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**NEW INDUSTRIAL HEAT PUMP APPLICATIONS TO A
PETROCHEMICAL PLANT**

PHASE IIA

FINAL REPORT

1995

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**For
U.S. DEPARTMENT OF ENERGY
OFFICE OF INDUSTRIAL TECHNOLOGIES
WASHINGTON, D.C**

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**By
TENSA SERVICES, INC.
HOUSTON, TEXAS**

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TABLE OF CONTENTS

SUMMARY.....	1-1
INTRODUCTION.....	2-1
PROCESS DESCRIPTION.....	3-1
PINCH ANALYSIS	4-1
HEAT PUMP DESIGN.....	5-1
ECONOMIC ANALYSIS.....	6-1
ADDITIONAL BENEFITS OF THE HEAT PUMP	7-1
FUTURE.....	8-1
APPENDIX A.....	A-1
Introduction to Pinch Technology.....	A-1
APPENDIX B.....	B-1
Heat Pump Simulation	B-1
APPENDIX C.....	C-1
Vendor Quotes	C-1

Section 1

SUMMARY

This report summarizes the results of the U.S. Department of Energy (DOE) sponsored study, titled "Adv: ^{<950>} Pump Application and Evaluation," DOE Contract Number ~~DE-FC02-82-OR21400~~². The purpose of this study was to evaluate the energy conservation potential of a heat pump in an industrial site. The proper placement of the heat pump was based on the principles of Pinch Technology. Chevron's refinery at Port Arthur, Texas, was selected as the industrial site for this study. Two energy conservation options were identified for this site with a combined total savings of \$570,000 per year. This represents over 10% reduction in current thermal energy consumption of the process units, which were part of this study. The details of each option are described below:

HEAT EXCHANGE

The first option was a passive heat integration scheme. The heat exchange option calls for increasing the process to process interchanger area by 3376 ft². The economics for this option are:

Heat from steam saved:	3.81 MMBtu/hr
Heat from condensate saved:	7 MMBtu/hr
Percent reduction in heat requirement:	3.7% of the units studied
Investment:	\$101,280
Total savings:	\$236,582/yr
Payback period:	0.43 years

HEAT PUMP

The second option involves a semi-open cycle mechanical vapor recompression heat pump that compresses the steam generated from the reactor exhaust streams of the cyclohexane unit to provide part of the reboiling duty of the benzene column.

The economics is summarized below:

Heat from steam saved:	18.25 MMBtu/hr
Percent reduction in heat requirement:	6.3% of the units studied
Investment:	\$463,315
Net savings:	\$343,572/yr
Payback period:	1.3 years

The proposed heat pump system also reduces the cooling water requirement. In many plants where production is often restricted by the available cooling water capacity, this system is a valid option for debottlenecking. The proposed heat pump also provides environmental emission improvement by reducing the emission of NO_x, SO_x and CO₂. No economic credit has been taken for this additional benefit.

The cyclohexane unit along with the BTX unit or any other column with suitable reboiling range has a good heat pumping potential which should be very attractive for other similar sites to replicate this application.

Section 2

INTRODUCTION

Application of pinch technology to selected industries in an early screening study done by TENSA Services (DOE/ID/12583-1), identified potential for heat pumps in several industrial sectors. Among these, the aromatic production (SIC code 2911) was considered one of the promising sectors for the advanced heat pump placement. Based on this recommendation, the U.S. Department of Energy (DOE) awarded a feasibility study (Phase I) for a BTX plant. As part of this study, a heat pump potential was identified in this type of plant which is the subject of a DOE report (DE-91013698). However, that host site decided not to pursue subsequent phases of the study which consisted of detail design, installation, and monitoring of the heat pump energy savings. A replacement site was therefore selected to demonstrate the industrial heat pump application in the chemical industry.

The aromatic processing plant of Chevron USA, Inc., at Port Arthur, Texas, was selected as the replacement site. The objective of this study was to identify the energy savings potential through heat integration including advanced heat pumps and to develop a detailed design package and cost estimate for the project. This information is essential for the site management to study the economic viability and operability of the proposed heat pump scheme, and to decide to move on to the next phase of installation and testing of the project.

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

Historically, heat pumps were placed to reduce the energy consumption in a particular unit operation such as evaporation or distillation systems. While this approach usually resulted in energy savings in unintegrated processes, integration through heat exchange often had more cost effective energy saving potential.

Later experience has shown that inclusion of heat pumps in the integration scheme, also results in lower process energy consumption.

TENSA Services, in the above referenced DOE sponsored screening study, has developed a systematic method for determining the optimum placement and sizing of industrial heat pumps. This was done through the use of pinch technology, a thermodynamic analysis tool based on the heating and cooling characteristics of the process. Though originally developed for optimizing heat exchanger networks (HEN), pinch technology is now being applied to the integration of the entire range of process components including heat exchangers, heat pumps, heat engines, and various unit processes such as distillation columns, evaporation trains, dryers, etc. Pinch technology is gaining recognition as a valuable process design tool and has pointed the way to many energy-saving ideas in the process industries.

Section 3

PROCESS DESCRIPTION

Chevron's refinery at Port Arthur, Texas is one of the largest refining complexes in the country. This facility consists essentially, among many other units, of crude unit, catalytic reforming unit and downstream processing of aromatic feed (BTX Unit). This study covered the aromatic extraction unit (UDEX) and the cyclohexane unit, which are part of the BTX unit. These units are located right beside each other. The UDEX unit is a liquid-liquid extraction process for separating aromatics from light and heavy paraffins and olefins. In the cyclohexane unit, benzene product from the UDEX unit reacts with hydrogen in the presence of a catalyst to form cyclohexane. The UDEX unit can be further divided into three sections:

- Feed Preparation Section
- Extraction Section
- Product Purification Section

Feed Preparation Section

The feed preparation section contains two distillation columns, the reformate splitter (P-V1), and the depentanizer columns (P-V13) Figure 3-1. The reformate splitter is used to split the total reformate charge from the catalytic reforming unit into two streams, light and heavy reformate. The splitter overhead, or light reformate, contains nearly all of the benzene and some of the toluene that are desirable components to charge to the aromatic extraction section. The splitter bottoms, or heavy reformate, normally contains very little benzene and the rest is toluene. The benzene content of the heavy reformate (splitter bottoms) is maintained less than 0.5%, by controlling the tower reflux rate.

The primary function of the depentanizer is to remove light ends from the feed (light reformate) going to the extractor. The depentanizer separates the light ends, mainly C5's and lighter, from the light reformate and other aromatics feedstocks.

Once these undesirables are removed, the aromatic-rich depentanizer bottoms become the feed to the UDEX extraction section.

Extraction Section

The extractor (U-V101) is used to extract aromatics from paraffins and naphthenes Figure 3-2. The solvent used for the extraction is tetraethylene glycol. The solvent stripper (U-V102) separates or strips the hydrocarbons (primarily aromatics) from the rich solvent coming from the extractor. The stripped or "lean" solvent is then recycled back to the extractor. The non-aromatic hydrocarbons picked up by the solvent are also returned to the extractor as a reflux stream. The aromatics from the stripper overhead receiver are first cooled by cooling water in E-102 before flowing to the column where it is washed with water. The water washed aromatics is then pumped to storage tank T2417. The non-aromatics or raffinate, which is not absorbed by the solvent flowing from the top of the extractor enters first on the shell side of heat exchanger (E101 A/C) where it preheats the extractor feed, and then further cooled in the raffinate coolers (E102 A/B), before flowing into the solvent knockout drum (U-V107). The solvent which is heavier settles in the knockout drum (U-V107) and is recycled back to the extractor along with the fresh feed. Raffinate which settles at the top of (U-V107) and is pumped to the raffinate water wash columns (U-V126) and (U-V106). Washed raffinate is blended into the gasoline pool or used as cracking heater charge stock for the ethylene unit.

From the bottom of the extractor, the rich solvent is pressured through the tube side of (E103 A/B), where it is cooled by vaporizing the stripping water. The steam generated in (E103 A/B), provides the stripping steam to the solvent stripper. The rich solvent then flows to the flash drum that sits on top of the solvent stripper. Liquid from the flash drum enters the blind deck of the stripper where the second stage flashing occurs. The collective overhead vapors from the flash drum and top tray are condensed in the solvent stripper overhead condensers (E106 A/B), and then collected in the overhead receiver (U-V103). A dragstream of benzene recycle from the benzene tower overhead as well as water from raffinate water washes (U-V126, 106) flows into the stripper overhead condenser (U-V103). The hydrocarbons from the stripper receiver (U-V103) are pumped to the bottom of the extractor as reflux. Stripping water is pumped back to the stripper through (E103 A/B).

FIGURE 3-2: EXTRACTION SECTION - UDEX UNIT

Much of the lean solvent from the bottom of stripper returns to the extractor. While a slip stream of the lean solvent flows through the solvent regenerator (V111) to be cleaned and sent back to the stripper.

The aromatics are removed as a side stream from the stripper. After condensing through the fin-fan exchanger (E105 A/B), the aromatics are received into extract receiver (U-V104). The extract is pumped to the extract water wash column (U-V123) to remove remaining solvent. Extract from the water wash column is pumped into tank (T-2417).

Product Purification System

The product purification sections Figure 3-3, contains three clay towers (U-V112, 113, 113A), and the benzene column. The purpose of the clay towers is to remove trace quantities of sulfur compounds, olefins and diolefins from the water washed aromatics stream prior to fractionation.

Charge to the clay tower is heated in the tube side of heat exchanger (E137) by the condensate. It is further heated on the shell side of the clay tower/effluent exchangers (U-E113). The final part of the preheating process is accomplished with 650 psig steam in the preheater (E118A) which raises the clay tower temperature to about 442°F.

The clay tower effluent is sent to the benzene column after first exchanging heat with the clay tower feed. The purpose of the benzene column is to fractionate the extract into benzene and toluene streams. The benzene product is drawn as a side stream so the lighter component can be removed in the overhead rather than in the product stream. The benzene product leaving the tower is cooled in the benzene product cooler E115. The benzene column overhead vapors are condensed in the overhead air cooler (E132 A/E). A portion of the benzene column overhead is recycled back as drag stream to the solvent stripper overhead receiver in the extraction section, and the rest is refluxed back. The bottoms' product from the benzene column is first cooled in exchanger (E117A) and then sent to storage for gasoline blending.

FIGURE 3-3: PRODUCT PURIFICATION SECTION - UDEX UNIT

Cyclohexane Section

The cyclohexane unit Figure 3-4, contains three reactors (V201, V202, V203), separator (V204), and a stabilizer (V206). The purpose of the reactors is to react benzene with hydrogen in the presence of a catalyst to form cyclohexane.

The total benzene flow to the unit is charged to the three reactors. A third of the benzene feed normally goes to each reactor. The effluent of the first reactor (V201) goes to the second reactor (V202), combined with fresh benzene feed. The (V202) effluent combined with the rest of the benzene goes to (V203). Hydrogen is charged to the reactors in series. Both the make-up and recycle hydrogen combined with a third of the benzene feed, enters (V201). Also entering the first reactor is the recycle cyclohexane stream which helps to quench the reaction.

