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Energy
CONSERVATION

NEW INDUSTRIAL HEAT PUMP APPLICATIONS TO VEGETABLE DRYING

Phase I

Final Report

February 1991

Work Performed Under Contract No. FC07-89ID12859

For
U.S. Department of Energy
Office of Industrial Technologies
Washington, D.C.

By
Utah Power and Light Company
Salt Lake City, Utah

and

TENSA Services, Inc.
Houston, Texas

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Prepared for
U. S. Department of Energy
Idaho Operations Office, Idaho Falls, ID
Sponsored by the Office of the Assistant Secretary
for Conservation and Renewable Energy
Office of Industrial Technologies
Washington D.C.

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Section 1

SUMMARY

The key findings of this study are as follows:

- By changing the inlet and outlet configuration of the primary dryer, the dryer inlet temperature can be reduced by 100°F. This will have a significant impact on the product quality.

- A heat pump system which recovers energy from the primary dryer exhaust air can replace about half the heating duty in the primary dryer.

- A heat recovery system which recovers energy in the boiler flue gas can supply the rest of the heating duty in the primary dryer.

- With a fuel cost of $3.00/MMBtu and electricity cost of 3 c/kWh, the heat pump system will give the following economics:

Savings in Hot Utility:	45%
Total Hot Utility Savings:	$146,000/yr
Electricity Cost:	$69,750/yr
Net Annual Savings:	$76,250/yr
Total Investment:	$160,000
Payback:	2.1 years

This economics is based only on heat recovery in the combined heat integration and heat pump system, no credit has been given for improved product quality, flue gas heat recovery, and savings made in the subsequent chiller duty.

This study also confirmed an earlier study (DOE/ID/12583-1) performed by TENSA Services concluding that food drying processes have a very good potential for heat pumping. Similar plants produce 280 MMlb/year of product. Scaling up the case study results to the national capacity shows a total potential saving of 0.65 trillion Btu per year from similar heat pump applications.

Section 2

INTRODUCTION

Application of pinch technology to selected industries in an early screening study done by TENSA Services (DOE/ID/12583-1) identified potential for heat pumps in several industrial sectors. Among these, the food industry (SIC code 2034) was considered one of the promising sectors for advanced heat pump placement.

The site selected for this study is a vegetable dehydration plant and the participating utility is PacifiCorp. The objective of this study is to further identify the energy savings potential through advanced heat pumps and other energy conservation methods developed in the context of pinch technology.

The Department of Energy's (DOE) Office of Industrial Technologies has been sponsoring a program on advanced heat pumps. The goal of the Industrial Heat Pump program is to advance industrial heat pump technology to increase the energy use efficiency in U.S. industry. This project was directed primarily towards more efficient utilization of heat pumps by improved integration into industrial processes.

Historically, heat pumps were placed to reduce the energy consumption in a particular unit operation such as evaporation or distillation systems. While this approach usually resulted in energy savings in unintegrated processes, integration through heat exchange often had greater energy saving potential and cost less. Later experience has shown that including heat pumps in the integration scheme produces the lowest process energy consumption.

TENSA Services, in the above referenced DOE sponsored study, has developed a systematic method of determining the optimum placement and sizing of industrial heat pumps. This was done through the use of pinch technology, a thermodynamic analysis of the heating and cooling characteristics of a process. Though originally developed for optimizing heat exchanger networks

(HEN), pinch technology is now being applied to the integration of the entire range of process components including heat exchangers, heat pumps, heat engines, and various unit processes such as distillation columns, evaporation trains, driers etc.. Pinch technology is gaining recognition as a valuable process design tool and has pointed the way to many energy-saving ideas in the process industries. An introduction to pinch technology is provided in Appendix A.

Section 3

PROCESS INFORMATION

The site selected for evaluation is a vegetable dehydration plant. The portion of the plant studied is shown in Figure 3-1. As shown in the figure, the product proceeds through a sequence of washing, cooking, mixing, conditioning, drying, cooling, and screening steps.

The plant uses a boiler to generate 350 psig steam, a portion of which is used in the primary dryer. Two steam turbines are employed to generate power for boiler feed pump and electricity for plant use. The plant also consumes 152 psia process steam, some of which is let down from 350 psig through the turbines and some of which is let down from 350 psig through throttling valves. The condensate from 350 psig steam used to heat drying air in the primary dryer is also flashed to obtain additional 152 psia steam. About half of the 152 psia steam is used in other processes which are not part of this study.

The current hot utility consumption is 14.1 MMBtu/hr in the process studied. Most of the heat (11.3 MMBtu/hr) is consumed in heating the inlet air to the primary dryer as shown in Figure 3-2. The balance of the heat is used to make hot water for the washing and cooking operations. The only cooling requirements are in the conditioning step and after the final dryer where product is cooled with chilled air. These cooling loads were small, however. The portion of the process studied does not presently have any process to process heat exchange. A summary of the process and utility data are shown in Table 3-1.

Accurate knowledge of the cost data for the plant is essential in order to confidently evaluate the payback. The economic data presented in Table 3-2 includes the cost of exchanger area on an installed basis and the number of hours per year that the plant operates. The boiler and turbine efficiencies are also listed in Table 3-2.

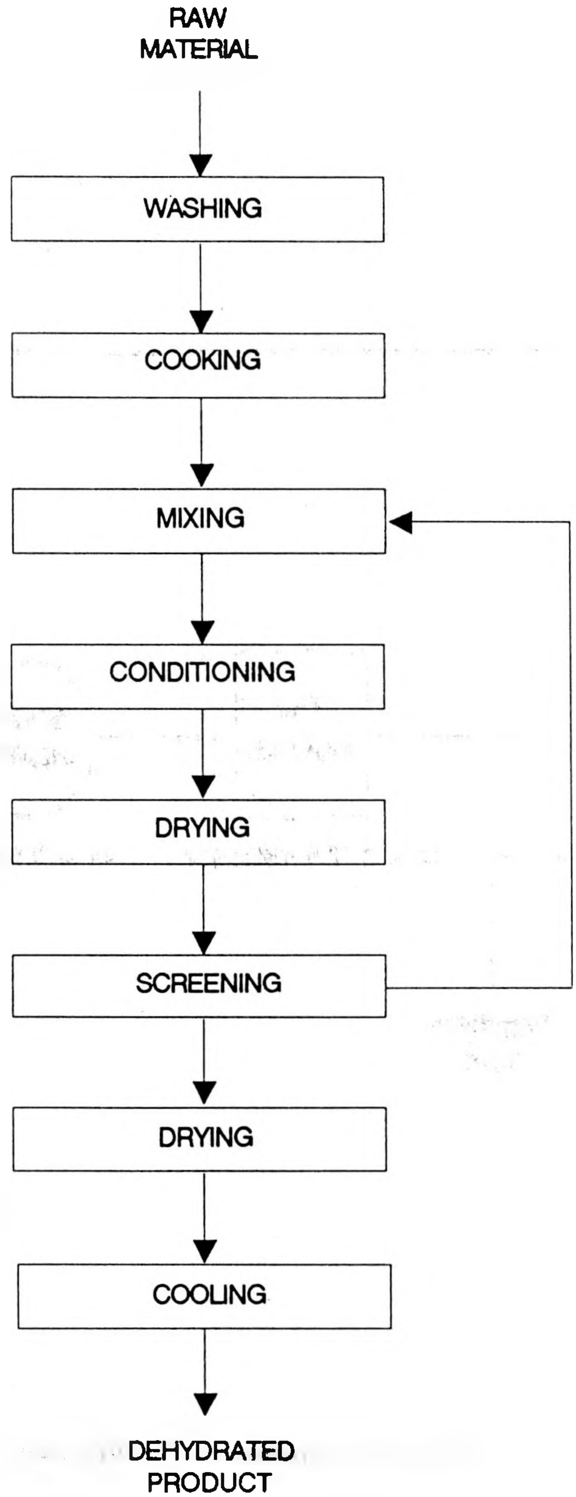


Figure 3-1 Block Flow Diagram of a Typical Vegetable Dehydration Process

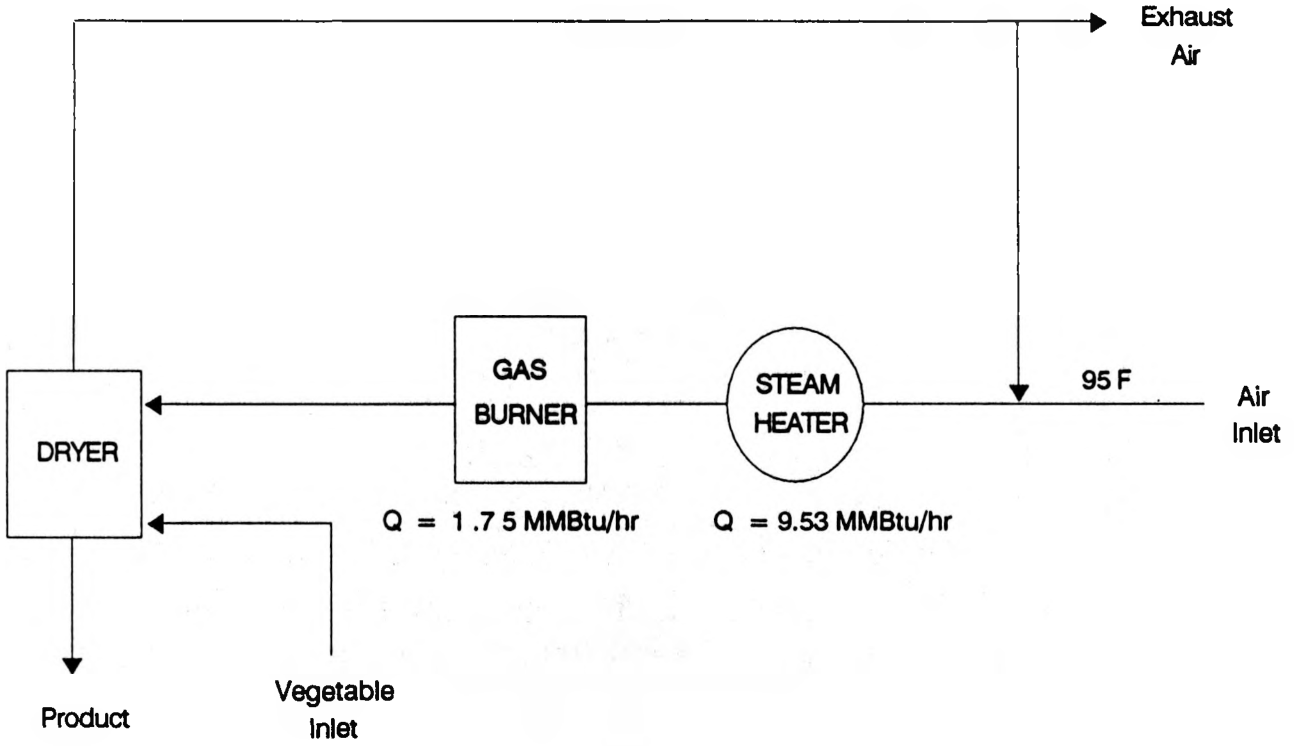


Figure 3-2 Schematic of Existing Dryer

Table 3-1

STREAM DATA

Stream Name	Temperature Source (°F)	Temperature Target (°F)	Assumed Film Coefficient Btu/hr°F ft2	Calculated Load (MMBtu/hr)
Product	110	85	50	0.03
Process Water	65	180	50	0.86
Primary Dryer Air	118	400^+	30	11.30
Secondary Dryer Air	95	250	30	1.75
Final Dryer Air	95	216	30	0.16

Table 3-2

ECONOMIC DATA

Heat Exchanger Area Cost ($\$/ft^2$)	= 10
Fuel Cost ($/MMBtu)	= 3.00
Electricity Cost ($/kWh)	= 0.03
Annual Operating Hours	= 7200
Boiler Efficiency	= 0.65

Section 4

TARGETING AND DESIGN

In this section the addition of heat exchangers and heat pumps will be evaluated relative to the potential for reduction of energy consumption in the present process configuration. Then a new configuration will be considered in which the drying temperature has been reduced to enhance product quality. Finally the potential for other energy savings will be evaluated. Pinch Technology was used in most of these evaluations and in the following discussions, a basic understanding of Pinch Technology by the reader is assumed. The targeting studies were carried out with the aid of a software package developed for EPRI by TENSA Services called HPSCAN (Heat Pump SCreening ANalysis).

4.1 HEAT EXCHANGER NETWORK

In this task, the theoretical minimum quantities of heat exchanger area and energy consumption for various DTmin were determined. This was done by first producing the hot and cold composite curves. These curves are a summation of the respective heating and cooling requirements of the process and the temperature levels at which these requirements exist. The plot of the composite curves for the process is shown in Figure 4-1.

The only hot streams requiring cooling are the product streams in the conditioning and final cooling steps (Figure 3-1). These cooling duties combined are two orders of magnitude less than the major heating requirements and are barely visible on the composite curve. Also, they are too small to be economically recoverable. Consequently, the heating utilities dominated the "hot" curve in the present configuration (Figure 4-1).

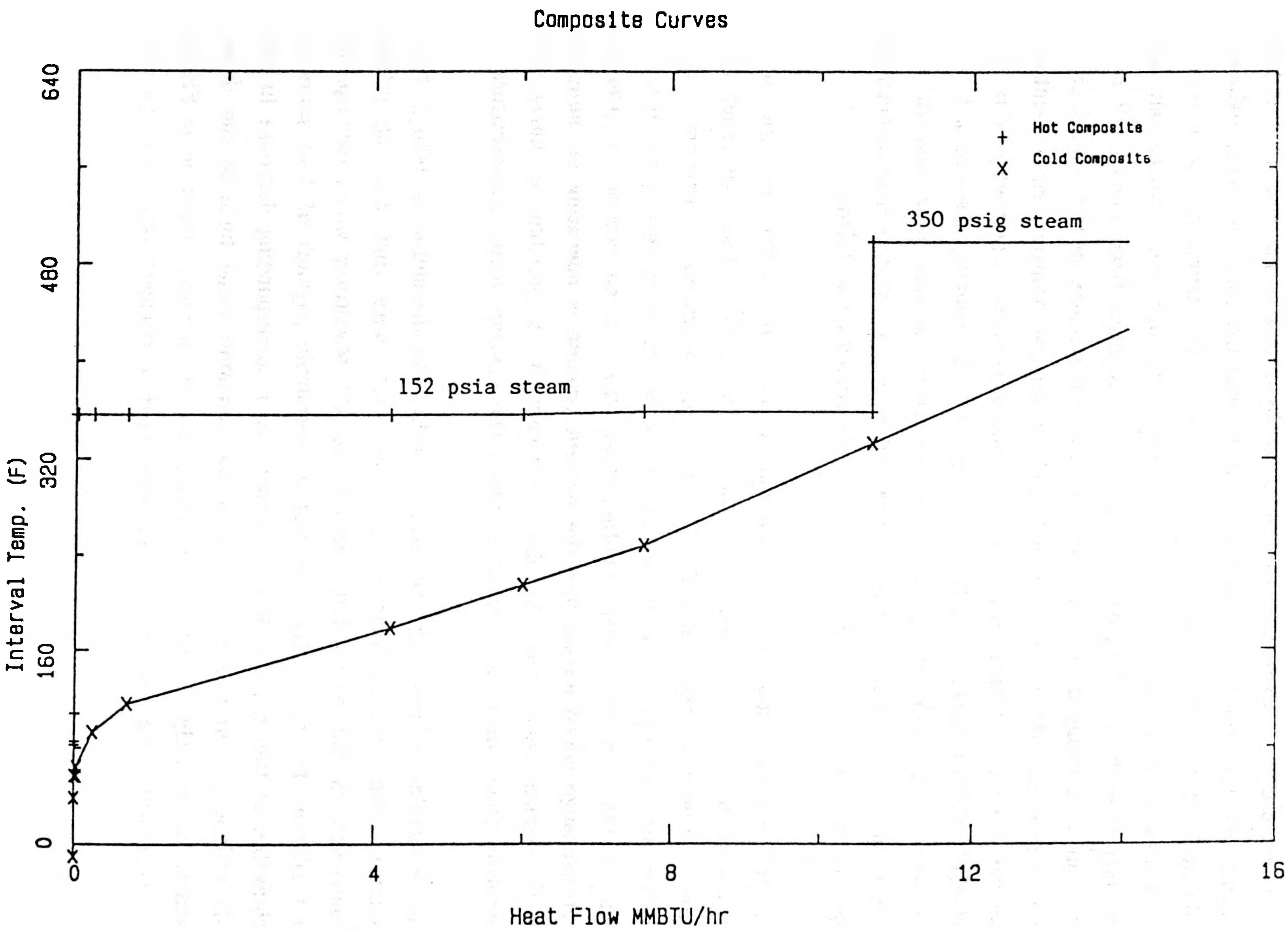


Figure 4-1 Utility and Process Composite Curves

Considerable heat is available for reclamation from condensing the water vapor in the dryer exhaust. This stream was not placed on the composite curve because it is exhausted to the atmosphere and is not one of the hot streams requiring cooling. However, if it is assumed that the dryer exhaust could be cooled to 95°F thereby extracting heat, the composite curve would be changed to that shown in Figure 4-2 where the hot dryer exhaust stream has been labeled as a "pseudo stream". It is apparent from Figure 4-2 that heat can be exchanged between the hot and cold streams in the shaded area. By reducing the DTmin, the hot and cold curves are brought closer together and the shaded area becomes larger. Also, and more importantly, the hot and cold utilities become smaller. The penalty for reducing the hot and cold utilities is the need for greater heat exchange surface area due to the greater amount of heat exchanged and the reduction of the heat exchanger approach temperature which is related to the reduction in DTmin.

