INDUSTRIAL HEAT PUMP
APPLICATION AND EVALUATION

Phase II A
Final Report

July 1996

Covering:
Startex Mills
Spartan Textile Facility
Spartanburg, South Carolina 29304

Work Performed Under Contract DE-FC07-89ID12862
With
Duke Power Company
Charlotte, North Carolina

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**FIGURES:** (FOR SECTIONS 1.0-6.0)

**APPENDIX A:** PINCH TECHNOLOGY

**APPENDIX B:** FUEL CONSUMPTION BASIS

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This report summarizes the result of the DOE Phase IIA project (DE-FC07-89ID12862) for the Spartan Mill's textile facility at Startex, South Carolina. A rigorous Pinch analysis was conducted on the finishing mill facility, and the plant wide steam system. Both direct heat exchange via heat exchangers, and a heat pump application were identified for a total energy savings of 11.6 MMBtu/hr. This translates to a saving of $245,000 per year. The proposed heat exchange option recovers heat from the plant waste water, tenter frame exhaust and steam can exhausts to preheat the process water. The proposed heat pump system also utilizes the plant’s waste water system as heat source and provides heat to supplement the existing plant’s hot water system. Based on this estimate, both direct heat exchange and heat pump options appear to be technically feasible and economically attractive.

In addition to energy savings, the identified projects also have an environmental impact of reducing the waste water temperature, removing air-borne lint in the exhaust, and reduction in NO\textsubscript{X} and CO\textsubscript{2} emissions from the fuel burning. Annual emission reduction is estimated to be 22 tons and 5,500 tons respectively for NO\textsubscript{X} and CO\textsubscript{2}. No credit has been given to any of these environmental benefits.

The plant data was further confirmed by taking new measurements. The results of the study was then modified to reflect this revised data in Section 6.0 of this report.
Section 2

PROCESS DESCRIPTION

2.1 Process Overview

This project focused on the wet finishing section of the Startex Mill. Figure 1 shows a typical simplified block flow diagram of the finishing mill. Based on the fabric and the product specifications, the mill might use different combination of processing steps. Greige goods from other part of the plant go through a series of treatment in the finishing mill, including singeing, desizing or scouring, bleaching, mercerizing, and drying. Singeing is a process of burning off protruding fiber from yarn or fabric by passing it over a direct gas flame. Desizing removes sizes applied to warp prior to weaving. Scouring is an operation to remove the sizing and tint used on the warp yarn in weaving and in general to clean the fabric prior to dyeing. The bleaching operation consists of several washing steps to remove the natural and artificial impurities to obtain clear white.

Mercerizing is to treat cotton yarn or fabric to increase its luster and affinity for dyes. The final step of the finishing process is drying Figure 2 cans and gas tenter frames dryers. In the steam drying cans, the surface of cans are heated by condensing steam inside the cans to provide heat through conduction to the fabric. In the tenter frame the heat for drying is provided by the combustion of natural gas, mainly from the combustion air. The combustion exhaust combined with building air, which is pulled by draft in the dryer, provides heat to the fabric and also carries the moisture out of the system.

2.2 Process Water System Overview

About 200 GPM of hot process water is required in various washers in the finishing mill process. A process water flow diagram is shown in Figure 3. The fresh city water, coming in at an annual average temperature of 60°F (Summer 75°F, winter 45°F), is first preheated by the waste water in two Tranter Heat Exchangers and also in the caustic evaporator condenser. The preheated fresh water is collected in a
storage tank and from there it is sent to the steam heater to raise its temperature to 200°F. The hot water is used in various washers in the process, which may also utilize direct steam injection to maintain the desired wash water temperature. The overflow of the wash water from each washer is eventually collected as waste water at about 190°F.
Section 3

STEAM SYSTEM

3.1 Steam System Overview

The steam system supplies steam to both finishing mill and greige mill. An average plant steam production of 42,000 lb/hr is recorded by steam meter in the boiler house, when both the mills are operating. The finishing mill consumption is measured to be an average of 29,000 lb/hr steam, the remaining 13,000 lb/hr goes to the greige mill. Each steam user in the finishing mill was identified, and its corresponding usage is analyzed in the steam modeling section.

3.2 Steam System Modeling

The APLUS (Analysis of PLant Utility System) software developed by TENSA Services was used to simulate the steam system as shown in Figure 4. There is no direct measurement for each steam user, therefore, each individual steam usage is calculated based on measured parameters and assumptions. This usage has been reviewed by Spartan's personnel and revised according to their comments. A summary of the usage follows:

- Overall steam generation
  42,000 lb/hr (measured average)

- Finishing mill steam usage
  29,000 lb/hr (measured with existing steam meter)

- Greige mill steam usage
  13,000 lb/hr (42,000 - 29,000 = 13,000)
  – 2,560 lb/hr through steam turbine (vendor data)
  – 968 lb/hr let-down to provide total of 3,528 lb/hr to the deaerator (APLUS simulation).
  – 945 lb/hr of GR1 represents 10% of the Greige process steam requirement where the condensate does not return (assumed).
- 8,515 lb/hr of GR2 represents 90% of the Greige process steam where condensate return to the boiler house.