The combined feed of benzene, recycle cyclohexane, recycle hydrogen, and make-up hydrogen is heated to about 280° F, before entering the reactors. The combined feed of reactor #1 exchanges heat with the #3 reactor effluent in heat exchanger (E201 A/B). The intercooler system cools the effluent from #1 and #2 reactors by generating low pressure steam. Condensate flows to intercooler exchangers (E202) and (E203) from the intercooler receiver (V214). The low pressure steam generated in the intercoolers returns to the intercooler receiver. Through an intercooler receiver pressure controller, steam is allowed to enter and condense in the intercooler condenser (E213), it then returns to (V214) in a closed loop. The intercooler receiver is just like a steam drum.

After the #3 reactor effluent stream has given up heat to the combined feed to #1 reactor in the heat exchanger (E201 A/B), it is further cooled in the reactor effluent condensers (E204 A/B) with cooling water. The reactor effluent then enters the product separator (V204). Some of the liquid product is recycled back to the first reactor and the remaining amount is sent to the stabilizer column. The vapor containing most of the hydrogen is returned to the reactors after getting compressed by the recycle hydrogen compressors. The hydrogen spill from (V204) is cooled in the separator gas exchanger (E205) and the gas chiller (V206). This allows cyclohexane to condense out of the gas streams and be recovered in the refrigeration receiver (V205). The recovered cyclohexane then flows into the stabilizer charge stream.

FIGURE 3-4: CYCLOHEXANE UNIT

The purpose of the stabilizer column is to remove light ends from cyclohexane product. The stabilizer distills the light ends (primarily H_2 and CH_4) overhead from the cyclohexane product. The column operates with partial condenser and no liquid overhead product is drawn.

The stabilizer feed absorbs heat in the tube side of the exchanger (E222 A/B), from the bottoms stream and passes through the shell side of the stabilizer feed preheater (E207). The stabilizer overhead vapors are partially condensed in the stabilizer overhead condenser (E208), and collected in the stabilizer receiver (V207). The stabilizer bottoms go through the shell side of the heat exchanger (E222) to give up heat to the stabilizer feed stream. It is then further cooled with cooling water in cooler (E223) and then further cooled in (E-210) before flowing to the storage tank.

Basis of Study

This study is based on the total reformat charge of 35,000 BPOD (barrels per operating day) to the reformat splitter. The designed rate for total extract production is 11,400 BPOD, with 8,100 BPOD of benzene and 3,300 BPOD of Toluene-Xylene mix. Table 3-1 summarizes the economic data used in this study.

Table 3-1
ECONOMIC DATA

650 # Steam (\$/MMBtu)	= 3.02
125 # Steam (\$/MMBtu)	= 2.50
Condensate (\$/MMBtu)	= 2.00
Cooling Water Cost (\$/MMBtu)	= 0.19
Electricity Incremental Cost (\$/kWh)	= 0.024
Annual Operating Time (hours/year)	= 8,400
Heat Exchanger Installed Cost (\$/ft ²)	= 30

Section 4

PINCH ANALYSIS

Pinch Analysis is a systematic method of analyzing the energy requirements of a plant. An important result of Pinch Analysis is the ability to set performance targets prior to design. It is possible from process data alone to confidently predict the optimum energy and surface area requirement for any process. The details of Pinch Technology and the development of various curves used in Pinch Analysis are shown in Appendix A.

Starting from the individual streams' temperature and enthalpy information, the first step in pinch analysis is to construct "composite curve" of all the hot and cold streams in the process, by simple addition of the heat contents over the temperature intervals in the problem. Figure 4-1 shows these curves for all the hot and cold streams identified in this study. The overlap between the two composite curves represents the maximum amount of heat recovery possible within the process. The "over-shoot" of the cold composite represent the minimum amount of external heating (hot utility Q_H) requirement. The "over-shoot" of the hot composite represents the minimum amount of external cooling (cold utility Q_C) 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 ΔT_{min} . As ΔT_{min} increases, utility requirements increases, however, the heat exchange surface needed to recover the energy decreases as a result of the larger approach temperature. For any ΔT_{min} , the composite curves define the minimum utility requirements for the process and thus establish the energy targets for the process.

A grid representation of the existing heat exchanger network is shown in Figure 4-2. In this figure, the lines with the arrows pointing to the right represent the hot streams that must be cooled to their target temperatures. The lines pointing to the left the cold streams that must be heated. Two dumbbells connected with a line represent the process to process exchangers. The circles with H or C represent the heaters and coolers respectively.

Figure 4-1
Hot and Cold Composite Curves

Figure 4-2
EXISTING HEN AT $DT_{min} = 20^{\circ}F$

Figure 4-2 shows the existing heat exchanger network at the optimum Dt_{min} of 20°F . the pinch interval temperature at this Dt_{min} is 275°F , as shown by the straight line in the middle.

This representation allows the violations to be identified based on pinch design principles. The shaded dumbbells indicate violations of process to process heat exchange, being exchanging heat across the pinch. Other violators of pinch are those utilities heaters or coolers, being heated below the pinch or cooled above the pinch.

Figure 4-3 is the MER (Maximum Energy Recovery) design based on pinch design rules, to correct the violations in the existing network. This design call for eight new heat exchangers. However, it results in multiple reboilers in the benzene reboiling stream, which is normally impractical to implement. The matches between splitter and benzene column are not economical, due to long distance (500 ft.) separating them.

Two design options evolved from the MER and are presented in Figure 4-4 and 4-5. Option 1 requires four new heat exchangers and an increase in surface area of the existing heat exchanger between clay feed and effluent stream. Table 4-1 summarizes the economics of this design.

Table 4-1
SUMMARY OF HEAT INTEGRATION ECONOMICS

Chevron	HX Option 1	HX Option 2
650 # Steam, MMBtu/hr saved	3.81	2.55
Cost Savings, \$/yr	96,812	64,796
125 # Steam, MMBtu/hr	0	0
Cost Savings, \$/yr	0	0
Condensate, MMBtu/hr saved	7	7.4
Cost Savings, \$/yr	122,304	129,293
Reduction in Cooling Water Load, MMBtu/hr	10.83	9.99
Cost Savings, \$/yr	17,466	16,112
Total Savings, \$/yr	236,582	210,201
Total HX Surface Area, ft^2	3,376	1,398
HX Installation Cost	101,280	41,940
Payback, yrs	0.43	0.2

Figure 4-3
MER DESIGN AT $DT_{min} = 20^{\circ}F$

Figure 4-4
REVISED HEN DESIGN OPTION 1

Figure 4-5
 REVISED HEN DESIGN OPTION 2

Heat Pump Selection

Pinch Technology plays a major role in the selection of streams for heat pump application. Although the criterion for the selection of streams for heat pump application is described in detail in Appendix A, a brief description is provided in the following subsection:

Selection of Streams for Heat Pump Application:

In order for heat pump to be appropriately placed according to Pinch principles, the heat pump must meet the following criteria:

- The temperature of the source stream must be less than the pinch temperature.
- Preferably, the source stream must be a condensing vapor (so that it has a large heat duty over a small temperature range).
- There must be a heat demand above the pinch to utilize the heat duty.

The heat pump potential is evaluated using the "Grand Composite Curve" or GCC of the process. The GCC gives a "profile" of energy requirement of the process. The corresponding GCC for the process is shown in Figure 4-6. The pinch temperature is shown on the GCC where the curve touches the temperature axis. Above the pinch temperature, the process is a net heat sink, where only the hot utility is required. Below the pinch, the process is a net heat source, where surplus heat has to be removed. The shape of the GCC indicates that a large amount of rejected heat is available below the pinch as condensing heat. This heat can be lifted-up to a higher temperature level to satisfy part of the heat demand through semi-open cycle heat pumps. A semi-open cycle heat pump uses a process stream, usually a condensable vapor below the pinch, as the working fluid. By compressing this condensable vapor to a higher pressure to raise its condensing temperature, this low temperature heat source is upgraded to match a process heat requirement above the pinch.

The source of streams available for the heat pump application below the pinch temperature are the overhead condensing streams of the splitter and the stripper. Another possible heat source for heat pump is the condensing steam from the intercooler receiver (V214). The benzene reboiler stream has the lowest lift required and is the most appropriate for heat sink. Three heat pump options are therefore studied.

Figure 4-6
Grand Composite Curves at $DT_{min} = 20^{\circ}F$

One matches the stripper overhead with benzene reboiler (Figure 4-7), the second matches splitter overhead with benzene reboiler, and the third option is to match the intercooler steam with benzene reboiler.

During the course of this study, the Chevron Company announced a major cut back in production. Chevron then went through an extensive process realignment for the reduced production. This plant-wide modification diverted to other more profitable operations, instead of supplying all of it to extraction unit. As a result of this policy, the feed to the UDEX unit varies day by day, depending on the demand of other operations in the plant. The details of these modifications were not revealed to TENSA. However, the charge rate in the UDEX unit was significantly reduced to less than one half of its originally designed capacity which was used earlier in the pinch analysis. However, the benzene column was kept at designed flow because aromatics charge are also available from other plant operations. The cyclohexane production has been increased from 1500 BPOD to 2200 BPOD.

With these process realignment in the plant, the available condensing heat in either columns (splitter and stripper) will be reduced at least by half. Further study of the condensing curve of the overhead streams indicated that the overhead condensing occurs in a much wider temperature range than first estimated. These can be seen clearly in Figures 4-8 and 4-9. From Figure 4-8, it is evident that the major portion of the condensation in the overhead stream from the stripper occurs between 200° F and 170° F, instead of 230° F to 220° F. This temperature difference is due to impurities present in the overhead. As a result, the temperature lift in the proposed heat pump system is increased by 50° F. This makes the heat pump system less attractive than what was identified earlier. Similar conclusion can also be drawn for the splitter heat pump system as shown in Figure 4-9.

The only viable heat pump option, is the reactor exhaust streams in the cyclohexane section. The reactor exhaust streams are presently collected and cooled in a closed loop intercooler system to control the feed stream temperature to the next reactor. Currently, low pressure steam is being generated from the reactor exhaust streams. Since the cyclohexane unit itself does not present any opportunity to utilize this steam, at present the low pressure steam generated is condensed by cooling water in the intercooler condenser.

Figure 4-7 Semi-open Cycle Heat Pump Arrangement

CHEVRON STRIPPER

Figure 4-8
CHEVRON Stripper Overhead

Figure 4-9
CHEVRON Splitter Overhead

A heat pump which compresses the low pressure steam generated in this unit to a higher pressure to be used in the benzene column reboiler, is therefore a better alternative to replace the originally proposed heat pump system.

The lessons learned during this phase was the realization that a good simulation program is essential to reconcile all the plant measured data. Even in the simple case of generating a condensing curve, the simulation results can reveal useful information that is very desirable in order to correctly evaluate the heat pump potential. A pure component condenses at constant temperature at fixed pressure. A mixture condenses over a temperature range according to the changing composition. In this study, we have learned that even a small portion of impurity can cause a drastic change in the condensing behavior. The common practice of assuming a 10° F variation in condensing temperature has to be checked during detailed design phase.

