PROCESS ENERGY EFFICIENCY IMPROVEMENT

IN WISCONSIN CHEESE PLANTS

by

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A thesis submitted in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

MECHANICAL ENGINEERING

at the
UNIVERSITY OF WISCONSIN - MADISON

1997

ABSTRACT

Wisconsin continues to lead the nation in the production of cheese, providing about 30% of the national supply. The industry is less secure than it once was, however, due to narrowing profit margins and competition from large plants in California and elsewhere. In 1994, the production of cheese consumed 87.7% of the milk produced in Wisconsin. Thus the success of the Wisconsin cheese industry is closely linked to the success of the Wisconsin dairy industry at large, which provides about three billion dollars per year in gross income.

In this study, the utility use of two representative cheese plants is examined. Utility costs generally represent about 11% of the total manufacturing cost of cheese. Utility use is one of the few variables that a plant manager can influence to improve plant profitability.

The largest fraction of the energy used at a typical plant is devoted to processing whey, a byproduct of significant food value. Two types of equipment are commonly used to process or remove moisture from whey: evaporation systems and spray dryers. Through the course of this research project, models for both evaporation and spray drying equipment were developed. Pinch analysis has been applied to investigate heat recovery options such as open cycle heat pumps and heat exchange units.

Significant opportunities for reduction of utility use have been identified for both the evaporation system and the spray drying system examined. Making use of the low temperature vapor rejected from the evaporation system by preheating raw milk provides two simultaneous benefits. Steam use by the

pasteurizer is decreased, and the cooling rejection load on the cooling tower (equipped with a 60 hp motor) is eliminated. Cost savings associated with these benefits are estimated to exceed $70,000 per year.

In the spray dryer system studied, outdoor air is heated to 240°F before being introduced into the drying chamber. Exhaust air leaves the chamber at 155°F. The use of either direct or indirect heat exchange between the supply air before it reaches the burner and the exhaust air is explored. Energy cost savings in the range of $40,000 to $70,000 are predicted for this opportunity.

In addition to the heat recovery analysis just described, this study explores an alternative control strategy to reduce electricity costs related to maintaining a cold storage warehouse through demand shifting. By sub-cooling the stored cheese during the off-peak period, it is possible to meet all or most of the cooling load in the warehouse as the sub-cooled cheese returns to its normal storage temperature. In this way, operation of the cooling equipment during peak-time can be avoided or reduced significantly. This control strategy has been examined using a finite difference model of the warehouse. The model has demonstrated the approach to be feasible. It has been estimated to result in reduced annual electric costs of about $15,000.

ACKNOWLEDGMENTS

First I wish to thank my team of advisors. Sandy Klein, Doug Reindl, and Doug Reinemann were all tremendously insightful and fully engaged in providing direction and technical assistance to me. I am most indebted to my principal advisor, John Mitchell. Not only has he played a dominant role in guiding this work, but more importantly, he has helped me and others in the lab to maintain humor and perspective through what can be an arduous process.

Though not directly involved in this project, I would like to acknowledge the role of Bill Beckman as director of the Solar Energy Lab. The lab is well known for high achievement and dedication to energy research that can lead this country to a smarter, more environmentally friendly future.

I wish to thank Anton Muljadi and Jeannie Leggett, both assistants to Doug Reinemann. Each of them were very supportive to this effort.

I am very grateful to Steve DeGeest of the C.E. Rogers company. He offered generous assistance and information related to the processing equipment under study.

Special thanks to Steve Ejnik, and Jim Mitchell, plant managers at Marshfield and Blair. Each of them took time out of their busy schedules to offer tours of their plants and to answer numerous questions.

Finally, I wish to thank the Energy Center of Wisconsin for their generous financial support of this project.

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NOMENCLATURE

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C	Capacitance Rate (Btu / hr)
c_p	Specific Heat (Btu / (lb °F))
COP	Coefficient of Performance
DM	Dry matter (mass dry matter / mass moisture)
h	Heat transfer coefficient (Btu / (hr ft^2 °F)) [chapter 5]
h	Specific enthalpy (Btu / lb) [chapter 3]
k	Thermal conductivity (Btu / (hr ft °F))
L	Length (ft)
$\dot{m}$	Mass flow rate (lb / hr)
Nu	Nusselt number
Pr	Prandtl number
Q	Heat flow rate (Btu / hr)
Ra	Rayleigh number
SHR	Specific heat ratio
T	Temperature (°F)
TS	Total solids (mass dry matter / mass moisture)
UA	Conductance (Btu / (hr °F))
V	Velocity (ft / s)
x	Mass fraction

Greek

g	Kays & London parameter	
e	Effectiveness	
r	Density (lb / ft^3)	

Subscripts

bulk	Precooled stored cheese	
c	critical	[chapter 5]
c	cold	[chapter 4]
cond	Condensate	
daily,prod	Amount of cheese produced each day	
h	hot	
r	radiation	
sol-air	effective temp. accounting for dry bulb temp. and solar gain	
∞	Ambient	
wh	Warehouse	

CHAPTER

ONE

INTRODUCTION

1.1 Overview

The ways in which consumers in the United States use energy is a growing concern for a number of reasons. One of the reasons relates to the national security risk associated with building an economy that is heavily dependent on an imported commodity (oil). A second reason for concern relates to the environment. Transporting environmentally degrading materials involves a measure of risk. In addition, combustion of fossil fuels has been directly linked to global climatic change. A third reason for concern is economic. Using valuable resources inefficiently can weaken the market position of a product, an industry, or of our country. Finally, utilities are often interested in minimizing electric demand to avoid the cost and environmental impact of building new generation facilities.

The U.S. energy economy is often broken down into three categories of use:
1) Transportation, 2)Industry, and 3) Commercial and Residential. Of these three, the energy used by industry is the largest. In 1994, 33.72 quadrillion Btu's were attributed to this sector (DOE homepage). Public interest and awareness of energy-use issues has largely been limited to the transportation and building sectors.

The impetus behind this study has been to consider energy use by the industrial sector of the state of Wisconsin, with a particular focus on the cheese industry. The project was initiated in the hope of documenting the potential for electric demand savings through heat pump application or alternative refrigeration technology. Capturing such potential should assist the Wisconsin cheese industry to remain competitive in a market with narrow profit margins, and would secure the societal and environmental benefits associated with energy savings.

1.2 The Wisconsin Cheese Industry

Until 1993, Wisconsin led the nation in annual milk production. In that year, California and Wisconsin each produced 15.2% of the nation's total. Since 1993, California has gained a small edge in milk production. Nevertheless, gross income from milk production in Wisconsin remains very close to three billion dollars per year. (Wisconsin 1995 Dairy Facts, p. 25)

As of 1994, however, Wisconsin had retained its lead in US cheese production, providing 30% of our total national production. The production of cheese consumed 87.7% of the milk produced in Wisconsin in 1994. This rate of milk utilization for the production of cheese has been steadily increasing since 1985. Thus the success of the cheese industry in Wisconsin is closely linked to the success of the dairy industry of Wisconsin at large.

The composition of the Wisconsin cheese industry has undergone broad changes as this increase in production has occurred. In 1950, there were 1,279 cheese plants in operation in Wisconsin. By 1970 the number had dropped to 334. In 1994 there were just 153 cheese plants in operation (see figure 1.1.) This decrease in number of plants has occurred in the midst of an increase in production of nearly 400% (see figure 1.2.) The clear trend then is toward fewer cheese plants of larger production capacity.

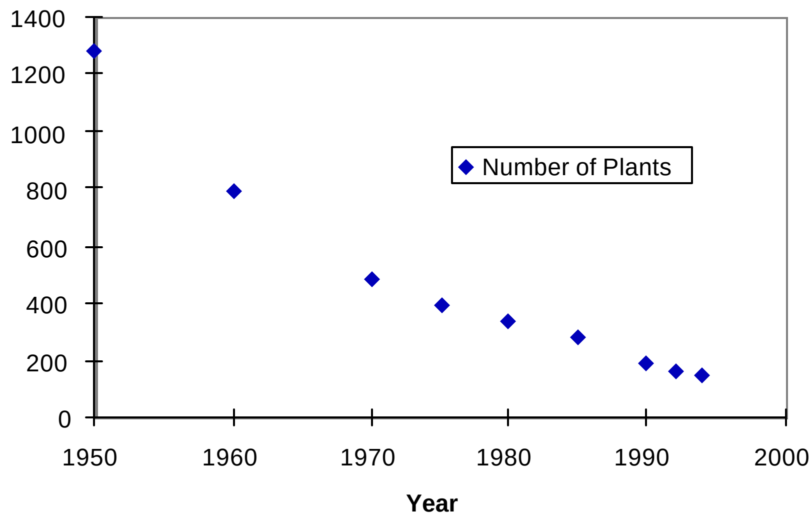

Figure 1.1 Cheese Plant Population from 1940 -1994

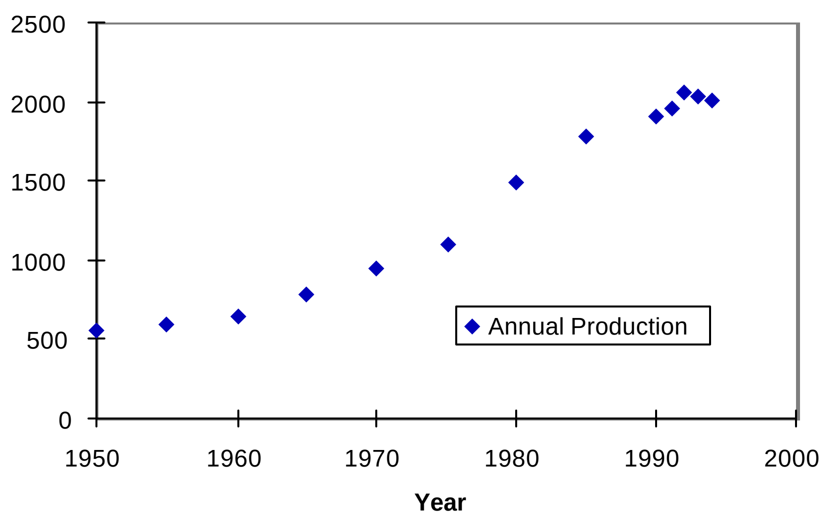

Figure 1.2 Annual Wisconsin Cheese Production from 1950-1994

1.3 Profitability in the Cheese Industry

Four major factors impinge on the profitability of a cheese making operation. These include the cost of milk, the total cheese and whey product manufacturing cost, the selling price of the cheese, and the selling price of the whey products. Of these four, a plant manager has influence only over the manufacturing cost.

Table 1.1 Average Production costs for Cheddar cheese and whey powder for a model plant processing 960,000 milk/day (Barbano, p.54).

Cost Item	**Cost per cwt milk ($/cwt) ***	**Percentage of Total Costs (%)**
Labor		
Supervisory	0.064	2.6
Direct Fixed	0.063	2.5
Direct Variable	0.764	30.9
Total Labor	**0.891**	**36.0**
Capital Costs		
Deprec. + Interest	**0.544**	**22.0**
Utilities		
Electricity	0.085	3.4
Fuel	0.164	6.6
Water & Sewage	0.031	1.3
Total Utilities	**0.280**	**11.3**
Materials		
Laboratory	0.007	0.3
Production	0.218	8.8
Packaging	0.199	8.0
Cleaning	0.051	2.1
Total Materials	**0.475**	**19.2**
Repair & Maintenance	**0.050**	**2.0**
Prop. Tax & Insurance	**0.183**	**7.4**
Production Inventory	**0.021**	**0.8**
Other Expenses	**0.032**	**1.3**
TOTAL	**2.476**	**100.0**

***** cwt = 100 lb

Barbano et al. have produced economic models of typical cheese plants of different capacities to gain insight into economy of scale factors influencing cheese plant profitability. Table 1.1 provides the manufacturing cost breakdown from their model for a 960,000 lb milk / day capacity cheese plant.

From table 1.1 it can been seen that 11.3% of the operating costs for a cheddar plant are utility related. Of this 11.3%, 6.6% is related to fuel use. It will be shown that for a plant such as Marshfield that operates a steam driven evaporation system, most of this fuel is consumed by the evaporation system.

The Barbano study goes on to analyze the profitability of cheddar plants when subject to the price factors over which a plant manager has no control. Using economic data from 1987 to 1990, their results show that small plants have little hope of competing economically with large plants in the production of commodity cheddar cheese. Additionally, the study shows that cheese plants typically operate with a very narrow profit margin, and that in many cases it is the profitability of the whey processing operation that allows plants to operate with a net gain rather than a net loss. Figure 1.3 shows the results of a 1987 study of the impact of plant size and hours of operation on profitability.

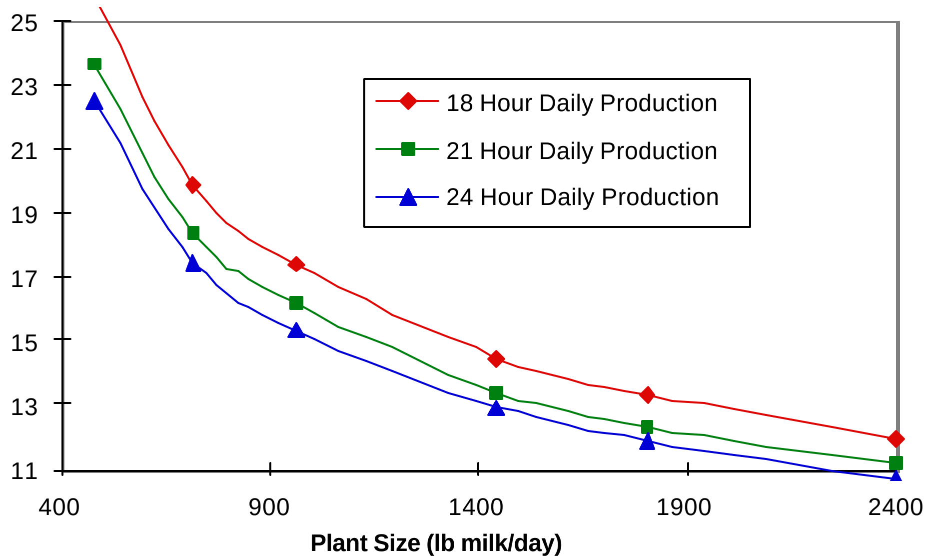

Figure 1.3 Average Production Costs for Different Size Cheddar Cheese Plants (Barbano, p.46)

1.4 Whey Production

Whey is the watery portion or serum that remains after coagulation that separates from the curd during cheese making. It is a dilute liquid containing lactose, protein, minerals, and traces of fat. It contains approximately 6% total solids, of which 70% or more is lactose and about 0.7% is whey proteins (Zadow, p.5).

For two reasons, utilization of whey rather than its disposal has gained attention. First of all, there are environmental concerns associated with disposing of whey into streams or sewage plants. It is estimated that the BOD (Biological Oxygen Demand) of 100 lb of whey is equivalent to the waste produced by 21 people every 24 hours (Gillies, p.11). A second reason that whey utilization has gained attention is due to its potential as a food source rich in lactose and protein.

Twenty years ago, two-thirds of whey was disposed of as waste (Zadow, p.13). At that time, since cheese plants were smaller and more widely dispersed, it was recommended by state agencies that the whey be sent back to the farms for use as feed for livestock or to be spread on fields to build and fertilize the soil (Gillies, p.10). The practicality of this approach to whey disposal has quickly become unfeasible as cheese plant capacities have grown.

Presently about 50% of whey produced is utilized as a food product for either human or animal consumption (ADPI Survey, p.16). This number has held steady during a period in which the production of cheese (and also the amount of whey generated) has grown at a significant rate. Table 1.2 show the various products resulting from processing of raw whey and the amounts produced .

Table 1.2 Estimated U.S. Whey Solids Production and Quantity Further Processed (ADPI Survey)

Year	1972	1974	1976	1978	1980	1982	1984	1986	1988
Total Whey Production (Solids Basis)	1830	1987	2219	2327	2591	2901	2969	3425	3625
Whey Solids Further Processed									
A-Conc. Whey Solids	70	61	146	144	86	142	130	47	37
B- Dry Whey									
Human Food	377	453	480	515	534	611	725	890	940
Animal Feed	385	399	182	196	156	179	173	141	197
C- Mod Dry Whey Prod									
Red.Lact./Min. Whey			154	196	189	163	88	90	122
Whey Protein Conc.			8	9	4	71	96	78	136
D- Whey Solids in Wet Blends			74	96	144	144	136	116	128
E- Whey Solids Utilized for Lactose	141	210	170	183	224	230	198	216	258
Total Whey Further Processed	973	1123	1214	1339	1337	1540	1546	1578	1818
Total Whey Further Processed as % of Total Whey Processed	53%	57%	55%	58%	52%	53%	52%	46%	50%

(In millions of pounds)

In its dilute form, whey has little or no market value. Thus, the use of whey as a food product requires processing of some type. In general, the whey is first concentrated either by evaporation or by a filtration system. It can then be dried to a powder either with a spray drying system or a roller drying system. Each of these processing steps is highly energy intensive.

Finally, it should be noted that whether or not a cheese plant processes its whey (either by concentration or drying) has a significant effect on the overall profitability of the plant. This is a key factor in the wide profitability margin between large and small cheese plants, as small plants cannot afford the capital investment required to purchase whey processing equipment. This is illustrated in figure 1.4. Using price data from January 1987 to December 1990, the Barbano model shows that cheese production alone resulted in a net loss, while whey powder production almost consistently produced profit. In fact, the profit generated from whey processing enabled the overall cheese plant modeled to return a net profit rather than a net loss at several points during that period.

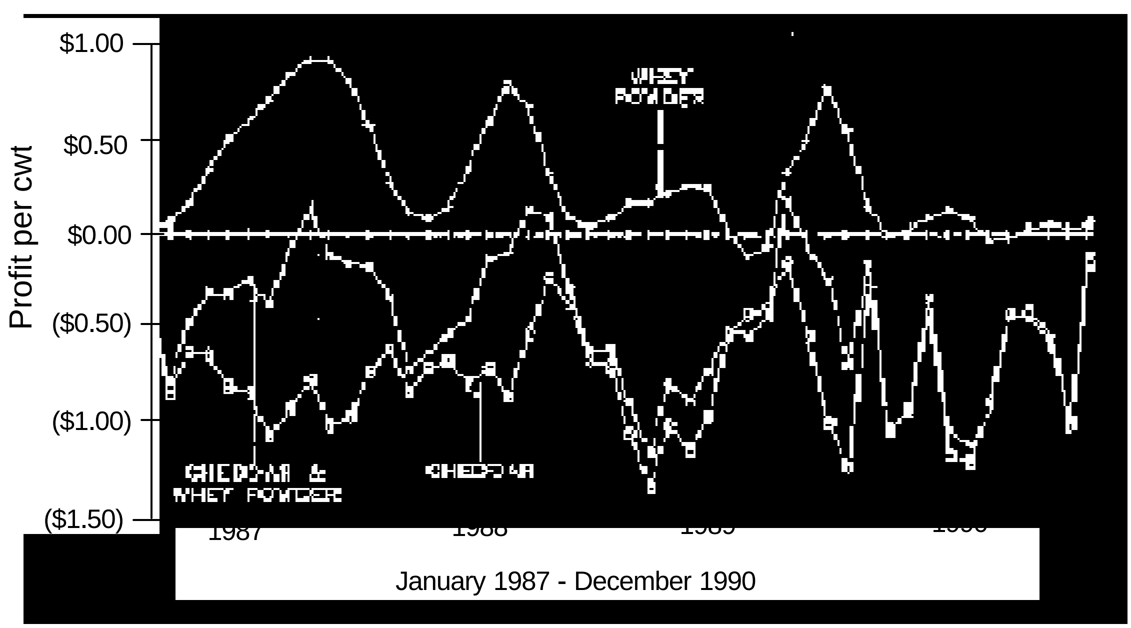

Figure 1.4 Profitability of Cheese and Whey Production (Barbano, p. 64)

1.5 Methodology for Energy Analysis

Within a cheese plant, a number of energy demands can be readily identified. Lighting, pumping, heating and cooling are the dominant ones. Of these energy uses, heating and cooling account for all of the natural gas usage and a large portion of the electric demand is used by refrigeration equipment. For this reason, this study examines the heating and cooling process requirements of cheese production and whey processing to determine whether potential exists for integration of these requirements.

The chief tool used in this study for determining the potential for thermal integration limits for a set of process streams is known as pinch analysis (Linnhoff, p.33). It is a graphic method in which all streams requiring cooling are collapsed into a composite hot stream and all of the streams requiring heating are collapsed into a composite cold stream. These streams can then be plotted on a temperature - heat flow (T - Q) diagram to show the trajectory of the streams. Once an adjustment of these plots is made to account for the minimum temperature difference allowed for heat exchange, the streams can be plotted on the same set of axes and moved laterally until they touch at one point.

This point at which the composite plots touch is the pinch point. Heat should not be sensibly transferred across the pinch temperature. And a heat pump should not be utilized unless it utilizes heat capacity below this temperature to supply a heating requirement above this temperature. Also from a pinch plot, the theoretical minimum hot and cold utility usage can be determined. This enables a designer to understand the degree to which further integration is possible. Violating either of the “rules” just described makes attainment of the minimum theoretical utility requirements impossible. Further explanation of pinch methodology is provided in appendix A.

Pinch plots will be generated for the process equipment responsible for the majority of the energy use at two representative Wisconsin cheese plants. In the first plant, the whey evaporation system is analyzed. In the second plant, the whey spray dryer is examined.