- Detail of each finishing mill users.
  - Caustic Evaporator 2,180 lb/hr (Vendor design data).
  - Steamer 1,430 lb/hr.
    - Based on a computer model developed by the North Carolina State University for wet processing. Important parameters for the steamer used in the model are provided by Startex Mill.
      - Fabric velocity: 130 yd/min.
      - Fabric weight: 6.43 oz/square yd.
      - Inlet fabric WPU: 1 (lb water/lb fabric)
      - Outlet fabric WPU: 1 (lb water/lb fabric)
      - Exhaust temp. 185°F
  - J Boxes (2) 2,769 lb/hr
    - Based on computer modeling of J-Box. Same fabric velocity, fabric weight, and inlet/outlet WPU as in steamer.
      - Exhaust temperature: 169°F
  - Dry cans on two lines: 5,096 lb/hr
    - Based on computer model with fabric velocity of 130 yd/min, and 6.43 oz/yard of fabric weight.
      - 45% moisture removal from both pre-dry cans and dry cans.
      - Exhaust temperature 200°F
      - Exhaust humidity (lb water/lb dry air): 0.06 (assumed)
        - The pounds of steam used per pound of water removal is 2.2. This is in the range of vendor reported value of 1.5 to 2.5.
  - Washers, 879 lb/hr,
    - Based on computer model.
  - Steam Heater, 11,234 lb/hr. This is calculated based on 200 GPM of hot water need at 200°F as follows (assuming fresh water temperature is 60°F on a yearly average):
    \[
    \text{Heat requirement:} \frac{(200 - 60 \degree \text{F}) \times 500 \text{ lb/hr} \times 200 \text{ GPM} \times 1 \text{ Btu/lb} \degree \text{F}}{\text{GPM}} = 14.0 \times 10^6 \text{ Btu/hr.}
    \]
Heat pick-up from Heat Exchanger:
Tranter: \((110 - 60) \times 500 \times 50 \times 2 = 2.5 \times 10^6 \text{ Btu/hr.}\)
Evaporator Distillate Exchanger 75\% of the time:
\((110 - 60 ^\circ F) \times 500 \times 50 \times 0.75 = 0.938 \times 10^6 \text{ Btu/hr.}\)

**Additional heat by Steam Heater:** \(14-2.5-0.938 = 10.56 \times 10^6 \text{ Btu/hr.}\)
20 psi steam requirement:
\[ \frac{10.56 \times 10^6}{940} = 11,234 \text{ lb/hr.} \]

- Boiler make-up water usage 13,719 lb/hr (13928 lb/hr, measured).
The close correlation between the measured and calculated values of make up water usage indicates that the APLUS steam system model is in good agreement with the plant operation.

There is 6500 lb/hr of steam usage unaccounted for in the plant, 15\% of the total steam production. A possible place for steam loss is through the steam traps and piping leakage. Proper maintenance and repairing should be able to reduce this by 50\%.
Section 4

PINCH ANALYSIS

Pinch technology is a systematic way of analyzing both the heating and cooling characteristics of a process. It is based on Thermodynamics principles and has been successfully applied to numerous industrial processes. However, it is beyond the scope of this report to describe the theory. Therefore, an introduction to pinch technology is attached in Appendix A for reference. An important result of the pinch analysis is the ability to set performance targets prior to the actual design. It is possible from the process data alone, to confidently predict the minimum thermal energy required for any process.

Starting from the individual streams' temperature and enthalpy information, the first step in pinch analysis is to construct "composite curves" of all the hot and cold streams in the process, by simple addition of the heat contents over the temperature intervals in the process. Figure 5 shows these curves for finishing facility.

The overlap between the two composite curves as shown by hatch mark, represents the maximum amount of heat recovery possible within the process. The "over-shoot" of the cold composite represents the minimum amount of external heating (hot utility) requirement. The "over-shoot" of the hot composite represents the minimum amount of external cooling (cold utility) requirement.

The closest point between the two composite curves is known as the "pinch" point. The temperature approach between the composite curves at the pinch is defined as "$\Delta T_{\text{min}}$". As $\Delta T_{\text{min}}$ increases, utility requirements increase, however, the heat exchange surface needed to recover the energy decreases as a result of the larger approach temperature. For any $\Delta T_{\text{min}}$, the composite curves define the minimum utility requirements for the process and thus establish the energy targets for the process.

For this process, at a minimum approach temperature of 20 °F, the hot utility requirement can be reduced from 28.8 MMBtu/hr to about 17.6 MMBtu/hr, resulting...
in a saving of 11.2 MMBtu/hr through heat exchange. The plant is currently recovering 3.4 MMBtu/hr heat from the waste water. Therefore, additional 7.8 MMBtu/hr of heat recovery through direct heat exchange is the target for the plant.

The heat pump potential for the plant can be evaluated utilizing the "Grand Composite Curve" or GCC for the process. The GCC gives a "profile" of the energy requirements of a process. The corresponding GCC for the finishing process is shown in Figure 6. The pinch temperature is shown on the GCC as the curve touching the temperature axis. Above the pinch temperature, the process is a net heat sink, where only hot utility is required. Below the pinch, the process is a net heat source, where surplus heat has to be removed.

