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Survey of Industrial Coal Conversion Equipment Capabilities: Heat Recovery and Utilization

W. R. Gambill

W. R. Reed

MASTER

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Chemical Technology Division

SURVEY OF INDUSTRIAL COAL CONVERSION EQUIPMENT CAPABILITIES:
HEAT RECOVERY AND UTILIZATION

W. R. Gambill and W. R. Reed*

*
Engineering Division

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FOREWORD AND ACKNOWLEDGMENTS

This is one of four reports describing surveys of industrial coal conversion equipment capabilities which were conducted for and supported by the Equipment Branch (James Powell, Branch Chief, and T. K. Lau, Project Manager) of the Fossil Energy Major Facilities Project Management Division (MFPM)* of the Energy Research and Development Administration (ERDA).*

This report describes the results of the survey relating to equipment for heat recovery and utilization; other reports in this series include the following:

W. A. Bush and E. C. Slade, Survey of Industrial Coal Conversion Equipment Capabilities: Valves, ORNL/TM-6071, Oak Ridge National Laboratory (1978).

J. P. Meyer and M. S. Edwards, Survey of Industrial Coal Conversion Equipment Capabilities: High-Temperature, High-Pressure Gas Purification, ORNL/TM-6072, Oak Ridge National Laboratory (1978).

W. R. Williams, J. R. Horton, W. F. Boudreau, and M. Siman-Tov, Survey of Industrial Coal Conversion Equipment Capabilities: Rotating Components, ORNL/TM-6074, Oak Ridge National Laboratory (1978).

The authors of this report appreciate the personnel in the process heat-exchange equipment industry who have given their time and effort to reply to our questionnaire and to discuss the availability and problems of equipment required by the coal conversion industry. Without their cooperation, this report would not have been possible. We also express our appreciation to those users of heat recovery equipment who shared their views with us, and to those in the Union Carbide Nuclear Division who have provided assistance during this study.

*Effective October 1, 1977, ERDA became part of the Department of Energy (DOE); MFPM has since become part of the Division of Coal Conversion of DOE.

SURVEY OF INDUSTRIAL COAL CONVERSION EQUIPMENT CAPABILITIES:

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ABSTRACT

A scoping survey of the capabilities of industrial heat recovery equipment was conducted to determine their adaptability to proposed coal conversion complexes. Major categories of heat exchangers included shell-and-tube, periodic-flow and rotary regenerators, heat pipe arrays, direct phase contactors, and steam and organic Rankine cycles for power generation from waste heat. Primary applications encompassed feed-effluent and other process stream interchangers, combustion air preheaters, and heat recovery steam generators (waste heat boiler-superheaters).

It is concluded that the single area providing the greatest potential for extending U.S. industrial heat-recovery equipment capabilities as related to coal conversion processes is a research, development, and testing program to acquire more physical property and heat transfer data and more reliable design correlations.

1. INTRODUCTION

Because the specific investment costs (dollars per 10^6 Btu of input coal) for converting coal or lignite to electricity, synthesis gas (syngas), substitute natural gas (SNG), methanol, and liquid fuels are about the same, the product costs will depend largely on the process thermal efficiencies. One factor obviously affecting the subject efficiencies is the degree of heat recovery and utilization reliably and economically achievable in such

plants. It is primarily for this reason that a scoping survey of industrial equipment capability relevant to coal conversion processes was considered desirable.

The survey was conducted by reviewing the general literature and the current catalogs and brochures of numerous companies specializing in the design and fabrication of large and/or severe-service heat exchange equipment. After discussions with many individuals conversant with the present state of the process heat transfer field, a questionnaire was sent to 23 companies requesting their views on the design and fabrication of 12 potentially critical heat recovery units selected from three recent evaluation design studies of coal conversion plants. The results of the initial survey are described in this report.

Steps involved in managing thermal energy can include such diverse operations as extraction, transport, storage, recovery, utilization, and ultimate rejection. These operations are characterized by different fluid-temperature levels and thermodynamic availabilities. This report addresses only recovery and utilization; applications such as heat extraction from methanation catalyst beds, transport within a plant by an intermediate fluid, storage for load leveling, and rejection to the atmosphere or surface waters are mentioned, if at all, only in passing.

The problems of designing and fabricating reliable heat exchange equipment for coal conversion plants are not fundamentally different from those affecting other exchangers. Factors that currently assume more-than-ordinary importance in assessing the capability in this area are simply those concerning which the least is known. Uncertainties include stream compositions, thermophysical properties, rates of fouling, corrosion and

erosion, and methods for designing for heat interchange between process streams which may include unusually large amounts of solids and/or highly viscous liquids. These problems are solvable only by specific research and development, testing, and operating experience.

2. SUMMARY

A scoping survey of the capabilities of industrial heat recovery equipment was conducted to determine their adaptability to proposed coal conversion complexes. Major categories of heat exchangers included shell-and-tube, periodic-flow and rotary regenerators, heat pipe arrays, direct phase contactors, and steam and organic Rankine cycles for power generation from waste heat. Primary applications encompassed feed-effluent and other process stream interchangers, combustion air preheaters, and heat recovery steam generators (waste heat boiler-superheaters).

A review of past experience in operating coal conversion plants and of some proposed designs indicates that many flows will probably be in the laminar or transition regimes, and that overall heat transfer coefficients for exchangers involving at least one process stream will be low [≤ 100 Btu/(hr)(ft²)(°F)]. Additional research is needed to obtain more physical property data (especially viscosities), improved heat transfer correlations for multiphase streams (e.g., coal-oil and coal-water slurries), and rates of fouling, corrosion, and erosion.

A questionnaire was sent to 23 companies that specialize in large, severe-service units requesting their views on the design and fabrication of 12 potentially critical heat exchangers selected from three evaluation design studies of coal conversion plants. Fluid streams included coal-oil,

syngas-char, flue gas, water-steam, air, and various liquid and gaseous mixtures. The following variations were included: temperatures, 40° to 1700°F; pressures, 65 to 4280 psia; flow rates, 314×10^3 to 13.3×10^6 lb/hr; and heat loads, 76×10^6 to 2920×10^6 Btu/hr. The information contained in the questionnaire responses (≈65% response rate), combined with information obtained from discussions with individuals knowledgeable in the process heat transfer field, indicated that design of the subject exchangers would be more difficult than their fabrication.

The collective experience for shell-and-tube units extends to:

Shell diam = 16 ft	Area per shell to ≈40,000 ft ²
Tube length = 80 ft	Approach Δt to 1°F
Tubesheet thickness = 25-1/2 in.	Weight = 180 tons
No. tube passes = 28	

Overall vendor fabrication capabilities are quite broad. Designs are based on the process and property data provided and the materials of construction specified by the buyer. There are no warranties concerning material lifetime or the effect of fouling on performance.

It is concluded that the single area providing the greatest potential for extending U.S. industrial heat-recovery equipment capabilities as related to coal conversion processes is a research, development, and testing program to acquire more physical property and heat-transfer data and more reliable design correlations. Liaison with heat exchanger vendor and user representatives should be maintained and extended to obtain their ideas on a regular basis regarding program directions.

3. SCOPE

3.1 Process Streams

Heat recovery and utilization are possible in two general types of equipment. Process vessels such as chemical reactors involve heat interchange because they either contain exothermic reactions or heating is used to sustain endothermic reactions. It is often necessary to protect the vessel or a catalyst from excessively high temperatures. Quenching fluids are introduced in some cases ("cold shot" cooling), but heat removal surfaces are often installed. Jackets and coils have been used and heat pipes have been considered. Heat extraction within a process vessel itself is excluded from the scope of this report. The second general type of equipment used for heat recovery is the heat exchanger or waste heat boiler which utilizes the flows between process vessels and is the subject of this report. The size and geometry of a process vessel often determines whether internal heat recovery surfaces or an external exchanger is preferable.

Most coal conversion plants include a section for gasification of coal and/or char. Some plants also include sections for producing and refining liquids for fuels, chemicals, and feedstocks. In addition, solid effluents such as sulfur, char, ash, and unreacted coal are normally generated.

Coal gasification plants include at least some of the following units:

1. coal treatment;
2. gasification (in one or more stages) at 900 to 3300°F;
3. catalytic shift conversion (in one or two steps) at 700 to 800°F;

4. gas purification at -60 to 260°F; and
5. methanation at 600 to 900°F.

The temperature drops between units provide opportunities for heat recovery, and the methanation step will require internal heat removal.

Several auxiliary processes serving coal conversion plants are partially proprietary in nature and are not discussed in detail in this report. They include the power, oxygen, sulfur recovery, and cryogenic separation plants. Cryogenic processes often utilize power recovered from the adjacent balance of plant as well as internal heat interchanging.

Coal liquefaction plants differ widely depending on the liquids and associated gases produced. Such plants may include the following operations: (1) feed preparation, (2) dissolution or pyrolysis, (3) gasification, (4) distillation, (5) filtration and drying, (6) hydrotreating, (7) ammonia removal, and (8) auxiliary units for fuel gas production and provision of utilities. Heat recovery is possible from hot streams in operations 2 to 8.

Streams from which heat is recoverable include:

1. Solids: char, ash, and slag, in some cases slurried with water or oil.
2. Liquids: distillation bottoms and product streams.
3. Gases: air, syngas, SNG, and steam.
4. Multiphase mixtures: gasifier or carbonizer off-gas with suspended solid char and solid ash and with varying amounts of liquid oil and tar; slurries of slag in water and dissolver liquid-solid products.

Streams being heated may include:

1. Solids: moist coal feed.
2. Liquids: feed streams to reactors and distillation columns.
3. Gases: air, oxygen, and steam feed streams.
4. Multiphase mixtures: reactor slurry feed composed of coal in oil with hydrogen gas, water for steam generation, filter cakes.

A list of the major coal conversion chemical reactions and a brief description of each¹ is given in Table 1.

3.2 Application Examples

Twelve severe-service heat exchangers were selected from three evaluation design studies of coal conversion plants. The twelve exchangers, listed in Appendix C, have been the subject of inquiries and discussions with manufacturers of heat exchangers. The evaluation studies, sponsored by ERDA, were based on processes that have undergone bench-scale or pilot plant testing.

The first three exchangers (A1, B1, and C1) are waste heat boilers and steam superheaters from a study conducted by the Ralph M. Parsons Company² of a Fischer-Tropsch complex. The proposed flows, temperatures, and pressures for one of two trains in a commercial-capacity plant are shown, but multiple shells would actually be used. The exchangers D2, E2, and F2 are taken from a study made at ORNL³ of a Synthoil unit producing 100,000 bbl/day. Item D2 is a feed-effluent exchanger and the other two are waste heat boilers. As shown, they are for a single-train, full-capacity plant. The last five items, H3 through L3, are from a study⁴ of a hydrocarbonization plant producing clean fuels equivalent in heating value to 100,000 bbl of fuel oil per day. Exchangers H3 through K3 are

Table 1. Principal coal conversion reactions

Reaction	Temperature range		Pressure range (psia)	ΔH_r^a		Catalyst
	$^{\circ}\text{F}$	$^{\circ}\text{C}$		Btu/lb-mole	kcal/g-mole	
Devolatilization						
COAL VM $\rightarrow$ C + CH ₄	600-1000	315-540	15-300	Endothermic ^b		None
Carbon-steam						
C + H ₂ O $\rightleftharpoons$ CO + H ₂	1600-1900	870-1040	200-600	58,300	32.4	None
C + 2H ₂ O $\rightleftharpoons$ CO ₂ + 2H ₂	1600-1900	870-1040	200-600	44,300	24.6	None
Water-gas shift						
CO + H ₂ O $\rightleftharpoons$ CO ₂ + H ₂	700-770	370-410	360-450	-16,600	-9.2	Iron oxide with Cr ₂ O ₃ activator, cobalt oxide
Hydrogasification						
C + 2H ₂ $\rightleftharpoons$ CH ₄	850-1450	455-790	1000-3000	-38,200	-21.2	Alkali carbonates (gas product); Co oxide and Mo oxide (liq product)
Methanation						
3H ₂ + CO $\rightleftharpoons$ CH ₄ + H ₂ O	600-900	315-480	300-700	-94,300	-52.4	Raney Ni (Ni and Al), Ni
Methane reforming ^c						
CH ₄ + H ₂ O $\rightleftharpoons$ 3H ₂ + CO	1250-1650	680-900	150-450	97,200	54.0	Ni, Ni oxide
Methanol synthesis						
2H ₂ + CO $\rightleftharpoons$ CH ₃ OH	500-600	260-315	750-4500	-41,800	-25.2	ZnO, Cr ₂ O ₃ , CuO
Fischer-Tropsch synthesis						
$n\text{CO} + 2n\text{H}_2 \rightarrow (\text{CH}_2)_n + n\text{H}_2\text{O}$	430-645	220-340	325-380	-70,900	-39.4	Fe, Fe with Cu and K activators
$2n\text{CO} + n\text{H}_2 \rightarrow (\text{CH}_2)_n + n\text{CO}_2$				-88,200	-49.0	
Carbon combustion						
C + O ₂ $\rightarrow$ CO ₂	1600-2600	870-1425	15-150	-170,100	-94.5	None
2C + O ₂ $\rightarrow$ 2CO	1600-2600	870-1425	15-150	-49,300	-27.4	None

^aHeat of reaction at temperature = average of stated temperature range. Minus sign denotes exothermicity with carbon as graphite, other species as gases.

^bTo devolatilize to coke, average value is 1500 Btu per pound of coal.

^cFor heavier hydrocarbons, C_nH_m + nH₂O $\rightleftharpoons$ nCO + (n + 0.5m)H₂; for reforming naphtha (BR=100-300°F), t = 1250-1650°F, and P = 15-370 psia.

gas-to-gas units that are subject to significant pressures on both tube-and-shell sides. L3 is a gas-to-gas exchanger operating at near atmospheric pressure.