The relationship between the reduction in the hot utility and the heat exchanger surface area required is shown in Figure 4-3. The area under this curve represents the infeasible region of operating conditions, i.e., insufficient exchanger area and/or energy are available to satisfy the process requirements. In the present configuration, there is no process to process heat exchange which means that the present process is operating at point A. By recovering heat from the dryer exhaust, it is possible to move the operating point down the curve and reduce the heating utility requirements.

The economics of reducing the heating utility requirements by adding heat exchanger area are summarized in Table 4-1. Note that the hot utility requirement is reduced until it stabilizes at 12.5 MMBtu/hr which corresponds to a DTmin of 12°F. Also note that a reasonable payback of 1.44 years is achievable at this point. Of course there is a corresponding decrease in the cold utility, but in this case it is of no economic value because the dryer exhaust is presently vented to the atmosphere at no cost. Point B on Figure 4-3 represents the relationship between heat exchanger area and heating

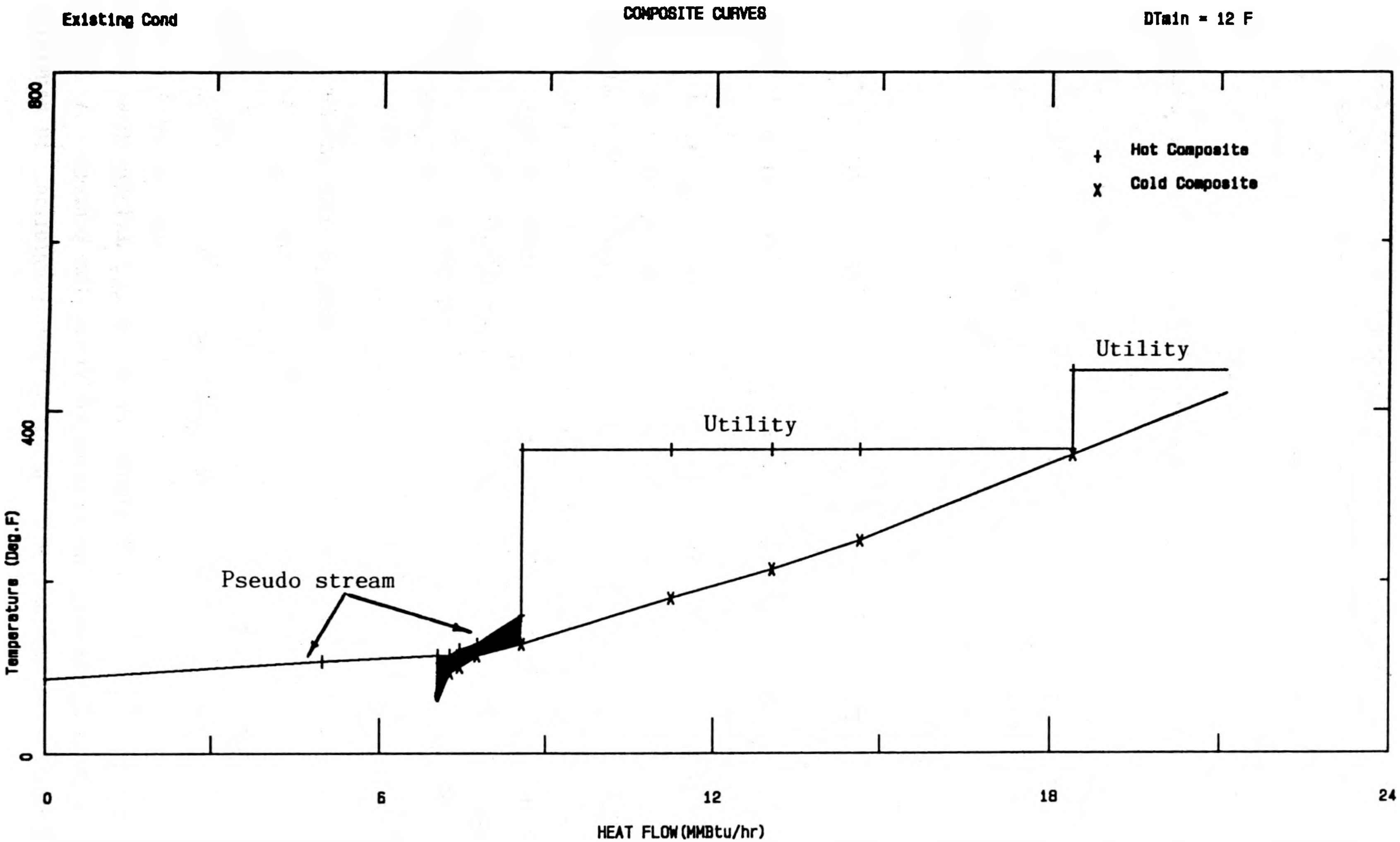


Figure 4-2 Utility and Process Composite Curves with Pseudo Process Stream (DTmin = 12 F)

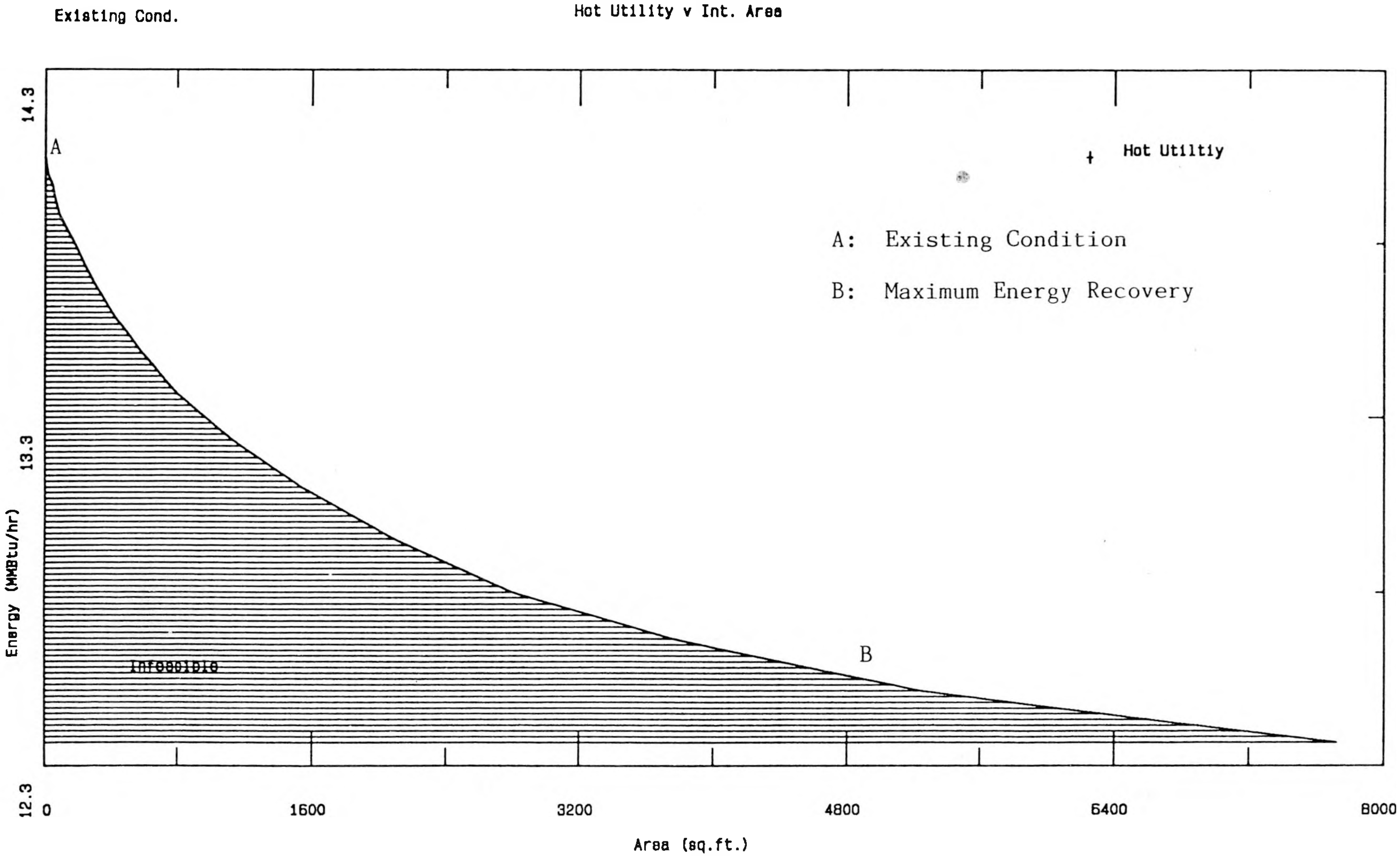


Figure 4-3 Hot Utility Vs. Heat Exchanger Area Curve

Table 4-1

ECONOMIC COMPARISON AT DIFFERENT HEAT INTEGRATION LEVEL FOR THE EXISTING CONFIGURATION

DTmin (°F)	Q_h (MMBtu/Hr)	Q_c (MMBtu/Hr)	ΔQ (MMBtu/Hr)	Exchanger Area (ft^2)	Investment ($)	Savings ($)	Payback (Yr)
100	14.1	8.59	0	0	0	0	0
80	13.9	8.48	0.2	70.3	703	4.320	0.2
50	13.5	8.06	0.6	545	5,450	12,960	0.42
30	13.0	7.58	1.1	1770	17,700	23,760	0.74
20	12.8	7.30	1.3	3104	31,040	28,080	1.11
15	12.6	7.17	1.5	4148	41,480	32,400	1.28
12	12.5	7.08	1.6	4992	49,920	34,560	1.44
10	12.5	7.03	1.6	5699	56,990	34,560	1.65

Q_h is the total hot utility requirement

Q_c is the total cold utility requirement

ΔQ is the reduction in hot utility

utility requirements corresponding to a DTmin of 12°F. A target energy savings of 1.6 MMBtu/hr is indicated by Figure 4-3; however, only part of this targeted savings can be realized because the temperature requirements for heat in the primary, secondary and final dryers do match the characteristics of the heat from the dryer exhaust stream. Therefore, although a simple payback of less than two years can be achieved, the effect on the heating utility is not as great as is desired.

4.2 HEAT PUMP

To evaluate the potential for heat pumps, it is useful to create a "Grand Composite Curve" or GCC for the process.

The GCC for maximum energy recovery with the pseudo stream is shown in Figure 4-4. The GCC gives a "profile" of the energy requirements of a process. It also provides important information on the heat pump potential present in a process.

To be effective, a heat pump must be "appropriately" placed. In the context of pinch technology, this means that the heat pump must move thermal energy from below the "pinch" temperature, where there is a surplus of thermal energy, to above the pinch where there is a need for thermal energy. Another requirement is that the temperature lift dictated by the structure of the GCC must be attainable by available compressors. Stated another way, this criteria demands that the GCC exhibit a long narrow nose at the pinch temperature, with a large heat source below the pinch and a large heat sink above the pinch. The GCC for this vegetable drying process exhibits the desired characteristics for heat pumping and two potential heat pump configurations will now be examined.

4.2.1 Semi-Open Cycle Heat Pump

A semi-open cycle heat pump uses a process stream, usually a condensible vapor below the pinch, as the working fluid. By compressing this condensible vapor to a higher pressure to raise its condensing temperature, this low

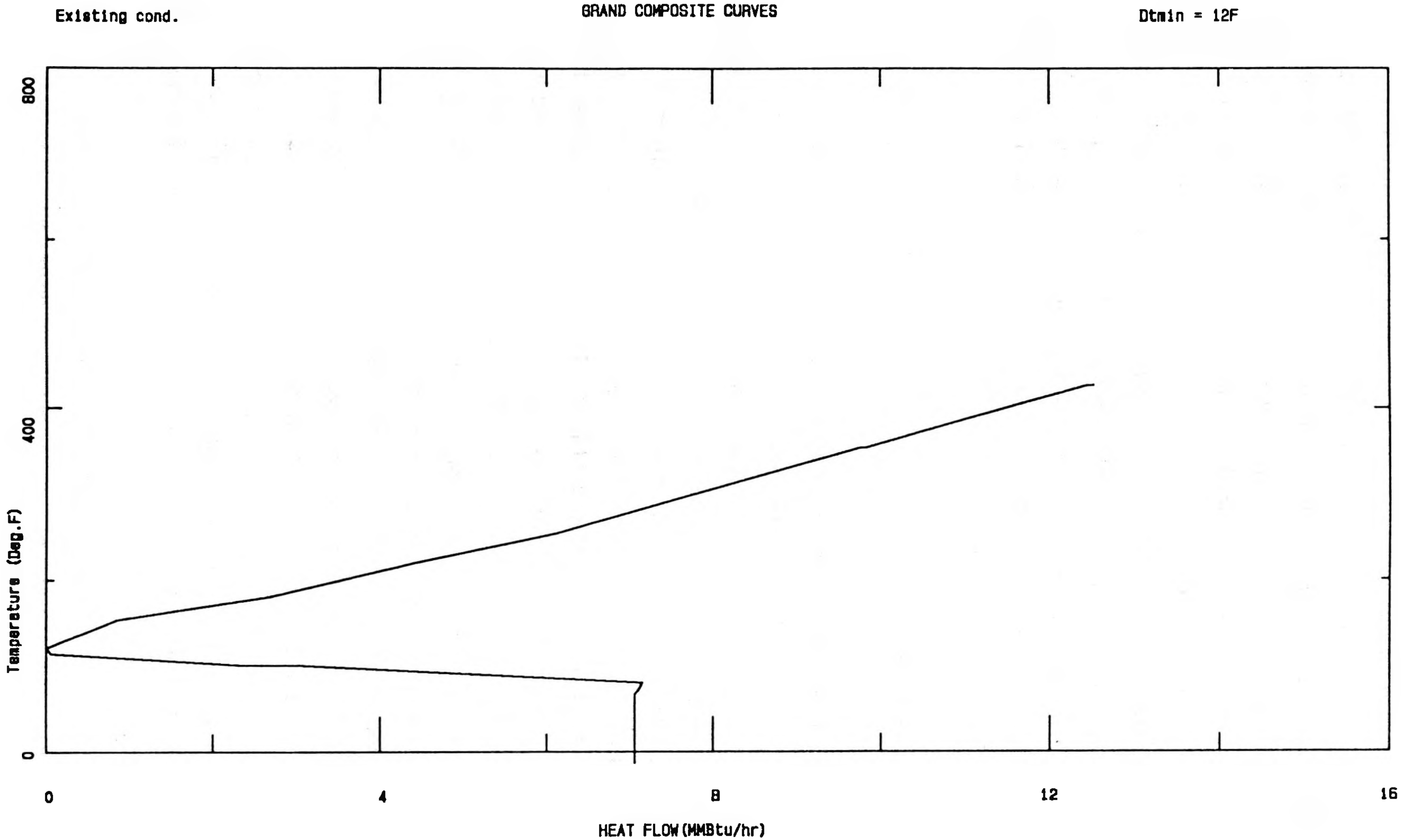


Figure 4-4 GCC of The Existing Configuration at DTmin = 12 F

temperature heat source is upgraded to match a process heat requirement above the pinch.

At a DTmin of 12°F, there are two streams below the pinch, the product stream and the exhaust air stream. However, the product stream is of dehydrated material and has a very small duty associated with it. Therefore, it is not suitable for use as a working fluid or as a heat source. The exhaust air stream has about 6.5 MMBtu/hr of heat load. Most of this heat is latent heat associated with the water vapor in the exhaust air. However, to recover this latent heat load through a semi-open cycle heat pump, a very large compressor must be employed due to the presence of the air. A semi-open cycle heat pump using the dryer exhaust air as the working fluid is, therefore, not considered economically feasible.