1.6 Food Impact Concerns

An important consideration for any engineering study is the potential for the effect, if any, of design modifications on the quality of the food product. In terms of thermal integration, there will always be a number of infeasible stream matches for such reasons. For instance, one would not consider direct sensible heat exchange between a pasteurized and an unpasteurized product stream unless a substantial pressure difference existed to prevent contamination of the pasteurized stream. The significance of this point is reflected in the following statement of Barbano:

> “The first priority in profitable operation of a Cheddar Cheese factory, or any other dairy product manufacturing plant , is to make a high quality product. If you cannot consistently accomplish this then all of the other items related to improving profit are of very little importance (Barbano, p.64).”

1.7 Conclusions

The success of the dairy industry is closely linked to the success of the cheese industry. The current cheese industry of Wisconsin is composed of a relatively small number of plants that have large production capacities. Given the fact that cheese plants operate with narrow profit margins, even modest savings on utility costs can make the difference in whether or not a plant remains economically competitive.

It is for these reasons that this study of industrial energy use has focused on the larger capacity cheese plants. Using pinch analysis, two representative cheese plants are analyzed to explore possibilities for further thermal integration. These investigations have largely been focused on the whey processing equipment at each plant due to the large proportion of utility use by this equipment. At the first plant, a multi-stage falling film evaporation system with thermal vapor recompression is modeled and analyzed. In the second plant, the integration possibility of a vertical spray drying unit with a secondary bed dryer is explored.

A final element of this study relates to potential for savings in electric demand charges. Every cheese plant includes at least short term cold storage of the cheese product. The cooling demand for maintaining this storage space is generally met with electric powered cooling equipment. As a relatively large consumer of electricity within the plant, this cooling demand significantly contributes to the monthly peak demand charge paid by the plant. Refrigeration equipment also contributes to high electricity costs due to operation during the peak demand period when high billing rates are in effect.

The possibility of operating the cooling equipment only at night and allowing the warehouse to “float” through the high billing rate period has been studied. A transient analysis of the warehouse to explore the potential for exploiting the thermal storage capacity of cheese has been conducted.

CHAPTER

Two

CHEESE PLANT DESCRIPTION

2.1 Flow Sheet of a Cheddar Cheese Plant

To analyze the energy use of an industrial plant, one needs to be familiar with the process and with the product and utility flows through this process. This section will discuss the basic elements of a cheese plant, along with the energy considerations for these processes relevant to this study.

There are a series of steps common to all cheese making operations. Details of each of these steps are provided below.

a.) Receiving -- The raw milk is received from the farm in bulk tank trucks. Prior to transport from the farm it is cooled to 40°F. At the cheese plant it is unloaded into cooled storage silos for holding until use.

b.) Treatment -- The raw milk is tested for fat content and is mixed with either cream or skim to achieve a standardized fat content. Standardization is necessary to produce a consistent quality of cheese using milk which has a seasonal quality variation. Also, the fat content is controlled to produce “low fat” cheeses. The milk is then pasteurized, generally using a method known as

HTST, for High Temperature Short Time. In this method, the milk is heated to 165°F, held at that temperature for 15 seconds, and then rapidly cooled down to 88°F to reduce protein damage.

c.) “Make Vats” -- The milk is pumped into large vats where it is mixed with starter bacteria and rennet. The mixture is carefully heated to 101°F and held at that temperature for a specific time interval as coagulation takes place and the curds are formed and cooked. Knives are built into these vats to agitate the mixture and to reduce the curd size, thereby allowing easy drainage of the whey.

d.) Cheddaring -- The curd is presented to a machine with a series of conveyors. In this machine the curd is drained, matted, and milled. Salt is applied to the curd once the proper level of acidity is reached to inhibit further production of lactic acid.

e.) Packaging -- The curds are placed in vacuum towers where the remaining whey is pulled off and the curds are knit together. The resulting 40 lb blocks of cheese are sealed in plastic and enclosed in cardboard boxes. These boxes are then placed on pallets.

f.) Cold Storage -- The pallets of cheese blocks are placed in storage where they undergo a seven day cooling period and are then maintained at 38°F until transport for retail packaging or long-term storage.

g.) Whey Separation -- The whey is centrifuged to separate the whey cream.

h.) Whey Processing -- The liquid whey that remains is then processed for use as animal feed or as a human food product, depending on equipment availability and existing market demand.

Time is an important factor in many of the processes described above. The duration of each step is carefully controlled to achieve a desired acidity level in the curd. Table 2.1 below presents the temperatures, durations, and acidities typical for each step in the process. The process at Marshfield differs only slightly from these values.

Table 2.1 Summary of Acidities, Temperatures and Operational Times of Traditional Cheddar Cheese. As Taken From Cheese Logs (Scott, p.219)

Process Stage	Acidity (pH)	Temp (°F)	Length (min.)	Time Elapsed (hr)
1. Raw milk	0.16	50		
2. Heat Treatment	0.16	161	0	0
3. Starter	0.18	86	20	20
4. Rennet	0.20	86	45	1:05
5. Cut	0.115	86	15	1:20
6. Stirring and scalding	0.135	100	40	2:00
7. Pitching	0.155	100	55	2:55
8. Draining whey	0.25	95	25	3:20
9. Cheddaring	0.62	90	65	4:25
10. Milling	0.65	86	15	4:40
11. Salting		84	20	5:00
12. Hooping		82	10	5:10
13. Pre-press	0.75 (whey)	79	40	5:50
14. Dressing			10	6:00
15. Pressing	1.15 (whey) 5.2 (curd)	75		
16. Storage		45-60	21+ days	

2.2 Energy Use within a Cheese Plant

Of interest for this study is the degree to which each of the processes described above require heating or cooling. Several of the steps listed above are clearly not important to such a study. Receiving, cheddaring, packaging, and whey separation have no heating or cooling requirement. Only those steps that have utility demands will be examined more closely. Figure 2.1 illustrates these processes and indicates where hot and cold utilities are required.

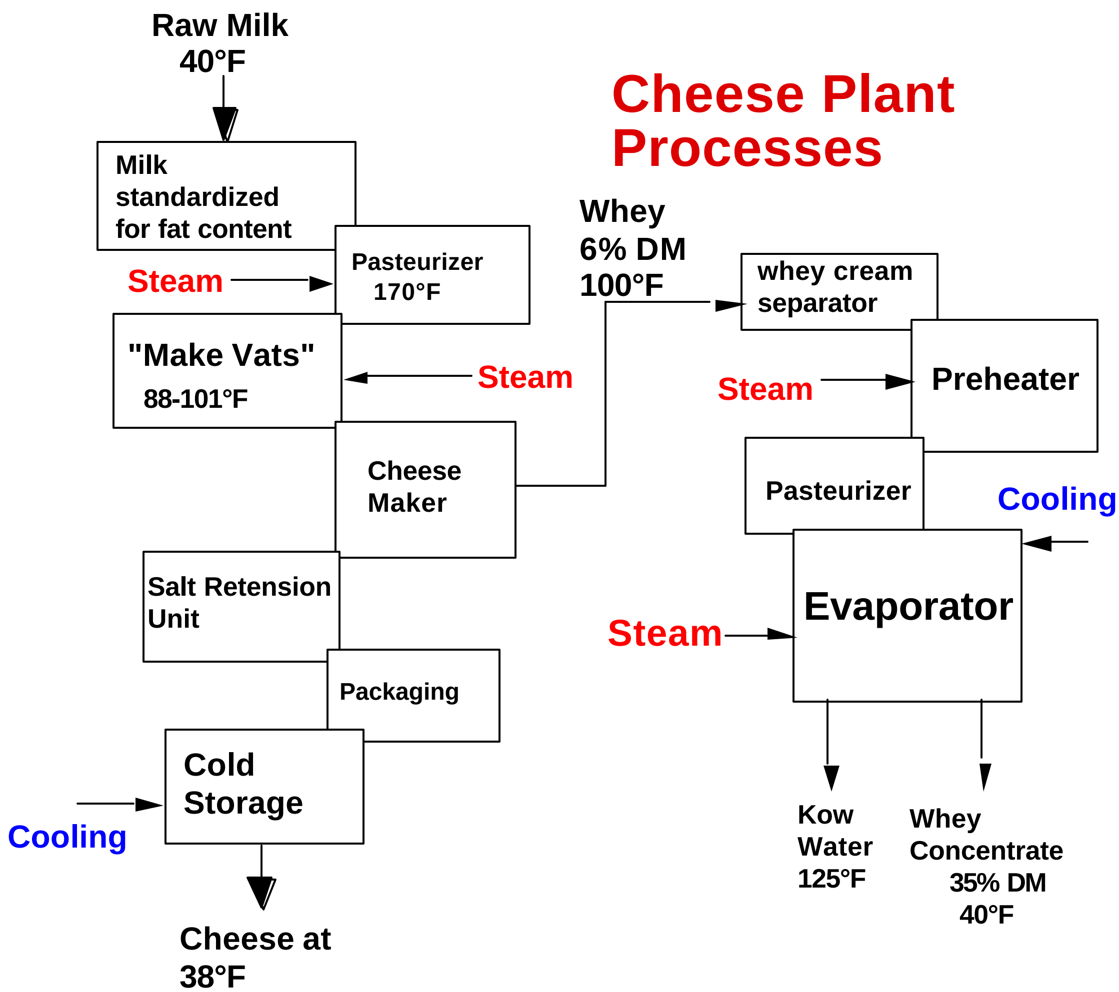

Figure 2.1 Cheese Plant Flow Diagram

Treatment. While the standardizing process is of no interest, certainly the pasteurizing process is important. In a typical HTST operation, the 40°F milk is heated to 165°F using high pressure steam from a boiler. The milk is then cooled down to 88°F in a plate heat exchanger where the heat is transfered to cool raw milk. Due to the very high mass flow rate of milk, this heating requirement and heat rejection requirement are indeed significant. It should be noted, however, that heat recovery opportunities from the 165°F milk are severely constrained since it is now pasteurized.

Make Vats. The 88°F mixture that is pumped into the make vats is carefully heated to 101°F and held at that temperature until the curds are fully formed. The product from these vats is handled at 101°F, so no heat rejection is needed. The heating requirement is minimal since the mixture is heated through only a 13°F temperature rise.

Cold Storage. The finished pallets of cheese are cooled in a large storage warehouse maintained at 38°F. The product holding temperature is reached after a seven day cooling period. Thus a significant cooling load exists in chilling the product. This is in addition to the building envelope, ventilation, and infiltration loads.

Whey Processing. The liquid whey byproduct has market value only if it is concentrated or powdered. For the purpose of concentrating whey, the cheese industry uses falling-film type evaporation systems due to the temperature sensitivity of the product. To fully dry the whey to a powder, condensed whey from an evaporation is fed to a spray dryer. Both of these processes are highly energy intensive due to the heating required.

2.3 Physical Properties of Cheese and Whey

In the analysis that follows, several physical properties of both cheese and whey are required. The values assumed for each of these properties are listed below. (Both the reference values and the approximate empirical relations are taken from Eck, p.331).

2.3.1 Cheese Properties (Cheddar)

Thermal Conductivity	0.1791 Btu / (ft hr °F)
Density	68.05 lb / ft^3
Specific Heat	0.502 Btu / (lb °F)

2.3.2 Whey Properties

The only physical property needed for the whey was its specific heat. Since whey can be of widely varying concentrations, its properties must be indexed to its concentration level. The following empirical relation has been suggest by Eck (where x represents mass fraction):

$$c_p = 1.0 x_{H_2O} + 0.5 x_{fat} + 0.3 x_{solids} \qquad \text{(Btu / (lb °F))}$$

2.4 Description of Plants Studied

What has been described so far has been of a general nature. This study focused on the two particular cheese plants located in Wisconsin that are described below.

2.4.1 Marshfield

The cheese plant at Marshfield produces mostly cheddar along with some mozzarella. Each day the plant consumes 1.6 million lb of raw milk to produce 156,000 lb of cheese. The plant uses an advanced cheddaring system to make the cheese. With advanced cheddaring, a single machine using a series of conveyors drains, mats, mills, and salts the curd (Mesa-Dishington, p.12). Whey is concentrated with a five stage falling-film evaporation system that uses thermal vapor recompression to enhance its steam economy. The plant has a cold storage facility capable of holding 3.5 million pounds of product that is maintained at 38 ± 2°F. Monthly average utility costs are approximately:

Electric:	$19,100
Gas:	$36,500

Based on information supplied by the plant manager at Marshfield, it is estimated that over 70% of the steam generated by the boiler is consumed by whey processing operations. Table 2.2 shows the steam distribution through the plant for each type of steam requiring equipment. These consumption rates occur throughout the 18 hour daily operating period.

Table 2.2 Steam Consumption at Marshfield

Equipment Name	Steam Demand (lb/hr)	lb Steam Req. per lb Cheese
1. Plant Hot Water Heater	1,575	0.18
2. Make Vats	1,250	0.14
3. H.T.S.T.	4,500	0.52
4. Flash Cooler	975	0.11
5. Evaporator Final Heater	1,600	0.18
6. Whey Pre-Heater	4,600	0.53
7. Evaporator	9,500	1.10
TOTAL	**24,000**	**2.77**

2.4.2 Blair

The cheese plant at Blair produces cheddar cheese almost exclusively. The plant consumes 2.2 million lb of raw milk each day to produce 220,000 lb of cheese. The plant uses an Advanced Cheddaring System to make the cheese. Whey is dried to a powder at this plant in a facility that is physically separate from the cheese making process. First the liquid whey is condensed with a four-effect falling-film evaporation system that uses mechanical vapor recompression. The condensed whey is fed to a 22' diameter spray dryer and then is passed through a bed drier. The storage facility at this plant was not studied. Monthly average heating fuel bills for operating the whey drying facility only are $43,700 (summer).

2.5 Conclusions

This study has focused attention on two specific cheddar cheese plants in Wisconsin. The processes analyzed at each plant, though, are somewhat typical for plants of similar capacity. It is hoped that conclusions drawn from the analysis of these plants will be useful as a guide for economizing efforts at other plants of similar capacity.

It is clear that the cost of energy for heating and cooling during the cheese making process itself is relatively small, with the exception of the energy used by the pasteurizer. By a large margin, the biggest cost of energy in a large cheese plant occurs in the concentration and drying of whey. The electric demand required to maintain the cold storage necessary at any plant is also significant. The remainder of this paper will focus specifically in these areas. Chapter three analyses an evaporation system, chapter four analyses a spray dryer, and chapter five examines energy savings opportunities related to the cold storage space.

CHAPTER THREE

EVAPORATOR ANALYSIS

3.1 Description of Marshfield Evaporator

The whey evaporator in use at the Marshfield Plant is known as a five-stage falling-film evaporation system with thermal vapor recompression. This long title will be explained piece by piece.

The heart of any evaporation-based concentration system is essentially a heat exchanger with some hot utility on one side and the material to be concentrated on the other. For temperature sensitive materials of low viscosity, falling-film heat exchangers, or calandrias, are used to minimize the exposure time of the product to the heat source. Such falling-film calandrias are simply shell and tube heat exchangers that are positioned vertically so that as product is distributed to the tube openings at the top of the calandria, gravity pulls the liquid down through the tubes. These tubes are enclosed in a tube chest (shell) that is injected with hot vapor as the heat source. As the vapor condenses around the outer surface of the tubes condensed liquid is drawn downward by gravity where it is pumped off. The calandria tubes at the Marshfield plant are twenty feet tall.

In a calandria such as is described above, one pound of steam can produce one pound of vapor evaporated from the product if the feed is at its boiling point and the steam is saturated. This vapor can then be fed into a subsequent calandria where it can produce an additional pound of vapor. Obviously, to do so requires that the saturation pressure and thus temperature in the second calandria be lower than the first. This sequential pressure drop across a series of calandrias is maintained by vacuum pumps placed after the final effect. This method of placing multiple calandrias in tandem is known as multi-effecting. The Marshfield plant has a five-effect system.

A second technique used to improve the economy of evaporation systems is to displace all or part of the utility steam supplied to the first stage with vapor from the last stage or some intermediate stage. Since the vapor produced further along in the process has a lower saturation pressure it must be recompressed in order to be useful. In practice this compression is achieved either with steam (using a steam jet ejector) or with a mechanical compressor. The use of steam for this purpose is known as Thermal Vapor Recompression (TVR), while the use of mechanical compressors is known as Mechanical Vapor Recompression (MVR). Centrifugal compressors or turbo fans are most commonly used for this purpose.

Two things must be considered when utilizing vapor compression equipment to improve evaporator economy. First, current technology allows a maximum pressure rise corresponding to a 20 to 28°F rise in the temperature of the vapor. This prohibits the use of final stage vapor as a heating source since the temperature drop across evaporation systems generally exceeds 30°F. (The temperature drop across the Marshfield system is approx. 75°F.) This means that the vapor source for recompression is generally an intermediate stage. Drawing vapor from an intermediate stage reduces the amount of vapor that can be produced in all subsequent effects.

To perform an energy analysis and explore the possibility of further evaporation system integration, it is necessary to know the temperatures and flow rates of product, vapor, and condensate at each stage of the system. At Marshfield, all of this information is not available. What is known is the feed rate, the stage temperatures, and the concentration of the final product, as well as the total condensate production rate. To proceed with this analysis, therefore, requires the development of a model to estimate these unknown values. Table 3.1 shows the information that is known for two operating points of the system. A diagram of the Marshfield evaporation system is presented in figure 3.1.

Table 3.1 Operating Data for Marshfield Evaporator

	Operating Point #1 (historical)	**Operating Point #2 (current)**
Daily Milk Use at Plant (lb / day)	1,550,000	1,520,000
Whey Feedrate to Evap. (lb / hr)	81,000	79,500
Rate of Prod. of Whey Concentrate (lb / hr)	14,523	20,408
Feed Concentration (Total Solids)	6.3 %	6.3 %
Total Condensate Production (lb / hr)	64,125	60,010
Product Concentration (Total Solids)	33%	24.3 %
Stage 1 Temperature (°F)	178	176
Stage 2 Temperature (°F)	155	156
Stage 3 Temperature (°F)	136	140
Stage 4 Temperature (°F)	125	128
Stage 5 Temperature (°F)	104	106

Note: Stage temperatures correspond to the steam saturation temperature within the calandria tubes.

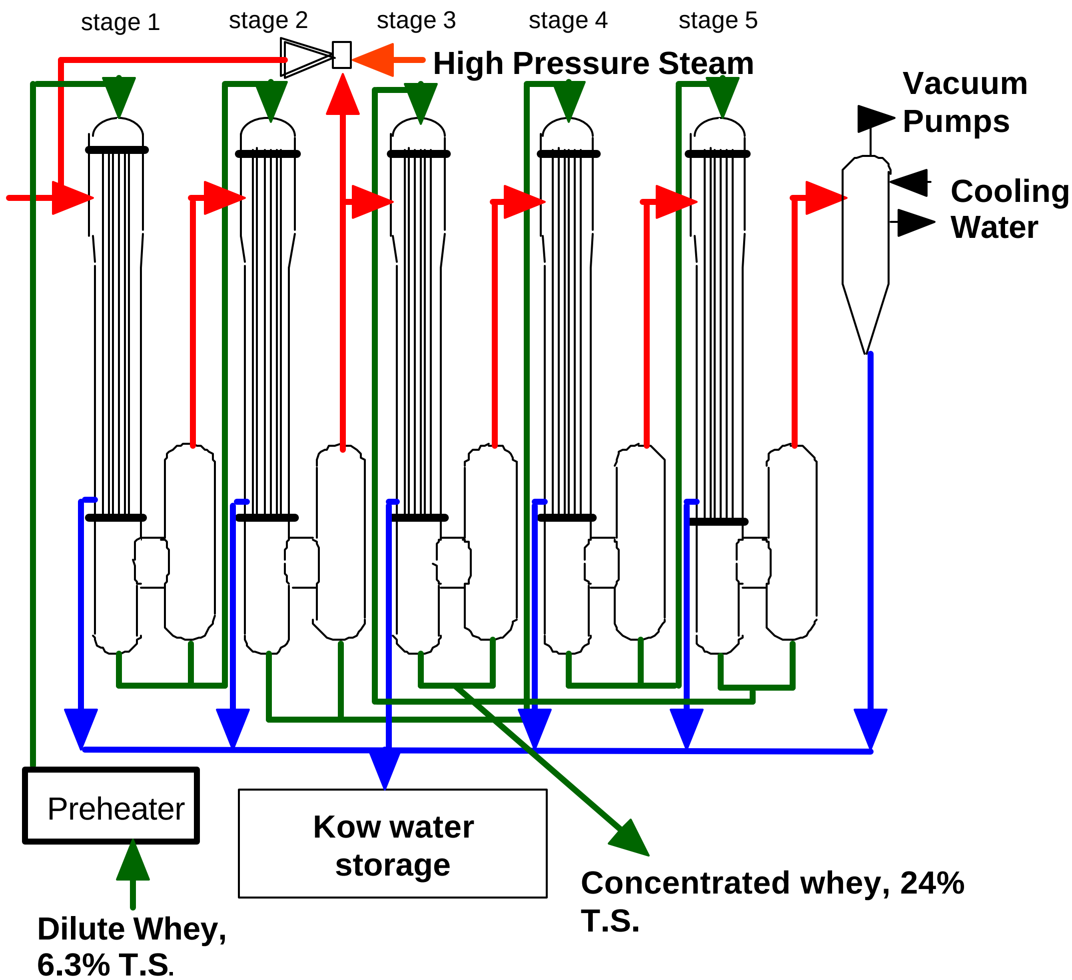

Figure 3.1 Diagram of Marshfield Evaporation System (where T.S. is total solids)

3.2 Theoretical Model of the Marshfield Evaporator

The processes taking place within an evaporation system can all be described with mathematical models. Models have been constructed for the evaporation system at Marshfield to provide estimates for the unknown flow rates when provided with the information that is known. This section describes the components that make up the model. The model described here has been programmed using Engineering Equation Solver (EES), a computer program developed at the University of Wisconsin - Madison to solve sets of algebraic equations (Klein, 1996). A listing of the EES file containing this model is included in Appendix B.