To be effective, a heat pump must be "appropriately" placed. In the context of pinch terminology, this means that the heat pump must move thermal energy from below the "pinch" temperature, to above the pinch.

4.1 Heat Recovery Scheme

Once the target hot utility consumption is known, pinch technology also provides design principles to achieve the target. Figure 7 shows the existing heat exchangers in the plant in a grid form. All the hot streams which require cooling are shown by a horizontal line with an arrow pointing to the right.

The numbers associated with the line on both ends indicate the temperature of the stream. Similarly, horizontal lines with an arrow pointing to the left indicate the cold streams that have to be heated.

In the middle of the diagram is a vertical line representing the pinch line that separates the system into above the pinch temperature (left half) and below the pinch temperature (right half) regions. At $\Delta T_{\text{min}} = 20^\circ\text{F}$, the pinch temperature is 95°F, which corresponds to 105°F for all the hot streams and 85°F for all the cold streams. Two circles joined by a vertical line connecting the hot and cold streams represent a heat exchanger. Also shown in the figure on the left is the heat content (MMBtu/hr) for each stream.

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The plant is currently recovering heat from waste water (2.5 MMBtu/hr), and caustic evaporator condenser (0.938 MMBtu/hr) to preheat the city water before it gets further heated to the desired temperature in the steam-heater. According to pinch design principle, both these heat exchangers can be better utilized for heat recovery. There are other energy savings' opportunities in the plant that can be improved upon like heating below the pinch and cooling above the pinch as indicated by the shaded circles.

Maximum heat recovery (MER) can be achieved according to the pinch design principles as shown in Figure 8. This MER can save a total of 11.2 MMBtu/hr of heat, which allows 7.8 MMBtu/hr more than the existing heat recovery.

In a retrofit situation, consideration has to be given in the area of operation flexibility and process constraints. This will somewhat offset the final design from the target of maximum heat recovery. The MER design requires heat exchange between the evaporator condenser and boiler feed water make-up. In view of the distance (over 400 ft apart) involved, the amount of heat recovered, and the schedule of evaporator condenser operation, it is decided to leave the existing heat exchanger between the evaporator condenser and the fresh water feed as is. A revised design evolved from the MER is shown in Figure 9. This design has a hot utility savings of 10 MMBtu/hr, which is 6.6 MMBtu/hr more than the current operation. Also shown in Figure 9 is the heat pump system that can provide additional hot utility (steam) saving. The following section describes in more detail the heat exchange and heat pump design.

4.2 Heat Exchange Option

The heat exchange option utilizes waste heat from drying cans exhaust, tenter frame exhaust, and plant waste water to preheat plant hot process water from 60 °F to 160°F.

The drying cans exhaust and tenter frame exhaust are first used to cover the low temperature end of the heating requirement, followed by waste water heat exchanger. Normally, if the exhaust temperature is high, a direct heat exchange option should be
used to recover the heat content in the exhaust stream. However, the airborne particles (particularly lint and oil) in the tenter frame and drying cans exhausts can build up on heat transfer surface, reducing effectiveness and in some instance it has even caught fire. In order to recover the heat content and avoid the pitfalls, an indirect scheme is used through an air washer system.

The exhaust heat from the tenter frame and steam can dryer is first cooled by spraying water over it in an air washer, to get rid of lint and oil in the exhaust. The use of air washer is a common practice in the textile industry to control the air quality and humidity in an air conditioning system. The heat released from the exhaust will heat up the water. The hot water is then used as a heat source to preheat the process hot water from 60 °F to about 85 °F.

The waste water from 190°F to 119°F will be used to preheat the process water from 85°F to 160°F. The existing caustic condenser will remain as is to preheat part of the process water to 110°F. The arrangement of the heat exchangers is shown in Figure 10. A special heat exchanger with automatic flushing control to prevent any deposition on the heat exchanger surface is recommended. This type heat exchanger has been proven in the textile industry to handle waste water from wet processing.

4.3 Closed Cycle Heat Pump

The heat source for the heat pump system is also from the plant waste water. The plant has about 200 GPM of waste water at 190°F. The higher temperature end of the waste water is used in the direct heat exchange option. The waste water comes out of the heat exchange option at 120°F. There is about 4 MMBtu/hr of heat available for heat pumping if the temperature is allowed to cooled further to 85°F. The available heat is used to evaporate the refrigerant in the evaporator of a heat pump system.

This vaporized refrigerant is then compressed in the compressor to a higher pressure to raise its condensing temperature. The condensing heat, thus available at higher temperature, could be utilized to provide heat to the heat sinks (process hot water and boiler feed water) in the plant.
Figure 11 shows the arrangement of the heat pump system in combination with the direct heat integration option to give the optimum heat recovery strategy. The heat pump transfers heat to the sinks where direct heat integration is not possible. The heat pump absorbs 3.7 MMBtu/hr of waste heat from the waste water. With about 400 kW of power input, the heat pump can save 5 MMBtu/hr of steam usage in the deaerator and in the steam heater.
Section 5

SUMMARY AND RECOMMENDATIONS

The analysis has shown very attractive heat recovery potential through both direct heat exchange and heat pump.