3.2.1 Feed-effluent exchangers

Heat recovered from a process stream may be used to heat another process stream. When both streams are associated with one equipment item or process, the heat recovery is accomplished in a feed-effluent (F-E) exchanger. An F-E exchanger will usually be more attractive than alternate third-media liquid systems if:

1. conditions remain safe if feed and effluent inadvertently contact each other;
2. distances and elevations are such that the available pressures can maintain a flow, and thermal losses from the lines are not excessive (i.e., adequate temperature differences can be maintained); and
3. process-fluid properties can be determined with sufficient exactness so that an economical equipment size can be determined. Attempts to size equipment conservatively often fail to compensate for inadequate data.

Another report⁵ described some F-E exchangers applicable to coal conversion in which a multiunit exchanger train is used to recover heat from the raw syngas, initially at 1000 psia and 1700°F, flowing from the high-pressure gasifier. Steam, generated for gasifier feed by cooling the syngas in a waste heat boiler, is later superheated; the syngas is used intermediately to preheat the oxygen feed to the gasifier. The service for the exchangers is potentially severe because the raw syngas stream contains 20-wt % char as well as hydrogen sulfide and hydrogen.

It has been proposed that:

"These three heat exchangers are vertical shell-and-tube single-pass units. Gas with entrained char flow is from top to bottom of each exchanger. To minimize tube abrasion, the tube inlets are fitted with abrasion-resistant ceramic ferrules to protect the turbulent entry area. The tube materials of construction are selected on the basis of proven suitability at the specific set of operating conditions.

These syngas coolers are patterned after units in petroleum fluid catalyst line exchangers. These exchangers are also similar to the high-pressure tubular boilers cooling ash-laden syngas under similar conditions in a coal-based ammonia plant."⁵

Item D2 in Appendix C illustrates one of the most difficult design problems in the coal conversion plants that has been studied. The F-E exchanger was proposed as a means of increasing the thermal efficiency of a Synthoil reaction unit by transferring heat from the hot, crude Synthoil reactor product to the reactor feed stream prior to additional preheating of the feed in a fired heater. Unit D2 is a shell-and-tube (S&T) exchanger that is difficult to design because of the fluids which must be handled.

Work with Synthoil reactor feed preheaters at the Pittsburgh Energy Research Center (PERC) has been described.⁶ It was indicated that a 1/2-ton per day (TPD) unit was to be the basis for scale-up to a 10-TPD unit. Efforts to minimize the cost of the preheater were complicated when tests showed that the optimal residence time for the feed in the preheater might be as long as 20 min. The long residence time was considered necessary to accomplish predigestion (solvation). The unit selected for the 10-TPD plant is a series of three 8-in.-ID by 14-ft-long tubular vessels heated in a furnace.

The F-E exchangers for the ORNL Synthoil evaluation³ were selected but with reservations as to their practicality. The exchanger design is complicated by the high solids content of the feed slurry (35 wt %) and by the presence of hydrogen gas. There is concern regarding coking of effluent on the tube surfaces. Some literature⁷ has indicated potential problems as a result of the swelling of coal particles in a slurry, with unusual viscosity peaking at temperatures similar to those encountered on both sides of the D2 exchanger. Yavorsky⁸ indicates that viscosity peaking would not occur on the effluent side of the heat exchanger but in the converter; however, there is a possibility that such peaking would occur on the feed side. ERDA is sponsoring research⁹ in this area.

Other F-E exchangers listed in Appendix C are items H3, I3, J3, and K3; each is a gas-to-gas unit. The operating pressure in I3 (2500 psia) would limit the applicability of several equipment types, but there are many possible applications of S&T exchangers for high-pressure gas-to-gas service.

3.2.2 Flue gas to combustion air exchangers

An exchanger of this type is item L3 in Appendix C. The combustion air preheater was sized⁴ to consist of six units transferring a total of 1074×10^6 Btu/hr. It is fairly typical of air preheaters in the utilities industry. The units selected are of the "heat wheel" or Ljungstrom type with vertical shaft orientation and a rotating heat-storage matrix. This preheater would exhibit a small amount of fresh-air leakage into the flue gas, resulting in a lower temperature flue-gas exhaust than the heat load would indicate.

3.2.3 Waste heat boilers

Waste heat or heat recovery boilers may be utilized to generate steam for utilities (e.g., to drive turbines) or as a process feed material (Fig. 1). They may also be used to vaporize other liquids such as organic heat-transport fluids (at temperatures generally less than 750°F). Waste heat boilers (WHBs) have been built for flue-gas flow rates up to 1 million scfm. In the Claus sulfur-removal process, the reaction furnace feeds an integral waste heat boiler operating at a typical inlet gas temperature of 2300°F. Two tube passes are used in the units, which generate steam on the shell side.

The process evaluation reports selected for this survey used boilers for generating steam. The first three items, A1, B1, and C1 in Appendix C were proposed² for use in a gasifier section. Communications from vendors have confirmed the availability of such fire-tube steam generators and superheaters. The inlet and outlet channels and the tube sheets would be refractory lined to protect the metal from the hot syngas. One manufacturer suggested the use of a double tube-sheet arrangement to separate the temperature and pressure requirements for the steam generator. The same type of arrangement was also proposed for steam superheaters, steam generation between two closely spaced tube sheets to assure cooling, and allowing the use of thin tube sheets to minimize lamination and other stress problems. A steam drum could be elevated above the WHB, and one such drum could serve if two boilers were furnished.

Other WHBs shown in Appendix C are Item E2, which is a 350-psig generator, and Item F2, which is a 50-psig generator. Both are of water-tube construction.³ Item E2 would probably be one or two U-bend units with one

ORNL-DWG 77-19642

Fig. 1. Waste heat boiler for heat recovery from hot gas.

steam drum. Some vendors suggested switching the gas to the tube side (downward flow) because of the high loading of particulates.

Information¹⁰ concerning the current state of the art in WHBs was received from one manufacturer. It indicates that process steam is usually generated at 125 to 650 psig. For power generation, pressures of 650 to 1650 psig are used. In the case of coal conversion processes, the higher pressure steam may be used for a process feed material. Either natural or forced convection units are available, depending on space limitations. For water-tube construction, inlet gas temperatures are generally limited to about 1800°F and gas pressures must be low (typically below 2 psig) unless an S&T-type exchanger is used. For hydrogen service in some ammonia plants, U-tube S&T exchangers utilize refractory-lined shells exposed to inlet gas temperatures of 1500 to 1800°F. Water jackets are used in some cases to protect the shell, and special efforts are made to protect the tubes by maintaining high tube-side velocities. Such S&T units for water-tube boilers are apparently most economical when steam pressure is above 1200 psig. Waste heat boilers of fire-tube type construction are generally less expensive, but are limited to steam pressures under ~1000 psig (a limit currently being increased). Europeans have reportedly¹¹ installed such fire-tube boilers for 1500-psig steam generation. They may be either vertically or horizontally oriented. The tube sheets are vulnerable.

Waste heat boilers generating steam at high pressures require a feed-water purity similar to that used in utility plant boilers. A recent article¹¹ stressed the internal corrosion that can be caused by dryout in horizontal water-tube boilers and the external corrosion of tubes in fire-tube boilers. In both cases, adequate feed-water purity can be a major

problem. Ribbed or rifled-bore tubing can be used to increase the dryout heat flux in water-tube designs.

In coal conversion plants, exchanger manufacturers are concerned about the erosive or abrasive action of the char that may be entrained in the gas. Generally, as is the case with other exchangers, WHBs are not guaranteed with respect to corrosion and erosion failures.

4. EQUIPMENT TYPES

4.1 Regenerators

4.1.1 Batch type

Batch-type thermal regenerators (i.e., exchangers with thermal storage) are so identified because they are operated in an alternating batch mode. Cold and hot-gas streams are alternately valved to direct a given stream to one of two, or sometimes more, regenerator chambers so that the streams may alternately exchange heat with a high-heat-capacity matrix. The matrix is stationary and is usually made of a ceramic material, either crude pebbles or rubble or, for lower pressure drop, preformed checker-work elements of simple or intricate shapes. For gas flows, the containment vessels are necessarily large. Only a small fraction of the heat stored in the matrix is removed during the cooling cycle because of the low thermal conductivity of the matrix. Batch-type regenerators have been used extensively in the steel and glass industries for preheating air to 1650 to 2200°F, usually using a checker-work matrix. One summary¹² describes blast furnace stoves utilizing 9 x 4.5 x 2.5 in. firebrick assembled to form flues 3.25 in. sq. An ambient air flow of 100,000 scfm is preheated to 900 to 1200°F (1000°F av) using four stoves; one unit preheats the air while three are being heated by hot gas. The shells for the stoves are

large vessels (24 ft diam and 100 ft high). They contain 124,000 ft² of surface area. Bypassing air flow is used as a control to maintain a reasonably constant preheat temperature. Open-hearth furnace thermal regenerators require larger flue openings because of contaminants.

A less-expensive pebble-bed regenerator has been used successfully¹² to preheat air to 2800°F in a 25-TPD blast furnace producing pig iron and ferroalloys (manganese, chromium, and silicon). Work on the magneto-hydrodynamic (MHD) process^{13,14} has favored batch-type regenerators at very high exit-air temperatures (~3000°F) and at pressures up to 8 atm. Checker-work matrices are used.

Disadvantages of the batch regenerator include the need for large switching valves (a possible requirement for control valves if reasonably constant preheat temperatures are needed), difficulty in cleaning, cross contamination of streams unless purging between cycles is performed, and sluggishness in bringing the regenerator to temperature. Nevertheless, this type of equipment is relatively attractive for use with high-temperature gas flows bearing particulates that might plug the smaller passages in equipment such as rotary regenerators. It would be most attractive for streams having large temperature differences and moderate pressures. Although shop-fabricated units are available, the sizes most economical for a coal conversion plant would probably require field fabrication. Methods for the thermal design of batch regenerators have been summarized by Mueller¹⁵ and Schack.¹⁶

4.1.2 Continuous type

Combustion air is heated in an air preheater upstream of the furnace-boiler of an electric generating plant. The device may be the "heat wheel" type (known also as a Ljungstrom or Lugat preheater). In these units, a

heat-sink matrix rotates and cyclically exposes a sector of the matrix, first to a hot flue-gas stream and then, as it turns further, to a cooler inlet-air stream. (See Fig. 2.)¹⁷ The heat-wheel matrix is usually made of ceramic coated material in those sections that are to operate above 600°F to 700°F. The matrix may be made of two layers, one of which is a more corrosive-resistant material to resist the acid that condenses on the cooler surface where air enters the wheel. If the layer is showing corrosion, the corroded matrix may be replaced, or in some units it may be removed, reversed, and reinstalled to present a fresh surface to the cold blast of air. Leakage rates typically grow with corrosion of the seals and the matrix and may reach 15 to 20% before corrective measures are taken. Other options for the heat wheel are soot blowers and washing jets for keeping the surface clean to maintain low pressure drops; however, the added deposits can help in heat transfer because of their heat capacity. Wheels are normally used in moderate-temperature environments and at low pressures.

A device resembling the heat wheel is the "Rothemuhle air heater." (See Fig. 3.) It utilizes a heavy stationary matrix for transient storing of heat. A sector of the frontal area of the disc-shaped matrix is subjected to a blast of cool air diverted through the disc by slots in two rotating hoods, one on either side of the disc. The hoods direct hot air through that portion of the disc not exposed to the slotted openings. The two currents of air are swept in a circular motion. Flow-stream orientations are reversed in some installations. The primary advantage of the Rothemuhle air heater seems to be the smaller rotating mass (20% of the total unit weight). In carbon steel it is used to 800°F. In both types of exchangers, a considerable area must be sealed if hot and cool gas streams are to be kept separate. Heat wheels for the utilities industry

Fig. 2. Heat and moisture recovery using a heat-wheel type of generator.

ORNL-DWG 77-19635

Fig. 3. Arrangement of counterflow regenerative air heater with stationary plates - Rothemuhle type. (Adapted from Steam/Its Generation and Use, Babcock and Wilcox Co., New York, 1975.)

normally experience pressure differences of >40 in. of water. Heat wheels with thermal efficiencies to $\sim 85\%$ and diameters to ~ 70 ft have been reported.¹⁷ The same source also reports that purging has been used to limit contamination in a laminar-flow heat wheel to as little as 0.04% and particle contamination to less than 0.2% . Single heat wheels have been built to preheat as much as 10 million lb of air per hour.

Equations and design curves for rotary air preheaters are available in several references.^{15,18} Smaller power plants may use tubular air preheaters, or the heat wheel and the Rothemuhle preheater described above.

Costs for rotary preheaters vary considerably depending on pressure-drop limitations and moisture loadings in the plant to be served. Estimated prices for large units are currently about $\$0.45/\text{lb}$ per hour of flue gas ($\sim \$2/\text{scfm}$).

4.2 Direct Contact Coolers and Condensers

In cooling and/or condensing hot gases and vapors such as pyrolysis gases, considerations relating to pressure drop, fouling, and economics may render indirect heat exchange through metal walls undesirable or infeasible. An important application for which direct contact quenching of a gas with a liquid has been used is cooling an ethylene-furnace effluent process stream. Such quenching cools the stream rapidly through a critical temperature range to prevent an undesired continuation of the reaction (i.e., to freeze the chemical equilibrium). Direct contact cooling is also employed while simultaneously scrubbing entrained dust from dirty hot gases in barometric steam condensers, and routinely in wet cooling towers.

This section deals with cool-liquid--hot-gas contactors; those involving cooling of a hot solid with a cool liquid are described in Sect. 4.5.

A strict interpretation of energy recovery would exclude liquid-gas direct contactors, because even when additional steam is needed for a downstream reaction step, as in some processes preceding the carbon monoxide shift conversion, the initial gas stream is diluted and thermal energy is not actually recovered. Low-pressure steam can be used to heat low-pressure water by direct contact in a line mixer or holdup tank, but this is usually done only on a small scale.