Section 5

HEAT PUMP DESIGN

Heat Pump Potential

The heat pump design on the cyclohexane unit and the benzene column is shown in the attached process flow diagram Figure 5-1. The 25 psig steam generated from the reactor exhaust streams flows from the intercooler receiver to a steam compressor. A centrifugal steam compressor compresses this 25 psig steam to 53 psig, which has a corresponding condensing temperature of 300° F. This compressed steam is then used to supply part of the benzene column reboiler duty.

The benzene column has two reboilers in place (E-134A & E-134B). Currently, only one is used for the reboiling service, the other is used as a spare. The proposed heat pump system is designed to take advantage of the existing spare reboiler. With a fixed amount of surface area, and a known overall heat transfer coefficient between steam and benzene column bottoms fluid, the total heat transferred is directly proportional to the log mean temperature of the two streams, which is also decided by the compressor discharge pressure of the compressed steam. With the increased cyclohexane production in the plant, the steam generated is also increased from the original amount of about 12,000 lb/hr to the measured 19,000 lb/hr. At this flow rate of steam, the discharge pressure of the compressor is determined to be 53 psig to best utilize the existing spare reboiler. A simulation of the heat pump system is attached in Appendix B.

A knock-out drum is used before the steam compressor to ensure that no liquid gets into the compressor. The steam condensate from (E-134B) does not return to the intercooler receiver, but flows to the existing condensate return lines. This will save the additional piping cost for the condensate. The make up condensate required at the intercooler receiver will be supplied by the existing condensate line connected to the intercooler receiver.

TENSAS SERVICES, INC		NOTE:	
CHEVRON STEAM COMPRESSOR		— EXISTING LINE	
PFD OF CHEVRON HEAT PUMP SYSTEM		— NEW LINE	
DWG NO 01-86-C-01		NEW EQUIPMENT	

FIGURE 5-1 STEAM COMPRESSOR

The design is flexible enough to allow both the benzene tower and the rectors system to operate in the present mode, should the heat pump fail. In this case, the system will revert to the present mode of operation. The benzene column will utilize more of 125 psig plant steam in E134A, and the generated steam from the reactor exhaust will be condensed against the cooling water.

The controls for this heat pump are shown on the Process and Instrument Diagram (P&ID) in Figure 5-2. The thick line on this figure refers to the new piping for the heat pump. The block valves on lines from the intercooler receiver allow the operator to switch from the heat pump mode to the normal mode. The compressor has a kick back line going to the suction of the knock-out drum to be used for the control. The following tables summarize the new equipment specifications for the heat pump operation:

Table 5-1
COMPRESSOR PROCESS SPECIFICATION

Compressor Inlet Temperature	= 266.8° F
Compressor Inlet Volume	= 9.6 MMSCFD
Compressor Inlet Pressure	= 39.7 psia
Compressor Brake Horsepower	= 446 hp
Compressor Discharge Pressure	= 67.7 psia
Compressor Discharge Temperature	= 398.6° F
Compressor Ratio	= 1.71

Table 5-2
HEAT PUMP OPERATION

Reboiler (HEX 4) Heat Exchanger	
Shell Inlet Temperature and Pressure	= 258° F and 9 psig
Shell Outlet Temperature and Pressure	= 265° F and 9 psig
Tube Inlet Temperature and Pressure	= 398.7° F and 67.7 psia
Tube Outlet Temperature and Pressure	= 297.2° F and 64.7 psia
Total Duty = 18.24 MMBtu/hr	
Total Area = 6,485 ft ²	

New Knock Out Drum (SP-100):

Drum Temperature = 267° F top and bottom

Drum Pressure = 25 psig

Filling Rate = 19,000 lb/hr

Inside Diameter = 4'6"

Drum Length T-T = 10'0" with mist pad.

Section 6

ECONOMIC ANALYSIS

Tables 6-1 to 6-3 contain the break down of the major heat pump components' costs including piping, instruments and installation cost estimates.

The overall savings capital cost and simple payback of the heat pump are shown on Table 6-4. The total savings for the heat pump option is 18.25 MMBtu/hr of heat provided by the steam and 17.11 MMBtu/hr of cooling provided by the cooling water. The net savings after subtraction of compression work are \$343,572/year. Table 5-4 shows the economics based on these savings. The simple payback period is 1.35 years, which should be very attractive for other similar sites to replicate this idea.

Table 6-1
HEAT PUMP INVESTMENT

Compressor Cost (ATLAS COPCO Quote)	\$170,000
Motor (WP II) With Explosion Proof Box (Vendor Quote)	\$59,000
Gear Box	\$25,000
Compressor and Motor Installation (50% of Equipment Cost)	\$127,000
Knockout Drum (Vendor Quote)	\$6,133
Knockout Drum Installation (Estimate)	\$3,000
Piping	\$15,000
Piping Installation (Estimate)	\$15,000
Valves and fittings	\$17,102
Installed Instruments (Estimate)	\$26,080
TOTAL INVESTMENT	\$463,315

Table 6-2
VALVES AND FITTINGS

	(\\$)
2 of 3 X 4 Relief Valves	6,092
2 of Drain Valves	82
1 of 2" Control Valve	2,350
1 of 6" Gate Valve	1,012
1 of 12" Gate Valve	2,414
1 of 10" Check Valve	1,735
1 of 1" Gate Valve	90
1 of 1" Control Valve	1,485
Fittings	1,842
TOTAL	17,102

Source(s): Greson Technical Sales and Service

Table 6-3
INSTRUMENTS

	(\\$)
2 Control Loops	4,000
3 Flow Indicators	1,623
6 Temperature Indicators	672
2 Pressure Indicators	1,688
Level Controller	2,325
Local Processing Unit	4,732
Installation	11,040
TOTAL	26,080

Source(s): Greson Technical Sales and Services

Table 6-4
HEAT PUMP SAVINGS AND PAYBACK

Total Hot Utility Savings (MMBtu/hr)	18.25
Total Cold Utility Savings(MMBtu/hr)	17.11
Compression Work (hp)	446
Cost of Steam (\$/MMBtu)	2.5
Cost of Cooling (\$/MMBtu)	0.19
Power Cost (\$/kWh)	0.024
Plant Annual Operating Hours	8400
Total Hot Utility Savings	\$383,250/yr
Total Cold Utility Savings	\$27,308/yr
Compression Cost	\$66,986/yr
Net Savings	\$343,572/yr
Investment	\$463,315
Simple Payback	1.35 YEARS

Section 7

ADDITIONAL BENEFITS OF THE HEAT PUMP

The proposed heat pump scheme has made it possible to reduce the cooling water load by 17.11 MMBtu/hr. This is a significant reduction for many plants whose production rate are sometimes limited by its cooling water capacity.

The proposed heat pump also provides improvement in environmental emissions by reducing the emission of NO_x , SO_x and CO_2 . No economic credit has been taken for this additional benefit.

Section 8

FUTURE

The current uncertainty regarding the future of this refinery, prevents any firm decision by Chevron if they would like to proceed to the next Phases IIB & III of this study. The Phase IIB consists of procurement and installation of the equipment and finally, after the heat pump is commissioned as part of the Phase III, data will be collected to verify the actual savings against the savings predicted by this study.

Appendix A

BASIC PRINCIPLES AND APPLICATIONS OF PINCH TECHNOLOGY

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

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

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

A.1 PROCESS HEATING AND COOLING

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

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

A.2 THE HEAT TRANSFER PINCH

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

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

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

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

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

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

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

A.3 HEAT PUMPS

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

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

Sometimes, it is possible to use a process vapor stream as the working fluid in heat pumps. These heat pumps are called semi-open cycle heat pumps. The most common semi-open cycle (type 1), is called the mechanical 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_{min}) between them. Consider the simplest case, where the heat transfer resistance associated with all the hot streams is equal to that associated with all the cold streams. The interval temperature of a hot stream at an actual temperature of T_H is defined to be $T_H - (DT_{min}/2)$; and that of a cold stream at an actual temperature of T_C is $T_C + (DT_{min}/2)$. Where the heat transfer resistance of the streams are different, it is necessary to ascribe an appropriate " DT_{min} contribution, " DT_{cont} " between 0 and DT_{min} , to each stream. Heat transfer between such streams is permitted only if $(T_H - T_C) > DT_{min}$ i.e. if the interval temperature of the hot stream is greater than or equal to that of the cold stream.

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

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

Q_H = Hot Utility Load

Q_C = Cold Utility Load

T_1 = Interval Temperature

H = Enthalpy

Figure A-10: The Grand Composite Curve

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

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

A.6 HOT AND COLD COMPOSITE CURVES AND AREA TARGETING

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

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

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

Increasing
Heat
Integration

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

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

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

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

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

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

C_A = Cost per Unit Area

C_H = Cost per Unit of Hot Utility

H = Hours of Operation per Year

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

Figure A-14: Area Targets for the Example Problem

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

A.7 APPROPRIATE/INAPPROPRIATE INTEGRATION OF HEAT ENGINES

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

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

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

Figure A-16: Appropriate Heat Engine Integration

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

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

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

A.8 PLACEMENT OF DISTILLATION COLUMNS AND EVAPORATORS

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

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

Figure A-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.

Appendix B

HEAT PUMP SIMULATION

ECHO PRINT OF INPUT DATA

*CHEVRON STEAM COMPRESSOR

C- HEAT PUMP DESIGN

C- KNOCKOUT DRUM

FLASH 2 = KNCK, 1, -11, -12, VERTICAL, K CO(FT/SEC) = .35

HEAT EXCHANGER 20 = PREH, 11,33, -21, -34, TEM OUT (F) = 275,

C- TEMPERATURE APP = 10

U(BTU/HR/FT²/F) = 80

DELTA PRESSURE = 2,2

COMPRESSOR 3 = COMP,21,-33,

ENTROPY OPTION,EFFICIENCY = 0.75

PRESSURE OUT (PSIG) = 60,

ELECTRIC DRIVE,

SIZE LINES

USE STREAMS = 1,11,34,

DIRECTION OF FLOW = 3*HORIZONTAL

MAXIMUM DELTA PRESSURE PER HUNDRED FEET = 3*0.5

LEN = 20, 20, 400,

SIZE LINES

C- USE STREAMS = 49,64,50,155

C- DIRECTION OF FLOW = 2*HORIZONTAL,DOWN,UP

C- MAXIMUM DELTA PRESSURE PER HUNDRED FEET = 3*2

C-STREAM MANIPULATOR 1 =SM1,101,-1,-2,

C- MULTIPLY,

C- SAME FACTOR,

C- FACTORS = 1.1,

C- HEAT EXCHANGER MODULE WHERE COMPRESSOR IS BENZENE REBOILER HEATER

HEAT EXCHANGER 4 = E134B, 34,-35,

C- HEAT EXCHANGER SPECIFICATION

TEMPERATURE OUT (F) = 300,

C- U(BTU/HR/FT²/F) = 75

C- SIMPLE, RATE, NUMBER OF TUBES = 500, TUBE PASS = 2,

C- EXCHANGER TYPE = AES, SHELL DIAMETER (IN) = 32,

C- SHELL FOULING FACTOR = .001, TUBE FOULING FACTOR = .002,

C- TUBE MATERIAL = 1, TUBE LENGTH = 16,

DELTA PRESSURE = 3,3,

C- AREA = 2200

C- TUBE PITCH (IN) = 1.0, TUBE INSIDE DIAMETER (IN) = .584,

C- METHOD = BELL, BAFFLE SPACING (IN) = 10

C- BAFFLE CUT = .25

COND CURVE,

CURVE INCREMENT = 20,

C- INLINE FORTRAN STATEMENTS

C- TO CALCULATE THE PAYOUT OF THE BEST HEAT PUMP

CHEVRON STEAM COMPRESSOR

PAGE 2

C- CALCULATE COMPRESSION COST FOR BOTH HEAT PUMPS

F- $C3KW = GET(COMP(3), CAL\ ELE)$

F- $C3COST = C3KW * 8600 * .024 / .95$

C- CALCULATE COMPRESSOR INVESTMENT

F- $C3BHP = -1 * GET(COMP(3), CAL\ WOR)$

F- $TEMP1 = GET(STR(21), TEM)$

F- $TEMP2 = GET(STR(33), TEM)$

C- CALCULATE TEMPERATURE LIFT

F- $C3DT = TEMP2 - TEMP1$

C- CALCULATE COMPRESSOR INVESTMENT

F- $C3INV = 2.528 * 342 * C3BHP * C3DT ** (-.48777)$

C- COMPRESSOR COST TIMES 3 FOR INSTALLATION AND DRIVER

C- CALCULATE HEAT EXCHANGER INVESTMENT (\$40/FT²)