5.1 Basis of Economics

Table 1 lists the cost information used in this analysis.

Table 1

<table>
<thead>
<tr>
<th>Economic Data</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Incremental Natural Gas Cost ($/Therm)</td>
<td>0.325</td>
</tr>
<tr>
<td>Average Boiler Combustion Efficiency</td>
<td>81.5%</td>
</tr>
<tr>
<td>*Overall Boiler System Efficiency</td>
<td>70%</td>
</tr>
<tr>
<td>Steam ($/MMBtu)</td>
<td>4.64</td>
</tr>
<tr>
<td>Incremental Electricity Cost ($/kWh)</td>
<td>0.03637</td>
</tr>
<tr>
<td>Annual Operating time (hours/year)</td>
<td>6240</td>
</tr>
<tr>
<td>*Net steam produced/total fuel burned</td>
<td></td>
</tr>
</tbody>
</table>

5.2 Heat Exchange:

Based on the economic data in Table 1 the simple payback of the heat exchange system is as follows:

- Additional Energy Saved: 6.6 MMBtu/hr
- Annual Savings: 191,000 $/yr.
- Estimated Installation Cost of Equipment: $188,900
- Simple Payback: 1 year
The installation cost includes cost for air washer, waste water pump, temperature gauges, control panel, and heat exchangers. A multiplier of two on the equipment cost is used for the installation cost estimate. Startex is already looking into the direct heat exchange project at present.

5.3 Heat Pumping: (Closed Cycle)

The closed cycle heat pumping payback is:

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Energy Delivered</td>
<td>5.0 MMBtu/hr</td>
</tr>
<tr>
<td>Electric Power Input</td>
<td>400 kW</td>
</tr>
<tr>
<td>Annual Savings</td>
<td>54,000 $/yr.</td>
</tr>
<tr>
<td>Estimated Installation Cost</td>
<td>$160,000</td>
</tr>
<tr>
<td>Simple Payback</td>
<td>3.0 years</td>
</tr>
</tbody>
</table>

The total additional energy saving potential identified in this analysis is 11.6 MMBtu/hr, which is about 28% of the overall steam requirement in the whole plant. The total saving for these options is $245,000 per year. Payback of about 1.4 years is expected if both of these options are implemented.

It is reasonable to assume that 50% of the unaccounted steam usage could be reduced by properly maintaining the steam traps, which will give an additional $94,100.00 per year of savings. A detailed plant steam audit is needed to identify the places of steam loss.

Currently, 2,560 lb/hr of steam is let down through steam turbine for part of the deaerator steam requirement. If the proposed heat recovery scheme is carried out, the deaerator steam usage will drop to the extent that generating pumping power for BFW from steam turbine, will no longer be economical. In that situation, Startex should replace the existing turbine drive on one boiler feed water pump to a motor drive similar to the spare pump. The existing spare pump is motor driven.

In addition to the hot utility savings, the reduction in fuel usage will also reduce the NOX and CO2 emissions in the plant. Annual reduction is estimated to be 22 tons and...
5500 tons respectively for NO\textsubscript{X} and CO\textsubscript{2}. However, no credit has been given for this.

The heat exchange option not only save energy but also reduce the waste water temperature. This will cut down the waste water treatment cost in the future. Currently, no credit has been given for this.

From the preliminary analysis, both heat exchange and heat pump show attractive paybacks. The heat pump system has to be optimized based on the measurement and confirmation of some of the process information, i.e., temperatures, humidity, flow rates of tenter frames and can dryers exhaust, waste water temperature, etc.

The economics of each option and the economic sensitivity to the cost of steam and electricity is summarized in Table 2.
### Table 2
Economic Analysis of Heat Recovery System

<table>
<thead>
<tr>
<th>Electric Cost (C/kWh)</th>
<th>3.637</th>
<th>3.637</th>
<th>3.637</th>
<th>4</th>
<th>4</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual Operating Hours</td>
<td>6240</td>
<td>6240</td>
<td>6240</td>
<td>6240</td>
<td>6240</td>
<td>6240</td>
</tr>
<tr>
<td>Steam Cost ($/MMBtu)</td>
<td>4.42</td>
<td>5.08</td>
<td>5.5</td>
<td>4.42</td>
<td>5.08</td>
<td>5.5</td>
</tr>
</tbody>
</table>

**Heat Integration**

<table>
<thead>
<tr>
<th>Savings (MMBtu/hr)</th>
<th>8.05</th>
<th>8.05</th>
<th>8.05</th>
<th>8.05</th>
<th>8.05</th>
<th>8.05</th>
</tr>
</thead>
<tbody>
<tr>
<td>Annual Savings ($/yr.):</td>
<td>222025</td>
<td>255179</td>
<td>276276</td>
<td>222025</td>
<td>255179</td>
<td>276276</td>
</tr>
<tr>
<td>Installed Cost ($)</td>
<td>160000</td>
<td>160000</td>
<td>160000</td>
<td>160000</td>
<td>160000</td>
<td>160000</td>
</tr>
<tr>
<td>Simple Payout (years):</td>
<td>0.7</td>
<td>0.6</td>
<td>0.6</td>
<td>0.7</td>
<td>0.6</td>
<td>0.6</td>
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</table>