A review article¹⁹ briefly describes design rules of thumb and equations for direct contact cooler-condensers such as baffle-tray columns (sheet trays with no perforations or weirs), crossflow-tray columns (with bubble caps, valves, or holes), spray chambers (with solid- or hollow-cone nozzles), packed columns (e.g., with Raschig rings, Berl saddles, or Pall rings), and pipeline contactors. Four cases are considered: simple gas cooling, gas cooling with partial coolant vaporization, and cooling with partial or total gas condensation. The design relations are quite approximate; many are derived by analogy from mass transfer data or correlations, and none are directly applicable to tower diameters exceeding 6 ft (usually restricted to ≤ 2 ft diam). At best, the equations are based on data obtained with air-water, air-oil, and H_2 -light hydrocarbon-oil. A more comprehensive study should include the degree of backmixing and of entrainment in large-diameter spray chambers, and the axial drop-size distribution in pipeline contactors with turbulent mist flow. Data obtained in studies of water-injection line desuperheaters for steam and with other gas-liquid combinations indicate that the uniformity of initial phase dispersion is important, because thermal effectiveness decreases rapidly with nonuniformity of dispersal.

A summary has been published²⁰ of an equilibrium-stage design approach for the final cooling of cracked gas streams in water quench towers, which usually operate at 90 to 250°F; ranges of tray thermal efficiency values are included.

Since direct-contact coolers are custom designed in the larger sizes, adequate generalizations concerning their costs are not available.

4.3 Recuperators

4.3.1 Shell-and-tube and similar exchangers

The shell-and-tube (S&T) exchanger is the most common type. Its use in the chemical industry has been at high pressures and temperatures and in large sizes. The principal weakness of the S&T stems from the velocity changes of the process flow streams in the shell side of the exchanger (where baffles are often used), and in the tube side where velocity changes are necessary at the tube sheets. The velocity changes are less significant when either gases or liquids are involved, but multiphase flow can be a problem, because phase separation can occur at turning points and at changes in the cross-sectional area. When solids are present, accumulations may reduce the available heat transfer area, and high-velocity particles may cause erosion. Erosion has been countered in some S&T exchangers by using deflector plates, extended tubes, or special ferrules. Uncertain viscosities are also a problem in S&T exchanger design; flows may be limited severely in regions of the heat exchanger.

An alternate type of heat exchanger that is efficient for viscous fluids is the plate type. The device is made of thin plates, stamped in such a way that various flow patterns are produced when plates are clamped against each other. The compact units can be engineered to suit the flow media characteristics by varying corrugation patterns in the plates. The

critical Reynolds number is reported to be low (100 to 400). The plate exchanger is limited to low pressures (~ 300 psig) because the broad plates are difficult to restrain. It is also limited to low temperatures ($\sim 450^\circ\text{F}$ max) because gaskets are required to seal the plates as a means of segregating and sealing the fluids from the surroundings. Sizes are limited to about $15,000\text{ ft}^2$ per unit. Designs capable of a 2°F approach temperature difference and an attainable recovery of 90% or more are available. Costs compare favorably to the S&T exchangers when stainless steel or higher grade materials are required.

Use of the spiral-plate exchanger is also somewhat limited, as with the plate-and-frame exchanger, although a welded closure can be provided to avoid the temperature restraints that are inherent with gaskets. This type of exchanger employs a deeply grooved plate or membrane that permits two opposing streams to flow countercurrently in spiral paths. Cross-flow arrangements are also available, as are special models for overhead condenser service. Some units may be as large as 2000 ft^2 , and pressures of 150 psig may be attained in standard models (higher on special units).

Similar to the spiral-plate exchanger is the spiral-tube exchanger that uses a multitude of tubes coiled side by side to form a wall which is then coiled into a spiral shape that is tightly confined by a casing. Each end of each tube is brazed or welded to a header that extends through a cover plate on one end of the casing. The two process streams are in counterflow. Sizes currently do not exceed $\sim 500\text{ ft}^2$, and pressure ratings for standard units do not exceed 4500 psi on the tube side or 300 psi on the shell side. Gaskets may be eliminated by welding the unit closed, but cleaning could be a problem. With gaskets, the maximum temperature would be limited to 450°F .

A survey of costs for large heat exchangers compiled from manufacturers' estimates and quotes (ten companies) obtained in early 1975 has been published.²¹ A summary is shown in Fig. 4. All costs are given in 1976 dollars. The average costs varied from \$7 to \$50/ft² for S&T units with the following specifications: carbon steel construction; surface areas of 20,000 to 35,000 ft²; tube-side pressures of 200 to 2000 psia; shell-side pressures of 100 to 4000 psia; 3/4 to 1-in.-OD tubes, 30 to 40 ft long, arranged with a pitch/diameter ratio of 1.25; and one-pass tube-and-shell sides. For induced-draft air-cooled units with aluminum fins and tube pressures of 100 to 4000 psia (other parameters are the same as above), the average cost range was \$15 to \$19/ft² for carbon steel construction and \$22 to \$30/ft² for 304 stainless steel (costs based on bare-tube area).

Unit costs have been compared²² for heat exchangers that are identical except for the materials of construction used for the tubes and the 1/4-in. lining of the tube sheets. Although promoting the use of titanium, data were given for 14 of the 40 metals that had been considered. Figure 5 shows results that were given for a range of sizes for heat exchangers made of 3/4-in. OD by 16-BWG tubes that were 20 ft long, unless otherwise noted. The heat exchangers portrayed all have mild steel shells, are of TEMA*-type BEM, and were designed according to the ASME Code, Sect. VIII, Division I.

The industrial furnace market currently uses both recuperators and regenerators for heat recovery. The regenerators described above have internal thermal-storage sink matrices that are alternately heated by hot flue gas and then cooled by atmospheric combustion air directed through

*Tubular Exchanger Manufacturer's Association.

ORNL-DWG 77-19648R

Fig. 4. Approximate heat exchanger costs as a function of shell-and-tube side pressures (1976 dollars).

ORNL-DWG 77-19647 R

Fig. 5. Cost comparison for shell-and-tube heat exchangers, mild steel shells, and tubing.

them for preheating. The recuperator, on the other hand, utilizes a metallic surface between the two continually flowing streams. The surfaces may be constructed to maximize radiant or convective heat transfer. Two sections may be installed in series, a radiant section at the hot (gas inlet) end and a convective section at the cooler end.

The radiant section of a stack type recuperator consists of two concentric cylinders. With significant lateral pressure differentials, the high-pressure stream is sent through the inner cylinder, and the low-pressure stream is sent through the annulus to maintain the cylindrical shape of the intervening wall. Two radiant sections can be arranged in series, with the cooled-gas end in counterflow and the hot-gas end in parallel flow. This arrangement can be used to protect the recuperator wall from excessively high temperatures at the gas inlet end. The basic radiant type is generally used for hot stream temperatures above 1600 to 1800°F; at these temperatures, thermal radiation from carbon dioxide and steam are the principal transfer modes. Below ~1300°F, convection sections are needed. Convective tubular exchangers, either vertical or horizontal, with either gas or air through the tubes, are installed to receive flue gas already cooled in the radiant zone. Generally, the radiant type recuperator is used for low-pressure service (<50 in. of water), but higher pressure units (2 psig) can also be built. Both field-fabricated and shop-fabricated units are available. Manufacturer's literature suggests that radiant type recuperators may be used with gas entering at temperatures up to 2500 to 2600°F. Convective recuperators with flue gas through the tubes can handle dirty gas continuously at temperatures to 2000°F. The tubes used are 2.5 to 4 in. in diameter and each is equipped with an expansion

joint. The shell may also have an expansion joint to allow for thermal expansion. For a 950°F preheat temperature, such devices are usually designed for flue-gas velocities of 10 ft/sec and air velocities of 35 ft/sec at standard conditions.

In addition to the tubular recuperators, modular canal type recuperators are made for low-pressure service. The tubular modules are suspended in large brickwork "canals" or flues containing a horizontal flow of flue gas. The stream to be heated passes through vertical tube bundles. Flow is generally 2- to 4-pass through the tubes with flexible connections between transfer boxes at the bottom of two adjacent bundles. Square or rectangular bundles are used. Heating fluid temperatures to 2200°F are used to preheat air to 1200°F.

Yet another type of recuperator is the cage type, so called because the tube bundle resembles a cylindrical cage with an open center. This recuperator is used for higher temperatures encountered (e.g., a max gas inlet temperature of 2500°F in the glass industry). The cage is housed in a refractory-lined vessel. Heat transfer is primarily by radiation. Air may be handled at high pressure in these units (250 psig).

Recuperators may be designed for preheating fluids other than air (e.g., hydrogen). If the heating stream is dirty, radiant-type recuperators are attractive because there are fewer obstacles to trap dirt.

Schematic drawings of these recuperators are shown in Figs. 6-9.

4.3.2 Rod-baffle exchangers

The Phillips Petroleum Company developed the rod-baffle exchanger (RBE) to replace three large S&T waste heat boilers that had failed because of excessive tube-bundle vibration, and to ensure high-reliability service

ORNL-DWG 77-19638

Fig. 6. Diagram of a convective-type recuperator.

ORNL-DWG 77-19639

Fig. 7. Diagram of a combined radiation and convective-type recuperator.

ORNL-DWG 77-19640R

Fig. 8. Diagram of a small radiation-type recuperator fitted to a radiant tube burner.

ORNL-DWG 77-19641

Fig. 9. Plate-type recuperator.

on North Sea petroleum platforms. A total of 32 units, with shell diameters to 6 ft and lengths to 65 ft, are in service or are being built;²³ they are fabricated by several companies with license rights. A matrix of rods comprising a baffle set is arranged perpendicularly to the tubes, which are on square pitch. The rod diameter is equal to the minimum spacing, or ligament, between adjacent tubes, and the baffle set touches and constrains each of the tubes from four directions. Because of the longitudinal flow characteristics, close-tolerance fabrication is important, and the clearance between the baffle ring (to which the ends of the rods are welded) and the shell is held to a minimum. Virtues include minimal tube vibration, low shell-side pressure drop and fouling rate, and ease of cleaning.

Phillips is engaged in an experimental test program to develop design correlations for RBEs. Available results, which have been confirmed by field data obtained with four of the installed units, indicate a shell-side pressure drop equal to about one-half that of the same shell and bundle, segmentally plate baffled, in the same service. Tests with water, air, and oil have shown that the ratio of the heat transfer rate to the shell-side pressure drop, as measured by $(U/\Delta P_s)$, is ~ 1.8 times higher for an RBE than for a double-segment baffled exchanger. Because there are no flow stagnation zones, the entire heat transfer surface is effective. Although local heat-transfer rates are somewhat lower than those for the active crossflow stream in a segmentally baffled exchanger, the high-surface effectiveness of the RBE compensates. In addition, vortices shed downstream of each baffle rod enhance heat transfer at a low energy dissipation rate.

The 32 units are in service as WHBs, gas, oil, and water coolers, and partial condensers; some have operated for as long as 6 years without vibration failures. Costs are within a few percent of comparable segmentally

baffled S&T units.²⁴ The RBE should be attractive for use in dirty services characteristic of coal conversion plants.

4.3.3 Heat pipe arrays

Heat pipes are high axial conductance, heat transport devices that have undergone considerable development since their discovery about 12 years ago. Such a pipe is a closed duct containing a working fluid that transports heat from one end (evaporator) to the other end (condenser) by liquid vaporization in the evaporator, axial vapor flow in the central core region, vapor condensation in the condenser, and condensate return along the inner wall by gravity and/or capillary forces to the evaporator. Schematic drawings of these pipes are shown in Figs. 10 and 11.

The relevant literature is abundant; a good overall treatment²⁵ and a lucid summary of the several operating limits imposed by fluid dynamics considerations (limits relating to boiling, capillary pumping, sonic flow, and entrainment)²⁶ are available. The potential of the heat pipe in coal conversion processes has been addressed by Ranken in his overview.²⁷ Design concepts include the incorporation of heat pipes into indirectly heated coal-steam gasifiers, into methanators for heat removal, and for energy recuperation in general; these concepts are especially applicable in first-generation plants where the shift reactor effluent gas is cooled to a near-ambient temperature for sulfur removal and is then reheated for methanation. Also included²⁷ are descriptions of a heat-pipe recuperator concept²⁸ devised to increase the efficiency of heat recovery from methanator product gas and of possible ways to fabricate high-temperature ceramic (silicon carbide) heat pipes.

Earlier results of a Morgantown Energy Research Center (MERC) - Los Alamos Scientific Laboratory (LASL) experimental test program, using up

Fig. 10. Schematic of heat pipe.

Fig. 11. Heat-pipe bundle incorporated in gas-to-gas recuperator.

to three heat pipes to thermally couple a fluidized-bed coal-air combustor at $\sim 2200^{\circ}\text{F}$ with a fluidized-bed coal-steam gasifier at $\sim 1800^{\circ}\text{F}$ have been reported.²⁹ In this program, 14-in.-long, 0.84-in.-OD heat pipes made of Inconel 601 and lined with a wick structure of 316 stainless steel screen were operated at $\sim 2000^{\circ}\text{F}$ with pure sodium-fill fluid. Based on the measured decrease in condenser-end wall thickness (0.109 in., initially) caused by erosion-corrosion in the hostile gasifier environment, complete consumption of the pipe wall would have occurred at ~ 3500 hr (4.8 months). To extend the lifetime, proposals have been made to use protective coatings, sleeves, or composite wall construction.