4.2.2 Closed Cycle Heat Pumps

A closed cycle heat pump employes a working fluid as the heat transfer medium. It requires both an evaporator, where the working fluid absorbs heat from a heat source to be evaporated, and a condenser, where the compressed working fluid releases heat to a heat sink. The delivery temperature in the condenser of a closed cycle heat pump is limited by the refrigerant used. With R-12 as working fluid, the practical delivery temperature is about 160°F. With R-114, the heat pump can deliver heat at about 230°F . This enforces a restriction on the usage of the recovered heat in the evaporator. If a 10°F LMTD (Logarithmic Mean Temperature Difference) in the condenser is allowed, the recovered heat from the exhaust air can be used to heat the dryer inlet air to 220°F. However, the heat integration option at DTmin of 12°F can take the inlet air up to about 141°F, leaving the heat pump option to cover the difference between 141°F and 220°F. This amounts to only 2.93 MMBtu/hr of heat to be supplied by the heat pump which includes the electricity input. This combined heat integration and heat pump system can payback in about 3 years. An alternative configuration is presented next which has many benefits over this configuration.

4.3 ALTERNATIVE CONFIGURATION

During the course of the study, it became apparent that a lower drying temperature would allow more heat recovery and at the same time provide an improvement in product quality. An alternative configuration was defined which made both these goals achievable. This configuration is based on experience in lumber drying.

To air dry any product requires the following: (1) The relative humidity of the drying air must be lowered sufficiently to allow the air to accept the moisture from the product, and (2) the drying air must have sufficient enthalpy to evaporate the moisture. The first requirement relates strongly to the drying rate. In general, the lower the relative humidity of the drying air, the faster the product will dry. The average drying temperature also has an effect because the higher the temperature for a given relative humidity, the faster heat is transferred to the water and the higher the drying rate. The second requirement relates to the combination of air flow volume and temperature drop in the dryer. The sensible heat given up by the temperature drop of the drying air must be sufficient to vaporize the water in the product being dried.

Application of heat pump systems to lumber drying has recently become popular in the U.S.. The main attraction to the lumber industry is the possibility of lower drying temperatures for enhanced lumber quality. The main reason that heat pumps work so well is the low drying rates required which translate to lower heat pump temperature lift requirements. In conventional lumber drying, the first drying requirement is met by heating drying air to lower the relative humidity so that the air can remove water from the lumber. Closed cycle heat pumps can cost effectively supply low humidity air at a lower temperature because they can remove moisture from drying air by condensing the water against the heat pump evaporator then reheating the dry air against the heat pump condenser. In this way, the required low relative humidity is achieved at a lower drying air temperature. At the same time, the heat of vaporization of the water is recovered in the heat pump evaporator. The lower drying air temperature made possible by this arrangement makes the temperature lift requirement of the heat pump economically attainable. The second requirement is met by increasing the air flow over the lumber to compensate for the lower temperature.

Vegetable drying differs from lumber drying in a number of aspects. Chief among them, from an energy standpoint, is the high drying rates used in vegetable drying relative to lumber drying. (These higher rates may be due to convention rather than need.) However, lumber and vegetable drying are similar in that the quality of both products can be enhanced by lower drying temperatures. This analysis attempted to reach a cost effective compromise between the desire for lower drying temperatures and the desire to maintain an acceptable drying rate.

In this particular plant, over 80% of the plant hot utility is consumed in the primary dryer where 350 psig steam and fuel gas are used to supply the heat. The temperature of the air entering the dryer is at 400°F or higher, and a lower temperature operation may improve the quality of the product. Also recovering some of the heat of vaporization with a heat pump will have a significant effect on the plant hot utility requirements, as will be shown.

An analysis was undertaken to determine the best procedure for the vegetable drying plant under study. The existing dryer was not to be replaced. However, a new dryer inlet and outlet arrangement was found which not only reduced the dryer temperature by about 100°F, while maintaining the same drying rate, but which also increased the benefits from using a heat pump.

The same targeting procedure for determining the potential for heat integration and heat pumping was followed for the new configuration as was used for the original configuration (Section 4.1 and 4.2). Table 4-2 summarizes the results for heat integration targeting. It shows, as in the original configuration, that heat integration should be increased to DTmin of about 12°F.

Table 4-2

ECONOMIC COMPARISON AT DIFFERENT HEAT INTEGRATION LEVEL FOR THE NEW CONFIGURATION

DTmin (°F)	Q_h (MMBtu/Hr)	Q_c (MMBtu/Hr)	ΔQ (MMBtu/Hr)	Exchanger Area (ft^2)	Investment ($)	Savings ($)	Payback (Yr)
80	15.1	6.31	0	0	0	0	0
50	14.8	6.09	0.3	193	1,930	6,480	0.30
30	13.7	4.93	1.4	2637	26,370	30,240	0.87
20	13.1	4.35	2.0	5279	52,790	43,200	1.22
15	12.8	4.06	2.3	7571	75,710	49,680	1.52
12	12.6	3.89	2.5	9641	96,410	54,000	1.79

Q_h is the total hot utility requirement

Q_c is the total cold utility requirement

ΔQ is the reduction in hot utility

Figure 4-5 shows the GCC of the new configuration at a DTmin of 12°F. The original and new configurations can be readily compared on the basis of the GCC because the GCC depicts the hot and cold energy profile of a system. Figure 4-6 shows the GCC's for the existing and new configurations on the same scale at a DTmin of 12°F. Note that the new configuration has the effect of lowering the overall temperature profile of the system. This is reflected in the GCC by lower temperature (about 100°F) at the hot end of the GCC. Moreover, the "nose" on the GCC has been shortened somewhat which means that the heat pump can economically recover more of the exhaust heat in the new configuration than in the original configuration.

A heat integration and heat pump design based on the new configuration is shown in Figure 4-7. This design first recovers heat through heat exchange at the lowest practical DTmin and then utilizes two heat pumps to recover additional heat. A total heat savings of 6.75 MMBtu/hr of energy is reclaimed through this system. The design is based on the minimization of compressor operating costs and limiting the flow of drying air to accommodate the existing dryer. The first heat pump system requires 104 KW of power and delivers 1.84 MMBtu/hr, and the second one requires 219 KW and delivers 2.75 MMBtu/hr. The COP's are 5.2 and 3.7 respectively. A $160,000 capital cost was estimated by Nyle Corporation for the system modification.

It may be possible to reduce the temperature further by passing more air through the dryer. This would make the heat pump more cost effective by reducing the temperature lift requirement. However, this could increase the required residence time of the vegetable in the dryer and adversely affect quality. Also, a new larger volume dryer would be required which is beyond the scope of this project.

The economics for this heat pump system is greatly influenced by the fuel and electricity costs. A sensitivity study of the heat pump economics is shown in Figures 4-8 and 4-9. With a fuel cost of $3/MMBtu and electricity cost of $0.03/kWh, the simple payback is 2.1 years. This is on the basis of fuel savings minus operating costs alone and ignores quality improvement.

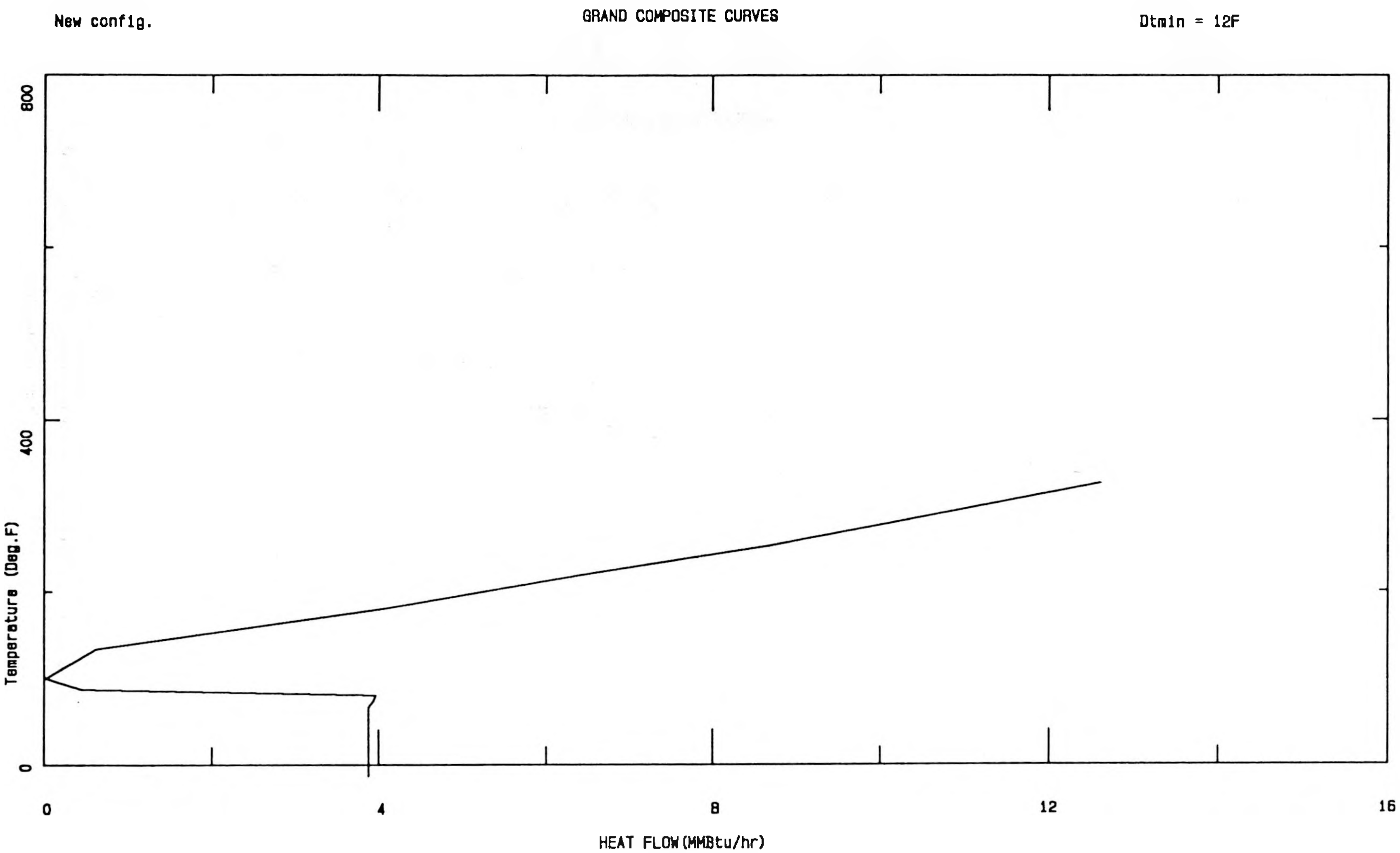


Figure 4-5 GCC of The New Configuration at DTmin = 12 F

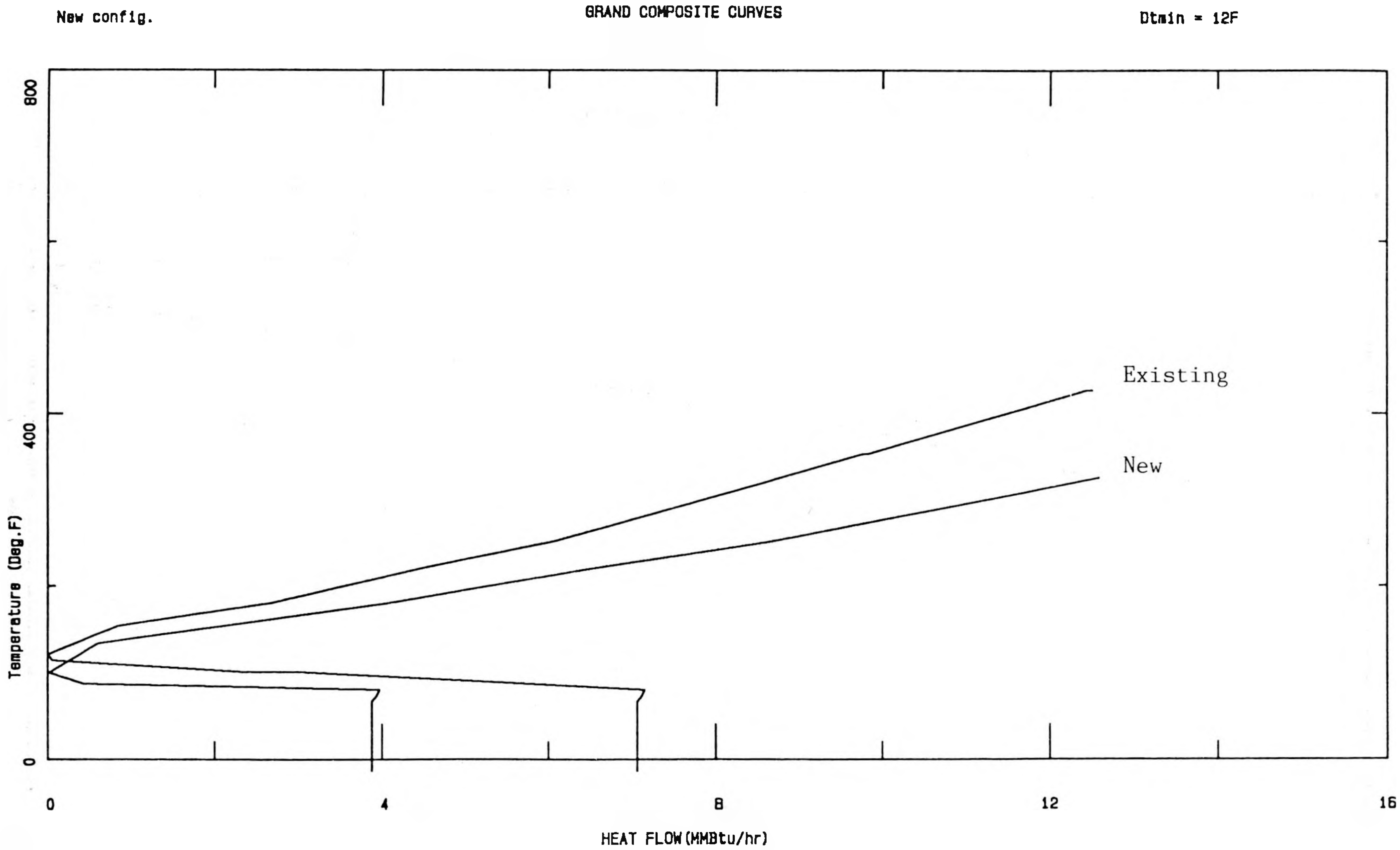
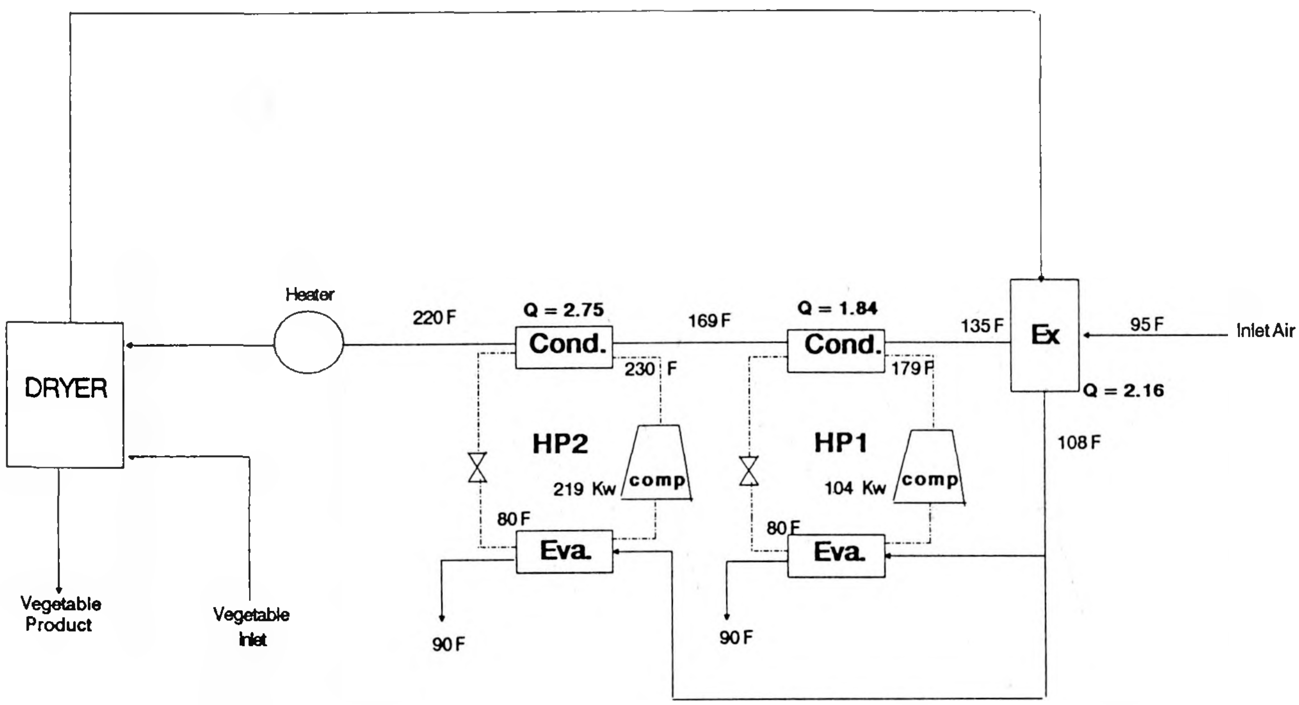