Preheater: The preheater model assumes that the steam supplied leaves as condensate with no temperature change. It is assumed also that the steam is saturated. The resulting energy balance is:

$$\dot{m}_{steam}(h_{steam} - h_{condensate}) = \dot{m}_{whey}\, c_{p,whey}\,(T_{whey,out} - T_{whey,in})$$

Calandria: An individual calandria can be modeled as an open system with three inlet streams and four outlets. The inlet streams include: 1) the whey feed, 2) the vapor feed, and 3) the preheat whey stream (The preheat heat exchange unit on the Marshfield evaporator is not in use and is not included in the model.) The outlet streams are: 4) the vapor raised in that stage, 5) the condensate formed as the feed vapor condenses, 6) the whey, and 7) the preheat whey stream. Figure 3.2 below shows each of these streams.

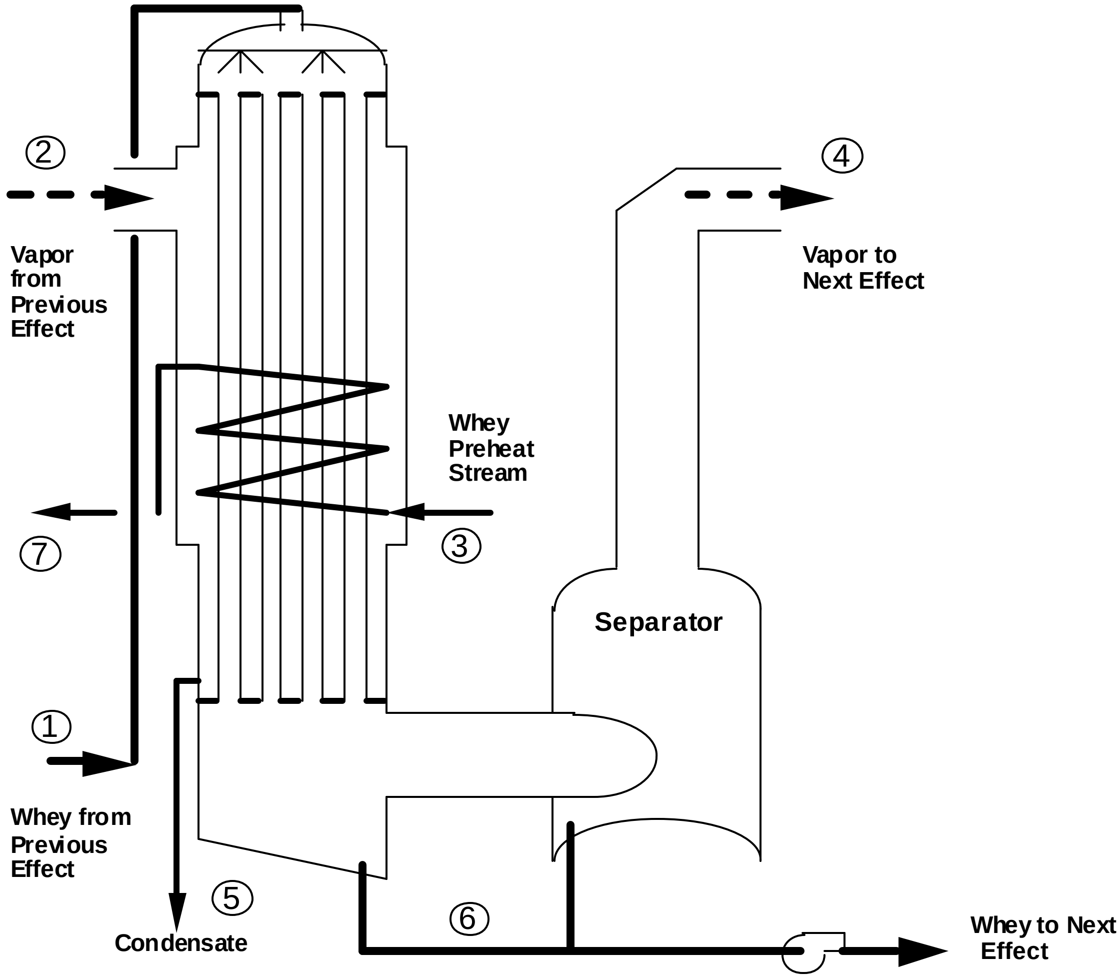

Figure 3.2 Calandria Flow Diagram

A mass balance and an energy balance are required for each stage. These are presented below.

Calandria Mass Balances (steady state)

$$\dot{m}_{whey,in} = \dot{m}_{whey,out} + \dot{m}_{vapor,out}$$

$$\dot{m}_{vapor,in} = \dot{m}_{condensate,out}$$

Calandria Energy Balance (steady state)

$$\dot{m}_{vapor,in}\, h_{vapor,in} + \dot{m}_{whey,in}\, h_{whey,in} + \dot{m}_{preheat,in}\, h_{preheat,in} - \dot{m}_{vapor,out}\, h_{vapor,out}$$

$$- \dot{m}_{whey,out}\, h_{whey,out} - \dot{m}_{cond,out}\, h_{cond,out} \dot{m}_{preheat,out}\, h_{preheat,out} = 0$$

Lastly, the energy balances of the calandrias were modified to account for thermal inefficiencies. A percentage of the vapor introduced to the shell side of each calandria condenses on the outside wall of the unit due to heat loss through the walls, providing no boiling of the product. The heat loss is a function of the temperature difference between the effect and the operating environment. Thus the losses in each effect were modeled as linear relations assuming an ambient temperature of 120°F. The loss coefficient was estimated from the operating point data.

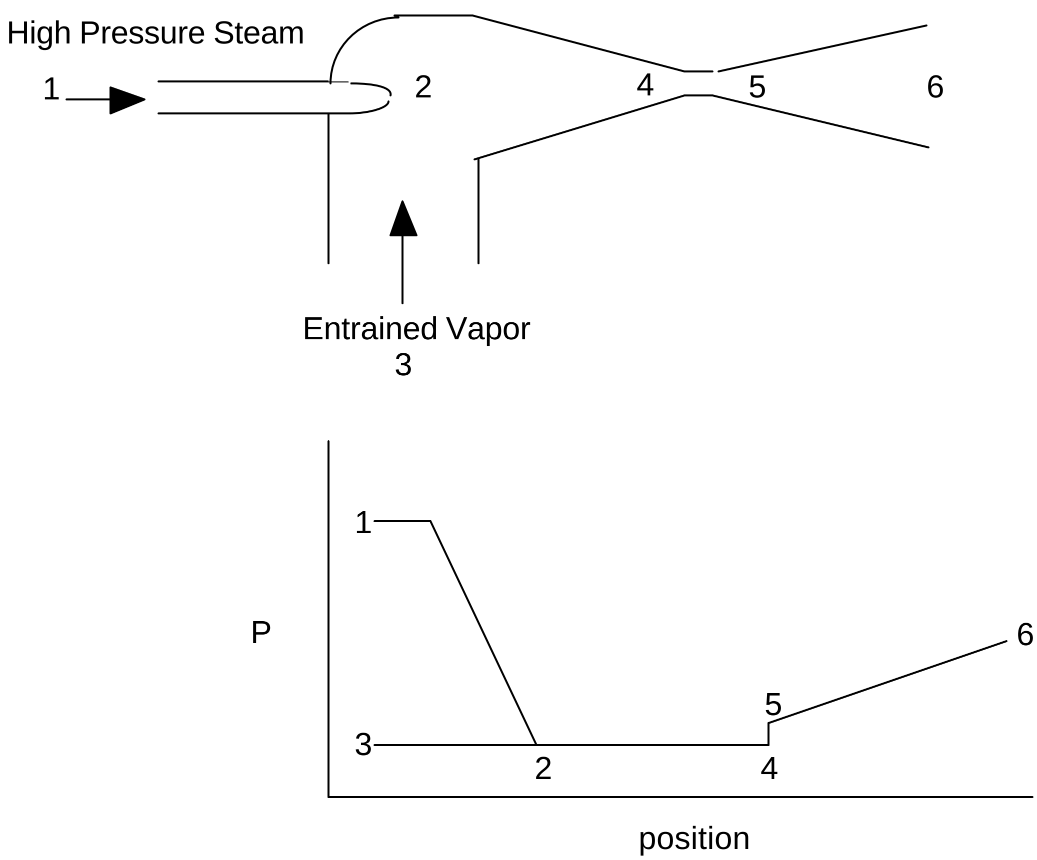

Figure 3.3 Steam Jet Ejector Diagram

Steam Jet Ejector: A steam jet ejector is a device that allows high pressure steam to be mixed with low pressure steam. This mixing is made possible by expanding the high pressure steam, thus creating a high speed low pressure stream capable of entraining a low pressure vapor stream. Figure 3.3 is a diagram of a steam jet ejector along with a plot to indicate the pressure changes occurring along the length of the unit (Stoecker, p. 197)

The first step in modeling a steam jet ejector is to determine the steam velocity as it is flashed to the suction pressure. The inefficiency in this process is accounted for by using an isentropic efficiency value of less than 1 to determine the enthalpy of the expanded stream. The difference in enthalpy of the expanded steam compared to the high pressure steam is attributed to a kinetic energy gain of the expanded steam.

Isentropic Nozzle Efficiency

$$h = \frac{h_{steam,hp} - h_{steam,lp}}{h_{steam,hp} - h_{steam,lp,isentropic}}$$

Nozzle Energy Balance

$$h_{steam,hp} - h_{steam,lp} = \frac{V^2_{steam,lp}}{2}$$

Having done this, momentum, mass, and energy balances can then be carried out to arrive at the enthalpy of the mixture of supply steam and entrained vapor.

Mass Balance

$$\dot{m}_{vapor,mixture} = \dot{m}_{steam} + \dot{m}_{vapor,entrained}$$

Momentum Balance

$$\dot{m}_{steam} V_{steam,lp} = (\dot{m}_{steam} + \dot{m}_{vapor,\,entrained}) V_{vapor,mixture}$$

Energy Balance

$$\dot{m}_{steam}\left(h_{steam,lp} + \frac{V_{steam,lp}^2}{2}\right) + \dot{m}_{vapor,entrained}\, h_{vapor,entrained} = (\dot{m}_{steam} + \dot{m}_{vapor,\,entrained}) h_{vapor,mixture}$$

Condenser: The condenser is modeled to provide the cooling duty required of the cooling tower for heat rejection. The cooling duty is simply the mass flow rate of vapor from the last effect multiplied by the latent heat of vaporization. If the duty on the cooling tower can be reduced, cost savings associated with partial capacity operation can be realized.

Flash Cooler: As the concentrated whey enters the low pressure environment within the flash cooler, flashing (or instantaneous vaporization) takes place. This vaporization absorbs energy from the whey stream, thus causing it to cool rapidly. This effect can be represented with the following mass and energy balances.

Mass Balance

$$\dot{m}_{whey,\,in} - \dot{m}_{whey,out} - \dot{m}_{vapor,\,out} = 0$$

Energy Balance

$$\dot{m}_{whey,\,in}\, h_{whey,in} - \dot{m}_{whey,\,out}\, h_{whey,out} - \dot{m}_{vapor,out}\, h_{vapor,out} = 0$$

3.3 Results of Evaporator Model

In the complete model, one parameter is not known. This parameter is the ratio of the mass flow rate of steam supplied to the steam jet ejector divided by the mass flow rate of vapor entrained. The model solves for this when a final concentration value of the concentrated whey is provided. The resulting entrainment ratio estimated by the model is 0.76. This value is in excellent agreement with manufacturer performance data for a suction steam saturation temperature of 140°F and a temperature rise of 28°F (Graham Manufacturing, Inc. data indicates an entrainment ratio of 0.79)

As previously mentioned, the purpose of developing the model has been to estimate flow rates for each of the flows within the process. Table 3.2 below provides the results of the model for each of the operating points for which data is available.

Table 3.2 Flow Rate Predictions of EES Model

Flows (lb / hr)	Operating Point 1	Operating Point 2
Vapor Produced in #1	21,693	20,232
Vapor Produced in #2	21,363	19,368
Vapor Produced in #3	8,566	6,201
Vapor Produced in #4	8,132	6,237
Vapor Produced in #5	6,722	5,541
Vapor Entrained from #2	11,875	11,875
Mass Flow Whey into #2	59,306	58,603
Mass Flow Whey into #3	23,089	27,390
Mass Flow Whey into #4	37,943	38,631
Mass Flow Whey into #5	29,811	32,200
Mass Flow Final Product	14,523	20,408

3.4 Minimum Utility Target

In this study, a methodology known as pinch analysis (Linnhoff, p. 33) has been applied to determine the theoretical limits to heat integration. In pinch analysis all of the streams requiring heating are collapsed in to a composite “cold” stream. Likewise, all of the streams requiring cooling are collapsed into a composite “hot” stream. These composites are modified to account for the minimum temperature difference necessary for effective heat transfer and plotted together on Temperature v. Total Enthalpy coordinates. The cold composite is adjusted laterally so that the two composites meet at one point. The temperature at this point is called the “pinch” temperature. The significance of the pinch point is that, to arrive at theoretical minimum utility demand for a process, heat should not be transferred across the temperature associated with the pinch. Heat pumps, on the other hand, should only be utilized such that energy from below the pinch is delivered at a temperature above the pinch. This methodology is summarized in Appendix A. The spreadsheets that produced these plots are presented in Appendix C.

The first, and perhaps most important, step in performing a pinch analysis is identifying the set of streams available for integration. Included here are three separate analyses of the Marshfield evaporation system. The three are included to indicate the evolutionary approach to identifying the most useful selection of streams to consider.

3.4.1 Iteration 1

The first iteration excluded only the condensate streams. This was done since the condensate is valuable at the temperature at which it is produced. The streams included are listed below. The resulting pinch plot is shown in figure 3.4.

1. Preheat of the raw whey (cold)

2. Whey evaporation in each stage (5 cold streams)

3. Vapor condensation in each stage (5 hot streams)

4. Cooling of concentrated whey (hot)

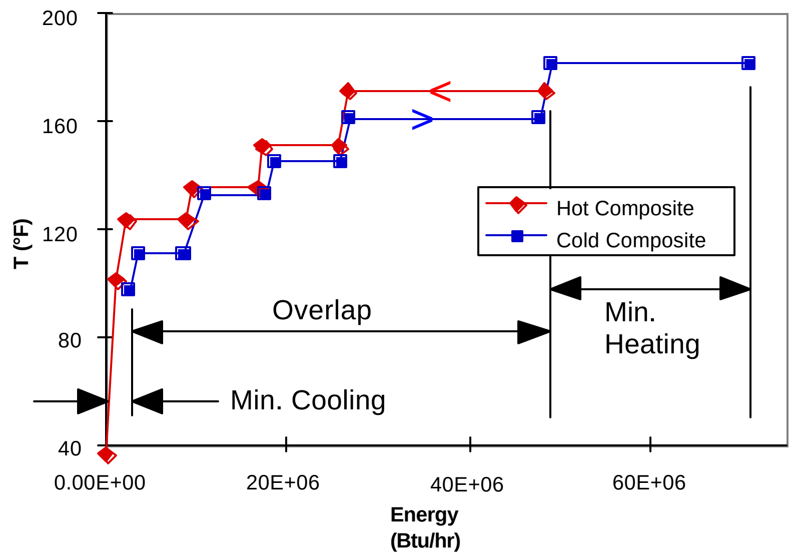

Figure 3.4 Pinch Plot -- First Iteration (? T_{min}=10°F)

Figure 3.4 is a good example of what a well integrated set of process streams should look like. The region labelled "overlap" shows how the heat rejection from the hot composite is used to heat the cold composite stream. In a pinch diagram, it is the degree to which these composites do not overlap that is significant. The magnitude of the non-overlapping parts of the composites represents the minimum heating or cooling required from an outside source. In this diagram, this amount is relatively small. Based on this diagram, it was determined that, since we are not interested in changing the evaporation and condensation streams within the calandrias, these would best be left out of the analysis so that attention could be focused on those streams with practical opportunity for heat exchange.

The theoretical minimum utility requirements in this case are:

Minimum Cooling = 0.25 x 10^7 Btu / hr

Minimum Heating = 2.26 x 10^7 Btu / hr

These values are then compared with the current utility usage for this set of streams:

Current Cooling = 4.83 x 10^7 Btu / hr

Current Heating = 6.84 x 10^7 Btu / hr

3.4.2 Iteration 2

For the second pinch analysis, the evaporation and condensation streams within the calandrias were excluded with the exception of the vapor which must be condensed following the final effect. The streams included in this analysis are listed below. The resulting pinch plot is shown in figure 3.5.

1. Preheat of raw whey
2. Cooling of the final product
3. Condensation of vapor from stage 5
4. Preheating of supply water for cleaning use

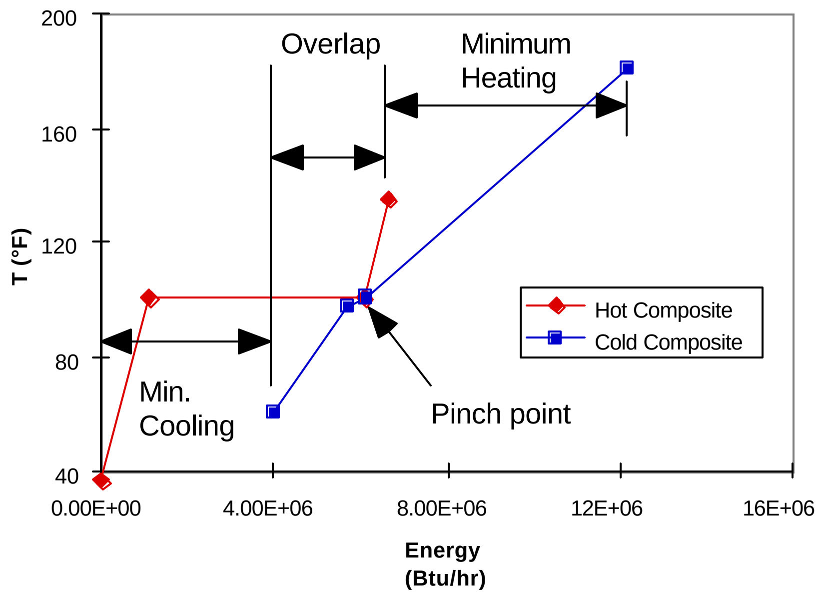

Figure 3.5 Pinch Plot -- Second Iteration (? T_{min}=10°F)

Iteration 2 demonstrates that, although the heating and cooling requirements of the process are substantial, the opportunity for sensible heat transfer only displaces a small portion of the heat rejection requirement. To fully displace the heat rejection load on the cooling tower, one would need to consider a heat pump to deliver the excess energy to a temperature above the pinch. There is not sufficient capacity above the pinch, however, to absorb all of the available heat. So this arrangement would be less than ideal.

The theoretical minimum utility requirements in this case are:

Minimum Cooling = 4.0 x 10^6 Btu / hr

Minimum Heating = 5.56 x 10^6 Btu / hr

These values compare with the current utility usage for this set of streams:

Current Cooling = 6.64 x 10^6 Btu / hr

Current Heating = 8.20 x 10^6 Btu / hr

3.4.3 Iteration 3

Upon examining the results of iteration 2, it became apparent that for the streams considered, the potential for sensible heat integration is limited. It is apparent that the final stage vapor possesses a significant amount of heat to be rejected, but that due to its low temperature (106°F) it requires the presence of a low temperature sink to be useful. Even at the higher temperature a heat pump could elevate it to, there exists insufficient sink to utilize it fully. Looking beyond the evaporation system to the cheese making operation, the hot utility requirement of the pasteurizer becomes an obvious potential sink for this low temperature heat source. For this to be feasible though, it would be necessary to confront an important health consideration. Since the vapor is at low pressure, there exists the potential for leakage of raw, unpasteurized milk into the condensate stream which would render the condensate unfit for cleaning purpose. Heat recovery equipment would need to be designed to prevent any leaks from occuring.

This iteration, then, is simply a repeat of iteration 2 with the addition of the raw milk heating stream from 40°F to 96°F. The results are shown in figure 3.6.

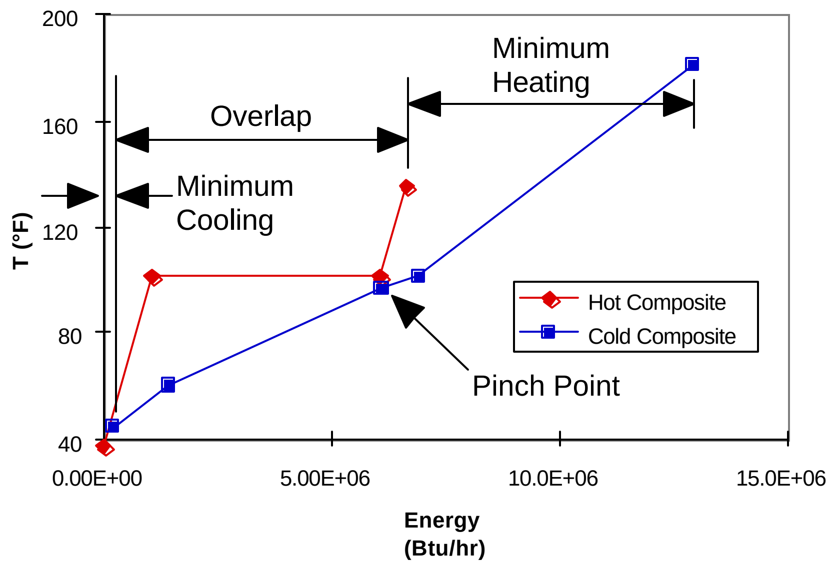

Figure 3.6 Pinch Plot -- Third Iteration (? T_{min}=10°F)

In this diagram, the minimum cooling requirement has been reduced to almost zero. Should it be possible to capture fully this integration potential using heat transfer equipment, the cooling tower could be removed from the process. About one-half of the energy required to heat the cleaning water would be supplied by heat recovery.