F- $AREA3 = GET(HEA\ EXC(20), ARE)$

$AREA1 = GET(HEA\ EXC(4), ARE)$

C-F- $AREA2 = GET(HEA\ EXC(5), ARE)$

F- $HP1ARE = (AREA1 + AREA3) * 40$

C- CALCULATE TOTAL INVESTMENT

F- $HP1TOT = 2.0 * C3INV + HP1ARE$

C- CALCULATE ENERGY SAVED

F- $SAV3 = -1 * GET(HEA\ EXC(20), CAL\ DUT)$

C-F- $SAV2 = GET(HEA\ EXC(5), CAL\ DUT)$

F- $SAV1 = GET(HEA\ EXC(4), CAL\ DUT)$

F- $HP1SAV = SAV3 * 2.5 * 8600 / 1E6 - C3COST$

C- NATURAL GAS AT \$2/MMBTU, 8400 HOURS/YEAR, 82.5% EFF

C- PAYOUT CALCULATIONS

F- $HP1PAY = HP1TOT / HP1SAV$

F- WRITE (6,2)C3DT

F- WRITE (6,5)HP1TOT

F- WRITE (6,6)HP1SAV

F- WRITE (6,10)C3INV

F- WRITE (6,16)C3BHP

F- WRITE (6,20)HP1PAY

F- WRITE (6,45)C3COST

F- WRITE (6,50)SAV1

F- WRITE (6,65)AREA1

CHEVRON STEAM COMPRESSOR

PAGE 3

F- 2 FORMAT(1X,///,1X,'COMPRESSOR 3 TEMP LIFT (F) = ',F16.2)
F- 5 FORMAT(1X,///,1X,'HEAT PUMP 1 INVESTMENT (DOLLAR) = ',F16.2)
F- 6 FORMAT(1X,///,1X,'HEAT PUMP 1 SAVING (DOLLAR/YR) = ',F16.2)
F- 10 FORMAT(1X,///,1X,'COMPRESSOR 3 INVESTMENT (DOLLAR) = ',F16.2)
F- 16 FORMAT(1X,///,1X,'COMPRESSOR 3 BHP = ',F16.2)
F- 20 FORMAT(1X,///,1X,'HEAT PUMP 1 PAYOUT (YR) = ',F8.2)
F- 45 FORMAT(1X,///,1X,'COMPRESSOR 3 POWER COST = ',F16.2)
F- 50 FORMAT(1X,///,1X,'HEX4 DUTY = ',F16.2)
F- 60 FORMAT(1X,///,1X,'HEX5 DUTY = ',F16.2)
F- 65 FORMAT(1X,///,1X,'HEX4 AREA = ',F16.2)

GENERAL,

COMPONENTS = 62,

TP (F,PSIG) 1 = 268,25

FLOW (LB/HR) 1 = 19000.,

GENERAL

C- INITIALIZE RECYCLE STREAM

C- STREAM 151 IS A RECYCLE STREAM

FLOW 33 = FLOW 1

TP (F,PSIG) 33 = 320, 55

RECYCLE STREAM = 33,

CONVERGE ON TEMPERATURE = 33

CONVERGENCE TOLERANCE = .01

MAXIMUM = 10

PRINT FREQUENCY = 5

GENERAL

C PRINT STREAM = NONE

C AUX FILE

C CHECK INPUT

END

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ASA46792 LICENSEE:- TENSA SERVICES HOUSTON ALR 10 386/20 10:31:54

CHEVRON STEAM COMPRESSOR

PAGE 4

FORTRAN COMPILATION

ELAPSED TIME- 00:00:01.260 IN COMPILATION

ELAPSED TIME- 00:00:00.550 IN LINKAGE

CHEVRON STEAM COMPRESSOR

PAGE 5

STREAM THERMODYNAMIC INITIALIZATION

STREAM NUMBER	KVALUES	ENTHALPIES	DENSITY
1	STD	STD	STD
33	STD	STD	STD

THE FOLLOWING STREAMS ARE FEED STREAMS TO THE PROCESS

STREAM NUMBER

1

THE FOLLOWING STREAMS ARE PRODUCT STREAMS FROM THE PROCESS

STREAM NUMBER

12
35

CHEVRON STEAM COMPRESSOR

PAGE 6

++SYSTEM VARIABLES++

NUMBER OF COMPONENTS 1

COMPONENTS USED... 62 WATER

ENTHALPY DATUM IS IDEAL GAS AT 32 DEGREES F

ENERGY SIGN CONVENTION: WORK PRODUCED BY THE EQUIPMENT IS POSITIVE

HEAT TRANSFERRED TO THE EQUIPMENT IS POSITIVE

STANDARD CONDITIONS ARE STP 60.00 DEG F AND 14.696 PSIA

NTP 0.00 DEG C AND 1.00 ATM

API 15.00 DEG C AND 1.00 ATM

LIQUID GAL AND BBL ARE U.S. GAL AND BBL

RECYCLE ACCELERATION STREAMS 33,

RECYCLE LOOP CONVERGENCE TOLERANCE 0.0100

MAXIMUM RECYCLE LOOPS 10

AUTOMATIC RECYCLE CALC. SEQUENCE USED

+++++ GENERAL THERMOPHYSICAL PROPERTY OPTIONS +++++

VALUES	ENTHALPIES	DENSITIES	VAPOR FUGACITIES
=====	=====	=====	=====
STD	STD	STD	STD VAP

VISCOSITY		THERMAL CONDUCTIVITY	
LIQUID	VAPOR	LIQUID	VAPOR
=====	=====	=====	=====
NBS81	NBS81	NBS81	NBS81

SURFACE TENSION

=====

STD

WATER IS TREATED AS AN IMMISCIBLE COMPONENT

CHEVRON STEAM COMPRESSOR

PAGE 7

++++UNIT CALCULATION SEQUENCE ANALYSIS

INLET STREAMS	EQUIPMENT			OUTLET STREAMS
	TYPE	NUMBER	NAME	
1	FLA	2	KNCK	11 12
*****BEGIN RECYCLE LOOP 1*****				
*****RECYCLE STREAM = 33 CONNECTING FROM (3) COMP				
TO (20) PREH				
11 33	HEAEXC	20	PREH	21 34
21	COMPRE	3	COMP	33
*****END RECYCLE LOOP 1				
34	HEAEXC	4	E134B	35

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PAGE 8

/ CALLING EQUIPMENT 2 + FLA = KNCK
KVALUES - STD ENTHALPY - STD DENSITY - STD
ELAPSED TIME - 00:00:00.05 - IN EQUIP NO + 2 +

CHEVRON STEAM COMPRESSOR

PAGE 9

++++++ BEGIN RECYCLE LOOP 1

++++ BEGIN ITERATION 1 OF RECYCLE LOOP 1 +++++

NOW CALLING EQUIPMENT 20 + HEAEXC = PREH

KVALUES - STD ENTHALPY - STD DENSITY - STD

ELAPSED TIME - 00:00:00.16 - IN EQUIP NO + 20 +

NOW CALLING EQUIPMENT 3 + COMPRE = COMP

KVALUES - STD ENTHALPY - STD DENSITY - STD

ELAPSED TIME - 00:00:00.22 - IN EQUIP NO + 3 +

CHEVRON STEAM COMPRESSOR

PAGE 10

ITER 2 RECYCLE LOOP 1

```
+++++ BEGIN ITERATION 2 OF RECYCLE LOOP 1 +++++
NOW CALLING EQUIPMENT 20 + HEAEXC = PREH
KVALUES - STD ENTHALPY - STD DENSITY - STD
ELAPSED TIME - 00:00:00.00 - IN EQUIP NO + 20 +
NOW CALLING EQUIPMENT 3 + COMPRE = COMP
KVALUES - STD ENTHALPY - STD DENSITY - STD
ELAPSED TIME - 00:00:00.05 - IN EQUIP NO + 3 +
```

CHEVRON STEAM COMPRESSOR

PAGE 11

ITER 3 RECYCLE LOOP 1

+++++ BEGIN ITERATION 3 OF RECYCLE LOOP 1 +++++
NOW CALLING EQUIPMENT 20 + HEAEXC = PREH
KVALUES - STD ENTHALPY - STD DENSITY - STD
ELAPSED TIME - 00:00:00.00 - IN EQUIP NO + 20 +
NOW CALLING EQUIPMENT 3 + COMPRE = COMP
KVALUES - STD ENTHALPY - STD DENSITY - STD
ELAPSED TIME - 00:00:00.06 - IN EQUIP NO + 3 +

WEGSTEIN CONVERGENCE ACCELERATION FOR STREAM 33

	CALCULATED WEGSTEIN Q	OLD ACCELERATED LBMOL/HR	CALCULATED LBMOL/HR	NEW ACCELERATED LBMOL/HR
1	1.000000	907.8488	907.8488	907.8488
STREAM 33 TOTAL			1054.657	1054.657

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CHEVRON STEAM COMPRESSOR

PAGE 12

ITER 3 RECYCLE LOOP 1

++++LOOP 1 HAS CONVERGED AFTER 3 ITERATIONS++++

NOW CALLING EQUIPMENT 4 + HEAEXC = E134B

KVALUES - STD ENTHALPY - STD DENSITY - STD

CHEVRON STEAM COMPRESSOR

PAGE 13

^AEXC 4 = E134

++++ HEAT EXCHANGER IN CONDENSING CURVE MODE
 NUMBER OF TEMPERATURE INCREMENTS ON STREAM 1 =20
 EQUIPMENT NUMBER = 4

++++ FOLLOWING CALCULATIONS BASED ON END POINT OF HEAT EXCHANGER

++++ TOTAL HEAT TRANSFERRED FROM STREAM 1, BTU/HR 0.18555E+08
 AREA/SHELL = 1384.8 FT2
 CORRECTED LOG MEAN TEMP. = 267.99 DEG F

MOL. WT. STREAM 1 = 18.02 MOL. WT. STREAM 2 = 18.02

++++ CALCULATIONS BELOW ARE FOR EXCHANGER DIVIDED INTO INCREMENTS

++++ CONDENSING CURVE LMTD IS FOR A 1,1,1 SHELL/TUBE ARRANGEMENT.