Figure 4-6 Comparison of GCC's for Existing and New Configurations (DTmin = 12F)



Q = Heat Load in MMBtu/Hr

Figure 4-7 Heat Pumps Design Based on Proposed New Dryer Configuration

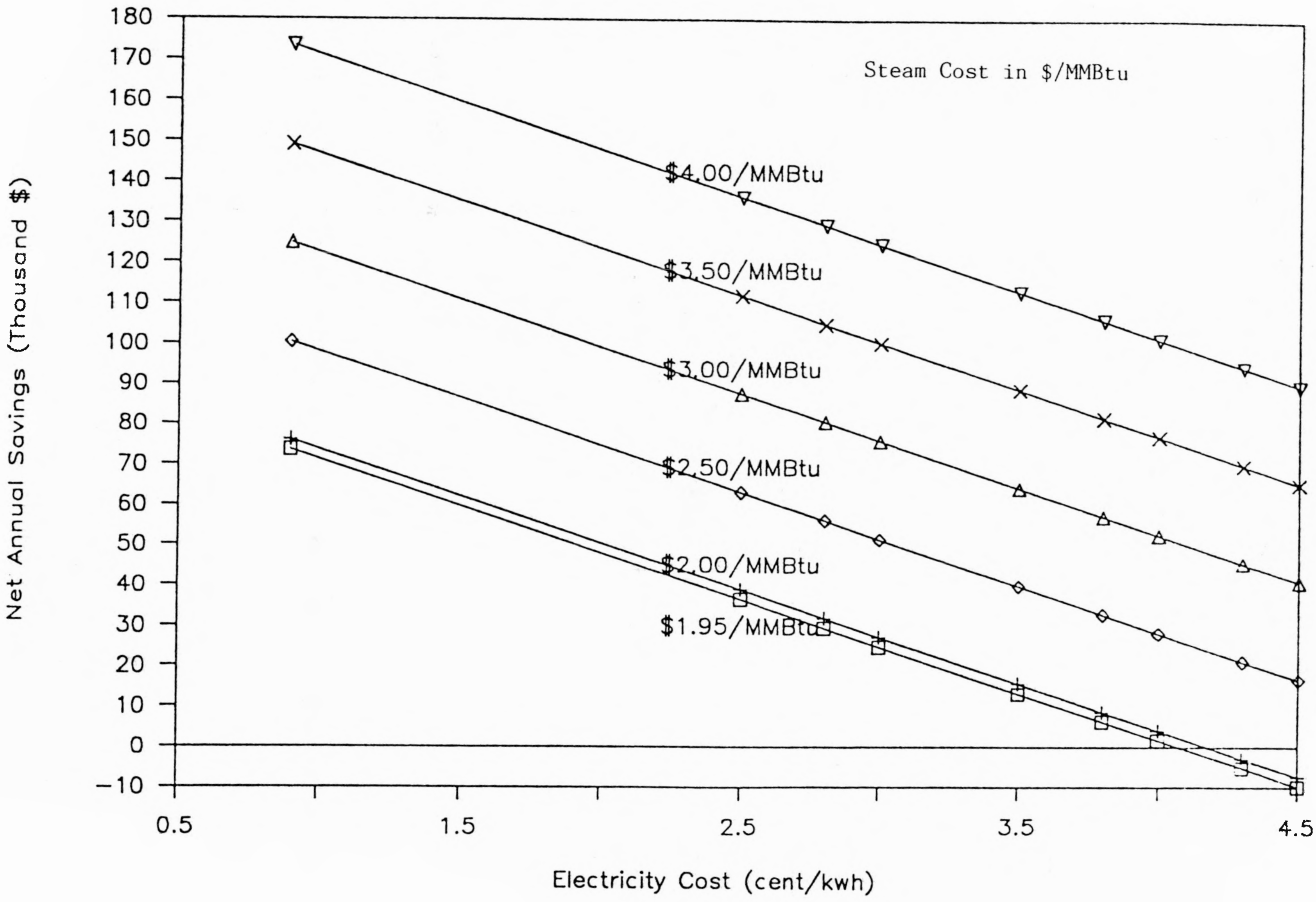


Figure 4-8 Closed Cycle Heat Pump Economics

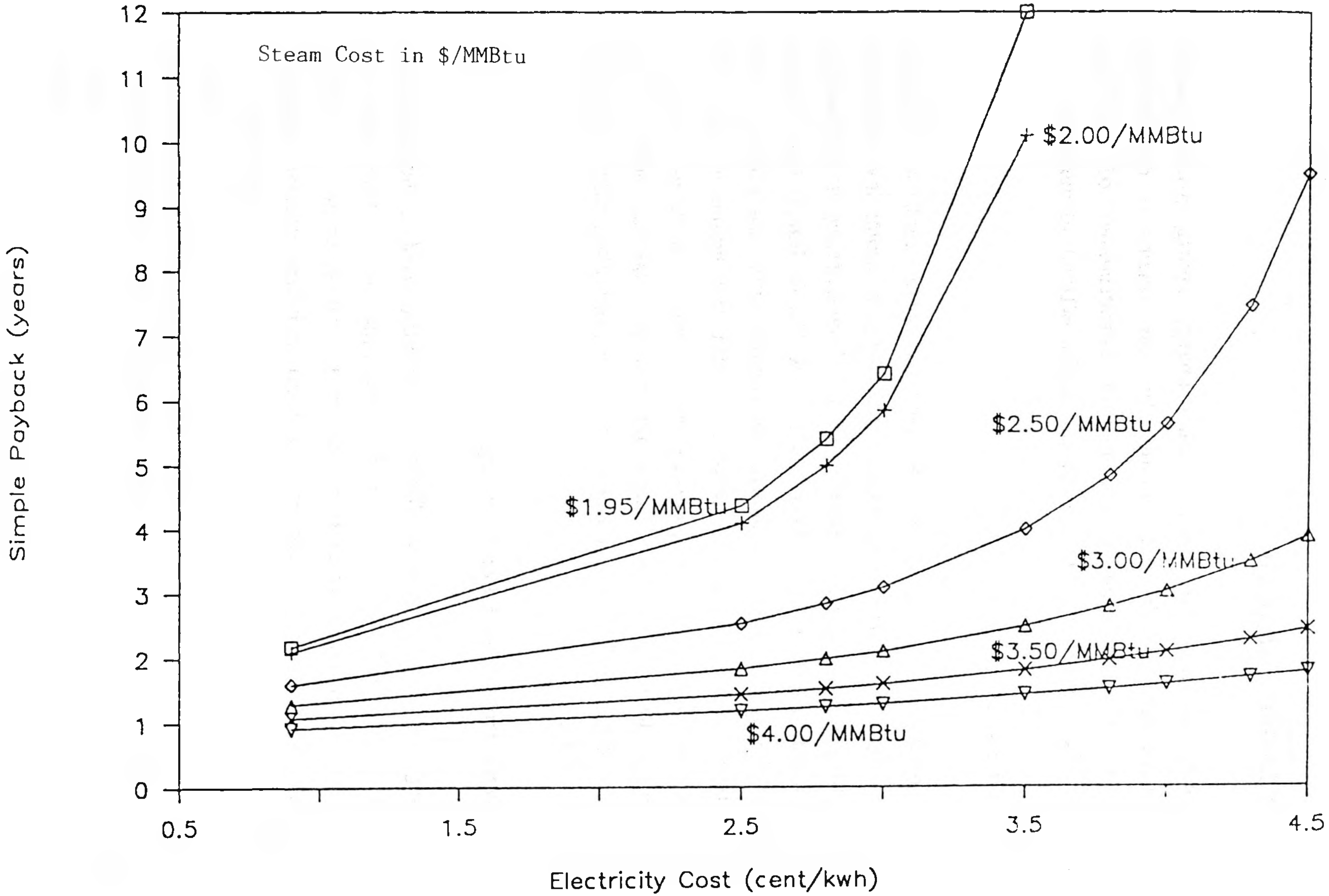


Figure 4-9 Closed Cycle Heat Pump Economics

4.4 OTHER SAVING POTENTIAL

During the course of this study, several other energy savings projects were identified. Although not directly related to the vegetable dehydration process, some of these will add to the overall attractiveness of the heat pump system. These potentials are presented briefly without elaboration.

(1) More Electricity Generation

With the proposed new configuration, the primary dryer inlet temperature drops by over 100°F. At that temperature, the 152 psia steam level will be sufficient to provide the necessary heating duty. The existing steam heater surface area, which is designed for bigger duty and LMTD (Logarithmic Mean Temperature Difference), should be enough for smaller duty and LMTD. The client can either reduce the steam generated from the boiler or use the additional high pressure steam to generate more power. With current low fuel cost, it seems that more power generation will be a better choice. The additional power generated can help reduce the averaged electricity cost for the heat pump system.

(2) Heat Recovery from The Boiler Flue Gas

Flue gas coming out of the boiler at 650°F is currently used to preheat the combustion air from 110°F to about 380°F. Flue gas leaves the stack at about 450°F, wasting a large amount of energy to the ambient. To fully utilize this high temperature heat source, a hot oil coil heat recovery system is suggested which runs through the flue gas prior to the air preheating to pick up about 6 MMBtu/hr of heat for the primary dryer. This would reduce the flue gas available at 450°F for preheating the combustion air because the flue gas temperature out of the air preheater will be only 250°F. Additional air preheater surface area might be required to perform the same heating duty. This arrangement could totally replace the remaining steam consumption in the primary dryer.

Section 5

CONCLUSION AND RECOMMENDATION

The advantages for the closed cycle heat pump system based on the new configuration on primary dryer can be summarized as follows:

- It reduces the dryer inlet temperature by 100°F.
- It improves the quality of the vegetable product.
- It reduces about 50% of the subsequent cooling duty at the end of the process.
- The fuel gas and high pressure steam heating duty can be replaced by 152 psia steam.
- The flexibility of utility system is enhanced. The client will have additional high pressure steam to generate power or to simply reduce the boiler output.
- The primary dryer heating duty can be totally replaced by the combination of heat pump system and heat recovery system in the boiler.
- The heat pump system gives the following economics: (based on a typical price of $3.00/MMBtu for fuel and 3 c/kWh for electricity)

Total Hot Utility Savings:	$146,000/yr
Electricity Cost:	$69,750/yr
Net Annual Savings:	$76,250/yr
Total Investment:	$160,000
Payback:	2.1 years

The implementation of the heat pump system and the heat recovery system are independent of each other. This study also confirmed an earlier study performed by TENSA Services concluding that Food drying process has a very good potential for heat pumping. Vegetables dehydration through similar processes amount to 280 MM lb/year. Based on this study and the national vegetable drying capacity, a total savings of 0.65 trillion Btu per year through heat pump replication can be realized.

APPENDIX A

APPENDIX A

BASIC PRINCIPLES AND APPLICATIONS OF PINCH TECHNOLOGY

This appendix provides an introduction to the basic concepts and terminology associated with "pinch technology". It also demonstrates the usefulness of pinch-based methods for industrial heat pump and heat engine placement. This is intended for readers unfamiliar with these technologies, to provide the necessary background for a general understanding of the main sections of this report.

A bibliography is attached to this appendix. It highlights recent articles that present a more detailed discussion of pinch technology and its application to heat pump placement and related subjects, such as:

- Overall Energy Efficiency (1),
- The Design of Heat Exchanger Networks (HEN's) (2),
- Integration of Heat and Power Systems with Chemical Processes (3),
- Heat Integration of Distillation Systems (4), and
- "Appropriate Placement" of Heat Pumps in Chemical Processes (5,6,7,8).

A.1 PROCESS HEATING AND COOLING

Within most processes in the chemical and allied industries, there are streams that require heating and streams that requires cooling. Any stream that requires heating is conventionally said to be "cold" (irrespective to its temperature level), and streams that require cooling are said to be "hot".

The heating and cooling duties within a process can be provided by "utilities" such as steam (for heating) and cooling water (for cooling) that are available on the site. However, the load on these external utilities can often be reduced by "heat integration" of the process - that is, by transferring heat from hot process streams to cold process streams by means of heat exchanger networks (HEN's). This is illustrated in Figure A-1. As the only heating and cooling duties that incur direct operating costs are those associated with utilities, heat integration generally leads to a reduction in process operating costs.

A.2 THE HEAT TRANSFER PINCH

In most processes, no matter how thoroughly they are heat integrated, there will always be a residual heating duty (Q_H) and a residual cooling duty (Q_C) that have to be met by utility heating and cooling. The size of these residuals can be reduced by increasing the heat transfer area within the process heat exchangers. This essentially allows smaller temperature differences between the matched hot and cold streams. In general, however, the residual utility loads would be finite even if the heat transfer area were infinite.

If both utility heating and cooling are required, the process may be considered to be made up of two parts:

- A higher temperature part which, after complete heat integration, acts as a net heat "sink" or acceptor.
- A lower temperature region which, after complete heat integration, has surplus heat to be rejected. It is thus a net heat "source".

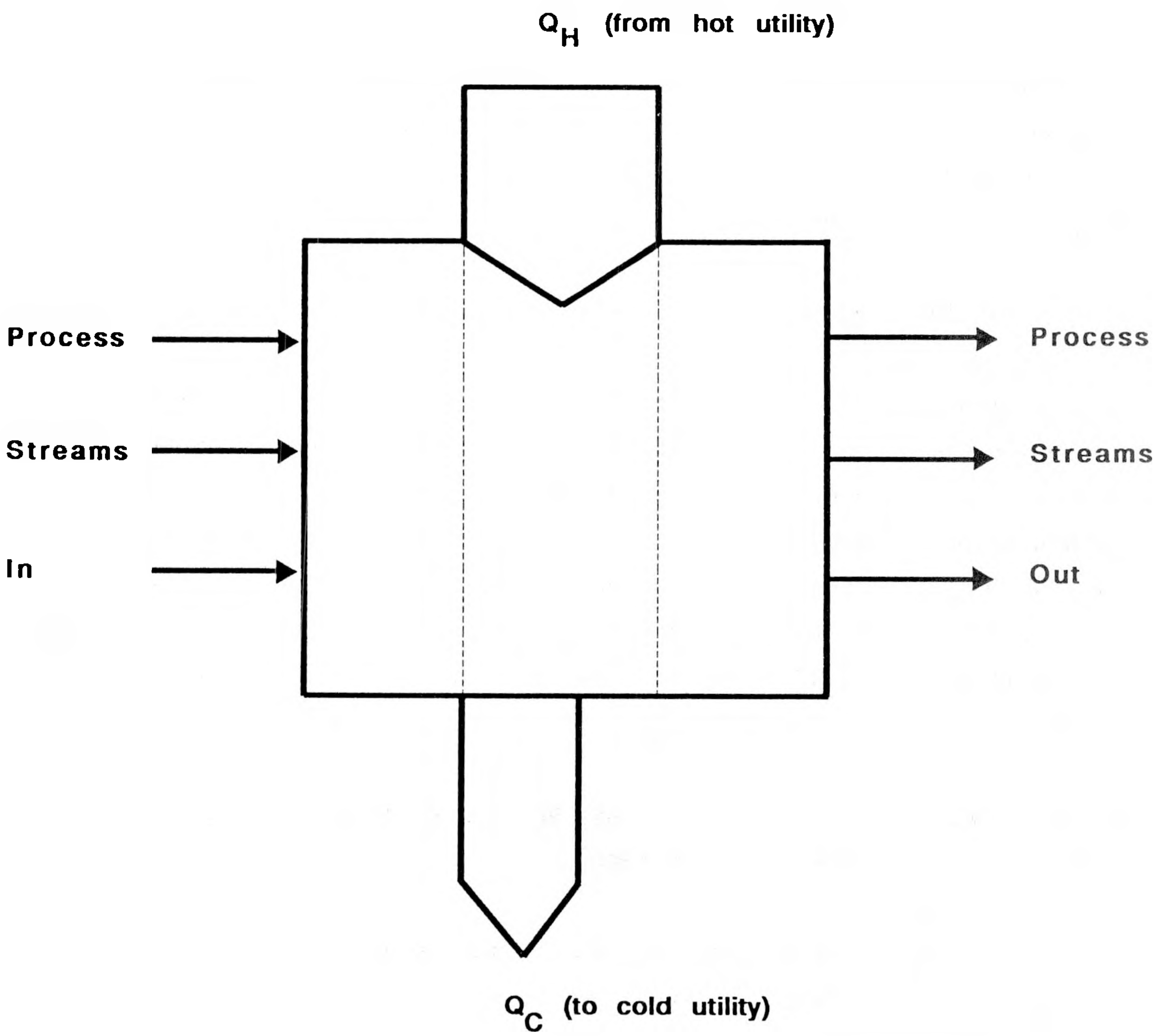


Figure A-1: Heat Transfer Between Process Streams Using A Heat Exchange Network

The temperature that separates the source and sink sections of the process is called the heat transfer "pinch" (Figure A-2). In a properly integrated process, there is no heat transfer from above the pinch to below the pinch. Also, the "temperature driving force" (i.e., the difference in temperature between the hot and cold streams) reaches its minimum value, designated DT_{min}, in the region of the pinch.