The theoretical minimum utility requirements in this case are:

Minimum Cooling = 0.20 x 10^6 Btu / hr

Minimum Heating = 6.64 x 10^6 Btu / hr

These values compare with the current utility usage for this set of streams:

Current Cooling = 6.64 x 10^6 Btu / hr

Current Heating = 12.7 x 10^6 Btu / hr

3.5 Integration Possibilities

The pinch diagrams of iterations two and three indicate not only the theoretical minimum heating and cooling requirements, but they also prove useful in suggesting ways in which these limits can be approached. Both diagrams identify a pinch temperature of 101°F. From both of these diagrams it is also clear that the majority of heat rejection requirement is due to the condensation of 5th stage vapor (the long flat line on the hot composite.) Condensation of this vapor represents a heating potential of 4.98 x 10^6 Btu / hr. So the 106°F vapor is the important heat source below the pinch point.

There are two cold streams below the pinch in iteration 2. The first is the proposed heating of city water to be stored for use during off-production cleaning cycles. These cleaning cycles require approximately 97,500 gallons of city water per day. For use, the water must be heated from the temperature at which it is received (approx. 55°) to anywhere from 140°F to 185°F, depending on the cleaning cycle for which it is used. With 106°F vapor, this water could be heated to 96°F, allowing a 10°F minimum temperature difference. Doing so would require the use of storage tanks (which are presently available and unutilized). To distribute this preheating load evenly through the operation period means that a flow rate of approximately 45,300 lb / hr be heated through the entire 18 hour operating period. This proposed sink could absorb 1.86 x 10^6 Btu / hr. This represents 37.3% of the heat provided by the condensing fifth stage vapor.

Also below the pinch is a very small segment of the whey preheat stream. Since the whey is received at 92°F, only a 4°F temperature rise could be achieved while allowing for a 10° minimum temperature difference. This proposed sink could absorb approximately 3.02×10^5 Btu / hr. This represents only 6.07% of the heat provided by the condensing fifth stage vapor.

Given the scenario in iteration 2, then, sensible heat exchange between the final effect vapor and a stream of city water to be stored for cleaning is suggested, along with preheating the feed whey slightly. However, these stream matches meet only 43.4% of the heat rejection required to condense the fifth stage vapor. The additional heat rejection could be done above the pinch with the application of a vapor recompression heat pump.

Current technology permits up to a 30°F temperature rise for recompressed vapor. Inspection of the pinch diagram for iteration 2 reveals that there is significant capacity for accepting heat at 136° or below. This is simply the energy difference between the hot and cold composites at the temperature of 131°F. This amounts to approximately 1.8×10^6 Btu / hr, or 36.1% of the heat rejection required.

Taken together, the heat pump, the preheating of the whey, and the preheating of the city water can utilize 79.5% of the energy given up by the condensation of stage 5 vapor. The condenser and cooling tower would still be necessary, but would service a heat rejection duty of only 20% of its current requirement.

Installing a heat pump, though, involves a very large first cost due to the compression equipment required. A heat pump option may not be cost effective with relatively inexpensive energy costs and should only be considered in the absence of less expensive first cost sensible heat exchange options.

Iteration 3 was undertaken after realizing the need to identify other low temperature heat rejection opportunities. Iteration 3 contains three cold process streams below the 106° pinch. In addition to the two streams described above, iteration 3 proposes preheating the raw milk before it reaches the pasteurizer. Preheating the raw milk from 42°F to 96°F, for instance, requires 4.38 x 10^6 Btu/hr. This accommodates 88.0% of the required heat rejection. Including this stream provides for sufficient "sink" to completely eliminate any heat rejection requirement, even without attempting the slight preheating of whey that was suggested in iteration 2. This suggests that the cooling tower that is currently accomplishing this heat rejection could be eliminated and energy transferred to the incoming milk rather than to the environment.

So the most attractive arrangement seems to be splitting the 106°F vapor from fifth effect into flows to two shell and tube heat exchangers. In the first, raw milk could be heated from 40°F to 96°F just prior to pasteurization. In the second, city water from the mains would be heated from 55°F to 96°F. It is suggested that a control strategy be designed to vary the flow rate of city water to maintain this temperature rise. This is necessary since the flow rate of vapor is not constant over the entire operating period.

Preheating the raw milk using condensing vapor from the evaporator does involve health considerations, however. The condensate produced in the evaporator is used in the first cycles of the off-production cleaning process. The condensate from the condenser is mixed with the other condensate streams. Should the condenser be replaced with a heat exchanger using milk as a fluid, the potential exists for raw milk to leak into the condensate stream since the condensing vapor is at lower pressure than the milk. If a leak occurred, the condensate from the final stage vapor would no longer be acceptable for cleaning purposes. This may actually be advantageous, though, because it is the lowest temperature condensate stream. Removing it from the stored condensate mixture would result in the balance of the condensate being held at a higher temperature. And there is more condensate produced each day than is required for cleaning purposes.

3.6 Estimated Savings

Once the integration opportunities have been identified, an economic analysis is necessary to determine the extent to which capturing the savings potential is economically feasible to pursue. The savings derived must be balanced against the first cost of investing in the equipment necessary to derive such savings.

An estimate of the savings derived from the heat exchange opportunities must include both the reduction in boiler demand as well as the avoided cost of operating the cooling tower. The boiler savings can be estimated from the heating energy avoided over the course of a year. There is a reduction of 4.98×10^6 Btu / hr of heating, and the plant operates 18 hours per day, 290 days per year. Over the course of a year, this amounts to 26.0×10^9 Btu. Since 1 therm = 1×10^5 Btu, this represents 2.60×10^5 therms. At $0.28 per therm, represents an annual savings of $72,800.

The cooling tower is equipped with a 60 hp motor that operates under partial load. Energy savings accrued from idling the cooling tower can be estimated by assuming 50% load for 18 hours each day that the evaporation system is in operation. This represents 1.17×10^5 kWh per year. At $ 0.027 per kWh, this comes to $ 3,150 per year. This brings the total utility savings to $76,000.

The capital investment cost necessary to capture these savings must also include several items. Two shell-tube heat exchangers (condensers) must be purchased and installed. Vacuum pumps will be needed to maintain the shell side at the low pressure necessary for the 106°F vapor. Such vacuum pumps are already in use with the current condenser and can likely be transferred. A 100,000 gallon storage tank will be required to hold the preheated city water until it is needed for the cleaning cycles.

While reliable estimates for the savings are available, reliable figures for the total capital costs would require equipment cost estimates from manufacturers. For this reason, the estimated savings will be used to demonstrate how much could be spent to achieve a reasonable payback on investment. Figure 3.7 below indicates how much could be invested to achieve payback within any investment period chosen, from 2 to 20 years.

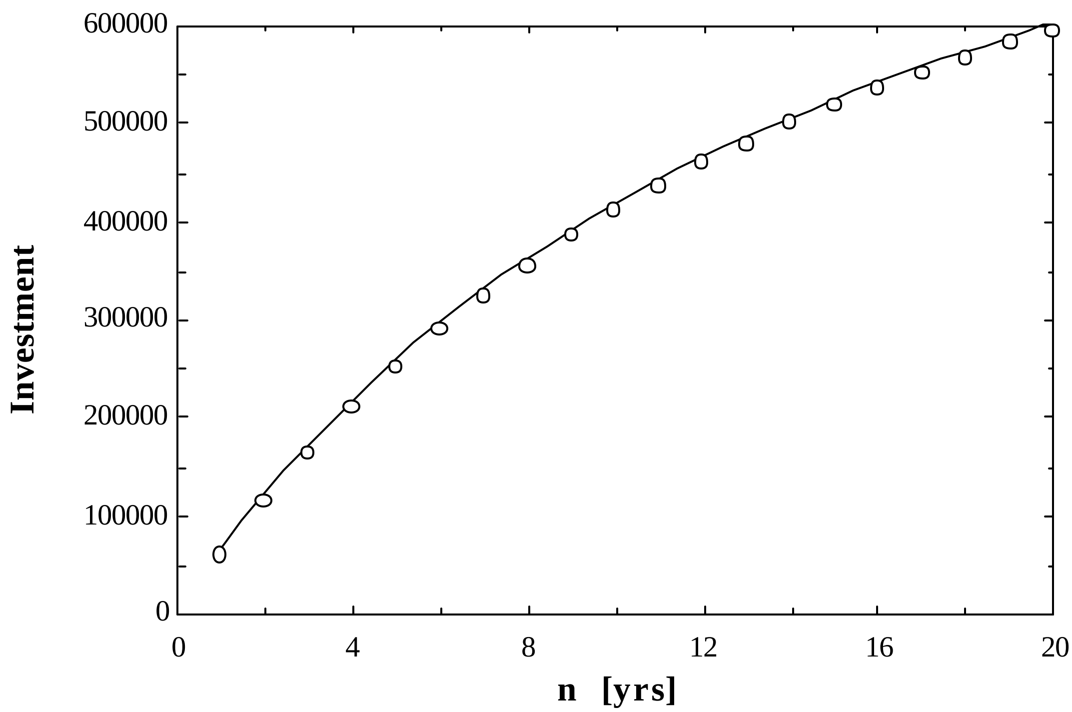

Figure 3.7 Payback Period vs. First Cost for Condenser Retrofit

From figure 3.6 it can be seen that investing $117,000 will achieve a 2 year payback. The plant manager could invest up to $414,000 if a ten year payback period were acceptable.

3.7 Conclusions

To evaluate the potential for heat recovery from the Marshfield evaporation system, the evaporator was modeled to fully identify the streams present. Pinch analysis methodology was employed to evaluate the potential for thermal integration among the streams considered. The pinch studies were done to explore results using different sets of potential streams.

The results of the pinch study have led to the conclusion that there exists an opportunity to exploit the heat removed from the condensing vapor from the last effect of the evaporator. Transferring this energy to the raw milk prior to pasteurization and to city water that will be stored for off-production cleaning can be done using readily available heat exchange equipment. Doing so not only reduces the boiler load during and after production, but also eliminates the need for operating a large cooling tower.

The estimated savings have been used to determine how much money the plant manager could invest to receive a variety of payback periods. For instance, if a ten year payback was required, an investment of $414,000 on equipment could be made.

CHAPTER FOUR

SPRAY DRYER ANALYSIS

4.1 Description of Blair Spray Drier

Nearly 62% of the whey that was processed in 1989 was for the production of dry whey for either human or animal consumption (ADPI Survey, p. 17). The production of dry whey is accomplished almost exclusively using spray dryers to remove the moisture remaining after preprocessing by evaporation or filtration. In a spray dryer, the product to be dried is introduced to the drying chamber in liquid droplet form. This is done either by spraying under very high pressure or by using an atomizer. At the same time, 240°F dry air is blown into the chamber. In this arrangement, there is a large driving potential for mass transfer due to both the large difference in vapor pressure between the dry air and the liquid whey as well as the large surface area of whey droplets exposed to air. The residence time in the chamber refers to the time required for the droplets to fall, due to gravitation and air flow, through the chamber. The amount of drying is, therefore, a function of the driving potential (vapor pressure differential), the mass transfer coefficients , and the residence time in the chamber.

The cheese plant considered at Blair uses a spray dryer with a central chamber that is 22 feet in diameter and 30 feet tall. The droplet residence time is approximately 25 seconds. From the outlet of the large vertical unit where the powder emerges at a moisture content of 11%, it is placed on a conveyer and delivered to a vibrating bed dryer for processing to remove the remaining moisture (see figure 4.2).

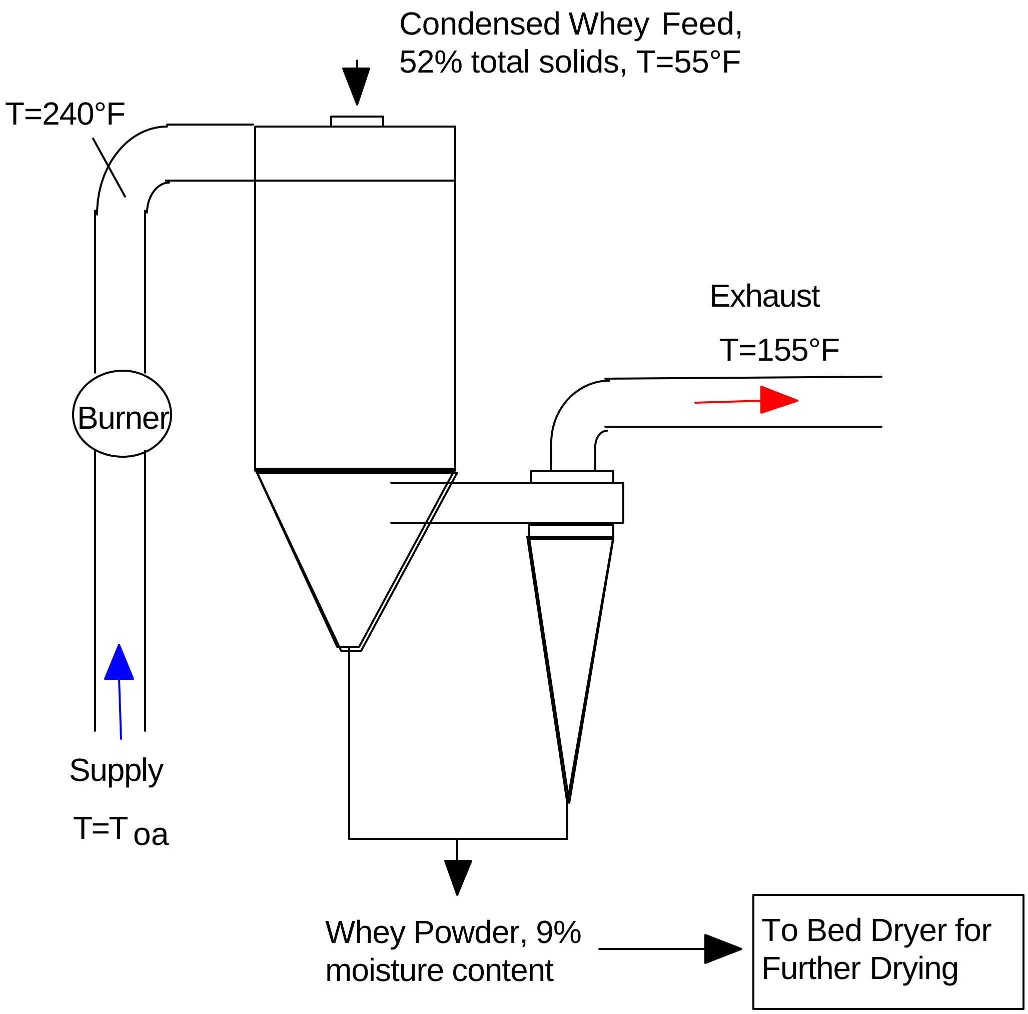

Figure 4.1 Current Configuratiuon of Spray Dryer Streams

Both the vertical drying chamber and the vibrating bed dryer are supplied with outdoor air that has been heated to 240°F ± 10°F. The heated air temperature is controlled to achieve consistent product moisture content.

4.2 Minimum Utility Target

The pinch analysis under the conditions described above is very simple. Only five streams are considered available for further integration. Table 4.1 provides the relevant data for each of these streams. The streams available for integration include:

1. The cool condensed whey
2. The supply air to the vertical unit
3. The supply air to the bed unit
4. The exhaust air from the vertical unit
5. The exhaust air from the bed unit.

Table 4.1 Process Streams Included in Blair Analysis

Stream	Flow Rate (lb / hr)	Specific Heat (Btu / lb hr °F)	Initial Temp. (°F)	Final Temp. (°F)
Cond. Whey	12,000	0.8	55	?
Vertical Supply Air	244,200	0.24	T_outdoor	240
Bed Supply Air *	66,156	0.24	T_outdoor	240
Vertical Exhaust Air	244,200	0.24	155	?
Bed Exhaust Air *	66,156	0.24	156	?

*** not shown in figures**

Table 4.1 indicates that several of the final temperatures of the streams are unknown. Under current operating conditions, these temperatures are known. However, it is not important that they remain at the present values. Using heat integration techniques may result in different values for these temperatures without impact on the performance of the dryer. The pinch plot of this stream set is shown in figure 4.1. The pinch diagram presumes that the exhaust streams are cooled to 100°F and that the whey is warmed to 125°F. In reality, these values are dependent on the effectiveness of the heat exchangers used.

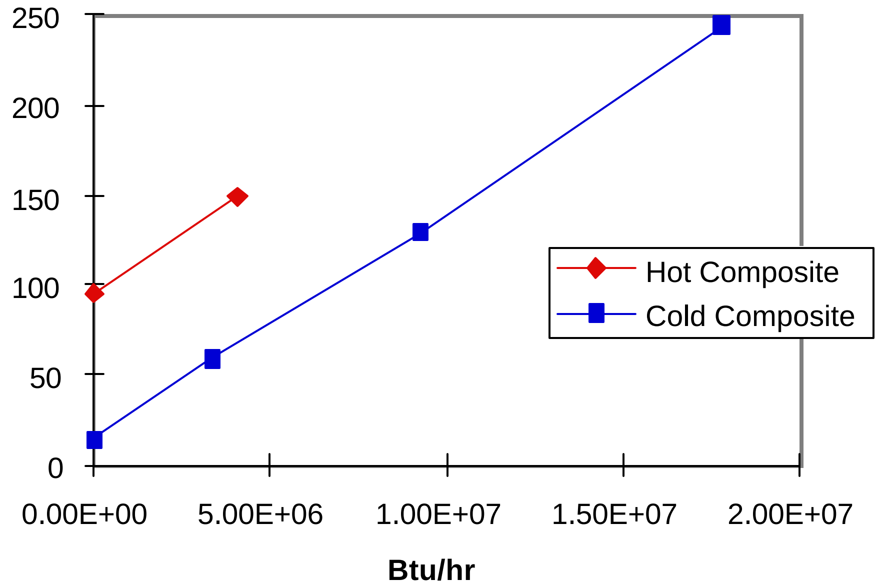

Figure 4.2 Pinch Plot for Blair Spray Dryer
(For outdoor air temperature of 10°F)

No "pinch" temperature is revealed in the plot shown in figure 4.1. Therefore, there are no restrictions on the temperatures of streams matches for heat exchange. Any hot stream can be cooled to exchange heat with a cooler stream. The overlap of the hot and cold composites indicates the potential for 4.10 x 10^6 Btu/hr of heating that can be displaced through integration when the outdoor air temperature is 10°F. Possibilities for achieving this integration potential are considered next.

4.3 Integration Possibilities

The capacitance rates of the supply and exhaust air streams of the spray dryer and bed dryer are approximately equal. For this reason, matching of the supply and exhaust air streams for heat exchange has been the focus of this study, rather than preheating the whey feed. Two alternatives are suggested and investigated for accomplishing this heat exchange. In the first, a cross-flow heat exchanger is used to transfer heat directly from the exhaust to the supply air streams. The second possibility entails indirect heat transfer from the exhaust streams to the supply streams utilizing an intermediary fluid. A third possibility is suggested that involves recirculation of a fraction of the exhaust air to mix with outdoor air as supply to the dryer.

The diagrams that follow portray only the large vertical spray dryer. Nevertheless, the configurations suggested for the vertical drying unit can be extended to the bed dryer as well. While the volume flow rate of the bed dryer is less than 30% of the flow for the vertical unit, its entering supply and leaving exhaust temperatures are analogous.