INC NO	STREAM 1		STREAM 2		ENTHALPY	AREA FT2	LMTD DEG F
	TEMP DEG F	VAPOR FRAC.	TEMP DEG F	VAPOR FRAC.	CHANGE BTU/HR		
0	300.0	0.0000	90.0	0.0000	0.00000E+00	0.0000E+00	0.0
1	302.3	0.0508	90.9	0.0000	0.92777E+06	88.06	210.7
2	302.5	0.1044	91.8	0.0000	0.92777E+06	87.92	211.0
3	302.6	0.1580	92.7	0.0000	0.92777E+06	88.24	210.3
4	302.8	0.2116	93.6	0.0000	0.92777E+06	88.56	209.5
5	302.9	0.2652	94.5	0.0000	0.92777E+06	88.89	208.7
6	303.0	0.3188	95.4	0.0000	0.92777E+06	89.22	208.0
7	303.2	0.3725	96.4	0.0000	0.92777E+06	89.54	207.2
8	303.3	0.4261	97.3	0.0000	0.92777E+06	89.88	206.5
9	303.5	0.4798	98.2	0.0000	0.92777E+06	90.21	205.7
10	303.6	0.5335	99.1	0.0000	0.92777E+06	90.55	204.9
11	303.8	0.5872	100.0	0.0000	0.92777E+06	90.89	204.2
12	303.9	0.6409	100.9	0.0000	0.92777E+06	91.23	203.4
13	304.0	0.6946	101.8	0.0000	0.92777E+06	91.57	202.6
14	304.2	0.7483	102.7	0.0000	0.92777E+06	91.92	201.9
15	304.3	0.8021	103.6	0.0000	0.92777E+06	92.27	201.1
16	304.5	0.8559	104.5	0.0000	0.92777E+06	92.62	200.3
17	304.6	0.9096	105.4	0.0000	0.92777E+06	92.98	199.6
18	304.7	0.9634	106.3	0.0000	0.92777E+06	93.34	198.8
19	337.9	1.0000	107.2	0.0000	0.92777E+06	86.65	214.1
20	440.8	1.0000	108.2	0.0000	0.92777E+06	66.61	278.6

++++ AREA/SHELL FOR 20 DIVISIONS = 1781.2 FT2
 TOTAL HEAT TRANSFERRED = 0.18555E+08 BTU/HR
 AVERAGE LMTD = 267.99 DEG F

-50 0 50 100 150 200
HEAT TRANSFERRED BTU/HR X 10 ** 5

CHEVRON STEAM COMPRESSOR

PAGE 15

ELAPSED TIME - 00:00:00.66 - IN EQUIP NO + 4 +

COMPRESSOR 3 TEMP LIFT (F) = 173.18

HEAT PUMP 1 INVESTMENT (DOLLAR) = 138101.38

HEAT PUMP 1 SAVING (DOLLAR/YR) = -96867.46

COMPRESSOR 3 INVESTMENT (DOLLAR) = 41261.26

COMPRESSOR 3 BHP = 589.67

HEAT PUMP 1 PAYOUT (YR) = -1.43

COMPRESSOR 3 POWER COST = 95469.09

HEX4 DUTY = -18555304.00

HEX4 AREA = 1384.77

ELAPSED TIME- 00:00:00.170 IN EQUIP NO 4

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A5A46792 LICENSEE: TENSA SERVICES HOUSTON ALR 10 386/20 10:31:54

CHEVRON STEAM COMPRESSOR

PAGE 16

SOLUTION REACHED

++++ SOLUTION HAS BEEN REACHED +++++

CHEVRON STEAM COMPRESSOR

PAGE 17

SOLUTION REACHED

STREAM SUMMARY

STREAM NUMBER	1	11	21	33
EQUIP CONXION	FEED-KNCK (0)-(2)	KNCK-PREH (2)-(20)	PREH-COMP (20)-(3)	COMP-PREH (3)-(20)
VAPOR FRACTION	1.0000	1.0000	1.0000	1.0000
TEMPERATURE F	268.00	268.00	275.00	448.18
PRESSURE PSIA	39.696	39.696	37.696	74.696
ENTHALPY BTU/HR	0.19559E+07	0.19559E+07	0.20209E+07	0.35216E+07
LB/FT3 T-P	0.93018E-01	0.93018E-01	0.87385E-01	0.14037
S.G.(60F) STP				
GAL/MIN STP				
MMSCF/DAY STP	9.6040	9.6040	9.6040	9.6040
MOLECULAR WT	18.015	18.015	18.015	18.015

FLOW RATES LBMOL/HR

WATER	1054.7	1054.7	1054.7	1054.7
-------	--------	--------	--------	--------

TOTAL LBMOL/HR	1054.7	1054.7	1054.7	1054.7
TOTAL LB/HR	19000.	19000.	19000.	19000.

STREAM NUMBER	34	35
EQUIP CONXION	PREH-E134 (20)-(4)	E134-PROD (4)-(0)
VAPOR FRACTION	1.0000	0.00000E+00
TEMPERATURE F	440.79	300.00
PRESSURE PSIA	72.696	69.696
ENTHALPY BTU/HR	0.34566E+07	-0.15099E+08
LB/FT3 T-P	0.13772	56.141
S.G.(60F) STP		1.0001
GAL/MIN STP		37.979
MMSCF/DAY STP	9.6040	
MOLECULAR WT	18.015	18.015

FLOW RATES LBMOL/HR

WATER	1054.7	1054.7
-------	--------	--------

TOTAL LBMOL/HR	1054.7	1054.7
TOTAL LB/HR	19000.	19000.

NING-----

THE FOLLOWING STREAM NUMBERS EXIST ON THE FLOW SHEET
 BUT DO NOT CONTAIN ANY MOLES-----

CHEVRON STEAM COMPRESSOR

PAGE 18

SOLUTION REACHED

+ EQUIPMENT SUMMARY - INDIVIDUAL DETAILS +

****FLASH SEPARATORS ****

EQUIPMENT NO. 2

EXTERNAL NAME KNCK
TYPE ISOTHERMAL
DUTY BTU/HR 0.00000E+00
SPECIFICATION

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CHEVRON STEAM COMPRESSOR

PAGE 19

SOLUTION REACHED

+ EQUIPMENT SUMMARY - INDIVIDUAL DETAILS +(CONTINUED)

****EXCHANGER/CONDENSERS****

EQUIPMENT NO.	4	20
EXTERNAL NAME	E134	PREH
U BTU/HR/FT2/F	50.00	80.00
AREA/SHELL FT2	1385.	4.700
NO. SHELLS	1.000	1.000
SHELL PASSES	1.000	1.000
TUBE PASSES	1.000	1.000
WATER GAL/HR	1.4850E+05	0.0000E+00
DELTA P-STR 1		
PSI	3.000	2.000
DELTA P-STR 2		
PSI	3.000	2.000
Q STR 1 BTU/HR	-1.8555E+07	6.5040E+04
MEAN T. DIFF F	268.0	173.0
CURVE SEG. F	20.00	0.0000E+00
UTI TEM IN DEG F	90.000	
UTI TEM OUT DEG F	105.00	
T OUT SPEC DEG F	300.00	275.00

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CHEVRON STEAM COMPRESSOR

PAGE 20

SOLUTION REACHED

+ EQUIPMENT SUMMARY - INDIVIDUAL DETAILS +(CONTINUED)

****PUMP/COMPRESSOR/EXPD****

EQUIPMENT NO.	3
EXTERNAL NAME	COMP
COMP. STAGES	1.000
WORK CAPACITY	
HP	1.00000E+06
OUTLET PRES.	
PSIA	74.696
DRIV TYPE	ELECTRIC
COMP TYPE	ENTROPY
THERMAL EFFIC.	0.7500
KW USAGE	439.4
REAL WORK HP	-589.7

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CHEVRON STEAM COMPRESSOR

PAGE 21

SOLUTION REACHED

M A T E R I A L B A L A N C E S U M M A R Y

COMPONENT NAME	FEED(S) LBMOL	PRODUCT(S) LBMOL	DIFFERENCE LBMOL	REL. ERROR
WATER	1055.	1055.	0.0000E+00	0.0000E+00
TOTAL FLOWRATE /HR	1055.	1055.	0.0000E+00	0.0000E+00

CHEVRON STEAM COMPRESSOR

PAGE 22

SOLUTION REACHED

STREAM NUMBER 1 CONNECTING (0) FEED
 TO (2) KNCK

STD K-VALUES STD ENTHALPY STD DENSITY FLOW RATES /HR

COMPONENT NAME	TOTAL LBMOL	LIQUID LBMOL	VAPOR LBMOL	TOTAL LB	TOTAL MOL PCT	KVALUE
WATER	1054.66	00.	1054.66	19000.0	100.00	
TOTAL	1054.66	00.	1054.66	19000.0	100.000	
TOTAL LB	19000.0	00.	19000.0			

STREAM NUMBER 1
 TEMPERATURE DEG F 268.00
 PRESSURE PSIA 39.696
 ENTHALPY BTU/HR 1.95585E+06
 VAPOR FRACTION 1.0000

PROPERTIES	TOTAL	VAPOR
MOLECULAR WT.	18.0153	18.0153
ENTHALPY BTU/LBMOLE	1854.4923	1854.4923
ENTROPY BTU/LBMO/R	1.1523	1.1523
CP BTU/LBMOL/R		8.3743
CV BTU/LBMOL/R		6.2804
SITY LB/FT3		0.093018
Z-FACTOR		0.984669
FLOWRATE FT3/SEC T-P		56.7396
MMSCFD STP		9.6040
VISCOSITY CP		0.010036
TH.COND BTU/FT/HR/F		0.020259

VISCOSITY CALCULATED BY METHOD OF ELY AND HANLEY (NBS81)
 TS IS STANDARD TEMPERATURE AT 60.00 DEG F

CHEVRON STEAM COMPRESSOR

PAGE 23

SOLUTION REACHED

REAM NUMBER 11 CONNECTING (2) KNCK
 TO (20) PREH

STD K-VALUES STD ENTHALPY STD DENSITY FLOW RATES /HR

COMPONENT NAME	TOTAL LBMOL	LIQUID LBMOL	VAPOR LBMOL	TOTAL LB	TOTAL MOL PCT	KVALUE
WATER	1054.66	00.	1054.66	19000.0	100.00	
TOTAL	1054.66	00.	1054.66	19000.0	100.000	
TOTAL LB	19000.0	00.	19000.0			

