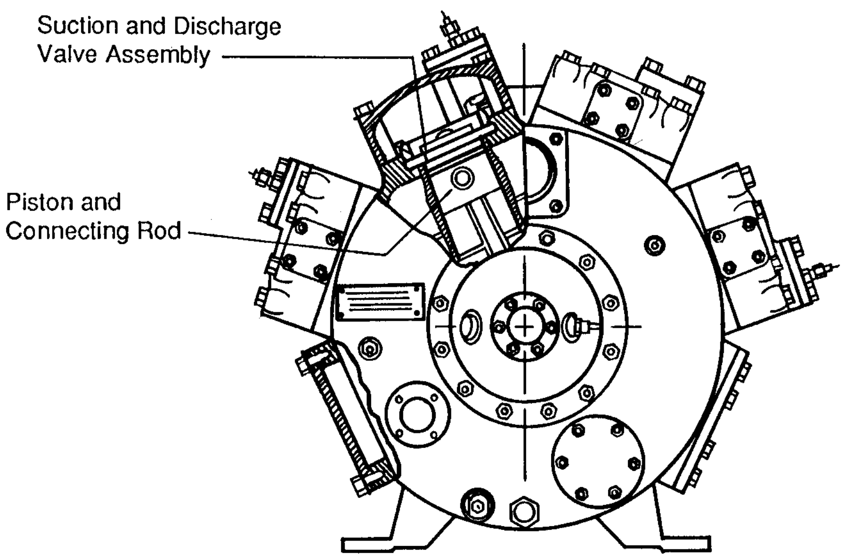


Figure 2-13. Open Reciprocating Compressor

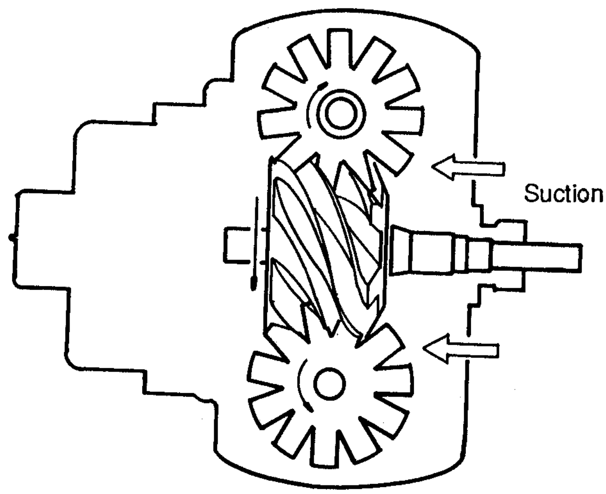


Figure 2-14a. Single Screw Compressor

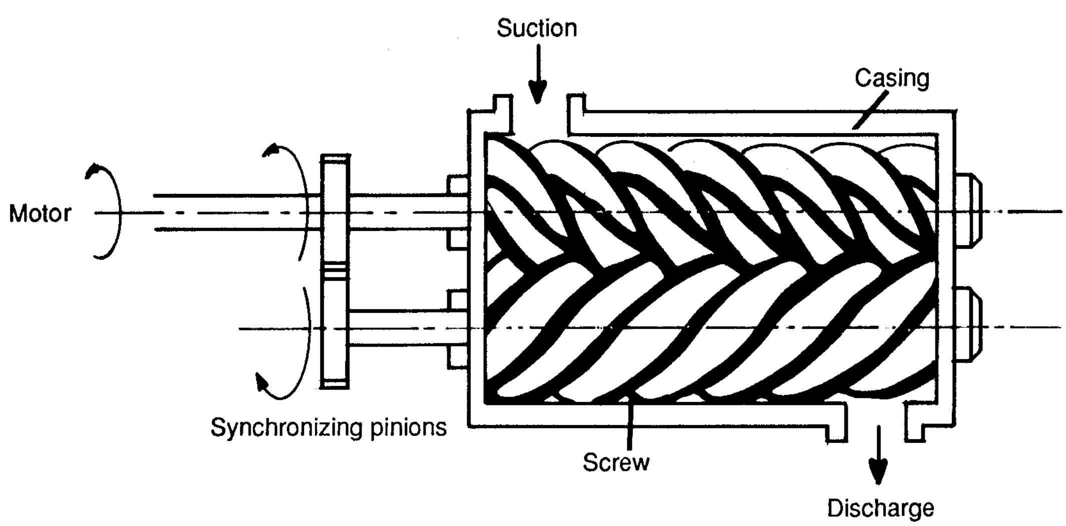


Figure 2-14b. Twin Screw Compressor

twin screw. Screw compressors are very robust machines and can operate well over a wide range of conditions. The twin screw machine is not affected by liquid droplets in the suction vapor, which can be a major advantage.

Screw compressors have been developed for use with steam as well as with conventional refrigerants. The applications tend to be those where shaft power is upwards of 50kW. Neither type uses suction or discharge valves. Most screw compressors are oil injected. The oil provides both lubrication and a seal between rotors and casing. Some designs are oil free which is particularly important for steam compression.

In the case of the single screw machine, the balanced compression process leads to very low bearing loads compared to other compressors. Both have stepless capacity control between 100% and 10% load although this can be at considerable expense in terms of efficiency.

Other Compressor Types. The rotary vane compressor, shown schematically in Figure 2-15a, has been used in a number of smaller heat pump applications. The reader may, however, be more familiar with its application as a compressor in vehicle air conditioning. This type of compressor is also known as the sliding vane type and tends to operate with low to moderate compression ratios, e.g., up to four, and discharge pressures up to 200 psig. The isentropic efficiency is often fairly poor.

Rotary lobe compressors (See Figure 15b) have been used in steam compression systems. They operate with compression ratios up to two and efficiencies of 60-75%. Their use is generally restricted to lower flow applications where the differential pressure across the compressor is less than 15 psi.

Heat Exchangers

A heat pump of the open type normally involves the use of at least one heat exchanger, while closed cycle units generally have three, assuming a subcooler is used in addition to the evaporator and condenser. Optimization of heat exchanger performance has a major influence on the cost-effectiveness of the overall heat pump system and it is important that attention is given to all aspects of heat exchanger design.

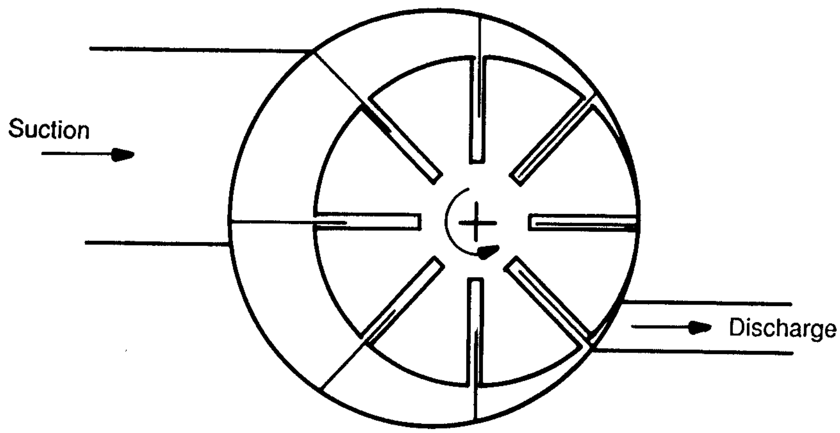


Figure 2-15a. Rotary Vane Compressor

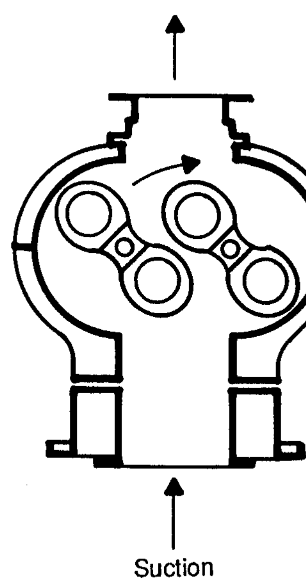


Figure 2-15b. Rotary Lobe Compressor

The two principal heat exchanger types used in heat pump systems are the shell and tube heat exchanger and the tube-in-plate heat exchanger (also known as a finned coil). For heat transfer to and from gas/air streams, the finned coil exchanger is used. For liquid heat sources and users, for some condensing duties, and where the heat pump is used to boil a fluid acting as the heat user, shell and tube heat exchangers are employed. Examples of heat exchangers are illustrated in Figures 2-16 and 2-17.

Another type of heat exchanger which may be considered for evaporator or condenser duty is the plate heat exchanger. Where heat sources other than a liquid or gas are used, for example the ground or solar energy, heat exchangers need to be tailored to the situation. Such situations are rare in industrial heat pump applications.

The nature of the heat source and heat user are the dominating factors in determining the type of heat exchanger to be chosen for the evaporator and condenser. As mentioned above, when air or a gas is the source or sink, a heat exchanger with a large amount of secondary surface on the gas side must be selected. If, on the other hand, a process liquid is involved, or a stream is undergoing evaporation or condensation, the heat transfer coefficients are likely to be of the same order as those on the refrigerant side, so plain tubes can be used. An exception to this rule is where water is the process fluid. Water normally gives a higher heat transfer coefficient than chorofluorocarbon (CFC) refrigerants, and a heat exchanger may be selected with finning on the refrigerant (shell) side (commonly low fin tubes) with the water flowing on the tube-side.

A number of the factors which have to be taken into account when selecting heat exchangers are listed in Table 2-1. The reader is advised to consult the design data of organizations such as the Heat Transfer Research Institute (HTRI), based in California, for detailed design procedures. Fouling is a very important aspect affecting heat exchanger performance, and is discussed later in this section.

Evaporators. There are two important categories of evaporators, flooded and direct expansion (DX). In the case of a DX evaporator, the refrigerant liquid supplied to the heat exchanger is completely boiled off and slightly superheated in the last part of the exchanger. A flooded evaporator has liquid present at all times which ensures good wetting of the heat transfer surface and offers improved heat transfer

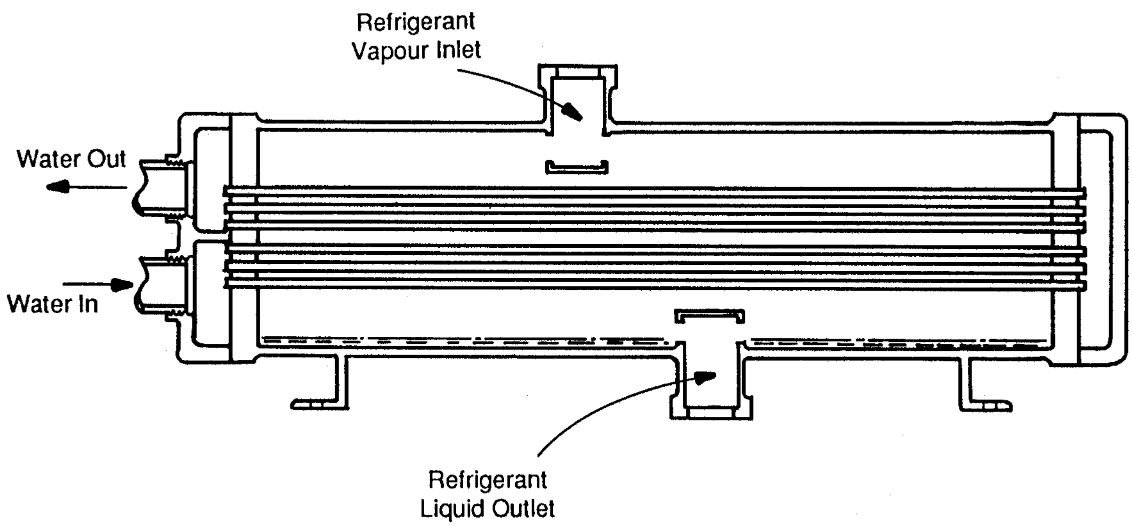


Figure 2-16. Shell and Tube Condenser

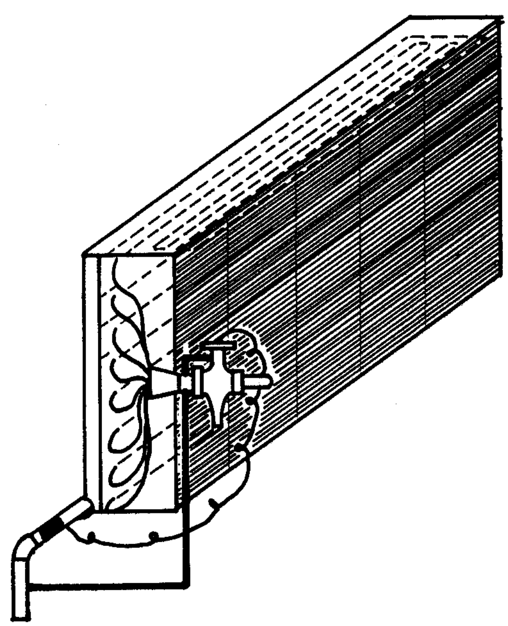


Figure 2-17. Finned Coil Evaporator

Table 2-1

CONSIDERATIONS WHEN SELECTING HEAT EXCHANGERS

Type of Fluids	Gas, Liquid, Vapor Clean, Dirty Wet/Dry Corrosive Elements Pressure Temperature Duty
Materials of Construction	Carbon Steel Stainless Steel Alloy Plastic (Teflon, Epoxy, etc.)
Cleanliness Requirements	Fouling resistance Ease of access to heat transfer surface
Cost	First Costs Maintenance Costs

coefficients. However, with a flooded evaporator it is necessary to separate the unevaporated liquid from the vapor. Figure 2-18 illustrates a shell and tube heat exchanger in flooded and DX modes.

From a thermal performance viewpoint, flooded evaporators are favored as they ensure the best heat transfer. However, there are a number of practical problems related to droplet separation and oil removal that tend to make flooded evaporators more expensive.

Condensers. These are almost invariably shell and tube heat exchangers for liquid heat users and finned coils for gaseous heat users. In the case of shell and tube condensers, the refrigerant is always condensed on the shell side with the liquid being heated located in the tubes. It is good practice to use a receiver vessel beneath the condenser to prevent vapor bubbles from passing to the expansion valve or excessive liquid banking up inside the condenser.

The Expansion Valve. The liquid feed to the evaporator is controlled by the expansion valve. In the majority of small heat pumps, thermostatic valves are used. In larger systems float switches or level control devices are common.

Thermostatic expansion valves are used with DX evaporators and are designed to ensure that vapor entering the compressor is always slightly superheated.

Where a flooded evaporator is used, a level controlled expansion valve is used. This maintains a constant level of liquid refrigerant in either the evaporator or the condenser.

Refrigerants

For open cycle systems, the process fluid being compressed differs from one industrial sector to another, although steam remains the most common. The range of refrigerants or working fluids which can be used in closed cycle heat pumps is wide. Note should be taken, however, of the potential impact concerning the effect of chlorofluorocarbons (CFC's) which include several common refrigerants, on the Earth's ozone layer. This is briefly discussed later in this chapter.

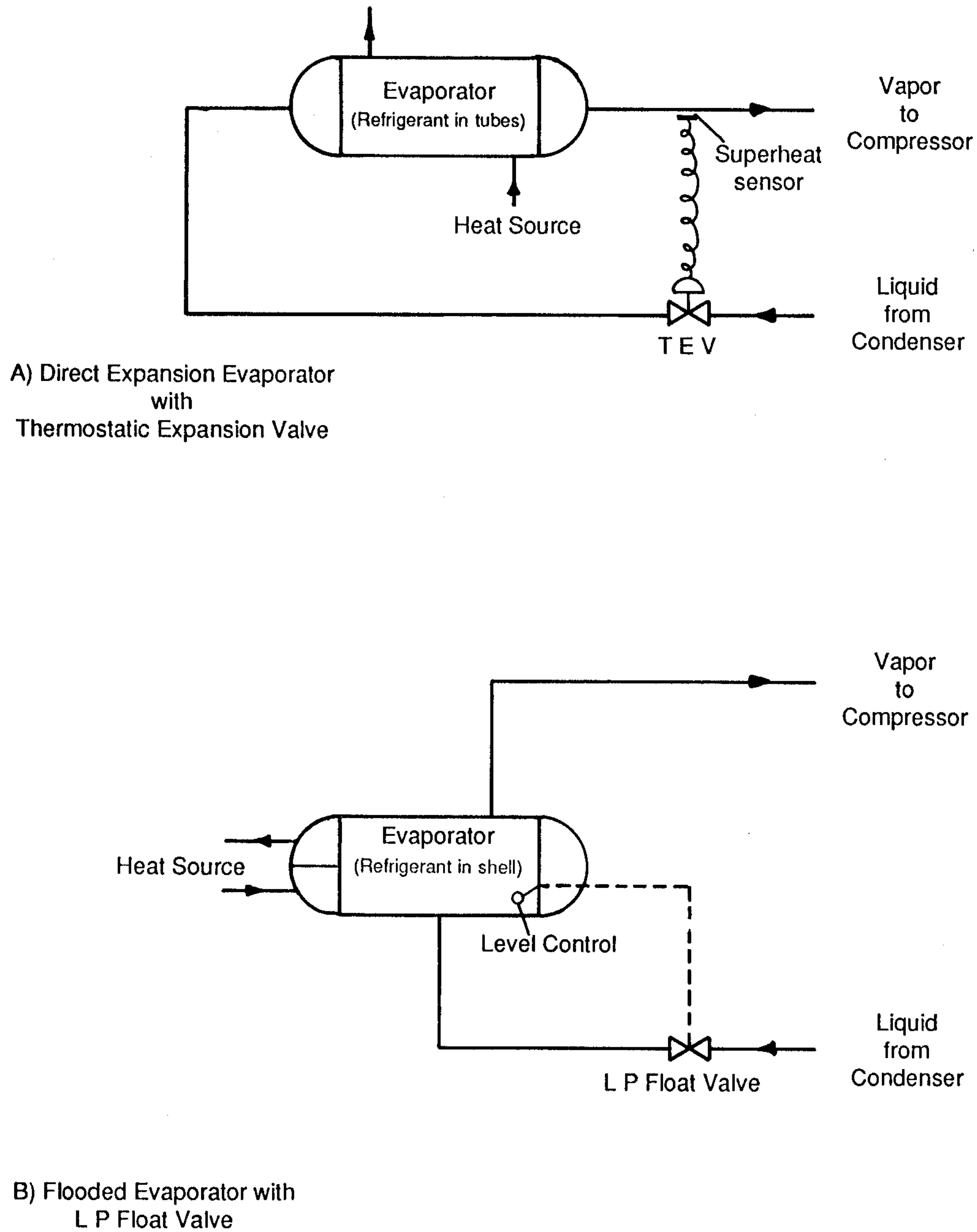


Figure 2-18. Flooded and Direct Expansion Evaporators

Refrigerants for closed cycle heat pumps are identified by an 'R' prefix, followed by a number, in accordance with the ASHRAE Standard List of Refrigerants. The most common refrigerants used in heat pumps are listed in Table 2-2.

Refrigerant property data are well documented which facilitates the sizing of equipment. In many heat pumps significant quantities of lubricating oil can circulate with the refrigerant. This oil can modify the refrigerant properties, and recently data have been made available on the thermodynamic properties of refrigerant/oil mixtures.

The choice of refrigerant for a given situation depends on a number of thermodynamic and practical factors. The evaporating and condensing pressures are of prime importance. The evaporating pressure should be above atmospheric pressure as it is difficult to prevent air in-leakage in a system under vacuum. The condensing pressure should not be above the maximum rating of the compressor. This appears to be an obvious point and, in principle, is not a restriction. However, most refrigeration compressors are only rated up to about 300 psig. As these are relatively cheap because of high volume production, it is expensive to demand higher pressures. It is also important that the condensing pressure is not close to the critical pressure of the refrigerant. In such circumstances, the latent heat of vaporization of the refrigerant is significantly reduced. This results in increased refrigerant flowrates and reduced COP.

The refrigerant chosen will affect the size of compressor required and the heat transfer coefficients in the evaporator and condenser. Toxicity, flammability and chemical stability are generally acceptable with the refrigerants listed, and the final selection is made on the basis of optimizing the cycle efficiency for refrigerants with acceptable costs.

In normal use, the refrigerants listed in Table 2-2 should prove reliable. If manufacturers recommended operating temperatures are exceeded, breakdown of the refrigerant can occur, leading to formation of acidic products which might attack the compressor, heat exchangers, or other components. Breakdown of refrigerant will occur at lower temperatures in the presence of mineral lubricating oils. For heat pumps operating above 210°F, it is necessary to use special synthetic oils or oil free compressors. Above about 270°F the refrigerant breakdown problem prevents use of CFC refrigerants, leaving water as the only obvious choice.

Table 2-2

COMMON REFRIGERANTS USED IN HEAT PUMPS

Refrigerant Number	Chemical Name	Chemical Formula	Boiling point at Atmospheric Pressure, °F
R11	Trichloroflouromethane	CCl_3F	74.7
R12	Dichlorodiflouromethane	CCl_2F_2	-21.6
R22	Chlorodifluoromethane	$CHCLF_2$	-41.4
R113	Trichlorotrifluorethane	$C_2Cl_3F_3$	117.6
R114	Dichlorotetrafluoroethane	$C_2Cl_2F_4$	38.4
R717	Ammonia	NH_3	-28
R718	Water	H_2O	212

For open cycle systems, it is important to have a knowledge of the fluid being compressed and any impurities which might be present. The possibility of corrosion must also be examined. This may necessitate all-stainless steel construction.

The Chlorofluorocarbon Problem. Legislation concerning limitations on the future production and use of chlorofluorocarbons (CFC's) will affect the refrigeration and heat pump industry, by most accounts, significantly. This is in spite of the fact that the proportion of total CFC production used in such equipment is relatively small (compared to other uses, such as foam blowing, aerosols, etc.).

It is difficult to predict the impact on the refrigeration and heat pump industries, but it is likely that many commonly used fluids will become unavailable. However, new research is expected to provide other alternatives.

When designing the heat pump, the refrigerant is an important component and any possible restrictions on this use should be investigated and understood.

Operational Limitations

It is important to recognize some of the technical limits to practical and economic operation of heat pump systems. The comments made here apply to "second generation" heat pumps i.e. ones that can be produced commercially today. Some of these limits will be exceeded by advanced cycles, which are described later in this section. It must be stressed, however, that such advanced cycles are a long way from reliable commercial production.

Top Temperatures. The temperature to which a process stream can be heated in a heat pump (the top temperatures) is one of the main operational limitations.

In the case of CFC refrigerants used in closed cycle vapor compression systems, the top temperature safely achievable is of the order of 250°F, sufficient to enable low pressure steam to be generated in the condenser. Refrigerants such as R114 and the modern synthetic lubricants are chemically stable at compressor discharge temperatures of 260°F and long term durability has been demonstrated in reciprocating and screw compressors as well as in centrifugal machines. Above this temperature the refrigerant breaks down chemically, making the heat pump inoperative. Refrigerant degradation is worst in the presence of oils. With oil free compressors, top temperatures of 280°F may be possible with CFC refrigerants.

Attempts have been made to use other fluids such as methanol in closed cycle heat pumps at condensing temperatures up to 320°F. To date, however, no commercial systems are available.

The most promising fluid for operation above 300°F is steam, either in open or closed cycle configurations. It is realistic to operate with delivery temperatures of 320°F, and tests on twin screw compressors have shown that delivery temperatures of 480°F are achievable.

Temperature Lift. It is not only top temperature that is important. The temperature lift of a heat pump has a direct bearing on the COP and hence the economic performance. Clearly if the lift is too high it is pointless trying to use a heat pump. It is difficult to give specific figures for unacceptable temperature lift because the economics depend on site specific conditions, particularly fuel and electricity prices. In general, if the lift is above 100°F it is unlikely to be a very economical application except in special circumstances.

Pressure Limitations. The choice of refrigerant is often dominated by pressure considerations. To keep compressor costs down it is important to avoid discharge pressures above 300 psig. For this reason different refrigerants are used as top temperature increases. Up to a condensing temperature of 120°F, R22 or ammonia are good fluids to choose. Between 120°F and 160°F, R12 is best. Above this temperature R114 or R500 is used. The thermal degradation problems described previously are reached before these fluids reach 300 psig condensing pressure.

There can also be pressure limitations on the suction side of the compressor. The problems arise because refrigeration compressors are the preferred choice to keep capital costs low. For instance, if R12 is used on a low temperature lift heat pump operating at a suction temperature between 120°F and 160°F, the suction pressure would exceed the rating of many designs.

Fouling/Corrosion. Heat exchanger fouling and corrosion are among the major problems encountered with first generation heat pumps. If either the heat source or heat user is too dirty or corrosive it may be impossible to identify a practical or economical design. It must be remembered that a fouled evaporator or condenser can cause a significant loss of COP and capacity.

Fouling and corrosion can be dealt with in a number of ways, and these should be considered before proceeding with a potential application. In the case of many heat sources or sinks, upstream filtration can remove particles. Alternatively, intermittent on-line or off-line cleaning may be carried out. Where condensation occurs, consideration must be given to the nature of the volatiles which condense onto the heat exchanger surfaces, since these can cause corrosion. Corrosion can be prevented through proper selection of materials, although costs can be high.

Fouling on the refrigerant side is not normally a problem if the circuit design is correct. Care must be taken to avoid a build up of oil in evaporators or air in condensers, particularly if the heat pump has a suction pressure below atmospheric.

Off-Design Behavior. The design or performance of a heat pump can often be affected by off-design operation. Such occasions include reduced product throughput or transients experienced during plant start up or shut down. It is vital that all conceivable modes of operation are considered at the design stage, not just the most common conditions.

Location. The location of heat pumps is generally subject to constraints similar to those for other thermal and mechanical equipment. The relative locations of the heat source and heat user will have a significant effect on capital cost since these affect piping or ducting runs.

The physical size of the heat pump also has a bearing on its location. If the major components are too heavy or too large, installation above ground level may be undesirable or uneconomical.

Provision of appropriate services, principally to power the prime mover, also influences location, as do environmental conditions. The evaporator often is the most critical component in determining the performance of heat pump applications, and its placement needs attention in order to optimize heat transfer and pressure drops.

ADVANCED SYSTEMS

A review of the current research and development activities of the industrialized nations (in particular, Japan, the United States and Europe) reveals a high level of activity on heat pumps. This will lead to new designs based on derivatives of existing cycles and novel cycles. This section briefly considers some of the contenders.

Systems Employing Non-Azeotropic Refrigerants

In conventional heat pump cycles the refrigerants evaporate and condense at constant temperatures. However, in many applications heat is supplied to the evaporator and removed from the condenser over a temperature gradient - this is illustrated in Figure 2-19a. It can be seen that at one end of the heat exchanger a large temperature differential exists. If this can be reduced, the COP can be raised.

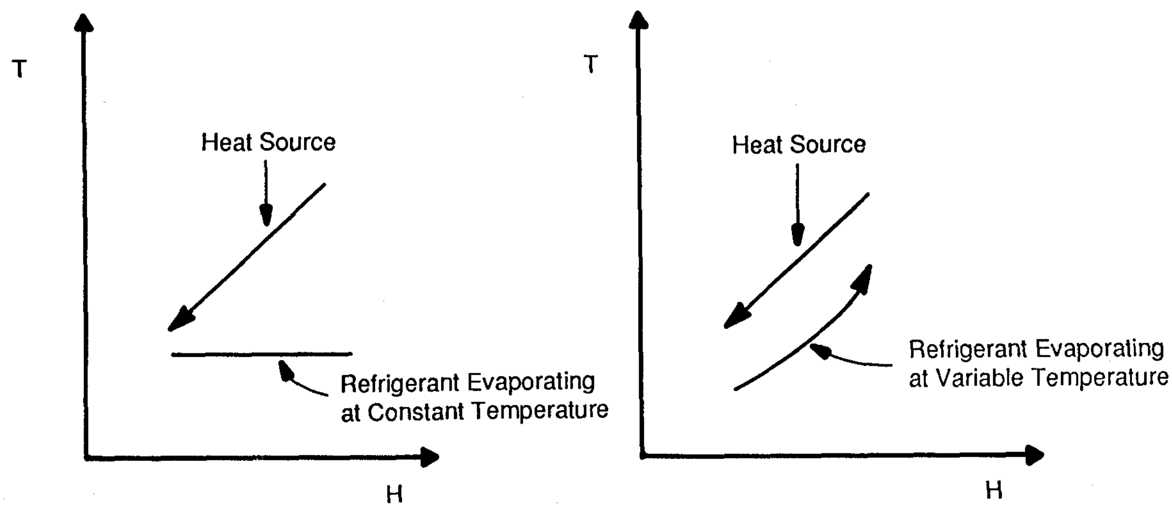
A non-azeotropic mixture evaporates over a temperature range, and the heat exchanger temperature profiles are modified as shown in Figure 2-19b. This has the effect of reducing the large temperature differential.

An example of the improvement possible using a refrigerant mixture is evident from the tests conducted by engineers in France, where much work in this area is being carried out. Sixteen different mixtures were investigated experimentally, and it was found that increases in capacity of up to 30% and in COP of up to 12% were achieved. The capacity increase was significant because this implies a saving on capital cost compared to a unit using a single component refrigerant.

Absorption Cycle Systems

While the vapor compression cycle heat pump dominates the market, the majority of R&D funding is being allocated to absorption systems. This is to a large extent aimed at penetration of the domestic heating market, but industrial systems are also receiving attention.

Like the vapor compression system, top temperature is a limit, as is temperature lift. It is proposed that 2-stage systems could overcome the latter drawback, without major cost increases. Also, new working fluid pairs may contribute toward



Conventional

Non-Azeotropic Mixture

(a)

(b)

Figure 2-19. Non-Azeotropic Cycle

the former. Targets of a delivery temperature of 400°F and PER's of up to 2.5 have been set by one research team in Austria. This might include other forms of 'chemical' heat pump where chemical reactions, rather than pure absorption, take place.

Interestingly, the vapor compression cycle can be linked with the absorption cycle to give a much improved temperature lift, without changing the compressor inlet and outlet conditions. The working fluid pair of ammonia and water appear suitable for many applications. A unit using a monoscrew compressor has been constructed in the Federal Republic of Germany.

The Brayton Cycle

Water has already been mentioned as a refrigerant. The other cheap and abundant compound which can be used as a refrigerant is air. Air is used in the Brayton cycle; i.e., in a gas turbine-compressor assembly, to produce similar heat pumping duties to the Rankine cycle, but at higher temperatures.

The Brayton cycle, applied to a spray dryer has been tested experimentally for heat pump duties, but has yet to see commercial service. It is probable that many of the turbo-compressor components could be adapted from systems made for other applications; one prototype is in fact using a turbocharger. The main interest is in the high temperature possibilities it seems to offer.

The United States Department of Energy has funded the installation of a Brayton Cycle heat pump for a solvent recovery application. This is in operation now and research aimed at broadening the range of applicability continues.

Finally, it is worth highlighting some aims of Japanese heat pump research. These include developing heat pumps capable of operating at up to 570°F (with a COP of 3), produce heat at 185°F from a source at 120°F with a COP of 8, and to give 100 MMBtu/h heat output for district and process heating. These ambitious goals are supported by significant financial support, which shows how important the Japanese consider industrial heat pumps to be.

Chapter 3

THE INDUSTRIAL HEAT PUMP MARKET - BACKGROUND AND APPLICATIONS

CHAPTER SUMMARY

The majority of industrial heat pumps are driven by electric motors and clearly represent additional electricity sales. How important are these sales to utilities?

This chapter contains a description of the existing industrial heat pump market. The types of systems in use and their applications are highlighted. The connected load associated with these installations is estimated and compared to that of residential and commercial units. To widen the perspective of the analysis, data about the markets in Western Europe and Japan are presented.

The manufacturers and other parties who may help with the design of a heat pump system are listed.

Finally, some thoughts about the future for industrial heat pumps are presented.

THE UNITED STATES INDUSTRIAL HEAT PUMP MARKET

To better understand the existing industrial heat pump market, a survey of manufacturers, utilities, and government agencies was completed during the preparation of this Resource Guide. The parties were contacted by telephone or mail and asked about their knowledge of existing installations, their activities in the industrial heat pump market and their knowledge of other useful information sources. The following analysis is based on the information gathered during that survey. The data presented is not intended to be completely comprehensive, however, it is thought to be a true reflection of the existing heat pump market. It should also be noted that much of the information was based on manufacturers data and the installations were not checked to ensure that they were all still in operation.

Numbers, Types and Applications

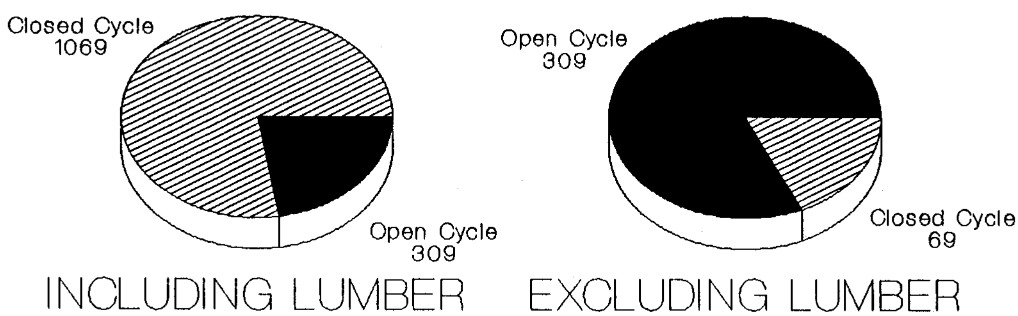
In Chapter 1, the industrial heat pump was defined as one having a process related heat source or heat sink and an 'industrial scale' heat output (approximately 250,000 Btu/h and above). Existing heat pump applications meeting these basic criteria were found to cover a wide range of industries, process, and sizes. The types of heat pumps in use include open cycles, semi-open cycles, and closed cycles.

It is interesting to note that the most numerous application of heat pumps, other than for space heating, is probably desalination or other purification of water using MVR evaporators. There are thousands of these units operating worldwide. Typical installations appear in resort hotels, water bottling plants, and on ships, producing anywhere from 200 to 50,000 gallons/day of water. However, the majority of these units are stand-alone and have the sole purpose of producing water. As these installations do not generally involve industrial processes, they have not been included in the analysis.

Types and Numbers. The heat pumps of most interest to utilities are those which require shaft work, because potentially they could use an electric driver. The survey identified 1378 of these units, with an estimated shaft power requirement of 440 MW and an estimated heat output of 2.7×10^{10} Btu/h. The split between open and closed cycles is shown in Figure 3-1. The open cycles in use are mainly applied to evaporation processes while the closed cycles in use are mainly applied to lumber drying. (Note that, strictly speaking, open cycles applied to evaporation should be called semi-open cycles because they involve the use of a heat exchanger. However, they are more commonly referred to as open cycles and for the remainder of this chapter, the term 'open cycle' is used to describe both open and semi-open cycles.)

Of the heat driven cycles (thermocompression and absorption) thermocompression systems are the most common. The main application is again evaporation processes. The survey did not address the numbers of these installations in detail, but it is estimated that about 400 units are in operation. There were no absorption type heat pumps in operation although it is believed that up to 20 absorption chillers may have been adapted for heat pump duties. Heat driven systems have been excluded from the remainder of this discussion.

Number of Units by Cycle Type



Source: Linnhoff March

Figure 3-1. Breakdown of U.S. Industrial Heat Pump Installations by Cycle

Applications. Figures 3-2 to 3-4 show how the number of industrial heat pumps is broken down by industrial sector.

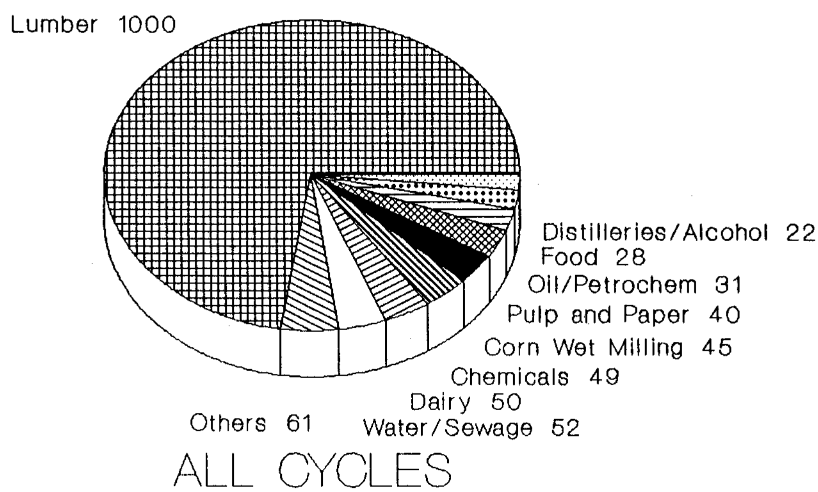
The single largest application of industrial heat pumps is lumber drying. In this process (see Figure 3-5) the moisture content of wood is reduced by circulating warm air around timber stacked in an insulated building (kiln). The warm, moist air being exhausted from the kiln is the heat source for the heat pump. Heat subsequently delivered by the condenser is used to heat air entering the kiln. Apart from lumber drying, this technique is being used in leather processing, fish drying, fruit drying, and drying of molded paper products. Typically these applications are small scale with average drying capacities being 5-6000 pounds of water per day, corresponding to a heating load of about 250,000 Btu/h. This heat output puts the majority of these systems right at the bottom of the industrial scale, but some applications have a heat output of up to 2 MMBtu/h. This market represents well proven technology. The benefits to the process operator in these cases is not just related to lower operating costs. Improved product quality, reduced material loss and lower first cost when compared to conventional systems are among some of the other reported advantages.

The other closed cycle applications are generally on a larger scale, with heat outputs ranging from 1 to 25 MMBtu/h with typical COP's of four to eight. These systems have been applied in a wide range of industries. Heat sources used include hot process streams (liquid and vapor), solar heated water, sewage, and heat from refrigeration condensers. Typical heat users are process water, water for cleaning purposes, HVAC systems, and process fluids. The upper temperatures for fluids being heated are in the range of 180-200°F. These installations are based on conventional refrigeration cycle technology, using CFC working fluids (refrigerants).

The open cycle heat pumps identified are mostly related to evaporation processes where the vapor produced by evaporation of a liquor is compressed and used to drive the evaporator (see Figure 3-6). These 'MVR' systems are mainly used for water based solutions. All the applications in Figure 3-4, except for those in the oil/petrochemical industry, involve the compression of water vapor.