Heat-pipe heat exchangers made from arrays of laterally finned heat pipes are typified by the heat recovery units³⁰ produced by Hughes Aircraft Co. (Electron Dynamics Div.) in Torrance, California. Their "Heatbank" exchangers presently use aluminum or copper tubes and fins and Freons 11, 12, or 113 as internal working fluids; the corresponding maximum hot-gas temperature is $\sim 450^{\circ}\text{F}$, but tests are being conducted with more expensive carbon steel heat pipes containing Freon 112 or water for use to $\sim 600^{\circ}\text{F}$. Current assemblies, which have seen use primarily in heating, ventilating, and air conditioning (HVAC) applications, are available with heat pipe lengths to 11.5 ft, assembly heights and depths (8 rows deep) to ~ 4.5 ft and 1.7 ft, respectively, and 4 to 14 fins/in. A central partition plate is used to separate (e.g., cool) inlet air from the hotter gas stream. The pipes are usually inclined slightly (tilt angle of ~ 6 degrees), with the evaporator at the low end, to augment condensate return by gravity. Thermal effectiveness should normally range from about 50 to 80%. With face velocities of 400 to 600 fpm, gas-side pressure drop is usually ≤ 1 in. of water. If the hot gas is corrosive, the heat pipes can be coated

externally with thin layers (~ 2 mils thick) of a phenolic compound (Heresite) or Teflon; in one application,³¹ nickel plating over copper tubes and fins was used. The maximum gas flow rate (each side) is currently limited to $\sim 30,000$ scfm. Approximate exchanger costs vary from \$0.50/scfm at high flow rates to \$0.95/scfm at low flow rates. Primary advantages include high heat transfer rates, compactness (high ratio of surface area to volume), no cross-stream contamination, and no need for auxiliary power to pump a heat-transfer fluid.

Thermal recovery units are also made by the Q-dot Corp. (Dallas, Texas). Their "Q-Pipe" exchangers are similar in most respects to those described above. The standard units, which include steel as well as copper and aluminum heat pipes, encompass an operating (external fluid) temperature range of -60 to 900°F . Flow partitions are fabricated from 14 or 16 gage aluminum or 304 stainless steel. Fins, spaced at 8 to 14 in., are 7- to 9-mil-thick alloys of copper or aluminum. The maximum gas flow rate in operating costs is currently $\sim 20,000$ scfm. Exchangers made with aluminized steel heat pipes are rated to a maximum gas temperature of 1325°F , and one unit has been in service at $\sim 1125^{\circ}\text{F}$ for a year. A summary of Q-Pipe exchanger applications³² indicates that most are used in the HVAC area, and that the average installed unit contains 125 heat pipes.

Similar heat-pipe arrays are also produced by Isothermics, Inc. (Augusta, New Jersey). Its standard "Thermo-Coil" units are suitable for gas temperatures to 450°F with aluminum pipes and fins, and to 800°F with copper pipes and fins. Such standard exchangers use 5 to 11 fins/in., arrays 2 to 16 rows deep, face velocities of 400 to 670 ft/min, and have been built for one-side flow rates of 30,000 to 40,000 scfm. Special

systems are said to be deliverable for flows to 10^6 scfm, temperatures to 1500°F , and thermal efficiencies to $\sim 90\%$. Such custom units use carbon steel or stainless steel pipes and fins with plate or welded fin configurations.

Some additional design considerations and correlations applicable to heat-pipe heat exchangers are also available.^{33,34}

4.4 Extended Surfaces

When one of the streams in an exchanger limits the heat transfer rate, benefit can often be derived by using an extended surface in contact with that stream. Practically, success depends on the medium being relatively clean, because extended surfaces can foul rapidly and can be difficult to clean. Fins, pegs, spines, or studs are often used on the outside of round or flattened tubes or hollow plates. Gas flow over the outside surfaces of such elements is generally preferred to flow inside the element so that the exchanger area can be decreased. The use of external rectangular fins is advantageous (compared to a plain wall)³⁵ when $\frac{2k}{hb} \geq 5$, where

k = fin thermal conductivity, $\text{Btu}/(\text{hr})(\text{ft})(^\circ\text{F})$,

h = film heat transfer coefficient, $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$,

b = fin thickness, in.

Fins should not be spaced closer than twice the thickness of the boundary layer to avoid interactive boundary-layer merging between adjacent fins. As the film heat transfer coefficient increases, the fins approach the fluid temperature and little benefit is derived from finning.

Explicit mathematical relationships may be used to determine the fin shape that will minimize fin weight for a given heat flow. Unfortunately,

the cost and the pressure drop associated with a given tube and fin design are not directly related to fin weight or to any other simple property of the assembly. For this reason, cores are often designed, built, and tested to evaluate the "life cycle" cost of an extended surface.

Particular designs such as finned-tube air coolers and radiators have been used for many years and have benefited by competitive efforts to reduce the manufacturing cost and to improve the reliability of the extended surface. For example, automatic brazing has been developed for high-volume production of automobile air conditioning heat exchangers.

In the chemical industry, there has been wide acceptance of hairpin heat exchanger sections (Fig. 12) consisting of two longitudinally finned, double-pipe heat exchangers connected in series. The arrangement is good for viscous fluids and moderate pressures. The heat transfer coefficient can be improved by using a series of longitudinal fins rather than a single continuous fin the length of the tube. The shorter fins are twisted slightly at the downstream end to interrupt laminar or transition flows at intervals. Three or more longitudinal fins can center the inner tube in the simple pipe that forms the exchanger shell. The fins are tightly bonded by mechanical means to the central tube and may be of a material different from the tube. A common variant of the double-pipe exchanger is the hairpin exchanger, which has multiple, longitudinally finned pipes, as depicted in Fig. 13. These units are usually supplied with outer-pipe diameters of 4 to 8 in.

Another common use of extended surface for heat recovery is in recuperators where gas exchanges heat with air. The Belgian DEKA surface, used for many years in Europe and elsewhere for air preheaters, is integrally cast in sections with the parent conduit material (a hollow plate). Extended

Fig. 12. Hairpin exchanger of double-pipe longitudinally finned tube construction with internal return bend. (Courtesy of Brown Fintube Co., Tulsa, Okla.)

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Fig. 13. Hairpin exchanger using multiple longitudinally finned tubes. (Courtesy of Brown Fin tube Co., Tulsa, Okla.)

surfaces in this case are sometimes used on both the inside and the outside of the conduit wall. Other integral extended surfaces have been developed that are made by skiving thin leaves of material from a duct plate of parent metal.

In addition to the integral fins just described, studs or pins may be welded to metal surfaces. A corrugated surface may be bonded or otherwise attached to a plain surface (either flat or tubular). Helical fins may be welded to tubes. Trane Co. and Curtiss-Wright have done much work to develop compact surfaces for special applications. ERDA has sponsored research on compact surfaces for use in the fields of uranium enrichment (gaseous diffusion plants), ocean thermal energy conversion (OTEC), and dry cooling towers. Extended surfaces for heat transfer are generally available either in-house or as purchased components. Although the field is constantly growing, good introductions are available for simple extended surfaces and compact heat exchangers.^{36,37}

4.5 Solids Coolers

Various coal conversion processes deal with such hot solids as ash, char, catalyst, heat carrying media, or mixtures of these materials.

For waste materials that present no problem of contamination, a fluid having good heat transfer properties (e.g., water) can be mixed with the solid. One method of recovering heat from ash is to utilize the slag-quench bath water, flashing steam from the water in a special vessel. In a proposed gasifier section for a Synthoil plant, 50-psig steam is generated in a slag tank, leaving behind a slag slurry (23-wt % ash at 298°F); considerable heat and water escape with the waste slurry. In the agitated-quench bath water, the ash-containing slag is converted to a frit or sand-

...like material which can easily be conveyed by the water. Variations in heat recovery from the water include the use of jacketed vessels or coils for steam generation or for process heating.

In cases in which contamination of the solid by a bath liquid would seriously lower its value, a fluidized-bed heat exchanger might be used. The fluidized bed requires a contaminating gaseous fluidizing medium. A compatible fluid can be separated later from the carry-over solids downstream of the cooler. An ERDA-sponsored study³⁸ proposed that a fluidized-bed char cooler would receive char at 555 psia and 1600°F. With the recovered heat, 600 psia steam could be generated in a tube bundle immersed in the bed of char fluidized with steam (see Fig. 14.) Although fluidized-bed coolers are not standardized commercial equipment, they have been built for special applications such as cooling alumina.

Several configurations of air-cooled, cement-clinker heat recovery devices are used in the cement industry.³⁹ An example is shown in Fig. 15. These units require large flows of air that are used to heat incoming feed material in intimate contact. The air flow is cleaned of carry-over solids between stages. The equipment is presently limited to near-atmospheric operation.

Heat recovery from solids does not appear to be practical with the metallic conveyor-belt coolers now in use (e.g., in the food industry). The belts may be made of metals such as stainless steel but low metal temperatures are required to prevent annealing. Although hot solids may be handled, the cooling medium, generally water, is allowed only a slight temperature rise ($\sim 2^\circ\text{F}$). The coolant would not be useful as a heat source in other parts of the plant, but it would be uncontaminated. Metallic conveyor

ORNL-DWG 77-19645

Fig. 14. Conceptual fluidized-bed char cooler. (Adapted from D. Kunii and O. Levenspiel, Fluidization Engineering, Wiley, New York, 1969.)

Fig. 15. Shaft-type clinker cooler, Walther-Beratherm system.

belts currently in use are not made for operation at pressures significantly above atmospheric.

4.6 Rankine Cycles for Converting Waste Heat to Electrical or Mechanical Energy

Low-temperature cycles using organic working fluids are typified by the Organic Rankine Cycle (ORC) system offered by Kinetics Corp. (Sarasota, Fla.). A waste-heat stream is used to generate steam, which condenses in an S&T exchanger, generating Freon vapor to drive a rotary positive displacement expansion engine. The expander isentropic efficiency, at a rated ΔP (with Freon 12) of 315 psi, is 80%. The Freon vapor is condensed in a water-cooled exchanger and recirculated to the vaporizer. Units producing 10 kW to 10 MW, or mechanical power for pumping or compression, are said to be available. Cycles have been operated on Freons 11, 12, and 114 at availabilities exceeding 99%. The system is presumably applicable to any waste-heat source temperature of 150°F or higher, a difference between heat source and heat sink temperatures of 100°F or above, and typical pay-out times of 3 years or less have been estimated. Conservative approximations of output power under usual circumstances can be made with the following simplified equations, as described in a company brochure.

For hot water,

$$HP = 0.0035 Q (T_h - T_c), \quad (1)$$

where HP is the ORC system shaft output (hp), T_h is the hot-water supply temperature (°F), T_c is the cooling-water temperature (°F), and Q is the hot-water flow rate (gpm).

For atmospheric-pressure steam,

$$HP = 0.038 W, \quad (2)$$

where W is the steam flow rate (lb/hr), and it is assumed the cooling (condensing) water is in the normal ambient temperature range. Based on an estimate of April 1977, a system generating 10 kW(e) costs about \$12,500, or \$1250/kW(e) for a 185°F hot-water stream and with cooling water at 85°F. A 10 MW(e) system is projected to cost $\sim$ \$4 million, or \$400/kW(e).

High-temperature cycles using steam as the working fluid are exemplified by the waste-heat steam-power systems provided by Thermo Electron Corp. (Energy Systems) in Waltham, Massachusetts. These packaged, closed-cycle systems use steam from a heat recovery boiler to drive a turbine generator, following which the steam is condensed and recirculated as water to the boiler. With clean (low-particle loading) exhaust gases, the boiler can be made more compact by using externally extended surfaces and forced water circulation. For dirty hot gases, bare tubes and natural water circulation are used, and steam or air soot blowers and collection hoppers are provided. Units generating 500 kW to 15 MW, operating as either condensing or back-pressure systems, are available. Turbine-inlet steam conditions can vary from 15 psig at 250°F (saturated) to 600 psig at 700°F. For a nominal 1500-kW system with turbine-inlet steam at 450 psig, 700°F, and a condenser temperature of 115°F, the steam rate is 13 lb of steam per kilowatt hour. These units typically cool fairly clean waste gases at 1000 to 1500°F down to $\sim$ 350°F, or to a temperature slightly above the dew point. Standard units are guaranteed, excluding materials, for 1 year. The most recurrent problem is boiler-surface fouling by particles in the hot-gas stream. Based on estimates of June 1977, the installed cost, excluding foundations, is about \$650/kW(e). Thermo Electron is also testing prototype units using organic working fluids such as thiophene.

4.7 Summary of Typical Vendor Offerings

Table 2 summarizes a range of typical vendor offerings, as reflected by vendor catalogs and brochures examined in mid-1977. The vendor identifier numbers may be found in Appendix A.

The following abbreviations and acronyms have been used in the subject tables:

ACFM — actual cubic feet per minute

EX — example

FG — flue gas

MM — million

S — shell

S&T — shell and tube

T — tubes

WG — water gauge

The examples cited are illustrative only and are not to be viewed as comprehensive.

5. OTHER TYPES OF HEAT EXCHANGE EQUIPMENT

Many types of heat exchanger equipment are manufactured for use in the oil, chemical, and allied industries.⁴⁰ Several types have only limited potential for heat recovery in coal conversion plants but will be found in auxiliary units. A brief description of several selected equipment types follows.

Cascade coolers, sometimes called trickle or trombone coolers, are made of horizontally oriented tube runs connected in a serpentine fashion and form a vertical bank that is cooled by water trickling over it.