STREAM NUMBER 11
 TEMPERATURE DEG F 268.00
 PRESSURE PSIA 39.696
 ENTHALPY BTU/HR 1.95585E+06
 VAPOR FRACTION 1.0000

PROPERTIES	TOTAL	VAPOR
MOLECULAR WT.	18.0153	18.0153
ENTHALPY BTU/LBMOLE	1854.4923	1854.4923
ENTROPY BTU/LBMO/R	1.1523	1.1523
CP BTU/LBMOL/R		8.3743
CV BTU/LBMOL/R		6.2804
SITY LB/FT3		0.093018
CTOR		0.984669
FLOWRATE FT3/SEC T-P		56.7396
MMSCFD STP		9.6040
VISCOSITY CP		0.010036
TH.COND BTU/FT/HR/F		0.020259

VISCOSITY CALCULATED BY METHOD OF ELY AND HANLEY (NBS81)
 TS IS STANDARD TEMPERATURE AT 60.00 DEG F

CHEVRON STEAM COMPRESSOR

PAGE 24

SOLUTION REACHED

STREAM NUMBER 21 CONNECTING (20) PREH
 TO (3) COMP

STD K-VALUES STD ENTHALPY STD DENSITY FLOW RATES /HR

COMPONENT NAME	TOTAL LBMOL	LIQUID LBMOL	VAPOR LBMOL	TOTAL LB	TOTAL MOL PCT	KVALUE
WATER	1054.66	00.	1054.66	19000.0	100.00	
TOTAL	1054.66	00.	1054.66	19000.0	100.000	
TOTAL LB	19000.0	00.	19000.0			

STREAM NUMBER 21
 TEMPERATURE DEG F 275.00
 PRESSURE PSIA 37.696
 ENTHALPY BTU/HR 2.02089E+06
 VAPOR FRACTION 1.0000

PROPERTIES	TOTAL	VAPOR
MOLECULAR WT.	18.0153	18.0153
ENTHALPY BTU/LBMOLE	1916.1619	1916.1619
ENTROPY BTU/LBMO/R	1.3378	1.3378
CP BTU/LBMOL/R		8.3721
CV BTU/LBMOL/R		6.2864
SITY LB/FT3		0.087385
Z-FACTOR		0.985848
FLOWRATE FT3/SEC T-P		60.3970
MMSCFD STP		9.6040
VISCOSITY CP		0.010140
TH.COND BTU/FT/HR/F		0.020452

VISCOSITY CALCULATED BY METHOD OF ELY AND HANLEY (NBS81)
 TS IS STANDARD TEMPERATURE AT 60.00 DEG F

CHEVRON STEAM COMPRESSOR

PAGE 25

SOLUTION REACHED

REAM NUMBER 33 CONNECTING (3) COMP
 TO (20) PREH

STD K-VALUES STD ENTHALPY STD DENSITY FLOW RATES /HR

COMPONENT NAME	TOTAL LBMOL	LIQUID LBMOL	VAPOR LBMOL	TOTAL LB	TOTAL MOL PCT	KVALUE
WATER	1054.66	00.	1054.66	19000.0	100.00	
TOTAL	1054.66	00.	1054.66	19000.0	100.000	
TOTAL LB	19000.0	00.	19000.0			

STREAM NUMBER 33
 TEMPERATURE DEG F 448.18
 PRESSURE PSIA 74.696
 ENTHALPY BTU/HR 3.52163E+06
 VAPOR FRACTION 1.0000

PROPERTIES	TOTAL	VAPOR
MOLECULAR WT.	18.0153	18.0153
ENTHALPY BTU/LBMOLE	3339.1279	3339.1282
ENTROPY BTU/LBMO/R	1.7388	1.7388
CP BTU/LBMOL/R		8.6069
CV BTU/LBMOL/R		6.5041
SITY LB/FT3		0.140369
Z-FACTOR		0.984134
FLOWRATE FT3/SEC T-P		37.5992
MMSCFD STP		9.6040
VISCOSITY CP		0.012914
TH.COND BTU/FT/HR/F		0.026531

VISCOSITY CALCULATED BY METHOD OF ELY AND HANLEY (NBS81)
 TS IS STANDARD TEMPERATURE AT 60.00 DEG F

CHEVRON STEAM COMPRESSOR

PAGE 26

SOLUTION REACHED

STREAM NUMBER 34 CONNECTING (20) PREH
 TO (4) E134B
 STD K-VALUES STD ENTHALPY STD DENSITY FLOW RATES /HR

COMPONENT NAME	TOTAL LBMOL	LIQUID LBMOL	VAPOR LBMOL	TOTAL LB	TOTAL MOL PCT	KVALUE
WATER	1054.66	00.	1054.66	19000.0	100.00	
TOTAL	1054.66	00.	1054.66	19000.0	100.000	
TOTAL LB	19000.0	00.	19000.0			

STREAM NUMBER 34
 TEMPERATURE DEG F 440.79
 PRESSURE PSIA 72.696
 ENTHALPY BTU/HR 3.45659E+06
 VAPOR FRACTION 1.0000

PROPERTIES	TOTAL	VAPOR
MOLECULAR WT.	18.0153	18.0153
ENTHALPY BTU/LBMOLE	3277.4583	3277.7256
ENTROPY BTU/LBMO/R	1.7239	1.7239
CP BTU/LBMOL/R		8.5964
CV BTU/LBMOL/R		6.4944
SITY LB/FT3		0.137722
Z-FACTOR		0.984211
FLOWRATE FT3/SEC T-P		38.3220
MMSCFD STP		9.6040
VISCOSITY CP		0.012798
TH.COND BTU/FT/HR/F		0.026267

VISCOSITY CALCULATED BY METHOD OF ELY AND HANLEY (NBS81)
 TS IS STANDARD TEMPERATURE AT 60.00 DEG F

CHEVRON STEAM COMPRESSOR

PAGE 27

SOLUTION REACHED

REAM NUMBER 35 CONNECTING (4) E134B
 TO (0) PRODUCT

STD K-VALUES STD ENTHALPY STD DENSITY FLOW RATES /HR

COMPONENT NAME	TOTAL LBMOL	LIQUID LBMOL	VAPOR LBMOL	TOTAL LB	TOTAL MOL PCT	KVALUE
WATER	1054.66	1054.66	00.	19000.0	100.00	
TOTAL	1054.66	1054.66	00.	19000.0	100.000	
TOTAL LB	19000.0	19000.0	00.			

STREAM NUMBER 35
 TEMPERATURE DEG F 300.00
 PRESSURE PSIA 69.696
 ENTHALPY BTU/HR -1.50987E+07
 VAPOR FRACTION 0.00000E+00

PROPERTIES	TOTAL	LIQUID
MOLECULAR WT.	18.0153	18.0153
ENTHALPY BTU/LBMOLE	-14316.2344	-14316.2344
ENTROPY BTU/LBMO/R	-21.1645	-21.1645

WARNING +++ CV MAY BE IN ERROR

BTU/LBMOL/R	20.6483
BTU/LBMOL/R	20.9783
DENSITY LB/FT3	56.1408
Z-FACTOR	0.2744E-02
FLOWRATE GAL/MIN T-P	42.1972
S.G.(TS/60) GPA STP	1.0001
VISCOSITY CP	0.374799
TH.COND BTU/FT/HR/F	0.562603
SURFACE TENSION DYNE/CM	48.9037

VISCOSITY CALCULATED BY METHOD OF ELY AND HANLEY (NBS81)

TS IS STANDARD TEMPERATURE AT 60.00 DEG F

CHEVRON STEAM COMPRESSOR

PAGE 28

SOLUTION REACHED

***** LINE SIZING RESULTS FOR GAS STREAM 1 *****

LINE IS ISOTHERMAL AND HORIZONTAL

METHODS USED : FRICTION DELTA P FRICTION FACTOR ELEVATION DELTA P
 DARCY MOODY PHASE DENSITY

VAPOR FLOW RATE	LB/HR	0.1900E+05		
TEMPERATURE	DEG F	268.0		
INLET PRESSURE	PSIA	39.70		
INTERNAL DIAMETER	IN.	10.02	DEFAULT	
MAX DELTA P PER 100 FT	PSI	0.5000	SPECIFIED	
LINE LENGTH	FT	20.00	SPECIFIED	
ELEVATION	FT	0.0000E+00	DEFAULT	
PIPE WALL CODE	STD		DEFAULT	
PIPE ROUGHNESS	FT	0.1500E-03	DEFAULT	
DENSITY (VAPOR)	LB/FT3	0.9301E-01		
MOLECULAR WEIGHT (VAPOR)		18.02		
VISCOSITY (VAPOR)	CP	0.1004E-01		
Z-FACTOR(AT INLET PRESSURE)		0.9847		
SONIC VELOCITY		1609.	FT/SEC	

		LARGER	TARGET	SMALLER
NOMINAL DIAMETER	IN.	12.00	10.00	8.000
INTERNAL DIAMETER	IN.	12.00	10.02	7.981
DELTA P (FRICTION)	PSI	0.1479E-01	0.3696E-01	0.1183
TA P (ELEVATION)	PSI	0.0000E+00	0.0000E+00	0.0000E+00
DELTA P (TOTAL)	PSI	0.1479E-01	0.3696E-01	0.1183
PRESSURE DROP/100FT	PSI	0.7395E-01	0.1848	0.5914
MEETS DELTA P SPEC	YES	YES	NO	
OUTLET PRESSURE	PSIA	39.68	39.66	39.58
FLUID VELOCITY	FT/SEC	72.25	103.6	163.3
TRANSITION REYNOLDS NUMBER		0.6123E+06	0.5022E+06	0.3908E+06
REYNOLDS NUMBER		0.9964E+06	0.1193E+07	0.1498E+07
FLOW REGIME		TURBULENT	TURBULENT	TURBULENT
FRICTION FACTOR		0.1411E-01	0.1431E-01	0.1468E-01

THESE RESULTS ARE SUBJECT TO THE LIMITATIONS AND ACCURACY OF THE
 METHODS USED. LITERATURE REFERENCES ARE GIVEN IN THE USER MANUAL.

STEAM COMPRESSOR

PAGE 30

SOLUTION REACHED

***** LINE SIZING RESULTS FOR GAS STREAM 34 *****

LINE IS ISOTHERMAL AND HORIZONTAL

METHODS USED : FRICTION DELTA P FRICTION FACTOR ELEVATION DELTA P
 DARCY MOODY PHASE DENSITY

VAPOR FLOW RATE	LB/HR	0.1900E+05		
TEMPERATURE	DEG F	440.8		
INLET PRESSURE	PSIA	72.70		
INTERNAL DIAMETER	IN.	7.981	DEFAULT	
MAX DELTA P PER 100 FT	PSI	0.5000	SPECIFIED	
LINE LENGTH	FT	400.0	SPECIFIED	
ELEVATION	FT	0.0000E+00	DEFAULT	
PIPE WALL CODE		STD	DEFAULT	
PIPE ROUGHNESS	FT	0.1500E-03	DEFAULT	
DENSITY (VAPOR)	LB/FT3	0.1377		
MOLECULAR WEIGHT (VAPOR)		18.02		
VISCOSITY (VAPOR)	CP	0.1280E-01		
Z-FACTOR(AT INLET PRESSURE)		0.9842		
SONIC VELOCITY		1783.	FT/SEC	
* * * * *				
		LARGER	TARGET	SMALLER
NOMINAL DIAMETER	IN.	10.00	8.000	6.000
INTERNAL DIAMETER	IN.	10.02	7.981	6.065
DELTA P (FRICTION)	PSI	0.5064	1.614	6.608
TA P (ELEVATION)	PSI	0.0000E+00	0.0000E+00	0.0000E+00
DELTA P (TOTAL)	PSI	0.5064	1.614	6.608
PRESSURE DROP/100FT	PSI	0.1266	0.4035	1.652
MEETS DELTA P SPEC		YES	YES	NO
OUTLET PRESSURE	PSIA	72.19	71.08	66.09
FLUID VELOCITY	FT/SEC	69.99	110.3	191.0
TRANSITION REYNOLDS NUMBER		0.5022E+06	0.3908E+06	0.2886E+06
REYNOLDS NUMBER		0.9358E+06	0.1175E+07	0.1546E+07
FLOW REGIME		TURBULENT	TURBULENT	TURBULENT
FRICTION FACTOR		0.1452E-01	0.1483E-01	0.1539E-01

THESE RESULTS ARE SUBJECT TO THE LIMITATIONS AND ACCURACY OF THE
 METHODS USED. LITERATURE REFERENCES ARE GIVEN IN THE USER MANUAL.