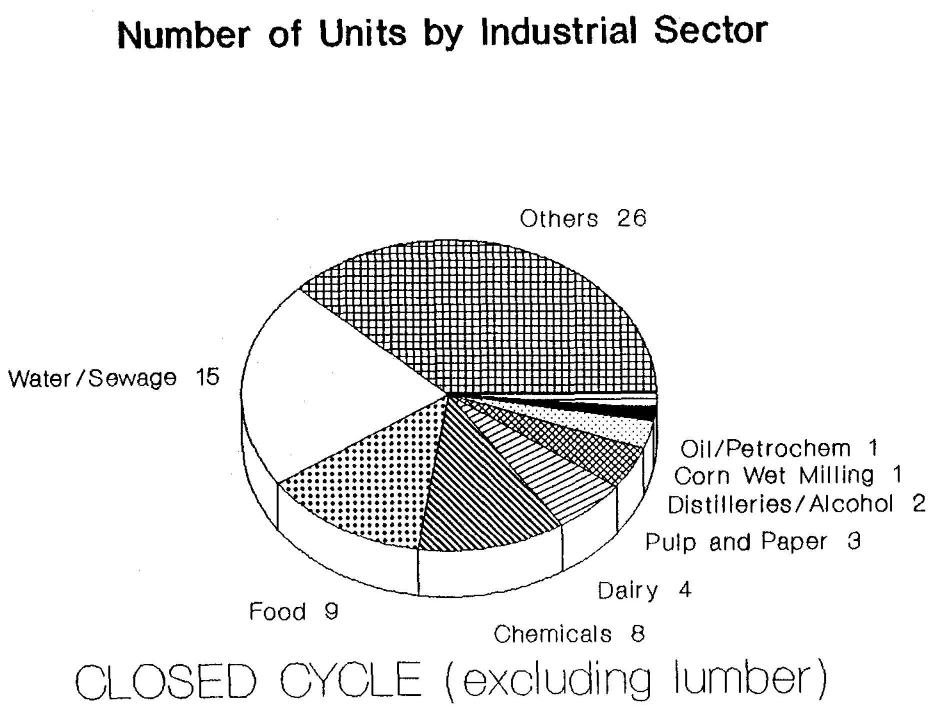
The oil/petrochemical industry MVR heat pumps are applied to distillation columns. Generally, overhead vapors from a column are compressed and used to reboil the column (see Figure 3-7). Applications are restricted to columns which separate

Number of Units by Industry Sector



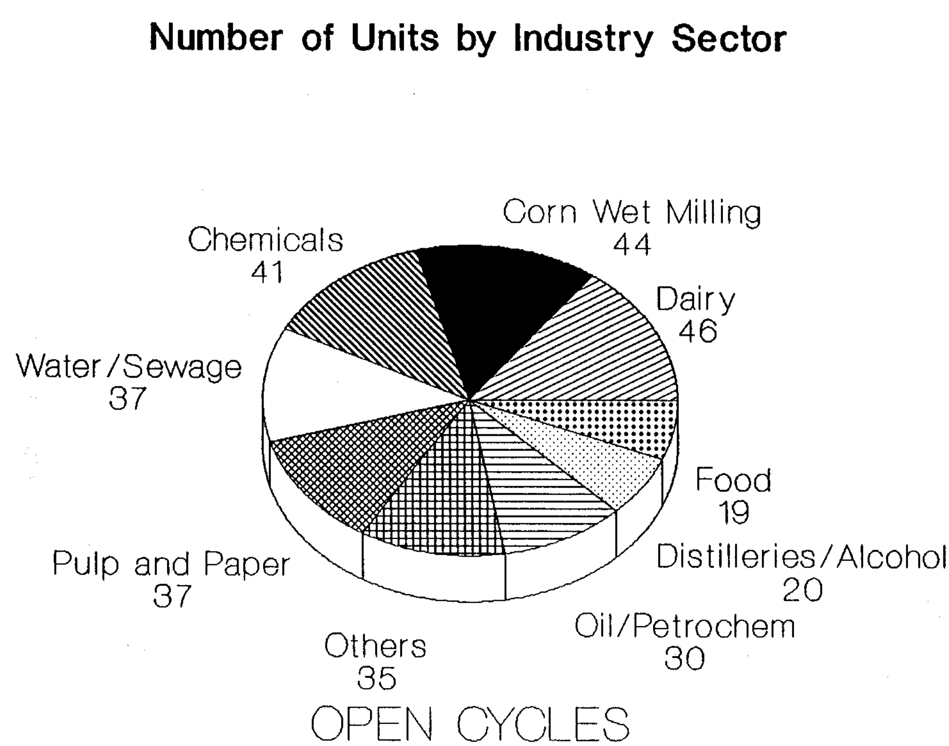
Source: Linnhoff March

Figure 3-2. Breakdown of U.S. Industrial Heat Pump Installations by Industry Sector



Source: Linnhoff March

Figure 3-3. Breakdown of U.S. Closed Cycle Industrial Heat Pumps by Industry Sector



Source: Linnhoff March

Figure 3-4. Breakdown of U.S. Industrial Open Cycle Heat Pumps by Industry Sector

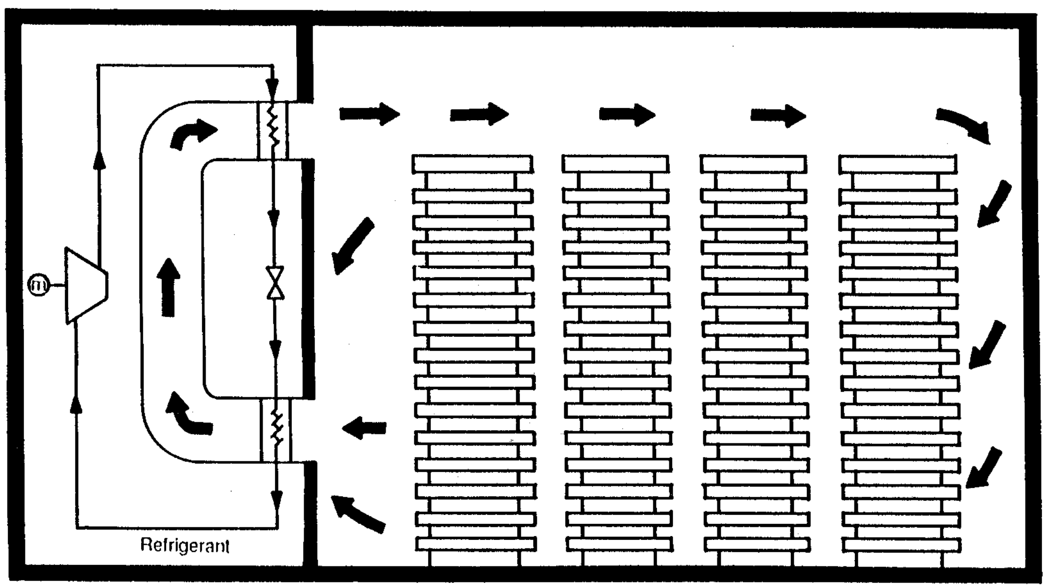


Figure 3-5. Heat Pump in a Lumber Drying Kiln

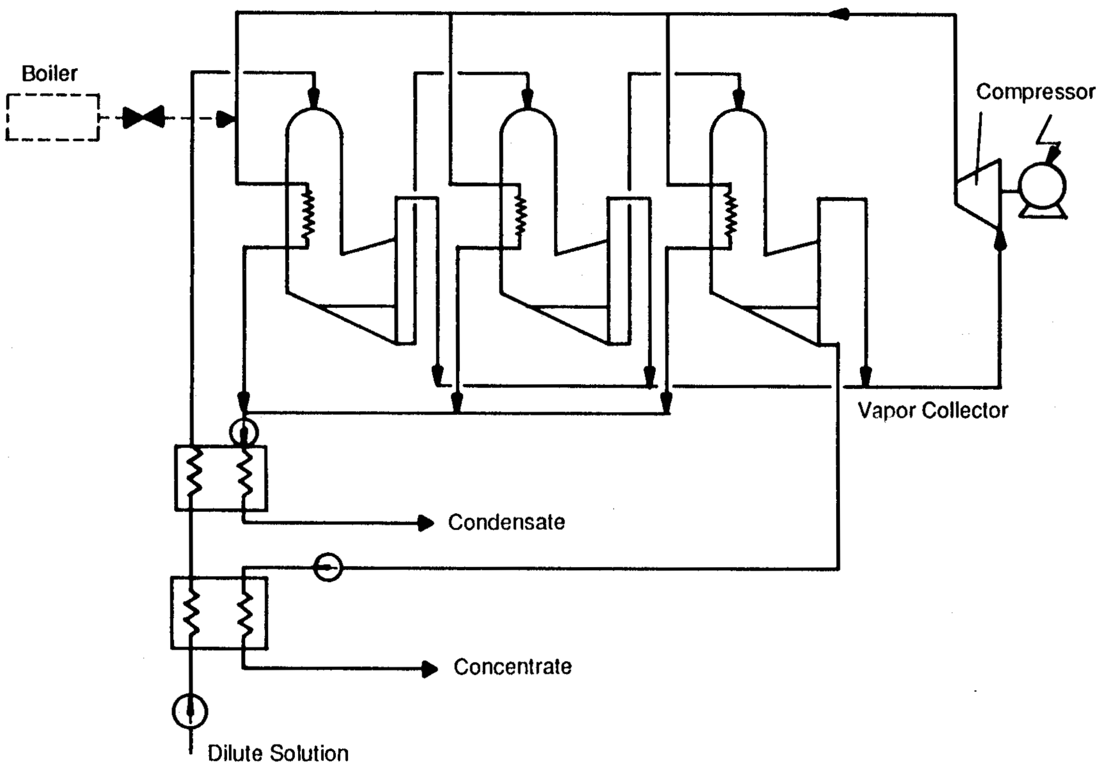


Figure 3-6. Heat Pump in an Evaporation Process

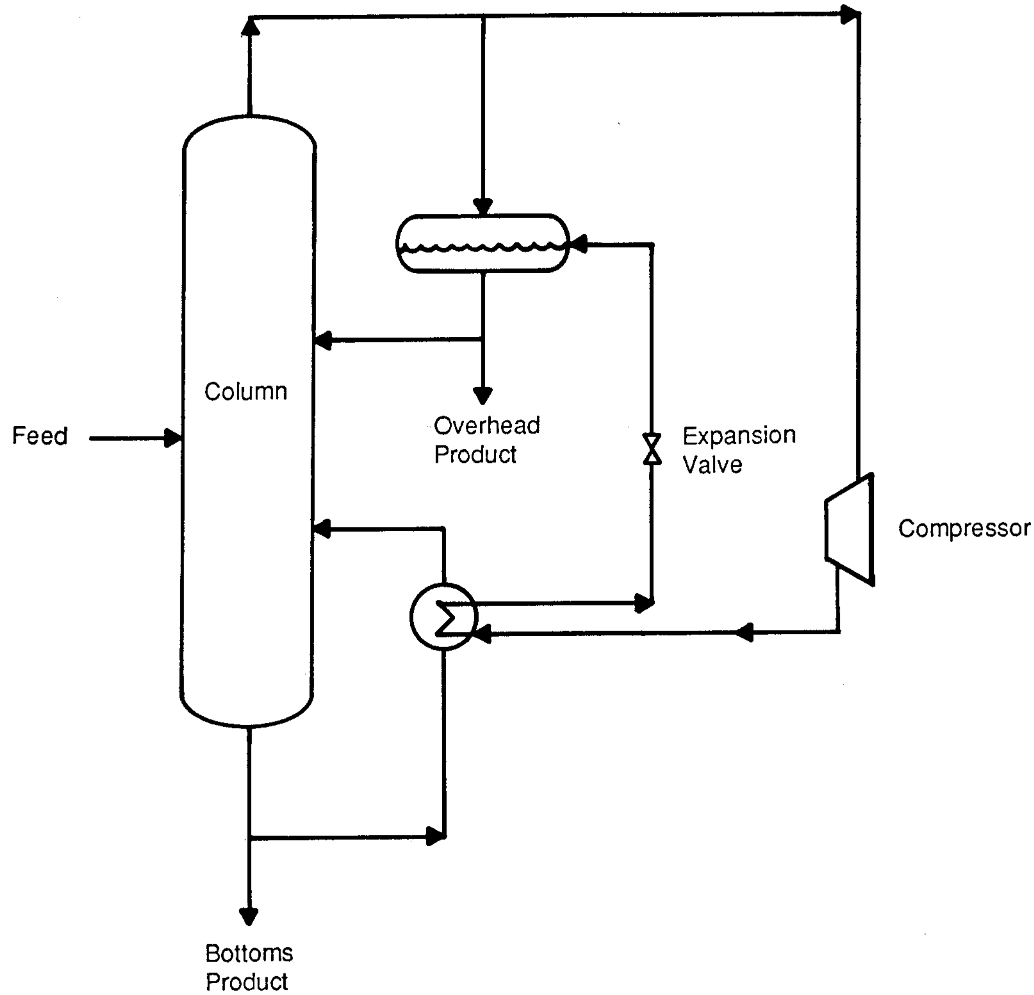


Figure 3-7. Heat Pump in a Distillation Process

components with similar boiling points, such as n-butane and iso-butane, and therefore, have a low temperature difference across the column.

Open cycles are generally the largest in terms of heat delivered. It is not uncommon for an evaporation rate of 100,000 lb/h to be achieved in MVR evaporators. The heat supplied by the vapor after compression, therefore, exceeds 100 MMBtu/h. The COP's of these systems are also high, for example, a unit operating with a compression ratio of 1.5 will have a COP of approximately 20. This means that 100 MMBtu/h of heat can be delivered to the process by using only 5 MMBtu/h of shaft work (1.5 MW). Vapor recompression evaporators represent a proven technology. Apart from very significant reductions in the energy consumption of the process, other reported benefits include: no requirement for additional steam raising capacity, no cooling water requirements, possibility for improved product recovery and improved product quality. The scale of the oil/petrochemical industry also leads to large installations. MVR systems integrated with distillation columns have heat delivery rates of up to 100 MMBtu/h.

Thermocompression heat pumps are typically used in evaporation processes and can be regarded as an alternative to MVR systems. The performance of these systems is not as good as that of MVR systems because a relatively large amount of motive steam is required to compress the process vapor. To ensure maximum benefit from these units, they must be integrated with the process in a way which allows all the exhaust vapors to be utilized. However, they are simple, cheap, and can have good paybacks.

Table 3-1 highlights some actual applications for each type of cycle by industry sector.

The timing of the MVR installations is illustrated in Figure 3-8. It can be seen that the build rate was low up to the mid/late seventies but picked up quite dramatically in the early eighties, probably due to the increase in fuel prices that occurred in 1979. However, this rate declined in the mid-eighties, just as in Europe. This, again, is probably related to the fall in energy prices that occurred in 1985/1986. (It should be noted that year of installation data was only available for about 70% of the MVR installations identified.)

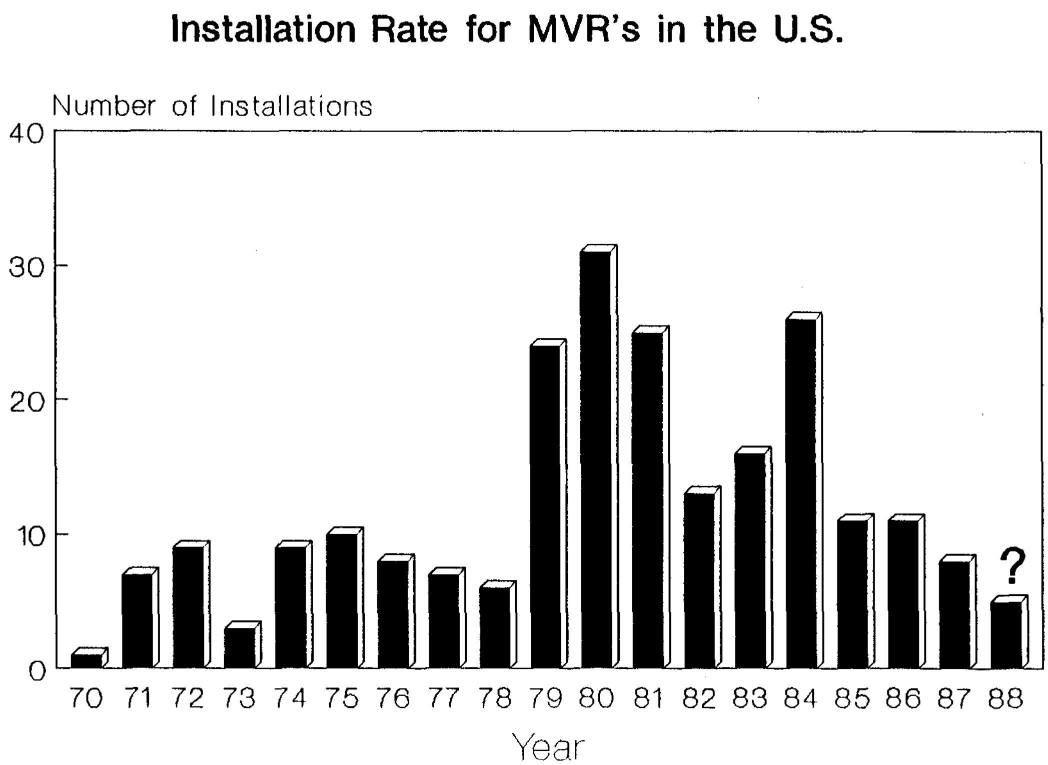
Table 3-1

HISTORICAL HEAT PUMP APPLICATIONS BY INDUSTRY SECTOR

Industry	Activity	Applied Process	Heat Pump Type
Petroleum Refining and Petrochemicals	Manufacture of petroleum/petro-chemical products	• Separation of C_3's • Separation of C_4's	Open Cycle Open Cycle
Pulp and Paper	Pulp manufacture	• Concentration of black liquor and other pulping process liquors	Open Cycle
	Paper manufacture	• Process water heating	Closed Cycle
Food/Beverage	Manufacture of alcohol	• Concentration of waste liquids	Open Cycle
	Brewing	• Concentration of waste beer	Open Cycle
	Manufacture of starch/sugar	• Concentration of steepwater	Open Cycle
		• Concentration of syrup	Open Cycle
	Manufacture of dairy products	• Concentration of milk	Open Cycle
		• Concentration of cheese whey	Open Cycle
	Manufacture of juices	• Concentration of apple juice	Open Cycle
		• Concentration of tomato juice	Open Cycle
	Manufacture of most food products	• Process/cleaning water heating	Closed Cycle
	Soft Drink manufacture	• Concentration of effluent	Open Cycle

Table 3-1 (cont.)

Industry	Activity	Applied Process	Heat Pump Type
Chemicals	Salt manufacture	• Concentration of salt solution	Open Cycle
	Sodium sulfate/ ammonia chloride/ sodium carbonate/ boric acid manufacture, etc.	• Product concentration	Open Cycle
	Process effluent	• Concentration of waste fluids	Open Cycle
	'Waste' vapor streams	• Compression of vapor for use elsewhere in the process	Open Cycle
	Pharmaceuticals/ other low temperature processes	• Process water/ fluid heating	Closed Cycle
Utilities	Nuclear power	• Concentration of radioactive waste	Open Cycle
	Power generation	• Concentration of cooling tower blowdown	Open Cycle
Others	Manufacture of fresh water	• Desalination	Open Cycle
	Electroplating type industries	• Heating of process solutions	Closed Cycle
		• Concentration of effluent	Open Cycle
	Lumber	• Drying	Closed Cycle
	Textiles	• Process water heating	Closed Cycle
	General Industrial	• Wash/process water heating	Closed Cycle
		• Water heating for space heating	Closed Cycle



Source: Linnhoff March

Figure 3-8. Build Rate of U.S. MVR Heat Pump Installations

The Importance to Utilities

It is estimated that the industrial heat pump in use at present require about 440 MW of shaft work to drive them. About 85% of this work is supplied by an electric motor. This compares to the estimated 10,000 MW associated with over 3 million electrically driven residential and commercial units.

Although lumber drying installations are the most numerous, they do not represent the largest power user. As shown by Figure 3-9, open cycles are by far the largest power user with about 380 MW associated with their drivers. The MW usage by industry sector for open cycles is broken down in Figure 3-10. The values for power consumption in Figure 3-10 represent the total power required to drive the heat pump compressors in each industrial sector. Not all of this power is supplied by electric motors. About 40% of the oil/petrochemical load, 25% of the corn wet milling load, and 10 % of the pulp and paper load is non-electric.

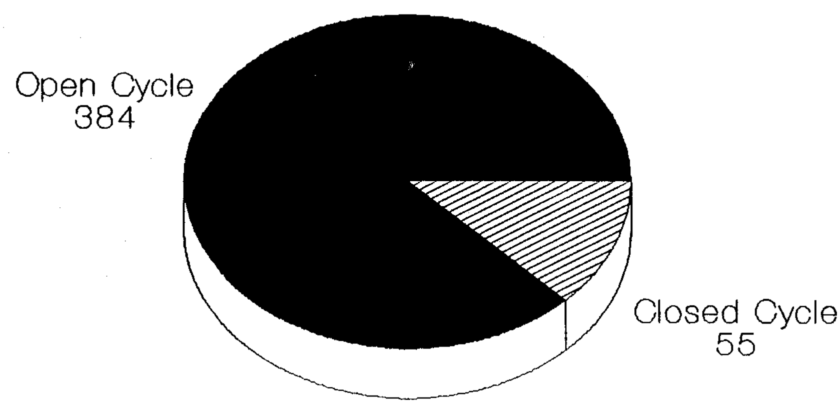
The oil/petrochemical sector has the highest shaft power requirement, but such units have a higher probability of being driven by a gas or steam turbine. The pulp and paper, water treatment, corn wet milling, and chemicals industry share the majority of the remaining load. However, in the water treatment category, about 90% of the load is related to utilities, the main application being concentration of cooling tower blowdown. The food and dairy sectors have a large number of installations but their average size is much smaller than those in the sectors mentioned above.

Although the 440 MW of load associated with industrial heat pumps is small compared to the 10,000 MW of residential load, it appears that in most cases, electric motors are the preferred driver for heat pump installations. The alternative drivers most commonly used are steam turbines, but gas turbines are used for large duties of 5000 hp and above. These figures suggest that electric utilities have a high likelihood of gaining additional electricity sales from new industrial heat pump installations and will benefit from promoting their use.

HEAT PUMP INFORMATION SOURCES

There are many manufacturers of heat pump systems and heat pump components, all of whom could be useful contacts during the evaluation and design of heat pump systems. A list of these manufacturers is included in Appendix A for reference.

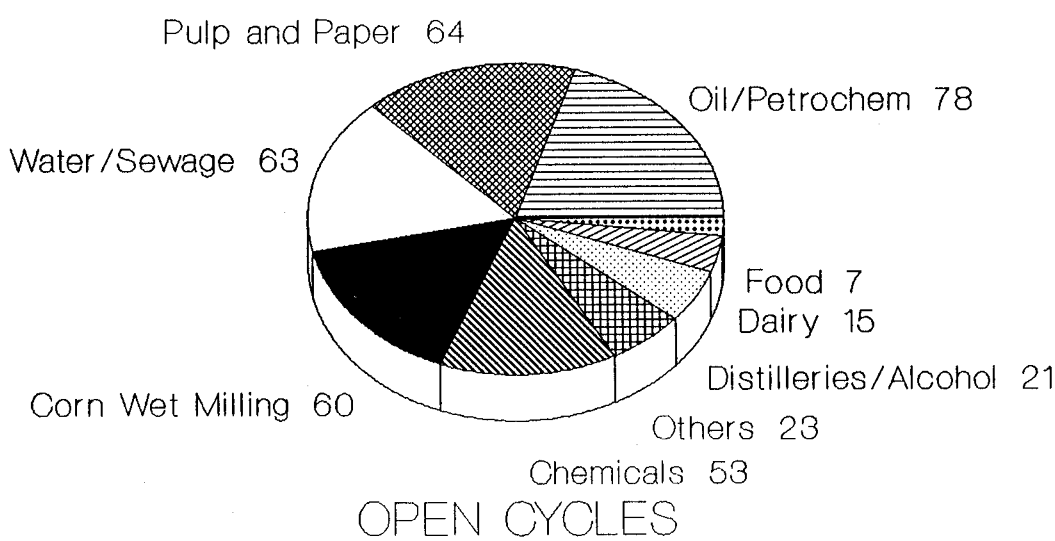
Power Consumption by Cycle Type (MW)



Source: Linnhoff March

Figure 3-9. Heat Pump Power Consumption by Cycle Type

Power Use by Industrial Sector (MW)



Source: Linnhoff March

Figure 3-10. Heat Pump Power Consumption by Industry Sector for Open Cycles

THE WORLDWIDE INDUSTRIAL HEAT PUMP MARKET

Industrial heat pumps are used all over the world with the most applications outside the United States located in Western Europe and Japan.

Numbers and Types of Installations

A recent survey of industrial heat pumps in Europe identified 850 units with a total heat output of 0.85×10^{10} Btu/h (this excludes a large number of the smaller lumber drying types). Of these, about 410 are open cycles and 440 closed cycles based on mechanical compression. There are only two or three absorption systems in use. This compares with the 378 non-lumber systems in the United States having an approximate heat output of 2.7×10^{10} Btu/h.

Two things are obvious: the average size of the units is smaller and closed cycles form a much larger percentage of the total market. This is a reflection of the smaller average size of manufacturing processes which utilize heat pumps and also the fact that European energy prices have traditionally been higher than those in the United States. The higher energy prices can justify the lower COP's of closed cycle units. Figure 3-11 shows which industries are utilizing heat pumps.

The predominant choice of driver is the electric motor. However, there are a significant number of gas engines being used to drive closed cycle systems. For systems with lower COP's, the opportunity for heat recovery from the engine exhaust and cooling jacket can improve the economic case for the installation. This is reflected by the higher PER (primary energy ratio, defined in Chapter 2) of engine driven systems with heat recovery, as compared to electric motor driven systems.

It is of interest to note the timing of these installations (see Figure 3-12). In the late 1970's, the installation rate began to increase, reaching a peak in 1982 and finally falling to a lower level in 1985. This fall in installation rate has been attributed to two factors: the fall in energy prices in mid 1985 and the unsatisfactory performance of many of the 'first generation systems' described in Chapter 1 (i.e. mechanical breakdowns, corrosion, oversizing, etc.). Avoiding the 'first generation' problems must be a prime concern of everyone interested in successfully promoting heat pumps in the United States. This is discussed further is Chapter 6.

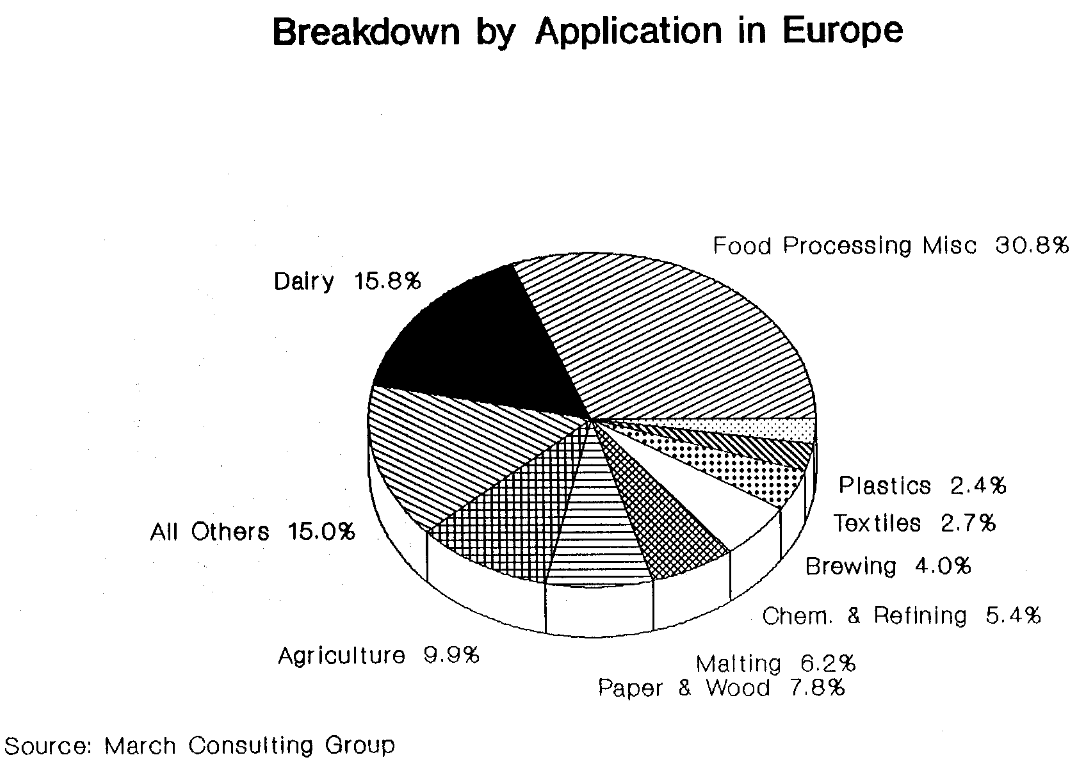
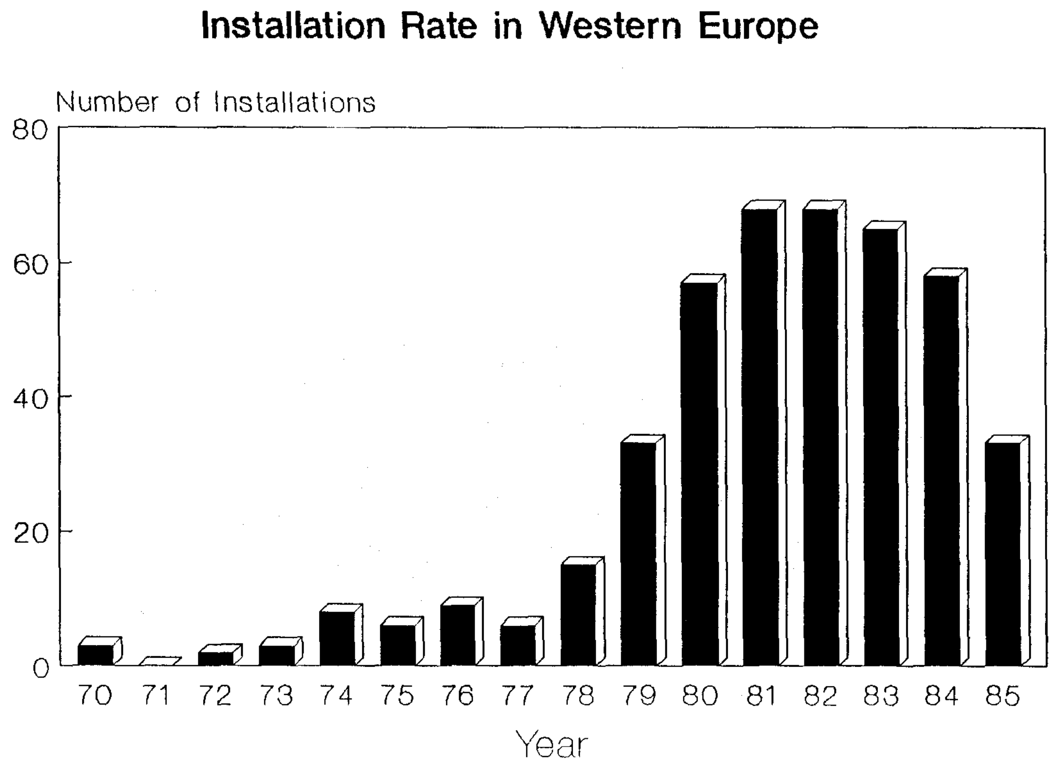


Figure 3-11. European Industrial Heat Pump Installations by Industrial Sector



Source: March Consulting Group

Figure 3-12. Build Rate of European Industrial Heat Pump Installations

In Japan a great deal of research effort has been put into the development of heat pumps for domestic, commercial and industrial applications. As Japan is very dependent on imported primary fuel, heat pumps are regarded as a promising technology for reducing reliance on imported fuel.

Again, the predominant types of systems are open and closed cycle mechanical compression systems with electric drivers. However, there is an increasing number of absorption systems being installed. These typically use waste heat, steam, or gas as their energy source. It is estimated that approximately 250 closed cycle units have been ordered over the past five years and that about 200 MVR units are in operation at present. The number of operating absorption systems is thought to be about 20. No information was available on the number of thermocompression heat pumps.

Applications

The types of applications encountered in both Europe and Japan are very similar to those in the United States. Applications for closed cycle units generally involve recovery of heat from hot liquid or air streams, with the upgraded heat being used for wash water, process water, boiler feed water, or other air or gas process fluids. Open cycles are again applied to the evaporation of water based solutions. The absorption systems in use have been used for combined chilling and heating duties as well as heating only duties. In some cases the useful heat leaving the absorption heat pump is in the form of low to medium pressure steam.

One application of large scale heat pumps that has been used in both Europe and Japan, but not yet on the same scale in the United States, is district heating. These are not strictly considered to be industrial heat pumps. However, these units have very large heat outputs, up to 100 MMBtu/h. They represent successful applications of very large scale closed cycle systems. In the United States, large heat pump based HVAC installations are in use now, but are restricted to heating individual buildings or building complexes with heat outputs in the 5 MMBtu/h range. These systems do not qualify as industrial heat pumps for the reasons described in earlier sections, but since HVAC applications represent a large potential market for high output closed cycle heat pumps, it is useful to know that they have been used successfully elsewhere.

THE FUTURE FOR HEAT PUMPS IN THE UNITED STATES

What does the future hold for the United States industrial heat pump market? Clearly this is a difficult question to answer, but it is possible to explore some of the factors which will have an effect.

Energy Prices

This is without doubt the single largest factor influencing the future of industrial heat pumps. If there is a trend towards higher oil prices this will certainly be of help, especially as the price of electricity is expected to rise more slowly than that of fuel. As fuel prices increase, heat pumps with lower COP's will become more attractive and the scope for applying heat pump technology will increase.

The absolute level of energy prices affects interest in all energy saving projects, not just heat pumping. However, the relative price of fuel and power is of particular interest when considering heat pumping (or cogeneration). There are significant regional variations in this ratio. Areas where power is relatively cheap compared to fuel will have more attractive heat pump economics.

One method of comparing the running cost of a heat pump with the existing heating plant is to work out the cost of heat delivered by each system. An indication of the sensitivity of heat pump economics to fuel price is provided by Tables 3-2 and 3-3 and Figure 3-13. (It should be noted that the cost of heat presented in these Tables does not take into account auxiliary equipment or distribution losses, etc.)

As an example, assume that a heat pump is proposed to replace a heating duty currently using low pressure steam. If fuel is \$3.00/MMBtu, and the boiler is 80% efficient, the cost of delivered heat is \$3.75/MMBtu (as shown in Table 3-3). Now, from Table 3-2 it can be seen that with a power cost of 4¢/kWh a heat pump with a COP of 3 delivers heat costing \$3.91/MMBtu, actually more expensive than that from the boiler. If, however, the heat pump COP was 5, the cost of heat reduces to \$2.35/MMBtu resulting in a saving of \$1.40/MMBtu. If the COP was 20, as it might be for an MVR system, the saving becomes \$3.02/MMBtu.

For each combination of fuel and power price, there is a COP for which the cost of supplying heat from a heat pump and a boiler is equal. This is the 'breakeven'

Table 3-2

COST OF HEAT DELIVERED BY AN ELECTRIC HEAT PUMP

	Cost of Heat Delivered $/MMBtu					
Power Cost	Heat Pump COP					
¢/kWh	3	5	8	10	20	30
2	1.95	1.17	0.73	0.59	0.29	0.20
3	2.93	1.76	1.10	0.88	0.44	0.30
4	3.91	2.35	1.47	1.17	0.59	0.40
5	4.89	2.93	1.83	1.47	0.73	0.50
6	5.86	3.52	2.20	1.76	0.88	0.60

Table 3-3

COST OF HEAT DELIVERED BY A BOILER

Fuel Cost $/MMBtu	Cost of Heat Delivered $/MMBtu*
2	2.50
3	3.75
4	5.00
5	6.25
6	7.50

*Assuming a boiler efficiency of 80%

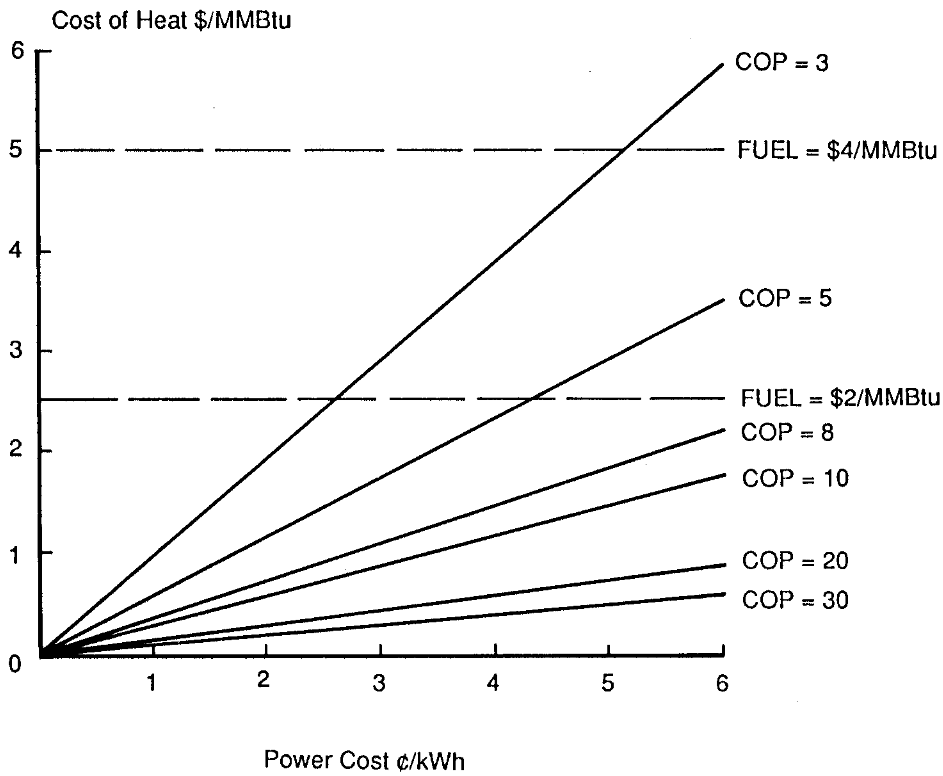


Figure 3-13. Cost of Delivered Heat

COP, and can be determined from Figure 3-14. In order for a heat pump to offer operating cost advantages, its COP must be better than the breakeven COP for the prevailing energy price conditions. For a given power cost, the breakeven COP reduces as fuel gets more expensive.