**Heat Pump Steam**

<table>
<thead>
<tr>
<th>Savings (MMBtu/hr)</th>
<th>5</th>
<th>5</th>
<th>5</th>
<th>5</th>
<th>5</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Input (kW)</td>
<td>360</td>
<td>360</td>
<td>360</td>
<td>360</td>
<td>360</td>
<td>360</td>
</tr>
<tr>
<td>Annual Savings ($/yr.):</td>
<td>56202</td>
<td>76794</td>
<td>89898</td>
<td>48048</td>
<td>68640</td>
<td>81744</td>
</tr>
<tr>
<td>Installed Cost ($)</td>
<td>190000</td>
<td>190000</td>
<td>190000</td>
<td>190000</td>
<td>190000</td>
<td>190000</td>
</tr>
<tr>
<td>Simple Payout (years):</td>
<td>3.4</td>
<td>2.5</td>
<td>2.1</td>
<td>4.0</td>
<td>2.8</td>
<td>2.3</td>
</tr>
</tbody>
</table>

| Overall Savings ($/yr.): | 278228 | 331973 | 366174 | 270073 | 323819 | 358020 |
| Overall Installed Cost ($) | 350000 | 350000 | 350000 | 350000 | 350000 | 350000 |
| Overall Payout (years): | 1.3   | 1.1   | 1.0   | 1.3 | 1.1 | 1.0 |

The heat pump payback is more affected by the steam cost than the electricity cost. An increase in current steam cost by 50 cents (or 35 cents in gas cost) will reduce the heat pump payback from 3 years to 2.5 years.
Section 6

UPDATED PLANT DATA AND RESULTS

Subsequent to the earlier work, Startex with the help of Duke Power personnel took more plant measurements to update the data.

6.1 Plant Data

The plant steam system varied throughout the year as shown in Table 3.

Table 3
Steam Flow Measurement

<table>
<thead>
<tr>
<th>Flow Rate</th>
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<tbody>
<tr>
<td>43,300 lbs./hr</td>
<td>Maximum</td>
</tr>
<tr>
<td>21,700 lbs./hr</td>
<td>Minimum</td>
</tr>
<tr>
<td>37,350 lbs./hr</td>
<td>Typical Selected August 1, 1995.</td>
</tr>
</tbody>
</table>

It was decided to use 37350 lbs./hr. as a typical average steam consumption. The steam balance was revised to match this as shown in Figure 12.

The only revision in the process water flow diagram was temperature coming out of the steam heater is 190 °F instead of 200 °F, Figure 13.

Some of the other plant operating parameters and climatic conditions which effects the plant economics was also firmed up per Table 4.

Table 4
Process Design Basis

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Waste Water Temperature</td>
<td>= 175 °F</td>
</tr>
<tr>
<td>Waste Water Flow (Max.)</td>
<td>= 226 GPM</td>
</tr>
<tr>
<td>Waste Water Flow (Min.)</td>
<td>= 152 GPM</td>
</tr>
</tbody>
</table>

REPORT NO: DOE/ID/12862–3
Process Water Requirement (Max.) = 172 GPM
Process Water Requirement (Min.) = 97 GPM

Plant Flow At Max. Flow Rate = 85% of the time
Plant Flow At Min. Flow Rate = 15% of the time

Fresh Water Temperature Winter = 40 °F
Fresh Water Temperature Summer = 80 °F

Duration of Summer = 7 months
Duration of Winter = 5 months

The fuel consumption and other data on Tenter frame dryers are in Appendix B.

Based on these revised data, both the existing heat exchanger network and the proposed modifications were revised. Figure 14 and Figure 15 show in detail the proposed overall heat recovery system. Figure 16 and Figure 17 show in detail the scheme for drier exhaust heat recovery system. Table 5 shows the revised net savings of the proposed heat recovery heat pump system. Table 6 shows the installed cost of both heat exchanger and heat pump. The simple payback for this system came to 1.6 years.

6.2 Results

The heat exchange and heat pump economics were modified based on these new plant measurements. The payback for both heat exchange and heat pump is as follows:

<table>
<thead>
<tr>
<th>Combined Heat Exchange and Heat Pump</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Winter Energy Saved (Max.)</td>
<td>= 12.09 MMBtu/hr</td>
</tr>
<tr>
<td>Winter Energy Saved (Min.)</td>
<td>= 7.76 MMBtu/hr</td>
</tr>
</tbody>
</table>

REPORT NO: DOE/ID/12862–3
Summer (Max.) = 11.61 MMBtu/hr
Summer (Min.) = 6.547 MMBtu/hr

Total Annual Savings = $321,278/year
Power Required (Max. Case) = 400 kWh
Power Required (Min. Case) = 217 kWh
Total Power Cost/year = $75,553/year
Net Savings = $245,725/year

Table 6
Cost

Installed Cost of Heat Exchanger = $188,900
Installed Cost of Heat Pump = $200,000
Total Installed Cost = $388,900

Simple Payback = \( \frac{388,900}{245,725} \) = 1.58 years

The combined heat exchange and pump confirms to have very attractive payback. It was decided to show Startex Mill the sensitivity of electric and fuel gas prices on the sensitivity of payback time.