CHEVRON STEAM COMPRESSOR

PAGE 31

SOLUTION REACHED

*** SEPARATOR SIZING RESULTS FOR VERTICAL SEPARATOR 2 *****

NOTE: NO LIQUID PRESENT AT THESE CONDITIONS.

LIQUID DENSITY USED IS AT DEW POINT 266.4 F

VAPOR STREAM = 11 LIQUID STREAM = 12

CONDITIONS

LIQUID	FLOW RATE	T-P	0.00000E+00	FT3/MIN
VAPOR	FLOW RATE	T-P	3404.4	FT3/MIN
TOTAL	FLOW RATE	T-P	3404.4	FT3/MIN
TEMPERATURE			268.00	DEG F
PRESSURE			39.696	PSIA

PROPERTIES

DENSITY	LIQUID	57.382	LB/FT3
DENSITY	VAPOR	0.93018E-01	LB/FT3
Z-FACTOR	VAPOR	0.98467	

DESIGN PARAMETERS

K CONSTANT	SPEC	0.35000	FT/SEC
SURGE TIME	DEFAULT	2.0000	MIN
DISENGAGING HEIGHT	DEFAULT	3.0000	FT
MIN LIQ TO INLET HEIGHT	DEFAULT	2.0000	FT
MIN LEVEL NOZZLE HEIGHT	DEFAULT	1.5000	FT
MIST ELIMINATOR	DEFAULT	0.50000	FT
DESIGN PRESSURE	DEFAULT	43.666	PSIA
ALLOWABLE STRESS	DEFAULT	15015.	PSIA
JOINT EFFICIENCY	DEFAULT	1.0000	
CORROSION ALLOWANCE (CA)	DEFAULT	0.10417E-01	FT
WEIGHT PERCENT ALLOWANCE(WA)	DEFAULT	20.000	

RESULTS

MAX. DROPLET SETTLING VELOCITY	8.6860	FT/SEC
ACTUAL VAPOR VELOCITY	8.3724	FT/SEC
DIAMETER BASED ON SEPARATION	2.8840	FT
SELECTED INTERNAL DIAMETER - PLATE STEEL	2.9375	FT
LENGTH	8.0000	FT
HIGH LIQUID LEVEL (HLL) HEIGHT	1.5000	FT
HLL TO INLET HEIGHT	3.0000	FT
DISENGAGING HEIGHT	3.0000	FT
ACTUAL PLATE THICKNESS	0.15625E-01	FT
ACTUAL HEAD THICKNESS	0.15625E-01	FT
MIN SHELL THICKNESS INCLUDING CA	0.13254E-01	FT
MIN HEAD THICKNESS INCLUDING CA	0.13251E-01	FT
APPROX WEIGHT OF STEEL VESSEL(EMPTY)	619.45	LB
APPROX WEIGHT OF VESSEL (FULL)	3730.6	LB
APPROX TOTAL WEIGHT INCLUDING WA	3854.4	LB

CHEVRON STEAM COMPRESSOR

PAGE 33

SOLUTION REACHED

* SKETCH OF VERTICAL FLASH DRUM 2 (TO APPROX. SCALE ONLY) ****

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PAGE	INDEX OF FLOWSHEET CALCULATIONS
8	FLA 2 (KNCK)
9	++++ BEGIN ITERATION 1 OF RECYCLE LOOP 1 ++++
9	HEAEXC 20 (PREH)
9	COMPRES 3 (COMP)
10	++++ BEGIN ITERATION 2 OF RECYCLE LOOP 1 ++++
10	HEAEXC 20 (PREH)
10	COMPRES 3 (COMP)
11	++++ BEGIN ITERATION 3 OF RECYCLE LOOP 1 ++++
11	HEAEXC 20 (PREH)
11	COMPRES 3 (COMP)
12	+++LOOP 1 HAS CONVERGED AFTER 3 ITERATIONS +++
12	HEAEXC 4 (E134)
16	SOLUTION REACHED
17	STREAM SUMMARY
19	EQUIPMENT SUMMARY - HEAT EXCHANGERS
20	EQUIPMENT SUMMARY - PUMPS
21	MATERIAL BALANCE SUMMARY
22	DETAILED STREAM PRINT
28	LINE SIZING RESULTS FOR LINE 1
29	LINE SIZING RESULTS FOR LINE 11
30	LINE SIZING RESULTS FOR LINE 34
31	SEPARATOR SIZING RESULTS FOR FLA 2
	ACCOUNTING INFORMATION - SEE BELOW

CHEVRON STEAM COMPRESSOR

PAGE 35

ACCOUNTING INFORMATION

COMPUTER TYPE IS 80386 MS-DOS

2476 15SEP92 10:31:54 WV86 1 DESI

56 17 0 B6C8 1 1000000000 0

CHEVRON STEAM COMPRE

56 CHEMSHARE CALCULATION UNITS (CCU) USED IN THIS RUN.

*** END OF SIMULATION ***

Appendix C
VENDOR QUOTES

COOPER

Gardner-Denver® Compressors

Your Air Of Confidence™

Available In 5 To 600 HP, Up To 2,595 CFM And Pressures To 10,000 PSIG

TELEFAX MESSAGE

Houston Sales Office
Gardner-Denver Industrial Machinery Division
3325 South Rice Avenue
Houston, Texas 77056
Phone: (713) 626-2220 • FAX: (713) 963-0510

DATE: 9-23-92

TO: Tensa Services

ATTN: Peter Chao

SUBJECT: Port Arthur Project

NO. OF PAGES: 1
(Including cover sheet)

AJAX® PUMPS • DUROFLOW® • GARDNER-DENVER® •
JOY® COMPRESSORS • OPI® • SUTORBIT®

FROM: J. L. Smith

Mr Chao -

We regret to inform you we have
nothing large enough to fit your steam
booster application on the subject project.

We would appreciate an opportunity to
quote any air compressors on this project
Thank you

J. L. Smith

3701 Kirby Drive, Suite 888
Houston, TX 77098
TEL: 713/522-1081
FAX: 713/522-9652

A **DOVER** DIVERSIFIED COMPANY

September 14, 1992

TENSA Services, Inc.
6200 Savoy Drive Suite 540
Houston, Texas 77036-3323

Attention: Mr. Peter Chao

Subject: Chemical Plant, Port Arthur, TX
Cyclohexane Intercooler Compressor
Contract No. DE-FC-12860-1
A-CCC Proposal No. S 1806

SEP 15 1992

Gentlemen:

A-CCC is pleased to propose the following oil free rotary screw compressor for your application.

To meet your required flow of 3623 ACFM, we offer our Model 105L2 rotary screw single stage gas compressor. Given below is the performance of this machine per your process conditions.

	<u>CASE 1</u>	<u>CASE 2</u>
ACFM	3.623	3.623
P1, PSIA	39.7	44.7
T1, Deg. F	275	275
MW	18.02	18.02
P2, PSIA	75	75
RPM	9374	9374
BHP	673	641
T2, Deg. F	438	411

Thank you for considering A-CCC for this application.

Please call me if you have any questions.

Very truly yours,

A-C COMPRESSOR CORPORATION

Rick Stegemeier

RS:jt
attachments

SCOPE OF SUPPLY

A. Steam Compressor

To meet your flow requirements we offer one (1) Steam Compressor General Arrangement with the General Arrangement consisting of the following:

<u>Item</u>	<u>Qty</u>	<u>Description</u>
.1	1	Model 105L2 Rotary Gas Compressor composed of the following: <ul style="list-style-type: none">- Cast steel casings.- Carbon steel rotors.- Sleeve type journal bearings and tilting pad thrust bearings active side(tapered land inactive side).- Carbon ring shaft seals.
.2	1	Speed increasing gear.
.3	1	800 HP 1.0 S.F. electric motor.
.4	1	High speed, nonlubed, flexible disc type coupling and manufacturer's standard fiber glass non-sparking guard, coupling gear to compressor.
.5	1	Low speed, nonlubed, flexible disc type coupling and manufacturer's standard fiber glass non-sparking guard, coupling motor to gear.
.6	1-lot	Local mounted, manifolded, carbon steel lube oil supply and carbon steel return piping.
.7	1-Lot	Local mounted carbon steel jacket cooling water supply and return piping.
.8	1-Lot	Local mounted, manifolded, carbon steel seal air piping.
.9	1	Gauge board consisting of instrumentation providing temperature and pressure readouts and alarm and shutdown switches. Flow and level indicators and other instrumentation having no customer interconnections are <u>not</u> gauge board mounted.
.10	1	Structural steel subbase under Items A.1-A.9 above.

Note: Items A.1-A.10 are factory assembled and piped to form a complete "Compressor General Arrangement."

The following support systems and software are in addition to the General Arrangement above and are provided in the quantities shown:

Item	Qty	Description
.11	1	Lube oil console with <ul style="list-style-type: none"> - carbon steel reservoir. - carbon steel piping. - single oil cooler. - twin oil filters. - dual electric motor driven oil pumps.
.12	1	Inlet silencer, externally lagged and of carbon steel construction.
.13	1	Discharge silencer. Construction same as inlet above. Note: Silencers are vertical and field mounted by Purchaser.
.14	1-Lot	Software furnished in the following quantities: <ul style="list-style-type: none"> 3 Instruction manuals. 1 Torsional and Lateral Vibration Analysis.
.15	1-Lot	Manufacturer's standard tests.
<u>.01</u>	1	<u>Optional adder</u> -Acoustic enclosure over compressor. Field erected by Purchaser. Estimated noise level with housing is 85 dBA. Without housing approximately 100 dBA. Enclosure comes complete with ceiling exhaust fan vented to atmosphere and lower side-wall air intake louvers.

NOTE: All valves are of carbon steel construction.
All flanges are of the slip on type.

The following are specifically not included in the scope of this Proposal:

- Process piping, supports, hangers, aftercoolers, separators; check, shut-off, and relief valves.
- Interconnecting piping, tubing, wiring, etc., between boundary limits of General Arrangement and Lube/seal Oil Console.
- Interconnections between Purchaser's utilities(steam, water, air, electrical) and the equipment offered.
- Freight. Shipping terms are FCA. Point of Manufacture.