Apart from the obvious influence of energy price, the data presented in the Tables shows clearly the importance of COP. For a power cost of 4 ¢/kWh, improving the heat pump COP from 5 to 10 results in a reduction in the cost of heat from $2.35/MMBtu to $1.17/MMBtu. There is, therefore, great incentive to design heat pumps with COP's as high as possible. The next section discusses some technological improvements which help to improve COP and, therefore, heat pump economics.

One other important element of assessing a heat pump application is ensuring that correct monetary value is associated with the steam or other utilities being saved by the heat pump. The 'marginal cost' of generating or saving utilities is determined by the equipment configuration in the utility system. Calculating marginal costs can be a difficult task. Further definition and explanation of marginal costing is given in Chapter 4.

Technological Improvements

There continues to be further development of the individual components of heat pumps resulting in improved COP's and lower capital costs.

Areas of most interest include compressors and heat exchangers which account for the major capital and operating costs. Other heat pump cycles may also come into more widespread use. An example of such beneficial development has been observed in the field of steam compressors used in MVR applications. Over time the isentropic efficiency of single stage centrifugal compressors has increased from under 70% to about 80%. This results in about 10% less horsepower being used to compress the same amount of steam, thus reducing operating costs. Also, for steam compression applications, a turbo-blower type compressor has been developed. These units can handle large vapor volumes over pressure ratios of up to 1.3 with a high efficiency of 80+ %. These units have a welded construction, operate at low speeds and offer a lower capital cost than centrifugal units. Developments of this type are helping to improve the economics of MVR systems. It is reasonable to expect that similar improvements in the efficiency of compressors used in closed cycle applications will occur.

To reduce heat exchanger costs, it is necessary either to make them smaller or to change their construction. In some instances it may be possible to improve the heat transfer rates in exchangers by using tube inserts or 'high efficiency' surfaces. These devices improve the effective heat transfer co-efficients in the exchangers and result in lower exchanger costs.

The overwhelming majority of heat pump cycles in use at present are based on mechanical compression. The alternative most likely to become attractive is the absorption system. There are many absorption based chillers in use, but not many absorption heat pumps. However, these systems are in industrial use and development work aimed at improving performance is continuing.

On a more general level, there are many national and international agencies involved in development work for all the basic cycles, as well as hybrid and novel cycles. The objective of this work is to develop 'cycles' with improved performance and economics.

Awareness and Perception of Heat Pumps

Clearly the best advertisements for heat pumps are successful installations. There are many of these in operation now, particularly MVR's. The use of heat pumps will increase as potential users are made aware of these successes.

The industrial heat pump is generally perceived as a unit which is installed after a process is built to save energy. However, the heat pump should be regarded as an integrated part of the process and designed as such with ALL its potential benefits taken into consideration including reduced cooling duty, less steam demand, improved product quality, etc.

Applications

All the information presented on the United States and world markets is historical. The applications have evolved as people recognized the opportunities. There may be other types of applications in industry which have not yet been recognized. Also, some of the existing applications may not really represent the best opportunities for heat pumps.

In Chapter 1 of this Resource Guide, the importance of finding applications which really are the best technical and economical solutions was highlighted. The next chapter describes how Pinch Technology can be used to find those good applications and to determine how heat pumping stands up to its competitors. The Pinch Technology approach to finding good heat pump opportunities and other energy cost reduction measures offers significant potential to improve the applicability and competitiveness of industrial heat pumps.

New Installations

The economic case for heat pumps will always be better for units installed in new plants than for retrofits. The reasons for this include:

- The acceptable payback period for new plants is generally larger than that for retrofits. This means that heat pumps which do not meet a companys' payback criteria in a retrofit situation are more likely to be acceptable for new plants.
- The capital investment for the heat pump can be credited with reduced capital expenditure for boilers, coolers or other utility equipment which have reduced duties.
- Installation of equipment at the time a plant is being constructed is often simpler, easier and possibly less expensive than in retrofit situations.

New plants offer a good potential market for heat pumps. It is, therefore, important to ensure that process operators considering new plants are made aware of the technology and its benefits.

Chapter 4

THE INDUSTRIAL HEAT PUMP IN PERSPECTIVE

CHAPTER SUMMARY

Heat pumps have a useful role to play in improving the energy efficiency of industry. However, it is important to realize that they represent just one element in a whole array of energy saving and process improvement technologies. In the long term heat pumps will only establish a good technical and economic reputation if they are used in the right situations. If they are sold in situations where simpler alternatives have better economics the whole technology is in danger of developing a bad reputation.

This section shows how the heat pump fits into the energy saving jigsaw, using the malt kilning process as an example. The various competing technologies are examined and ways of comparing them are discussed. In particular Pinch Technology is introduced as a powerful tool for finding good heat pumping opportunities. Its role in the systematic generation and evaluation of alternatives is defined and discussed.

WHY CONSIDER ALTERNATIVES?

The technology of heat recovery has an interesting (and sometimes frustrating) general characteristic. It is almost certain that for every heat recovery scheme proposed, there will be another alternative. Whenever waste heat is available from a process there is a whole range of energy saving options. Each will use different technologies, save different amounts of energy and have different payback periods.

It is very tempting for a heat pump specialist to identify a cost effective opportunity in a process and suggest that as the best option. During the past 10 years many heat pumps have been installed in this way. In some cases the purchaser will have made the correct decision. However, in other situations, the opportunity for a morecost effective investment will have been missed. For example, a simple shell and tube heat exchanger could have recovered heat from a hot process stream instead of a heat pump being used to recover heat from a waste heat stream. The simple heat exchanger could have been the better option, but the heat pump may have been more obvious!

It is important to remember that the heat pump is competing against many other energy saving technologies. For the use of industrial heat pumps to become more widespread they must be perceived as technically and economically successful, just as their residential counterparts are. Technical success will come from good specification and design as described in Section 6. Economic success will come from finding good applications and ensuring that the heat pump really is the best option. This in turn means that comparing the heat pump with other energy saving possibilities is a crucial stage in the assessment of a heat pump installation.

WHAT ARE THE ALTERNATIVES?

Energy cost saving opportunities can be categorized in the following areas:

- Improve the performance of the existing equipment by better maintenance (e.g. fix steam leaks), good housekeeping (e.g. insulation, turning energy users off), better controls (e.g. temperature controllers).
- Install passive heat recovery equipment, such as heat exchangers.
- Modify the process to reduce energy requirements (e.g. mechanically reduce the moisture content of wet material prior to thermal drying).
- Install a heat pump.
- Install a lower cost utility system (e.g. cogeneration).

Of course if all this looks too formidable, then doing nothing is another alternative! Good housekeeping is an obvious way to save and should be pursued on an ongoing basis. The way in which these other options fit together will be explained with reference to a case study. The malt kilning process provides a good example of how the many different energy saving possibilities have been applied. Since the 1960's the energy consumed by this process has been reduced from around 2.5 to 1.4 MMBtu/1000 lb. This has been achieved through implementation of various energy saving schemes, including the use of heat pumps.

Malt is used in the production of beer and the malting process essentially involves the drying of a cereal grain after it has been soaked in water. A simplified flowsheet for the process is shown in Figure 4-1. Dry cereal grain, most commonly barley, is taken from storage and 'steeped' in water tanks until germination occurs. At a certain point the germination must be stopped and this is achieved by drying, or kilning, the grain in a stream of hot air.

In the kiln, grain is loaded on a grid to a height of 3 to 6 feet and heated air passes through the bed, removing moisture. Initially the air is heated to 170°F until the grain (or malt as it is now called) is free from surface moisture. At this point the air temperature is increased to 180-210°F for final drying. When fully dry, the malt is sent to storage.

The temperature and moisture content of the exhaust air varies throughout the batch kilning cycle. Initially the moisture content is high and the temperature low, the air being close to saturation. About halfway through the cycle, the moisture content of the exhaust air starts to fall and the temperature rises. This transition point is known as the 'break.'

To achieve the energy savings mentioned earlier, opportunities have been identified in all the categories defined above. These opportunities are described in the following sections.

Process Control and Housekeeping Improvements

The first opportunities identified by maltsters was that there was scope to improve the operation of their existing plant. For example, it was discovered that the way in which the bed of malt was laid had a significant effect on the energy consumption. An uneven bed could lead to channeling of air. Consequently, it was possible for some air to pass through the bed without having done any useful drying. The solution was simply to ensure even laying of the malt and even air distribution.

In some maltings the same vessel is used for steeping, germination and kilning. In these cases the water used for steeping must be drained from the vessel, although pools of water tended to remain, needing removal by the drying process; clearly this is inefficient. By designing the multi-purpose kilning vessels to allow easy water drainage this problem was eliminated.

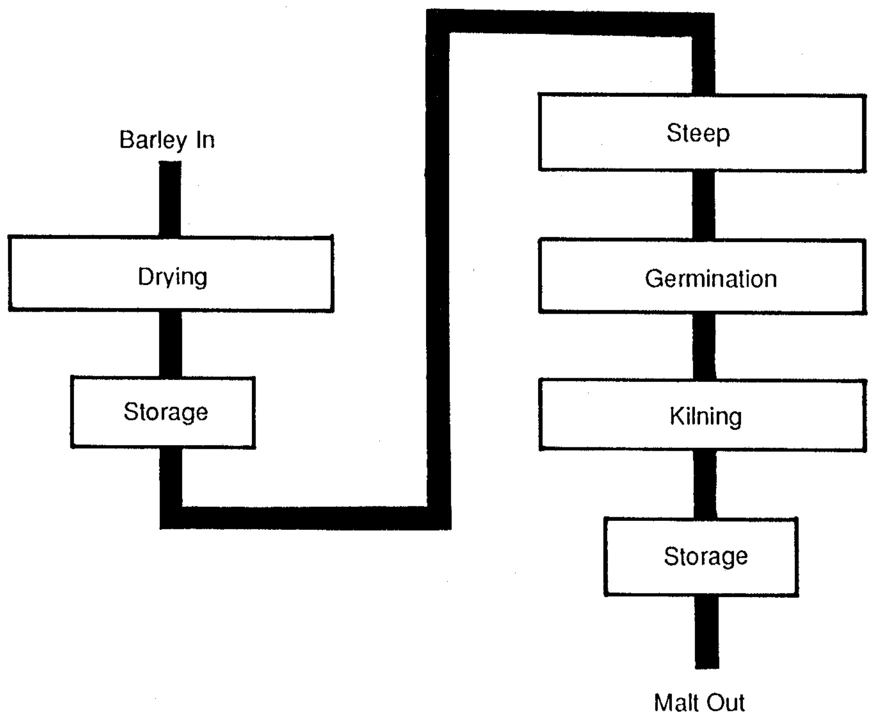


Figure 4-1. Simplified Flowsheet for the Malt Kilning Process

Obviously the energy cost is proportional to the amount of moisture removed and therefore, overdrying the malt incurs additional costs. By carefully controlling air flowrates and temperatures, particularly after the break, the costs associated with overdrying can be eliminated.

These relatively simple, low cost process improvements are typical of many opportunities which exist in industrial processes. Until they are attended to, it is usually wrong to consider the more expensive heat recovery or cogeneration options. As a general rule, one must avoid recovering heat that should not be there in the first place.

Heat Recovery

Having done the simple things, it is then necessary to address the more complex opportunities. Recovery of heat from the exhaust air to heat inlet air will reduce the overall energy consumption of the process. This can be done directly (i.e. by recycling air) or indirectly (i.e. using heat exchangers). Both methods have been used in the malting process.

Direct Heat Recovery. Usually, fresh air is heated and used to do the drying. This has a low moisture content and there is a high driving force for mass transfer of water from the malt to the air. The exhaust air is hot and can be used to replace fresh air. However, in the early stages of the cycle the exhaust temperature is low and the moisture content is high. Use of air in this condition would not save much energy and would not promote rapid drying. However, after the 'break' the air is hotter and drier and can substitute directly for fresh air. In many installations more than one kiln is in use and the exhaust air from the 'post break' kiln is fed as preheated air to the 'pre break' kiln. This technique does not require any heat exchangers, only appropriate ducting and dampers. A schematic is shown in Figure 4-2.

Indirect Heat Recovery. The direct method did not require any heat exchangers, but the energy saving only occurred for the 'post break' part of the cycle. Indirect recovery schemes allow some level of heat recovery throughout the cycle because there is no mixing of process streams.

One option, shown in Figure 4-3a, recovers heat from the exhaust using a single air-to-air heat exchanger. In some cases the exhaust and inlet air ducts may not

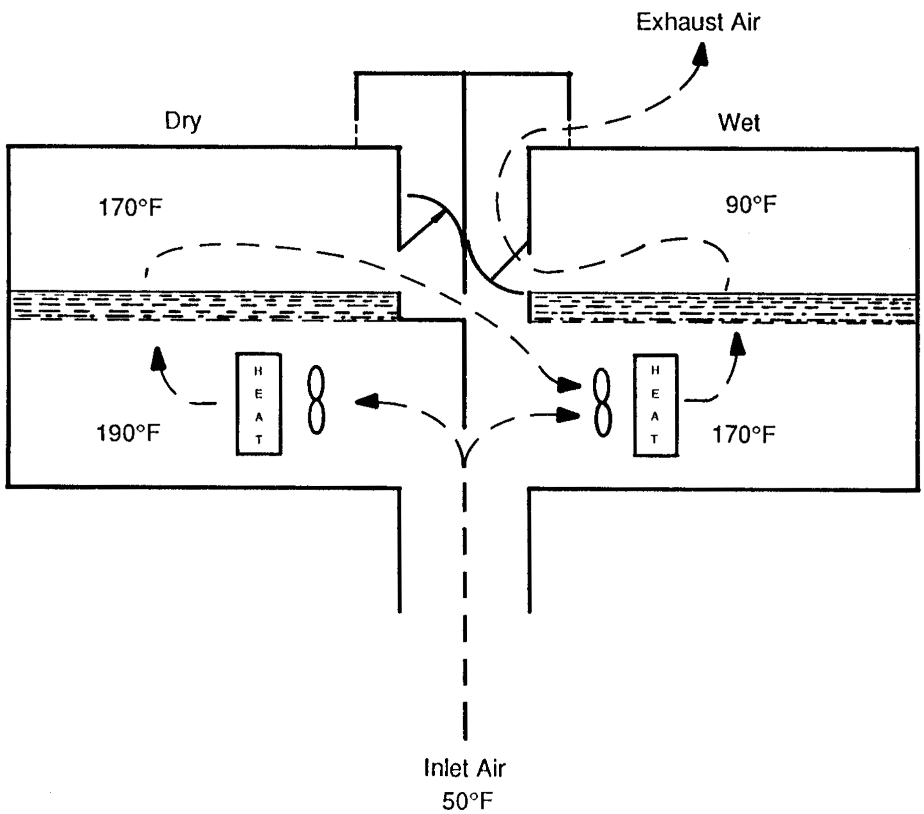


Figure 4-2. Direct Heat Recovery in the Malt Kilning Process

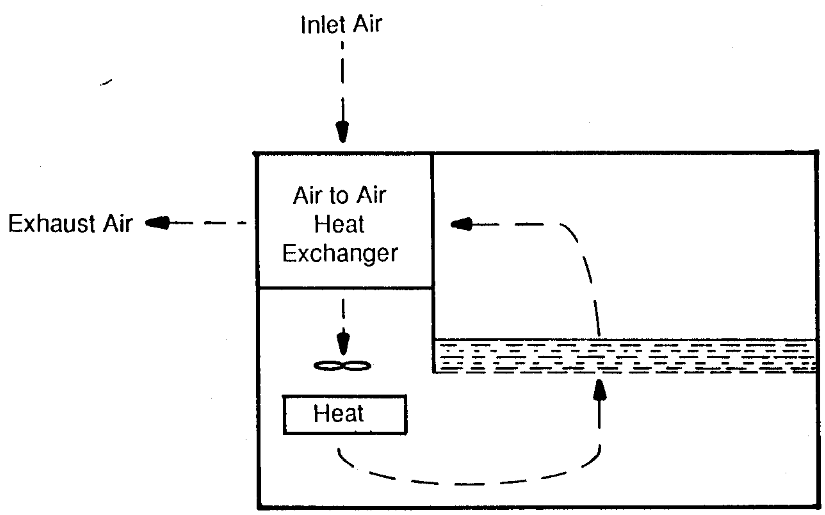


Figure 4-3a. Indirect Heat Recovery in the Malt Kilning Process - Single Air-Air Exchanger

be close together. As extensive ductwork modifications can be costly, the option shown in Figure 4-3b could be a cheaper solution. In this scheme heat from the exhaust is recovered into an intermediate fluid, typically glycol/water. This fluid is then pumped to an exchanger in the inlet air duct, to heat the inlet air. This system requires two heat exchangers instead of one. However, the liquid pipework will be less expensive to install than ducting for air.

<u>Heat Pumping</u>

The scope for passive heat recovery is limited by the relatively low temperature of the exhaust air in comparison to the required inlet air temperature. A heat pump can be used to increase the amount of heat recovery by up-grading the heat in the exhaust stream to a higher level. A typical installation, such as that in Figure 4-4, will also include some passive heat recovery. Such installations can be driven by electric motors or combustion engines. The use of combustion engines offers the possibility of additional heat recovery from the engine exhaust and cooling jacket. Note that the optimum solution has passive heat recovery first followed by heat pumping. Although a heat pump could do the whole job, it would lead to both higher running costs and more capital investment.

<u>Utility System Design</u>

The utility system supplies the process heat that cannot be obtained from process heat recovery and therefore plays an important role in determining the operating costs of a process. Again, many different types of utility systems can be used in the malt kilning process. Typical systems include direct firing of oil or gas in the air stream or the use of hot water for indirect heating.

<u>Direct and Indirect Heating</u>. Direct heating represents a very efficient way to use fuel, with the thermal efficiency reaching over 90%. However, the disadvantage is that the malt comes into contact with combustion products which contain nitrogen oxides. In some circumstances this can be detrimental to product quality.

The use of hot water for indirect air heating removes this problem, but the boiler efficiency may be lower, only 80%. Also the losses associated with hot water distribution further reduce the thermal efficiency.

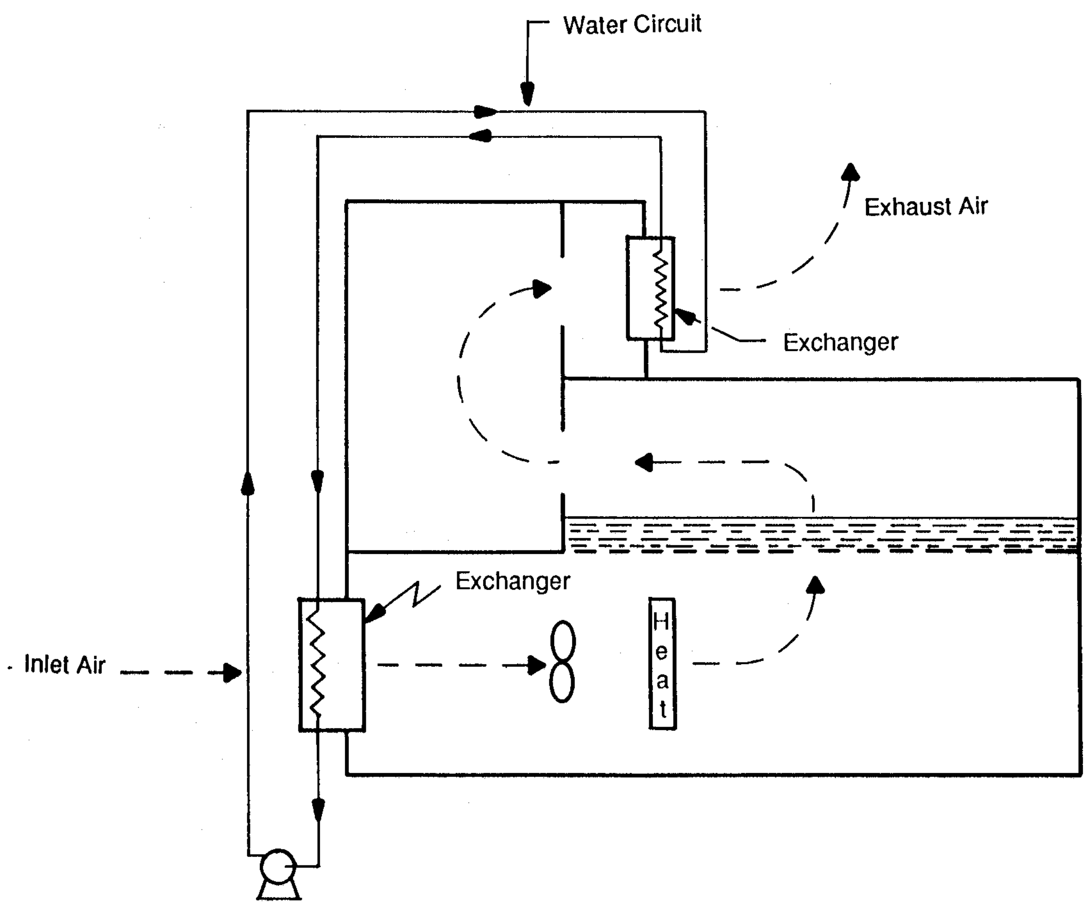


Figure 4-3b. Indirect Heat Recovery in the Malt Kilning Process - Run Around Coil

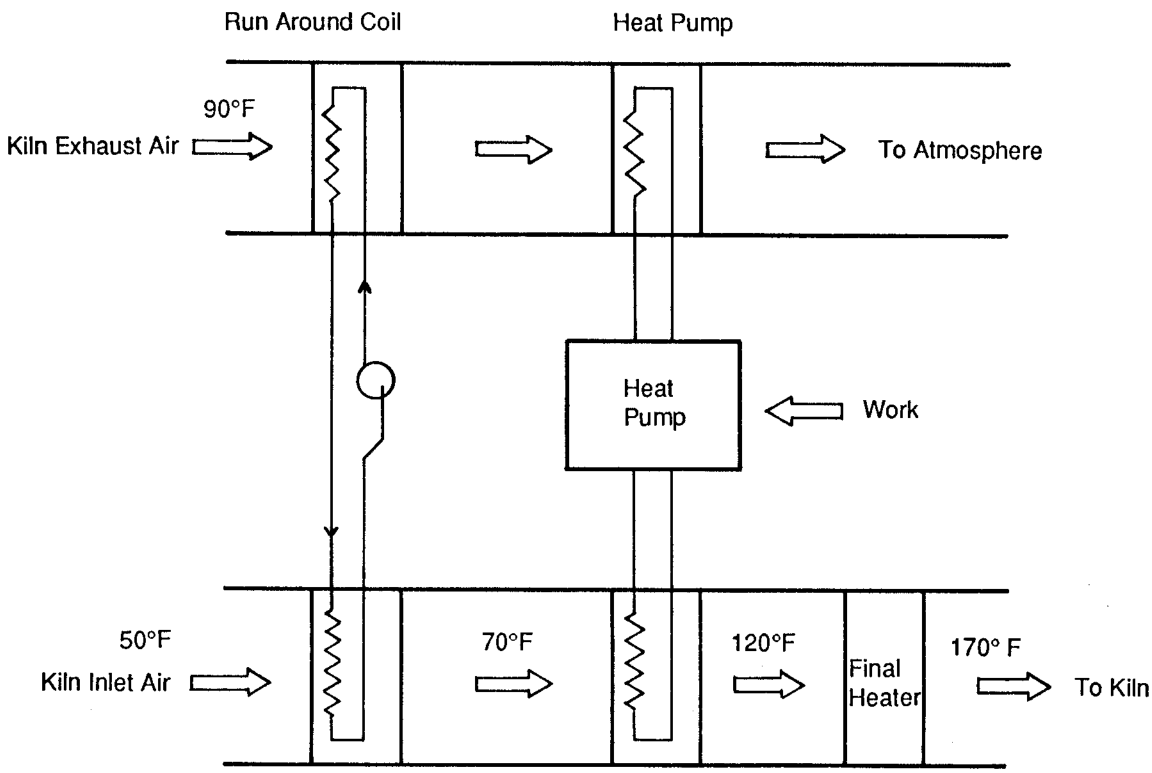


Figure 4-4. Heat Pump Applications in the Malt Kilning Process

Cogeneration. Cogeneration represents another technique for reducing energy costs. However, unlike most of the other techniques that have been described, the cost savings do not come from reducing the primary energy consumption of the plant. In a cogeneration system fuel is burned to produce shaft work (or electricity), in addition to heat for the process. The work (or electricity) is more valuable than the fuel that was burned to produce it, resulting in a reduction in operating costs.

A very common example of a cogeneration scheme is a boiler producing high pressure steam which is expanded to a lower pressure in a steam turbine. This type of installation could be used to supply low pressure steam for air heating in the malt kilning process.

COMPARISON OF ALTERNATIVES

All the measures described in the preceding section will save energy related operating costs. But, with so many alternatives, what selection criteria should be used for determining which projects to implement? It is important to realize that projects cannot be assessed in isolation; there can be interaction between them. For example, reducing the steam demand of a process by installing heat recovery equipment will affect the economics of a proposed cogeneration plant. Understanding these interactions is important when comparing projects and developing an energy cost reduction strategy.

Interaction Between Energy Saving Projects

As demonstrated by the malt kilning example, installation of one project can mean that another is rejected. For example, if an air to air heat exchanger is used to recover heat from the kiln exhaust, then run around coils cannot be used. This is a simple either/or situation. However, some of the interaction between projects can be more subtle and it is possible to define 'exclusive' projects, 'domino' projects, and 'independent' projects.

'Exclusive' projects are similar to the one described above. Two different options exist for doing basically the same job; when one is selected the other must be rejected. Other types of 'exclusive' projects occur when one heat source could be used to satisfy two alternative users, or two alternative heat sources could satisfy a single heat user. Clearly, when a decision is made to match two particularly streams, the other match is no longer possible.

'Domino' projects are projects which, if implemented, could have a downstream effect on other heat recovery projects. For example, reducing the flowrate of air to a malt kiln would reduce the air heating duty. However, it would also have a 'downstream' effect, in that there would be less heat to recover in the exhaust. This would then effect the economics of heat recovery from the exhaust.

One category of 'domino' projects involves changing the quantity of heat which is available for recovery in other projects or possibly even the time at which heat is available (in batch processes). The other category involves changing the cost of the utility being saved. For instance, installation of a cogeneration scheme can reduce the cost of steam to a level where other energy reduction schemes no longer have good paybacks. It is in this sense that a cogeneration scheme 'competes' with heat pumping.

'Independent' projects are ones which have no effect on one another. Either, neither, or both can be installed and each must be justified independently.

It is important that the status (independent, exclusive or domino) of each potential project is established in order that a strategy for implementing all those measures can be developed. Generally it will be best to implement domino projects first so that the basis on which all subsequent projects are developed is firm. Installation of a domino project after other projects are installed can be detrimental to the economics of those projects. The decision on which exclusive and independent projects would then be implemented can generally be based on economic and operability criteria.

The Importance of Energy Prices

Obviously the cost of primary fuel has a significant impact on the viability of <u>all</u> energy saving projects. If fuel is expensive there will be a great incentive to save energy, and vice versa. The capital cost of heat recovery equipment remains relatively constant as the price of energy varies, sometimes quite dramatically; consequently, the interest in energy saving projects rises and falls with fuel prices.

Changes in the absolute cost of fuel will affect all energy saving projects equally. Lower costs will limit the interest in energy saving projects to the smaller, simple schemes. Higher costs will promote interest in larger more complex schemes. Changes in energy prices after a project is installed can obviously effect the payback. Therefore, the sensitivity should be investigated during the feasibility study. For straightforward heat recovery projects, a fall in fuel cost will increase the payback period. But, for cogeneration schemes the payback could be reduced (because oil will probably drop in price more than electricity). The sensitivities will, therefore, be different for different types of projects and should be understood so that there are no surprises in the future!

The absolute value of fuel cost is not the only important factor. The ratio of fuel to power costs should also be considered, and is particularly important for heat pumping and cogeneration schemes. If the cost of fuel is high relative to power, there will be less incentive to cogenerate and heat pumping will look more attractive. Of course, if the cost of fuel is low relative to power the reverse is true. The economics of heat pumping and cogeneration are very sensitive to changes in the fuel/power price ratio and, again, these sensitivities should be understood up front.

The cost of fuel clearly affects the value of savings, but the fuel value is not necessarily the same as the value of the heat being saved! It is very important to establish the real value of the heat being saved. This is not as simple as it may sound. In complex utility systems the calculations can be quite difficult. These 'real values' are known as marginal utility costs and are discussed in greater detail below.

<u>Marginal Utility Costs</u>. The marginal cost or value of a utility is defined as the incremental cost incurred or saved by generating or saving a 'unit' of utility. The simplest example of marginal costing is steam raising. Assume, for instance, that steam (15 psig) is being raised in a boiler which is 85% efficient and that the energy required to raise steam from the feed water is 1 MMBtu/1000 lb. The fuel that must be fired to raise that steam is actually 1/0.85 = 1.18 MMBtu/1000 lb. So saving 1000 lb/h of steam will result in a fuel saving of 1.18 MMBtu/h. If fuel costs $3.50/MMBtu, the marginal value of steam will be $4.13/1000 lb.

It is important to remember that in retrofit situations only the variable costs associated with utility generation are considered in the marginal costing. Fixed costs associated with capital, labor, buildings etc. are not affected by saving or generating incremental amounts of utilities. (The exception to this is when, for instance, boiler capacity is exceeded and new capital expenditure would be required).

The situation starts to get more complex in cogeneration schemes. The simplest example for illustrative purposes is generating low pressure steam by first expanding high pressure steam through a turbine. In this case assume that the energy required to raise high pressure steam (600 psig, 750°F) is 1.1 MMBtu/1000 lb. If the boiler is 85% efficient the fuel that must be fired is 1.29 MMBtu/1000 lb. At the fuel price above, the value of high pressure steam is $4.53/1000 lb. However, when the high pressure steam is expanded to low pressure, some power is generated which reduces the site electrical demand. The value of this saved power should be credited to the marginal cost of low pressure steam. The actual power generated by the expansion to 15 psig is 0.195 MMBtu/1000 lb (57 kWh/1000 lb). If the incremental cost of imported power is 4 ¢/kWh, the marginal cost of low pressure steam is $2.24/1000 lb.

This clearly demonstrates the importance of knowing the real costs of utilities. A project saving low pressure steam but based incorrectly on the marginal cost of high pressure steam will have a payback period almost twice as long as predicted. In more complex utility systems which have several steam levels, gas turbines, use of steam for boiler feed water heating and so on, it is not always clear what the marginal costs of utilities are. In these situations a thorough analysis must be done by someone experienced in utility system operation. To assist with this, EPRI has funded the development of a computer program called 'APLUS.' APLUS simulates the performance of plant utility systems. The user builds up a model of a utility system by interconnecting units such as boilers, turbines, headers and expansion valves. The use of interactive graphics simplifies this task. The program then calculates the operating cost of the system for the specified utility flowrates. By making incremental changes in the utility flows, the appropriate costs, or savings, are calculated. APLUS is available through the EPRI Software Center.

Generating the Alternatives

So far in this section we have discussed:

- the areas in which energy savings may be available,
- the types of projects that may be possible,
- some of the factors which affect project selection,
- independent, exclusive and domino projects,
- the importance of fuel prices, and
- marginal utility costs.

This information helps to define the environment in which the various energy saving technologies compete. It also serves to highlight the importance of considering technologies which are alternatives to heat pumping in order to ensure that the heat pump really is the best option. However, an important subject must still be addressed - how can the various options be generated? Ideally a systematic approach should be used. This will ensure that all the alternatives are found and highlight good heat pumping opportunities.

Pinch Technology is such an approach and has been developed into a powerful tool for analyzing process energy saving options. It allows identification of good heat recovery and heat pumping opportunities and provides a means of determining the impact of various heat recovery options on the entire process.

A description of Pinch Technology and how it can be used to find good heat pumping projects appears in the following sections. The description is a brief and simplified treatment of a technology which has wide ranging implications for industrial energy utilization. However, the essential aspects of the technology are presented and their importance explained. Detailed information about Pinch Technology can be found in a companion document to this Resource Guide, which will be published by EPRI in late 1988, entitled 'Pinch Technology Primer.' The Bibliography also lists useful references.

PINCH TECHNOLOGY - THE RIGOROUS TOOL FOR COMPARATIVE ANALYSIS

Pinch Technology offers a new approach to the design of industrial processes. It gives a clear picture of the energy flows in a process and produces designs which consume less energy and cost less to build than those developed using conventional techniques. It is based on a fundamental thermodynamic analysis of the process and gives proper consideration to actual site energy and capital costs. The technology allows the optimal level of heat recovery to be determined for any process, and guarantees that a design achieving this level of recovery can be obtained. The techniques have been applied to batch and continuous processes in many industrial sectors with outstanding results. Typical energy savings are in the range of 15 to 70% as shown in Table 4-1 and capital savings for new designs normally result.

In addition to improved heat recovery, the technology also addresses all aspects of site heat and power (utility) systems including boilers, steam systems, steam and gas turbines, furnaces, heat pumps, cooling towers, refrigeration systems and the other elements which make up a site utility system! It is the new insight into the ways in which all these systems interact that allows designs produced using Pinch Technology to 'beat the learning curve' described in the preface to this Resource Guide.

The way in which the technology can be used to find and assess energy cost reduction opportunities is explained with reference to the categories of opportunities identified earlier in this chapter; passive heat recovery, process modifications, heat pumping and lower cost utility systems. These will be illustrated using the previous example - the malt kilning process.

AN INTRODUCTION TO PINCH TECHNOLOGY

The technology is based on several very important concepts. These include

- Setting energy consumption and capital cost targets for the process using 'Composite Curves,'
- Designing heat recovery networks to meet these targets using the 'Pinch Principle' and the 'Pinch Design Methodology,'
- Modifying the process to make it inherently more energy and capital efficient using the 'Plus/Minus' principle,
- Placing process equipment in the process using the 'Appropriate Placement Principle,' and
- Selecting site utility systems using the 'Grand Composite Curve.'

However, the first important step in any pinch analysis is 'Data Extraction.'

Table 4-1

RESULTS OF PINCH TECHNOLOGY STUDIES

COMPANY	PROCESS DESCRIPTION	RESULTS
A. E. Staley Co.	Corn Wet Milling starch/syrup production	Energy Savings of 20% at 30 month payback.
Cadbury Typhoo	Batch processing of dairy products.	30% energy savings at less than 1 year payback.
Ciba Geigy Chemicals	Continuous resin manufacturing process. New design.	Overall 15% energy plus capital cost savings.
Ectona Fibres	Continuous cellulose acetate processing.	$170,000/year energy savings at 1 year payback.
Long John International	Whisky distillery. Batch and continuous operations.	Energy savings 50% at less than 2 years payback. Significant debottlenecking.
Olin Corporation	Syngas/Petrochemical complex.	Energy savings in excess of 20% identified.
Pfizer, Inc.	Food additives.	Energy savings of 30% identified at good payback.
Shell UK	Crude Oil refinery preheat train.	12 operating cases designed for. Attractive energy savings.
Southern California Edison	Tissue paper mill.	Savings of 25% identified with better payback than proposed cogeneration scheme.
Unilever	Edible oil refinery. Totally batch operation. Wide range of feedstocks.	Energy savings 70% plus 15% increased capacity.
U.S. Brewery	Beer manufacture.	Fuel savings of 45% identified.

Data Extraction

In order to complete a pinch analysis it is necessary to identify the process streams which need to be heated and cooled. Any stream can be identified as either a 'hot' stream (one which gives up heat) or a 'cold' stream (one which receives heat). These streams can be defined in terms of a starting or 'supply' temperature, an ending, or 'target' temperature and an enthalpy change. For example, a hot stream might cool from a supply temperature of 190°F to a target temperature of 90°F, with an enthalpy change of 200,000 Btu/h. If this hot stream is matched, in a heat exchanger, with a cold stream heating from 70°F to 120°F and an enthalpy change of 200,000 Btu/h, the exchange can be represented on a temperature-heat content (enthalpy) diagram as shown in Figure 4-5.

Data extraction is a very important step because the resulting stream data form the basis of the pinch analysis. Having good data at this stage is essential.

The malt kilning example is shown in Figure 4-6. Two kilns are in operation, one pre-break (i.e., the exhaust air is cool and moist) and one post-break (i.e., the exhaust air is warm and dry). In this problem there are four streams, two hot and two cold. The two cold streams are the air supplies to each kiln and the hot streams are the exhaust streams from each kiln. Table 4-2 shows the stream data. Note that stream 2, the exhaust air from the first kiln, has a substantial amount of latent heat to give up because it is close to saturation. This is reflected by splitting the stream into two sub streams (segments) which correspond to the condensing curve.

How Much Passive Heat Recovery?

Having obtained stream data for all the process streams it is possible to set targets for the amount of passive heat recovery possible and the utility requirements of the process. The tools used for this purpose are the hot and cold composite curves. In Figure 4-7 all the hot streams defined in Table 4-2 have been combined into a single curve known as the 'hot composite curve' and all cold streams have been combined to form the cold composite curve. Both curves are plotted on the same temperature-enthalpy axes.