When a pinch design is implemented, the hot and cold utility requirements reach their minimum values, Q_{Hmin}, Q_{Cmin}, appropriate to the selected value of DT_{min}.

A few processes do not have heat transfer pinches. These require either only heating or only cooling from external utilities, but not both. Such processes are said to exhibit "threshold" characteristics.

A.3 HEAT PUMPS

Heat pumps provide a means of upgrading heat (i.e. raising its temperature) by the input of work. They may therefore be regarded as heat engines running in reverse. The principle behind the heat pump is illustrated in Figure A-3, where an ideal heat pump extracts an amount of heat Q from a temperature T_1 and elevates it to the temperature T_2 by the input of reversible work W_{REV}. The best known real heat pumps are reverse Rankine cycles. Low pressure vapor generated at some source temperature, T_s, is compressed to a higher pressure at which it condenses, releasing its heat at a higher target temperature T_t.

The work input for such a system is generally provided by mechanical compression with the system operating in a closed cycle (i.e. the working fluid repeatedly passes through evaporation, compression and condensation stages). This is depicted in Figure A-4.

Sometimes, it is possible to use a process vapor stream as the working fluid in heat pumps. These heat pumps are called semi-open cycle heat pumps. The most common semi-open cycle (type 1), is called the mechanical

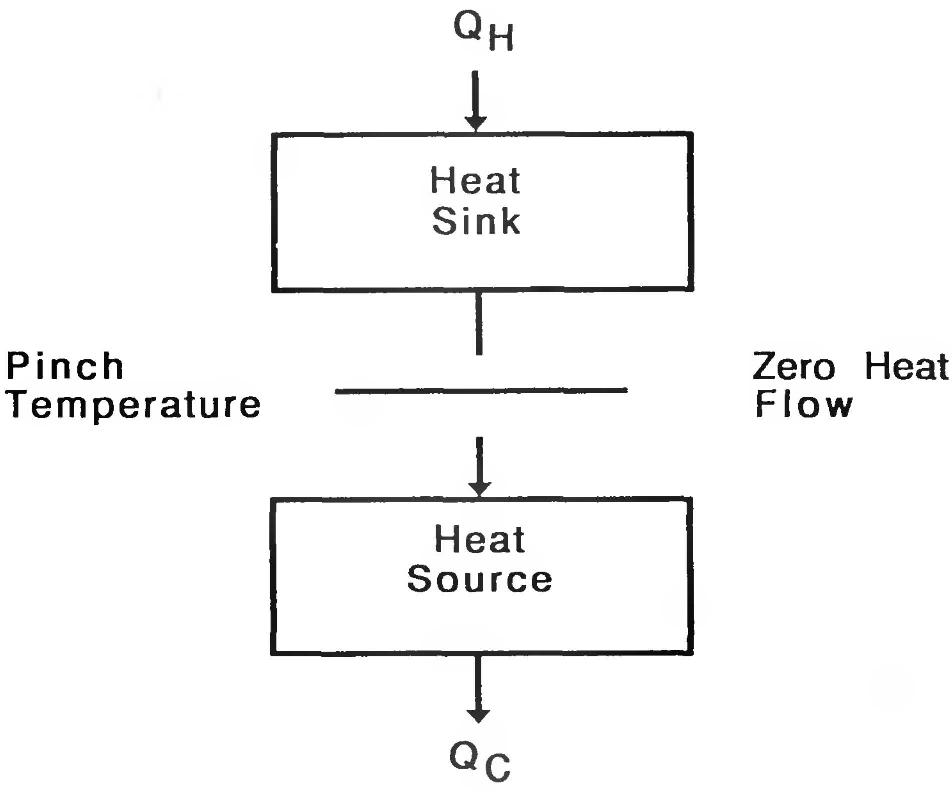


Figure A-2: The Heat Transfer Pinch

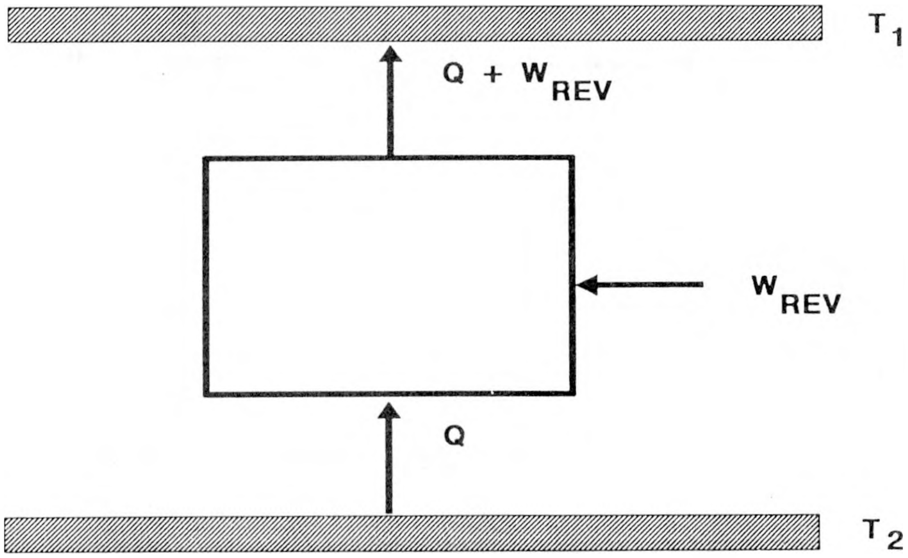


Figure A-3: Basic Principles of the Heat Pump

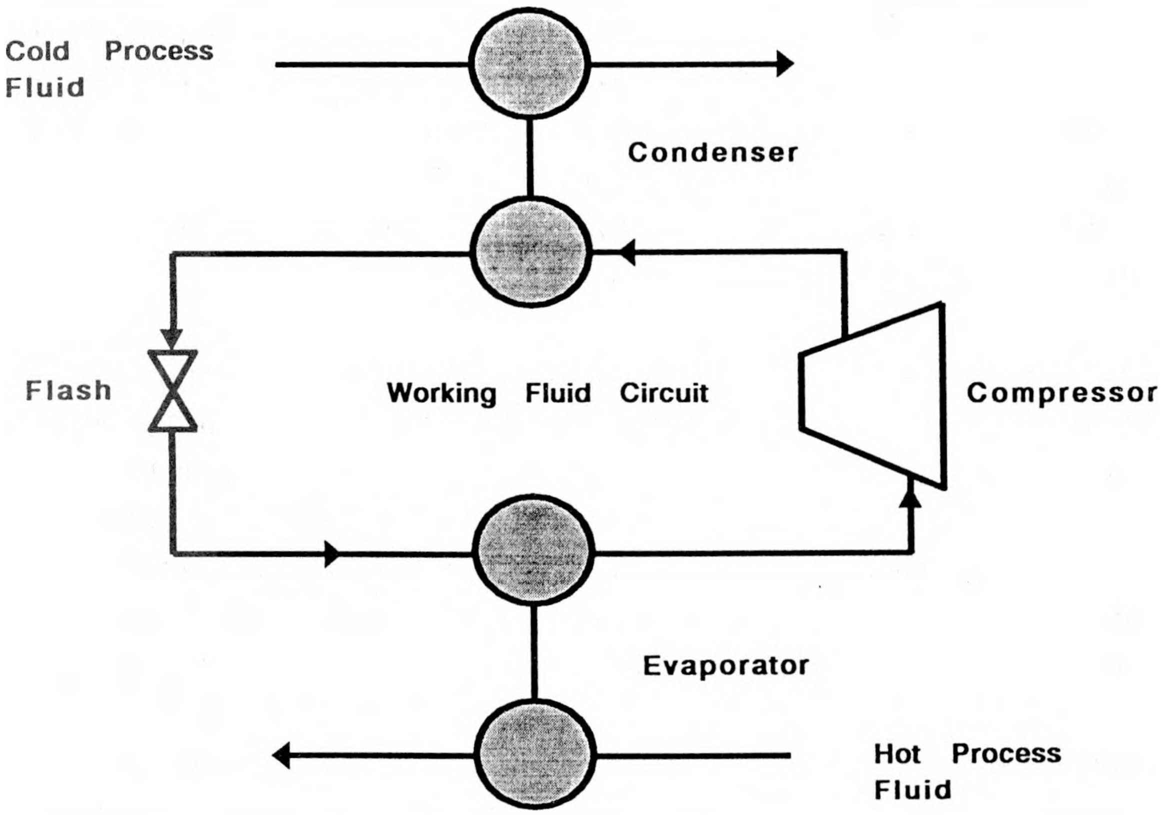


Figure A-4: Closed Cycle Heat Pump

vapor recompression (MVR) heat pump. The hot process vapors are compressed in a compressor and then condensed in the heat pump condenser to satisfy a process heating requirement at an elevated temperature (see Figure A-5). A less common type of semi-open cycle heat pump (Type 2) has the opposite configuration, i.e., an evaporator instead of a condenser. A liquid stream is vaporized in the evaporator and then compressed to a higher temperature in a compressor (see Figure A-6). This type of heat pump cycle is recommended when a low temperature heat source is available to evaporate a liquid process stream which is required in the vapor phase at a higher temperature. It is important to note that semi-open cycles are only feasible when the process fluid undergoes a phase change; condensation for Type 1 systems, and evaporation for Type 2 systems.

These are the main types of heat pumps considered in this report. In addition to these, there are a number of other types of heat pumps either commercially available or under development. These include chemical heat pumps (which use exothermic and endothermic reactions as a means of upgrading heat), absorption heat pumps (which use low grade heat to drive an evaporation/condensation cycle to elevate the available "waste heat" to a useful level) and electromagnetic heat pumps.

Current state-of-the-art heat pumps tend to be limited in the operating temperature range for available working fluids. Moreover, economic considerations generally limit the practical temperature lift in heat pumps to around 60°F.

A.4 APPROPRIATE PLACEMENT OF HEAT PUMPS

The pinch concept leads to useful insights into the appropriate use of heat pumps in industrial processes. Because the below pinch region is a net heat source any heat pump must accept heat in this region if it is to reduce the external cooling requirements of the process. By a similar argument the heat pump must reject its heat to the net heat sink above the pinch to reduce the demands on external utility heating. A heat pump which satisfies these criteria is said to be "appropriately placed" (Figure A-7).

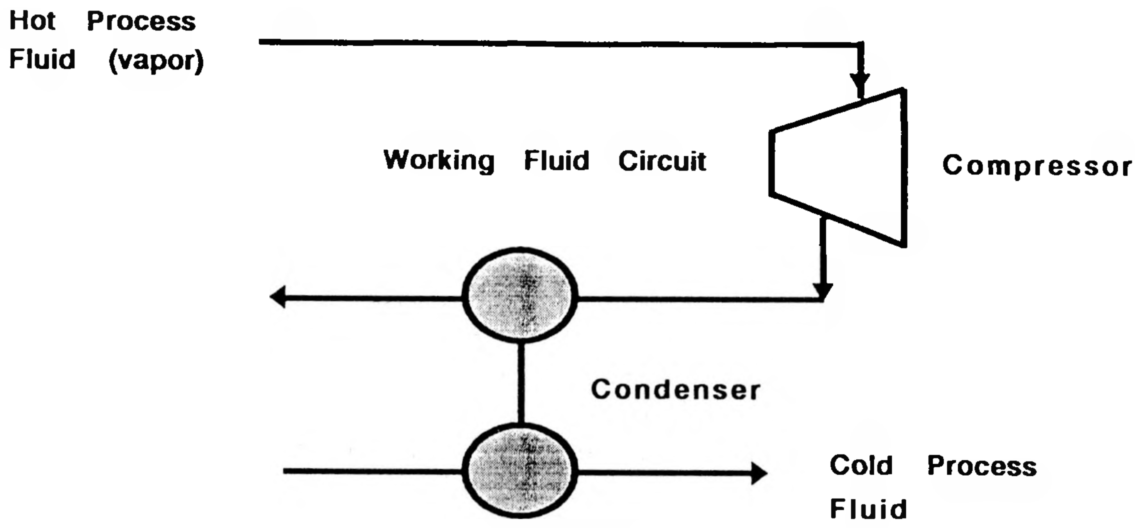


Figure A-5: Semi-Open Cycle Heat Pump (Type 1)

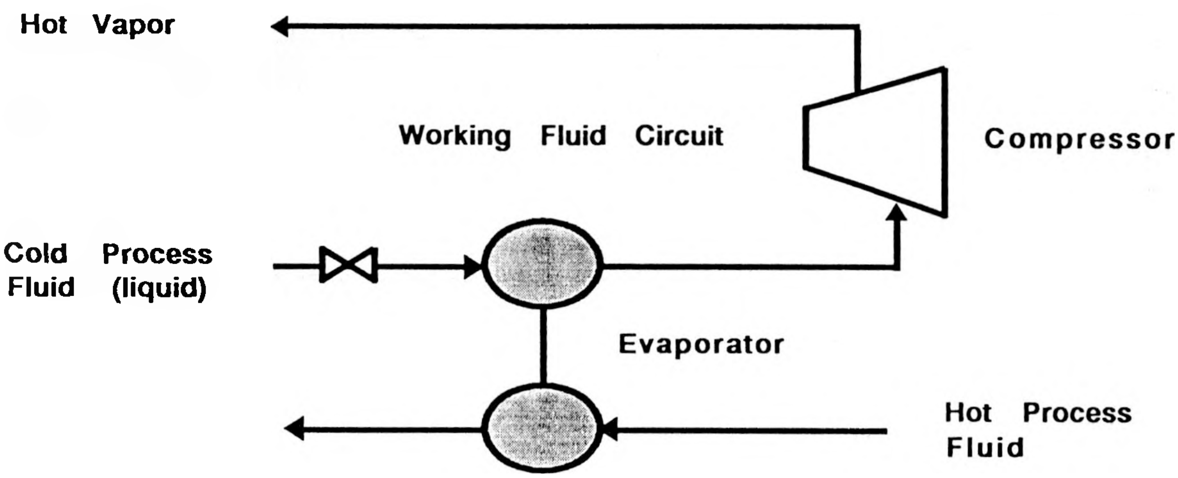


Figure A-6: Semi-Open Cycle Heat Pump (Type 2)