In order to facilitate the mill to decide to proceed to the next phase, a sensitivity analysis was performed to see the effects of steam price (which was tied to gas price) and the electric price on the payback time Figure 18. Table 7 shows the payback sensitivity in tabular form.
### Table 7
Payback Sensitivity Analysis

<table>
<thead>
<tr>
<th>Steam Cost ($/Mlb.)</th>
<th>Payback (years)</th>
<th>$0.0325/kWh</th>
<th>$0.04/kWh</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.00</td>
<td>1.93</td>
<td>2.11</td>
<td></td>
</tr>
<tr>
<td>4.50</td>
<td>1.64</td>
<td>1.77</td>
<td></td>
</tr>
<tr>
<td>5.00</td>
<td>1.43</td>
<td>1.53</td>
<td></td>
</tr>
<tr>
<td>5.50</td>
<td>1.27</td>
<td>1.35</td>
<td></td>
</tr>
</tbody>
</table>

- Heat Pump payback is affected more by steam cost than electric cost.
- An increase of steam cost by $0.50 (or gas cost by $0.35), will drop the payback to 2.5 years.

This modified result was presented to Startex Mill for convincing them to proceed to the next Phase II B. Inspite of the attractive payback, which impressed the plant personnel, they were unable to proceed to the next Phase II B of design and construction of the heat pump, due to other business reasons.
FIGURES

FOR

SECTIONS 1.0 - 6.0
Figure 1
SIMPLIFIED BLOCK FLOW DIAGRAM OF TEXTILE WET PROCESSING
Figure 2 - DRYING STEP OF THE FINISHING PROCESS
Figure 3 - STARTTEX PROCESS WATER FLOW DIAGRAM.
GRAND COMPOSITE CURVE

GRAND COMPOSITE CURVE (GCP) FOR THE STANFLEX PROCESS AT DTWHT = 20°F

Figure 6
at DT min = 200 F

Figure 7 - Existing Heat Exchangers for Startup

- BFW Heat
- BFW-MU-Pre 175
- Steam Heater Make-Up
- Can-Fresh Pre
- Hot Water from Waste
- STM Ht 200
- Cell Exhauster
- Tent Exhauster
- EVA Condenser 168
- Waste Water 190
- Waste Water 190
- Ah (MWb/m^2)

1.82
1.9
1.963
0.938
2.2
6
0.88
3.56
0.938
5.5
5.5
At 0.1 min = 200°F

Figure 8 - Maximum Heat Recovery for Startup

- BFW Heat 220°F to 175°F
- BFW 175°F to 147°F
- Slm Heater Make-Up 147°F
- Can Fresh Pre

- Hot Water From Waste 80°F
- STM Hot 200°F
- Can Exhust 88°F
- Tent Exhust 4°F
- Evap Condenser 168°F
- Waste Water 190°F
- Waste Water 190°F

All (Min/hr)
Figure 9 - Heat Recovery for Start at DTM, n = 20°F

- BFW Heat
- Sim Heater Make-Up
- Can-Fresh-Pre
- Hot Water from Waste
- STM HI, 200
- Can Exhaust
- Tank Exhaust
- Evaporator 168
- Waste Water 190
- Waste Water 190

AH (MMBtu/hr)
Figure 10 - HEAT PUMP SYSTEM DESIGN
Figure 12: STEAM SYSTEM OF STARTEX MILL

Legend:
- Condensate receiver tank
- Boiler Make-Up Water
- Atmospheric receiver tank
- Bleach basement
- Washers
- Caustic
- Evaporator
- Dryer Chips
- J-Boxes
- Steam Heater
- Process
- Condensate and returning
- Blowdown Boilers
- Flow rate in lb/hr

Symbols:
- X: Unknown
- ST: Steam Treatment
- GR1: Gas Receiver 1
- GR2: Gas Receiver 2
- Fy: Fabric

Important Numbers:
- 38,525
- 219°F
- 35,483 (136°F)
- 12,104 (60°F)
- 23,379 (175°F)
- 13,667
- 4,843
- 3,105
- 2,560
- 2,500
- 1,430
- 879
- 5096
- 2,769
- 2,180
- 1,175
- 37,500
- 23,588
Figure 13: STARTEX PROCESS WATER FLOW DIAGRAM
Figure 14: EXISTING HEAT EXCHANGER FOR STARTER AT DMIN = 30°F