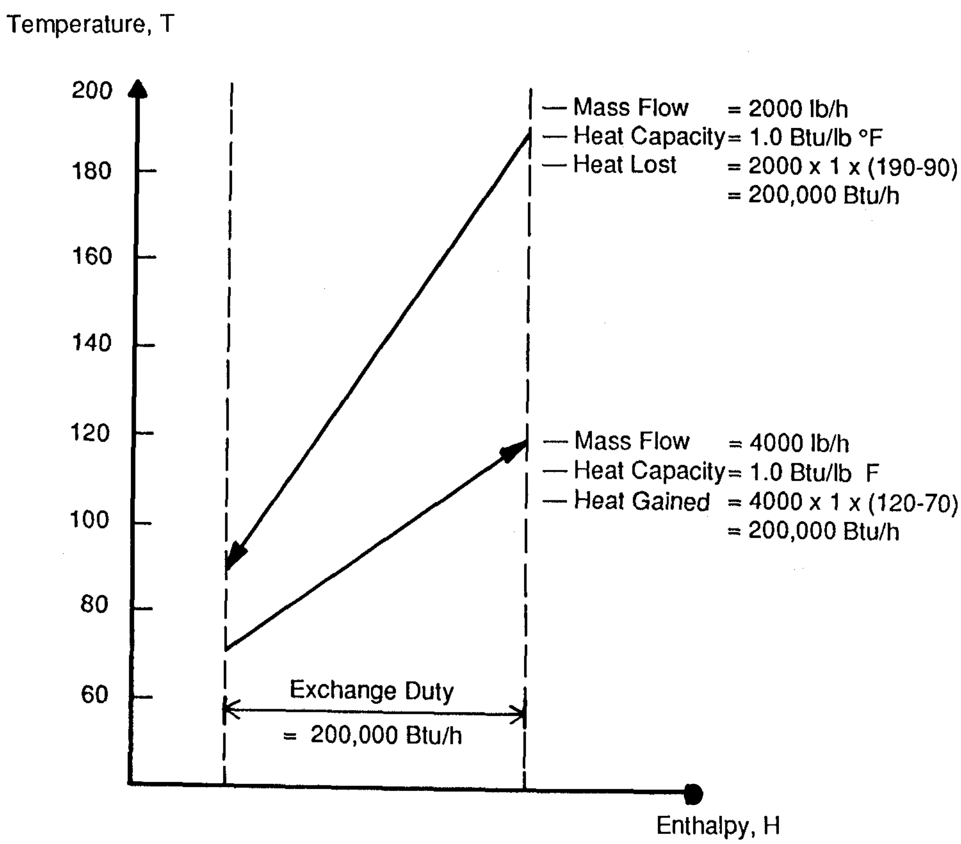


Figure 4-5. Matching Streams on the T-H Diagram

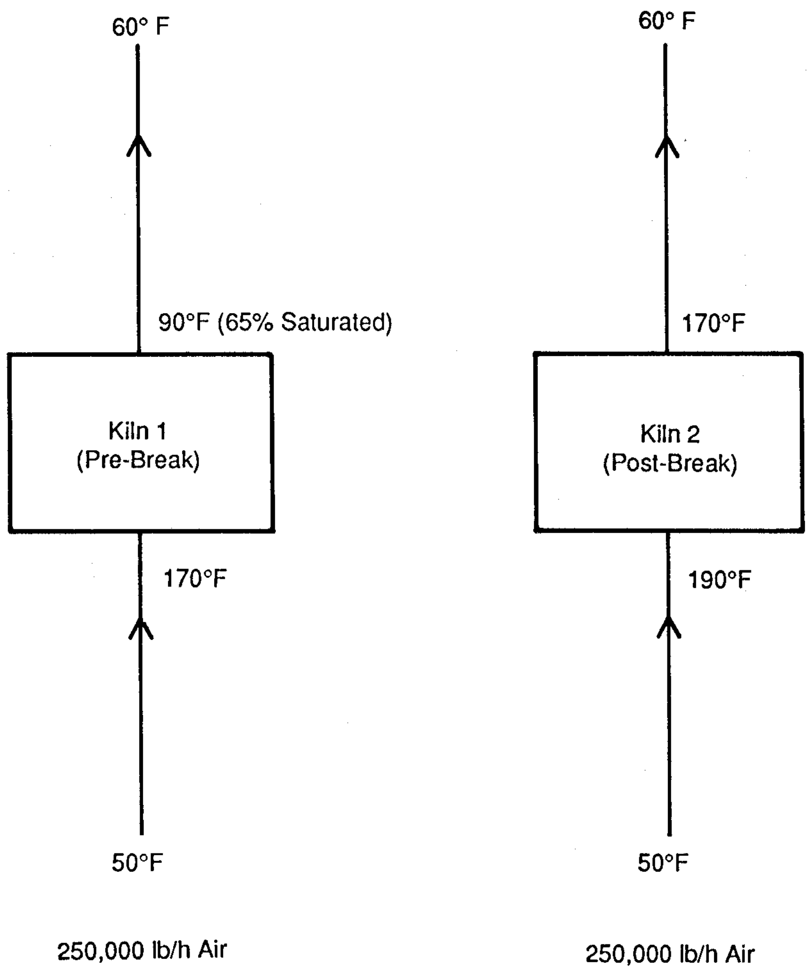


Figure 4-6. Process Data for the Malt Kilning Example

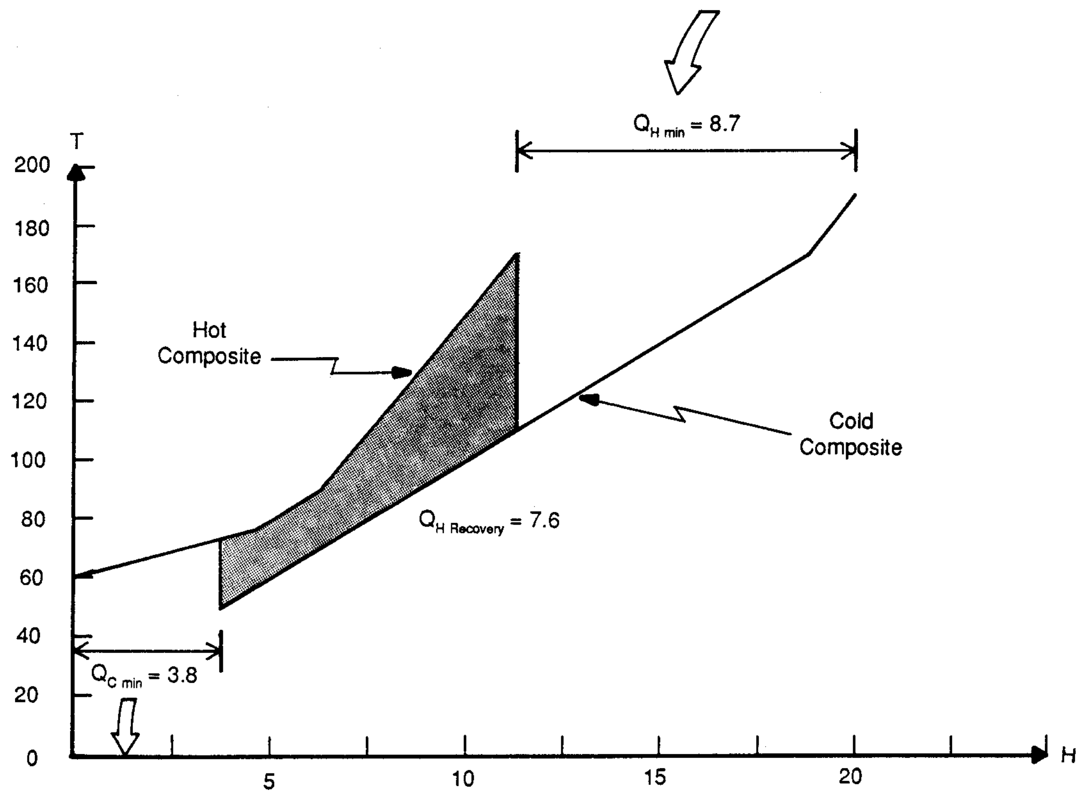


Figure 4-7. Composite Curves for the Malt Kilning Example

Table 4-2

STREAM DATA FOR MALT KILNING EXAMPLE

Stream Number	Stream Name	Stream Type	Supply Temperature °F	Target Temperature °F	Enthalpy Change MMBtu/h
1	Kiln 1 Air	Cold	50	170	7.5
2	Kiln 1 Exhaust	Hot	90 77	77 60	0.9 3.6
3	Kiln 2 Air	Cold	50	190	8.8
4	Kiln 2 Exhaust	Hot	170	60	6.9

The overlap between the hot and cold composite curves is a measure of the maximum scope for passive heat recovery between hot and cold process streams in the plant. This is analogous to the overlap between the single streams in Figure 4-5.

At the hot end of the composite curves, there is an 'overshoot' of the cold composite curve where no corresponding hot streams are available for heat exchange. This overshoot represents the minimum amount of utility heating which must be supplied and is known as the 'hot utility target,' Q_{Hmin}. Similarly, the overshoot of the hot composite curve at the cold end represents the minimum amount of utility cooling which is required. This is the 'cold utility target,' Q_{Cmin}. The composite curves allow the minimum energy consumption of a process to be predicted <u>before</u> the design of the heat exchanger network is determined.

For the example, the hot utility target is 8.7 MMBtu/h, the heat recovery target is 7.6 MMBtu/h, and the cold utility target is 3.8 MMBtu/h. It should be noted that in this particular case, the cooling is achieved by venting the exhaust air to atmosphere.

<u>Developing Heat Recovery Projects</u>

Now that targets have been established a heat recovery network which meets those targets must be designed. Pinch Technology provides specific rules on how this can be done.

The point at which the composite curves are closest together is known as the process 'Pinch.' At this point, the temperature difference between the curves is a minimum and is called ΔT_{min}. The pinch divides the system into two thermodynamically separate systems as shown in Figure 4-8. The system above the Pinch is short of heat and the system below the Pinch has a surplus of heat. When the target heating and cooling duties (Q_{Hmin} and Q_{Cmin}) are supplied, the two systems are in heat balance.

If, however, the process consumes X units of hot utility above the target, the extra X units must be transferred across the boundary into the cold system. By the same reasoning, the X units of additional heat must then be rejected into the cold utility to maintain the energy balance around the cold system. It follows from this argument that 'to meet the energy targets, heat must not be transferred across the pinch.' This conclusion is known as the 'Pinch Principle.' There are two

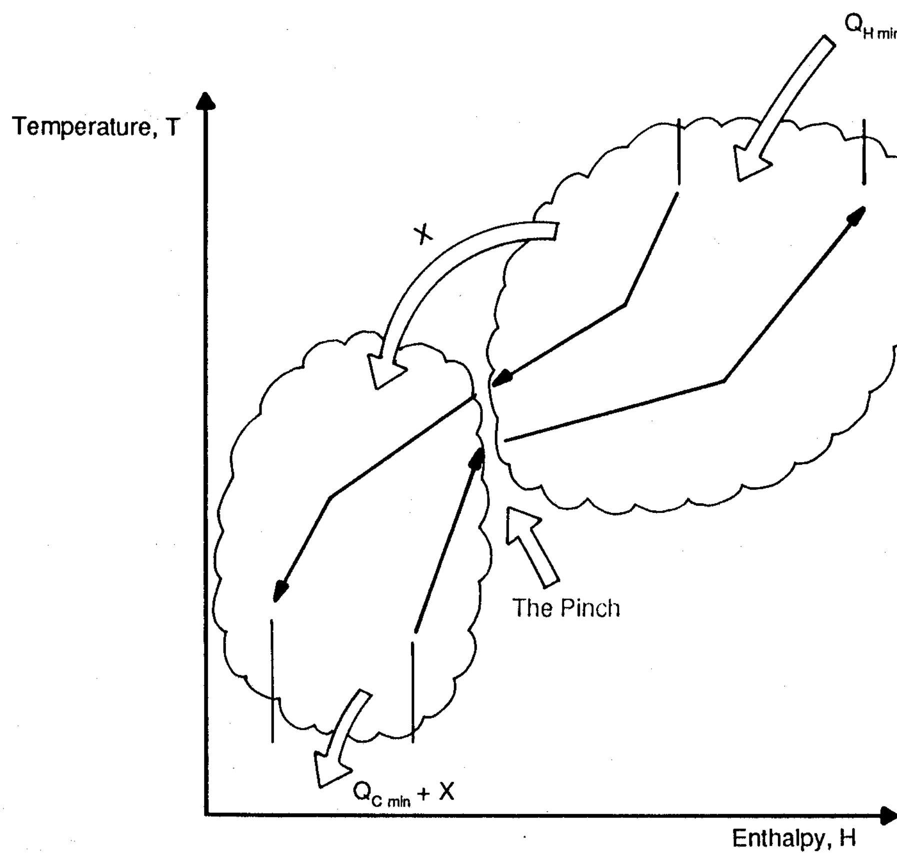


Figure 4-8. The Pinch

corollaries to this principle. For minimum energy consumption the designer must not:

- use a cooling utility above the pinch, or
- use a heating utility below the pinch.

To design a network which guarantees the target energy recovery level the above and below pinch region are kept separate. This results in above pinch and below pinch designs. In both cases the process-to-process heat exchange matches are made first, with the utility to process matches completing the solution. The above and below pinch networks are then brought together and refined. The 'design grid' is a simple and convenient way of representing the heat exchanger network. Streams are represented by horizontal lines. Hot streams are shown running left to right at the top of the grid while cold streams are shown running right to left at the bottom of the grid. Process heat exchangers are shown as circles on the relevant streams connected by a vertical line. Utility-process exchangers (heaters and coolers) have symbols, 'H' for heater and 'C' for cooler. Exchanger heat loads are shown underneath the symbols with stream temperatures being shown at various points in the grid. Figure 4-9 shows the completed grid design for the malt kilning example.

The procedure for determining which streams to match is known as the 'Pinch Design Method.' It provides specific rules and design tools which enable networks meeting the targets to be designed. There is no longer any guesswork involved in designing heat recovery systems. From the very outset the energy recovery level is known and achievement of that level is guaranteed.

Applying the Pinch Design Method to a problem will generate a consistent set of projects which can then be assessed for their technical and economic merit.

<u>How Can The Process Be Modified To Reduce Its Energy Consumption</u>?

Figure 4-8 showed how the pinch divides the process into two thermodynamically separate systems. The above pinch system is in heat balance when hot utility Q_{Hmin} is supplied and the below pinch system is in heat balance when cold utility Q_{Cmin} is supplied.

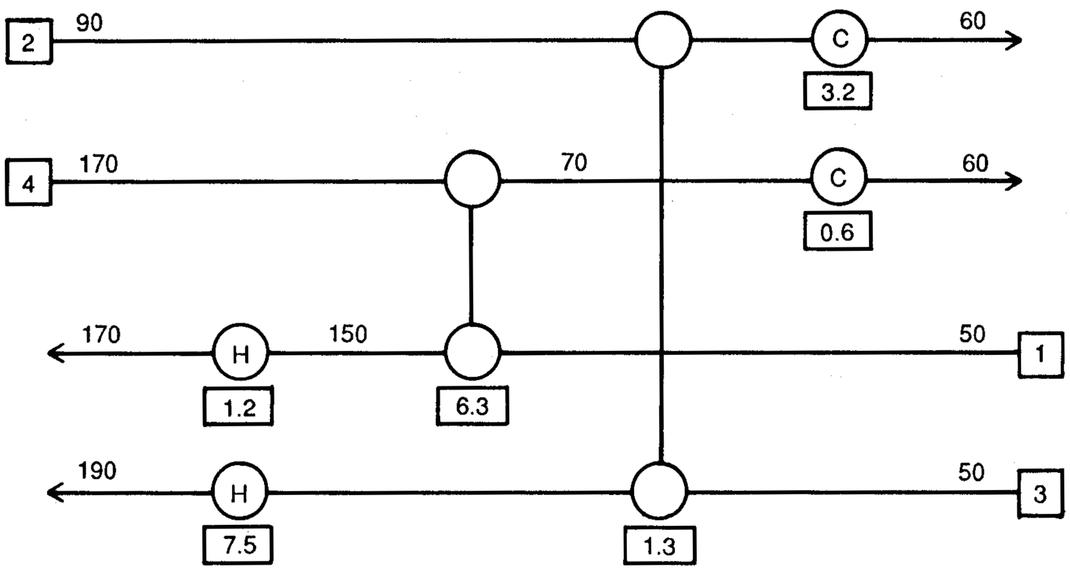


Figure 4-9. Final Network Design for the Malt Kilning Example

Is there any way in which the targets can be reduced? Consider the system above the pinch in Figure 4-10. The only way that the hot utility target can be reduced is either by reducing the heat duty of cold streams (-) or by increasing the heat content of hot streams (+). Similarly below the pinch, the only way that the cold utility target can be reduced is either by reducing the heat content of hot streams below the pinch (-) or by increasing the heat duty of cold streams below the pinch (+). This is the 'Plus/Minus' principle.

As an example, consider the composite curves in Figure 4-11a. The process represented by these curves includes independent evaporation and condensation steps. The condensation occurs below the pinch and the evaporation above the pinch. How can the process be modified to reduce energy consumption? If the temperature of the condensing vapor could be increased to a value above the pinch temperature, the heat would become available above the pinch, thus reducing the utility targets as shown in Figure 4-11b. This is in line with the Plus/Minus principle because the amount of heat available from hot streams above the pinch has increased. In practice, the necessary temperature increase can be achieved by increasing the condensation pressure.

This simple example illustrated how process modifications suggested by the composite curves can be translated into actual changes in equipment design or process operation.

<u>Where Do Heat Pumps Fit In</u>?

The energy targets developed so far only take into account the scope for passive heat recovery. What about opportunities for heat pumping?

A heat pump, as defined earlier, is a device which accepts heat at a low level and delivers it at a higher level. The Pinch Principle provides guidelines on how to integrate heat pumps within a process and tools for sizing heat pumps and deciding on the heat sources and uses.

<u>Appropriate Placement Of Heat Pumps</u>. There are three options for placing a heat pump in the process a) both the heat source and user above the pinch, b) both below the pinch, or c) the heat source below the pinch and the heat user above the pinch. Figure 4-12a shows a heat pump operating entirely above the pinch. Q_{in} units of heat are supplied to the heat pump at a low temperature level, W units of

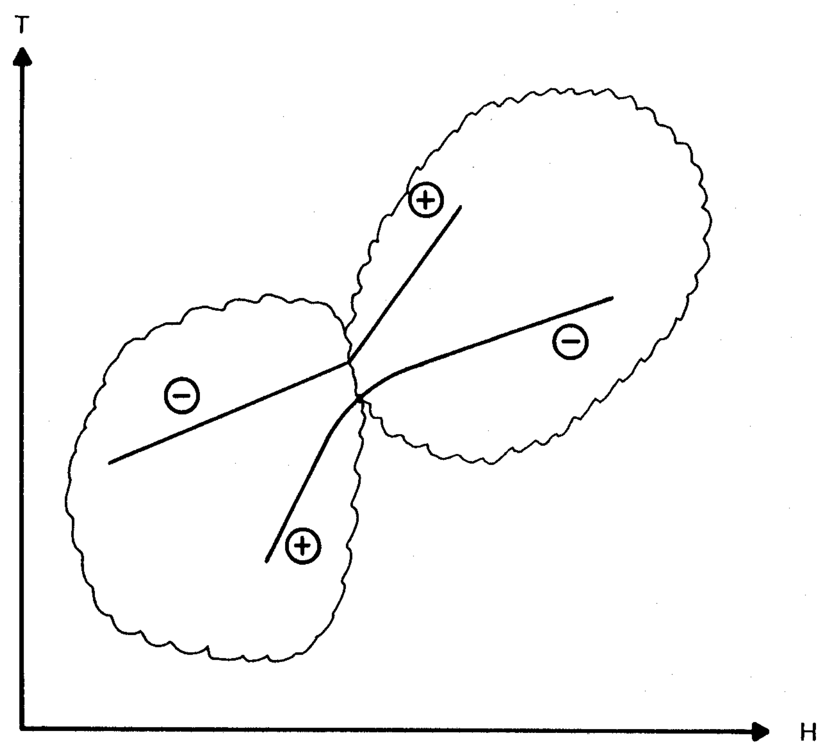


Figure 4-10. The Plus/Minus Principle

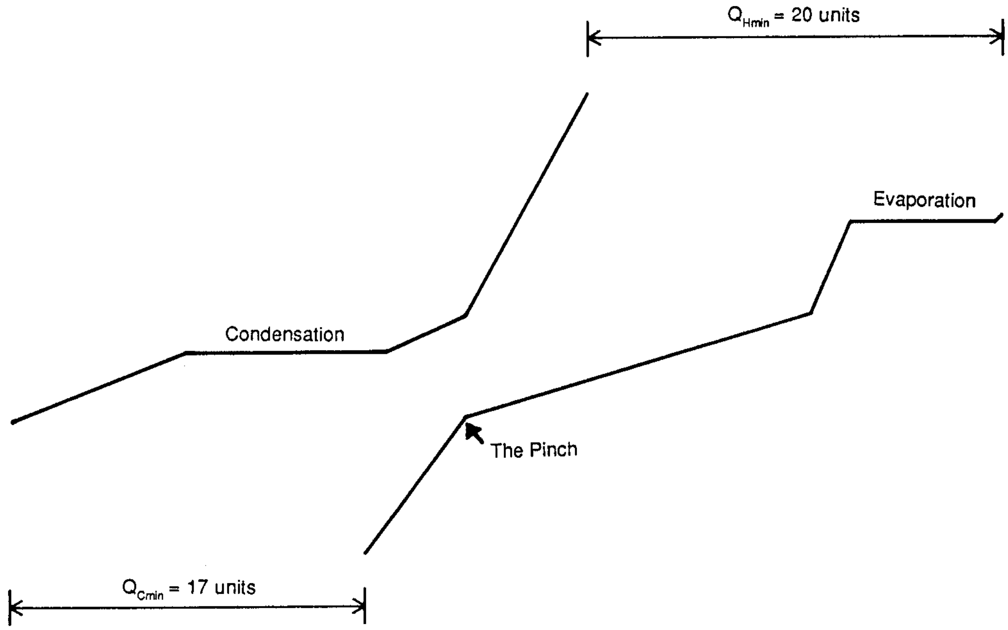


Figure 4-11a. Composite Curves for Basic Process

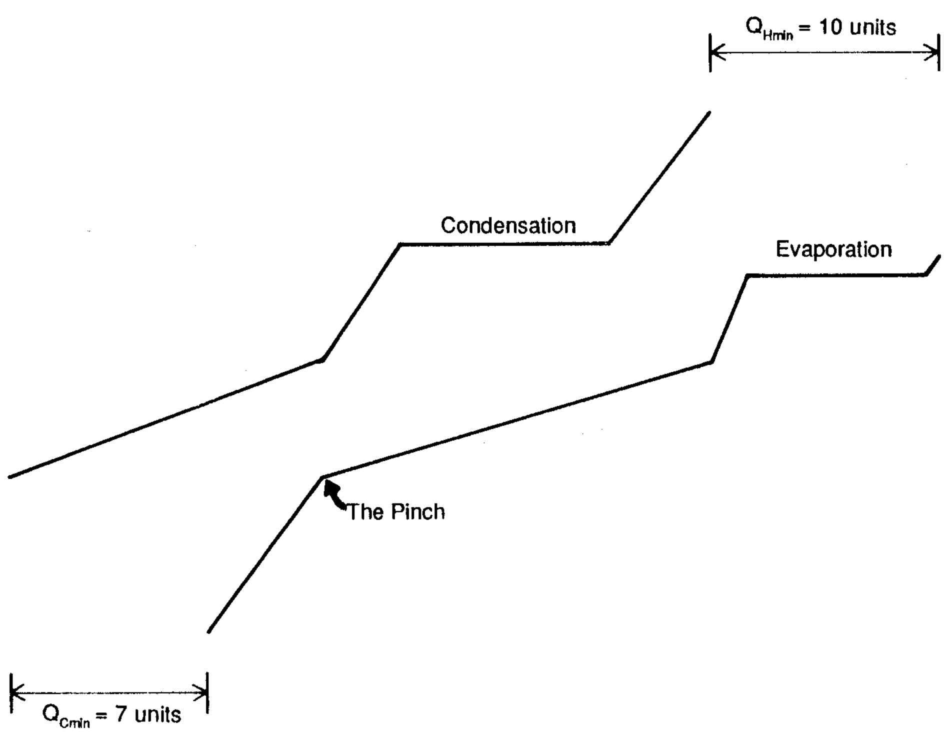


Figure 4-11b. Composite Curves for Modified Process

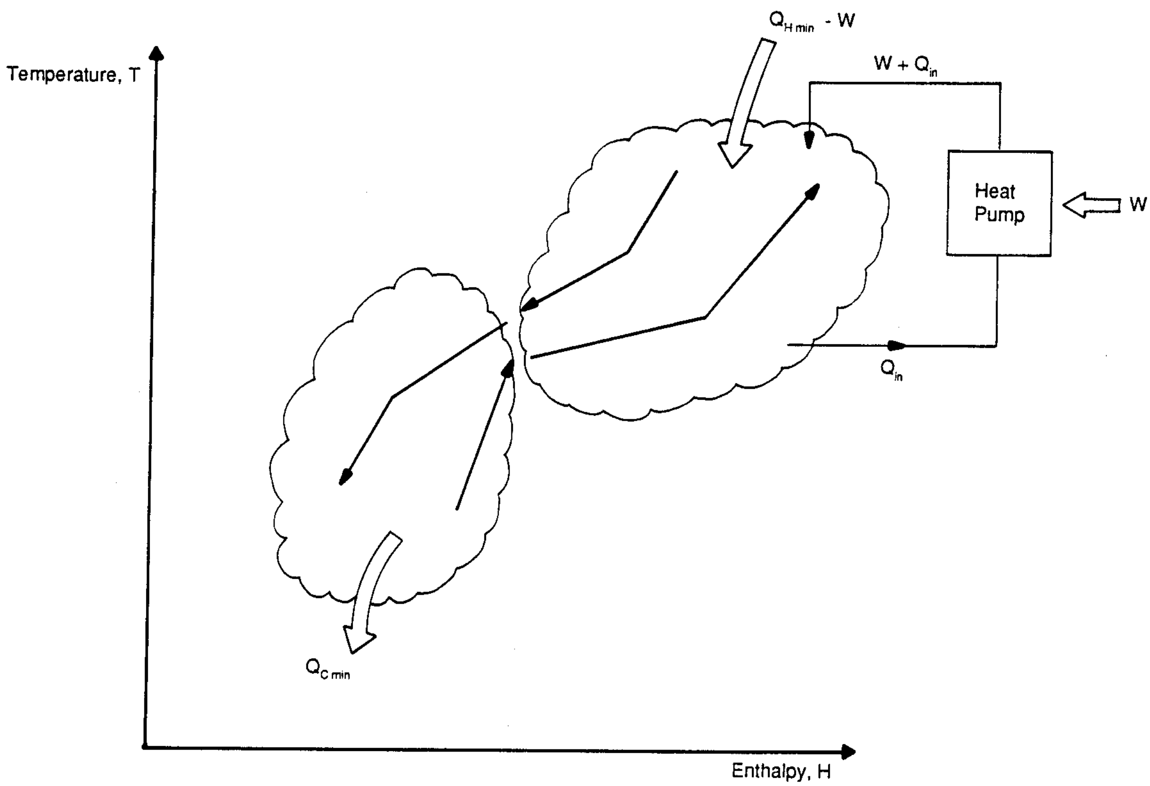


Figure 4-12a. Heat Pump Operating Above the Pinch

work are used to drive the heat pump, and $W + Q_{in}$ units of heat are delivered at a higher level. What is the effect of this on the heat balance above the pinch? The hot utility requirement has been reduced by W units. In other words, W units of work (electricity) have been used in place of W units of steam. This will not be cost effective unless work (electricity) costs less than steam. As this is very rarely the case, the conclusion is that a heat pump is not appropriately placed if both the heat source and user are entirely above the pinch.

A similar argument for a heat pump operating entirely below the pinch is shown in Figure 4-12b. In this case, hot utility use is not reduced at all. Work (electricity) is being degraded into heat and rejected to the cooling water. Again, there is no benefit and this configuration can never be correct.

Now consider a heat pump operating around the pinch, as shown in Figure 4-12c. Q_{in} units of heat are supplied below the pinch, W units of work are used to drive the heat pump and $W + Q_{in}$ units are delivered above the pinch. In this case, the hot utility requirement is reduced by $W + Q_{in}$ and the cold utility requirement is reduced by Q_{in}. This is clearly beneficial and the heat pump has a chance of being economically attractive.

This analysis results in a very important conclusion: a heat pump is 'appropriately placed' only when it accepts heat below the pinch and delivers it above the pinch. How is this of help when designing a heat pump system? The appropriate placement rule, when used in conjunction with a tool called the 'Grand Composite Curve' provides a systematic approach to finding good heat pumping opportunities.

<u>Heat Pumps And The Grand Composite Curve</u>. The Grand Composite is a representation of all the process streams--hot and cold--as a single line on temperature-enthalpy axes and shows the levels at which heat must be supplied to and rejected from the process. This is significant because it allows the amount of hot or cold utility required at any temperature level to be determined. It also allows the operating temperatures and heat loads for heat pump evaporators and condensers to be determined. The Grand Composite Curve is derived from the same data as the composite curve and shares two common features: the same minimum utility requirements and the same pinch location. Figure 4-13 shows the Grand Composite Curve for the malt kilning process.

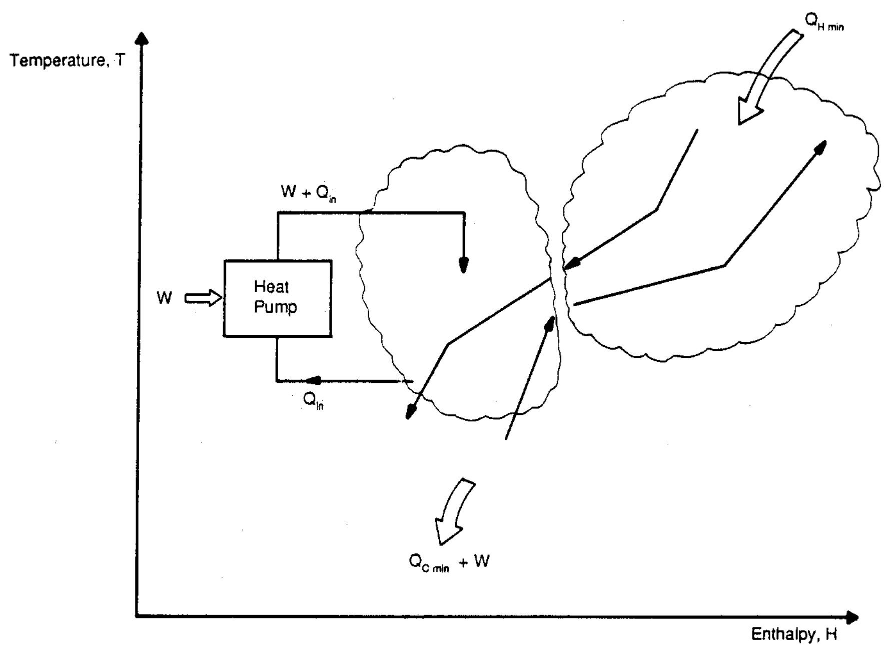


Figure 4-12b. Heat Pump Operating Below the Pinch

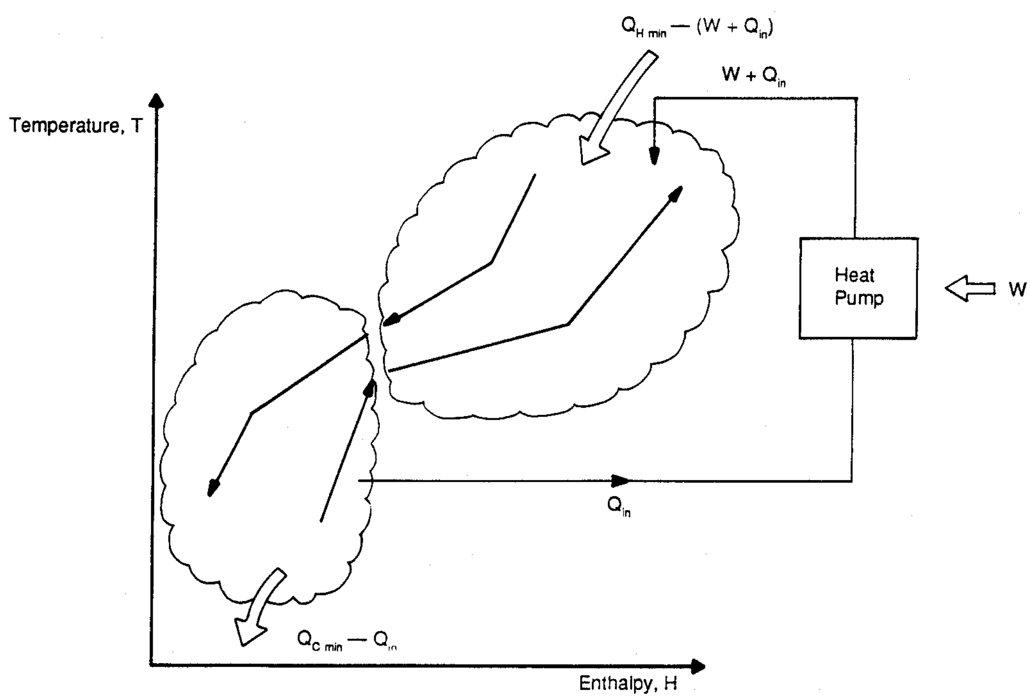


Figure 4-12c. Heat Pump Operating Around the Pinch

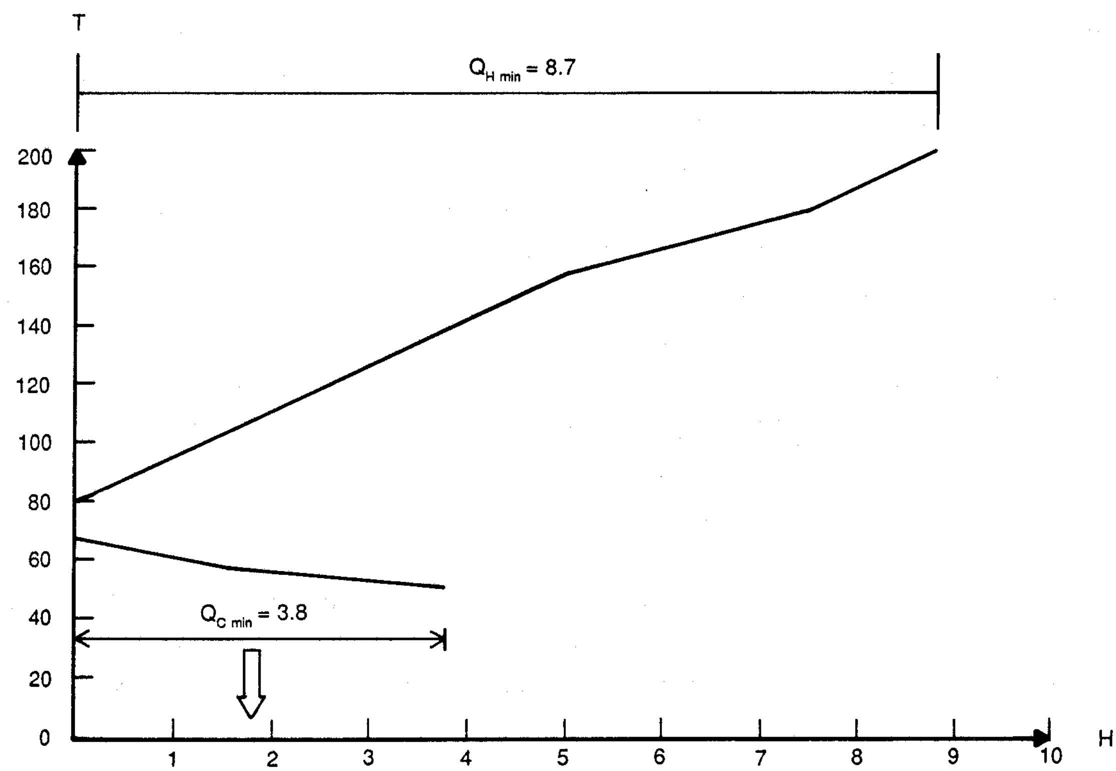


Figure 4-13. Grand Composite Curve for the Malt Kilning Process

If a heat pump is to be beneficial, it must pick up heat below the pinch and reject it above the pinch. It is clear from the grand composite curve in Figure 4-14a that if a heat pump evaporator were to operate at 40°F, 3.8 MMBtu/h of heat could be collected. This heat, plus the work input, must be delivered above the pinch, but where?

The Grand Composite also shows that, above the pinch, the minimum temperature level at which 3.8 MMBtu/h could be used is about 150°F. However, the work supplied to the heat pump increases the amount of heat that has to be rejected. This in turn means that the heat must be delivered at a temperature higher than 150°F in order to accommodate the additional load. In this case, the required delivery temperature is about 170°F. This heat pump would have a COP of approximately 3.5, would require about 450 kW (1.5 MMBtu/h) of driving energy and reduces the heat required by the process by 5.3 MMBtu/h. An alternative heat pump operating with an evaporator temperature of 45°F is shown in Figure 4-14b. This heat pump absorbs 2.0 MMBtu/h of heat below the pinch. The minimum temperature at which this amount of heat can be accepted is 120°F and the practical level for heat delivery (taking into account the work element) is about 130°F. This heat pump would have a COP of about 5.0, would require about 140 kW (0.5 MMBtu/h) of driving energy and reduce the heat required by the process by 2.5 MMBtu/h. Obviously there are many other possibilities, but they can all be explored using the grand composite curve prior to designing the heat pump or the heat recovery network.

When a good opportunity is found, the streams which must be matched with the heat pump evaporator and condenser are identified using the Pinch Design Method. Figure 4-15 is the grid diagram of the heat pump and heat recovery network for the system represented in Figure 4-14b.

Designing Utility Systems

Utility system design has a direct impact on total and marginal energy costs. As discussed earlier, reducing energy costs through modifying the utility system can have a 'domino' effect on the economics of other heat recovery projects, including heat pumps. It is, therefore, important that utility system design is well understood. Pinch Technology shows how appropriate utility systems are selected and sized.