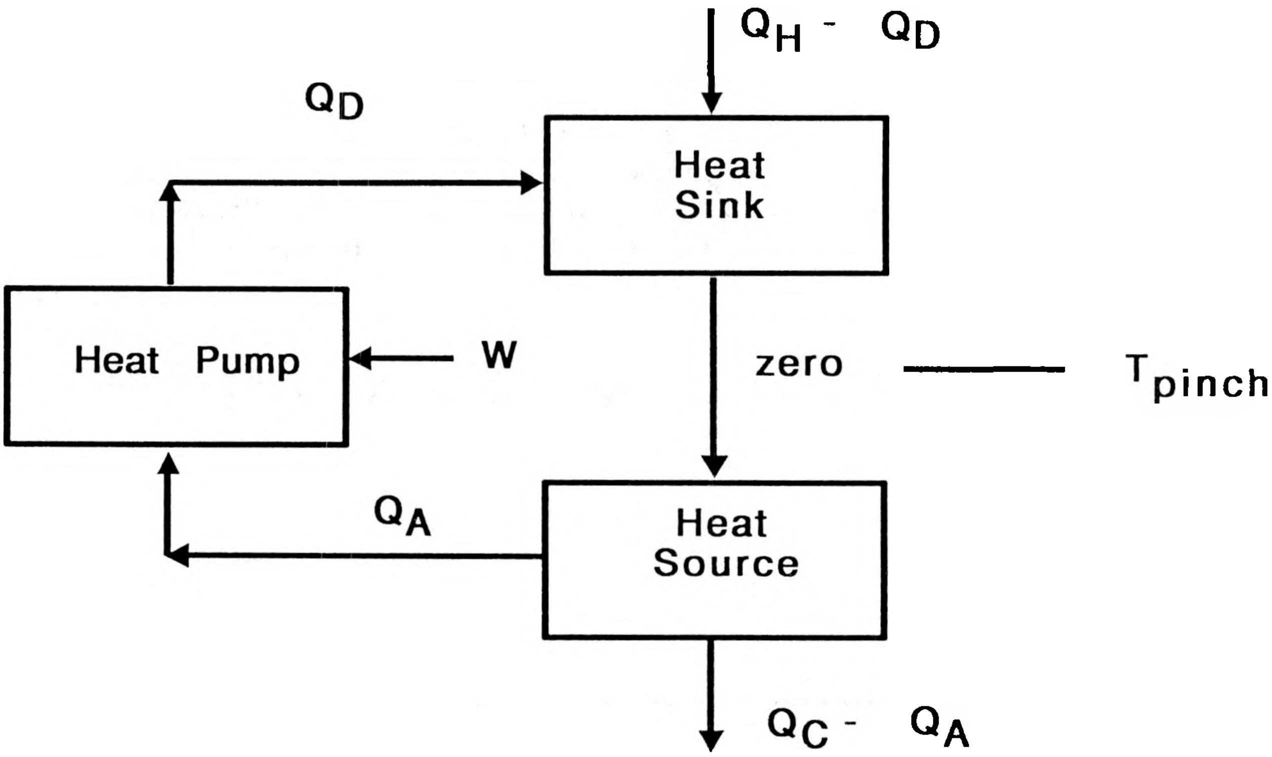


Figure A-7: Appropriate Heat Pump Integration

If a heat pump acts wholly above the pinch, it will reduce the hot utility requirement Q_H by an amount equal to the work input W of the heat pump (see Figure A-8). However, as the unit cost of providing work is normally greater than the unit cost of heating, such an arrangement is generally uneconomical. A heat pump acting entirely below the pinch (Figure A-9) has the net effect of degrading the work input W into waste heat that has to be rejected to the cold utility i.e. the net heat rejected rises from Q_C to Q_C + W, which is clearly undesirable.

Both Figures A-8 and A-9 represent "inappropriate placement" options for industrial heat pumps.

A.5 THE GRAND COMPOSITE CURVE (GCC)

As already noted, most industrial processes can be divided at a "pinch temperature" into net heat source and net heat sink regions with no heat flow at the pinch itself. However, it is possible to represent the net heat flow at every temperature level within the process by means of a "Grand Composite Curve" (GCC) or temperature enthalpy plot. An example of such a plot is given in Figure A-10.

The ordinate of the GCC is the so-called "interval temperature." This is a convention to put the hot and cold streams on a common temperature basis, after allowing for the necessary minimum temperature driving force (DT_{min}) between them. Consider the simplest case, where the heat transfer resistance associated with all the hot streams is equal to that associated with all the cold streams. The interval temperature of a hot stream at an actual temperature of T_H is defined to be T_H - ($DT_{min}/2$); and that of a cold stream at an actual temperature of T_C is T_C + ($DT_{min}/2$). Where the heat transfer resistances of the streams are different, it is necessary to ascribe an appropriate "DT_{min} contribution,"DT_{cont}, between 0 and DT_{min}, to each stream. Heat transfer between such streams is permitted only if (T_H - T_C) > DT_{min} i.e. if the interval temperature of the hot stream is greater than or equal to that of the cold stream.

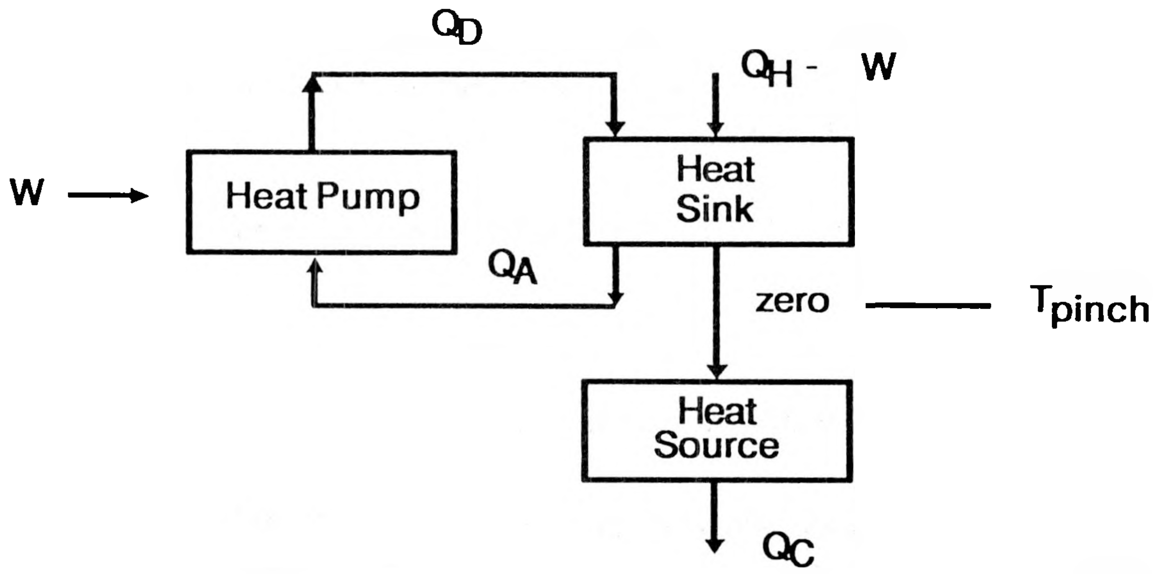


Figure A-8: Inappropriate Heat Pump Integration-above pinch

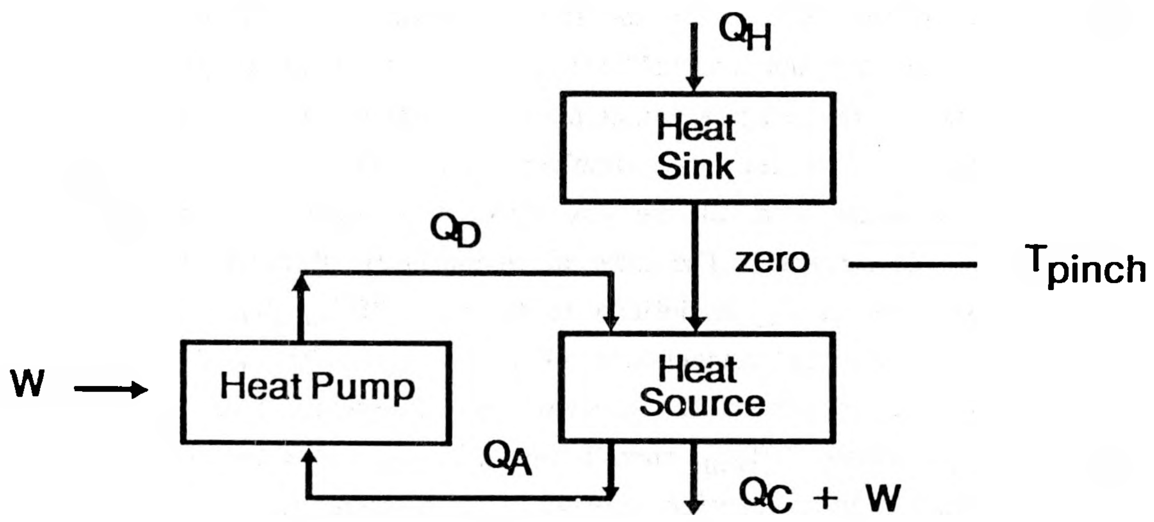
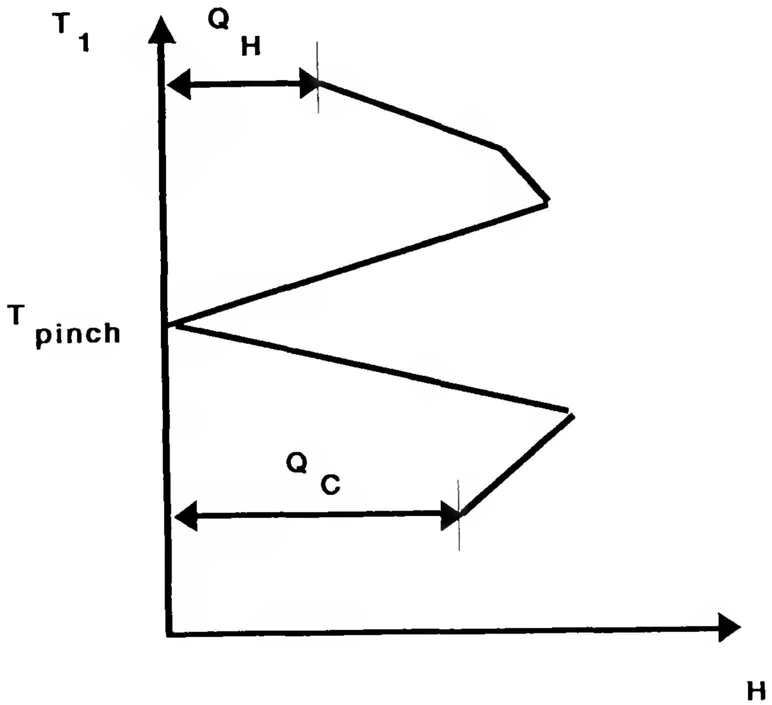


Figure A-9: Inappropriate Heat Pump Integration-below pinch



Q_H = Hot Utility Load

Q_C = Cold Utility Load

T_1 = Interval Temperature

H = Enthalpy

Figure A-10: The Grand Composite Curve

The abscissa on Figure A-10 represents the net heat flow through the process after allowing for all permitted heat integration of process streams. This takes the value of zero at the pinch, as described in the earlier discussion, and has the values of Q_H (i.e. net hot utility requirement) at the highest interval temperature in the process and Q_C (i.e. net cold utility requirement) at the lowest interval temperature.

The GCC is important in evaluating heat pumping opportunities because it allows a rapid assessment of the temperature levels available, and the amount of heat that can be heat pumped in a process. Thus, in Figure A-11 an amount of heat Q_A can be accepted by a heat pump from the process at an interval temperature T_A below the pinch. An amount of heat $Q_D = Q_A + W$ can then be delivered to the process above the pinch at interval temperature T_D.

A.6 HOT AND COLD COMPOSITE CURVES AND AREA TARGETING

The effect of heat integration on the process utility consumption and temperature driving forces is shown in the form of hot and cold composite curves in Figure A-12. The "hot composite curve" represents the summation of the heat loads associated with all streams that have to be cooled in the process ("hot" streams) and similarly, the "cold composite curve" represents the summation of all heating loads (i.e. "cold" streams). DT_{min}, the minimum temperature difference between the hot and cold composite curves, appears as the vertical distance between the hot and cold composite curves at their point of closest approach. DT_{min} is a measure of the level of heat integration.

Varying the extent of heat integration is represented by moving the hot and cold composite curves horizontally relative to one another. Doing so will change the vertical distance between them at their point of closest approach (i.e. the pinch), and thus corresponds to changing values of DT_{min}. The horizontal displacements between the composite curves at their high and low temperature ends are the corresponding values of Q_H and Q_C (the external heating and cooling requirements, respectively) for the process. A decrease in DT_{min} implies an increase in the level of heat integration. In general, as