Pinch = 80°F

Pinch = 110°F

AV (MMBTU/hr)
Figure 15: PROPOSED HEAT RECOVERY SYSTEM

- BFV-MU Pre 175
- STM IItr 200
- Can-Fresh Pre
- Cans Exhaust
- Tent Exhaust
- Eva Condenser 168
- Waste Water 190

ΔH (MMBtu/hr)
Figure 16: Heat Recovery System Design
Figure 17: DRIER EXHAUST HEAT RECOVERY ARRANGEMENT

- Air Washer
- Hot Water Tank
- Lint Strainer
- Process Water Heater
- City Water
- To Drain
- 60°F
- 95°F
- 110°F
- 75°F
- Exhaust
- Tenors & Can Driers
- Ambient
- 95°F
Figure 18: STEAM COST ($/Mibu.)
APPENDIX A
Appendix A

BASIC PRINCIPLES AND APPLICATIONS OF PINCH TECHNOLOGY

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

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

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

A.1 PROCESS HEATING AND COOLING

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

A.2 THE HEAT TRANSFER PINCH

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

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

- A higher temperature part which, after complete heat integration, acts as a net heat "sink" or acceptor.
- A lower temperature region which, after complete heat integration, has surplus heat to be rejected. It is thus a net heat "source".
Figure A-1: Heat Transfer Between Process Streams Using A Heat Exchange Network
The temperature that separates the source and sink sections of the process is called the heat transfer "pinch" (Figure A-2). In a properly integrated process, there is no heat transfer from above the pinch to below the pinch. Also, the "temperature driving force" (i.e., the difference in temperature between the hot and cold streams) reaches its minimum value, designated $DT_{\text{min}}$, in the region of the pinch.

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

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

A.3 HEAT PUMPS

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

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

Sometimes, it is possible to use a process vapor stream as the working fluid in heat pumps. These heat pumps are called semi-open cycle heat pumps. The most common semi-open cycle (type 1), is called the mechanical vapor recompression (MVR) heat pump.
Figure A-2: The Heat Transfer Pinch

Figure A-3: Basic Principles of the Heat Pump
Figure A-4: Closed Cycle Heat Pump
The hot process vapors are compressed in a compressor and then condensed in the heat pump condenser to satisfy a process heating requirement at an elevated temperature (see Figure A-5). A less common type of semi-open cycle heat pump (Type 2) has the opposite configuration, i.e., an evaporator instead of a condenser. A liquid stream is vaporized in the evaporator and then compressed to a higher temperature in a compressor (see Figure A-6). This type of heat pump cycle is recommended when a low temperature heat source is available to evaporate a liquid process stream which is required in the vapor phase at a higher temperature. It is important to note that semi-open cycles are only feasible when the process fluid undergoes a phase change; condensation for Type 1 systems, and evaporation for Type 2 systems.

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

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

A.4 APPROPRIATE PLACEMENT OF HEAT PUMPS

The pinch concept leads to useful insights into the appropriate use of heat pumps in industrial processes. Because the below pinch region is a net heat source any heat pump must accept heat in this region if it is to reduce the external cooling requirements of the process. By a similar argument the heat pump must reject its heat to the net heat sink above the pinch to reduce the demands on external utility heating. A heat pump which satisfies these criteria is said to be "appropriately placed" (Figure A-7).
Figure A-5: Semi-Open Cycle Heat Pump (Type 1)

Figure A-6: Semi-Open Cycle Heat Pump (Type 2)
Figure A-7: Appropriate Heat Pump Integration
If a heat pump acts wholly above the pinch, it will reduce the hot utility requirement $Q_H$ by an amount equal to the work input $W$ of the heat pump (see Figure A-8). However, as the unit cost of providing work is normally greater than the unit cost of heating, such an arrangement is generally uneconomical. A heat pump acting entirely below the pinch (Figure A-9) has the net effect of degrading the work input $W$ into waste heat that has to be rejected to the cold utility i.e. the net heat rejected rises from $Q_C$ to $Q_C + W$, which is clearly undesirable.

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

A.5 THE GRAND COMPOSITE CURVE (GCC)

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

The ordinate of the GCC is the so-called "interval temperature." This is a convention to put the hot and cold streams on a common temperature basis, after allowing for the necessary minimum temperature driving force ($DT_{\text{min}}$) between them. Consider the simplest case, where the heat transfer resistance associated with all the hot streams is equal to that associated with all the cold streams. The interval temperature of a hot stream at an actual temperature of $T_H$ is defined to be $T_H - (DT_{\text{min}}/2)$; and that of a cold stream at an actual temperature of $T_C$ is $T_C + (DT_{\text{min}}/2)$. Where the heat transfer resistance of the streams are different, it is necessary to ascribe an appropriate "$DT_{\text{min}}$ contribution," $DT_{\text{cont}},$ between 0 and $DT_{\text{min}},$ to each stream. Heat transfer between such streams is permitted only if $(T_H - T_C) > DT_{\text{min}}$ i.e. if the interval temperature of the hot stream is greater than or equal to that of the cold stream.
Figure A-8: Inappropriate Heat Pump Integration-above pinch