ESTIMATED PRICING AND DELIVERY

The following estimated prices apply to the equipment as listed in the Scope of Supply above.

I. Basic Equipment

<u>Quantity</u>	<u>Description</u>	<u>ESTIMATED Price</u>
(1-lot)	Compressor General Arrangement, per Items A.1-A.2, A.4-A.15	\$ 375,000 ± 20%
1	TEFC main driver, Item A.3	\$ 90,000 ± 20%
1	X-proof main driver, Item A.3	\$ 105,000 ± 20%

Optional Equipment or Services:

1	Acoustic enclosure, Item O.1	\$ 19,387 ± 20%
---	------------------------------	-----------------

Delivery:

Estimated delivery 48-52 weeks ARO

PRICES SUBJECT TO CHANGE UPON RECEIPT OF FINAL SPECIFICATIONS.

Put the screws to your compressor applications

The often-overlooked helical-screw compressor can offer CPI plants a number of benefits. Here is a checklist for selecting and installing one.

Charles Jackson*
Turbomachinery Consultant

The helical-screw compressor is rarely considered for compressor applications in chemical-process-industries (CPI) plants, which is surprising since it costs little to maintain, and operates at high efficiency.

The screw compressor is somewhat of a hybrid — it produces positive displacement, like a piston-type compressor, but it runs with a rotary motion, like a centrifugal compressor. High compression ratios are possible, yet low maintenance and steady, low-pulsation deliveries are achievable with a properly applied machine. And the device offers stable performance in applications where polymerization, or liquid or solid carryover into the gas, may occur.

*Mr Jackson wrote this article as a consultant for A-C Compressor Corp., West Allis, Wis.

Screw compressors have been used successfully in a variety of services, such as dirty acetylene, H_2/CO_2 mixtures, air, hexane, etc. Some services have had flushing facilities; some, in hydrogen service, have had seals with water injection combined with buffer seals; some limited heat of compression to 300–350°F; and some were in vacuum service.

Selection and installation

For detailed specifications for helical-screw compressors, refer to "Rotary-type positive-displacement compressors for general refinery services," API Standard 619, 2nd ed., May 1985. The following checklist provides some general guidelines on the selection and installation of such units:

1. Transient and steady-state torsional analysis should be performed before the shaft-ends, gear size and mass, coupling stiffness and mass, etc., are determined. This will allow

Figure — Helical-screw compressors are well-suited for applications in which (a) the gas mass flowrate varies but a fixed discharge pressure is needed, (b) the molecular weight of the gas might vary, or (c) minor contaminants must be dealt with

PLEASE DELIVER THE FOLLOWING PAGES TO:

NAME: Mr. Raymond Adam

COMPANY/LOCATION: Tensa Services Inc.

FAX NO.: 789-0907

RE: Steam Compression ACCI Prop CR8-441492
Kerr McKee

THIS TRANSMITTAL IS BEING SENT FROM:

NAME: John O. Higgins

FAX NO: (713) 359-3349

COMPANY/LOCATION: ATLAS COPCO HOUSTON/KINGWOOD

DATE: June 8, 1992

NUMBER OF PAGES INCLUDING COVER SHEET: 3

Raymond,

CASE I: We can do with a titanium impeller, also included on performance sheet is max P_2 run with a standard 17-4 PH stainless steel impeller.

CASE II: The mass flow is too small for us to make a P_2 of 128.1 psia. The max P_2 we can get with a TP 6/6 two stage is 105 psia. We would need more lb/HR and a third stage to reach 128.1 psia pressure.

Pricing for CASE I Model SC-6 is \$ 202,000.00 net with standard impeller. Titanium impeller adder \$ 27,500.00 driver price 250 HP motor 460 volt/3 ph/60 HZ TEFC NEMA frame motor is \$ 8,000.00 net. Shipment total package 40 wks ARO.

If you have any question, please call me.

Best Regards,

John O. Higgins
John O. Higgins *cmc*
Regional Manager
JOH/cmc

Address
One Kingwood Place
Suite 214
Kingwood, TX 77339

Telephone
(713) 359-6645
Telefax
(713) 359-3349

ATLAS COPCO

ONE-STAGE COMPRESSOR

CUSTOMER.: TENSA

MODEL: SC-6

DATE.....: 06-05-1992

ESTIMATED PERFORMANCE

REFERENCE: CR8-441492

DESIGN CONDITIONS

PBAR = 14.7

PH = 1

PI = 84.7

TI = 283

INLET FLOW = 1207

case I

Compressor Model

SC-6

Process Requirements and Expected Performance

Item:	SC-6
Operating Condition:	Design (Case I)
Gas Handled:	Steam
Barometer (PSIA):	14.7

Inlet Conditions

	17-4 PH	Titanium
Pressure (PSIA)	34.7	34.7
Temperature (°F)	265	263
Relative Humidity (%)	1.0	1.0
Molecular Weight MW	18.01	18.01
Cp/Cv K1	1.3	1.3
Compressibility Z1	0.975	0.975
Inlet Volume (ACFM)	1207	1207
Weight Flow lb/hr	6000	6000

Discharge Conditions

Pressure (PSIA)	66	72.2
Temperature (°F)	411	434
Pressure Ratio r	1.90	2.08
Brake Horsepower (BHP)	205	236
Input Speed (RPM)	3600	3600
Compressor Speed* RPM	52487	52487
Number of Stages	1	1
Impeller Dia.* (IN)	6.52	7.0

* Subject to change based on final engineering.

INGERSOLL-RAND
AIR COMPRESSORS

Houston Office

Ingersoll Rand Company
9301 S.W. Freeway #270
Houston, Texas 77074
P.O. Box 740468
Houston, Texas 77274-0468
(713) 995-3000
FAX: (713) 779-2107

September 9, 1992

Tensa Services, Inc.
6200 Savoy Drive, Ste. 540
Houston, Texas 77036

Attn: Mr. P. Chao

Re: Inquiry for Steam Booster
Port Arthur Chemical Plant
DE-FC-12860-1

Dear Mr. Chao:

We appreciate receiving your above referenced inquiry for a steam booster, but regret that at this time Ingersoll-Rand has nothing to offer in this inlet temperature and pressure range.

Please keep us in mind for any additional compressor requirements.

Very truly yours,
INGERSOLL-RAND COMPANY

J.S. Schillaci
Account Manager
Southwest Region Sales and Service

QUOTATION

P.O. BOX 6007
1205 West Main
Baytown, Texas 77520

(713) 427-6625

(713) 471-5713

FAX No: (713) 428-7524

Inquiry No. DE-FC-12860-1

Date 9-17-92

Net 30 DAYS
Terms Pending Credit Approval

Prices quoted are
F.O.B. DELIVERED

Delivery STOCK

TO TENSA SERVICES

Attn: Peter Chao

Fax: 789-0907

DEAR PETER:

THANKS FOR THE OPPORTUNITY TO QUOTE YOUR MATERIAL REQUIREMENTS. WE ARE PLEASED TO OFFER THE FOLLOWING PRICES FOR YOUR CONSIDERATION. WE ARE UNABLE TO UNDERSTAND EXACTLY WHAT YOU REQUIRED FROM YOUR SPECIFICATION SHEET.

12 TERMS FOR THIS INQUIRY ARE NET 30 DAYS, PENDING APPROVAL OF YOUR CREDIT. ALL PIPE IS QUOTED IN DOUBLE RANDOM LENGTHS. THIS BID IS VALID FOR 30 DAYS. PLEASE CHECK YOUR SPECIFICATIONS AGAINST WHAT WE ARE QUOTING.

10" 3/4	A53-B	SMLS. Pipe	17.27	seamless
10" 3/4	A53-B	ERW Pipe	14.35	welded.
8" 3/4	A53-B	SMLS. Pipe	11.87	
8" 3/4	A53-B	ERW Pipe	9.44	

PRICING IS PER LINEAL FOOT. ALL PIPE IS QUOTED AS DOMESTIC. IF YOU HAVE ANY QUESTIONS, PLEASE CONTACT MYSELF OR STEPHEN THOMAS.

THANKS,

By Gary Arner Jr.
GARY ARNER JR

Kennedy Tank and
Manufacturing Co., Inc.

TEL: 787-1311 FAX (317) 783-7772 1-800-445-1344
119 East Sumner Ave. P.O. Box 47070 Indianapolis, Indiana 46247-0070

QUOTATION

DATE: 10-14-92

PAGE OF

PAGES: 1 OF 1

KINETIC ENGINEERING CORPORATION
1055 SILBER ROAD, SUITE 101
HOUSTON, TX 77055

KENNEDY
QUOTATION: 92-2788

REFERENCE: TENSA-PETER CHAO

ATTN: PAT BERNHAGEN

PHONE NUMBER: 713-686-2200

FAX NUMBER: 713-686-1450

entlemen:

We appreciate this opportunity to submit for your approval and acceptance, this quotation on the following products:

QUANTITY	DESCRIPTION	PRICE EACH	EXTENDED PRICE
	<u>BUDGET ESTIMATE</u>		
1	VERTICAL PRESSURE VESSEL, 3'-0" I.D. X 8'-0" TANGENT TO TANGENT STRAIGHT SHELL, MADE OF CARBON STEEL WITH 1/16" CORROSION ALLOWANCE, DESIGNED FOR 75 PSIG @ 300° F. AND ASME STAMPED. INCLUDES 2-6" RF FLANGED OPENINGS, 2-2" RF FLANGED OPENINGS, 2-1 1/2" RF FLANGED OPENINGS, 1-12X16 ELLIPTICAL MANWAY, 316 MESH PAD 6" THICK, 4 SUPPORT LEGS, SANDBLASTED AND PRIMED ONE COAT RED OXIDE PRIMER. ESTIMATED SHIPPING WEIGHT: 1186#	6,133.	6,133.
	VERTICAL PRESSURE VESSEL, 7'-6" I.D. X 14'-0" TANGENT TO TANGENT STRAIGHT SHELL, MADE OF CARBON STEEL WITH 1/16" CORROSION ALLOWANCE, DESIGNED FOR 75 PSIG @ 350° F. AND ASME STAMPED. INCLUDES 4-6" RF FLANGED OPENINGS, 2-2" RF FLANGED OPENINGS, 2-1 1/2" RF FLANGED OPENINGS, AND 1-20" MANWAY. ALSO INCLUDES SUPPORT SKIRT, SANDBLASTED AND PRIMED ONE COAT RED OXIDE PRIMER ALL AS PER FAXED REQUEST DATED 10-8-92. ESTIMATED SHIPPING WEIGHT: 9665#	21,261.	21,261.
	NOTE: LEGS FOR ITEM #1 AND SKIRT FOR ITEM #2 TO SUPPORT BOTTOM OF TANK @ 24" ABOVE GRADE.		

JOINT

OUR PLANT

LEAD TIME
PAYMENT 9 WEEKS AFTER RECEIPT OF APPROVED DRAWINGS

TERMS: NET 30 DAYS

3

May we look forward to serving you?

Kennedy Tank and Manufacturing Co., Inc.

J. D. CLARK
VICE-PRESIDENT, SALES