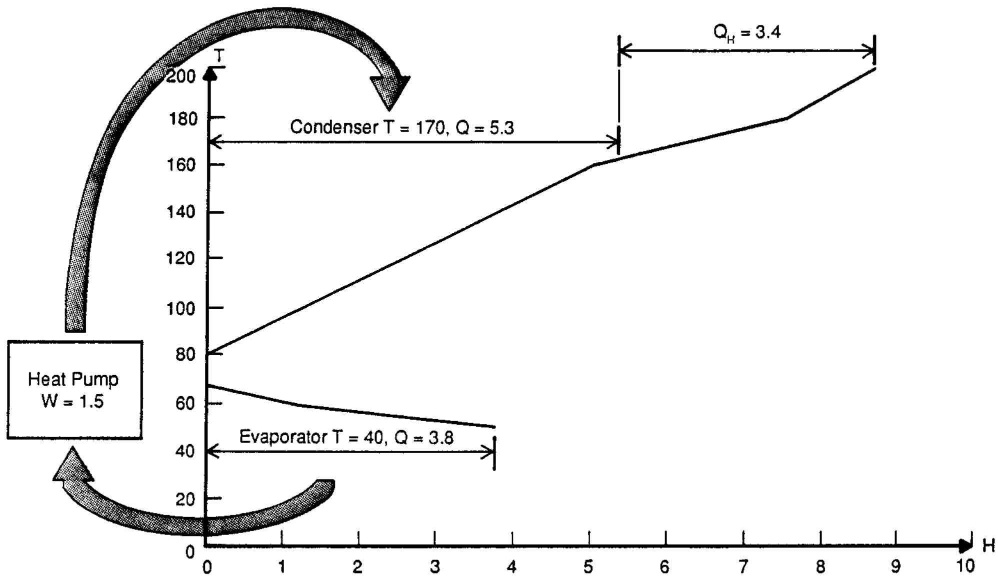


Figure 4-14a. Heat Pump Design A

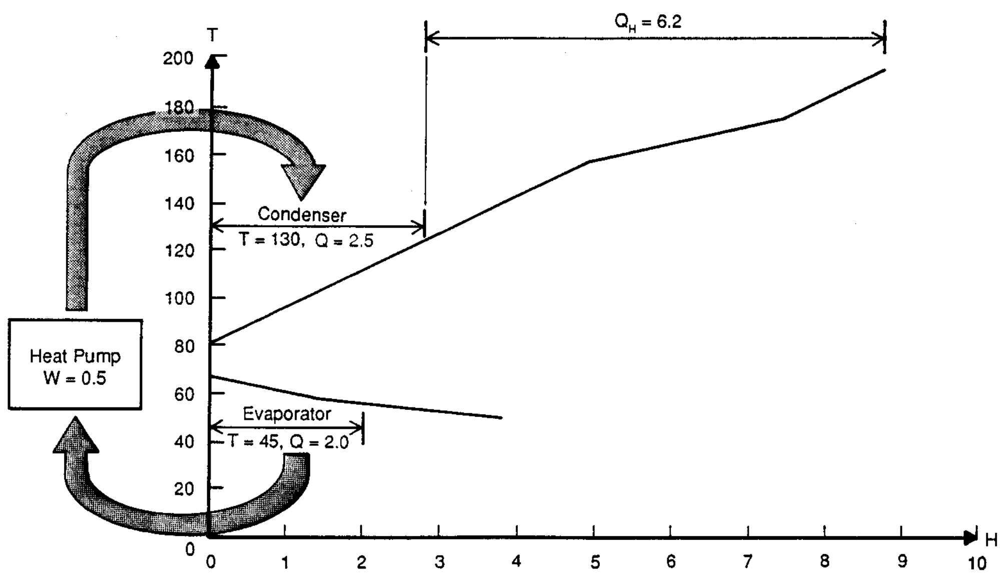


Figure 4-14b. Heat Pump Design B

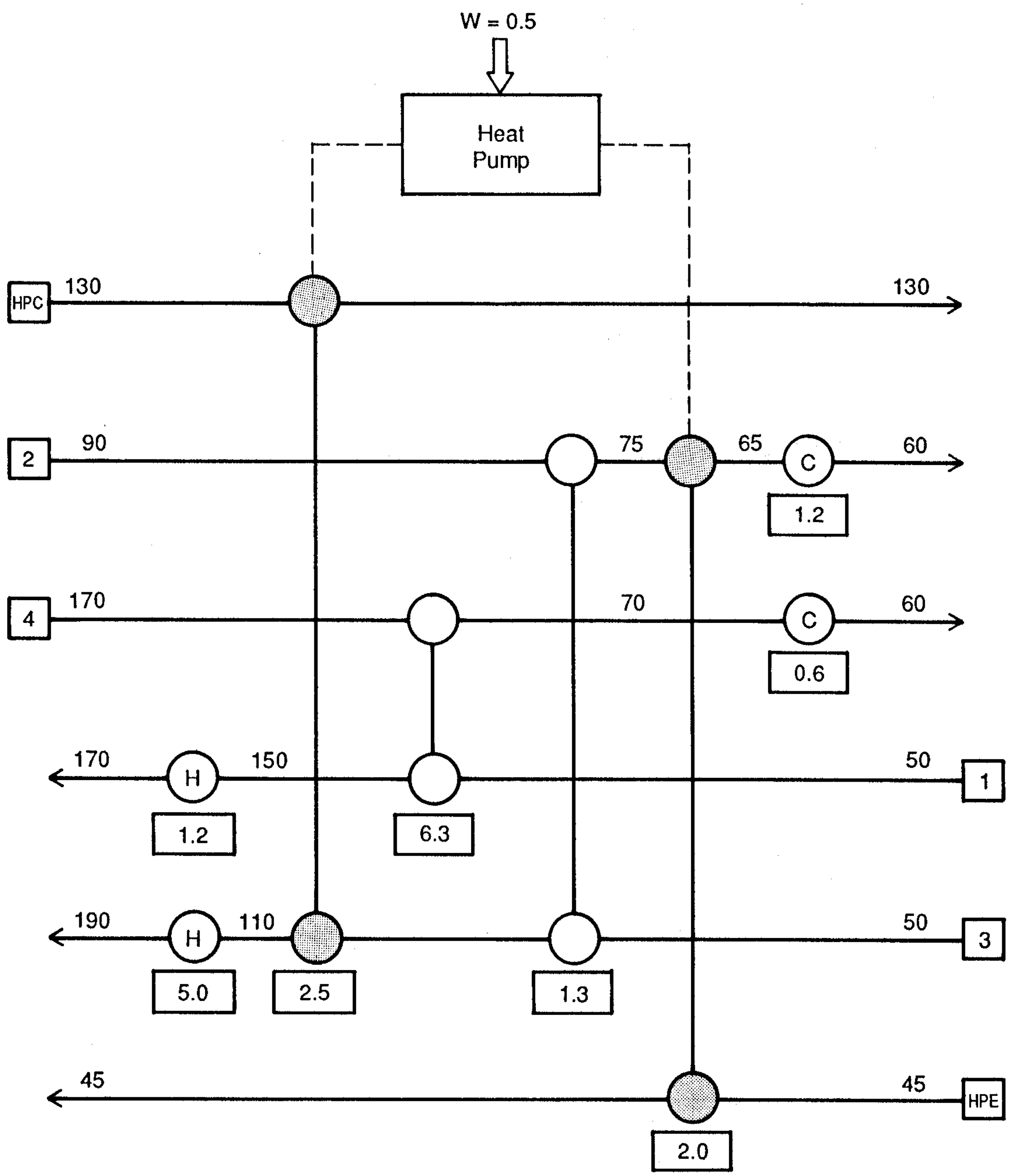


Figure 4-15. Final Network Design for the Malt Kilning Example Incorporating a Heat Pump

The heat duties and temperature levels of utilities can be determined using the grand composite curve, just as the heat pump system loads and levels were determined. This is particularly useful when sizing cogeneration schemes.

What are the options for supplying heat to the malt kilning process? The grand composite in Figure 4-16a shows that the maximum temperature at which the process needs heat is about 200°F. This can be supplied by low pressure steam at, say, 240°F. Because the hot utility target is known, the exact quantity of steam can be calculated, and from an estimate of boiler efficiency, the amount of fuel consumed can be calculated.It would also be possible to supply that low pressure steam by first raising high pressure steam and expanding it through a steam turbine to generate power (Figure 4-16b). Again, because the low pressure steam demand is known it is possible to back-calculate the amount of high pressure steam required, the fuel consumed and the power generated.

Steam-based options are not the only possibilities. A hot water loop could be used (Figure 4-16c). This eliminates the need for steam mains, traps, blowdown and other losses associated with steam systems and may provide a good solution. Another option worth noting is a reciprocating internal combustion engine (Figure 4-16d). This device generates exhaust gas and hot water (from the cooling circuit) which could satisfy the process heat demand, while generating more power than the steam turbine based system.

The ease with which utility options can be generated from the grand composite curve makes it the ideal tool for utility selection. A range of equipment types can be considered. Boilers, steam and gas turbines, reciprocating engines and furnaces represent just some of the hot utility systems which can be considered individually or in combination. Of course, cold utility systems are handled equally as easily; cooling water, mechanical refrigeration and absorption refrigeration representing some of the typical cold utility systems in use. In each case knowledge of the required load and level for each utility allow the operating costs of each utility configuration to be determined.

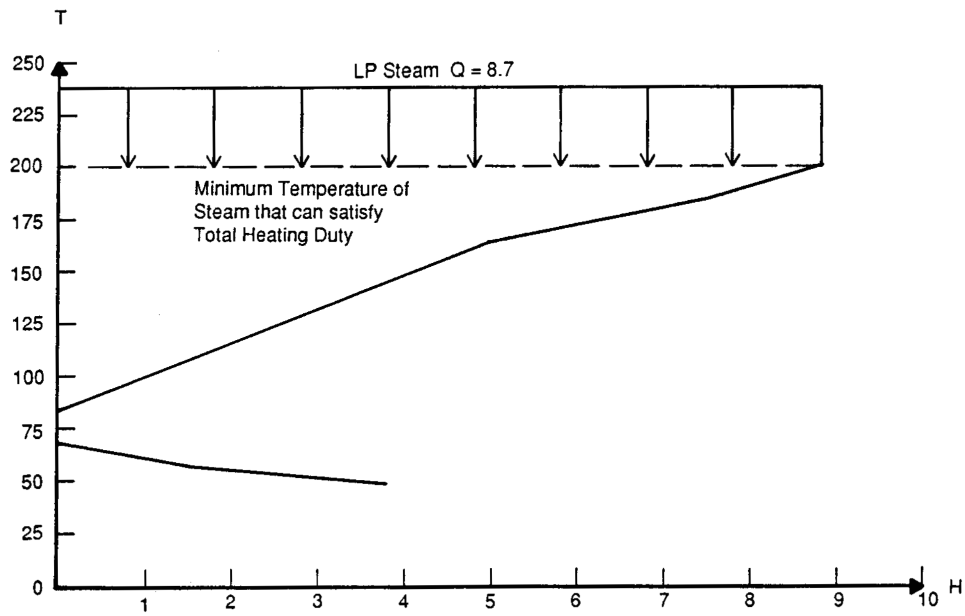


Figure 4-16a. Steam Based Utility System

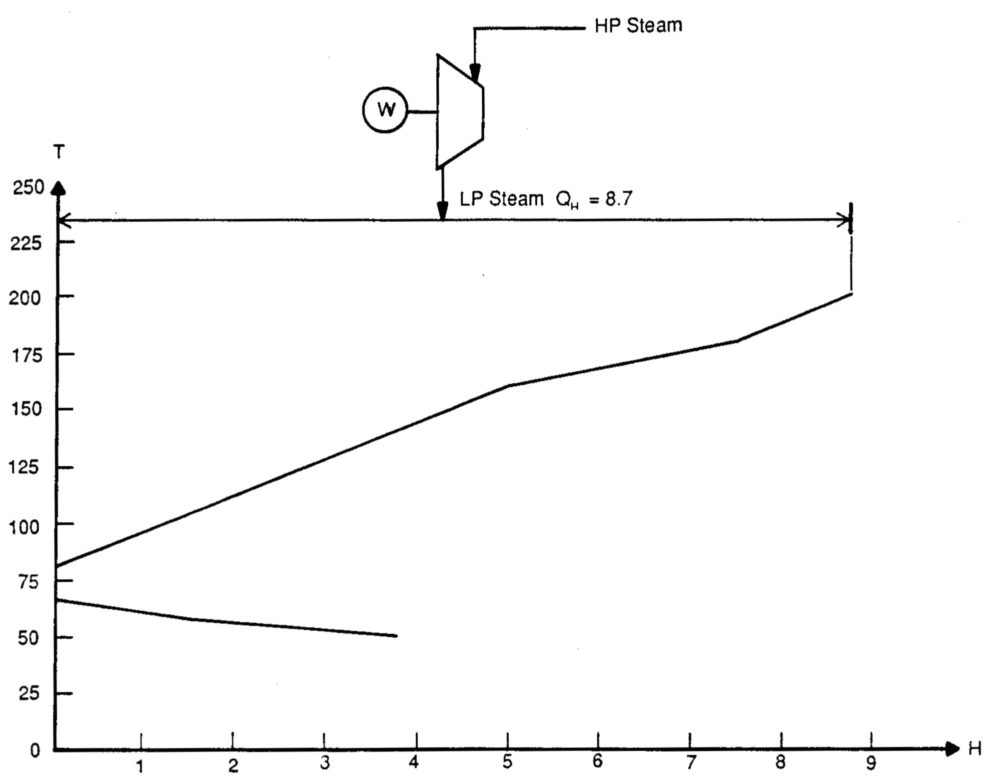


Figure 4-16b. Steam Turbine Based Cogeneration

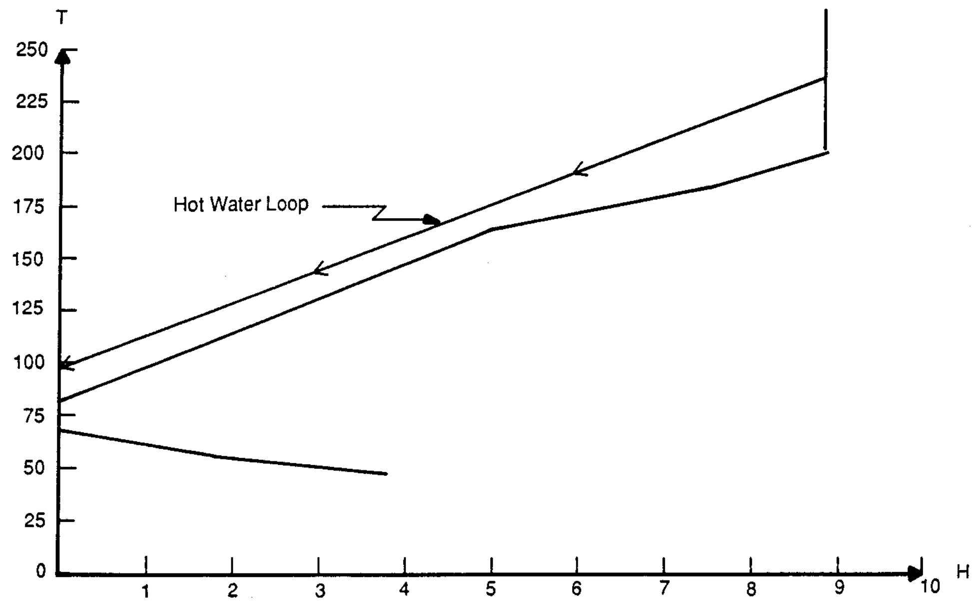


Figure 4-16c. Hot Water Loop

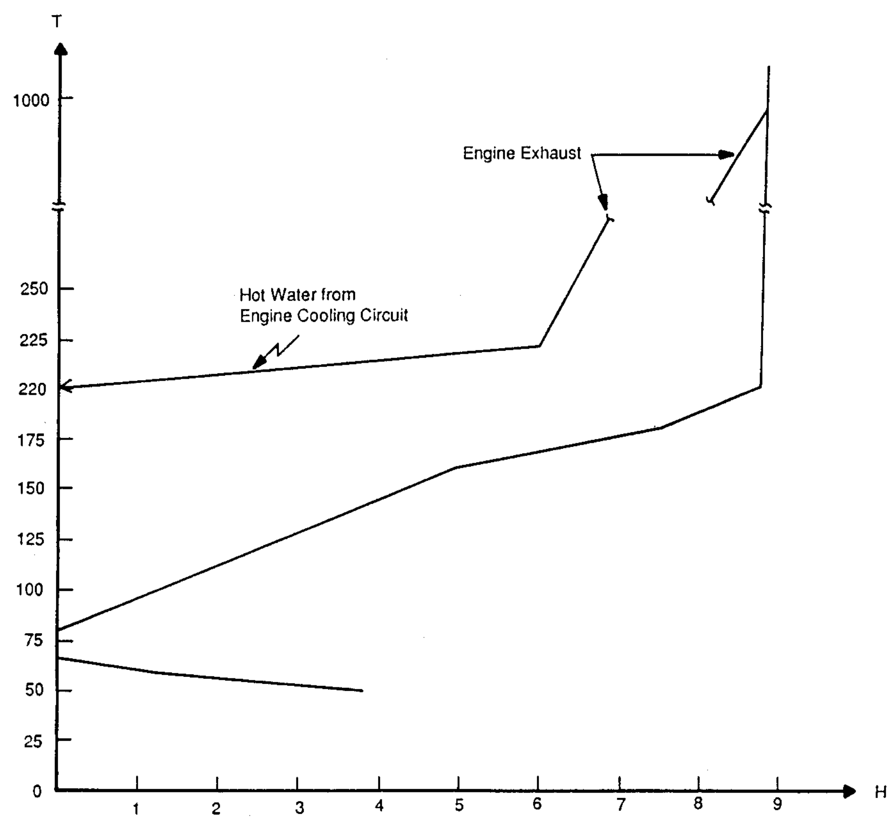


Figure 4-16d. Reciprocating Engine Based Cogeneration

Capital Cost Implications

Up to this point the process has been analyzed taking into account only energy costs. Clearly this is only one-half of the picture. For an energy cost reduction scheme to be attractive it must have an acceptable payback. How can the capital cost implications be determined? The composite curves indicate the temperature driving forces available for heat transfer, in addition to the energy targets. With a knowledge of the process stream film heat transfer coefficients, the driving forces can be translated into a heat exchanger surface area target. The cost of heat exchangers is a strong function of their surface area and with appropriate costing relationships, the area target can be translated into a capital cost target prior to design! Additional influences on cost are the number of heat exchangers and the number of shells (for shell and tube types). Both these factors can also be taken into account.

One design variable which has a strong influence on capital cost is ΔT_{min}, the minimum approach temperature in the heat recovery network. When the composite curves are close together, the temperature driving forces are low, resulting in a high capital cost, but the energy targets are low, resulting in a low energy cost. As the curves are moved further apart the temperature driving forces increase, reducing the capital cost and increasing the energy costs. This is shown in Figure 4-17. Both the annual energy cost and annual capital cost are important and the total cost of operating the heat recovery network is the sum of these two costs. Annual capital cost is the total installed cost of the heat exchanger network annualized over the life of the equipment. Clearly, there is a trade-off between the two cost elements and this general capital/energy trade-off is shown in Figure 4-18. There is an optimum value for ΔT_{min} which results in the lowest total annual cost. Figure 4-19 shows this trade-off for the malt kilning process, with the optimal value of ΔT_{min} being about 25°F. Note that this optimization is done prior to design of the heat recovery network and at this stage only considers passive heat recovery.

Energy saving will also be considered in retrofit situations. Typically an existing plant will consume too much energy for the amount of heat exchange surface area installed as shown in Figure 4-20. This implies that, by rearranging the existing heat exchangers, it is possible to reduce energy consumption to the ideal level. While this is theoretically possible, it is not practically possible in most cases. The realistic option involves adding additional heat exchanger area

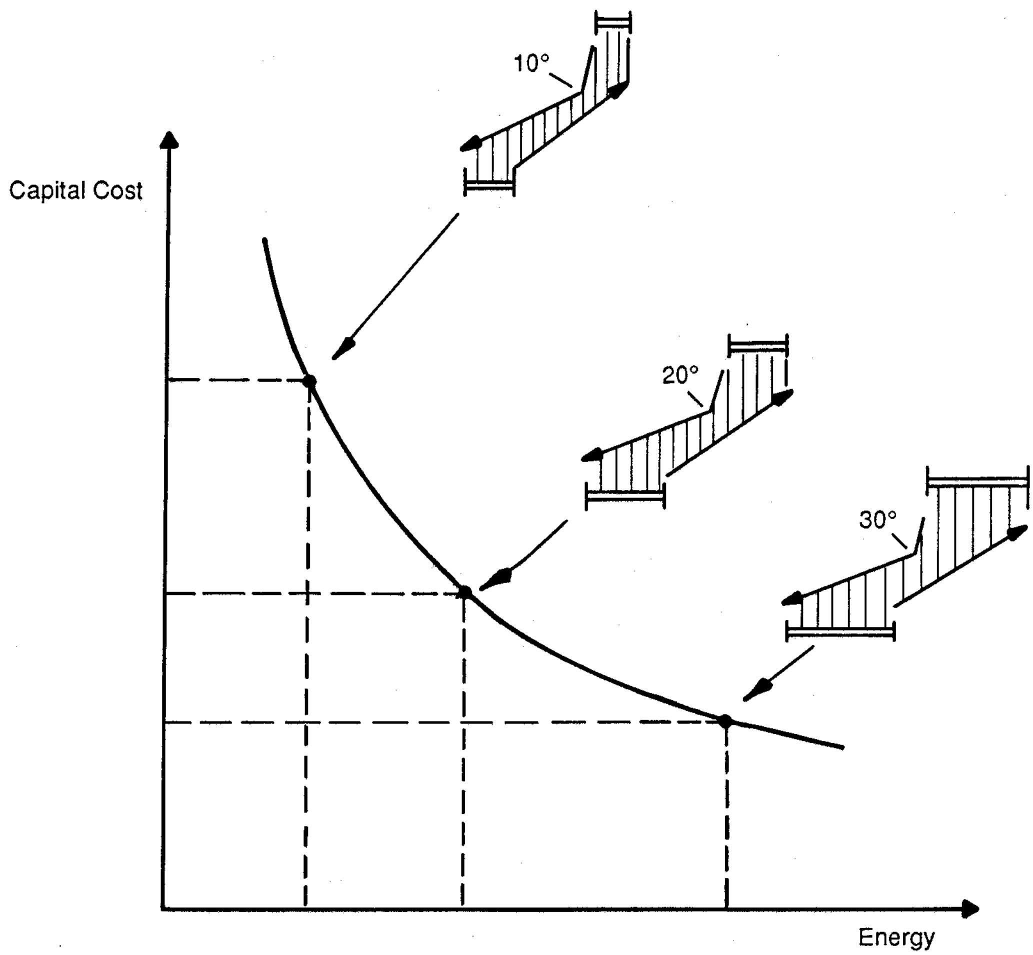


Figure 4-17. The Effect of ΔT_{min} on Energy Consumption and Temperature Driving Forces

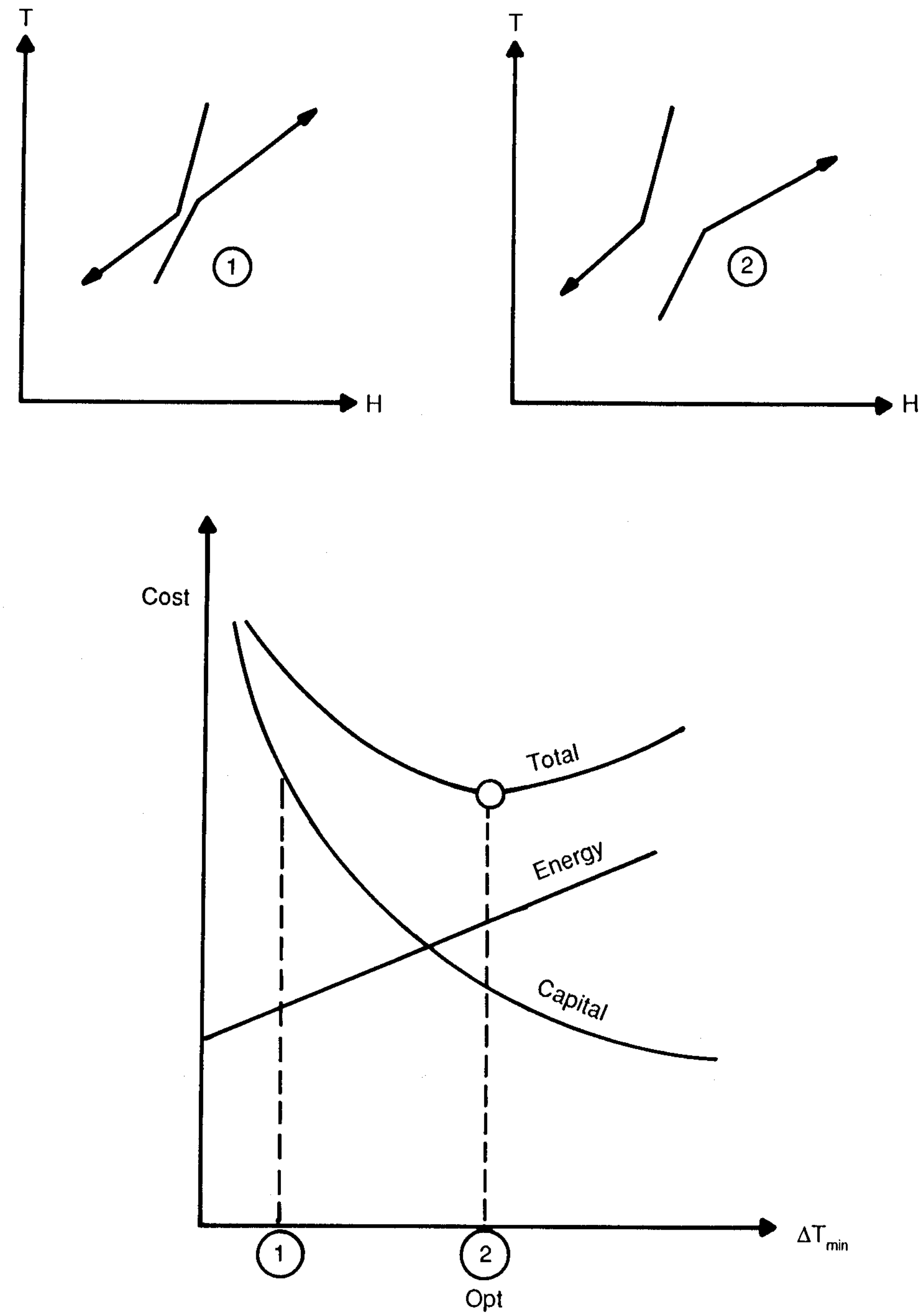


Figure 4-18. The Capital Energy Trade-off

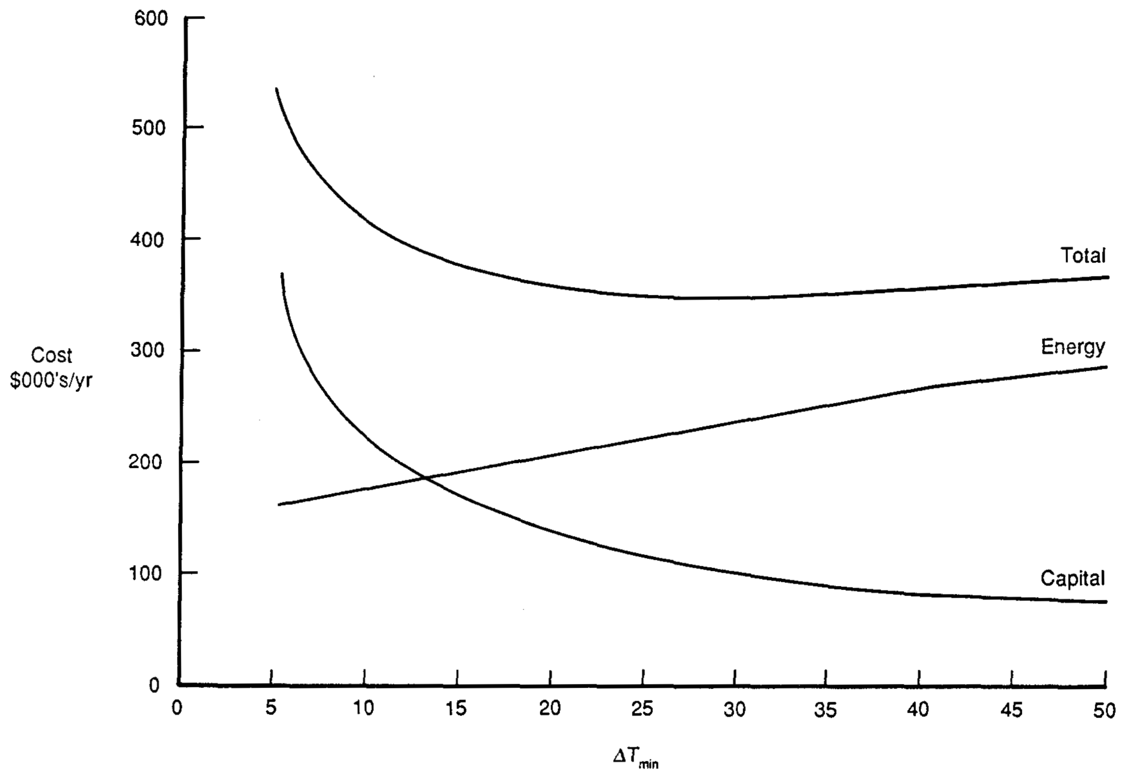


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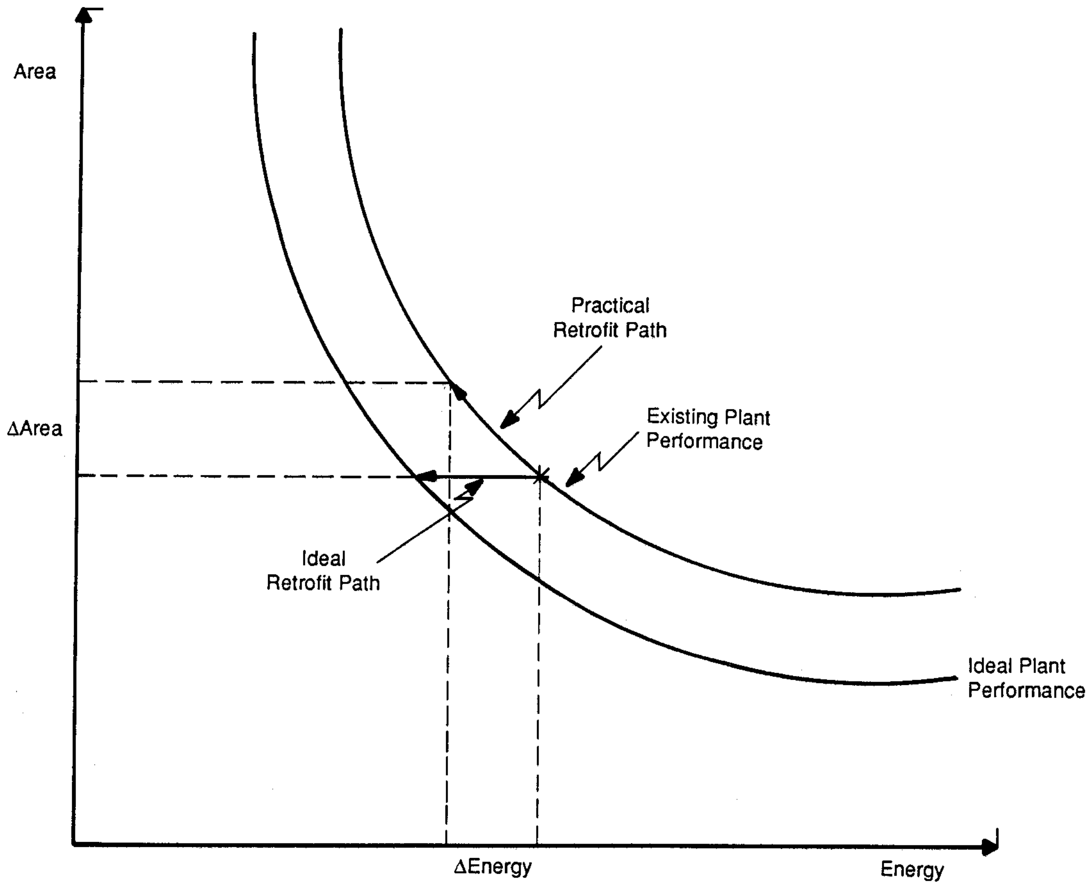
Figure 4-19. The Capital Energy Trade-off for the Malt Kilning Process



$$\text{Payback} = \frac{\Delta\ \text{Capital Cost}}{\Delta\ \text{Energy Cost}}$$

$$\Delta\ \text{Capital Cost} = \Delta\ \text{Area} \times \text{Cost of Area}$$

$$\Delta\ \text{Energy Cost} = \Delta\ \text{Energy} \times \text{Marginal Energy Cost}$$

Figure 4-20. Calculating Retrofit Payback Targets to reduce energy consumption. This route is shown as the 'practical retrofit path' in Figure 4-21. This clearly shows the relationship between the potential energy savings and the additional heat exchange area (and therefore capital cost). By calculating the incremental cost of additional area required and the associated energy savings it is possible to target for retrofit paybacks. At some point, however, the payback available from passive heat recovery projects may not be attractive and heat pumping may be a more economic option. EPRI has funded the development of a computer program called 'HPSCAN' which can assist with the comparison between improved heat recovery and heat pumping. The program uses the concepts of Pinch Technology to quickly investigate the trade-offs in energy and capital between simple heat integration and heat pumping. This allows the appropriate combination of heat integration and heat pumping to be determined for any given set of economics. HPSCAN is available through the EPRI Software Center.

WHY USE PINCH TECHNOLOGY?

Pinch Technology shows clearly the relationships between <u>all</u> the heat sources and heat sinks in a process. Because of this, it offers a consistent approach to identifying and designing passive heat recovery systems, selecting process modifications, designing utility systems, and placing heat pumps.

Without Pinch Technology, heat pump opportunities are identified in isolation from the rest of the process. Some of the dangers of this are highlighted below.

<u>Which Heat Is 'Waste' Heat</u>?

Traditionally, heat pumping opportunities have been identified by finding a source of waste heat along with a suitable sink. The economic performance of the heat pump system would be calculated and if the owners' payback criteria were satisfied, installation would proceed. Such an installation would certainly have reduced energy costs, provided that the economic analysis were correct. However, was a heat pump installation really the best opportunity? Could straightforward passive heat recovery have saved a similar amount of energy at a lower investment? The Pinch Technology concepts that have been described can answer that question.

It is quite possible that many waste heat streams only appear to be waste because of cross pinch heat transfer already present in the heat recovery network. Cross pinch heat transfer not only increases the utility consumptions, but can lead to hot streams being available as 'waste' heat sources which appear to be suitable for heat pumping. Similarly, this also leaves sinks available for the heat delivered by the heat pump. This is shown in Figures 4-21a and b. Figure 4-21a represents the situation where hot streams above the pinch are only used to heat cold streams above the pinch, and hot streams below the pinch are only used to heat cold streams below the pinch. There is no cross pinch heat transfer. In Figure 4-21b, however, hot streams above the pinch are being used to heat cold streams below the pinch. This is cross pinch transfer. Consequently, some hot streams below the pinch cannot transfer heat to cold process streams and therefore become a potential heat source for a heat pump. Similarly, above the pinch some cold streams that should be heated by hot streams above the pinch have now become a potential user for the heat delivered by a heat pump. The source and user would both disappear if the heat exchanger network were rearranged in line with the pinch principles, the utility requirements of the process would be reduced and the need for the heat pump would be eliminated. In many cases it is likely that the revamp of the heat exchanger network would achieve good savings at a better payback and lower investment than the heat pump option.

Clearly, 'waste' and 'wasted' heat must be carefully defined. 'Wasted' heat is heat that is not recovered because of poor design of the heat recovery network. 'Waste' heat is heat that must be rejected from a process after the economic level of heat recovery has been achieved in a properly designed heat exchanger network. Pinch Technology eliminates any confusion about 'waste' heat by showing clearly which heat should be recovered and where it should be used for both passive heat recovery and heat pumping options.

Understanding The Total Process

Sometimes looking at projects in isolation from the process can give misleading results. The global perspective that Pinch Technology provides is invaluable when trying to understand the impact of design decisions on the total process. In addition to addressing the integration of heat pumps, the technology provides rules for appropriately integrating other unit operations, such as evaporation and distillation.