Table 2. Summary of equipment availability

Exchanger category	Vendor	Equipment type	Temperature (°F)	Pressure (in. H ₂ O)	Common process streams	Approx current equipment size and capacity	Comments
Radiation recuperator	1	Stack, cage	Graph: 2600 FG, 1400 preheat	In annulus, 10 to 15	Dirty gas and air	6 ft diam by 60 ft long.	Cage refers to the appearance of the ring of tubes.
				Glass furnaces, 20 to 30			
	2	Stack	2500		Dirty gas and air	12 ft diam by 45 ft long (to 200 MM Btu/hr) - applies to all Vendor 2.	
		Cage			Tubes, air; shell, dirty gas	Inner shell: stainless steel construction; outer shell: insulated inside and lined.	
		Canal				Metal tubes; shell refractory lined.	Canal refers to the appearance of the brickwork canal in which the recuperator sits.
		Module				Shell lined; single unit to 3.2 MM Btu/hr; units may be combined.	
Convection recuperator	1	Tube	FG, 2000; air, 1300	FG, 0.6; air 8	FG vs air	Closure may be gasketed or welded.	
		Canal	FG, 1700; air, 1200	FG, 0.75; air 5 to 12			
	2	Tube	FG, 2200; air, 950			12 ft diam by 45 ft long.	
		Tube	FG, 1900; air, 950			12 ft diam by 45 ft long.	
		Canal	FG, 2200; air, 1200			3.5-in.-OD tubes centrifugally cast; 27 MM Btu/hr per unit; units may be combined.	
	22	Extended surface	Min 230; max, not given			Tubes 11.2 ft long; with glass tubes, TFE gaskets; horizontal orientation preferred for ease in cleaning.	Glass-tube section if necessary for corrosion resistance in cold end section. Metal, cast iron; gasket, asbestos. Water wash may be necessary below dew point.
	25	Plate	FG, 250 to 1000	3 or less		Module of 12 plates, 3.3 ft wide x 6 ft deep x 10 ft high; FG velocity 2400 fpm.	Vertical hollow-plate elements with air channeled through U-shaped paths within elements; FG, usually baffled flowing outside the plates; two 3750-scfm fans supply cooling air to a module. Chains may be used to clean plates. Ex: Cool 70,000 ACFM from 1000°F to 250°F using 10 modules, 10,548 ft ² , 32 ft 10 in. diam.
Recuperator with both radiation and convection sections	1	Common shell; one unit parallel flow and one unit counter flow	Very high preheat		FG and air		Natural draft can be used.
		Two shells; convection S&T FG in tube	Air, 1500				Shot cleaning available.
Heat wheel, metal	1	Horizontal or vertical	1100 std; higher available		FG and air	4-ft-diam to 17-ft-diam units made of 12 sections; 21 sizes.	Bearings oil lubricated, water cooled; matrix in two layers with cold layer reversible. Soot blower or wash used.
	5	Horizontal or vertical	Graph indicates 1000			Max diam, 56.5 ft; for boilers to 350,000 lb/hr; max gas flow, 1,070,000 lb/hr; for turndown, use optional bypass arrangement.	Bearings in oil bath; matrix in two layers with cold layer reversible. Soot blower or wash used.
Heat wheel, ceramic	2	Horizontal or vertical	"zero expansion" from 70 to 1600		FG and air		
Heat pipes	24	Horizontal or vertical	1325			6 in. to 16 in. deep; face areas to 54 by 192 in. (72 ft ²); dampers available; ex: 8000-scfm unit.	Cleaning is necessary for the unit; preferable to use filtered air.
	26	Horizontal or vertical	450			Typical face velocity, 500 fpm.	Currently not quoting for high-temperature service (above 450°F), but brochure shows 600°F.

Table 2 (continued)

Exchanger category	Vendor	Equipment type	Temperature (°F)	Pressure (psig)	Common process streams	Approx current equipment size and capacity	Comments
Shell and tube	1	Shell with two removable helical coils					Suitable for three streams; shell interior insulated.
		Removable tube bundle					For hydrocarbon gases; shell interior insulated.
		Double pipe			Good for slurry; for dirty gases in inner pipe		
	2	Rectangular shell with serpentine coils	1500	300	Tube, water; shell, dirty gas		Chains and soot blower used to clean.
	3	S&T					
	4	S&T		2050			
	6	S&T		S, 2725; T, 2150		Limits: tube sheet, 14 in. thick; channel forging: 55 in. ID, 23 tons.	Feed-effluent hydrocarbon service.
		S&T		S, 700; T, 5100			In NH ₃ service.
		S&T		S, 1210; T, 4000			
	7	S&T				Facilities: 350-ton lift; form, 4-in.-thick cold, 6-in.-thick hot; mill, 14 ft 8 in. diam; heat-treat furnace 13.75 x 15 x 54 ft to 2300°F.	
	9	S&T				Facilities: 50-ton lift; roll 4 in. thick; shell, 72 in. diam; tube sheet, 12 in. thick; heat treat furnaces; one 11 ft x 7 ft x 28 ft to 2000°F, one 4 x 10 x 30 ft to 2100°F.	
	12	S&T	1100	3000		8 ft diam.	
	15	S&T				12 ft diam, 3 in. thick.	
	16	S&T	500	150		576 ft ² .	
	17	S&T	320	75		Ex: 3700 ft ² .	
	18	S&T, coiled tube, and double pipe	2000	2000, std, 3000 special		5000 ft ² .	
	20	Helical coil					
	21	Glass S&T, bayonet, glass tube					Drainage of water is critical to prevent freezing.
Spiral plate	13	Embossed with spiral groove		150 std, higher available		5 to 2000 ft ² ; Ex: unit 42 in. diam by 60 in. long has 1000 ft ² .	For viscous materials.
		Counterflow			Gas		
		Crossflow			Liquid		
		Counterflow and crossflow			Condensing vapor		
		Overhead condenser					
Plate and frame	13	Corrugated plates clamped against gaskets	300 std, 400 special	300		Largest, 13,000 ft ² , 4.5 x 9.67 x 26 ft long.	Fouling at 10 - 30% S&T rate.
	14	Corrugated plates clamped against gaskets	450 with special gasket			Std max 5000 ft ² ; special 16,450 ft ² for 10,000 gpm has 16-in. connections; 2°F approach on one unit.	Turbulent flow possible at low Re No. (150), giving good results with viscous fluids (30,000 cP cited).
Waste heat boiler	1	For steel, glass, and chemical units; membrane walls common; auxiliary firing is option			FG vs steam; FG vs hot water; or FG vs gas	Low capacity, fire tube; high-capacity water tube	Clean with shot or with vibrators.
		Cool gas turbine exhaust			Two steam pressures are optional		
	4	Gas turbine exhaust cooling; modular construction	FG 2500, steam 950; finned tube, 1700 max	2500		Dimensions: 63 x 26 x 35 ft; up to 9 sections in enclosure including superheater and process heaters.	
	10	Modular units for inserting into hot-gas streams		Economizer at 512 psig		Can operate dry; Ex: highest throughput 49 MM Btu/hr with hot water; 25,000 lb/hr steam.	Use soot blowers or wash to clean.

Some water usually evaporates at the surface. Tubes may be made of ceramic material for cooling corrosive fluids. Equations for determining the approximate film coefficients and the corrections for mean temperature difference are available.⁴⁰

Pipe coils are often used for maintaining a desired temperature in a naturally convecting fluid. The fluid within the coil is generally circulated under pressure. The coil may be finned on the outside if fouling is not considered to be a problem.

Double-wall heat transfer plates or thermal panels are made with internal channels that can be economically embossed into the plate wall(s). The heat transfer plates are made to be attached either to the interior or exterior of a vessel wall. The use of a special "heat transfer cement" can aid in overcoming the resistance to heat flow between the plate and the vessel to which it is bonded or otherwise held.

A bayonet exchanger may be inserted into either a vessel or a shell, and it can be used to exchange relatively small heat loads. The device easily accommodates differential thermal expansion, because it is simply supported at only one end. When several elements are grouped into a unit, double tube-sheet construction is required. Each element is basically a straight open-ended inner tube extending into an outer concentric tube that is closed at one end. Flow is through the inner tube followed by a return pass through the annulus. Bayonet exchangers are used as surface condensers at low pressures because of the large diameter of the outer tube. They are also used to heat liquids (using steam) and are easily cleaned because of the simple shape and the large size of the tube.

Finned-plate surfaces are used in brazed aluminum heat exchangers (BAHX). Aluminum plates and finned components (e.g., corrugated thin sheets)

are brazed together to form compact alternate layers for the two heat-transfer fluids (often gases). Two streams are generally in crossflow through several layers. Headers distribute and collect the flows in various arrangements of passes that may be required. The success in brazing depends on many factors, some of which are not initially appreciated by the purchaser or the manufacturer. Quality control testing is necessary for critical services. BAHX types are useful, particularly in cryogenic air separation plants where aluminum is at high strength and where low approach temperatures are very desirable.

Evaporative coolers or condensers utilize two heat sinks. At full flow, a hot stream delivers heat through a tube bundle to a downward flowing mixture of falling water and induced air flow. Water is crudely separated from the air before the air is discharged upwards. Bypass arrangements permit some warm air to be recirculated to prevent freezing water during periods of low-heat load and cold weather. Evaporative coolers are said to be useful for moderate heat loads.⁴⁰

Falling film exchangers are normally conventional straight tube exchangers mounted vertically, with each tube having a weir or other liquid distribution device at the top. The resulting film of liquid inside the tube flows rapidly and is less prone to overheating or undercooling than a flow filling the tube.

Air-cooled heat exchangers have been used extensively for saving water, particularly for remote locations. Air-cooled condensers are specified for steam turbine exhausts in several proposed coal gasification plants. Special header piping is used in such condensers; the coolers are inclined in an A-frame arrangement, and condensate is drained to prevent freezing

problems. Available options include either induced- or forced-draft air flow. Induced draft fans result in higher plumes and less chance of air recirculation; however, this results in additional cost in film coefficient and some loss in convenience. Such exchangers are used for heat rejection rather than for heat recovery and are therefore not discussed further.

6. HEAT EXCHANGE THERMAL EXPERIENCE

Even in services typically cleaner than those characteristic of coal conversion processes, the difference between calculated and actual heat transfer rates can be significant. Illustrative of this difference is the summary paper by Smith,⁴¹ who tested several hundred heaters, coolers, and condensers in the Billingham petrochemical complex of Imperial Chemical Industries. Flows (always countercurrent) included pure and mixed liquids and gases at pressures of 1 to 360 atm abs. The ratio of the calculated (clean surface) to measured overall heat-transfer coefficients derived for each heat exchanger varied from 1.1 to 4.1. The deviation between calculated and observed values was attributed primarily to surface fouling (dirt deposition), and secondarily to stagnant fluid regions in the shells and to flow maldistribution among parallel tubes.

Many tube-side liquid and slurry process flows in coal conversion plant heat exchangers have been and are expected to be in the laminar or transition flow regimes. The correspondingly low heat transfer rates have been documented in a report⁷ which indicates that overall heat transfer coefficients for operating coal liquefaction plant heat exchangers have varied over the following typical ranges [units of $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$]:

Fired coal slurry preheaters: 1 to 3

Recycle gas coolers: 30 to 100

Heavy vapor condensers: 40 to 80

Reactor effluent coolers, pre-heating coal-paste feed: 10 (dirty) to 50 (clean)

Film coefficients for process streams in German coal-hydrogenation plants have been reported.⁴² For coal pastes and a velocity of 8 fps, the heat transfer coefficient (h) varied from 80 to 120 Btu/(hr)(ft²)(°F); for mixtures of middle oil and hydrogen (15 to 30 ft³ of hydrogen per pound of oil) at 6 to 15 fps, h ranged from 130 to 160 Btu/(hr)(ft²)(°F); and for heavy-oil letdown (HOLD) streams, the average h was 90 Btu/(hr)(ft²)(°F) at temperatures above 400°F and at velocities of 9 to 15 fps.

Similar data obtained for various types of heat exchange equipment in six continuous coal-tar distillation plants in England also indicate quite low heat transfer rates.⁴³ For about 30 heat exchangers (vapor-tar, oil-tar, pitch-tar, condensers, fired-heater convection tubes, and batch heaters), the range of surface heat flux was 160 to 3000 Btu/(hr)(ft²) and that of overall heat transfer coefficient was 0.4 to 26 Btu/(hr)(ft²)(°F).

Information concerning the S&T exchanger used to preheat the coal-water slurry at the Mohave generating station before it enters the centrifuges was obtained from M. I. Dina, Plant Engineer at Mohave.⁴⁴ The unit is horizontal, has two tube passes, and the slurry in the 2.465-in.-ID tubes is heated by second-point extraction steam at ~150 psia in the shell. A late 1975 test gave an overall heat-transfer coefficient of 144.4 Btu/(hr)(ft²)(°F), with a 47 wt % slurry flowing at 8.8 fps. Analysis of the data indicate the slurry film coefficient was about 15% of the value for water at the same velocity, bulk temperature (105°F), and tube diameter.

The mean Reynolds number was about 9800. Although no fouling has occurred, tube inlet erosion has been severe. Stainless steel ferrules at the tube inlets have worn out within 3 months, and hard rubber inserts were being tried.

The salient points are that fouling of various types can cause a wide divergence between calculated and operating heat transfer rates, that many streams are relatively viscous and in laminar or transition flow, and that overall heat transfer coefficients are low [$<100 \text{ Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$].

7. PROBLEM AREAS

7.1 Laminar-Flow Heat Transfer

The Reynolds moduli for liquid and slurry streams in heat exchangers used in coal liquefaction pilot plants (Bergius, CSF, and SRC processes) were estimated from available data.⁷ Except for the Cresap (CSF) fired preheaters for which the N_{Re} was apparently 10,000 to 15,000, the minimum moduli, based on maximum viscosities,^{3,42} varied from 6 to 1600, indicating stable laminar flow. Even when superimposed free convection is substantial, heat transfer coefficients are generally small for laminar flow in tubes of standard diameters.