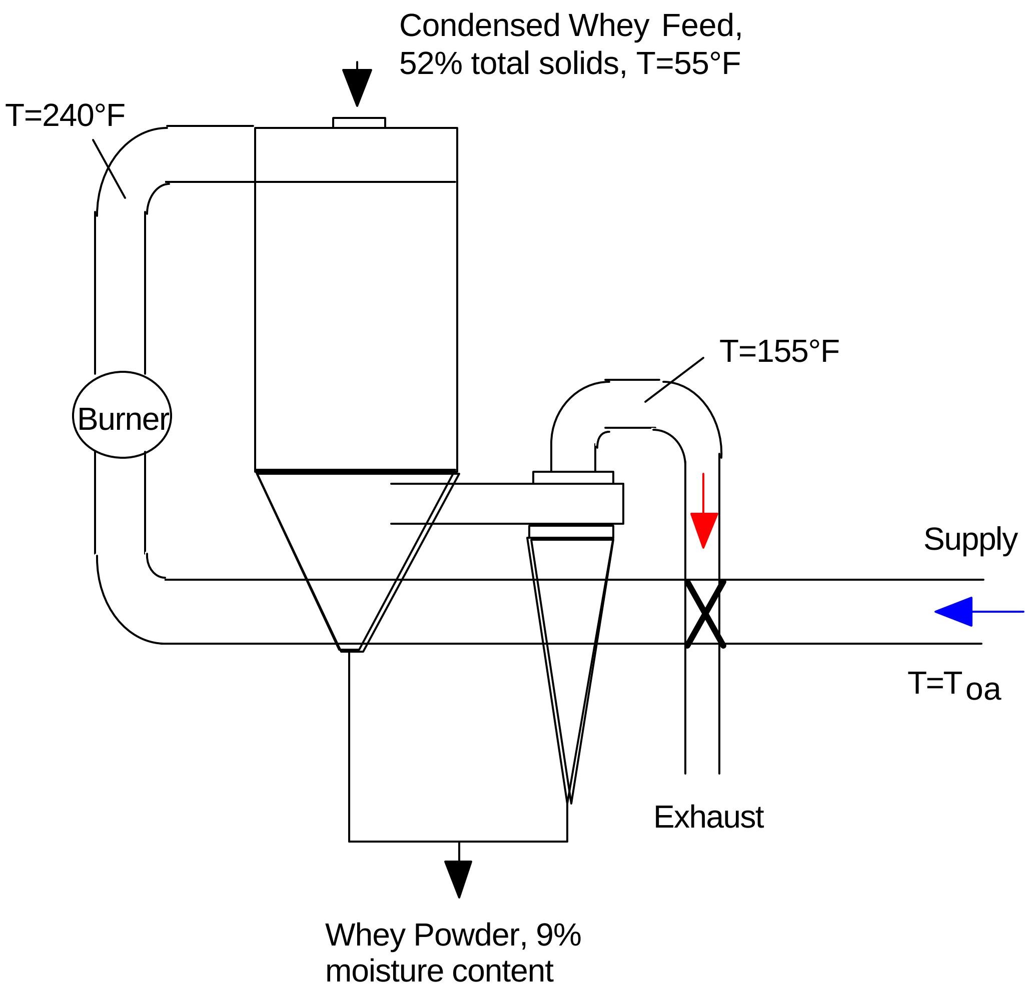

Figure 4.3 Cross-flow Heat Exchanger Installation

4.3.1 Cross-flow Air-to-Air Heat Exchanger

The simplest manner for heat transfer to take place between the exhaust air stream and the supply air stream is by using a cross-flow heat exchanger. To refit an existing plant with a cross-flow heat exchanger would, however, require significant modification of the ducting arrangement. Use of a cross-flow heat exchanger would also require a control strategy to allow for a defrost cycle during winter months when the freezing of condense from moist discharge air within the unit would likely occur. A diagram of a spray dryer implementing a cross-flow heat exchanger is shown in figure 4.2.

The maximum amount of energy savings achievable using a cross-flow heat exchanger is ideally the same as for the runaround loop heat exchange system described next. Detailed thermal analysis has been done for the runaround system. The difference between the two options is only the first cost of installation.

4.3.2 Run-around Loop

Ideally, to exploit waste heat from the exhaust air stream of a spray drier, the use of an air-to-air cross-flow heat exchanger would be suggested. However, making use of an intermediary transfer fluid can greatly simplify the equipment modification necessary. For this reason, the run-around loop configuration shown in figure 4.3 is suggested. An additional benefit of this system is that the need to defrost the heat exchanger in the supply air stream is likely to be reduced since the secondary fluid temperature will be lower than the exhaust air temperature (thus causing reduced condensate production.)

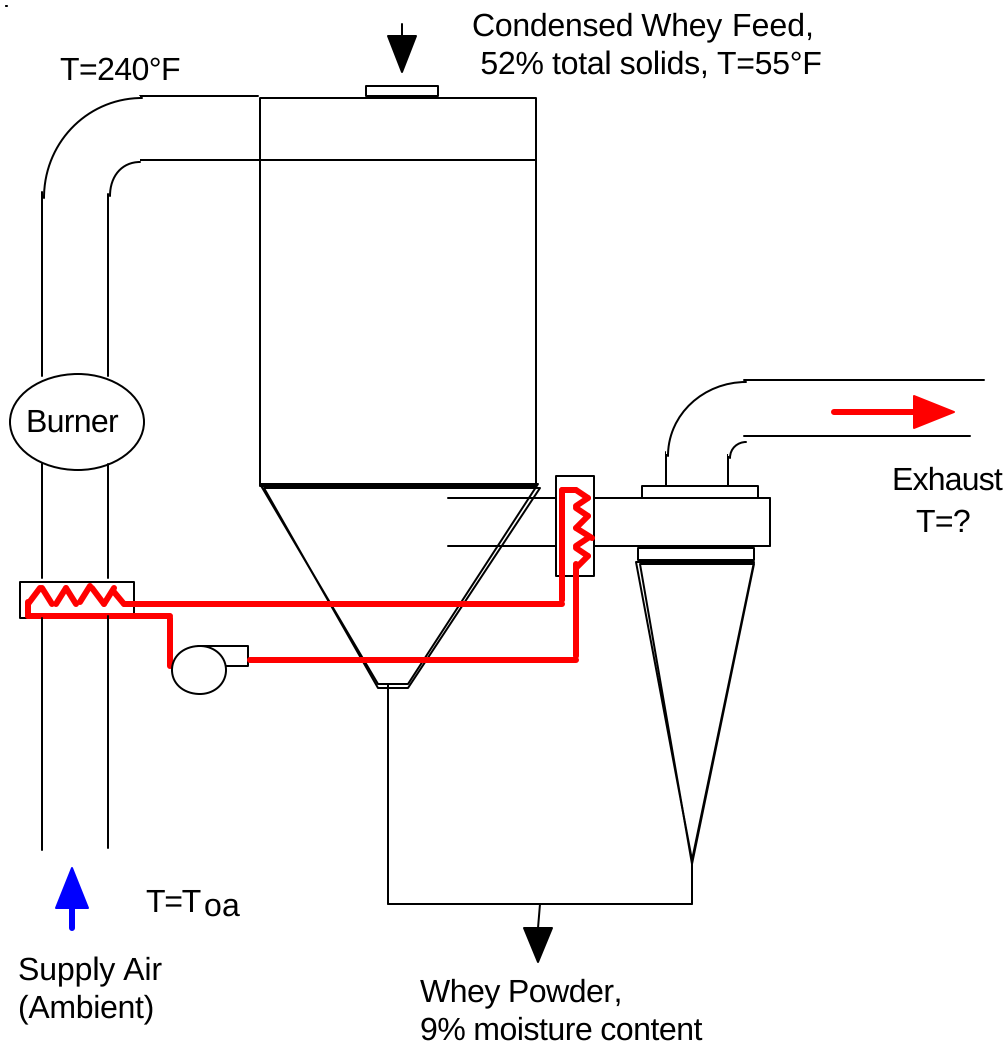

Figure 4.4 Run-around Loop Configuration

To explore the benefits of the run-around configuration shown in figure 4.3, a model of the system was developed using TRNSYS. TRNSYS is an computer program developed at the University of Wisconsin to perform transient yearly simulations using detailed weather data. The TRNSYS type (Fortran subroutine) that models heat exchangers is based upon the Kays and London mathematical description.

For a cross-flow heat exchanger, this is:

if $C_{max}=C_h$

$$g = 1 - \exp\left(\frac{UA}{C_{\min}}\frac{C_{\min}}{C_{\max}}\right)$$

$$e = 1 - \exp\left(-g\frac{C_{\max}}{C_{\min}}\right)$$

if $C_{min}=C_h$

$$g = 1 - \exp\left(-\frac{UA}{C_{MIN}}\right)$$

$$e = \frac{C_{\max}}{C_{\min}}\left(1 - \exp\left(-g\frac{C_{\min}}{C_{\max}}\right)\right)$$

in either case,

$$T_{ho} = T_{hi} - e\left(\frac{C_{\min}}{C_h}\right)(T_{hi-}T_{ci})$$

$$\dot{Q}_T = eC_{\min}(T_{hi} - T_{ci})$$

The TRNSYS model utilizes this heat exchanger description to calculate the hourly heat transfer from the exhaust stream to the supply stream over the course of an average year. The simulation results, therefore, provide an estimate of energy savings over the course of a year as a function of the capacity of the heat exchangers used. The model was run a series of times in which the capacity (UA) was parametrically varied. The results of this simulation series are shown in figure 4.4.

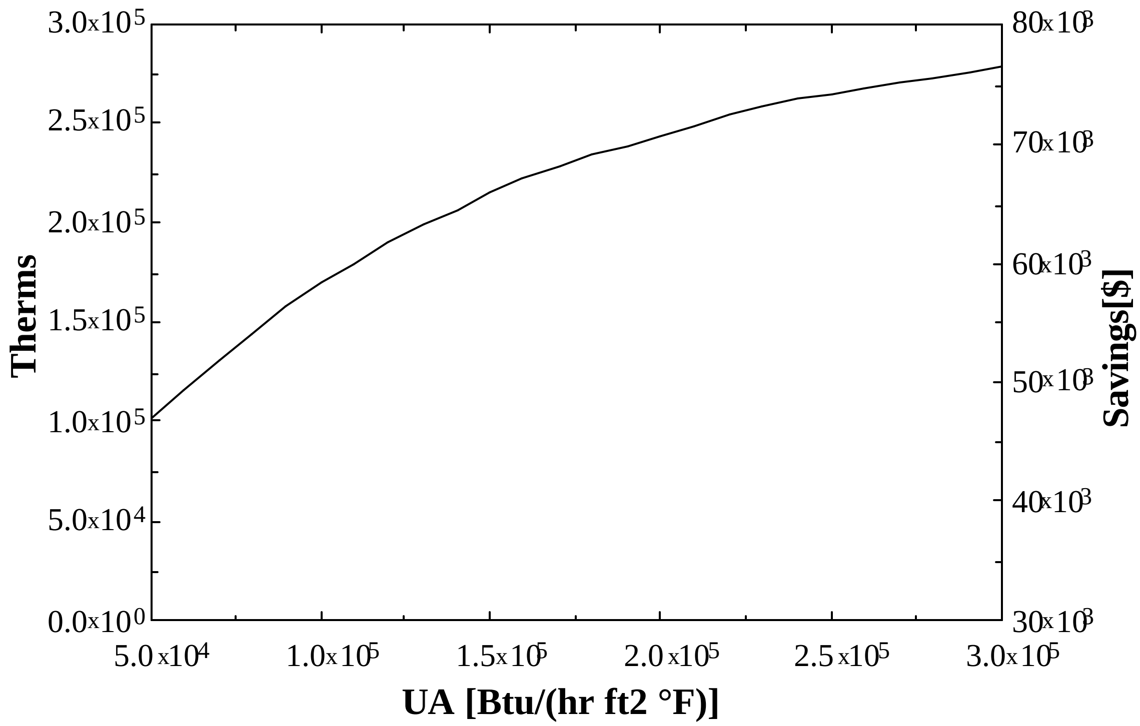

Figure 4.5 Annual Fuel Savings v. Heat Exchanger Capacity

Based upon figure 4.4 the economic feasibility of the opportunity can be evaluated once installation costs are known. Payback period can easily be determined for any combination of heat exchanger cost and capacity.

4.3.3 Exhaust Recycle

A third possibility for integrating the spray dryer involves recycling a portion of the exhaust air for reuse in the supply stream. An example of such a configuration is shown in figure 4.5. By mixing a fraction of the exhaust with outdoor air before heating, the supply air temperature and humidity are increased. In this way the load on the burner is reduced.

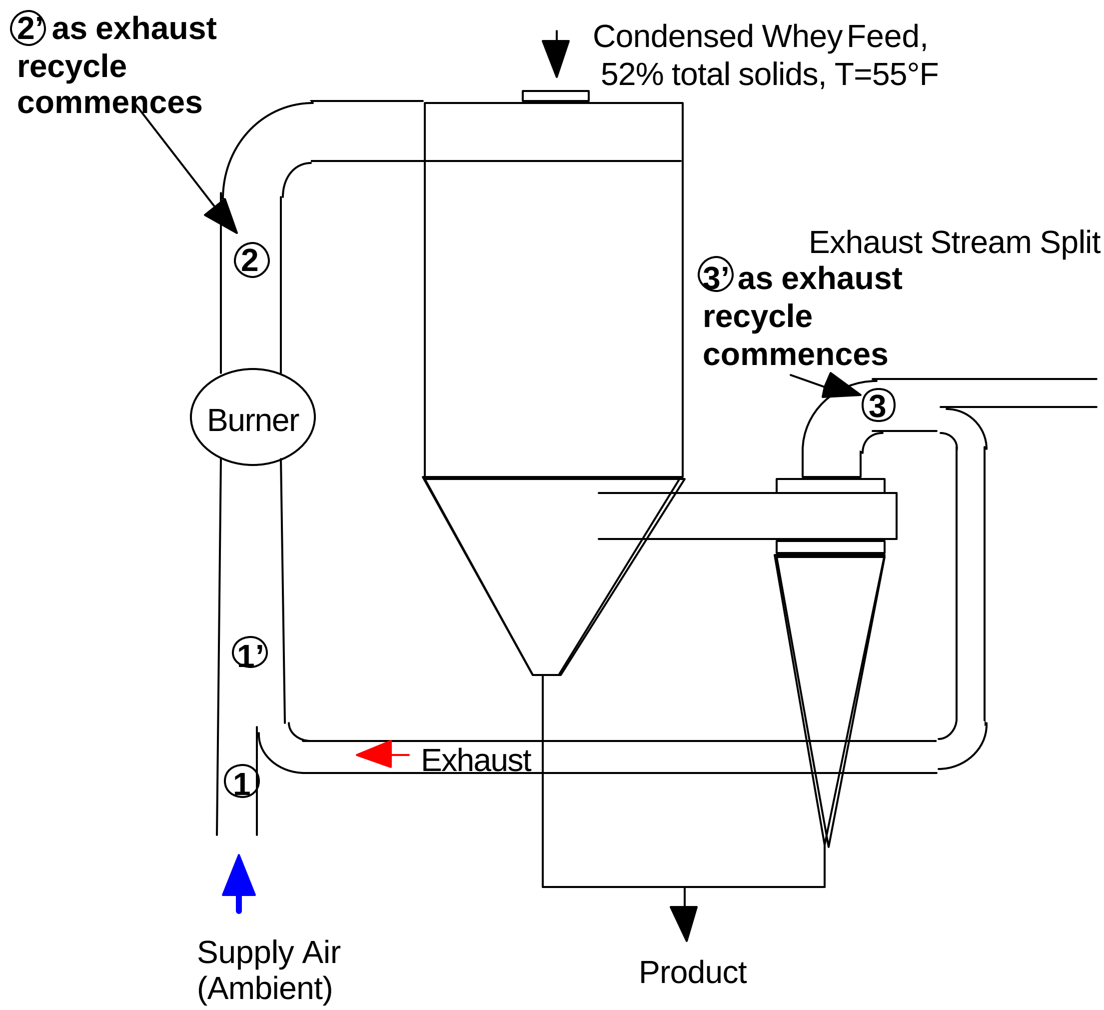

Figure 4.6 Diagram of Spray Dryer Exhaust Recycle Configuration (Circled numbers used to reference state points shown on figures 4.5 an 4.6)

To evaluate this option, one must consider both facets of the driving force within a drying operation. Not only is the temperature of the air blown into the chamber significant, but also its humidity level. The exhaust air will necessarily have a higher humidity ratio than the supply air. Mixing exhaust air with the outdoor air will increase the humidity ratio of the air entering the dryer. This will reduce the rate of drying . In fact, this effect is well known to plant operators. They report a significantly greater drying capacity during winter months when the humidity ratio of the outdoor air is very low as compared to summer months.

This process can be understood by considering the various state points on the psychrometric diagrams shown in figures 4.6 and 4.7. The state points referred to are shown in figure 4.5. To accomplish effective drying, the relative humidity of state point 2 should be as low as possible. Figure 4.6 demonstrates that with exhaust recovery, the relative humidity of the heated air flow will be elevated.

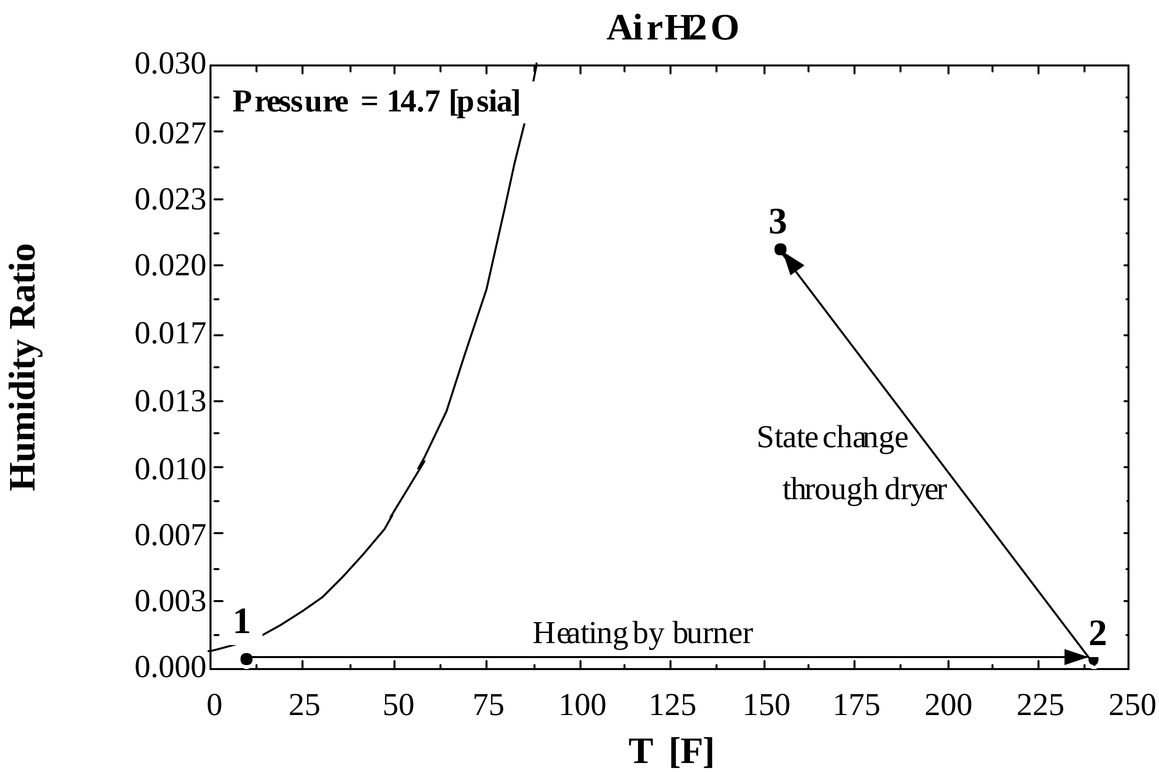

Figure 4.7 State Points of Air in Spray Dryer Under Current Operation

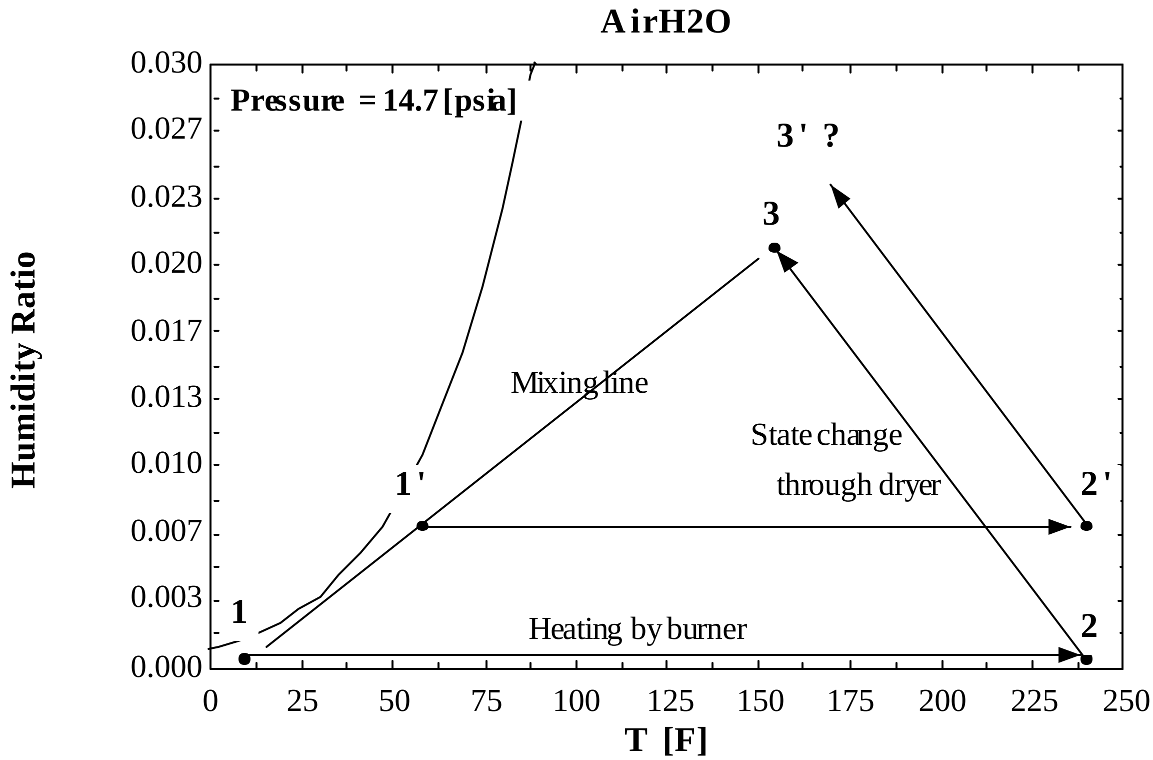

Figure 4.8 State Points in Spray Dryer Using Exhaust Recycle

For reasons just described, it is necessary to develop a detailed model the spray dryer to determine how variations in the supply air state affect the burner load. Previous work in this area has been summarized and extended by Zbicinski et. al. (1987). The Zbicinski mathematical model attributes to each particle some amount of ambient air, creating an air-droplet system. For each system, momentum, mass and energy balances are performed. Since the spray equipment produces droplets of varying diameters, the distribution of droplet sizes produced must be known in order to fully solve the model.