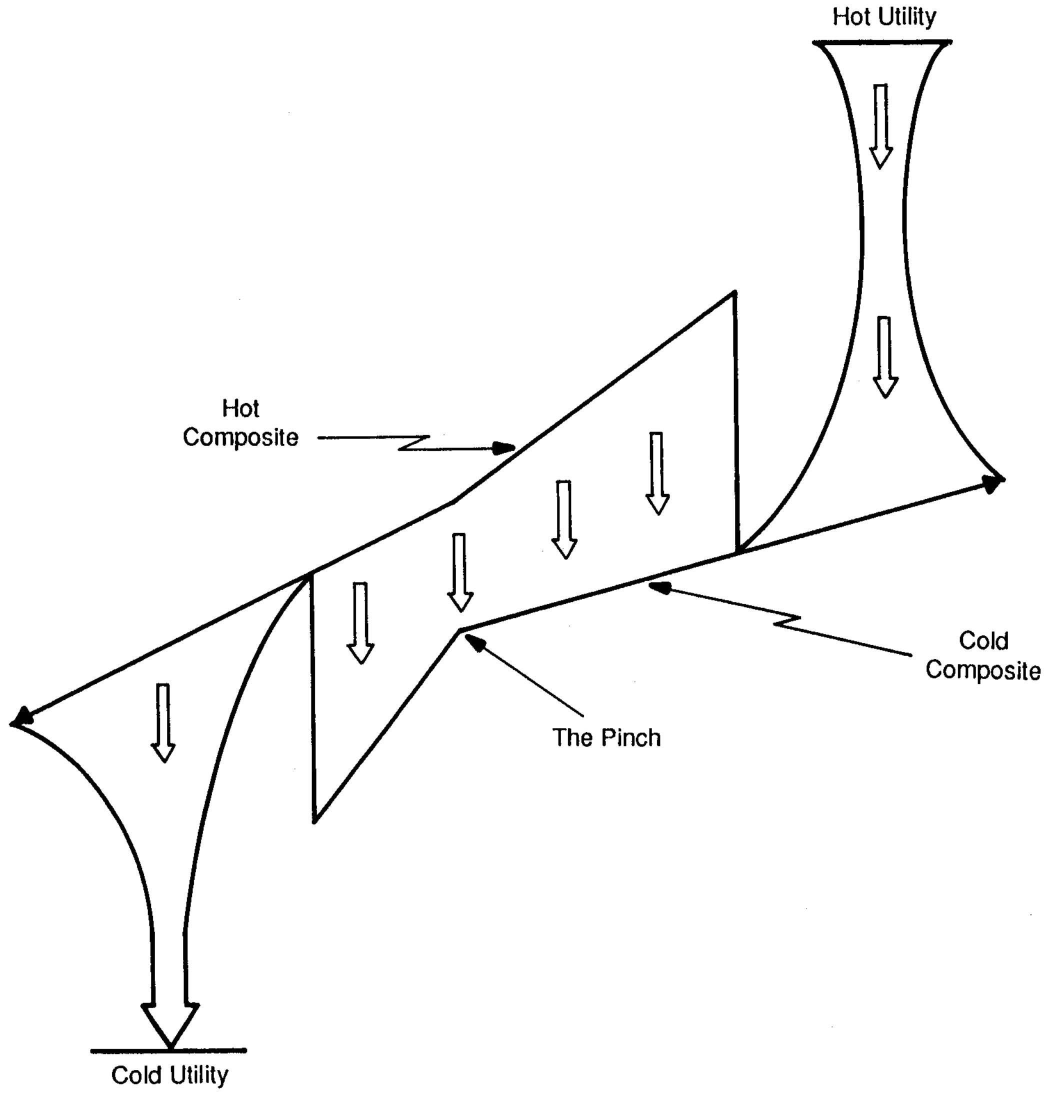


Figure 4-21a. Composite Curves Showing No Cross Pinch Heat Transfer

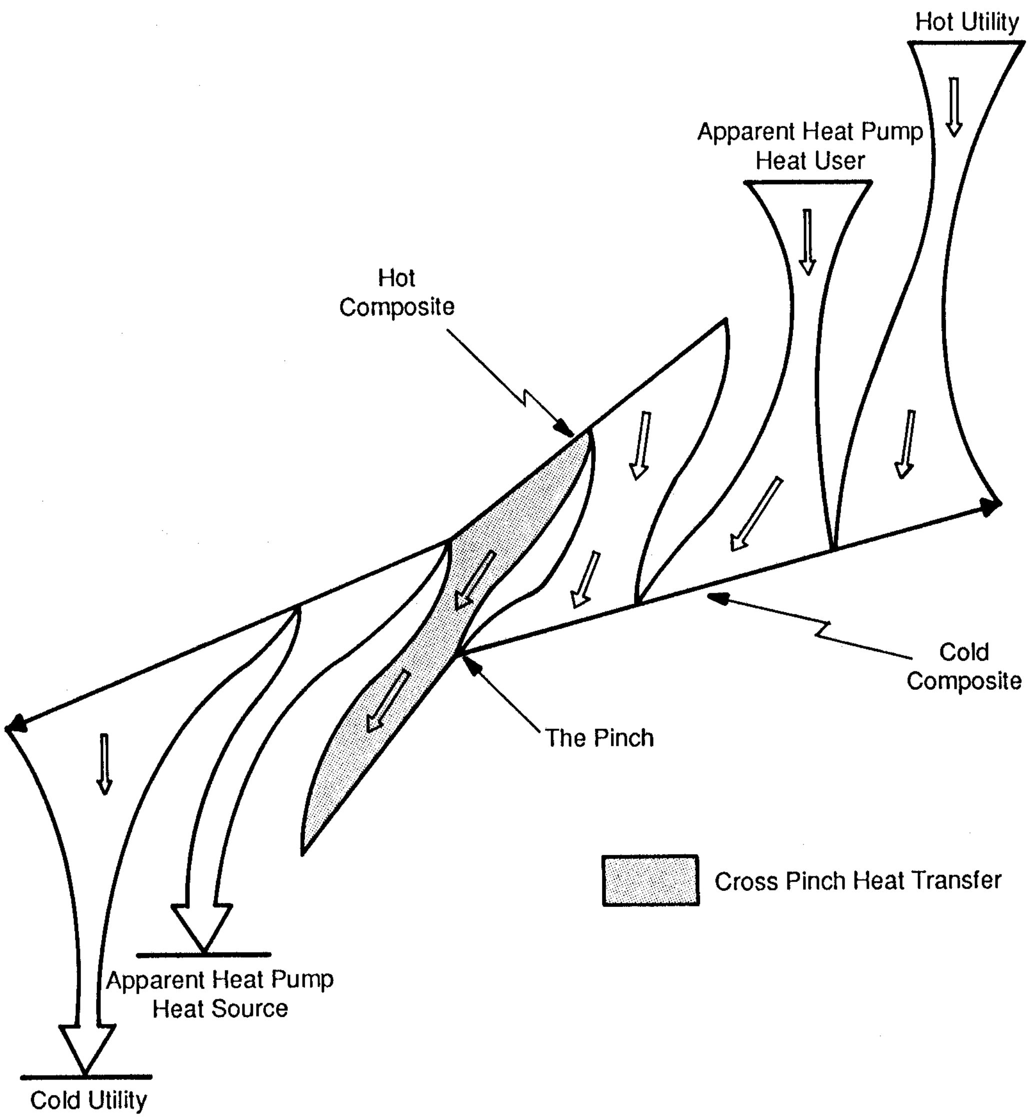


Figure 4-21b. Composite Curves Showing Cross Pinch Heat Transfer

THE BENEFITS OF PINCH TECHNOLOGY

Pinch Technology offers a structured approach to solving the energy cost reduction puzzle and has important implications for industrial heat pump applications.

The benefits of applying the technology fall in the areas of:

- Systematic generation of energy cost reduction alternatives including passive recovery, process modifications, cogeneration, and heat pumping.
- Understanding the overall process implications of a project rather than the local consequences.
- Finding good heat pump applications.
- Ensuring that the heat pump is appropriately integrated within the <u>entire</u> process.
- Correct selection of the temperature level and heat loads.

There is no other approach which gives as much insight into process energy utilization.

In summary, Pinch Technology is an invaluable tool for finding and assessing energy cost reduction schemes and guarantees that if a good heat pumping opportunity exists, it will always be found.

<u>A Final Note On Pinch Technology</u>

The main example presented was really quite simple. The malt kilning process has been evolved over many years and the kilns in operation now incorporate some or all of the energy saving options described in the preceding sections. Also, the heat pumps, where used, are indeed appropriately placed. How did Pinch Technology help in this case? It should be clear that all the heat recovery, utility, and heat pump options can be generated very quickly, allowing rapid feasibility assessment. Years of evolution are not required to arrive at the optimal solution. Also, with a relatively small problem the number of stream matching possibilities is quite small. With two hot and two cold streams there are only about 12 different ways in which all the streams can be matched. It would be possible for an experienced engineer to generate and assess those possibilities. However, in a problem with four hot and four cold streams the number of possible heat recovery networks increases to over 10^{13} and there is no way that this number of possible configurations can be generated and assessed. In these cases, using Pinch Technology is the only sensible approach to design the process energy network. Even with smaller problems, using Pinch Technology offers the best approach.

There are no hard and fast rules as to when Pinch Technology should be used and when it should not be used. However, as discussed, processes with a small number of process streams can be analyzed without using Pinch Technology more easily than those with a large number of streams. For example, a pinch analysis on the lumber drying process is unlikely to reveal any energy saving projects which have not already been discovered.

If Pinch Technology is not used explicitly to find cost reduction opportunities, then being aware of its guiding principles will certainly be of help whichever alternative approach is used.

Chapter 5

THE FEASIBILITY STAGE

CHAPTER SUMMARY

In the previous chapters of this Resource Guide the general background to industrial heat pumps was presented. This chapter begins the description of how to appraise specific heat pump opportunities.

The first part of heat pump appraisal is the Feasibility Stage. This consists of three steps; - Data Gathering, Generating Options, and Assessment of Options. Each step is described in this section. These basic steps must be considered part of an iterative procedure i.e. the steps are repeated in more detail as the best options are defined. The need for a structured analysis is stressed - if a project is randomly selected and appraised it is unlikely to prove the most cost effective opportunity.

Feasibility analysis is essentially a financial comparison of available options. Details are given of the way certain key parameters affect the economics of heat recovery systems.

HEAT PUMP FEASIBILITY AND DESIGN

Chapters 1 to 4 of this Resource Guide are intended to give a background to the field of industrial heat pumps and other energy cost reduction options. The remaining chapters are devoted to explaining the procedures required to assess a particular heat pump opportunity. The procedures must encompass the initial search for a possible application through to the commissioning and operation of a full scale industrial heat pump.

In this chapter the Feasibility Stage is discussed. The aim of this stage is to assess a factory or process to identify which package of heat recovery projects, including heat pumping, has the best financial return. The Feasibility Stage must investigate a wide range of options. The Design Stage described later in Chapter 6 describes in detail the steps that should be followed when designing and specifying a heat pump installation after its initial feasibility has been proven.

STEPS IN THE FEASIBILITY STAGE

The Feasibility Stage is essentially an economic comparison of potential energy saving projects. An interesting feature of energy recovery opportunities, as pointed out in Chapter 4, is that there are usually many different options. The existence of so many options makes a feasibility study more difficult to carry out properly, hence it is vital to use a structured approach.

The Feasibility Stage is probably the most important single step in the whole design procedure; if the wrong decision is made here then the wrong option will be carried forward and eventually built. The resulting project could well be a technical success but it might not represent the best investment. There are many examples of first generation heat pumps that were built in place of a simpler option. Clearly what must be avoided is the investment of $250,000 to save $100,000 when an alternative with a capital cost of, say, $150,000 could have saved the same amount.

The Feasibility Stage must include procedures to generate lots of potential projects as well as screening stages which reduce the number of options. This will avoid excess time and effort being spent on appraising projects which are poor from the beginning. More detailed appraisals should be restricted to the most promising projects. Because of this need for option elimination the feasibility stage must be considered an iterative procedure. That means that the basic steps in the analysis are repeated with increasing levels of detail. The first time around the minimum amount of work necessary to home in on the best set of options is done. Then the analysis is repeated in increasing detail in order to rank the better options.

The various steps in a rigorous feasibility analysis are outlined in Figure 5-1.

Step 1 is to gather relevant data about the host process. The data will be used as the basis for analyzing the process, and proposing and assessing the various energy saving opportunities. The data must include many technical details and also key commercial factors (e.g., what return on investment is acceptable).

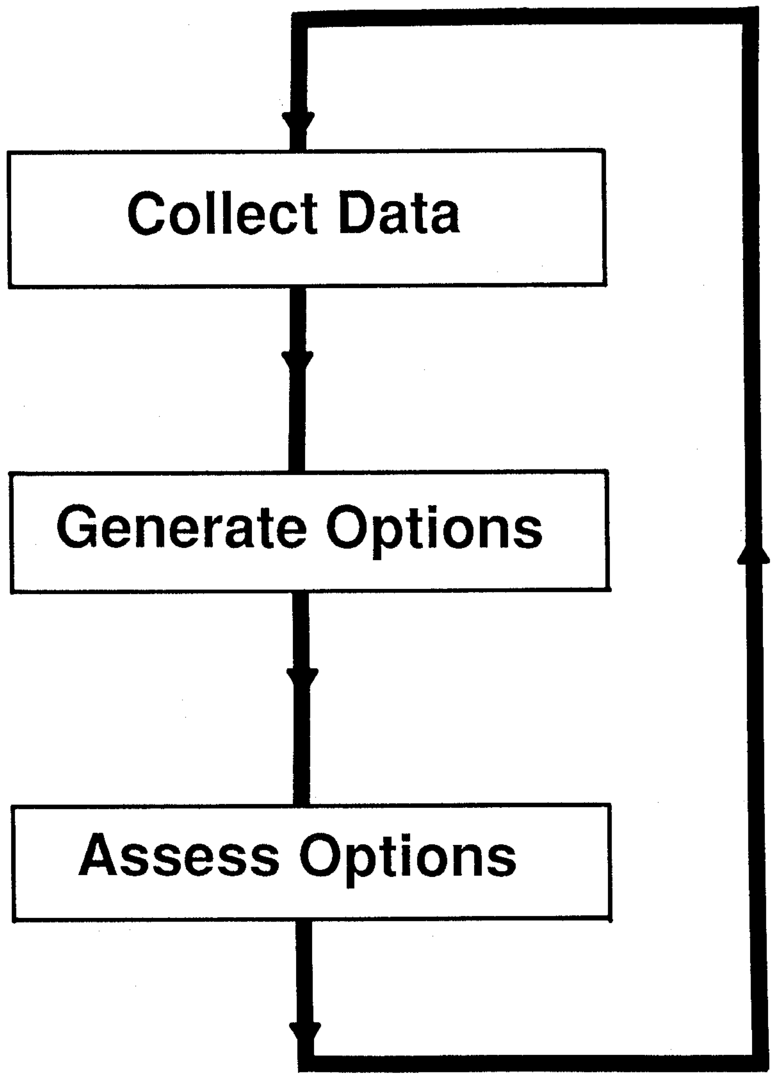


Figure 5-1. Feasibility Analysis

Step 2 is Generating Options. This is the stage when all relevant heat pump and other opportunities are identified. The techniques of Pinch Technology should be applied here.

Step 3 is Assessment of Options. The opportunities are investigated in sufficient detail to ensure costs and savings can be accurately calculated, and to ensure that a technically feasible scheme is finally proposed.

The second half of this section looks in detail at each of these steps. However, before doing that it is necessary to fully understand the ways in which heat pump economics in particular can be influenced.

PARAMETERS AFFECTING HEAT PUMP FEASIBILITY

As stated earlier, the Feasibility Stage is essentially a financial analysis of technically feasible options. The next sections examine how the most important parameters alter the payback period of a heat pump. Some of the parameters are only related to heat pump systems (e.g. COP, heating temperature range). Others are more general and better known (e.g. fuel costs, running hours).

Fuel Costs

The influence of both absolute and relative energy (fuel and power) prices on the economics of heat recovery projects was described in Chapter 4, and in Chapter 3 the sensitivity of heat pump operating costs to changes in fuel prices were described. In addition, the importance of attaching the correct marginal cost to the utilities being saved by heat recovery projects was discussed in Chapter 4. However, there are a number of energy price issues related particularly to heat pumps. These are discussed below.

The cost of electricity is very variable from one part of the country to another. Hence care must be taken in assuming that a successful heat pump in a plant in one state can be replicated in another. Similarly the converse must be true - don't assume that a design that proves negative in one situation will always be that way!

Using the correct fuel price data is not as easy as it would seem. A common error is to use average unit costs for electricity. However, many tariffs are calculated on a varying scale - extra electricity usage for a heat pump may be at a price lower than the average rate. It is important to use the marginal cost for the extra usage, not the average cost and any impact on demand charges should be investigated. Clearly the same applies to utilities being saved.

Heat Pump Capital Cost

The payback period of a heat pump will be directly proportional to capital cost. At the Feasibility Stage this figure is very difficult to estimate accurately and confidently. The capital cost must represent the fully installed price and include the cost of the heat pump itself plus all associated costs such as civil works, electrical and control wiring, instrumentation etc. Often the ex-factory heat pump package only represents about one third of the final price, so it is inevitable that the installed cost will vary from site to site.

A useful measure is the 'specific capital cost', which is the cost per unit of heating capacity (e.g. $/Btu/hr or $/kW). This allows easier comparison between heat pump systems of different size.

A good starting point in the analysis of capital cost is to look back at existing installations. Figure 5-2 shows values of specific capital cost for a variety of heat pump installations in the U.S. and in Europe. The main feature of the plot is the degree of scatter and the fact that economy of scale is not an obvious effect. This graph confirms the point stated above that it is not an easy matter to estimate capital cost.

At the Feasibility Stage there is one point in our favor. The analysis requires the comparison of many options. In general, as long as intelligent cost estimates are made, the final ranking order of the options will be correct and the best one will be identified, even though the absolute value of payback period may be accurate only to within ±10 to 20%.

Annual Running Hours

Another basic parameter is the annual usage of the heat pump. It is necessary to look at the annual availability of the source and the potential annual usage of the delivered heat and calculate how many hours/year these coincide and hence how many hours/year the heat pump can run. In many cases, estimates of usage have

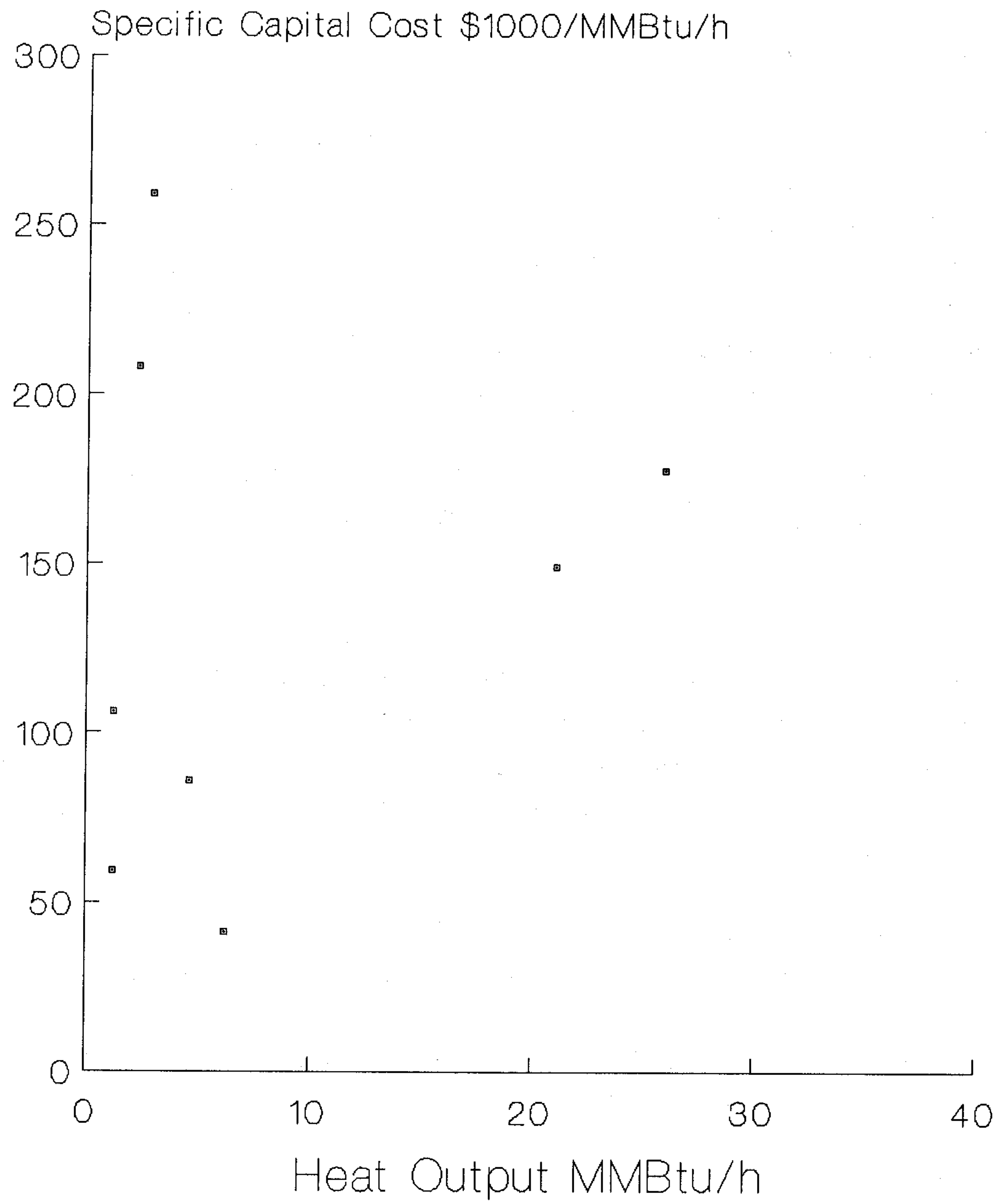


Figure 5-2. Heat Pump Capital Costs

tended to be rather optimistic, often by as much as 25%. If a heat pump has a design payback period of 2.5 years and operates for 25% less hours than planned the payback period will rise to 3.4 years; this is an unacceptable increase. Care must be taken to ensure the operating hours estimate is realistic and accounts for typical levels of unplanned shutdown as well as budgeted operation.

High annual running hours are an essential ingredient for a cost effective heat pump. Factories operating 7 days/week 24 hours/day (annual operating hours > 8,000) will have better opportunities than those operating a 5 day/week 3 shift pattern (6,000 hours/year). If usage is significantly less than 5,000 hours/year then a heat pump will usually require some extra non-energy benefit to achieve satisfactory payback periods. Hence single shift processes are usually not worth considering. Low running hours is one of the reasons why space heating heat pumps are not cost effective in countries where summer time air conditioning is not required.

Load Factor and the Sizing Issue

It is not only the total operating hours that will influence the amount of fuel savings from a particular heat pump. The heat pump load factor is equally relevant. If a heat pump runs 8,000 hours/year but only operates at a load of 25% of its full rated output the payback period will be very poor. Ideally a heat pump should run at 100% load all the time it is running. If this does not occur there is a financial penalty for 2 reasons,

- If a heat pump that supplies 250,000 Btu/hr only operates at an average of 175,000 Btu/hr then the capital investment is not being well used. Although the actual capital cost was, say, $100/kBtu/hr the EFFECTIVE capital cost is actually $143/kBtu/hr because of the low load factor.
- When a heat pump operates at part load, its efficiency usually falls. This is because the compressor does not operate well at part load and because various auxiliaries such as pumps or fans usually operate at full load all the time.

These two effects make the part load performance of a heat pump very poor in overall economic terms. If low load only occurs occasionally, say 1 hour/day during a cleaning cycle, then the situation is fine. If, however, the heat pump operates with a consistently low load factor the economics will be poor.

The importance of ensuring a high load factor cannot be stressed enough. The most common fault of first generation heat pumps was oversizing. The problem usually occurs because "fossil fuel design rules" are applied. If, for example, a boiler plant is being built to heat a process it will always be made larger than necessary, so it has contingency to supply all foreseeable peaks. This design philosophy is wrong for heat pumps because they are a more capital intensive heat source and hence should be operated at high load factors. Generally, there is no need for the heat pump to supply all the heat requirement; existing boiler plant can supplement the heat pump and follow the peaks and troughs in the process.

The correct design philosophy for heat pumps is "small is beautiful". Select a heat pump size that fits below the troughs in heat demand so it runs at full load for most of the time. Part load operation down to 75% is forgivable in a heat pump operating 8,000 hours/year but significant usage below this level will lead to poor economics. This important issue is discussed again in Chapter 6 in the Design Stage, Step Two.

Heat Pump Performance

The four parameters discussed above are all of a general nature and require little specialist understanding of the heat pump itself. It is also vital to have a good knowledge of the factors that affect the running cost of the heat pump system so that this can be compared with that of the existing utility system. The following six sections examine the various ways in which heat pump running cost can be altered.

Before trying to establish the way heat pump performance varies it is helpful to define some measure of performance. In Chapter 2 of this Resource Guide the concept of Coefficient of Performance, COP, was introduced and defined. The COP is the ratio of heat output to the compressor shaft power input; higher values of COP imply better heat pump performance. However, the COP is not the best measure of performance for use in the Feasibility Stage. There are several reasons for this

- The value of COP makes no allowance for the fact that the heat pump fuel and the existing fossil fuel usually have different prices.
- The COP does not include the cost of running auxiliaries such as pumps and fans.
- The concept is relatively unfamiliar to most non-specialists.

It is far better to establish the cost of each unit of heat supplied by the whole heat pump system. This can be simply expressed in \$/MMBtu or ¢/kWh and then a very easy comparison between different heat pumps and the existing utility system is possible. In order to calculate this figure it is necessary to know the instantaneous heat output, the power absorbed by the heat pump and any auxiliaries, and the price of each fuel involved. In that way quite different types of system can be compared as shown in Table 5-1. From the Table, which compares two electric heat pumps and a gas engine driven system it is clear that COP is an unsatisfactory method of comparison. Heat Pump 2 has the best COP but this is achieved with a very high auxiliary power so the overall running cost is highest. Heat pumps 1 and 3 both have the same COP but the fuels used are different and the gas engine heat pump has the opportunity for exhaust gas recovery. Remember that the ranking of these two plants will change if the fuel price ratio alters. All the heat pumps have a running cost just over 50% of that of a conventional fossil fired plant.

Temperature Lift. The temperature lift of a heat pump is the temperature difference between the heat source and the heat user. For example, consider a heat pump which heats a fluid from 150°F to 200°F; the heat source is a process stream at 100°F that is cooled to 75°F in the heat pump evaporator. In this case the temperature lift is 200-75 = 125°F. Note we use the higher figure for the heat user and the lower figure for the source as these values will define the heat pump evaporating and condensing conditions.

It is obvious from the "heat pulley" analogy discussed in Chapter 1 (see Figure 1-1) that heat pump performance improves as the temperature lift falls. The temperature lift is always a function of the host process streams. A key part of the Feasibility Stage is to find the best combination of potential heat sources and heat users. As a general rule it is best to minimize the temperature lift between streams in order to minimize the cost of heat from the heat pump.

Although the temperature lift is a host process characteristic it is sometimes possible for the heat pump designer to alter the process conditions in favor of the heat pump. In some circumstances it may be possible to alter the heat source conditions so that the heat is available at a higher temperature. If this leads to a waste heat source at 130°F being cooled to 105°F (instead of 100°F to 75°F) then the temperature lift falls from 125°F to 90°F and the heat pump performance will improve.

Table 5-1

COMPARISON OF HEAT PUMP PERFORMANCE

	Heat Pump 1 Electric	Heat Pump 2 Electric	Heat Pump 3 Gas Engine
Heat Pump Output KW	100	100	100
Compressor power input kW	25	20	25
Heat Pump COP	4	5	4
Auxiliary Electric input kW	5	15	10
Engine Waste heat kW recoverable (80% total)	0	0	60
Cost of Electricity ¢/kWh	5	5	5
Cost of Gas ¢/kWh	2	2	2
HEAT PUMP RUNNING COST ¢/kWh	1.5	1.75	1.6

Note, cost of heat from conventional gas fired boiler with 75% efficiency = $\frac{2}{0.75}$ = 2.7 ¢/kWh

Gas engine assumed to have 25% thermal efficiency

Evaporator and Condenser Sizing. The actual temperature difference over which the heat pump operates is the difference between the refrigerant condensing temperature and evaporating temperature (for closed cycle systems). This is greater than the temperature lift because it must include the heat exchangers driving forces. In the first example above, the lift was 125°F. However the refrigerant would have to condense at, say, 220°F to heat a fluid from 150°F to 200°F. Similarly the evaporating temperature may be 55°F. Hence the OVERALL TEMPERATURE LIFT is actually 220-55 = 165°F. By altering the design of the evaporator and condenser it is possible to change the overall lift. If larger heat exchangers are used a closer approach temperature of, say, 10°F is possible. If both the evaporator and condenser are altered the overall lift in this example falls from 165°F to 145°F.

The heat exchangers will need careful optimization at the Design Stage. In the Feasibility Stage it is sufficient to use figures based on previous experience to make comparisons between different designs. The approach temperature to be used depends on two main factors,

a) The type of fluid being heated or cooled (liquid, gas or two phase?, clean or dirty?).

b) The temperature lift.

Generally speaking larger approach temperatures are required for gases than for liquids; two phase mixtures (e.g. condensing vapor heat source) have lower approach temperatures than liquids. These rules must be adjusted if the fouling factor is high.

The temperature lift is also of importance in heat exchanger optimization. If the lift is large, say above 100°F then optimization of the approach temperature is less important than a heat pump based on streams with a lower lift. Table 5-2 compares a heat pump with a 50°F lift with one of 125°F. In the case of heat pump 1 the COP rises by 25% if the temperature approach in each heat exchanger is reduced from 20°F to 10°F. For a similar change in temperature approach, the COP of heat pump 2 only improves by 8%.

Table 5-3 shows some typical approach temperatures that can be used in the Feasibility Stage of a heat pump analysis.

Table 5-2

OPTIMIZATION OF HEAT EXCHANGER SIZES

		Heat Pump 1		Heat Pump 2	
Temperature Lift	°F	50		125	
Heat Exchanger Approach Temperature	°F	10	20	10	20
Overall Temperature Lift	°F	70	90	145	165
Heat Pump COP		5.0	4.0	2.7	2.5

Table 5-3

HEAT EXCHANGER APPROACH TEMPERATURES

	Temperature Lift, °F		
Fluid Type	> 100	50-100	< 50
Clean Liquid	20	15	10
Dirty Liquid	25	20	13
Clean Gas	30	25	20
Dirty Gas	40	32	25
Clean Condensing Vapor/Boiling Liquid	15	10	10
Dirty Condensing Vapor/ Boiling Liquid	20	13	13

Note: Values given for "dirty" fluids are typical figures for moderate fouling.

Heating Temperature Range. Heating temperature range (HTR) is a very simple and effective design concept that is of enormous value when analyzing heat pumps and other heat recovery options. It is interesting to note that HTR has little or no value to the designer of a fossil fueled heating system. The concept of HTR is an excellent example of the need for a different design philosophy for conventional plants and for heat recovery.

The definition of HTR is very simple; it is the temperature difference through which a heat user is raised. For example heating water from 50°F to 150°F involves an HTR of 100°F. HTR can be used as a characteristic of a heat user stream. As explained below a stream with a large HTR has a better potential for heat recovery than a stream with a small HTR.

The concept is most easily illustrated with an example; relevant data is shown in Table 5-4. A comparison is made between two heat users. Both are streams of water being heated to 150°F. In both cases the heating duty is 100 kBtu/hr. In the first case the flow rate is low but the HTR is high; the converse is true for the second heat user.

So, what is the difference? If you apply "fossil fuel rules of thumb" there is no difference! If a boiler supplies heat to these streams the fuel consumption will be the same. But in the case of a heat recovery system the difference can be enormous. Imagine a waste water heat source is available. The flow rate of this waste stream is 1500 lb/hr and it is available at 175°F. With this heat source it is possible to supply all the required energy to Heat User 1 simply by passive heat recovery. In the case of Heat User 2 it is only possible to extract energy from the source between 175°F and 145°F (allowing 15°F minimum driving force in the heat exchanger). This is only equivalent to 45% of the heat requirement of the heat user. Figure 5-3 illustrates the two heat recovery opportunities. In this example it is clear that the potential value of a waste user stream is dependant on the HTR of the heat user. When the HTR was large the heat recovery potential was far greater than when the HTR was small.

It is worth noting that the concept of HTR is a simplified special case of some of the rules that emerge from Pinch Technology.

Table 5-4

THE USE OF HEATING TEMPERATURE RANGE

	Heat User 1	Heat User 2
Inlet Temperature °F	50	130
Outlet Temperature °F	150	150
Mass Flow lb/h	1,000	5,000
Heating Duty kBtu/h	100	100
HTR	100	20

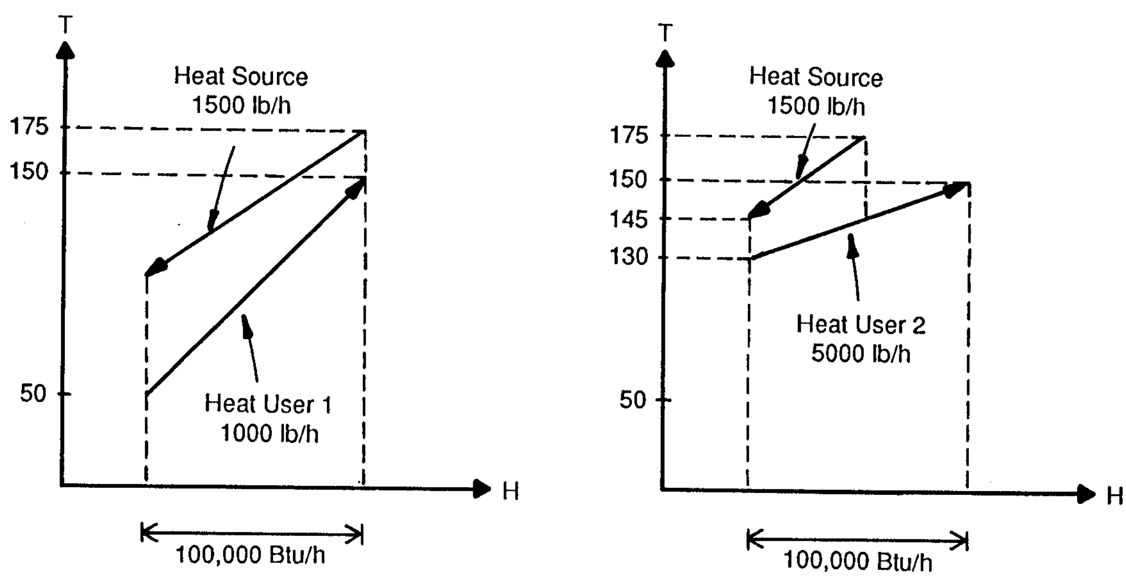


Figure 5-3. Heat Recovery Stream Diagrams for Table 5-4

Choice of Basic Cycle. In Chapter 2 of this Resource Guide some of the design variants for different heat pump types were examined. During the feasibility stage it is necessary to decide which cycles should be considered. The concept of HTR is a useful tool for this purpose. There is a corollary to HTR which is the temperature range of the heat source. This can be called CTR or cooling temperature range. The CTR is also an important selection tool for heat pump cycles.

Table 5-5 shows a matrix of different combinations of HTR and CTR. This table can be used to identify the best cycles. It must be noted, however, that the table is only a rough guide. The choice of cycle not only depends on HTR and CTR. The overall temperature lift and the heat pump capacity also have a great influence on this choice.

If both HTR and CTR are low then the simple 4 component cycle is the only choice unless the heat source is a vapor. In that case open cycle will be preferable. A refrigerant subcooler should be added as soon as the HTR is greater than 20°F and if it reaches 60°F a desuperheater can be added as well. If HTR remains low but CTR rises then a cascade of 2 heat pumps should be considered. If both HTR and CTR are high then a combination of cascaded heat pump with subcoolers is possibly the best option. On large systems it is worth considering a cascade of at least 3 heat pumps.

It is also interesting to note that engine driven systems can be most effective when the HTR is large. This is because the engine waste heat is used to do the final heating of the heat user. With a large HTR this means the heat pump itself will operate over a much lower temperature lift -- with a significant increase in COP.

Heat Pump Auxiliaries. The performance of a good heat pump can sometimes be spoiled by excessive auxiliary power. As discussed above, the cost of auxiliary power must be included in the calculation of heat pump operating cost. The major auxiliaries are pumps or fans to pass the heat source and heat user fluids through the heat pump evaporator and condenser. Other possibilities include oil pumps on screw compressors, crank case oil heaters and standby or supplementary heaters.

It is important not to penalize the proposed design with auxiliary loads that are required even without the heat pump. For example a heat pump may extract heat

Table 5-5

CYCLE SELECTION MATRIX

Cooling Temperature Range	Heating Temperature Range		
	Low < 20°F	Medium 20°F-60°F	High >60°F
Low < 20°F	Use open cycle if possible or 4 component closed cycle	Add a subcooler	Use subcooler and desuper-heater Consider engine drive
Medium 20°F-50°F	4 component closed cycle	Add a subcooler	Cascade of 2 with sub-coolers Consider engine drive
High > 60°F	Cascade of 2	Cascade of 2 with subcooler	Cascade of 3 plus sub-coolers Consider engine drive

from a process water stream. If the existing water pump is capable of passing the water through the heat pump evaporator then there is no extra auxiliary power and the pump need not be considered as a heat pump auxiliary. In fact in some cases there will be a saving in auxiliary power. For instance, the heat pump evaporator may reduce a cooling tower duty. In this case the cooling tower fans are no longer needed and the heat pump should be credited with an additional saving. Note that by measuring heat pump system performance in ¢/kWh these adjustments to the running cost are easily made.

As a general rule auxiliary power should not exceed about 15% of the full load heat pump compressor power. If it is higher than this then the pipework for source and user streams may be badly designed or undersized. The amount of pumping power can influence the optimization of heat exchanger sizes as discussed above. To achieve a smaller approach temperature, and hence improve the heat pump COP, a larger heat exchanger is required. This can imply extra auxiliary power and although the COP is improved the overall performance is not always better.

Another relevant point is that auxiliaries do not usually have any capacity control. If the heat pump itself has capacity control and is running at, say, 50% load then the auxiliary power becomes twice as important as it was at full load. This is a strong influence on part load performance and heat pumps running consistently at part load are unlikely to have good economics.

Effect of Constraints. In the Feasibility Stage it is necessary to make many judgments in order to make use of the various factors described above and hence to estimate the economic performance of a heat pump. At all stages one must consider site specific constraints (and in some cases, benefits) and modify the general data accordingly.

Heat pump location often has a strong influence on capital cost. If the source and user streams are conveniently close together and also close to a site to place the heat pump then capital costs will be low. However, in many cases the converse is true and it may even be necessary to include the cost of a building to locate the heat pump.

Corrosive or dirty source and user streams also represent extra costs. Heat exchangers may require expensive materials of construction or automatic cleaning facilities. The heat pump performance may also be affected as the heat exchanger approach temperatures will be greater than normal.

Excessive variations in heat load or poor matching of source and user loads are to be avoided. Similarly if the source and user are in different processes that operate at different times of day opportunities are limited. Plant start up must be considered. In many cases this is no problem, but occasionally extra plant is required to ensure proper operability.

Estimating Heat Pump Compressor Power Requirements. The most accurate method for determining compressor power required is to do a detailed thermodynamic calculation using the properties of the proposed refrigerant. However, for estimating purposes in the feasibility stage, a short cut method can be used. This method is based on the fact that the COP of 'real' heat pumps only reaches a certain fraction of the COP of an ideal heat pump. The COP of an ideal heat pump cycle (Carnot cycle) is easily calculated from the refrigerant evaporating and condensing temperatures. Referring back to Figure 2-1, the ideal heat pump COP would be:

$$\text{Carnot COP} = Q_{out}/W_{ideal} = \frac{T_2}{T_2-T_1}$$

Where T_2 is the condensing temperature of the refrigerant and T_1 is the evaporation temperature of the refrigerant (in absolute temperature, °Rankine). For example, a heat pump being used to heat water from 100 to 180°F with a heat source at 85°F cooling to 55°F (as shown in Figure 2-7) will have an evaporation temperature of 45°F and a condensing temperature of 190°F (assuming a temperature approach of 10°F in the evaporator and condenser). Thus the Carnot COP is given by:

$$\text{Carnot COP} = \frac{650}{650-505} = 4.5$$

In practice, typical actual COP's fall in the range of 0.65-0.75 of Carnot COP. (This variation is caused by a number of factors including compressor efficiency and refrigerant properties.) On this basis, the actual COP of the example heat pump would be approximately 3.0.