Augmentation of laminar-flow heat transfer in tubes would obviously be desirable. Hong and Bergles⁴⁵ have reviewed past studies in which curved tubes, rotating tubes, displaced promoters, surface roughness, internal extended surfaces, and swirl flow have been used. The results of tests using twisted-tape inserts, Noranda Forge-Fin (internally finned) tubing, and both Koch and Kenics static-mixer inserts have also been described.^{45,46} Large increases in heat transfer rate were obtained with modest increases

in pressure drop with the internally finned tubing. The relative merit of the inserts on the heat transfer/pressure drop ratio followed the sequence (decreasing order) of twisted tape $\rightarrow$ Kenics mixer $\rightarrow$ Koch mixer. With the simple twisted-tape configuration, the heat transfer coefficient was easily increased by a factor of 2 to 3. The small-diameter, constant wall-heat flux tests reported in refs. 45 and 46 are being extended to larger diameters and a constant wall-temperature boundary condition; this work is being done at Heat Transfer Research, Inc. (HTRI) using viscous liquids such as ethylene glycol.

A recent application of spirally twisted cupronickel tubing in out-board keel coolers for cooling engines and auxiliary equipment on very large boats and small ships has been reported.⁴⁷ The increased surface area and turbulent helical flow combine to double the heat-transfer rate compared with that in straight tubing with normally laminar flow. An extension of the efforts to enhance laminar-flow heat transfer rates in a nonfouling fashion with heavy oils and coal-oil slurries should be worthwhile.

7.2 Physical Property Data

Relevant physical property data or methods for estimating the properties are available for pure and mixed gases and for pure light liquids.^{48,49} However, more data are needed for heavy oils, many liquid mixtures, slurries, slags, refractories, and some materials of construction.^{50,51} For heat exchanger design purposes, it is important that vapor-liquid equilibrium (VLE) data and viscosities and thermal conductivities be known with reasonable accuracy. Assuming adequate VLE data to establish the relative flows and compositions of vapor and liquid streams, the thermal

design problem is largely reduced to calculating the heat transfer coefficient. As an illustration of the relative importance of the various properties, consider the heat transfer coefficient for turbulent, single-phase tube flow, for which $\underline{h}$ is proportional to $(k^{2/3} C_p^{1/3} \rho^{0.8} / \mu^{0.467})$. Densities (ρ) and heat capacities (C_p) are usually available or can be reasonably estimated, but a three-fold error in the thermal conductivity (k) results in a two-fold error in $\underline{h}$; likewise, an error in viscosity (μ) of 4.4-fold leads to a two-fold error in $\underline{h}$.

7.3 Heat Transfer Correlations

Knowledge of heat transfer coefficients is the primary requirement in designing heat transfer equipment, and adequate correlations for predicting coefficients are available for some coal conversion streams. The need for more reliable correlations for direct contact coolers and condensers, for coal slurry heaters and slurry-to-slurry interchangers, and for surfaces immersed in fluidized-bed gasifiers and combustors has been reported.⁵¹

A study of heat transfer to and from nonflocculated coal-water slurries in turbulent tube flow has been described.⁵² All the coal was pulverized to -250 mesh (maximum particle diameter $\sim 30 \mu\text{m}$) and slurries were prepared containing 13.0 to 33.5 wt % coal. The empirical correlation, which describes the limited data within $\pm 26\%$, was based on a Bingham plastic flow model. Viscometric data, in a form not ordinarily available, are required to apply the correlation.

7.4 Fouling and Corrosion

A recent Westinghouse report⁵³ is explicit concerning the ways in which fouling, corrosion, and erosion make the present heat recovery system

design for hot coal gas difficult and uncertain. Problems addressed include tar and ash-slag fouling, residual hydrocarbon deposition, carbon laydown, hydrogen and H_2S attack, nitriding, and aqueous and chloride-salt corrosion. The report statements that "Only actual test operation can provide data for guaranteeing exchanger cleanliness," and that "It remains for actual results in a real system with coal gas to determine the severity of the problem and the efficacy of the solution" are realistic.

In general, the problems of fouling in coal conversion process exchangers are unique and are anticipated to be more severe than those experienced currently in most other industries. Such fouling is a costly and largely unresolved problem throughout refineries and some petrochemical plants. The deposits decrease heat transfer rates, increase pressure drops, and impede process flows. Provisions must be made for periodic mechanical, chemical, or hydro cleaning of such exchanger surfaces. Fouling rate data and associated velocity dependencies, if such exist, are needed for a large variety of process streams and conditions if the onstream time of exchangers is to be reasonable. Only specific testing and/or accumulated experience in pilot plants can provide the needed fouling factors, as well as the corrosion and erosion data needed for material selection (corrosion) and application of specific mechanical design techniques (erosion).

8. COMPARISON OF GENERAL AND SPECIFIC VENDOR COMMENTS

8.1 General Comments

In replying to the questionnaire, many vendors prefaced their specific ideas and recommendations for the 12 heat exchangers with similar general comments. Vendor identifier numbers are defined in Appendix A. The preliminary and tentative nature of the estimates was stressed by vendors 4,

28, 30, and 46. A lack of experience in designing exchangers for streams with a high-solids content was specifically cited by vendors 29, 36, and 37.

Major problems were perceived to be those associated with the stream solids: that is, whether char will stick, plug, and erode, and what velocities to use (No. 4); characterization of the solids (No. 28); and possible deposition, fouling, plugging, and erosion compounded by heavy-carrier liquids (No. 6).

In the area of onstream times, the responses varied from relatively low (vendor 6) to perhaps 2 years if materials selection and exchanger design are proper and there is no flow-path plugging by solids (No. 3). Vendor 9 mentioned that onstream availability is not guaranteed; an operating interval of about 1 year was estimated by vendors 4 and 5.

No vendor stated a needed development time for adequate design, but 2 years was implied by vendor 36; this company also indicated that probably no one can design properly at this time, and that data on heat transfer with high-solids content streams should be obtained by initiating experimental studies with such organizations as HTRI and Particulate Solid Research, Inc. (PSRI). Vendor 5 suggested that appropriate modifications to their products could be developed within 1 year. The acquisition of operational data in demonstration-scale units was emphasized by vendor 6, and the ability to select materials resistant to char abrasion and trace-compound corrosion was stressed by vendor 3.

Exchanger warranties uniformly follow that recommended in the TEMA Standards (e.g., vendors 3 and 37). The warranty is typically extended to workmanship and materials and (at specified conditions and with clean surfaces) to thermal performance (vendors 9 and 28) for a 1-year interval.

Vendor materials recommendations are excluded from liability (vendors 28 and 37); a warranty may be extended to 5 years at a higher cost (vendor 9), or for other periods at a rate of 2 to 3% per year (vendor 37). Warranties exclude problems associated with solids and residues in the fluids (vendors 4 and 37).

Delivery times for large exchangers are estimated by the responding vendors to vary from a minimum of about 4 months (vendor 28) to a maximum of about 30 months (vendor 36), depending on market activity and the availability of materials such as nickel and chromium alloys. The average of the various fast and slow delivery times stated in the responses (vendors 4, 5, 6, 9, 28, 30, 36, 37, and 38) is about 1 year.

Other general comments in the responses, most given as overall recommendations, included the following:

1. Shell- and U-tube-bundle exchangers would probably be the predominant type (vendor 3).
2. Higher allowable pressure drops than some of those used in the evaluation reports would be desirable (vendors 28 and 30).
3. Unit weight limits in several U.S. shops extend to the 150 to 200-ton range (vendor 6).
4. The design should reflect the expected cleaning procedure (vendor 6).
5. Solids-laden streams should be routed in downflow through large-diameter vertical tubes; minimum and maximum velocity limits of such streams should be established; and use of sacrificial erosion shields and of onstream cleaning and maintenance should be considered (vendor 6).
6. A capability exists for gun drilling tube sheets up to 42 in. thick (vendor 28).

7. A transient (startup-shutdown) stress and fatigue analysis of units operating at high temperatures and pressures is desirable and can be done (vendor 28).
8. The optimal number of shells is best determined by an economic balance involving the costs, mechanical properties, and environmental resistances of several materials of construction, especially when high-pressure fluids are on the shell side (vendor 28).

8.2 Specific Comments

The specific comments and recommendations made by the vendors concerning the 12 potentially critical heat exchangers in the questionnaire are summarized below. Appendix D gives the heat load for each of the exchangers and briefly outlines the size and type of each unit as specified in the evaluation studies and as estimated by the responding manufacturers-vendors. The specification sheets, which include the flow rates, are included as Appendix C.

Item A1: As specified in a study of a Fischer-Tropsch complex, a vertical S&T exchanger in the Gas Cleaning Unit would cool 444,000 lb per hour of raw syngas and char at 1700°F and 483 psia while superheating 500-psig steam on the shell side. Vendor 9 recommended that heat recovery be accomplished by generating steam rather than superheating it, and pointed out that the variable heat transfer rate of dry superheated steam can lead to overheating the metal at these temperatures. Other vendors agreed that internal refractory linings on the tube-side channels are necessary. Vendors 4 and 28 saw merit in providing two shells rather than one. Vendor 38 proposed using two trains, each composed of two single-pass units in series. Vendors 6 and 28 suggested Incoloy as the tube material; vendor 38

suggested 1 1/4 Cr - 1/2 Mo. Vendor 28 proposed a horizontal unit with double-tube-sheet construction, permitting the generation of steam between the tube sheets to keep the primary load-bearing tube sheets relatively cool. For the same general purpose, insulating ferrules would be provided by vendor 28 to carry gas back of the shell-side tube sheet, which would minimize tube-sheet temperature gradients. Vendor 6 proposed sacrificial erosion shields to protect the tube sheets from the solids-laden gas. The variations in recommended heat transfer areas are not unusual (the ratio of maximum-to-minimum areas was 2.54) considering the variability in mechanical designs and in probable levels of vendor estimation efforts.

Item B1: This 1200-psig steam generator, as in Item A1 above, cools 1700°F raw syngas and char from a Fischer-Tropsch gasifier, but it has a larger flow rate, (~696,000 lb/hr). As originally specified, Item B1 was also a vertical S&T unit with water and steam on the shell side. Evaluators and vendors would expect the inlet channel to be refractory lined, and some suggest that both channels be lined. Vendor 28 proposed double-tube-sheet construction, with insulation on the inlet-tube-side sheet. Vendors 4, 28, and 38 recommended consideration of two or more shells. Vendors 6 and 28 proposed horizontal units, and vendors 4 and 6 proposed external steam drums. The proposals vary considerably in estimated surface area (the ratio of maximum to minimum surface areas was 3.47).

Item C1: The third example from the Fischer-Tropsch complex is a 1200-psig steam superheater, again heated by raw syngas and char at 1700°F. The gas-char stream flow rate is 2,791,000 lb/hr. The evaluation study called for an S&T vertical exchanger. The sizes proposed by two vendors bracketed the surface area selected in the original study. Vendor 28

proposed the use of ferrules and a cooled double-tube-sheet design in an effort to use thinner tube sheets, which would protect the load-bearing members from excessive temperatures. Incoloy tubes were suggested by vendors 6 and 28. Vendor 38 proposed 1 1/4 Cr - 1/2 Mo for both shells and tubes. Vendor 6 recommended a shell made of 304H stainless steel. Vendors 28 and 38 recommended a one-pass unit because of the temperature discontinuity across pass partitions in multipass units with high tube-side inlet temperatures. Vendor 38 proposed divided flow on the shell side. The ratio of maximum-to-minimum estimated surface areas for this unit was 1.8.

Item D2: Specifications for the feed-effluent exchangers proposed for the Synthoil reactors in the evaluation study were based on limited data obtained at PERC and elsewhere in bench-scale and pilot plant testing. Vendors responding to the questionnaire were probably not as aware of these data and of secondary operational requirements. The original study called for 32 units, each handling ~416,000 lb per hour of a three-phase feed mixture at 4280 psia on the shell side of an S&T exchanger. The number of shells suggested by vendors were 4 (vendor 4), 8 (vendor 38), 10 (vendors 6 and 28), and 20 (vendor 9). Vendor 38 proposed a U-tube bundle; vendor 28 suggested a special U-tube bundle with inlet flow centered into the bends with a return gas flow on the periphery because of the tube-side temperature differential between the inlet and outlet. Vendor 6 proposed that the fluids be reversed and that a horizontal U-tube arrangement of 304 stainless steel construction be used.

Vendors 6, 9, and 28 all suggest a differential or balanced pressure design for the tube sheets to minimize their thickness. An all-welded construction was also suggested by vendors 9 and 28. During World War II,

the Germans developed a tapered-ring gasket which was effective in such exchangers. Vendors 9 and 38 suggested a 304 stainless steel overlay over rolled plate for the shell and chromium-molybdenum forgings for the channels. The ratio of maximum-to-minimum surface areas estimated for Item D2 was 4.9.

Item E2: The gasification unit for a Synthoil complex has been designed for heat recovery using a hot gas and solids stream at 900°F and 254 psia to generate 350 psig steam in two S&T waste-heat boilers. Approximately 3,572,000 lb per hour of hot gas and solids are cooled on the shell sides of the two boilers in the original study. Vendor 30 suggested four vertically oriented S&T fixed tube-sheet units. Vendors 6 and 28 suggested one unit, vendor 38 suggested two, and vendor 9 proposed ten individual shells. Vendor 30 proposed all carbon steel construction. Vendor 38 proposed 304 stainless steel clad 1 1/4 Cr - 1/2 Mo material. Generally, other vendors favor carbon steel-molybdenum alloys (CS-Mo) for the gas-side materials. The larger surface areas proposed by vendors probably result from the low-pressure drop (2 psi) specified for the gas side. Vendors 4, 9, 28, and 38 all responded with U-tube designs. Vendor 38 suggested double divided shell flow. The ratio of the maximum to minimum estimated surface areas for this exchanger was 5.6.