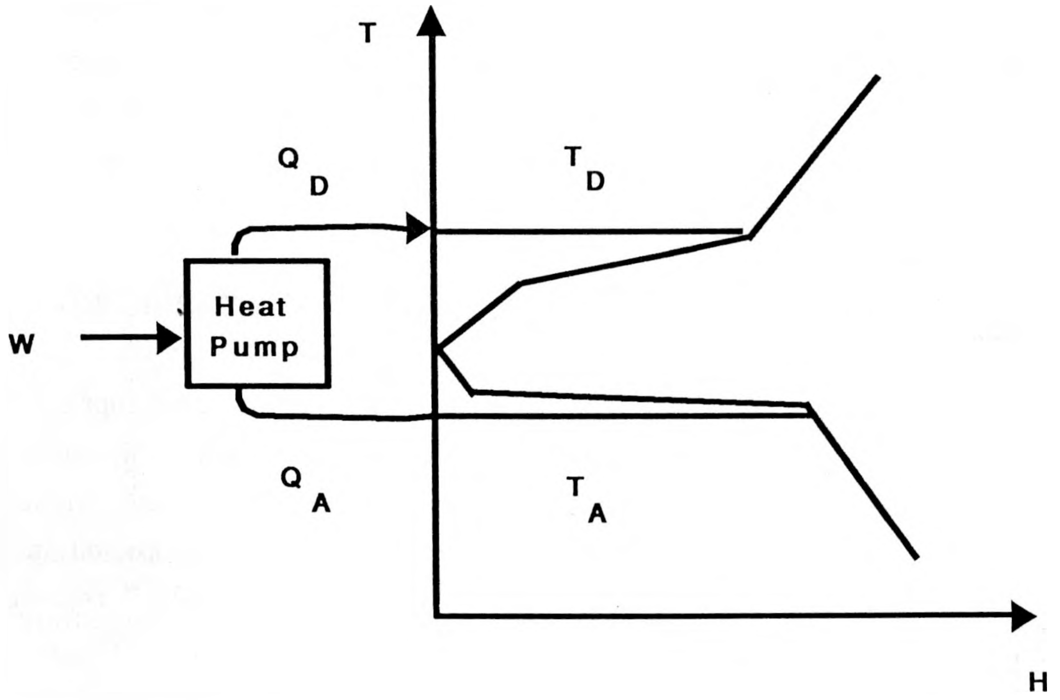


Figure A-11: The Grand Composite Curve and Heat Pump Integration

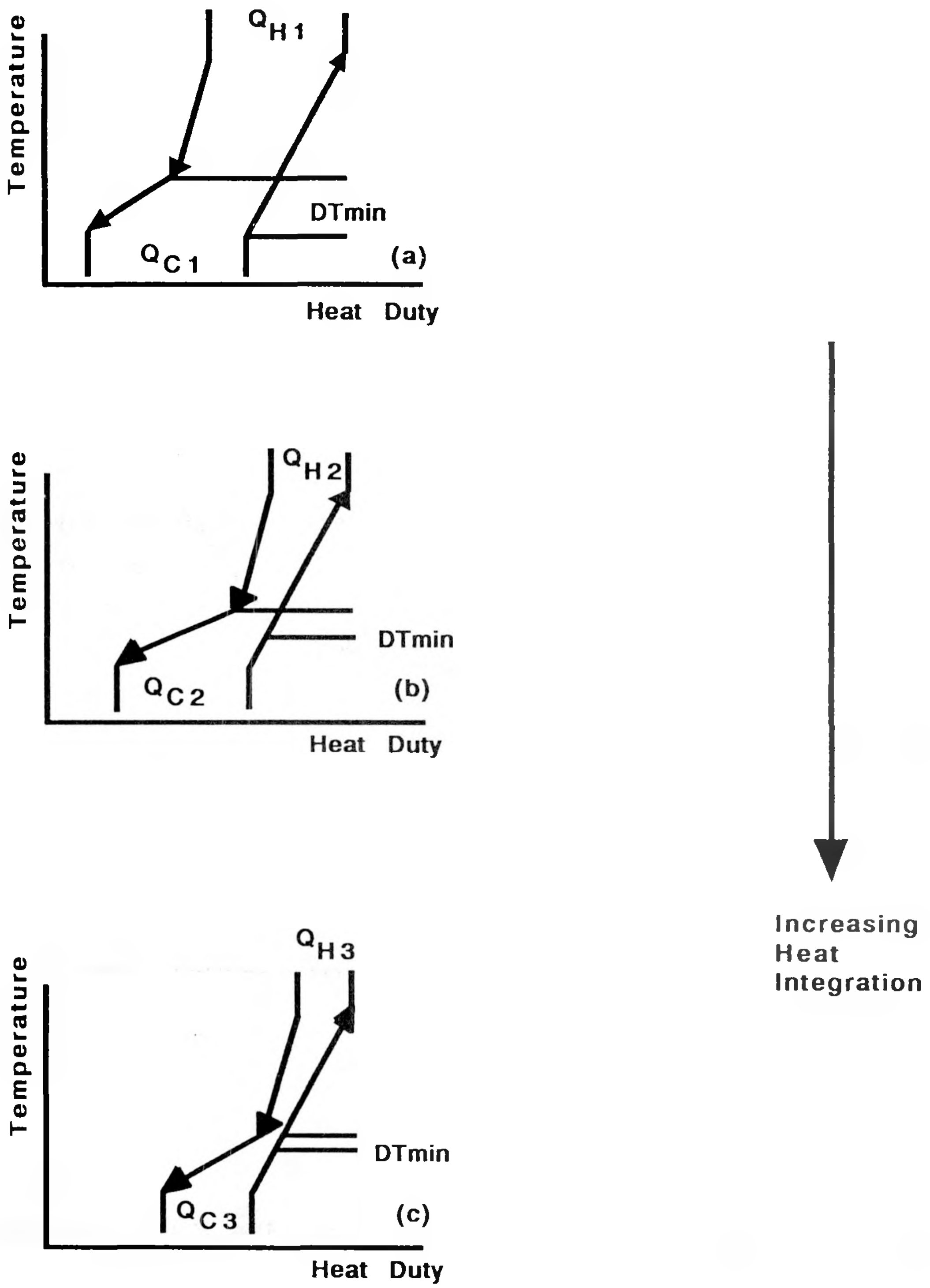


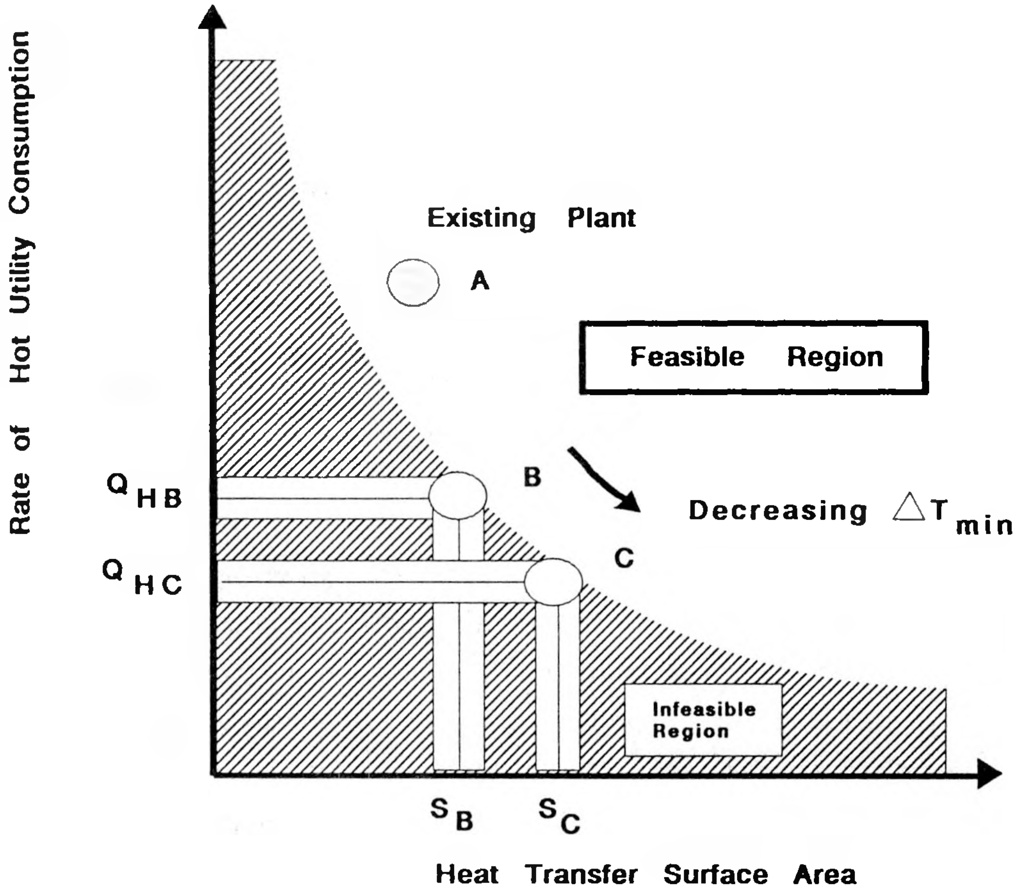
Figure A-12: Effect of Heat Integration on Utility Targets and Temperature Driving Forces

DTmin decreases the minimum hot and cold utility requirements also decrease. Progressively smaller values of DT_{min} are represented in Figure A-12(a) through A-12(c). However, a decrease in DT_{min} also implies a decrease in the overall driving force for heat transfer and a resulting increase in heat transfer surface area requirements.

Figure A-13 shows a plot of plant heat transfer area and the corresponding minimum hot utility consumption. Curves of this type can be generated for any given process using area targeting algorithms based on pinch technology principles (see below).

The curve shown in Figure A-13 separates the thermodynamically feasible region from the infeasible region. The region above the curve and to the right represents a process in which the available heat transfer area is greater than or equal to the minimum needed to achieve a specified hot utility usage level. To the left and below the curve the implied heat transfer area is less than the minimum requirement, implying that no practical process can correspond to any point in the feasible region, e.g. point A. The hot utility consumption and the heat transfer area requirements are directly related to the plant operating costs and capital costs, respectively. Therefore, the inverse of the slope of the straight line joining two points on the curve is a measure of the payback period for going from one level of heat integration to another. This is also illustrated on Figure A-13.

The subject of heat exchanger network (HEN) area requires further elaboration. Townsend and Linnhoff (9) provides a useful algorithm for estimating required HEN areas without having to design the HEN in detail. The principles behind the algorithm are illustrated in Figure A-14 in terms of heat transfer from the hot composite curve to the cold composite curve and between the process and the hot and cold utilities. For a given value of DT_{min}, there is a certain extent of horizontal "overlap" of the two composite curves. This represents the amount of heat that can be transferred from hot streams to cold streams within the process. Outside of the overlap region, utility heating or cooling is required.



$$\text{Incremental Payback Period [years]} = \frac{(S_C - S_B)\, C_A}{-\, H\, (Q_{HC} - Q_{HB})\, C_H}$$

C_A = Cost per Unit Area

C_H = Cost per Unit of Hot Utility

H = Hours of Operation per Year

Figure A-13: Hot Utility Consumption Versus Heat Exchanger Area

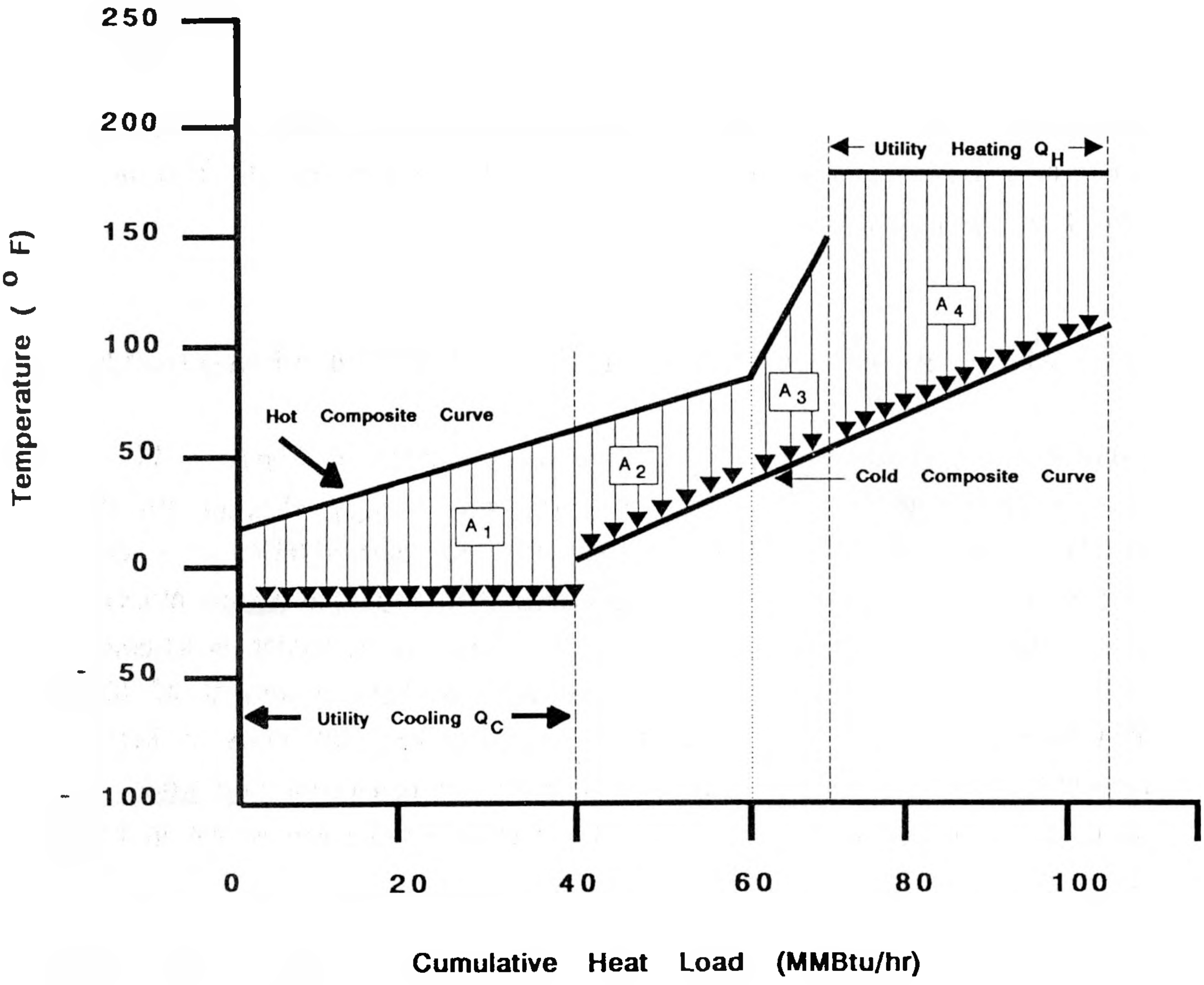


Figure A-14: Area Targets for the Example Problem

All heat transfer is represented as vertical lines on Figure A-14. This is an idealized representation implying that all matched hot and cold temperatures within the process HEN must be the same as the matched temperatures on Figure A-14. However, with this simplifying assumption, it is possible, using appropriate stream heat transfer film coefficients, to predict the minimum area for a HEN with surprising accuracy. For further details, reference (9) should be consulted.

A.7 APPROPRIATE/INAPPROPRIATE INTEGRATION OF HEAT ENGINES

Consider the hypothetical process shown to the right of Figure A-15. The process pinch and minimum utility requirements (Q_{Hmin}, Q_{Cmin}) are shown on the figure. To the left of Figure A-15 is a representation of a Carnot engine, for which both the heat acceptance and rejection temperatures are hotter than the process pinch temperature. This Carnot engine is assumed to have an efficiency (ratio of power produced to heat absorbed) of 33.3%. This means that for each 3W units of heat absorbed, 2W units of heat are rejected at a lower temperature and W units are converted into work. The heat and work flows associated with the Carnot engine are shown in Figure A-15.

The total hot utility requirement for the Carnot engine and process is (Q_{Hmin} + 3W). This requirement can be reduced by integrating the heat engine with the process above the pinch, see Figure A-16. In the "above pinch" integrated arrangement, the engine exhaust heat displaces the process hot utility usage and the total utility requirement falls to (Q_{Hmin} + W), a saving of 2W units of heat. Work W has effectively been produced at a marginal efficiency of 100% (neglecting mechanical and electrical losses) since the waste heat from the machine is usefully used to displace process requirements.