Having calculated the COP, it is easy to calculate the work (power) requirement from knowledge of the evaporator duty or condenser duty:

$$Q_{out} = Q_{in} + W = COP \times W$$

$$\therefore W = \frac{Q_{in}}{COP+1} \quad \text{or} \quad W = \frac{Q_{out}}{COP}$$

Thus, if the evaporator duty in the example is 1 MMBtu/h, the work required is 0.5 MMBtu/h (147kW) and 1.5 MMBtu/h are delivered in the condenser.

It should be stressed that this is only an approximate method and final evaluation of the operating cost of the heat pump should be based on detailed thermodynamic calculations. However, in the early stages of evaluation, this shortcut method will give an approximation of performance which will assist in screening the heat pump options.

THE FEASIBILITY STUDY

It is worth re-emphasizing that a heat pump feasibility study must be treated as a heat recovery feasibility study and address all the possible alternatives. Just considering whether a heat pump will be cost effective and technically feasible for a given application is not sufficient; it may provide a good solution and good savings, but other options such as passive heat recovery, process modifications, cogeneration or any combination of these may provide better technical and economic solutions.

This section describes the three steps necessary in a feasibility analysis, Data Collection, Generation of Options, and Assessment of Options. However, there is one important preliminary step, the '5 Minute Screening'.

The 5 Minute Screening

The structure of the feasibility study presented in this guide is based on an iterative approach i.e. you proceed through the 3 basic steps more than once. The first pass through the analysis should be considered as a quick initial screening.

This has the aim of quickly deciding, without committing much time and money, whether it is worthwhile looking for energy cost reduction opportunities. It addresses overall questions such as:

- Is energy cost really a problem?
- Is it likely that energy recovery opportunities exist?
- Is the energy bill large enough to justify spending time on studying?
- Is the interest in energy saving aimed at relieving capacity problems?

Asking these questions, and others of this type, up front will help to put the correct perspective on the objectives of completing a study.

Step One, Data Collection

Before any heat recovery projects can be identified, some data is required! This data relates to the process and the economic parameters at the site being studied.

It is interesting to note that Step 1 of the Design Stage, described in the next chapter of this Resource Guide, is also data collection. This has two implications. Firstly, it stresses how important data collection is in both the feasibility and design stages. Secondly, it implies that different data is required for the feasibility stage and design stages. At the feasibility stage, wide ranging data about the process and site are required. However, in the design stage only one option is being addressed and much more detailed and reliable data specific to that project should be used.

The necessary data falls into two main categories, data to help identify the recovery opportunities and data to compare these opportunities with the existing process performance.

The basic requirement for the technical assessment of heat recovery opportunities is a consistent heat and mass balance for the process. This is a quantification of the flow rate, temperatures and heat contents of all process streams. This can be done by first making a list of all streams and then establishing the data to attribute to each stream. During the data gathering for the "5 minute screening" it is quite sufficient to use approximate data for average operating conditions.

When the feasibility analysis has reached a more detailed level then it is necessary to verify the data and to look at the way it varies under different operating conditions.

In addition to the basic heat and mass balance data discussed above, further technical information is required. This includes details of the physical location of the streams, any variation in heat loads or temperature levels, the presence of fouling or blocking or corrosive constituents and potential hazards, e.g., chemical reactions that may occur if two streams become cross contaminated. The basic process data requirements are summarized in Table 5-6.

The information discussed above is sufficient for the technical evaluation of the relevant options. In order to carry out a financial analysis of these options and to compare them with the existing plant other more general data is needed.

It is important to know what return on investment, (payback period etc.) is acceptable to a particular company. Does this vary with the level of investment? Is investment in one process more favorable than in another, due to plans for expansion and so? How much does fuel and electricity cost? Will this change and what are the marginal costs?, etc.

Data on any extra benefits should also be gathered. An example is where the use of a heat pump releases boiler capacity eliminating the need, perhaps, to purchase additional boiler plant for other heating objectives. An alternative spin-off is the reduction in cooling water required for the heat source, with a corresponding reduction in treatment costs and the release of cooling capacity again perhaps, eliminating the need for new plant.

<u>Step Two, Generating The Options</u>

As soon as relevant host process data has been gathered it is necessary to generate options on at least two levels. The first level options relate to 'What cost reduction possibilities exist?' The second level options are related to 'How can the cost reduction be achieved?' For example, it may be determined that it is possible to heat process stream A with process stream B. This is a first level option, it has been established that the two streams can exchange heat. The suboptions related to this project are, should a shell and tube exchanger or a

Table 5-6

BASIC PROCESS DATA REQUIREMENTS

TYPE OF DATA	ITEM
Thermodynamic Properties	Supply Temperature Target Temperatures Heat Capacity Latent Heat (if applicable)
Physical Properties	Flowrate Phase Composition Pressure Heat Transfer Coefficient Fouling Tendency Corrosiveness

plate and frame exchanger be used? or, is an intermediate heat transfer fluid required, and so on.

The types of projects which fall under the first and second level categories are shown in Table 5-7. It could also be argued that third level options also exist i.e., if it is decided to match stream A and stream B in direct heat exchange, the further options are what type of exchanger, what material of construction, etc. These are really more detailed engineering and cost considerations which become more important at the design stage.

To make the option generating process easier, it is essential that a structured approach be used.

For generating first level options, the Pinch Technology approach described in Chapter 4 will quickly lead to identification of a thermodynamically consistent set of projects and a clear picture of the energy interactions in the process. However, it may be possible to utilize an alternative approach if it provides a structured comprehensive approach to identifying <u>all</u> the possibilities. Just looking for passive heat recovery opportunities or just looking for heat pumping opportunities, etc. may cause better projects to be overlooked.

In problems with a large number of streams, a comprehensive search for opportunities will be very difficult to accomplish without using Pinch Technology. In smaller problems a comprehensive search may identify good projects however, it would be very easy to identify inappropriate heat pumping or process modification opportunities as shown in Chapter 4. It should be noted that the same data about the process and site is required whichever approach is taken. Using Pinch Technology represents the most cost effective way of using that data.

To help generate second level options, the Option Tree shown in Figure 5-4 can be used. This shows the types of projects which may be identified. The Option Tree should be considered an 'aide memoir'. It does not generate specific options, but it reminds us of the general categories available.

The key to the best sub-options for passive heat recovery is the choice of heat exchanger type. The type of streams (whether gas or liquid, dirty or clean etc.) will define which type of heat exchanger is best suited.

Table 5-7

EXAMPLES OF FIRST AND SECOND LEVEL OPTIONS

Type of Project	1st Level	2nd Level
Passive heat recovery	Match stream A with stream B	direct heat exchanger indirect heat exchanger
Install a heat pump	Use stream A as the heat source and stream B as the heat sink	open cycle closed cycle electric drive
Utility supply	Use steam as the hot utility	low pressure boiler high pressure boiler plus steam turbine combined cycle
Process Modification	Reduce evaporation load in a drier	accept wetter product reduce moisture content of material entering drier by improved mechanical dewatering

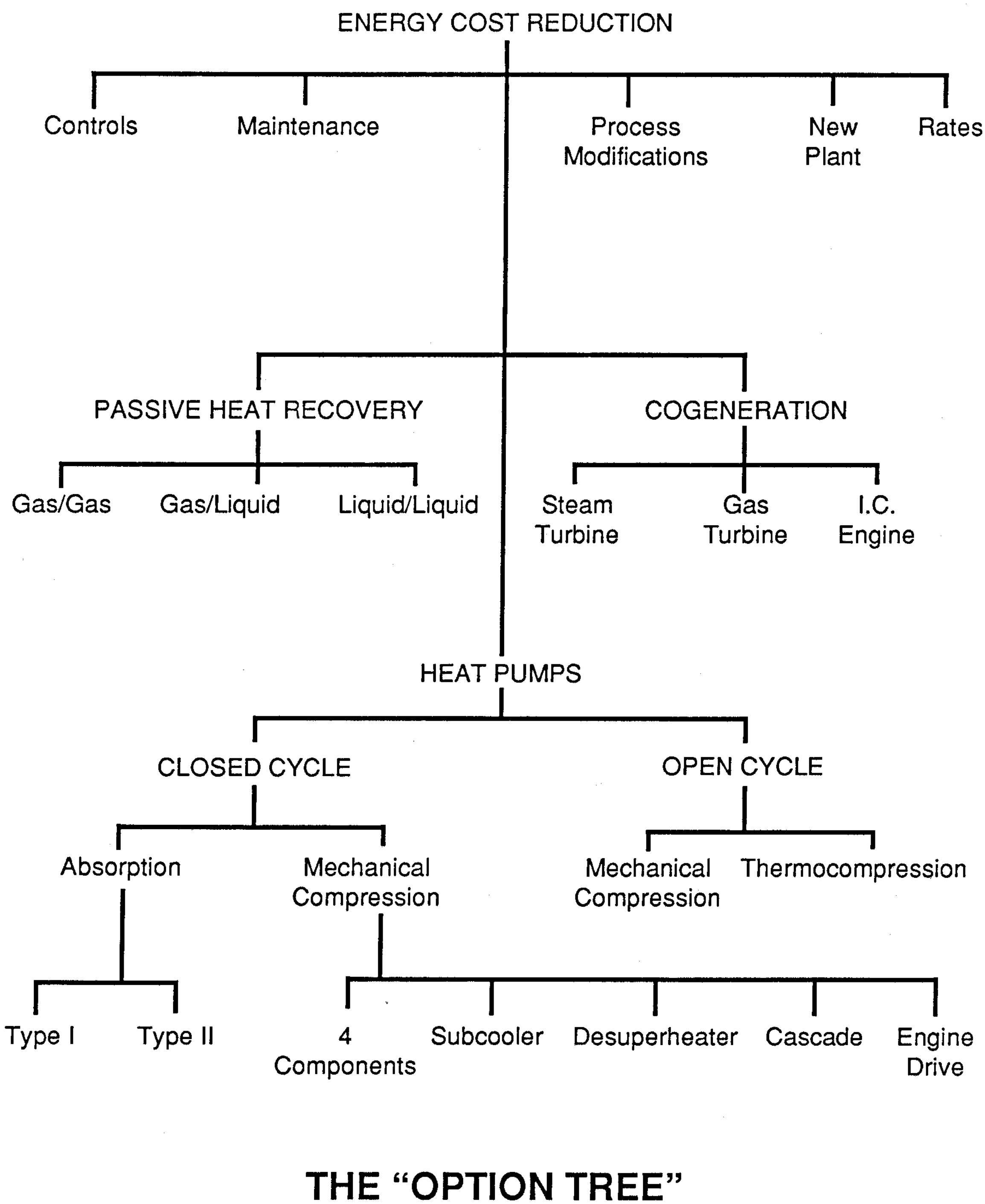


Figure 5-4. The Option Tree

In the case of cogeneration options the site heat to power ratio and the temperature at which heat is required are two of the basic criterion to be used. Steam turbines operate at heat to power ratios of about 10 while internal combustion engines operate at the other extreme with heat to power ratios of 1 and gas turbines fit in between at 3 to 4.

Following the heat pump branch of the tree, the first decision regards open or closed cycle. In general open cycle heat pumps will be more cost effective as they have both higher COP's and lower capital cost than closed cycles. Hence open cycle options should be generated if at all possible.

In the case of open cycle options, the mechanical compression route is the one usually followed. Thermocompression is only valid if the heat user is much larger in capacity than the heat source. The suboptions for MVR systems usually relate to temperature lift optimization. For example, if an evaporator is being retrofitted with MVR there are often two basic possibilities (i) use the existing heat exchanger and run with a high temperature lift (and low COP) or (ii) install a new, larger heat exchanger and obtain a significantly better COP. The economics of both options must be appraised in the feasibility stage. If the second option is proved best, the "fine tuning" of heat exchanger size versus COP can be left to the Design Stage (see Chapter 6).

The choice of closed cycle options follows a similar process. The selection matrix discussed previously (Table 5-5) helps choose between the various closed cycle sub-options such as subcoolers and cascaded cycles. The ratio of gas to electricity prices will influence whether or not engine driven options should be proposed. Absorption cycles are worth considering although it must be noted that experience of this technology in the United States is limited.

Step Three, Assessment of Options

Having generated a list of options each one must be assessed and three basic questions must be answered:

- is the option technically feasible?
- what is the annual cost saving?
- what is the capital cost?

Usually the capital cost is the most difficult of these questions to answer with confidence. Leaving that question till last is a good idea -- you may reject the option before the capital cost needs assessment!

Assessment of technical feasibility involves a search for practical factors that will prevent an option being possible. For example, a heat pump option involving a top temperature of 400°F may be found. This option can be dismissed immediately because 400°F is above the top operating temperature of existing cycles. Some practical factors may make an option difficult but not impossible. For example a heat source and user located far apart will involve extra cost. It is a matter of judgement whether this should be dismissed early in the analysis as technically infeasible or whether it should be carried forward with an appropriate adjustment to the capital cost.

The annual cost saving assessment first involves establishing a base case running cost for the existing system. Then the operating cost of the proposed option must be carefully assessed. Note we are calculating cost savings, not energy savings. By converting all energy savings into costs at an early stage it is easy to include tariff changes, maintenance costs and any other saving or extra cost.

The running cost of a heat pump usually consists of three elements; the prime energy, auxiliaries, and maintenance. In the case of mechanical vapor compression heat pumps, the compressor power is calculated from the heating duty and the COP. During a detailed feasibility assessment the COP should be properly calculated from thermodynamic and manufacturers data. However, during an initial screening it is useful to have some approximate data available. Table 5-8 gives some approximate COP's for cycles with different temperature lifts. Note that the table is based on overall temperature lift, i.e., including the temperature differences in the evaporator and condenser. These values should only be used as a rough guide to COP. The actual COP depends not only on overall temperature lift but also on the top temperature, the compressor efficiency, the refrigerant subcooler and so on.

For thermally driven systems, the cost of compressor power is replaced by the cost of the primary heat energy.

When the annual savings of the various options have been calculated it is often possible to reject a number of them either because savings are negative or because in comparison to other options the savings are too low.

Table 5-8

APPROXIMATE COP DATA

Overall Temperature Lift	COP	
	Small HTR (4 component heat pump or open cycle)	Large HTR (using subcooler or cascade)
20°F	20	20
40°F	10	11
60°F	7	8
100°F	4.5	5.5
140°F	3.5	4.7
180°F	2.5	3.6

Finally the capital cost must be estimated. Some guidance on this topic was given earlier in this section. Realistic estimates must be made and full allowance included for all installation costs. In general feasibility study estimates tend to be low because of too much optimism! When the capital cost is known an analysis of the payback period and return on investment of each option can be made. It can be dangerous to only use the simple payback period for comparison as this can give a distorted picture. For example a passive heat recovery option may save \$100,000/year with a payback of 2 years while a heat pump saves \$200,000 with a payback of 3 years. In the long term the heat pump may be the better option. For investments of this scale a proper discounted cash flow analysis is the only way of giving a true comparison.

In addition to establishing the technical and economic feasibility of each project, their status as either independent, domino, or exclusive (as defined in Chapter 4) should be determined. This will enable 'packages' of compatible projects to be developed. Care should be taken with domino projects because of the 'downstream' effect they can have on other projects.

At this point, some projects may have been eliminated and at least one package of projects will have been developed. Each of the promising projects should now be reassessed in more detail. For instance, instead of basing savings on the average flowrate of the streams being matched in a heat exchanger, the instantaneous flowrates should be investigated to make sure the savings can really be achieved. In general, the purpose of iterating in the assessment stage is to define the savings, technical feasibility, and estimated capital cost to the point at which there is sufficient confidence to proceed with detailed design.

In the initial stages, the level of detail does not have to be great. This enables a reasonably rapid and crude screening of the generated options. The use of screening criteria such as payback, capital cost, safety, etc. will reduce the number of options at each iteration until the necessary degree of confidence in the projects is reached.

Chapter 6

THE DESIGN STAGE

CHAPTER SUMMARY

This Chapter outlines the three step Design Stage that is required before going out to tender. The importance of detailed design is highlighted so that the heat pump purchaser understands why this stage is needed.

Step 1 of the Design Stage is data collection. Detailed information about the host process is required as the design basis for the heat pump. Relevant data include temperatures and flows, fluid properties, time related data and site information.

Step 2 concerns engineering details. A range of topics concerning system design, controls and component design are examined.

Step 3 of the Design Stage is writing a technical specification. The requirements of an industrial heat pump specification are explained.

THE NEED FOR DETAILED DESIGN

The feasibility stage has been completed and a potentially good heat pump opportunity has been identified. Now it is very tempting to rush off to a manufacturer and ask for a quotation.

However, this is precisely the wrong thing to do at this time and is probably the reason why so many 1st Generation heat pumps turned out to be unreliable and did not meet the purchaser's expectation in terms of performance. The purchaser is not yet ready to get a quotation because much vital information is missing. First it is necessary to examine the potential design in more detail.

The Design Stage really consists of three main steps. Firstly the basic design data from the host process must be confirmed and checked in more detail. The heat pump size should be reassessed and then various points of detailed engineering examined. Finally the technical specification should be written to allow different manufacturers to quote on a comparable basis.

Money and effort spent at this stage is usually invaluable. Correcting faults in a heat pump is at least 10 times more expensive after it has been built than at the design stage. The value of a good detailed design is of course true for any type of engineering project. However, what the purchaser must realize is that it is very risky to expect the manufacturer to do the detailed design under the misapprehension that this design is free. Clearly the manufacturer must recoup his design costs in the final tender price so they are definitely not free. Of greater worry is the fact that the manufacturer will always do the minimum amount of design work possible because he knows he may never win the contract (either because he loses it to opposition, or, as is commonly the case, the heat pump is never built). This means he will base the design on data from the feasibility study, which will often be approximate and usually not comprehensive enough. An alternative approach is to contract an engineering design company to do the detailed design. In this case, it is just as important to ensure that the engineering company is working on the correct basis.

There are many examples of heat pumps that were designed and built on very vague data from a feasibility study and the results were disastrous. Systems built were too large, corrosive constituents in the heat source were not identified, start up conditions had not been considered .... the list is almost endless!

Again, it must be stressed that this Resource Guide is aimed at "heat pump facilitators." The design stage that is recommended here does not enter the realm of pressure vessel ratings or compressor valve design. Clearly that is the responsibility of the manufacturer. What is necessary at the design stage is work that relates to the host process and decisions that the manufacturer cannot be expected to make himself. This is best illustrated by two examples.

If the heat source is a gas stream it should be tested for corrosive or fouling components. This is best done by the purchaser and defined in the specification. This saves separate measurements by each manufacturer on the tender list (which is clearly a waste of money) and it may give the purchaser a stronger legal position if the evaporator suffers corrosion at some later date.

A second example is standby strategy. Is it important to have a standby compressor available in case of a breakdown? This is a costly extra and it is impossible for the manufacturer to be certain whether or not it is necessary. This should be decided by the purchaser in the design phase and then clearly specified.

DATA COLLECTION

The first step of the design stage is to collect detailed data which will ensure that the heat pump can be properly specified. All the required data relates to the host process. The heat pump interacts with the host process in two places; energy is extracted from a waste heat source and is delivered to a heat user. In the feasibility stage it was good enough to know the approximate characteristics of these two streams in terms of temperature and flow rate. Now it is important to define more carefully the nature of these two streams. The required data falls into four categories, bulk properties, thermodynamic properties, time related data and site information. Table 6-1 summarizes these requirements.

Bulk Properties

Usually the streams will be a gas, a liquid or two phase mixture (i.e., a condensing vapor or a boiling liquid). The most common streams will either be water or air although many other fluids can be encountered. If the fluids are pure the data collection exercise is relatively easy. However, in most cases there are mixtures of fluids involved that make things much more complicated. For example, an open cycle heat pump that has water vapor as a heat source will usually have to incorporate design features for purging air because the heat source is often a mixture of air and steam. If large quantities of air are present then the host process may require modification to minimize the problem.

The temperature and flow rate of the source and user must be carefully measured under a variety of operating conditions. Usually the full load figures are the only ones used in the feasibility study. Of equal importance are values for typical part load operation, plant start-up, shut down, and any other relevant variations. It is necessary to relate the variations in the source with variations in the user. If they are part of the same process the variations may coincide. However, in many cases the streams may be in two separate and independent processes. This means that variations will be unrelated and, if they

Table 6-1

HOST PROCESS DATA REQUIREMENTS

Type of Data*	Item	Comments
Temperature and Flowrate	Averages Maxima Minima Start-Up/Shut Down Full Load/Part Load	Are variations large and frequent? Do variations in source and user coincide or are they unrelated?
Fluid Properties	Main Components Phase(s) Specific Heat Capacity Latent Heat Corrosive Components Fouling Components Pressures Available Pressure Prop Heat Transfer Coefficients	For vapors/two phase mixtures
Time Related	Annual Availability Daily/Weekly Load Profiles Seasonal Variations Frequency of Start-Up/Shut Down	Particularly if availability is variable Will this be difficult with a heat pump?
Site Information	Heat Pump Location Piping Routes Location of Services	Civil/Structural Details and Access Fuel/Power/Water, Etc.

*These apply to both heat source and heat user are significant, will affect the design of the heat pump. In the case of a condensing vapor heat source or a boiling liquid heat user it is necessary to measure the dew point or boiling point carefully. If there are impurities (such as air in steam) then the values can be significantly different to those of the pure fluid.

The pressure of the source or user stream will influence the design of the heat exchanger. It is important to establish the maximum allowable pressure drop through the heat exchanger. In some cases there will be enough pressure available to incorporate a heat exchanger with no modifications. In other situations an extra pump or fan may be required.

It is also important to identify any fouling or corrosive elements in the streams.

Thermodynamic Properties

Various thermodynamic and physical properties of the heat source and heat user must be established. Firstly, the main fluid constituents must be defined. This will enable the specific heat capacity and latent heat to be found from relevant data tables. Then minor fluid constituents must be investigated. These may not affect the thermodynamic properties but can influence the heat exchanger design because of corrosion or fouling. Information about the heat transfer properties are also required.

Time Related Data

The third group of measurements relate to time. The total annual running hours and annual load factor should be established. It is surprising how often the owner of a plant over-estimates the load factor of a piece of equipment. As this will define the heat pump size it is very important to get this correct. If usage is variable then daily or weekly load profiles are often useful. Conditions at start up, shut down and under emergency or breakdown should also be established as these may affect the heat pump design.

Site Related Data

The final set of data that is required relates to more general information about the site. A good location for the heat pump must be identified and relevant civil and structural engineering details established. If the heat pump is to be electric, then the location of the nearest suitable connection to the electricity network must be established. In some cases the heat source or heat user will not be adjacent to the heat pump so a suitable pipework or ductwork route must be agreed.

ENGINEERING CONSIDERATIONS

In the Feasibility Stage the best heat pump configuration was identified. More detailed data to define the host process has been collected in Step 1 of the Design Stage. Now it is useful to benefit from the experience of previous heat pump installations to ensure that various aspects of the engineering design are given careful consideration. This will ensure that mistakes made by other people are not repeated! First the heat pump must be considered at system level and then at component level.

The Sizing Issue, Again!

The importance of heat pump size has already been stressed. In general the system should be designed to run at a high load factor; peaks in the heat requirement can be more cost effectively supplied by conventional heating methods. Now that more detailed pictures of the heat source and heat user are available, it is possible to reassess the optimum size for the heat pump. This is best done by plotting representative heat load profiles for both source and user. In Figure 6-1 we see an example where the heat source is a stream of warm water at 90°F and the heat user is boiler make-up water. It is clear that the two process streams are not related to each other. The heat source is very steady and is always greater in capacity than the heat requirement. The need for make-up water varies quite considerably and has short duration peaks. From this example we can make the following assessments:

(a) It is wrong to base the heat pump size on the heat source as this would lead to a heat pump that is too large;

(b) similarly it is wrong to make the heat pump cater for the maximum heat requirement of 3.2 MMBtu/hr because heat loads over 2 MMBtu/hr only occur for 15% of the time;

(c) the minimum load of 0.5 MMBtu/h only occurs for 1 hour/day so it is reasonable to run the heat pump on part load during this period.

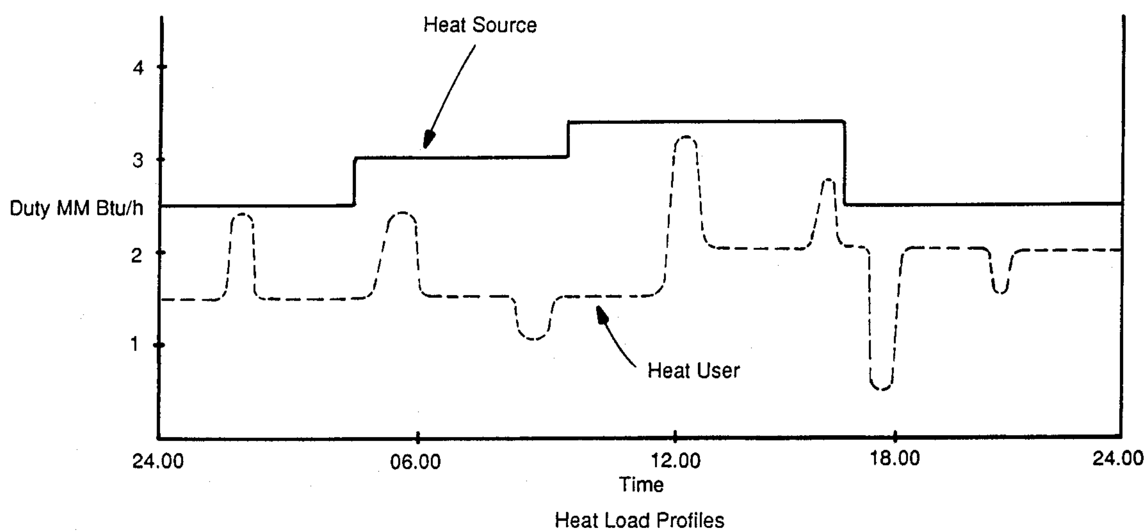


Figure 6-1. Heat Load Profiles

(d) the heat user has two distinct "average" levels of load. For 12 hours the load is about 1.5 MMBtu/h and for the other 12 hours the load is somewhat higher at 2 MMBtu/h. Ideally, the heat pump should be sized at one of these levels. It is not immediately obvious which is best. The shortest payback period will be for a heat pump sized to the lower figure. This will run at 100% load for more than 90% of each day. However, in the long term the slightly larger heat pump will save more money. A discounted cash flow analysis is needed to differentiate between the two options. In general it is safest to chose the smaller size as this will have less control problems than the larger one that must operate at 76% load for 12 hours/day. Use of a hot water storage tank can also be considered together with a heat pump of 1.75 MMBtu/h running at full load 24 hours/day.

Evaporators

Poor evaporator design has led to many problems in first generation heat pumps. The faults cause poor heat transfer coefficients and a loss of both COP and thermal capacity through low evaporating temperatures.

Direct Expansion Evaporators. A common heat pump configuration is to use a direct expansion (DX) evaporator in conjunction with a thermostatic expansion valve. In such an arrangement refrigerant is completely evaporated in one pass through the evaporator (see Figure 6-2a and b). This is different to the flooded evaporator where a mixture of liquid and vapor is present throughout the evaporator. DX evaporators suffer a number of possible problems that lead to poor heat transfer rates and disappointing heat pump performance. These relate to uneven distribution of liquid to the evaporator circuits, which leads to early drying out of the evaporator. This does not occur in a flooded evaporator where heat transfer is good throughout. It is recommended that flooded evaporators are used whenever possible.

Superheat Zone. Many systems are operated with a high superheat at compressor suction. This can be through evaporator design or a badly set expansion valve. In these situations a significant percentage of the evaporator surface can be dry on the refrigerant side as described above. This waste of surface area leads to lower than normal evaporating temperatures. Another serious side effect occurs with systems that have moisture in the heat source. The water vapor cannot condense in the superheat zone of the evaporator as it is too warm. This leads to particularly poor heat transfer in that part of the coil, and a further loss of capacity. The system should be designed for low superheat at the evaporator exit.

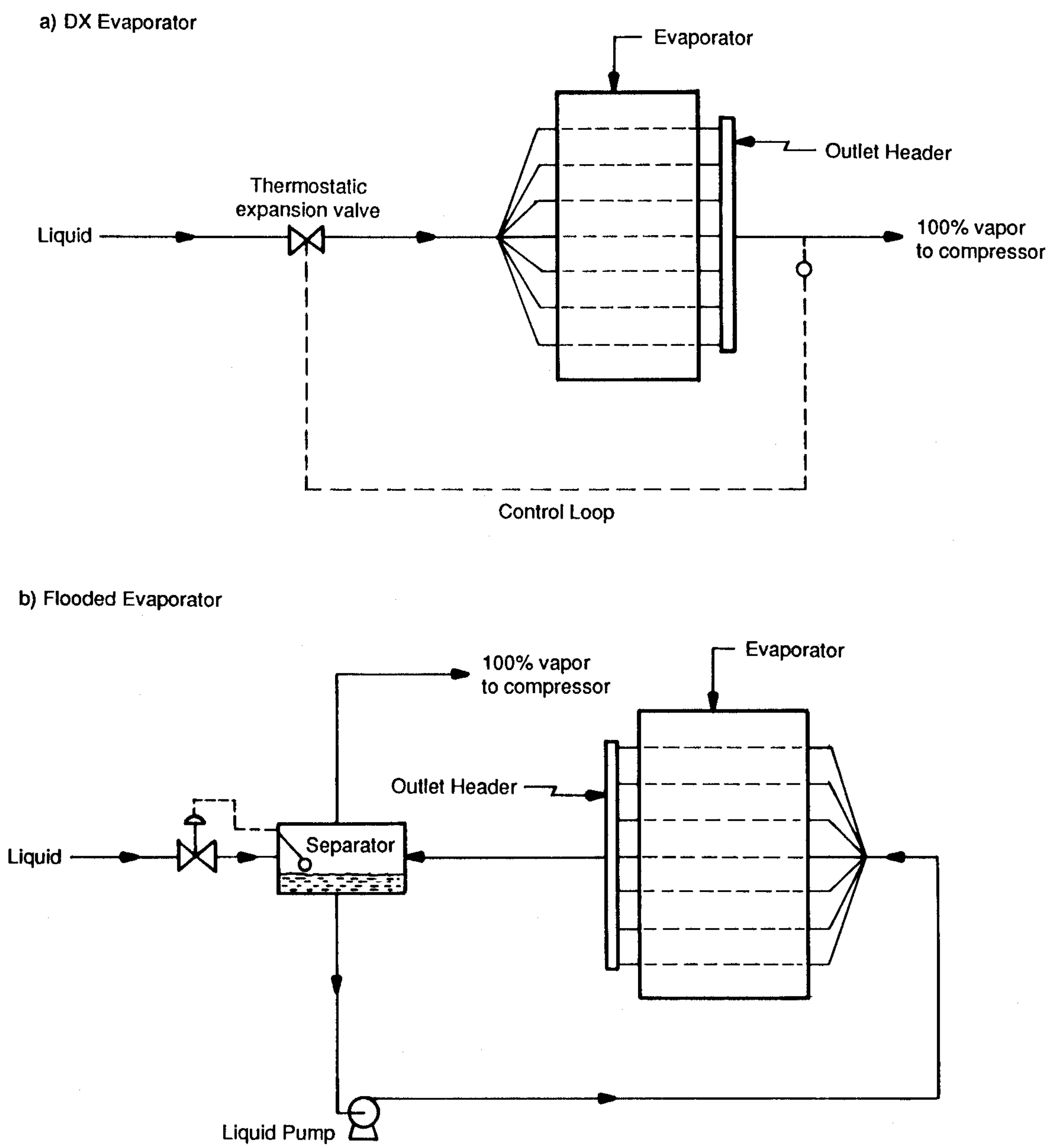


Figure 6-2. Evaporator Arrangements

If necessary, a suction line heat exchanger (using condensed liquid to heat the suction vapors) can be used to protect the compressor from liquid carry over.

Frost Build Up. Most designs of air source evaporators do not minimize the losses that occur through frost build up. It is vital to stop frost forming a bridge between adjacent fins. Useful design rules are:

(a) do not use closely spaced fins - 3 or 4 fins/inch is the minimum spacing that should be chosen. Some small sized systems have used 12 fins/inch and frost has been a tremendous difficulty.

(b) use coils with relatively large face area and a high air face velocity as this encourages water droplets to be blown off before freezing occurs.

(c) place coils horizontally with air blowing downward so gravity can assist the path of water droplets off the coil.

Effects of Oil. Evaporators are often designed on the assumption that the refrigerant is pure. This is not the case in most plant as oil is present as a vapor, in liquid droplets and in solution with the refrigerant. Oil can cause a reduction in evaporator capacity. The effects appear greater in many heat pump applications than in apparently similar refrigeration systems.

Oil return systems have often caused problems, particularly in flooded evaporators. When using R22 special care must be taken in designing the oil return equipment.

Undersizing. The COP of a heat pump is related directly to temperature lift. The temperature difference across the evaporator adds to this lift. Many heat pumps have used evaporators that have too little surface area. This undersizing is caused by use of refrigeration design rules and gives a non-optimized COP.

Fouling and Corrosion. Many heat sources have dirty or corrosive constituents. Great care must be taken to select the correct materials for the evaporator. In two malt drying heat pumps, plastic dipped finned coils were used as evaporators. Fouling occurred because of the close fin spacing and corrosion occurred because the plastic finish was not sound. One of the heat pumps had a new evaporator made from bare stainless steel tubes which was a much more satisfactory design. The other was actually scrapped when a leaking evaporator was added to a long list of technical problems.

Compressors

The compressor has seen major improvements during the last decade for heat pump applications. At first it was believed that completely standard refrigeration compressors could be used. This has been found to be wrong except for extremely robust machines that were previously overdesigned. Great care must be taken using the cheaper machines that are designed for packaged chillers and air conditioning plant. Some common faults have included:

- Valve failure through higher than normal forces from compressed refrigerant. Valve seats, springs and plates must be carefully chosen for heat pump applications.
- Bearing failure through high side thrusts on crankshafts, etc. Stronger bearings must be used.
- Shaft seal leakage.
- Failure of ancillary components such as baffles in oil separators due to higher than normal forces from compressed refrigerant.
- Frequent motor burnouts. With semi-hermetic compressors this has led to great difficulty in cleaning the refrigerant pipework and heat exchangers.

Compressor Efficiency. Is surprising how often a heat pump is designed without reference to compressor efficiency. Bearing in mind that the isentropic efficiency of compressors used for heat pumps varies between 40% and 80% this can lead to serious reductions in financial return. Using a low efficiency compressor has exactly the same effect on the heat pump performance as the use of a low efficiency boiler has on a steam system. Boiler purchasers are unlikely to buy a 75% efficient boiler if 80% is available - they certainly will not use one at 40%.

Compressor efficiency is not necessarily related to capital cost; often a cheaper machine is more efficient than an expensive one. The manufacturer should always be requested to supply isentropic efficiency data relating to three modes of operations:

- design point efficiency - this value allows quick comparison of performance between several machines
- off-design full load efficiency - if the heat pump is required to operate under different evaporating or condensing conditions the off-design figures are important. There is no point in buying a machine

with a very high design point efficiency if the off-design values fall significantly.

- part load efficiency - most compressors have very poor part load efficiency. Ideally, the heat pump should be sized in such a way that it doesn't run at part load. If this is not possible design calculations should take into account low efficiency when the compressor is unloaded.

The best way of presenting compressor efficiency data is through efficiency contours. These were described in Chapter 2, see Figure 2-11.

Condensers

Of the three main components of heat pumps the condenser has caused the fewest problems. The main causes for concern have been fouling or corrosion on the heat user side and undersizing. Similar issues affect evaporators and are discussed above.

In some systems two condensers are used, one for heat recovery and a second unit to reject heat to the atmosphere when heat recovery is not required. The way these vessels are piped together is very important. On at least one installation the heat recovery condenser was acting as a liquid receiver!

It is necessary to consider how the condensers may effect each other. In Figure 6-3 we see an air cooled condenser in parallel with a water cooled heat recovery condenser. The heat pump was designed to operate with just one condenser. In heat recovery mode, the condensing temperature could be 140°F and in air cooled mode, a typical value of a 95°F was expected. The condensers were well piped, compared to the liquid receiver problem described above, having generously sized liquid line connections to the common receiver. However, when the weather was cold and wet the air cooled condenser pulled the head pressure down, even though the fans were not operating, and the heat recovery condenser could not deliver suitably hot water. An extra solenoid valve is required in the air cooled condenser liquid line to prevent this from happening. However, further care must be given to switching from one condenser to the other. Ideally the liquid solenoid should be operated a short time after the vapor line valves are switched to ensure that the air cooled condenser can be drained of liquid. Without these simple practical considerations taken into account, the heat pump is unlikely to operate well.