The droplet size distribution of the spray system at Blair is not known, nor is it possible to make reasonable assumptions about the distribution in the absence of empirical data. For this reason, further examination of recycling the exhaust was deemed beyond the scope of this study.

4.5 Conclusions

Spray dryers are commonly used in the production of whey powder. These dryers consume a significant amount of the natural gas used by a typical cheese plant or whey processing facility.

Potential for recovery of waste heat has been identified at the spray dryer in use at the Blair cheese plant examined in this study. Exhaust streams from both the main drying unit and the secondary bed dryer contain recoverable amounts of thermal energy. Annual simulation of heat exchange equipment transferring heat from the exhaust streams to the supply streams of each of these dryers has demonstrated the potential for saving up to $80,000 per year in natural gas costs (see figure 4.4).

CHAPTER FIVE

WAREHOUSE DEMAND SAVINGS

5.1 Introduction to Demand Shifting

As our economy's demand for electricity grows, the capacity of electricity providers must expand as well. A utility must always have either generation potential or access to outside sources of electricity to meet the highest possible demand. Unfortunately, this demand is not distributed evenly through the course of a day or through the course of a year. To meet this peak demand, which may exist only for a short period during the air-conditioning season each summer, a utility may be required to invest capital in generation facilities that are under-utilized. It would be more desirable to have fewer plants operating steadily than to have many plants, some of which operate only sporadically.

Presented with this situation, utilities have designed rate structures to "encourage" customers to reduce their peak demands. First of all, customers are assessed a demand charge based on their highest demand for electricity either in the preceding month or in the preceding year. Second, the rates charged for use during the day, when demand is greatest, are significantly higher than for electricity used during the night. The rate schedule of the electric provider for the Marshfield cheese plant is as follows:

Off-Peak Billing Rate: $0.0220 per kWh

On-Peak Billing Rate: $0.027 per kWh

Peak Hours: 7:00 a.m. through 9:00 p.m., October through May

8:00 a.m. through 8:00 p.m., June through September

Demand Charge: $4.60 per kW of on-peak billed demand

*In addition, a customer charge of $100.00 per month is assessed.

Rate structures such as this have led to strategies by consumers to meet their electricity needs for less money, just as the utilities had intended. One way this has been done is through the use of thermal storage. For example, many commercial buildings now operate their cooling equipment through the night and “store” the cooling provided by producing ice. This ice can then be melted through the day to meet the cooling needs of the building.

Thermal storage systems can be classified as either full storage or partial storage. With a full storage system, all of the cooling load can be met by operating the chiller only during the off-peak period. Partial storage, on the other hand, means that the cooling equipment must be operated during the on-peak period but under reduced load. The remainder of the load is met with cooling stored during off-peak operation. The term “no-storage” is sometimes used to refer to a conventional system in which the cooling system operates at all times to meet present loads.

In this section, the potential for using cheese as a thermal storage medium is explored. The off-peak cooling is used to sub-cool stored cheese so that during the on-peak period a cooling load can be met as this cheese returns to its normal storage temperature.

5.2 Description of Warehouse and Demand Profile

Every cheese plant has associated with it a cold storage space in which the cheese is cooled and stored at low temperature. The ventilation and cooling equipment required to condition the air in this space accounts for a significant demand for electricity 24 hours a day, 365 days a year. The product cooling load is not a factor that can be changed, so the cooling requirements cannot be reduced. However, opportunity may exist for reducing the amount of money spent to meet this cooling load.

Finished cheese leaves the production area at a temperature of approximately 97°F. From this temperature, the product must be cooled to below 40°F. Current practice for 42.7 lb boxes of cheese requires that this cooling take place over a period of seven days. Long-term storage of cheddar cheese is commonly done at a temperature of 38°F. The storage temperature and duration of cooling period are both factors that can effect the shelf life of the final product.

At the Marshfield cheese plant studied, two cold storage areas exist. Together, these spaces have a maximum storage capacity of 3.5 million lb of cheese. On average, though, there is 2.8 million lb of cheese in storage, and the amount in storage never falls below 0.5 million lb.

The cooling load within the warehouse has a number of components. The largest cooling load is due to the daily addition to the warehouse of 156,000 lb of warm cheese. A number of smaller loads contribute as well. These include: the ventilation load, the envelope load, the occupancy load, the equipment load, the lighting load, and a latent load resulting from moisture absorbed by the cardboard packaging prior to entering the cold storage area. Estimates have been made for all of these. Figure 5.1 is a histogram showing the expected cooling demand for the average day of each month. The histogram makes apparent the conclusion that the constant cheese cooling load is the dominant load on the cooling system. The season fluctuation in the load is, nonetheless, significant. The largest demands occur during the hottest part of the summer.

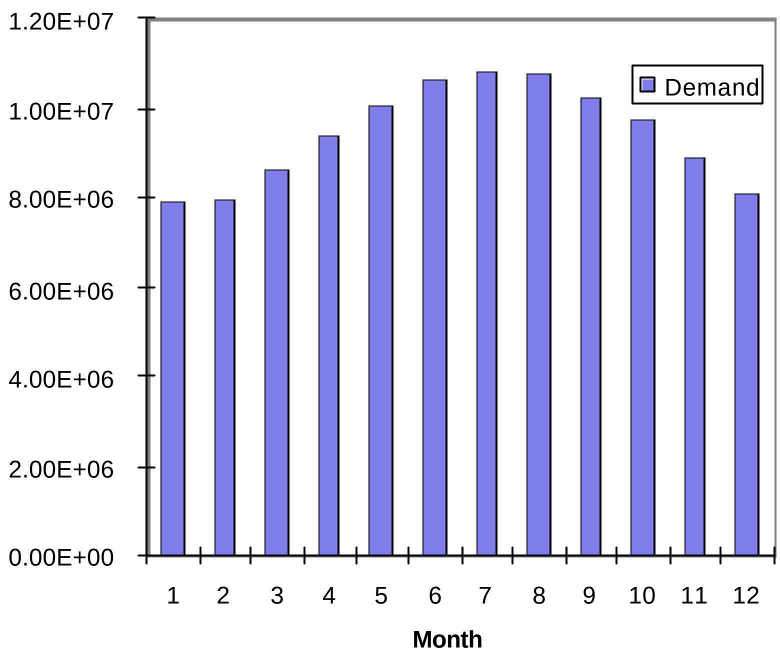

Figure 5.1 Monthly Average Daily Cooling Load For Marshfield Warehouse

The assumptions upon which this load histogram are based will now be described. The spreadsheet from which it was generated is presented in appendix D. First, while there are seven daily production amounts of cheese at different stages of cooling, the total temperature change for one day's production is used to estimate the total daily cooling load. So the daily sensible product cooling load becomes:

$$Q_{cooling} = m_{daily,prod.} c_{p,cheese} (97 - 38)$$

In addition, since the packaging occurs in a humid environment at a high temperature, it must be assumed that the cardboard packaging material absorbs moisture. This water will be released through evaporation in the cool, dry storage room, and represents a substantial latent load. A sensible heat ratio (SHR) of 0.6 has been selected. This relatively high SHR has been chosen to conform the load estimate in this analysis to the operating conditions observed at the plant.

To estimate the cooling load resulting from heat gain through the building envelope, the ASHRAE Sol-Air temperature method has been used. By this method, the cooling requirement is represented by the following relation:

$$Q_{envelope} = UA(T_{sol-air} - T_{warehouse})$$

where U is the conductance of the envelope in Btu / (hr ft^2 F), A is the surface area in ft^2, and $T_{sol-air}$ is a temperature difference in Fahrenheit used to account not only for outdoor air temperature but also for the solar radiation heat gain. Conductance values for the roof are taken as those of ASHRAE roof #1, steel deck with 3.33 in. of insulation. Conductance values for the walls are taken as those of ASHRAE wall #1, steel siding with 4 in. of insulation.

Table 5.1 Load Estimation Parameters

Load Component	Suggested Parameter
Infiltration	0.7 Airchanges / 24 hours
Occupancy	20,200 Btu / 24 hours

Equipment	2500 Btu / hr (per HP)
Lighting	1.25 W / ft^2

Estimates of the occupancy, infiltration, lighting, and equipment loads have been arrived at using the methodology suggested in the Refrigeration Load Estimating Manual of the KRACK corporation. Parameters taken from this manual are presented in table 5.1

A final aspect of the cheese cooling process relates to the packaging of the cheese. At the conclusion of the production process, the cheese is sealed in plastic and placed in cardboard boxes. The average weight of these boxes is 42.7 lb. The dimensions of these boxes are shown in figure 5.2. These boxes are then placed on pallets to facilitate transportation and storage. 54 boxes are placed onto each pallet. The boxes are stacked six high and arranged “chimney style” to allow for the circulation of air through the pallet to expedite cooling.

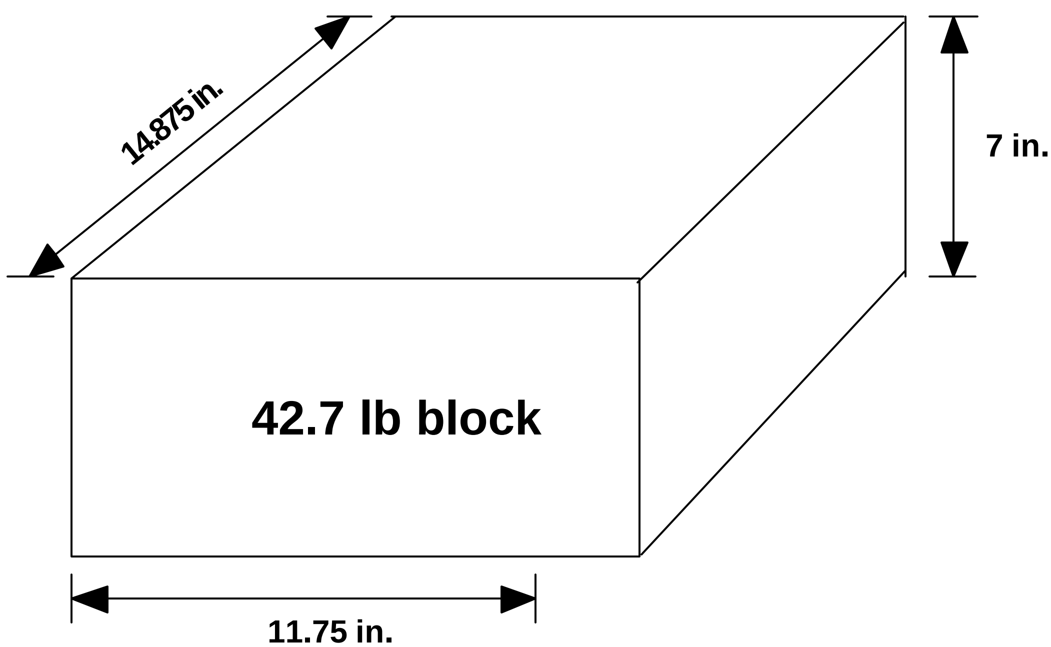

Figure 5.2 Dimensions of Packaged Cheese Block

5.3 Thermal Analysis of Stored Cheese

5.3.1 Description of Warehouse Model

To evaluate whether or not stored cheese is a feasible thermal storage medium, a finite difference model has been developed. This model involves seven individual finite difference terms representing a days production of cheese at each 24 hour interval of the seven day cooling period. These are coupled to a finite difference model of the cooled cheese in the warehouse. Adiagram of the thermal circuit modeled is shown in figure 5.3. Added to the energy balance are terms for the envelope load, the ventilation load, the lighting load, and the equipment load. This model is then solved for hourly intervals over a 24 hour period under design conditions. The final energy balance becomes:

$$Q_{refrigeration} = \sum_{i=1}^{7}(m_{prod,daily} c_{p,cheese} \frac{T^{+}_{cheese,i} - T_{cheese,i}}{\Delta t}) * (\frac{1 - SHR}{SHR})$$
$$+ m_{bulk} c_{p,cheese} \frac{T^{+}_{cheese,bulk} - T_{cheese,bulk}}{\Delta t}) - Q_{ventillation} - Q_{lighting} - Q_{motors} - Q_{envelope}$$

The SHR term above is included so that the latent load is accounted for in the energy balance. The delta T term represents a one-hour interval.

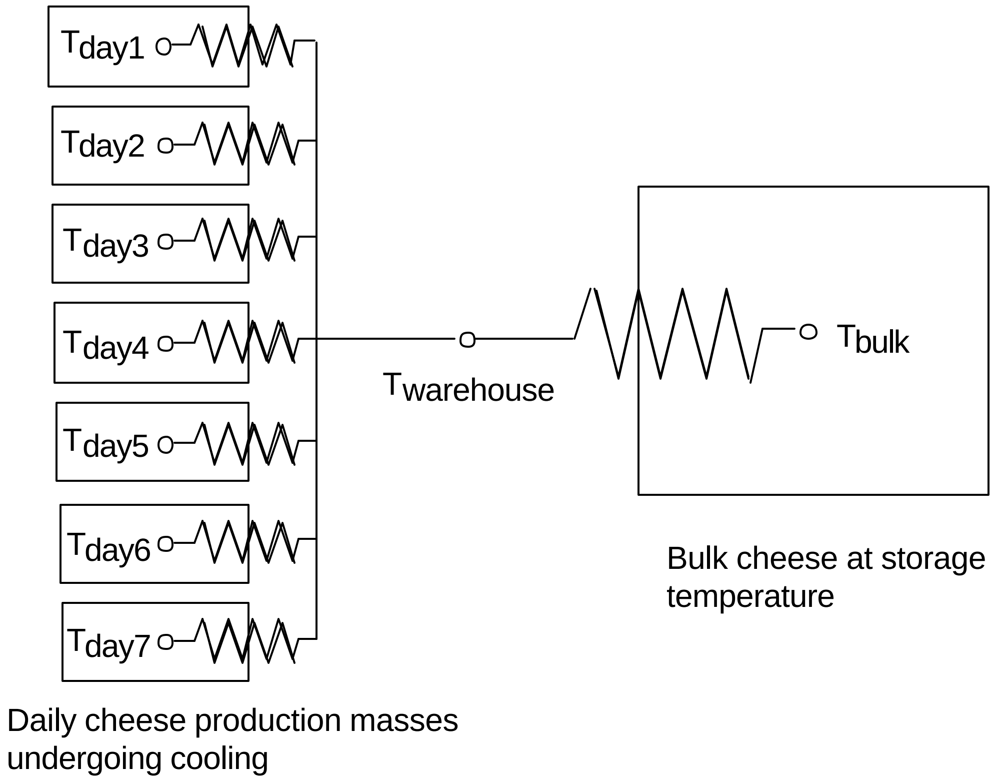

Figure 5.3 Thermal Circuit of Warehouse Model

The capacitances are treated as lumped. This means that the cheese blocks are assumed to be at uniform temperature at each time step. While this is clearly not a reasonable assumption to make, it greatly facilitates the modeling effort. The degree of error inherent in making this assumption is examined in the next section, and has been found to be acceptable for the purpose of this study.

Rate equations for each of the capacitances in the energy balance are needed to complete the system of equations. Under the lumped parameter treatment, Newton's law of cooling provides the necessary relation between the heat flows to or from each cheese mass and the temperature change of the mass. Thus,

$$Q = h_{effective} A_{\exp osed} (T_{warehouse} - T_{cheese})$$

While these rate equations complete the system of equations, two parameters that appear in the rate equations must be estimated. These are $h_{effective}$, the effective heat transfer coefficient from the cheese surface, and $A_{exposed}$, the surface area of the stored cheese that is exposed to air. Determination of these parameters is discussed next.

5.3.2 Determination of Heat Transfer Parameters

Determination of $A_{exposed}$ is based upon the geometry of the cheese boxes and the method in which these boxes are placed on pallets. Since the pallet area upon which the blocks are placed is greater than the bottom surface area of nine blocks, it is possible to arrange the blocks to allow for the circulation of air around the sides of each block. The sides of the cheese blocks are, therefore, assumed to be exposed. Due to stacking, the tops and bottoms of the cheese boxes are mostly covered by other boxes. But since the arrangement of adjacent layers is varied slightly, a small portion of the tops and bottoms of the boxes is assumed to be exposed. Also the tops of the boxes on the top layer of each pallet are exposed. Assuming that 10% of the tops and bottoms of the remaining blocks are exposed, it can be shown that 163.7 ft^2 of surface is exposed for each pallet of cheese. Given that each pallet holds 54 blocks at 42.7 pounds each, this translates into 0.071 ft^2 of exposed area per pound of cheese.

The general result for a lumped capacitance treatment yields the following relation for the temperature of an object after a certain period of time, t:

$$\frac{T_t - T_\infty}{T_{initial} - T_\infty} = \exp\left[-\left(\frac{hA}{rVc_p}\right)t\right]$$

Since the cooling period for cheese is known, as well as the exposed area, the mass of the cheese blocks, and the specific heat of cheese, this equation can be used to determine the effective heat transfer coefficient, $h_{effective}$. For a 42.7 lb block cheese with a seven day cooling period, a value of $h_{effective}$= 0.127 Btu / hr ft2 F has been found. This value is used to represent both the surface heat transfer resistance as well as the internal resistance.

5.3.3 Validity of the Lumped Capacitance Method

The assumption of a uniform temperature distribution through a solid is reasonable accurate only if the Biot number for the solid is less than 0.1. To calculate the Biot number requires the thermal conductivity of the material, the critical length of the material, and the surface convection coefficient. The thermal conductivity of cheddar cheese is 0.179 Btu / hr ft R. The critical length for a 42 lb cheese box (dimensions shown in figure 5.2) is found by dividing the volume by the surface area. It is 0.141 ft for the boxes commonly used. Determination of the heat transfer coefficient is described next.

Heat transfer from stacked boxes in a warehouse involves both convective transfer and radiative heat transfer. Faces of the box in direct contact with an adjacent box must be considered to be adiabatic. Natural or free convection relations have been employed to determine the convective coefficient. These are as follows:

for a vertical surface: $$\overline{Nu_L} = \left\{0.825 + \frac{0.387\,Ra_L^{\frac{1}{6}}}{\left[1 + (0.492/\Pr)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right\}^2$$

for the top: $$\overline{Nu_L} = 0.27 Ra_L^{\frac{1}{4}}$$

for the bottom: $\overline{Nu_L} = 0.54 Ra_L^{\frac{1}{4}}$

From these relations, a convective surface heat transfer coefficient has been arrived at by weighting the coefficients of each type with the amount of surface area in each category. The resulting average convection coefficient on the exposed surfaces was found to be 1.87 Btu / (hr ft^2 F).

Net radiative heat transfer occurs predominately from the box surfaces on the outer edges of the stack. A radiation heat transfer coefficient has been calculated using the method described by Beckman and Duffie (p.158). The formula for this radiation coefficient, h_r is:

$$h_r = \frac{s\left(T_2^2 - T_1^2\right)\left(T_2 - T_1\right)}{\frac{1-e_1}{e_1} + \frac{1}{F_{12}} + \frac{(1-e_2)A_1}{e_2 A_2}}$$

A radiation heat transfer coefficient has been calculated as 0.81 Btu / (hr ft^2 F).

These two coefficients, weighted by the surface area over which they operate, have been added to arrive at an overall effective coefficient. The value arrived at is approx. 2.3 Btu / (hr ft2 F). The EES file used to calculated the coefficients just described is shown in appendix E.

Once the heat transfer coefficient is known, the Biot number can be calculated:

$$Biot\_No. = \frac{hL_c}{k} = 1.81$$

This number is significantly greater than 0.1. This indicates that it may not be assumed that the dominant resistance to heat flow from a block of cheese resides at the surface of the block. Rather, there is appreciable internal resistance within the cheese block. This explains why there is significant difference between the heat transfer coefficient calculated based upon convection and radiation relations and the coefficient arrived at using the general formula for lumped capacitances.

5.3.4 Evaluation of Error Associated with Lumping

For the purpose of this study, the effective heat transfer coefficient arrived at using the general capacitance relation will be used. Since this is not a strictly rigorous approach, an estimate of the error associated with the use of this method is necessary. Finite Element Heat Transfer (FEHT), is a computer program developed at the University of Wisconsin - Madison to numerically model transient heat transfer in two-dimensional conduction problems (Klein, 1992). FEHT is used in this study to compare the cooling behavior of an unlumped block of cheese to the behavior the block would exhibit were it truly lumpable.

Case A: Non-lumped finite element model

The block has been modeled in FEHT using the material properties described in chapter 2. The boundary conditions at the surface included a convection coefficient of 2.3 Btu / (hr ft2 °F) and a fluid temperature of 38°F. The results of this model are shown in figure 5.4. Each line represents the temperature change over time at different points from the box surface to the box center. From the figure it can be observed that during the first 24 hour period there is nearly a twenty degree temperature difference within the block. This difference reduces to less than three degrees by the fifth day.