Item F2: In the Synthoil evaluation study, further cooling of the gas and solid mixture described for Item E2 is accomplished by generating 313,800 lb per hour of 50 psig steam in two S&T waste-heat boilers. Vendors 6 and 30 recommended carbon steel construction and reversed flows (gas on the tube side). Vendors 9 and 28 favor CS-Mo for the gas-side materials. Vendor 38 proposed 304 stainless steel clad 1 1/4 Cr - 1/2 Mo for the same service. Vendor 4 selected two shells for the job; vendors 6 and 28 each

selected one shell; vendor 38 chose three shells; vendor 9 favored ten transfer-line exchangers. The total areas proposed by each vendor are tabulated in Appendix D. The ratio of maximum-to-minimum surface areas for this unit was 5.4.

Item G2: The same gas and solid mixture flowing in exchangers E2 and F2 is finally cooled in 32 air-cooled heat exchangers in the Synthoil evaluation study. Only two vendors responded directly for the proposed service. Vendor 5 proposed renewal of earlier efforts to develop a heat wheel for elevated pressures such as are required; a compact unit should result. Vendor 6 favored using five modular tube bundles to be inserted into ductwork provided by the purchaser. The tubes would be vertical, of G-fin construction, with tubes welded to the tube sheets. The fluids would be reversed compared to their routing in the evaluation study. The vendors made no mention of expected pressure drops on either side. The surface area estimated by vendor 6 is only one-eighth that in the evaluation, which is probably too small.

Item H3: A gas-to-gas heat exchanger was proposed to exchange heat between recycle gas streams in a recent hydrocarbonization process evaluation study. Both streams are at a pressure of ~185 psia, and the two streams enter at 40°F and 120°F. The evaluation study specified 18 shells with cold gas on the tube side. Vendors responded with smaller area S&T heat exchangers; vendors 6 and 28 chose one shell, vendor 30 chose four shells, vendor 38 chose six, and vendor 9 chose ten. Vendor 9 suggested a floating-head arrangement to permit bundle removal. Vendor 28 suggested CS-Mo construction; others (vendors 6 and 9) chose carbon steel. Vendor 5 proposed a compact finned-plate recuperator in 8 modules, occupying a volume

of 88 x 17 x 8 ft. Construction would be Cor-Ten (A-242) or A-36 steel. The ratio of maximum-to-minimum estimated surface areas for this item was 5.1.

Item I3: The evaluation study for the hydrocarbonization complex specifies three S&T exchangers to cool primary flash vapor while heating a mixture of oil and hydrogen feed for a tar hydrotreating unit. The flash vapor at 2490 psia and 750°F is cooled by feed that contains about 13,000 lb per hour of solids and 484,000 lb per hour of liquid and gas. The two streams are at essentially the same pressure. Vendors, under no restraints as to shell size, proposed one shell for this service. Vendor 28 proposed molybdenum alloy construction as did vendor 38; except for the 304 stainless steel clad shell; vendor 9 proposed 2 1/4 Cr - 1 Mo, and vendor 6 suggested low-alloy materials. Vendors 6 and 9 favored all-welded construction. Vendor 9 proposed a design differential pressure of 300 psi between the S&T fluids. Vendor 5 suggested development of modifications to their low-pressure finned-plate recuperator. The ratio of maximum-to-minimum estimated surface areas for this unit was 4.9.

Item J3: Eight S&T exchangers were originally proposed for a fluidized-bed combustion system interchanging heat between pretreater off-gas (cooled) and recycle gas (heated). The units handle 3,764,000 lb per hour of pretreater gas with solids at 348°F on the tube side. Vendor responses included designs with fixed-tube sheets (vendors 6, 30, and 38), optional use of floating heads (vendors 9 and 28) and U-tubes (vendor 28). Vendor 28 proposed molybdenum alloy construction for the shell; other vendors (6, 9, and 38) proposed carbon-steel construction throughout. The estimated number of shells for this service were 1 (vendors 4 and 28), 2 (vendors 6 and 38),

4 (vendor 30), and 20 (vendor 9). Vendor 5 proposed using eight modules of their steel (A-242 or A-36) finned-plate recuperators, which, if combined, would occupy a volume of 88 x 17 x 8 ft. The ratio of maximum-to-minimum estimated surface areas was 4.7.

Item K3: The fluidized-bed combustion system proposed in the hydro-carbonization evaluation study utilized 18 shells to exchange heat between pretreater gas (cooled) and dryer heating air (heated). Approximately 3,110,000 lb of gas per hour is cooled from 1236°F to 1000°F. Vendor 6 favored a single 15,000 ft² S&T horizontal unit with insulated carbon steel channels, 304 H tubes, and a low-alloy steel shell. Vendor 2 portrayed two vertical S&T units (10 ft diam and 51 ft high). Bellows would be provided in the upper and lower flow-transition sections. Vendor 38 proposed ten S&T units made of 1 1/4 Cr - 1/2 Mo steel. Vendor 9 suggested that ceramic channel linings, heat shields, tube ferrules, and inlet nozzle liners would be required to ensure safe metal temperatures. Vendor 28 would pursue this item with a subsidiary if so desired. Vendors 9 and 38 would like to study this unit further. The ratio of maximum-to-minimum surface areas was 2.4.

Item L3: Six conventional rotary air preheaters were originally proposed for transferring heat from 720°F flue gases to combustion air. Vendors were unrestricted as to the maximum flow rate per shell and therefore as to the number of shells proposed. The widespread results were as follows: 1 shell for vendor 2, 10 for vendor 38, and 200 for vendor 9. Vendor 5 proposed two of their rotary air preheaters to handle the entire vertical air flow (9,307,000 lb/hr). Vendor 2 proposed a box structure containing ten vertical tube bundles of rectangular cross section with

flue gas flowing upwards through the tubes; air would flow in a two-pass arrangement. The elevated unit is large (53 ft high x 24 ft deep x 130 ft long). Vendor 9 suggested the use of 100 modules or trains in parallel, each module or train comprised of two S&T exchangers in series. Two-inch diameter tubes of CS-1/2 Mo material were proposed. Vendor 38 proposed ten S&T units, also with CS - 1/2 Mo tubes. The ratio of maximum-to-minimum estimated surface areas was 4.5.

Area Comparisons

The preceding ratios of maximum-to-minimum estimated surface areas for the individual exchangers are summarized in Fig. 16.

Fig. 16. Summary of heat exchanger surface areas.

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APPENDIXES

Appendix A: Catalogs, Brochures, and Letters Received

<u>Vendor and Contact</u>	<u>Description</u>
1. The American Schack Co., Inc. 9730 Rinaman Road (P.O. Box 11006) Wexford, PA 14090 Contact: D. R. Hirshberg Chief Engineer (412) 935-5725	Recuperators, S&T, heat wheel, waste heat boilers
2. Thermal Transfer Division of Kleinewefers 1100 Rico Road Monroeville, PA 15146 Contact: Bernard W. Savko Asst. Mgr. Project Engineering (412) 351-3013	Recuperators, gas coolers, heat wheel
3. Hughes-Anderson Engineering Corp. 1001 North Fulton Tulsa, OK 74115 (918) 836-1681	S&T
4. Struthers Wells Corp. P.O. Box 8 Warren, PA 16365 Contact: Edward D. Burns, Jr. Mgr. Special Products Dept. (814) 726 1000	S&T, waste heat boilers
5. C-E Air Preheater Combustion Engineering, Inc. P.O. Box 372 Wellsville, NY 94895 Contact: Eugene D. Krumm, Mgr. of Marketing Development and Dr. Charles K. Leeper, Vice- President of Engineering (716) 593-2700	Heat wheel

<u>Vendor and Contact</u>	<u>Description</u>
<p>6. C-E Lummus HTD (Heat Transfer Division) of Lummus The Lummus Company International Headquarters 1515 Broad Street Bloomfield, NJ 07003</p> <p>Contact: W. Schoonman Vice Pres. & Gen. Mgr. Heat Transfer Div.</p> <p>(201) 893-1515</p>	S&T
<p>7. Nooter Corporation P.O. Box 451 (1400 S. Third Street) St. Louis, MO 63166</p> <p>Contact: William Roth Sales Engineer</p> <p>(314) 621-6000</p>	S&T
<p>8. Fallon Engineering Co. 8040 San Fernando Road Sun Valley, CA 91352 (213) 768-3900</p>	General bulletin
<p>9. Engineers & Fabricators Co., Inc. (EFCO) P.O. Box 7395 (3501 W. 11th Street) Houston, TX 77008</p> <p>Contact: E. P. Hansen President</p> <p>(713) 869-3461</p>	S&T
<p>10. Sunrod Products, Inc. 437 Madison Avenue, 27th Floor New York, NY 10022</p> <p>Contact: Gunnar Jakobson President</p> <p>(212) 754-0887</p>	Waste heat boilers

<u>Vendor and Contact</u>	<u>Description</u>
11. Holcroft & Co. 12068 Market Street Livonia, Michigan 48150 (313) 261-8410	General bulletin
12. Bos-Hatten, Inc. c/o Maveric Co. 5652 Savoy Houston, TX 77036 (713) 784-3220 Bos-Hatten, Inc. French and Old Union Roads Buffalo, NY 14224 (716) 668-8111	S&T, finned tube
13. American Heat Reclaiming Corp. 1270 Avenue of the Americas New York, NY 10020 (212) 489-1900	Plate, spiral
14. APV Co., Inc. 395 Fillmore Avenue Tonawanda, NY 14150 (716) 692-3000	Plate
15. Gaston County Dyeing Machine Co. P.O. Box 308 Stanley, NC 28614 Contact: Mahendra Chudgar Project Engineer (704) 827-6751	S&T
16. American Standard Power and Controls Group 3801 Mars Avenue Birmingham, AL 35215 (205) 681-7984	S&T
17. Perry Products Corp. 25 Mount Laurel Road Hainesport, NJ 08036 (609) 267-1603	S&T

<u>Vendor and Contact</u>	<u>Description</u>
18. Ametek Schutte & Koerting Division Cornwells Heights, PA 19020 (215) 639-0900	S&T, double pipe
19. The Trane Co. LaCrosse, WI 54601 (608) 782-8000	Cryogenic, compact surface
20. Graham Manufacturing Co., Inc. 170 Great Neck Road Great Neck, NY 11021 (800) 645-3757, (516) 482-5200	Spiral, tubular
21. Corning Process Systems Corning Glass Works Corning, NY 14830 (607) 974-9000	S&T, bayonet, glass elements
22. Henry P. Thompson Co. 4866 Cooper Road Cincinnati, OH 45242 (513) 891-4242	Recuperators (tubular)
23. Union Carbide Corporation Linde Division P.O. Box 44 Tonawanda, NY 14150 (716) 877-1600	Special heat transfer surface
24. Q-dot Corporation 151 Regal Row, Suite 220 Dallas, TX 75247 (214) 630-1224	Heat pipe
25. Lear Siegler, Inc. Environmental Technology Division 74 Inverness Drive East Englewood, CO 80110 Contact: Victor M. Day National Sales Coordinator (303) 770-3300	Plate type recuperator, filters

<u>Vendor and Contact</u>	<u>Description</u>
26. Hughes-Electron Dynamics Division 3100 W. Lomita Boulevard Torrance, CA 90509 Contact: R. C. Prager (213) 534-2121	Heat pipe
27. Niagara Blower Co. Executive Offices 405 Lexington Avenue New York, NY 10017 (212) 697-6151	Water sprayed unit
28. Wyatt Industries, Inc. P.O. Box 5418 Dallas, TX 75222 (214) 747-8741 Sales Office: 6200 Kansas Street Houston, TX 77001 Contact: J. M. Walton (713) 861-6141	General, S&T, kilns
29. Riley-Beaird, Inc. P.O. Box 31115 Shreveport, LA 71130 Contact: W. E. Adams (318) 868-4441	General, S&T, kiln
30. Henry Vogt Machine Co. P.O. Box 1918 Louisville, KY 40201 Contact: Wilmer Reynolds, Jr. Assistant General Manager Heat Transfer Division (502) 634-1511	S&T, scraped surface
31. Wheco Division Therma Technology, Inc. P.O. Box 190 Channelview, TX 77530 (713) 452-5984	Fired heaters, waste heat boilers

	<u>Vendor and Contact</u>	<u>Description</u>
32.	Futura Titanium Corp. P.O. Box 5004 Westlake Village, CA 91359 (800) 423-5181, (213) 889-2420	S&T, coils
33.	Thermo-Electron Corp. Energy Systems 101 First Avenue Waltham, MA 02154 Contact: Thomas Speidel Applications Manager Energy Systems Group (617) 890-8700	Heat recovery/power generation system (using steam)
34.	Kinetics Corp. 1121 Lewis Avenue Sarasota, FL 33577 (813) 366-3050 46 Woodcrest Drive Scotia, NY 12302 Contact: James J. Zimmerman Vice President Marketing (518) 377-0214	Heat recovery/power generation system (using Freon)
35.	Babcock & Wilcox 161 East 42nd Street New York, NY 10017 (212) 687-6700	Rothemuhle air preheater
36.	Thermex, Inc. P.O. Box 7365 Houston, TX 77008 Contact: Anthony Docherty, President (713) 869-4821	S&T

<u>Vendor and Contact</u>	<u>Description</u>
37. Richmond Engineering Company, Inc. P.O. Box 25189 Richmond, VA 23260 Contact: Warren W. Hall, Manager of Heat Exchanger Engineering (804) 644-2611	S&T
38. Industrial Fabricating Co. Word Industries, Inc. 3159 S. Sheridan Rd. Tulsa, OK 74145 Contact: R. A. Frazier, Vice President Sales and Engineering (918) 272-2261	S&T

Appendix B: Vendor Contacts

Early in the study, one of the authors attended a seminar on energy management at which several manufacturers of heat exchangers (and other heat recovery equipment) were represented. Discussions regarding problems related to coal conversion and equipment are summarized in a trip-report letter that is reproduced in Appendix F of this study. Several of the attendees were members of Heat Transfer Research, Inc. (HTRI), an organization which conducts technical work sponsored by its members.