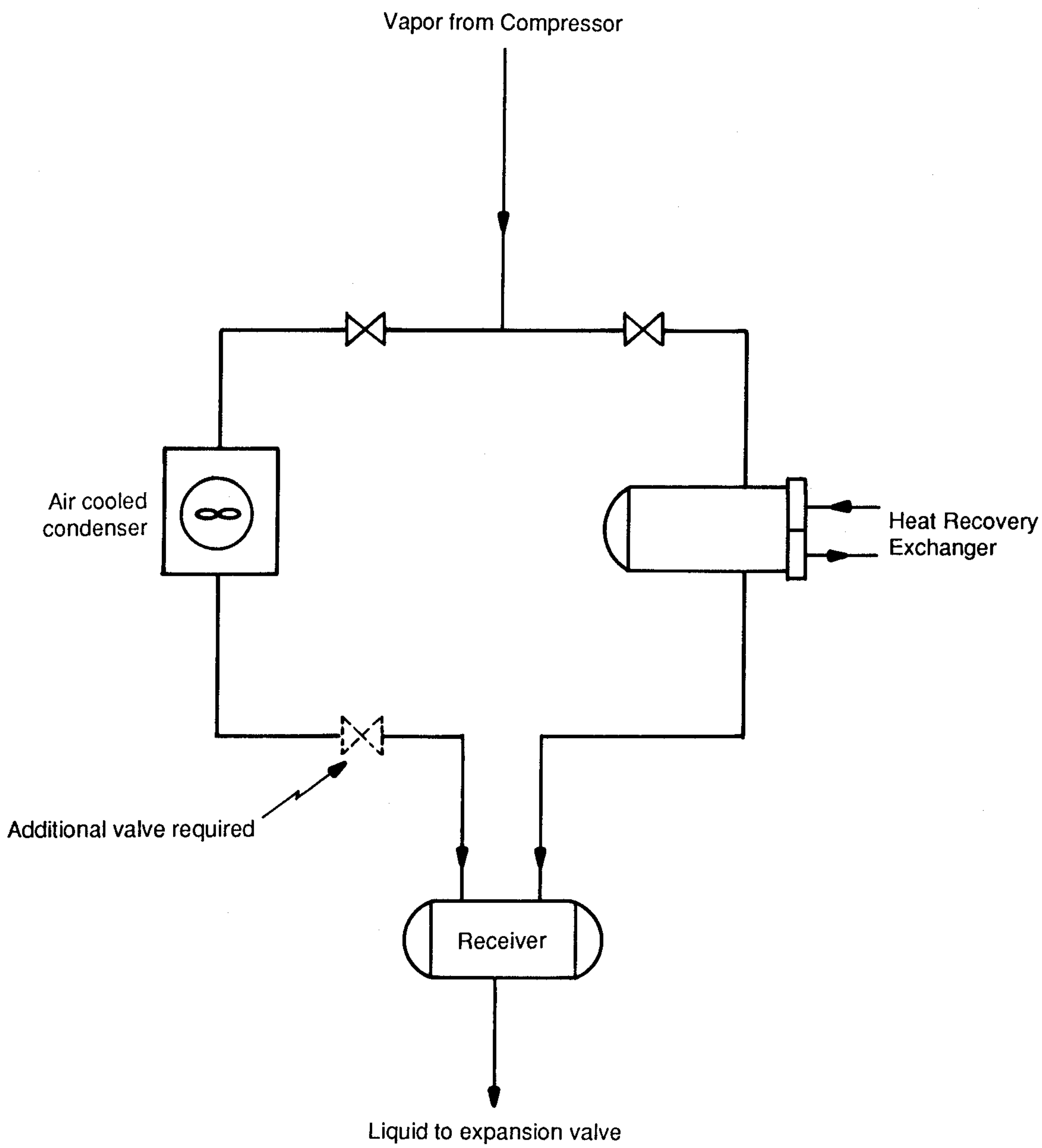


Figure 6-3. Condenser Arrangements

Expansion Valves

As with refrigeration systems a wide range of expansion valves can be used, including thermostatic expansion valves (TEV's), low and high pressure float controlled valves and electronically controlled valves.

TEV's do not suffer from the inefficiencies they cause in refrigeration plant, due to the need for head pressure control. However, superheat settings are often too high, causing poor evaporator performance as described above. Float controls will respond much better to variations in pressure ratio. This can be important when process requirements lead to variations in heat source or heat user temperature. Modern developments in electronically controlled expansion valves will be of great help to designers of second generation heat pumps.

Engine Driven Systems

Engine driven systems are often worth considering because they improve the overall energy savings potential, particularly for applications with high temperature lift. However the engine does lead to a lot of extra design and maintenance considerations:

Engine Air Intake Location. One of the classic lessons learned, and unfortunately relearned, during the last ten years has been the simple rule about engine combustion air intakes. These MUST be ducted from outside the engine room in a position where refrigerant cannot be ingested into the engine. The consequences of CFC refrigerants entering an engine combustion chamber are dramatic. The refrigerant is broken down by high temperatures into highly corrosive compounds of fluorine and chlorine. Severe engine damage is inevitable.

Engine Compressor Vibration. Several large systems have suffered major problems due to the wrong type of coupling between engine and compressor. Flexible compressor and engine mountings have caused many difficulties. The engine and compressor manufacturers must carry out a rigorous vibration analysis to design the mounting and couplings. Analysis should allow for unloaded compressor cylinders, and possibly unloaded engine cylinders resulting from spark plug failure.

Exhaust Heat Exchanger Design. The exhaust heat exchanger must not cause too great a back pressure otherwise engine performance is lost. Materials must be carefully selected to avoid corrosion, particularly if diesel engines are used.

Lubricating Oils. These must be carefully selected in conjunction with the engine manufacturers. It is necessary to check on the rate of consumption of lube oil - on some engines a lot of oil is burned resulting in high annual maintenance costs.

Refrigerant Leakage

Many heat pumps have suffered with refrigerant leakage problems. In general the cause has been poor manufacturing standards, lack of checking and lack of attention to detail. Leakage was a major factor contributing to the downfall of the German domestic heat pump. In a recent field trial in the UK seven out of eight systems suffered from significant leaks. It should be noted that these were systems that the manufacturers knew were to be carefully tested and appraised!

Temperature Lift of Open Cycle Heat Pumps

For many MVR processes it is possible to achieve very high COP by using the smallest practical temperature lift. COP's over 30 are practical in some industrial applications. In order to evaporate water at 212°F it is possible to use steam condensing at about 230°F. This will give a COP of around 20. It also implies a very low pressure ratio in the compressor and axial "fans" have been used in some cases. However, the main heat exchanger must be very large for such low temperature differences.

The trade-off between COP and heat exchanger size is the most important issue in MVR system design. It also illustrates well the need for a new philosophy when designing heat pumps in place of fossil fired systems. In a fossil heated evaporator steam at around 300°F (50 psig) is usually available. Compact heat exchangers are normally used for convenience and low cost. It is important not to fall into the trap of assuming that small heat exchangers should still be used for MVR systems.

As well as the obvious effect on heat exchanger design the issue of temperature lift often affects the mode of operation. Many evaporator heat exchangers become fouled and are cleaned periodically. For MVR plant the fouling could cause an unacceptable drop off in COP. Much more regular cleaning may be required. In breweries, for example, the main evaporator heat exchangers are cleaned about twice a week. This must be increased to a clean in between each brew, about every two hours, to achieve maximum COP.

Boiling Point Rise in Open Cycle Systems. Boiling point rise is a major consideration in the design of evaporation systems incorporating open cycle heat pumps (MVR's). Boiling point rise is the difference between the boiling point of a solution and the boiling point of pure water at the same pressure. For example, pure water boils at 212°F at one atmospheric pressure but a 20% salt solution boils at about 220°F at atmospheric pressure. If the boiling point rise is high, the vapor leaving the compressor must condense at a higher temperature (pressure) than it would in a system with low boiling point rise. This makes the system more expensive to operate.

The physical properties of the solution being evaporated should be carefully checked before recommending vapor compression evaporation.

Desuperheating in a Steam Compressor. When steam is compressed it reaches very high levels of superheat. In many MVR applications steam would reach more than 480°F at compressor discharge. This is unacceptable for most compressor types so it is necessary to inject water into the compressor to reduce the superheat. Control of water injection has proved difficult and it is worth buying good quality control valves for this job. The point at which water is injected is also of importance. Direct injection into the compressor may be best but this cannot be carried out in all types.

Instrumentation

As a general rule it is worth considering a relatively comprehensive set of measuring instruments on any industrial heat pump. This will ensure that regular and meaningful monitoring can take place and that the heat pump can be run under optimum energy saving conditions.

Temperature pockets should be included in the system whenever a significant change in temperature occurs. In the refrigerant circuit of a closed cycle heat pump this includes compressor suction and discharge vapors and liquid from the condenser. If a liquid subcooler is included the temperature after the subcooler should also be measured. Similarly the heat source and heat user stream temperatures should be monitored in and out of each heat exchanger.

The most important pressure measurements are the compressor suction and the discharge vapors. These allow proper assessment of the thermal performance of both the evaporator and condenser (by converting the pressure measurements into evaporating and condensing temperatures using refrigerant tables and comparing these values with heat source and heat user temperatures). Other pressure measurements are less usual although pressure drops through heat exchangers provide a useful way of identifying fouling problems.

Electric power input to the whole system and preferably to each major load should be recorded. Ideally, the mass flow rate of the heat source and/or heat user should also be measured to enable the COP to be calculated. An alternative is to measure the refrigerant flow rate although this is not commonly done.

Safety

The requirements for safety in terms of pressure relief, cut-outs, electrical isolation, etc. are the same as for refrigeration and standard codes should be adhered to.

TECHNICAL SPECIFICATION

The last step of the Design Stage is to write a technical specification. This will clearly define the heat pump that is envisaged and will be used as the basis for manufacturers' quotations. Many heat pumps have been based on a very vague specification. The whole point of the Design Stage is to ensure that attention is given to the kind of details that caused many first generation heat pumps to have problems. Clearly the effort will be wasted unless the specification reflects the work done. There can be a danger, however, that if the specification is too long and rigorous, manufacturers will find it necessary to add extra costs. The degree of detail must reflect the complexity and size of the proposed heat pump. A system with a heat output of, say, 5 MMBtu/hr will be very costly and must be very carefully specified. The smallest industrial systems (at around 250,000 Btu/hr heat output) will usually cost less than $50,000 (before installation) and a 50-page specification is obviously overkill.

A good specification for an industrial heat pump will have the following essentials:

- The host process will be clearly defined. All relevant data about the heat source and heat user will be described
- The preferred heat pump system, derived from the feasibility study, will be presented.
- Any important engineering requirements must be specified
- The specification must be structured in a way that will allow easy comparison of tenders.

One of the values of a good technical specification is that contract responsibilities are clearly defined. If the heat pump installation has problems in meeting the expected performance it is usually relatively easy to find out whether the manufacturer did not meet the specification or that the specification was in fact incorrect. If only a vague specification is given to the manufacturer (with many details only given verbally) then it is often very difficult to prove who is responsible for a problem.

The engineering specification for a large heat pump will usually include two sections; i.e., a General Specification and a Particular Specification. In addition to the engineering specification there must also be a variety of commercial conditions specified, in line with the preferred purchasing and contractual procedures. For smaller heat pumps the general specification (or large parts of it) can be omitted.

The general specification will include requirements about many of the heat pump components that must be met to satisfy the purchaser. For example, pressure vessel and pipework design codes, preferred types of electric motors or pumps, requirements for safety, etc. This type of specification will be familiar to all purchasers of process plant and will not be discussed in this guide.

The Particular Specification

This is the part of the Technical Specification that must be customized to the required heat pump. Usually the General Specification and the Commercial Specification will be based on standard material used regularly by the Purchaser or his Consultants. The Particular Specification for a heat pump is less familiar. The following sections are recommended:

(a) Design Basis. This section outlines all the relevant host process data collected in Step 1 of the Design Stage.

(b) Required heat pump performance. A clear statement should be given that specifies the required heat pump output under relevant process conditions. As well as the thermal output, any other important requirements should be clearly stated (e.g., the need to run at part load, the need for standby, etc.).

(c) Preferred heat pump design. In the Feasibility Stage the option tree was used to investigate the financial performance of various different heat pump designs. The preferred design must be specified and shown as an outline system. Where possible the outline should be made flexible as this will allow the manufacturer the opportunity to use his equipment (and expertise) in the best way. However, if the outline is too flexible then the opportunity to obtain the optimum heat pump design may be missed. Usually the system design will need relatively detailed specification (e.g., that a heat pump with a liquid subcooler is required) but the component design can be left unspecified (i.e., the manufacturer can choose the type of subcooler).

(d) Key Engineering Requirements. All key design features identified in Step 2 of the Design stage must be specified. These can be subdivided into system design, controls and component design.

(e) Heat Pump Installation. When the heat pump is installed it will be necessary to carry out various civil/structural/electrical modifications and also to break into the existing process streams that will be the heat source and heat user. Often the civil work, etc., is best carried out by someone other than the heat pump manufacturer. The preferred responsibilities must be specified. Any restrictions on access to the process streams must be stated (usually the host process must be stopped to connect the heat pump into the system - this may only be possible in an annual shutdown, or at weekends).

(f) Commissioning and Operational Details. Any requirements for specific performance and commissioning tests should be specified. A request for specific operator training and maintenance/operating manuals should also be given.

(g) Tendering Information. An important section of a good Particular Specification is the one that sets out the tendering requirements. If tenders are to be easily compared, it is useful to have identical tables of data from each manufacturer. A blank table of data should be drawn up and included in the specification.

Chapter 7

COMMISSIONING, OPERATION & MAINTENANCE

CHAPTER SUMMARY

In this chapter various important issues concerning the actual usage of heat pumps are discussed. In order to ensure proper operation it is necessary to carry out a proper commissioning program. Having established good operation it is vital that this is maintained. A comprehensive monitoring system is recommended to keep a regular check on energy savings and to identify potential problems as they arise. Finally, it is useful to be aware of the typical operational problems that can arise; these are discussed together with methods of diagnosis and cure.

COMMISSIONING OF A HEAT PUMP

The commissioning phase of a heat pump installation is critical to ensuring that the subsequent performance of the system is close to that specified, and remains so over a long period of time. Codes of Practice exist for commissioning of many types of plant, including air and water distribution systems and refrigeration equipment. All of these component systems are relevant to heat pump commissioning.

It is intended here to highlight some of the important points which must be taken into account during commissioning. It should, however, be noted that the responsibility for commissioning to the customer's satisfaction will normally rest with the equipment supplier/installer.

Gas and Vapor Distribution Systems

When the heat source or user is a vapor or gas (e.g. air) stream, a number of factors related to the distribution system must be taken into account. If these are neglected, a reduced COP could well result.

Preliminary checks are necessary in at least seven areas. These are:

- Velocities and pressure drops - these should be measured and compared with design data.
- System cleanliness - including ducts, fans, filters and coils.
- Flow control devices - correct operation of dampers or inlet guide vanes, if incorporated and any safety features such as hazard isolation doors or pressure relief systems.
- Leak tightness - ductwork and fittings such as access doors, joints etc. should be checked.
- Mechanical components - in particular the lubrication of any moving components should be checked, and the alignment of compressor and fan drives etc. confirmed.
- Electrical components - all electrical components should be checked for correct wiring, fuses, power supplies etc.
- Special equipment - items such as electrostatic filters, features involving gases other than air, and any other potentially hazardous items requiring outside specialist knowledge should be dealt with separately.

<u>Liquid Distribution Systems</u>

Excluding the refrigerant circuits and coolant/lubrication for compressors or drives, many industrial heat pumps also have liquid sources or users which require integrating into the system. Additionally, water systems may be used to clean heat exchangers in the system which may be liable to fouling - most commonly the evaporator. It is important that any automatic wash-down system has a high degree of reliability.

Procedures are generally similar to those for air distribution systems, but the first critical steps following installation are system filling and flushing. Flushing is carried out normally from high to low points, on a sectional basis using isolating valves. Chemical cleaning may be needed in some circumstances, although care should be taken to ensure all traces of chemicals are subsequently removed.

Leak testing is of course necessary, and frost protection should be incorporated (e.g. trace heating or antifreeze agents) if in a cold environment. Checks on pressure and temperature drops should be made, with thermal insulation applied where appropriate. Correct balancing of liquid flow rates is particularly important.

The Heat Pump Circuit

To obtain satisfactory operation of a closed cycle heat pump, carefully planned procedures must be followed associated with refrigerant charging.

Firstly, it is essential to make sure that the refrigerant itself does not become contaminated following delivery by the supplier. Secondly, it is important to keep the circuit and all associated components on the refrigerant side as clean as possible, and free from moisture, oil, any fluxes, dirt etc.

Leak testing and pressure testing are critical in ensuring the integrity of the refrigerant pipework, valves and other components. If a vacuum leak testing method is used, this will also assist in removing moisture from the system as it is pumped down. However, prior pressure testing will be necessary for most heat pumps, and this will also help identify gross leaks. Leakage has been a major problem with first generation heat pumps. The purchaser should insist on the highest possible standards of leak tightness. Ideally, the leak tests should be independently witnessed.

The heat pump circuit must again be evacuated prior to filling with refrigerant. Charging will normally be done by adding the refrigerant in its liquid phase. Both undercharging and overcharging will lead to reduced heat pump performance. It is important to check that the correct refrigerant is being used, or if a refrigerant mixture is selected, that the correct proportions are added.

Correct selection of the compressor lubricating oil also needs consideration. Oil-refrigerant compatibility is particularly critical in heat pump systems operating at higher temperatures, and it may be necessary to use one of the newer synthetic lubricating oils. Again, oil cleanliness cannot be too highly stressed.

It will generally be found, either soon after charging or after early heat pump operation, that the system will require purging of trapped air and non-condensible gases which may have been generated by some local contaminant reaction. It may be advisable to purge once or twice during this early operating period. If excessive or over frequent purging is needed, this can indicate the presence of more serious contamination, or a leak.

For both open and closed cycle units, compressor operation is the critical component. It is important to ensure that liquid carryover is avoided and that the lubrication system is operating correctly. The manufacturers recommendations regarding start-up should be followed.

MONITORING OF PERFORMANCE

Performance monitoring of a heat pump installation is essential for several reasons. The principle "raison d'etre" for the purchase of a heat pump in the first place is energy cost reductions. Performance monitoring will ascertain whether the predicted reductions are being achieved consistently. Should this not be the case, such monitoring will assist the operator and equipment supplier to pinpoint system components which are not performing to specification, for example the compressor, drive, evaporator, condenser, or some aspect of the control system.

Plant log sheets provide an effective way of gathering data on plant performance but they are only of real use if:

- the data recorded is accurate
- the information is analyzed correctly
- any problems identified are followed up

Key operating parameters, (temperature, pressure, hours run, power consumption etc.) should be included on the sheets along with details of their normal operating range so that out of limits operation can be seen readily

Performance Tests

The log sheets can only provide a rough guide to the operation of the heat pump. Periodically, review of detailed tests should be carried out. Essentially, a full consistent heat balance should be calculated from measured conditions and then compared with design and expected performance figures for the various heat pump components. In this way you can determine the energy savings and identify faults. The importance of the monitoring cannot be overstressed. Without analysis of measured data it is quite common to find a heat pump operating but achieving a heat recovery level much lower than it should be.

Figure 7-1 shows the measurements required to assess the performance of closed cycle vapor compression heat pumps. Using this data, equipment performance can be calculated, item by item.

<u>The Evaporator</u>. The mass flow rate of the heat source stream can be determined from the flow/pressure characteristics of the pump or heat exchanger (or both). The heat duty can then be calculated using fluid properties and the temperature fall $(T_2 - T_1)$.

The refrigerant evaporating temperature, T_E, will be the saturated temperature corresponding to the compressor suction pressure. The performance of the evaporator can then be compared with the expected performance to identify problems such as fouling.

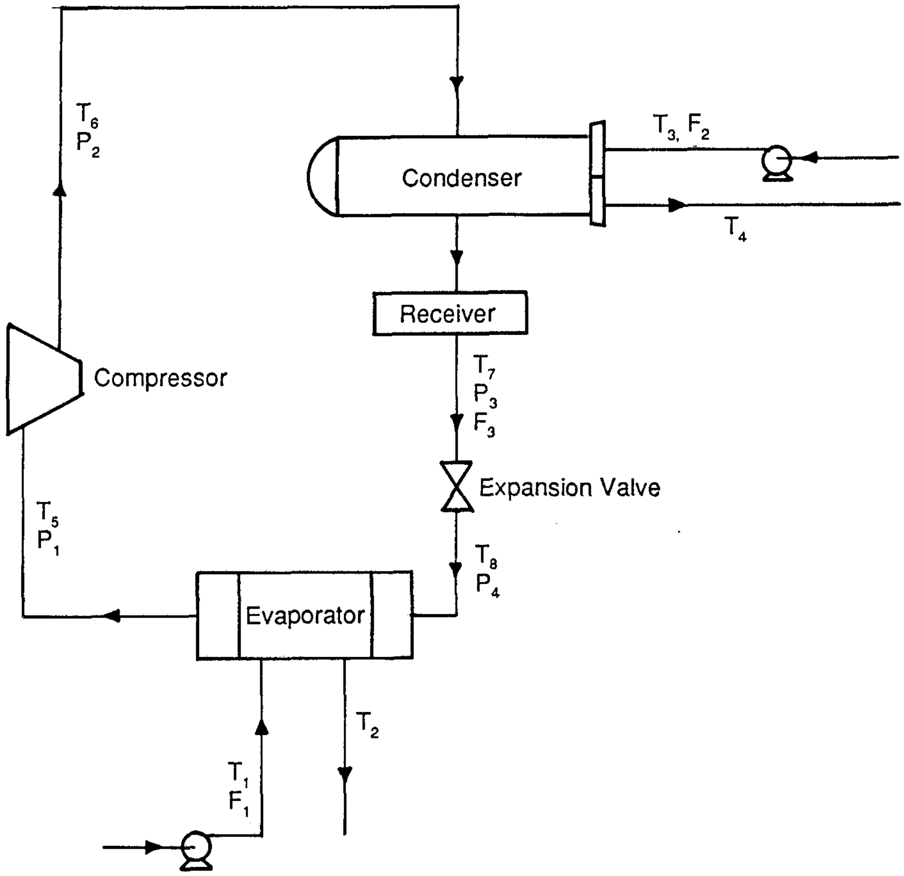
<u>The Compressor</u>. Compressor manufacturers provide performance data for their machines either as curves or in tabular form. Essentially, these relate the heat duty of the evaporator and condenser to the compressor power requirement.

The volume of vapor being compressed can be calculated from a knowledge of the evaporator duty (and the pressure drop across the flash valve in in closed cycles). The actual power taken by the compressor can be measured and then compared with the expected value. Motor efficiency and power factor should be taken into account.

<u>The Condenser</u>. The condenser can be analyzed in a similar way to evaporator. The heat duty should be equal to that of the evaporator plus the power required to compress the gas. Any inconsistencies between actual and expected performance is indicative of problems and should be investigated further.

<u>Automatic Plant Monitoring</u>

As mentioned above, monitoring assists considerably in minimizing the effects of malfunctions, by warning of their possible occurrence, where feasible. Automatic plant monitoring, using microprocessor-based systems, is now available, and this can relieve plant operators of many of the tedious tasks associated with data logging.



Key

T_1 = Temp. of Heat Source Entering Evap.

T_2 = Temp. of Heat Source Leaving Evap.

T_3 = Temp. of Heat User Entering Cond.

T_4 = Temp. of Heat User Leaving Cond.

T_5 = Temp. of Refrigerant at Compressor Suction

T_6 = Temp. of Refrigerant at Compressor Discharge

T_7 = Temp. of Refrigerant in Receiver

T_8 = Temp. of Refrigerant Entering Evap.

F_1 = Flow Rate of Heat Source

F_2 = Flow Rate of Heat User

F_3 = Flow Rate of Refrigerant

P_1 = Compressor Suction Pressure

P_2 = Compressor Discharge Pressure

P_3 = Pressure prior to Expansion

P_4 = Pressure after Expansion

Figure 7-1. Data Requirements for Heat Pump Performance Monitoring

Such systems can automatically and continuously monitor the essential parameters of the heat pump, and present the data on demand. It is also comparatively simple to incorporate a capability for providing at regular intervals data on plant performance and efficiency, possibly in the form of an overall COP.

A typical monitoring system would be called upon to produce the following data during each shift or at any other time interval:

- Running times.
- Compressor suction and delivery temperatures and pressures.
- Evaporator inlet and outlet temperatures.
- Condenser subcool and approach temperature.

Following an analysis of data over an extended period of, say, four weeks, such a system would generate figures for:

- Overall COP.
- Analysis of source and delivery temperatures (time and date-related).
- Analysis of running times at different loads.
- Total running time.
- Time to next scheduled service.

Such a system would not over-ride the normal trips supplied with the compressor and other components, and solely serves a monitoring function. Until confidence is gained in its performance, normal manual monitoring procedures can be carried out.

TYPICAL OPERATIONAL PROBLEMS

The operational problems encountered with heat pumps are basically similar to those familiar to refrigeration plant users. Faults can occur in any of the fluid streams (i.e. the source, the user and refrigerant) or the problem may be a failure of a specific heat pump component.

In both the heat source and heat user streams the most common problems are fouling or reduced flow rate. In both cases the log mean temperature difference observed in the relevant heat exchanger will be seen to rise above design values. If for example the evaporator is fouled the refrigerant evaporating temperature will fall below design values. Similarly if the evaporator is starved of air (assuming the heat source is air) the same symptom will be observed. Measuring the pressure drop across the evaporator differentiates these two problems; lack of flow leads to a low pressure drop whereas fouling is indicated by a high pressure drop.

There are four common problems related to the refrigerant stream. These are undercharge, overcharge, air contamination or oil contamination. Undercharge usually occurs through leakage. it can lead to significant drops in COP - eventually vapor is pumped uselessly round the circuit. If a liquid line sight glass is fitted undercharging is observed by the presence of bubbles in the liquid line. The absence of liquid subcooling at the condenser exit and low evaporating pressures are also indications of undercharge. Overcharging with refrigerant can either lead to an evaporator or a condenser with too much liquid inside it (depending on the type of expansion value used). If the evaporator is overcharged there is danger of liquid getting back to the compressor which could cause damage. If the condenser is overcharged the condensing pressure will rise and liquid subcooling will increase.

If air is present in the circuit it will always collect in the condenser. Air in leakage is quite likely in systems where the evaporating pressure is below atmospheric. The air prevents proper condensation taking place - which leads to high head pressure and reduced COP. The pressure of air can be confirmed by isolating the condenser from the refrigerant circuit and running the heat user stream through the condenser at a constant temperature. If no air is present the pressure inside the condenser will give a saturation temperature that is equivalent to the heat user temperature. If the pressure is higher than expected air is present and should be purged.

If oil is present it will collect in the evaporator and cause fouling problems. This is only a major problem on flooded evaporators and regular checks for oil fouling should be made. With CFC refrigerants automatic oil purge systems are required as the oil and refrigerant are miscible. With ammonia oil can simply be drained from the evaporator. A common misconception is that heat pumps "burn" oil like a car. Clearly this is impossible. If the compressor is requiring extra oil on a regular basis this implies oil is building up inside the system.

Various components in the heat pump can fail. The most likely ones are the expansion valve and the compressor. Expansion valves can stick open - usually leading to gas bypass and high evaporator pressure. Compressors can fail in a number of ways, some of them disastrous! On reciprocating machines, valve failure is common; this is often associated with high discharge temperatures. Centrifugal compressors are liable to surge if they are run at off design conditions. This may be observed by heavy operation or fluctuations in power absorption. If allowed to persist motor burnout or compressor rotor failure can occur.

Many other operational problems can occur. These are not necessarily specific to heat pumps. Control problems are common through probe failures or incorrect settings. A practical textbook on refrigeration is a good source of reference for other types of failure.

Chapter 8

HOW TO MAKE IT HAPPEN

CHAPTER SUMMARY

Chapters 1 to 7 of this Resource Guide have provided a comprehensive background to the field of Industrial Heat Pumps. This has included much valuable information relating to the performance, design, evaluation, and operation of heat pump systems. The way in which the heat pump competes with other energy saving opportunities was also discussed and the importance of ensuring that the heat pump is compared to these other options in an unbiased fashion was stressed. Pinch Technology was introduced as the best tool for identifying good heat pumping, passive heat recovery, and other energy cost reductions measures. Existing applications, and the current state of the heat pump market were also presented.

This information forms the basis of a design methodology covering the feasibility assessment, detailed design, and commissioning and operation of industrial heat pumps. Of course all this, while useful, is of limited value unless it is used effectively to promote heat pump technology to potential industrial users with the result that more heat pumps are installed.

So, just how can the information presented in this Resource Guide be used effectively? Chapter 8 contains some ideas!

THE NEED FOR SUCCESS

Poor reliability, due to equipment failure, has been a common experience for 1st generation heat pump owners, particularly those with closed cycle systems. In many cases, this has led to poor economic performance and a lower than expected return on investment. Another factor contributing to poor economic performance has been oversizing. Systems have often been designed to deliver peak heat demands, resulting in part load operation (at low efficiency) for the majority of the time. Apart from increasing running costs, this represents poor use of capital. If these mistakes continue to be made, the industrial heat pump will not gain more widespread use. However, these mistakes are not necessary. It is time to learn from the 1st Generation experience. Good designs, based on the guidelines presented in this Resource Guide will be reliable and meet expected economic performance.

To increase user confidence, in the short term, it is important to encourage successful case studies, but initially a conservative approach is required. It is better to have a small number of visible successes rather than a larger number of partial success or failures. Developing the market in this cautious way will lay solid foundations for the future.

HOW TO USE THIS RESOURCE GUIDE

The Resource Guide was not intended to be a completely comprehensive reference on the subject of heat pump technology. After reading the Guide, the reader may feel like an expert, but . . . this is not really the case! The Resource Guide gives a good overview of the technology and provides a good structure for the design and appraisal of heat pump systems, but . . . this doesn't mean that the reader is qualified to perform these tasks. That takes real expertise and experience.

So, what can the reader do after reading this Guide? He or she is in an excellent position to be a facilitator and promote the use of industrial heat pumps by,

- Advising clients on how to do it right.
- Asking the right technical questions.
- Recommending what external expertise is required at each stage.

This is a useful and important role of great benefit to potential users and manufacturers.

Finding the Market

What steps should be taken to get heat pumps installed? A first step is actually identifying good applications. This can be done by examining the industrial customer base in the service territory to identify companies with heat recovery and heat pumping potential. The industries listed in Chapter 3 can form a basis for the initial search but don't forget that there may be other applications which haven't been discovered yet. Concentrate on industries with long running hours i.e., 6000 + hours/year and look out for sites installing new equipment.

Generating and Evaluating Options

The decision to install a heat pump or any other energy cost reduction equipment will be based on an economic and technical comparison of the alternatives. Therefore, having found a suitable site, discuss all the energy cost reduction options with them. Don't just try to sell a heat pump - explain the 'Option Tree'.

It is very important to understand the operations on-site. Use the '5 Minute Screening' to quickly determine whether it is worth while looking for energy cost reduction opportunities. Identify how much energy is used and what it is used for. (It is interesting to note that in many industries, all the purchased energy is eventually rejected to cooling water or the atmosphere. For example, if cold water is processed into cold beer, there is no net energy change in the water. This implies that all energy purchased in a brewery must leave as 'waste' heat. This is true of many food processes and low temperature industries. There are however, exceptions, particularly when the process involves chemical reaction.) Find out about the site's financial criteria (i.e. payback, rate of return, maximum capital outlay) and other relevant criteria such as the need to debottleneck, limitations on boiler capacity, and so on. The background knowledge obtained in the '5 Minute Screening' is essential for understanding how well each proposed project meets the site's objectives.

To generate and evaluate projects, follow the three step feasibility study methodology described in Chapter 5.

1. Collect Data
2. Generate Options
3. Assess Options

Remember that this is an iterative procedure. In the early stages, approximate data is sufficient, but as projects are investigated in more detail, more accurate data is required. Use Pinch Technology, or an alternative thermodynamically consistent approach, to generate options. As discussed in Chapter 4, Pinch Technology offers an excellent approach to identifying heat pumping and all other heat and power options. Remember to classify the projects identified as either independent, exclusive, or domino. This helps with understanding of the interactions between projects and will allow a mutually consistent set of projects to be developed.

When assessing options in the early stages, use approximate capital costs and savings to calculate preliminary economics. Make sure that the correct marginal energy costs are used. Some projects may be eliminated at this stage because the paybacks will obviously be unacceptable. Use other screening criteria such as relative location of heat source and heat user to reduce the number of options. The quality of the savings and capital estimates will need to be improved as project definitions are firmed up.

When a potential heat pump application is found, the compressor power requirement can be estimated from the formula given in Chapter 5. Capital cost estimates are slightly more difficult, but Figure 5-2 gives an indication of the range of costs that may be encountered. Look for open cycle opportunities, they are likely to cost less to build and operate than closed cycles. If a closed cycle is appropriate, don't just consider a four component cycle. Look for subcooling and cascading opportunities which improve the COP.

The economic benefits of heat pump are particularly sensitive to the relative costs of fuel and power. When calculating payback or rate of return, ensure that the sensitivities to fuel price changes are understood. This will help to avoid surprises after the equipment is installed. Also don't forget to include credits for other benefits accrued by installing a heat pump (e.g. improved product quality, increased throughput, etc.) in the economic analysis. If the economic case for a heat pump is marginal, it may be worth considering a special rate structure for the heat pump load which swings the economics in favor of the heat pump.

During the detailed design phase, take care to select the most appropriate heat pump size. Go for long operating hours and a high load factor. Choose the system components carefully, particularly the compressor, and investigate heat pump performance for all the operating cases that are likely to be encountered.

Consideration should be given to working with an independent consultant. This will improve credibility and bring knowledge and experience to the project that may not be available within the utility. This in turn, will improve the quality of results and the benefits to the customer.

If Pinch Technology is used, the utility again has the option of working with an independent consultant or training its own staff to do pinch studies. The decision on which route is best should be based on the resources available within the utility, the perceived need for the skills and the frequency with which such studies will be done.

Developing relationships with equipment suppliers and vendors is essential. This will make obtaining capital estimates easier and also help in obtaining knowledge about other installations and applications.

Presenting the Results

For the energy cost reduction study to be a success, the customer must believe the results and believe that utilities involvement is of benefit to them. Therefore, communication with the customer must take place throughout the study and all efforts must be taken to ensure a good working relationship.

It is in the best interest of the utility to present the study results in an unbiased fashion. By recommending heat pumps when they are not the best option for a customer, the utility will lose its credibility. The results package should be presented in very clear terms showing exactly which projects are recommended, along with the project savings and payback (or return).

The study results should come as no surprise to the customer, having been developed in close cooperation with the customer to ensure that the appropriate objectives are satisfied.

Follow-up

After presenting the study results, follow-up by the utility is essential to ensure that the proposed projects are given appropriate consideration by the customer. Such follow-up includes being available to answer questions about the study and providing contacts with equipment vendors.

Finally, when an installation is operating successfully, make sure the system is monitored and publicize the results.

Good Luck!

Chapter 9

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Appendix A

INDUSTRIAL PROCESS HEAT PUMP EQUIPMENT SUPPLIERS

A-C Compressor Corporation
1126 South 70th Street
West Allis, Wisconsin 53214
414-796-3999

APV Equipment, Inc.
395 Fillmore Avenue
P.O. Box 366
Tonawanda, New York 14150
716-692-3000

ASEA Brown Boveri, Inc.
50 Chestnut Ridge Road
Montvale, New Jersey 07645
201-391-8930

Ambient Technologies
1250 Broadway
New York, New York 10001
212-244-1412

Application Engineering Corporation
801 AEC Drive
Wood Dale, Illinois 60191
312-595-1060

Aqua-Chem, Inc.
P.O. Box 421
Milwaukee, Wisconsin 53201
414-961-2749

Atlas Copco Comptec
20 School Road
Voorheesville, New York 12186
518-765-3344

Blaw-Knox Food & Chemical Company
Box 1041
Buffalo, New York 14240
716-895-2100

Budco, Inc.
21 Old Windsor Road
P.O. Box 357
Bloomfield, Connecticut 06002
203-242-6180

Continental Products, Inc.
P.O. Box 418228
Indianapolis, Indiana 46241
317-241-4748

Dedert Corporation
20000 Governors Drive
Olympia Fields, Illinois 60461
312-747-7000

E-Tech, Inc.
3570 American Drive
Atlanta, Georgia 30341
404-458-6643

Elliott Company
North Fourth Street
Jeannette, PA 15644
412-527-2811

FES, Inc.
Box 2306
York, Pennsylvania 17405
717-767-6411

Food Machinery Sales & Service, Inc.
P.O. Box 692
Canandaigna, New York 14424
716-394-5606

Friedrich Air Conditioning
Climate Master Division
2007 Beech Grove Place
Utica, New York 13501
315-724-7111

GEA Food & Process Systems Corporation
8940 Route 108
Columbia, Maryland 21045
301-997-9500

Goslin-Birmingham
P.O. Box 398
Birmingham, Alabama 35201
205-324-7511

Graver Water
2720 U.S. Highway 22
Union, New Jersey 07083
201-964-2600

HPD, Inc.
305 East Shuman
Naperville, Illinois 60566
312-357-7330

Hackman-MKT, Inc.
100 Pinnacle Way
Suite 165
Norcross, Georgia 30071
1-800-445-7462

Heat Exchangers, Inc.
8131 N. Monticello Avenue
P.O. Box 790
Skokie, Illinois 60076
312-679-0300

Hitachi Zosen U.S.A., Ltd.
150 East 52nd Street
20th Floor
New York, New York 10022
212-355-5650

Koch Engineering
161 East 42nd Street
New York, New York 10017
212-682-5755

LICON, Inc.
2442 Executive Plaza
P.O. Box 10717
Pensacola, Florida 32504
904-477-0334

Lewis Refrigeration Company
395-T West 1100 North
North Salt Lake, Utah 84054
801-292-0493

Marlow Industries, Inc.
10451 Vista Park Road
Dallas, Texas 75238
214-340-4900

Marvair Company
P.O. Box 400
Cordele, Georgia 31015
912-273-3636

McQuay, Inc.
13600 Industrial Park Blvd.
P.O. Box 1551
Minneapolis, Minnesota 55440
612-553-5330

Mechanical Technologies, Inc.
968 Albany-Shaker Road
Latham, New York 12110
518-785-2211

Mycom Corporation
19475 Gramercy Place
Torrance, California 90501
213-328-6279

Niro Atomiser Food & Dairy, Inc.
1600 County Road F
Hudson, Wisconsin 54016
715-386-9371

Nyle Corporation
P.O. Box 1107
Bangor, Maine 04401
207-989-4335

Patterson International
Dantherm Systems Division
208 East Adams Street
Cambridge, Wisconsin 53523
608-764-8300

Refrigeration Engineering Corporation
8799 Crown Hill
P.O. Box 3C
San Antonio, Texas 78217
512-824-2336

Resources Conservation Corporation
3101 N.E. Northup Way
Bellevue, Washington 98004
206-828-2400

Sullair
3700 East Michigan Blvd.
Michigan City, Indiana 46360
219-879-5451

Swenson, Inc.
15700 Lathrop Avenue
Harvey, Illinois 60426
312-331-5500

Thermo Electron Corporation
45 First Avenue
P.O. Box 9046
Waltham, Massachusetts 02254
617-622-1000

Vilter Manufacturing Corporation
2217 South First Street
Milwaukee, Wisconsin 53207
414-744-0111

York International Corporation
P.O. Box 1592
York, Pennsylvania 17405
717-771-7890