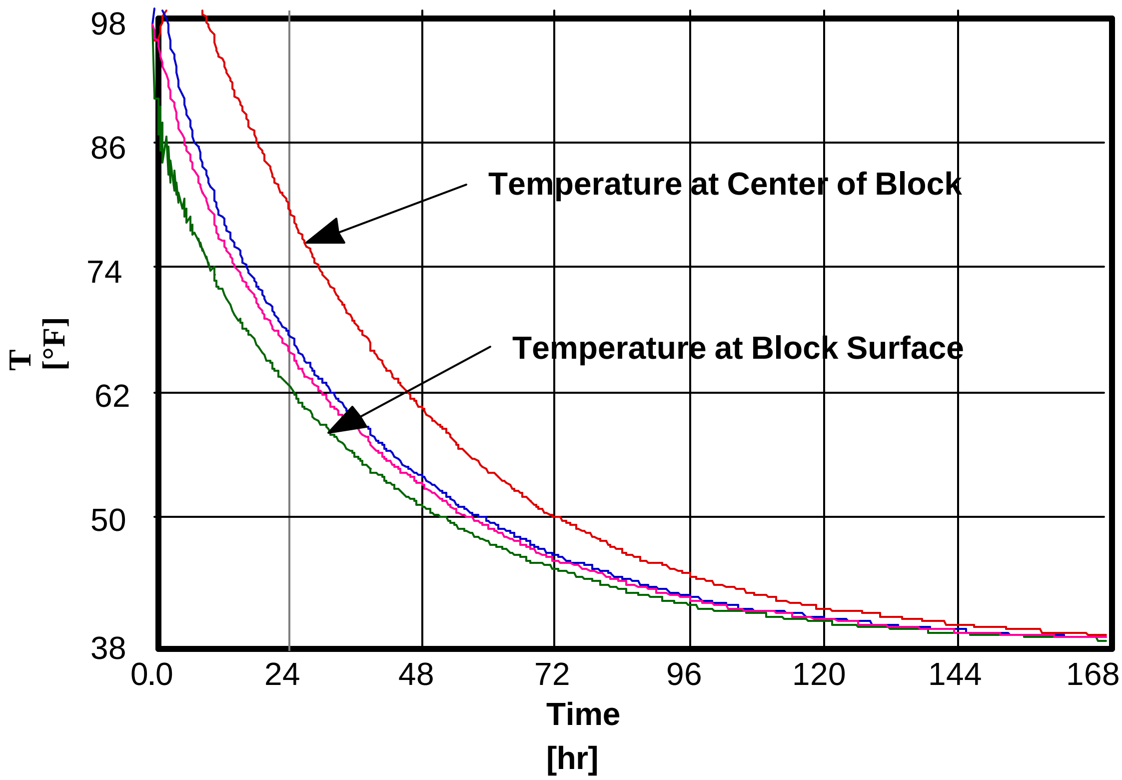

Figure 5.4 Transient Temperature Behavior of Unlumped Cheese Block

Case B: Lumped transient behavior

FEHT has also been used to model the cheese block as a lumped material for two sets of boundary conditions. In the first case, the air temperature surrounding the block is held constant at 38°F. In the second case, a square wave generator is used to model the warehouse temperature under a full storage control strategy in which the warehouse temperature essentially switches from a low setpoint temperature to a higher floating temperature at twelve hour intervals. Figure 5.5 shows the non-lumped temperature trajectory of figure 5.4 plotted together with the trajectory of the lumped cooling block for comparison. Figure 5.6 shows the plots of the lumped block under no storage (for which the warehouse temperature remains constant) and full storage control strategies plotted together.

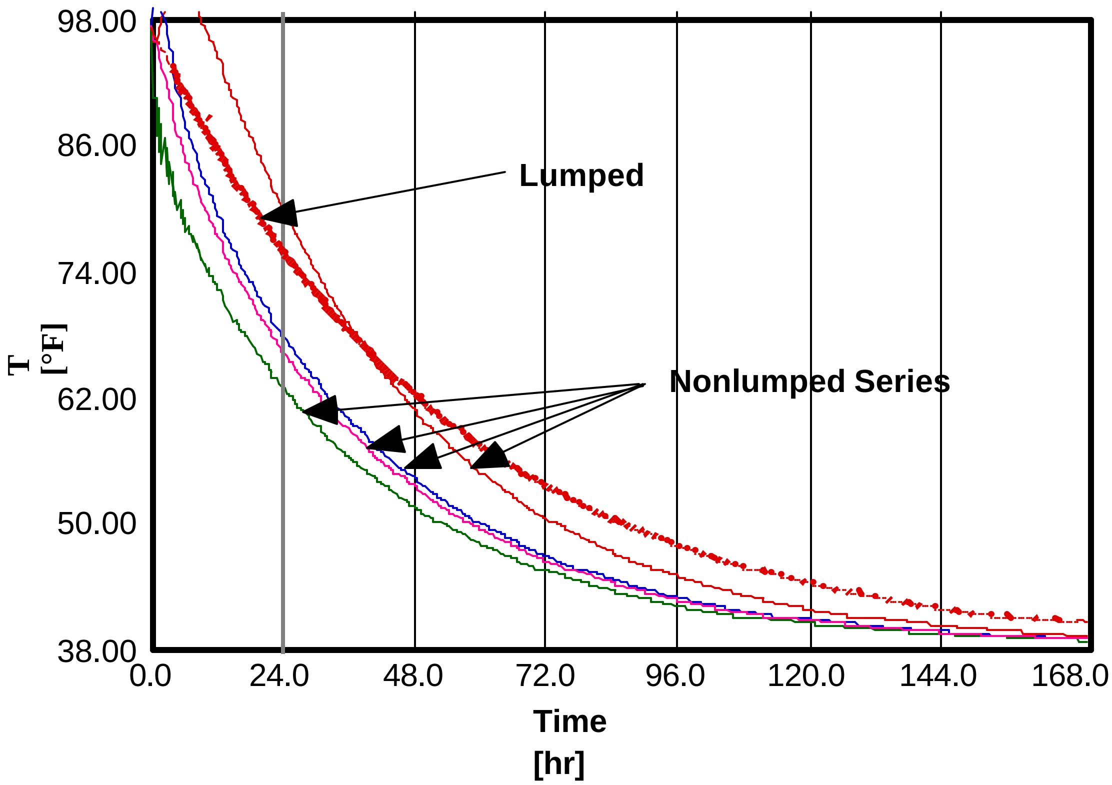

Figure 5.5 Lumped and Unlumped Cheese Block Cooling Trajectories

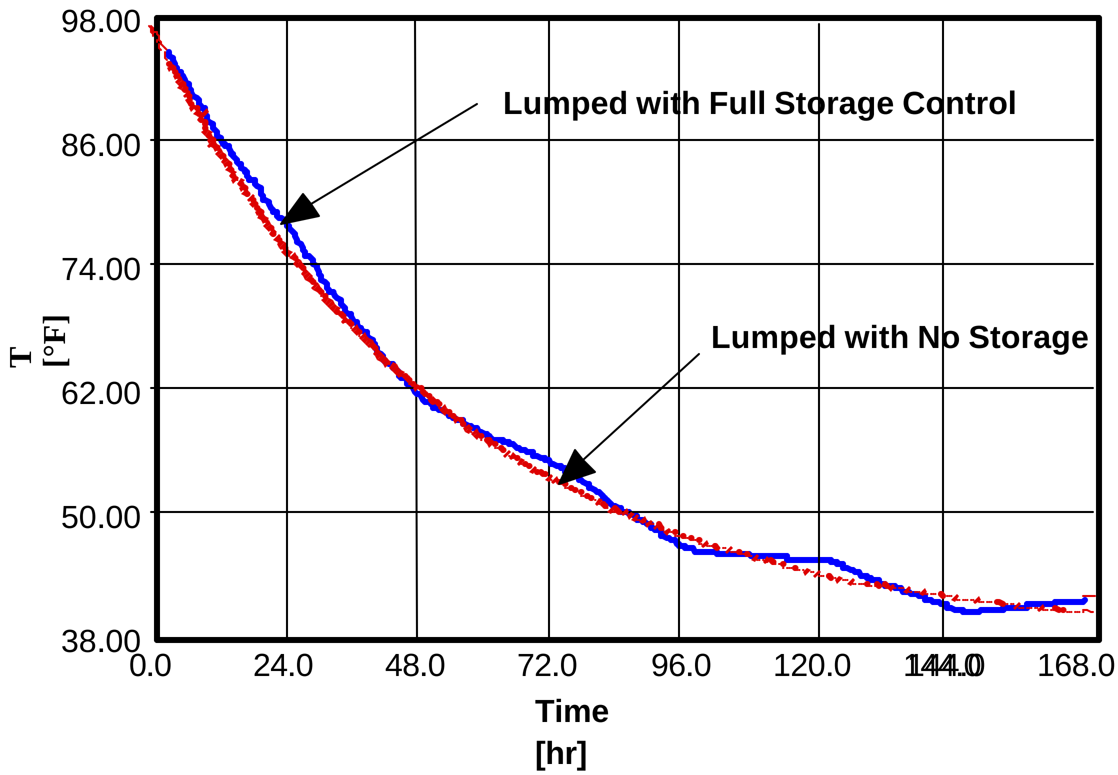

Figure 5.6 Lumped Cooling Trajectories Under Full Storage and No storage Control Strategies

Several observations can be made from the FEHT modeling results. First, upon comparing the non-lumped cooling trajectory with the lumped cooling trajectory depicted in figure 5.5, it is apparent that heat loss from the lumped model occurs at a slower rate than from the unlumped. From this it can be concluded that the lumped model will under predict the temperature swing of the bulk mass, though only slightly. Second, from the lumped trials shown in figure 5.6, it can be concluded that the full storage control strategy has only slight effect on the cooling trajectory of the cheese. The duration of the cooling period is essentially unaffected. Therefore, it can be concluded that, for the purpose of establishing the feasibility of the full storage control scheme, the lumped capacitance model can be used with acceptable results.

5.4 Results of Finite Difference Model of Warehouse

Now that the necessary parameters have been estimated and the use of the lumped capacitance model validated, the results of this finite difference model will be examined. The model was run under ASHRAE design conditions for the hottest month of the year. If a full storage control strategy can be shown to be feasible under these conditions, it can be safely assumed to be feasible under milder conditions as well. The EES file containing this finite difference model is presented in appendix F.

The output of interest from the model is the temperature response of the air in the warehouse and the bulk cheese in storage. This is the cheese that has already completed the seven day cooling period and is awaiting shipment. Current convention stipulates that this cheese remain at or below 40°F.

The temperature swing that occurs during a 24 hour period under a full storage control strategy is directly related to the amount of cooled cheese in storage. As previously mentioned, this amount can vary from 0.5 to 3.5 million pounds. Figure 5.7 depicts the model results for a 24 hour period with a storage mass of 2.8 million pounds and an outdoor temperature of 92°F. For comparison, figure 5.8 presents the results predicted for the same stored mass but when the outdoor air temperature is 40°F. And figure 5.9 presents the model's prediction for an outdoor air temperature of 90°F but with only 1.5 million pounds in storage. Note that the warehouse setpoint has been reduced for this run in order to bring the bulk temperature back to its starting point.

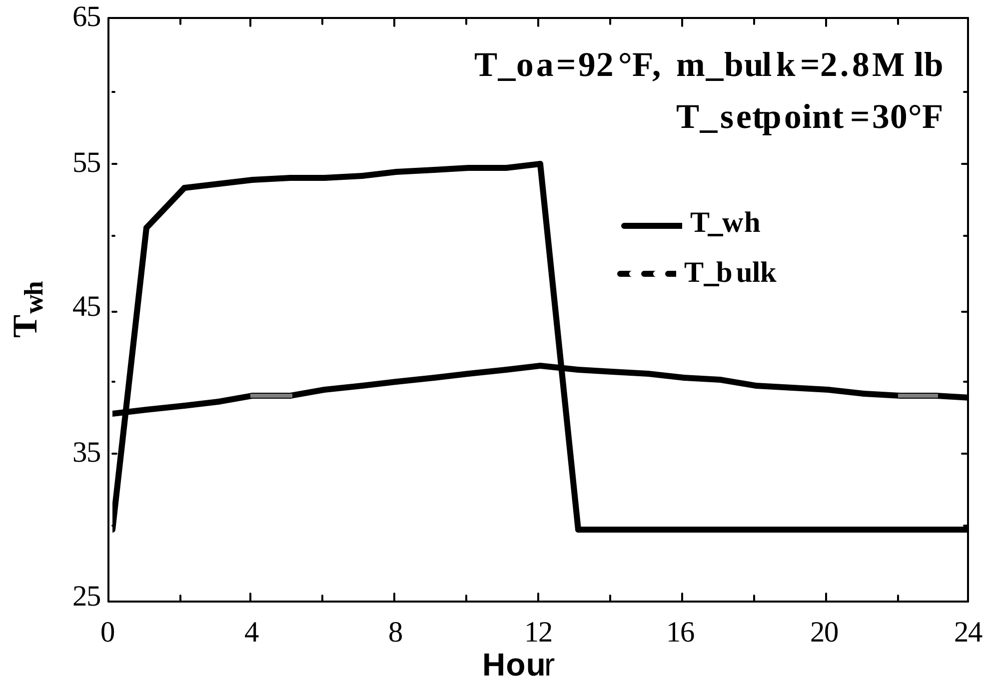

Figure 5.7 Warehouse Model Results for m_{bulk}=2.8 million lb, T_{oa}=92°F

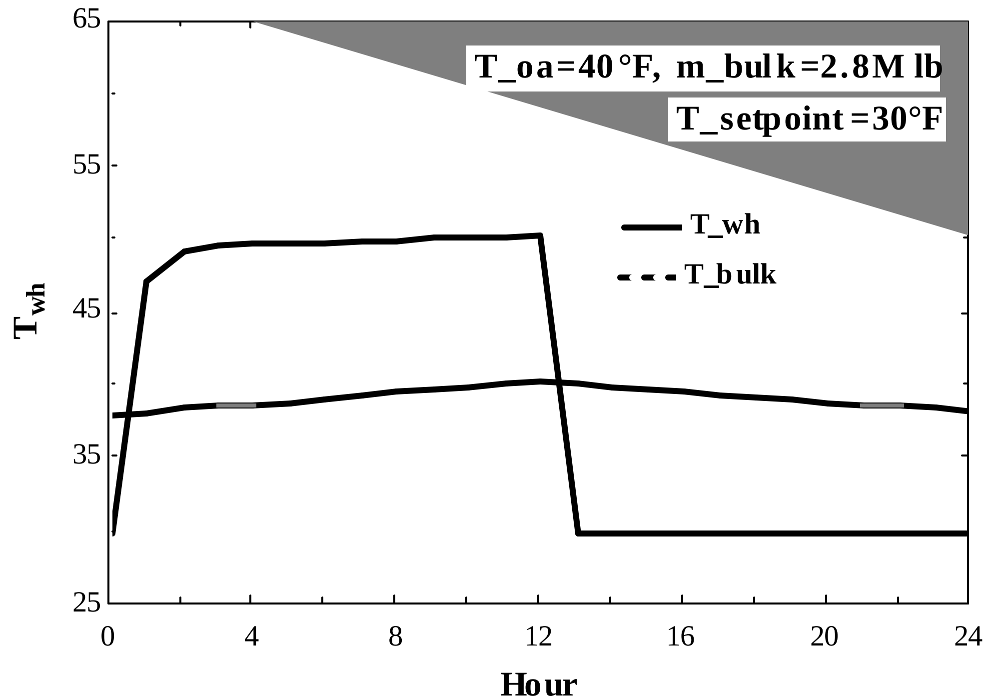

Figure 5.8 Warehouse Model Results for m_{bulk}=2.8 million lb, T_{oa}=40°F

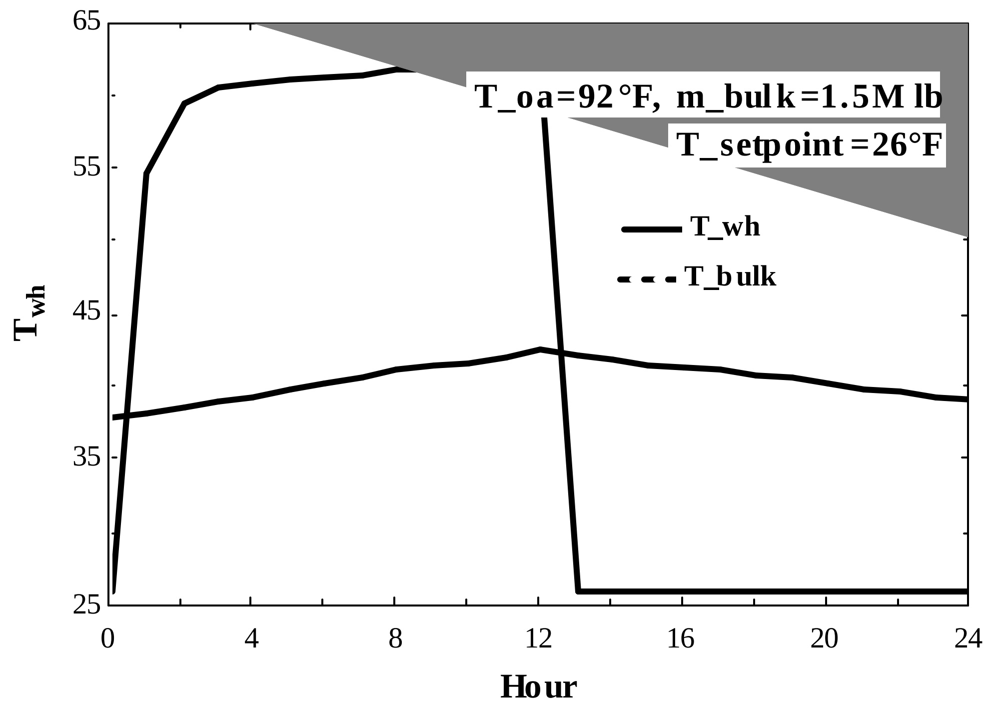

Figure 5.9 Warehouse Model Results for m_{bulk}=1.5 million lb, T_{oa}=92°F

Figure 5.10 presents the relationship between the magnitude of the bulk cheese temperature swing to the amount of cheese in storage. This data has been generated by a series of runs in which the bulk mass was varied.

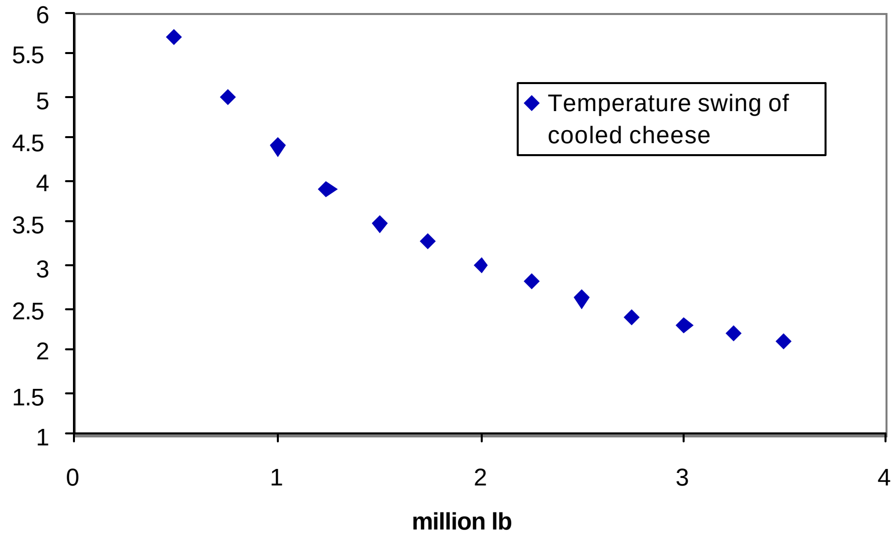

Figure 5.10 Modeling Results of Relationship of Amount of Cooled Cheese in Storage to the Temperature Swing with Full Storage Control (T_oa=92°F)

A temperature swing of large magnitude does not necessarily mean that full storage control is not practical. It simply means that the off-peak setpoint must be reduced. Operating the equipment at a cooler setpoint does, however, slightly reduce the COP of the cooling system. Figure 5.10 shows that a full storage strategy will not create a bulk cheese temperature swing larger than 4°F if there is one million pounds or more in storage. Though a setpoint of 28°F or lower may be required to safely allow shutdown of the cooling system during on-peak hours, while still prohibiting the temperature of the bulk stored cheese from increasing above 40°F.

5.5 Estimated Savings

The cost to meet the cooling load is based upon the monthly average daily energy use estimates described in section 5.2. To arrive at the electric demand necessary to meet this cooling load, it is necessary to know the system performance through the year. Monthly estimated of the system's Coefficient of Performance (COP) have been estimated using a simple thermodynamic model. The COP model assumes an insentropic compressor efficiency of 0.65 and a heat exchanger temperature difference of 12°F. The model is presented in appendix G. Table 5.2 presents the estimated annual energy costs of the cooling system under present operating conditions along with the estimated annual energy costs under a full storage control strategy.

Table 5.2a Costs Estimates for System Operation with and without Storage - January through July

Month	Jan.	Feb.	March	April	May	June
Average Daily Load (Btu/dy)	7.9E+06	8.0E+06	8.6E+06	9.4E+06	1.0E+07	1.1E+07
COP	8	8	8	5.7	4.2	3.2
Cost, No storage ($)	1,397	1,379	1,417	1,531	1,703	1,872
Cost With Storage ($)	190	174	207	308	460	619
Savings ($)	1,207	1,205	1,210	1,223	1,243	1,253

Table 5.2b Costs Estimates for System Operation with and without Storage - August through December

Month	July	Aug.	Sept.	Oct.	Nov.	Dec.

Average Daily Load (Btu/dy)	1.1E+07	1.1E+07	1.0E+07	9.7E+06	8.9E+06	8.1E+06
COP	3	3.1	3.8	4.9	7.8	8
Cost, No storage ($)	1,957	1,925	1,741	1,615	1,423	1,403
Cost With Storage ($)	696	667	502	382	213	195
Savings ($)	1,261	1,258	1,239	1,233	1,210	1,208

5.6 Economizer

In a cold climate, where outdoor temperatures are frequently below the setpoint temperature within a cold storage area, a possibility exists for meeting a cooling load by ventilating with outdoor air, allowing the cooling equipment to remain idle. The Marshfield plant is not currently taking advantage of this opportunity. Table 5.3 provides the yearly average number of hours, based on TMY data, when the ambient temperature would allow for economizer use for a number of setpoint temperature.