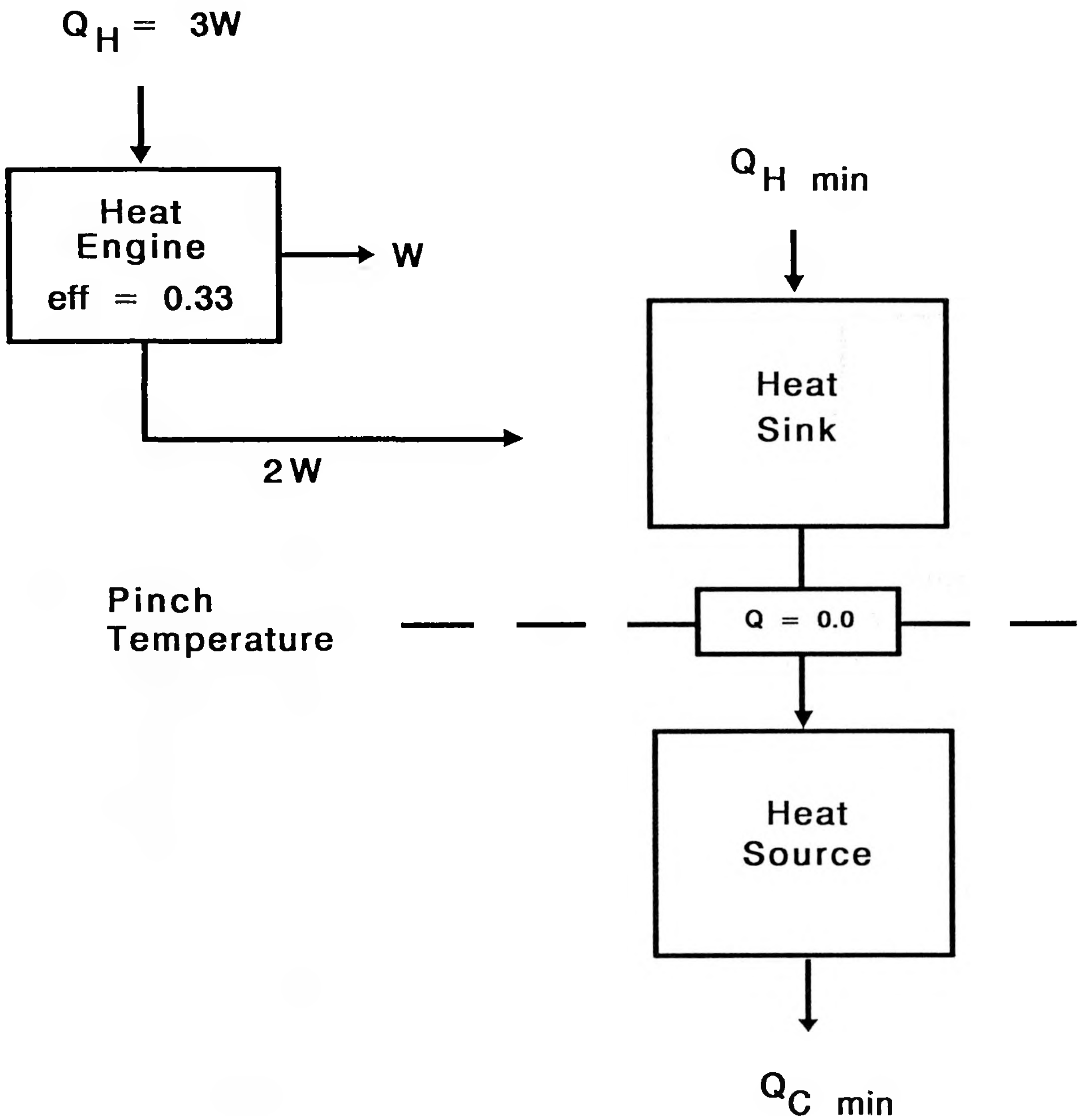


Figure A-15: Stand Alone Heat Engine Operation and Process Demand

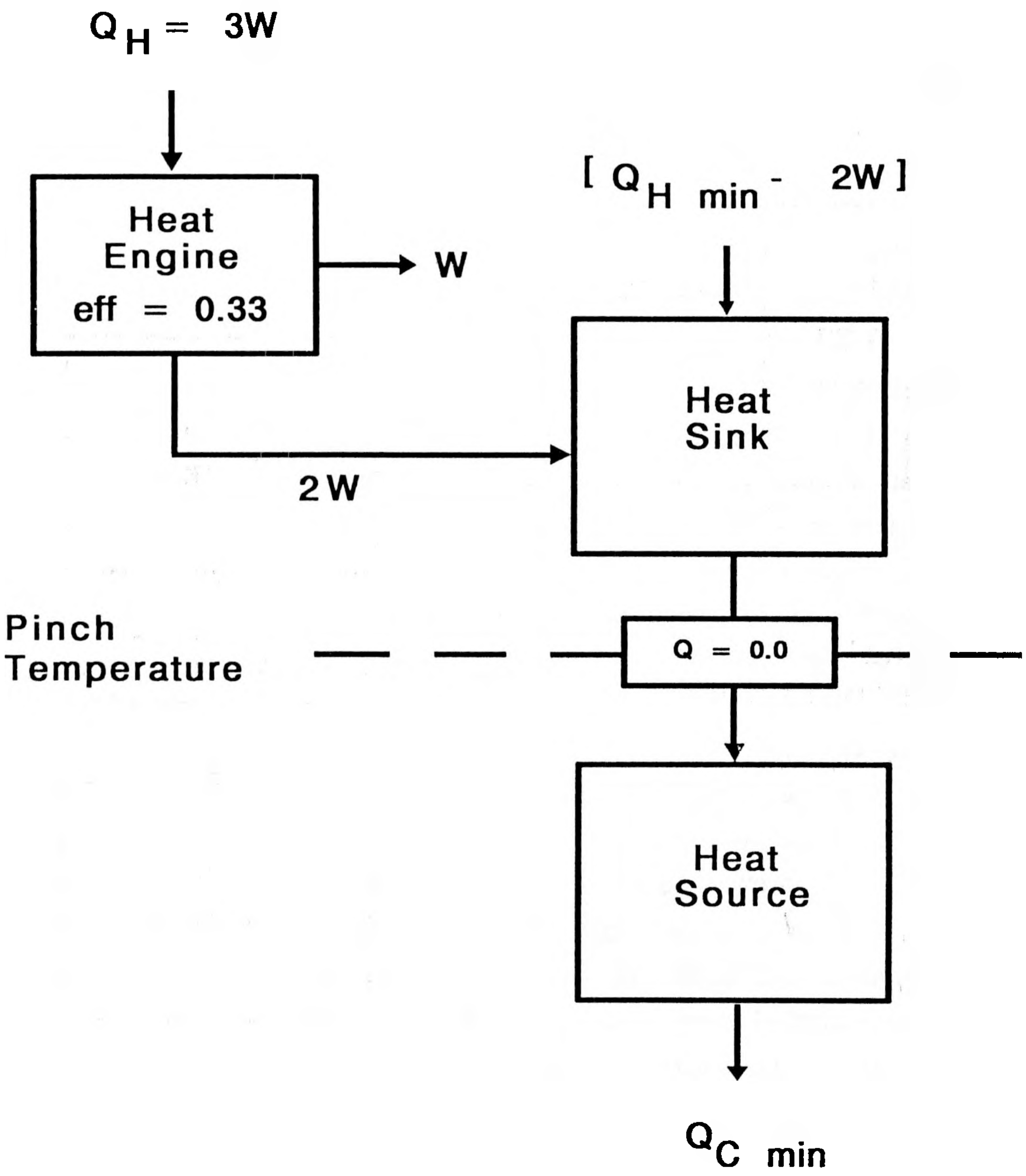


Figure A-16: Appropriate Heat Engine Integration

Figure A-17a illustrates the same process together with a Carnot engine for which both the heat acceptance and rejection temperatures are colder than the process pinch temperature. As in Figure A-15, a machine efficiency of 33.3% is assumed. The total hot utility requirement for the Carnot engine and process shown in Figure A-17a is (Q_{Hmin} + 3W). This requirement reduces to Q_{Hmin} when the heat engine is integrated with the process below the pinch, see Figure A-17b, since the engine heat requirements are supplied by waste process heat.

In Figure A-18, a Carnot engine is shown integrated with the process such that heat is absorbed from above the pinch and rejected below the pinch. In this heat integrated arrangement, the minimum utility requirement is not reduced from the total non-integrated requirements. In other words the heat engine violates the process pinch by transferring heat across it. As a result, the total hot utility requirement of the integrated system (Q_{Hmin} + 3W) is the sum of the two separate system requirements.

Figure A-16 and Figure A-17 illustrate the concept of "appropriate" integration of heat engines with a process. Appropriate integration involves operating a heat engine such that the engine heat acceptance and heat rejection are entirely above or entirely below the process pinch but not across the pinch. As the figures illustrate, this appropriate integration leads to substantial reductions in utility requirements over the inappropriately integrated case, Figure A-18. Inappropriate integration means the heat engine transfers heat across the process pinch.

A.8 PLACEMENT OF DISTILLATION COLUMNS AND EVAPORATORS

The rules for placing distillation columns and evaporators are essentially the same as those for heat engine placement. The entire distillation or evaporation system should be either above the pinch, or below the pinch to ensure the maximum scope for beneficial heat integration of the condenser and reboiler heat loads. Placing the system such that the reboiler is above the pinch and the condenser is below the pinch (see Figure A-19) is "inappropriate" as it degrades heat across the pinch.

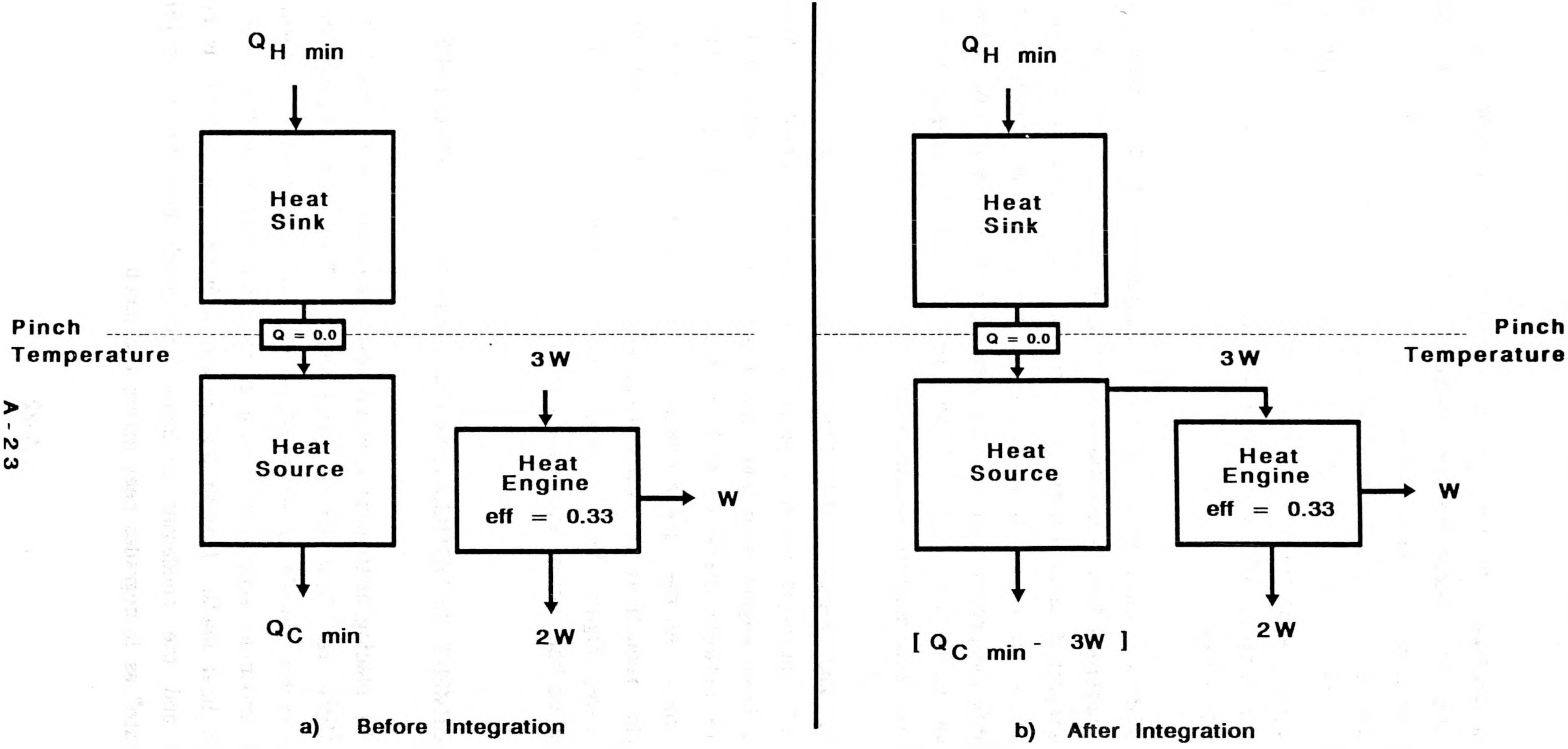


Figure A-17: Appropriate Heat Engine Integration Below the Pinch

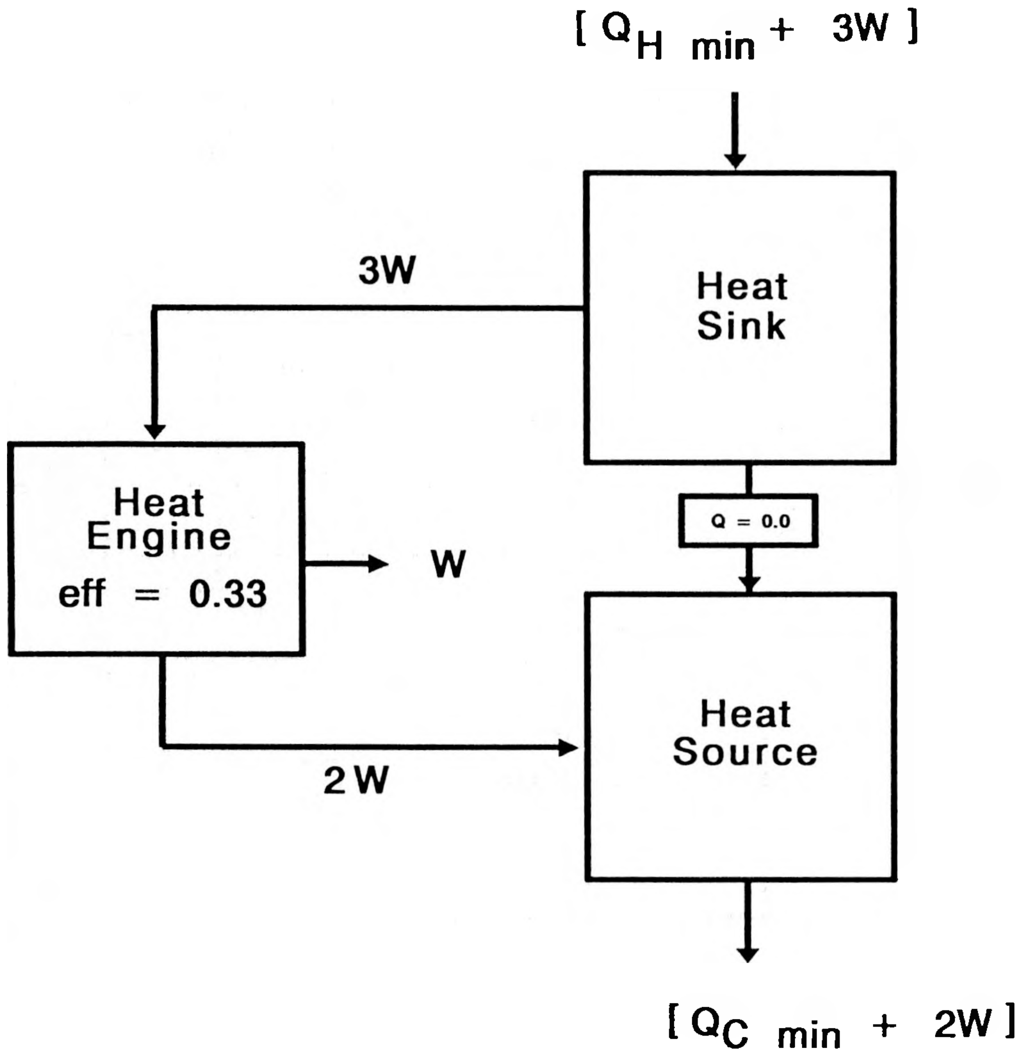


Figure A-18: Inappropriate Heat Engine Integration

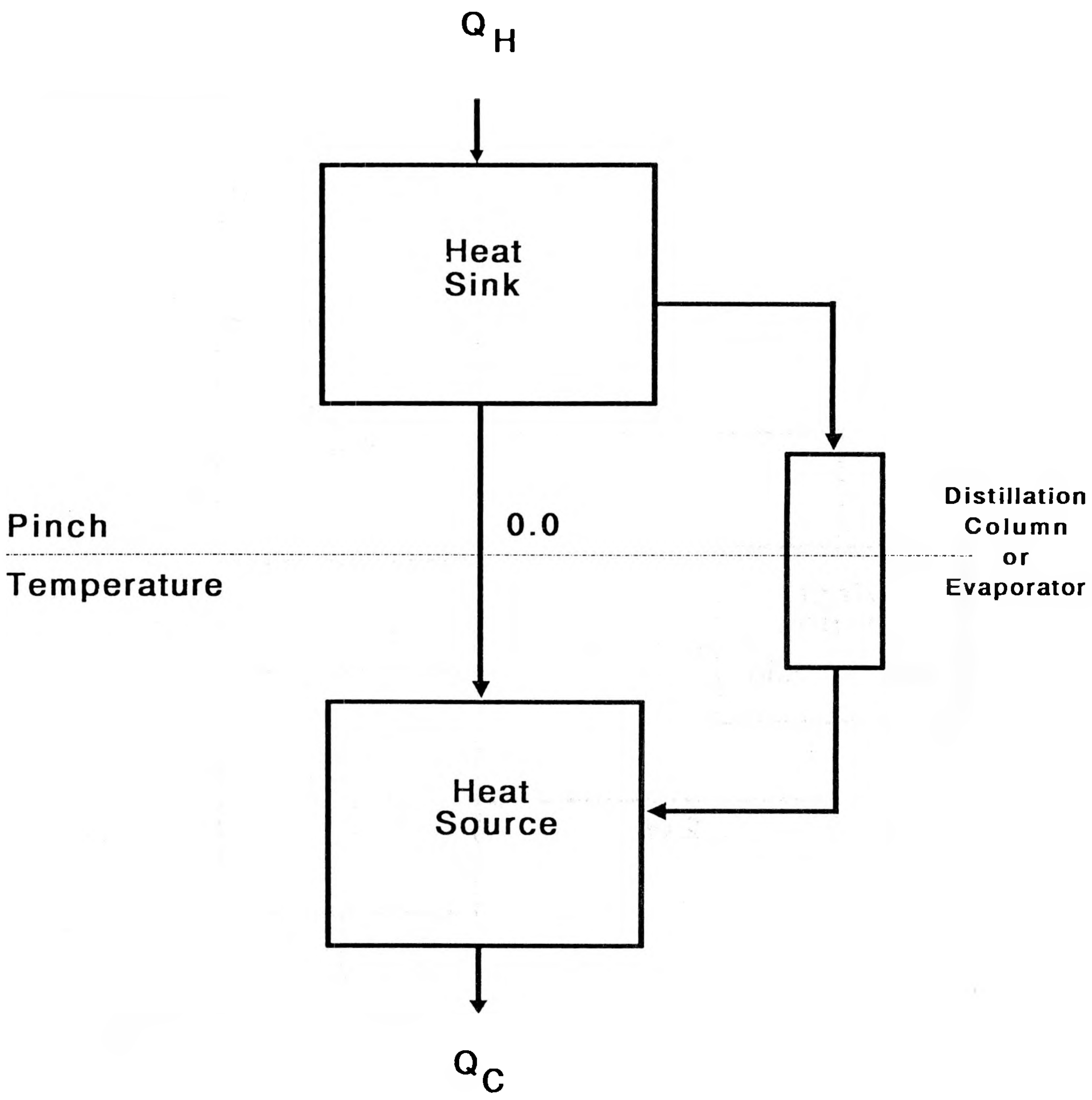


Figure A-19: Inappropriately Placed Distillation Column or Evaporator

The procedure for correcting inappropriate "cross pinch" placement of distillation columns (4) is based on the concept of "pressure shifting". Reducing the pressure of the column lowers the evaporation and condensation temperatures, and so may allow the temperature of an "above pinch" reboiler to be reduced to below the pinch. Conversely, raising the pressure may allow a "below-pinch" condenser to be raised to a temperature above the pinch. In either case, the result is that the "cross pinch" placement is eliminated and both the reboiler and condenser are restored to the same side of the pinch.

Correction of inappropriate evaporator placements uses precisely the same procedure.

A.9 SUMMARY

This appendix gave a quick review of pinch technology and its relevance to heat pump and heat engine placement in industrial processes. Methods to determine the appropriate positions for heat pumps and heat engines in any given process have been detailed.

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