Figure A-9: Inappropriate Heat Pump Integration-below pinch
$Q_H = \text{Hot Utility Load}$  
$Q_C = \text{Cold Utility Load}$  
$T_1 = \text{Interval Temperature}$  
$H = \text{Enthalpy}$

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

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

A.6 HOT AND COLD COMPOSITE CURVES AND AREA TARGETING

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

Varying the extent of heat integration is represented by moving the hot and cold composite curves horizontally relative to one another. Doing so will change the vertical distance between them at their point of closest approach (i.e. the pinch), and thus corresponds to changing values of $DT_{\text{min}}$. The horizontal displacements between the composite curves at their high and low temperature ends are the corresponding values of $Q_H$ and $Q_C$ (the external heating and cooling requirements, respectively) for the process. A decrease in $DT_{\text{min}}$ implies an increase in the level of heat integration. In general, as $DT_{\text{min}}$ decreases the minimum hot and cold utility requirements also decrease.
Figure A-11: The Grand Composite Curve and Heat Pump Integration
Figure A-12: Effect of Heat Integration on Utility Targets and Temperature Driving Forces
Progressively smaller values of $DT_{\text{min}}$ are represented in Figure A-12(a) through A-12(c). However, a decrease in $DT_{\text{min}}$ also implies a decrease in the overall driving force for heat transfer and a resulting increase in heat transfer surface area requirements.

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

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

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

- \( C_A \) = Cost per Unit Area
- \( C_H \) = Cost per Unit of Hot Utility
- \( H \) = Hours of Operation per Year

**Figure A-13:** Hot Utility Consumption Versus Heat Exchanger Area
Figure A-14: Area Targets for the Example Problem
All heat transfer is represented as vertical lines on Figure A-14. This is an idealized representation implying that all matched hot and cold temperatures within the process HEN must be the same as the matched temperatures on Figure A-14. However, with this simplifying assumption, it is possible, using appropriate stream heat transfer film coefficients, to predict the minimum area for a HEN with surprising accuracy. For further details, reference (9) should be consulted.

A.7 APPROPRIATE/INAPPROPRIATE INTEGRATION OF HEAT ENGINES

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

The total hot utility requirement for the Carnot engine and process is \((Q_{H\text{min}} + 3W)\). This requirement can be reduced by integrating the heat engine with the process above the pinch, see Figure A-16. In the "above pinch" integrated arrangement, the engine exhaust heat displaces the process hot utility usage and the total utility requirement falls to \((Q_{H\text{min}} + W)\), a saving of 2W units of heat. Work W has effectively been produced at a marginal efficiency of 100% (neglecting mechanical and electrical losses) since the waste heat from the machine is usefully used to displace process requirements.
Figure A-15: Stand Alone Heat Engine Operation and Process Demand
Figure A-16: Appropriate Heat Engine Integration
Figure A-17a illustrates the same process together with a Carnot engine for which both the heat acceptance and rejection temperatures are colder than the process pinch temperature. As in Figure A-15, a machine efficiency of 33.3% is assumed. The total hot utility requirement for the Carnot engine and process shown in Figure A-17a is \((Q_{H\text{min}} + 3W)\). This requirement reduces to \(Q_{H\text{min}}\) when the heat engine is integrated with the process below the pinch, see Figure A-17b, since the engine heat requirements are supplied by waste process heat.

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

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

A.8 PLACEMENT OF DISTILLATION COLUMNS AND EVAPORATORS

The rules for placing distillation columns and evaporators are essentially the same as those for heat engine placement. The entire distillation or evaporation system should be either above the pinch, or below the pinch to ensure the maximum scope for beneficial heat integration of the condenser and reboiler heat loads. Placing the system such that the reboiler is above the pinch and the condenser is below the pinch (see Figure A-19) is "inappropriate" as it degrades heat across the pinch.
Figure A-17: Appropriate Heat Engine Integration Below the Pinch
Figure A-18: Inappropriate Heat Engine Integration
Figure A-19: Inappropriately Placed Distillation Column or Evaporator
The procedure for correcting inappropriate "cross pinch" placement of distillation columns (4) is based on the concept of "pressure shifting". Reducing the pressure of the column lowers the evaporation and condensation temperatures, and so may allow the temperature of an "above pinch" reboiler to be reduced to below the pinch. Conversely, raising the pressure may allow a "below-pinch" condenser to be raised to a temperature above the pinch. In either case, the result is that the "cross pinch" placement is eliminated and both the reboiler and condenser are restored to the same side of the pinch.

Correction of inappropriate evaporator placements uses precisely the same procedure.

A.9 SUMMARY

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


A-27
APPENDIX B
Appendix B
Fuel Consumption Basis

■ Steam System
  - APLUS simulation
  - No direct measurements available for each user

■ Gas Usage for Tenter Frame
  - Burner size ($3.5 \times 4 = 14$)
  - Newly obtained gas bill indicates only $88,000$ per year at $4.80$ $$/MCF$
  - Tenter frame consumes about $2.8$ MMBtu/hr ($2.8$ MCF/hr) of gas.

■ Tenter Frame Exhaust Gas Flow, Humidity and Temperature
  - Provided by STARTEX