Table 5.3 Utilizability of an Economizer v. Setpoint Temperature

Setpoint Temp., °F	Ave # hours at or below	Fraction of year
32	2605	0.30
34	3000	0.34
36	3253	0.37
38	3486	0.40

During the operation of an economizer, the only energy cost is that associated with the operation of fans to force the ventilation.

The costs shown above can all be reduced considerably with the installation and use of an economizer. Assuming a setpoint of 32°F, an economizer can be used in place of the cooling system for 30% of the year. Its benefits are slightly diluted due to the fact that it is useful only during the winter, displacing the cooling system when the system's COP is at its peak. The energy cost savings under these conditions are estimated to be $4,400 per year.

5.7 Conclusions

The possibility of exploiting the thermal storage potential of cheese to shift electric demand has been explored. This has involved several aspects of analysis. First, the cooling demand of the cold storage facility has been evaluated. Second, the thermal behavior of stored cheese has been modeled. Finally, the results of this modeling of stored cheese have been extended for use in a finite difference model of the entire warehouse. This model has allowed for estimates to be made on the potential for load shifting.

The modeling effort has demonstrated the feasibility of sub-cooling the stored cheese in the warehouse during off-peak hours when energy rates are low in order to avoid operating the cooling system during the peak rate period. In this way energy costs can be reduced by operating with lower rates and, more importantly, by reducing the monthly demand charge.

Installation of an economizer has been suggested. At least 30% of the year, ambient temperatures are low enough to meet warehouse cooling demands simply by ventilating with outdoor air. Energy cost savings associated with an economizer have been estimated.

Table 5.4 presents a summary of the potential savings identified in this section.

Table 5.4 Summary of Predicted Savings

	Annual Energy Cost ($)	**Savings ($)**
No Storage	19.362	n / a
Full Storage	4,611	14,750
Economizer	n / a	4,339
Full Storage + Economizer	3,644	15,718

Both of these energy savings opportunities appear highly attractive. While each generates considerable savings, the investment required to undertake them is minimal. For the full storage control strategy, no additional cooling system is necessary since the cooling equipment already in place is of far greater capacity than is presently needed. The cost of an economizer is presumed to be small as well.

CHAPTER SIX

SUMMARY AND RECOMMENDATIONS

6.1 Summary of Study

The focus of this study has been to examine the use of electricity and natural gas at two representative cheese plants in Wisconsin. The main objective has been to identify and document opportunities to reduce utility costs for these plants. Possibilities for waste heat recovery and demand savings have been analyzed. These possibilities have fallen into three categories, including: 1) thermal integration of a multi-effect evaporation system, 2) thermal integration of a spray drying system, and 3) alternative control strategies to reduce the energy costs of cooling a cheese storage warehouse. Conclusions regarding the feasibility of the possibilities explored have been reached based upon the analysis.

6.1.1 Thermal Integration of Marshfield's Evaporation System

The evaporation system studied currently makes use of a sizable cooling tower to reject roughly five million Btu / hr of heat into the environment during operation. Pinch analysis has demonstrated that potential exists for this heat to be delivered to the 40°F raw milk prior to pasteurization for cheese making and to preheat a 12,000 lb / hr flow of city water for cleaning purposes. This would provide two simultaneous economic benefits. First, the cooling tower, which requires pumping and operation of a 60 hp fan, could be abandoned. Second, the heating requirement of the pasteurizer could be reduced significantly.

The annual savings of these opportunities fall into two parts, 1) avoided heating costs, and 2) avoided cooling tower operating costs. There is a reduction of 4.98 x 10^6 Btu / hr of heating, and the plant operates 18 hours per day, 290 days per year. Over the course of a year, this amounts to 2.60 x 10^{10} Btu. Since 1 therm = 1 x 10^5 Btu, this represents 2.60 x 10^5 therms. At $0.28 per therm, this amounts to an annual savings of $72,800.

Cooling tower operation is presently estimated to use 1.17 x 10^5 kWh per year. At $ 0.027 per kWh, this comes to $ 3,150 per year. Since this can be added to the fuel savings, the total utility savings for this opportunity comes to $76,000.

6.1.2 Spray Dryer Integration

Whey can be processed into a variety of marketable products. These include Concentrated Whey Solids, Dry Whey, Reduced Lactose Whey, Reduced Minerals Whey, Whey Protein Concentrate, Whey Solids in Wet Blends, and Whey Solids Utilized for Lactose. Among these different alternatives, dry whey is the by far most common as well as the most energy intensive to produce. Producing dry whey from concentrated whey solids is done exclusively with spray dryers.

Spray Dryers represent the largest consumer of energy in cheese plants that produce whey powder. These units draw a large volume flow of outside air to be heated by a direct fire burner. An exhaust flow leaves the unit of equal volume to the supply. This exhaust, while much cooler than the heated air supplied to the drying chamber is still at a temperature much higher than outdoor air, particularly in the winter.

The potential for utilizing a heat exchange system between the supply and exhaust air streams has been evaluated at the Blair whey processing facility. The results of this analysis indicate that implementing a heat recovery heat exchange system would likely reduce fuel use by 150,000 to 250,000 therms per year, depending on the size of the heat exchangers. At $0.28 per therm, this represents an annual savings of $42,000 to $69,000.

6.1.3 Cold Storage Warehouse Demand Savings

Two possibilities have been explored with respect to lowering energy costs related to the cold storage facility. The first opportunity examined relates to peak load reduction savings using stored cheese as a thermal storage medium. Analysis has shown this to be a feasible way to reduce significantly the operation of cooling equipment during the peak rate period, except for instances when the amount of cheese in inventory is especially low. Since the plant examined is not presently using an economizer, installation of such a devise represents another significant cost saving opportunity. An economizer takes advantage of the cooling potential of ventilating with outdoor air when the air temperature is below the storage room setpoint.

Analysis has shown that the implementation of a full storage control strategy could save the Marshfield plant nearly $15,000 per year. Likewise, implementing an economizer (without full storage) could save the plant over $4000 per year. If both opportunities are pursued, a total savings of $15,700 could be expected.

6.2 Conclusions

This study has demonstrated two important things. First, it has exhibited the strength of pinch methodology for helping to understand a complex process and for focusing attention on the right parts of such a process to improve overall efficiency. Second, it has demonstrated that significant opportunity exists for improving the energy efficiency within the Wisconsin food industry. This industry is not a new one in the state. Thus most of the equipment in use could benefit from similar efforts. The approach used for this study could easily be extended to other cheese plants and to other sectors within the Wisconsin food industry.

6.3 Recommendations

As expected, the opportunities promising the largest savings are also likely to require the most significant investment of capital. This study has focused on the technical feasibility of energy saving opportunities and has estimated the energy savings expected from each. The next step for each opportunity is to determine the installation costs to determine the economic feasibility.

Without cost information it is not possible to rank the opportunities identified in terms of pay back period as would be preferred. Rather, they will simply be ranked according to potential for utility costs savings. This ranking is presented in table 6.1

Table 6.1 Ranked Summary of Results

Rank	Description	Expected Annual Savings
1	Evaporator Integration Strategy	$76,000
2	Spray Dryer Integration Strategy	$42,000 - $69,000
3	Thermal Storage Warehouse Control	$14,750
4	Warehouse Economizer Installation	$4,340

6.4 Future Research

Several items of interest for further research in this area have been identified in the course of this study. These are briefly explained in the following section.

1) Further Development of Spray Dryer Models

With good models of the physics of industrial spray dryers, more detailed analysis of integration possibilities and alternative control strategies may be explored.

2) Product Quality Constraints with Respect to Thermal Storage

Before broad acceptance of thermal storage strategies become widely accepted for cheese warehouses, the potential effects, if any, of daily temperature swings on product quality must be better understood. This study has shown that the mass averaged temperature swings are not large. But the temperature swing of the cheese near the surface of the block may be significant.

3) Whey Processing for Small Plants

Although the trend in Wisconsin is clearly toward fewer plants of larger production capacity, there remain a number of small plants. It is uncommon for small plants to invest in capital intensive whey processing equipment. Thus, most of the whey produced at these plants is not utilized. Possibilities may be available, though, for numerous small plants to support a central whey processing facility. Or perhaps investment in a less expensive whey filtration system could concentrate the whey to the extent that it is feasible to transport.

APPENDIX A, Pinch Analysis Methodology

Pinch analysis is a method for arriving at maximum levels of heat integration for an industrial process. Details of this method follow.

The first step in pinch analysis is to extract the design data necessary to characterize each of the flow streams in a process. For each stream, one must identify the starting temperature, the final temperature, the specific heat, and the mass flow rate. Each stream can further be identified as either a hot stream (from which heat is removed) or a cold stream(to which heat is added.)

Once the stream data is known, each stream can be plotted on a temperature vs. total enthalpy (flow rate x enthalpy) graph. This results in a set of curves, each curve representing one flow stream. The slope of these curves is the heat capacity (mass flow rate x Cp) of the particular stream. All of the cold streams are then combined onto one curve. Where two or more streams traverse the same temperature interval, the slope of the composite curve is the sum of these streams. The resulting curve is know as the Cold Composite Curve. In a parallel manner a Hot Composite Curve is produced.

The two composite curves are then plotted on the same graph. It is desirable that these curves overlap one another in terms of enthaply to the maximum extent possible since overlapping means that heat exchange (process integration) is possible. However, there is a limit to how close the curves may be brought together defined by the minimum temperature difference for which heat exchange is feasible (delta T min). Once these curves have been moved as close as possible, there is a point where the temperature difference between them is equal to delta T min. This is known as the Pinch.

The amount by which the cold composite maximum enthalpy now exceeds the maximum hot composite enthalpy represents the heating that can only be accomplished by providing hot utility. This is referred to as the Minimum Hot Utility. Similarly, the amount by which the minimum cold composite enthalpy exceeds the minimum hot composite enthalpy represents the minimum amount of cooling utility necessary for the process. This is the Minimum Cold Utility. These minima are then design goals for a heat exchanger network (HEN) for the process. These minimum targets are presented graphically on a pinch composite diagram in figure B.1

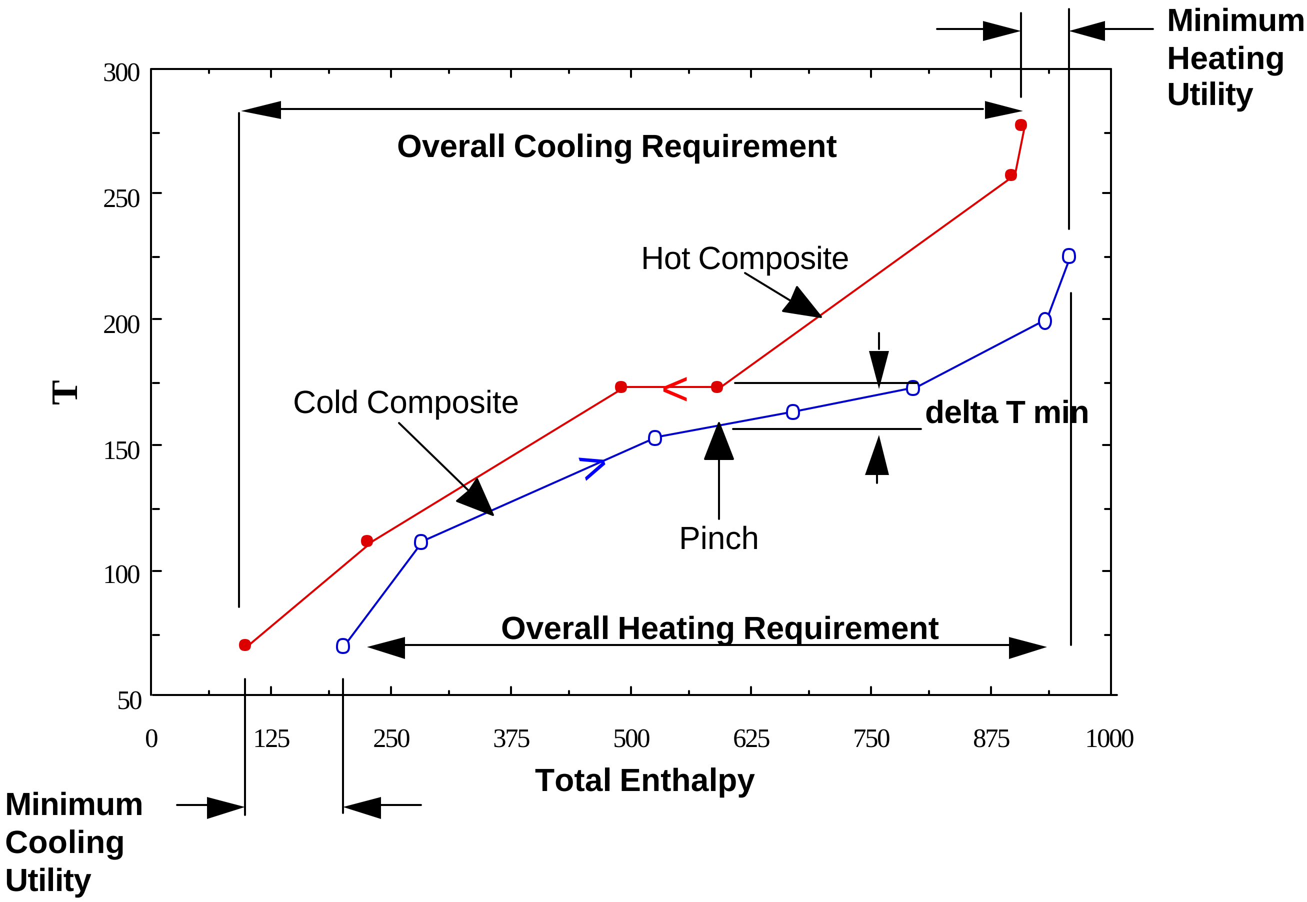

Figure B.1 Pinch Composites with Interpretation

To understand the significance of the pinch, one must produce a final graph, this time plotting the enthalpy difference between the hot and cold composites vs. temperature. This is known as the Grand Composite. The hot and cold composites are modified slightly for this plot. The hot composite above the pinch is shifted downward by delta T min. And the cold composite below the pinch is shifted upward by delta T min. The grand composite will therefore always show the enthalpy difference between the hot and cold composites as zero at the pinch.

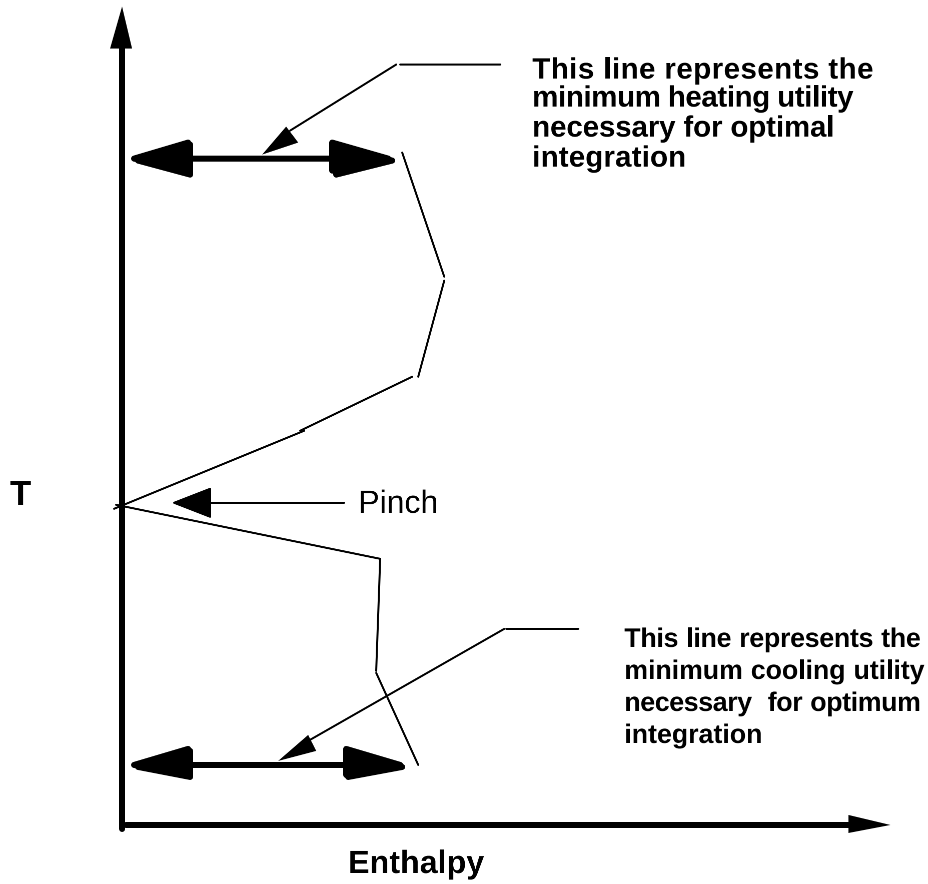

Figure B.2 Grand Composite Diagram

From what has now been said about the hot target, the cold target, and the enthalpy difference at the pinch, we arrive at conclusion that the pinch represents a boundary within the process between a net heat sink above the pinch and a net heat source below it. If one were to place a heat exchanger across this boundary, the result would be to increase the hot utility requirement and simultaneously to increase the cold utility requirement as well. For the same reason, one would never want to add hot utility below the pinch or cold utility above the pinch.

BIBLIOGRAPHY

American Dairy Products Institute (ADPI), Whey Products: A Survey of Utilization and Production Trends 1989.

American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE), 1993 ASHRAE Handbook - Fundamentals, 1993.

Barbano, D., Hurst, S., and Aplin, R., 1991. “Factors Influencing Costs and Profitability of Cheese and Whey Product Manufacture.” Proceedings, Cheese Cheese Research & Technology Conference, Center for Dairy Research, University of Wisconsin - Madison, March 6-7, 1991.

Billet, R, Evaporation Technology, Principles, Applications, Economics, VCH, Weinheim, 1989.

Cerda, J., Westerberg, A.W., Mason, D., and Linnhoff, B., “Minimum Utility Usage in Heat Exchanger Synthesis,” Chemical Engineering Science, Vol. 38, No. 3, pp.373-387, 1983.

Duffie, J.A. and Beckman, W.A., Solar Engineering of Thermal Processes, Wiley Interscience, New York, 1991.

Eck, A., Cheesemaking: Science and Technology, Lavosier Publishing Inc., New York, 1986.

Gillies, M.T., Whey Processing and Utilization: Economic and Technical Aspects, Noyes Data Corporation, Park Ridge, New Jersey, 1974.

Hansen, R, editor, Evaporation, Membrane Filtration and Spray Drying in Milk Powder and Cheese Production, North European Dairy Journal, 1985.

Hurst. S, R. Aplin, and D. Barbano. 1990. “Whey Powder and Whey Protein Concentrate Production Technology, Cost, and Profitability.” Cornell Program of Dairy Markets and Policy. Cornell University, Ithaca, NY.

Kays, W.M., and London, A.L., Compact Heat Exchangers, The National Press, Palo Alto, 1955.

Keey, R.B., Introduction to Industrial Drying Operations, Pergamon Press, New York, 1978.

Klein, S.A., et al., TRNSYS 14 Manual, July, 1994.

Klein, S. A. and Alvarado, F. L., “EES - Engineering Equation Solver,” F-Chart Software, 1996.

Klein, S. A., W.A. Beckman, and G.E. Myers, “FEHT A Finite Element Analysis Program,” F-Chart Software, 1992.

KRACK Corporation, Refrigeration Load Estimating Manual, 1992.

Linhoff, B., “Use Pinch Analysis to Knock Down Capital Costs and Emissions,” Chemical Engineering Progress, Vol. 90, No. 8, pp. 32-57, 1994.

Masters, K., Spray Drying Handbook, Longman Scientific & Technical, New York, 1991.

Mesa-Dishington, J. K., D. M. Barbano, and R.D. Aplin. 1987. “Cheddar Cheese Manufacturing Costs Economies of Size and Effects of Different Current Technologies.” Department of Agricultural Economics. Cornell University, NY.

Mitchell, J.W., Course Notes, 1996.

Papoulias, S.A. and Grossman, I.E., “A Structural Optimization Approach in Process Synthesis -- II,” Computers and Chemical Engineering, Vol. 7, No. 6, pp. 707-721, 1983.

Pendyala, V.R., and Patwardhan, V.S., “Design of a Vapour Compression Heat Pump in a Heat Exchange Network,” Heat Pumps -- Proceedings of the 4th International Conference, STI / Springer - Verlag, 1991.

Reay, D.A. and MacMichael, D.B.A., Heat Pumps, Second Edition, Pergamon Press, New York, 1988.

Scott, R., Cheesemaking Practice -- 2nd Edition, Elsevier Applied Science Publishers, New York, 1986.

Stoecker, W.F., Refrigeration and Air Conditioning, McGraw-Hill Book Co., New York, 1958.

Swaney, R, "Thermal Integration of Processes with Heat Engines and Heat Pumps," AIChE Journal, Vol. 35, No. 6, pp 1003-1016, 1989.

WDATCP, 1995. "Wisconsin 1995 Dairy Facts." Wisconsin Department of Agriculture, Trade and Consumer Protection. Madison, Wisconsin, USA.

Zadow, J.G., Editor, Whey and Lactose Processing, Elsevier Science Publishers Ltd, Essex, England, 1992.

Zbicinski, I., Grabowski, S., Strumillo, C. Kiraly, L, and Krzanoski, W., "Mathematical Modelling of Spray Drying," Computers and Chemical Engineering, Vol. 12, No. 2/3, pp. 209-214, 1988.