The following vendors of special equipment items were contacted by telephone:

Solids Coolers

Sandvik Conveyor, Inc.
J. Michael Vaccaro
Southern Regional Manager
Atlanta, Georgia
(404) 955-2999

Fuller Company
Catasaqua, Pennsylvania
Mr. W. W. Kramer, Chief Engineer - Fluidized-Bed Equipment
(215) 264-6731
Mr. Reuben J. Aldrich, Sales Manager - Fluidized-Bed Equipment
(215) 264-6506

Exchangers recently ordered for H-Coal Pilot Plant

Therma Technology, Inc.
Attn: B. Farnsworth
Senior Thermal Design Engineer - Brown Fintube
Tulsa, Oklahoma
(918) 584-2621

(See copy of summary letter in Appendix E.)

A questionnaire concerning 12 severe-duty heat exchangers was sent to each of the 23 vendors listed below. Substantive responses were received from those firms identified by asterisks.

*Yuba Heat Transfer Corporation

P. O. Box 3158
Tulsa, Oklahoma 74101
Attn: A. J. Cunningham, Sales Manager - Process Exchangers

*Wyatt Industries, Inc.

U. S. Industries Company
P. O. Box 5418
Dallas, Texas 75222
Attn: John T. Thorngren, Process Engineering

*Industrial Fabricating Company

Word Industries, Inc.
3159 S. Sheridan Road
Tulsa, Oklahoma 74145

Hunt Tool Company

1945 Cullen
Houston, Texas 77023

Delta Southern Company, Inc.

1435 Choctaw Dr.
Baton Rouge, Louisiana 70805

*Thermex, Inc.

P. O. Box 7365
Houston, Texas 77008
Attn: Anthony Docherty, President

*Riley-Beaird, Inc.

A Subsidiary of The Riley Company
P. O. Box 1115
Shreveport, Louisiana 71130
Attn: W. E. Adamo, President and General Manager

*Engineers and Fabricators, Co.

A Marley Company
3501 W. 11th Street
Houston, Texas 77008
Attn: Edward P. Hansen, President & Gen. Mgr.

*Western Supply Division

Therma Technology, Inc.
P. O. Box 2739
Tulsa, Oklahoma 74101
Attn: H. F. Koons, Vice President

*Richmond Engineering Company, Inc.

P. O. Box 25189
Richmond, Virginia 23260
Attn: Warren W. Hall, Manager of Heat Exchanger Engineering

*Thermal Transfer

Div. of Kleinewefers

1100 Rico Road

Monroeville, Pennsylvania 15146

Attn: Bernard W. Savko, Assistant Manager - Project Engineering

The American Schack Company, Inc.

P. O. Box 11006

Pittsburgh, Pennsylvania 15237

Attn: Mr. D. R. Hirshberg, Chief Engineer

Southwestern Engineering Company

Tyler Corporation

P. O. Box 54940

Los Angeles, California 90054

Attn: Tom C. Arnett, President

*Struthers Wells Corporation

P. O. Box 8

Warren, Pennsylvania 16365

Attn: Edward D. Burns, Jr., Manager - Special Products Dept.

Nooter Corporation

P. O. Box 451

St. Louis, Missouri 63166

Attn: Mr. Gene R. Smith, General Sales Mgr., Shop Fabrication

*C-E Lummus

The Lummus Company

Heat Transfer Division

1515 Broad Street

Bloomfield, N. J. 07003

Attn: W. Schoonman, Vice President and General Manager

*C-E Air Preheater

Combustion Engineering, Inc.

P. O. Box 372

Wellsville, N. Y. 14895

Attn: Chief Executive Officer

American Heat Reclaiming Corporation

1270 Avenue of the Americas

New York, N. Y. 10020

Attn: Chief Executive Officer

Fallon Engineering Company

National Headquarters

8040 San Fernando Road

Sun Valley, California 91352

Attn: Chief Executive Officer

*Hughes-Anderson Engineering Corporation
1001 North Fulton
Tulsa, Oklahoma 74115
Attn: E. D. Anderson, Executive Vice President

Sunrod Products Incorporated
437 Madison Avenue (27th Floor)
New York, N. Y. 10022
Attn: Mr. Gunnar Jakobson, President

*The Garrett Corporation
Attn: Willis E. Sullivan
Manager Program Development
Government Relations
1625 Eye Street, N.W.
Washington, D. C. 20006

*Henry Vogt Machine Company
Attn: Wilmer Reynolds, Jr., Assistant General Manager
Manager Heat Transfer Division
P. O. Box 1918
Louisville, Kentucky 40201

Appendix C: List of Severe-Service Heat Exchangers
for Coal Conversion Processes and
Questionnaire Specification Sheets

A tabulation of the 12 heat exchangers selected from three coal-conversion evaluation reports follows. Process specification sheets for these units were sent to vendor companies for their comments, estimates of required surface areas, and materials recommendations.

Item No.	Evaluation report ref.	Description	Original recommended No. of shells
1. A1	FE-1775-7 13-1601-01	500-psig steam superheater (for heat recovery in gas cleaning unit)	1
2. B1	FE-1775-7 13-1601-02	1200-psig saturated steam generator (for heat recovery in gas cleaning unit)	1
3. C1	FE-1775-7 13-1601-04	1200-psig steam superheater (for heat recovery in gas cleaning unit)	3
4. D2	ORNL-5209 12-1603	Reactor feed-effluent exchanger (for Synthoil unit)	32
5. E2	ORNL-5209 18-1601	350-psig steam generator (for gasification unit)	2
6. F2	ORNL-5209 18-1602	50-psig steam generator (for gasification unit)	2
7. G2	ORNL-5209 18-1603	Raw gas-air cooler (for gasification unit)	32
8. H3	ORNL-5212 12-1605	Recycle gas precoolers (for hydrocarbonization unit)	18
9. I3	ORNL-5212 16-1601	Feed pretreater (for tar hydrotreating unit)	3
10. J3	ORNL-5212 26-1601	Recycle gas preheater I (for fluidized-bed combustion system)	8
11. K3	ORNL-5212 26-1604	Flow dryer air preheater (for fluidized-bed combustion system)	18
12. L3	ORNL-5212 26-1605	Combustion air preheater (for fluidized-bed combustion system)	6

ITEM A1 - 500 psig Steam SuperheaterTUBE SIDE - Raw syngas and charHEAT LOAD - 148.5 MM Btu/hr P_1 483 psia P_2 482 psia T_1 1700 °F T_2 950 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	11,019
CO	28.01	180,524
CO ₂	44.01	66,411
H ₂ S	34.08	4,976
SH	33.07	
NH ₃	17.02	17
H ₂ O	18.02	32,094
N ₂	28.01	3,109
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	180
SO ₂	64.06	
C1	16.04	4,491
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		141,250
ASH	200	
COAL (MAF)	800	
TOTALS		302,821 Gas 141,250 Char (solid) 444,071 All

NOTES:

Fluid is syngas having MW of 19.2.

At inlet, solids loading is approximately 0.125 lb/ACFM.

At outlet, solids loading is approximately 0.20 lb/ACFM.

ITEM	A1	-	500 psig Steam Superheater	
SHELL SIDE	-	Steam		
HEAT LOAD	-	148.5	MM Btu/hr	
P ₁	510	psia	P ₂	485 psia
T ₁	670	°F	T ₂	1050 °F
FLUID COMPONENT			MW	FLOW RATE (lb/hr)
H ₂			2.02	
CO			28.01	
CO ₂			44.01	
H ₂ S			34.08	
SH			33.07	
NH ₃			17.07	
H ₂ O			18.02	715,826
N ₂			28.01	
Ar			39.95	
N ₂ + Ar			30.81	
O ₂			32	
COS			60.07	
SO ₂			64.06	
C1			16.04	
C2			30.07	
C3			44.1	
n-C4			58.12	
C5-400			105	
400-650			190	
650-850			320	
850-1050			440	
1050 +			600	
NAPHTHA			142.5	
LIGHT GAS OIL			205.5	
HEAVY GAS OIL			291.3	
RESIDUE			390	
RESIDUE			800	
CHAR				
ASH			200	
COAL (MAF)			800	
TOTALS				715,826

NOTES:

ITEM B1 - 1200 psig Saturated Steam Generator

TUBE SIDE - Raw syngas and char

HEAT LOAD - 231.7 MM Btu/hr

 P_1 483 psia P_2 482 psia T_1 1700 °F T_2 950 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	17,392
CO	28.01	282,761
CO ₂	44.01	103,996
H ₂ S	34.08	7,770
SH	33.07	
NH ₃	17.02	51
H ₂ O	18.02	50,294
N ₂	28.01	4,846
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	300 ^a
SO ₂	64.06	
C1	16.04	7,042
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		221,250
ASH	200	
COAL (MAF)	800	
TOTALS		474,452 Gas
		221,250 Char
NOTES:		695,702 All

Fluid is syngas having MW of 19.2.

At inlet, solids loading is approximately 0.125 lb/ACFM.

At outlet, solids loading is approximately 0.20 lb/ACFM.

^a300 was omitted on original specification

ITEM B1 - 1200 psig Saturated Steam Generator

SHELL SIDE - Steam

HEAT LOAD - 231.7 MM Btu/hr

P₁ 1215 psia

P₂ 1215 psia

T₁ 513 °F

T₂ 569 °F

FLUID
COMPONENT

MW

FLOW RATE
(lb/hr)

H ₂	2.02
CO	28.01
CO ₂	44.01
H ₂ S	34.08
SH	33.07
NH ₃	17.02
H ₂ O	18.02
N ₂	28.01
Ar	39.95
N ₂ + Ar	30.81
O ₂	32
COS	60.07
SO ₂	64.06
C1	16.04
C2	30.07
C3	44.1
n-C4	58.12
C5-400	105
400-650	190
650-850	320
850-1050	440
1050 +	600
NAPHTHA	142.5
LIGHT GAS OIL	205.5
HEAVY GAS OIL	291.3
RESIDUE	390
RESIDUE	800
CHAR	
ASH	200
COAL (MAF)	800

TOTALS

336,590	Steam out
10,410	Blowdown out
347,000	A11

NOTES:

ITEM C1 - 1200 psig Steam Superheater

TUBE SIDE - Raw syngas and char

HEAT LOAD - 931.8 MM Btu/hr

P_1 483 psia P_2 482 psia

T_1 1700 °F T_2 950 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	69,763
CO	28.01	1,134,293
CO ₂	44.01	417,259
H ₂ S	34.08	31,149
SH	33.07	
NH ₃	17.02	221
H ₂ O	18.02	201,716
N ₂	28.01	19,439
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	1,141
SO ₂	64.06	192
C1	16.04	28,262
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		887,500
ASH	200	
COAL (MAF)	800	
TOTALS		1,903,435 Gas
		887,500 Char
		2,790,935 All

NOTES:

Fluid is syngas having MW of 19.2.

At inlet, solids loading is approximately 0.125 lb/ACFM.

At outlet, solids loading is approximately 0.20 lb/ACFM.

ITEM	C1	-	1200 psig Steam Superheater
SHELL SIDE	-	Steam	
HEAT LOAD	-	931.8	MM Btu/hr
P ₁	1215	psia	P ₂ 1210 psia
T ₁	630	°F	T ₂ 950 °F
FLUID COMPONENT		MW	FLOW RATE (lb/hr)
H ₂		2.02	
CO		28.01	
CO ₂		44.01	
H ₂ S		34.08	
SH		33.07	
NH ₃		17.02	
H ₂ O		18.02	4,379,723
N ₂		28.01	
Ar		39.95	
N ₂ + Ar		30.81	
O ₂		32	
COS		60.07	
SO ₂		64.06	
C1		16.04	
C2		30.07	
C3		44.1	
n-C ₄		58.12	
C5-400		105	
400-650		190	
650-850		320	
850-1050		440	
1050 +		600	
NAPHTHA		142.5	
LIGHT GAS OIL		205.5	
HEAVY GAS OIL		291.3	
RESIDUE		390	
RESIDUE		800	
CHAR			
ASH		200	
COAL (MAF)		800	
TOTALS			4,379,723

NOTES:

ITEM D2 - Reactor Feed/Effluent Exchanger

TUBE SIDE - Reactor Effluent

HEAT LOAD - 2919.5 MM Btu/hr

P₁ 4280 psiaP₂ 4235 psiaT₁ 860 °FT₂ 627 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	1,508,424
CO	28.01	
CO ₂	44.01	
H ₂ S	34.08	115,040
SH	33.07	
NH ₃	17.02	15,194
H ₂ O	18.02	301,577
N ₂	28.01	
Ar	39.95	
N ₂ + Ar	30.81	687,688
O ₂	32	
COS	60.07	
SO ₂	64.06	
C1	16.04	2,443,880
C2	30.07	496,278
C3	44.1	330,380
n-C4	58.12	183,839
C5-400	105	114,566
400-650	190	290,187
650-850	320	967,904
850-1050	440	1,694,756
1050 +	600	3,219,360
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	376,400
CHAR		
ASH	200	556,540
COAL (MAF)	800	
TOTALS		13,299,014

NOTES:

ITEM	D2	-	Reactor Feed/Effluent Exchanger
SHELL SIDE	-	Reactor Feed	
HEAT LOAD	-	2919.5	MM Btu/hr
P ₁	4265	psia	P ₂ 4250 psia
T ₁	353	°F	T ₂ 623 °F
FLUID COMPONENT	MW	FLOW RATE (lb/hr)	
H ₂	2.02	1,611,897	
CO	28.01	7,115	
CO ₂	44.01	2,245	
H ₂ S	34.08	2,324	
SH	33.07		
NH ₃	17.02	5	
H ₂ O	18.02	111,526	
N ₂	28.01		
Ar	39.95		
N ₂ + Ar	30.81	687,688	
O ₂	32		
COS	60.07		
SO ₂	64.06		
C1	16.04	2,368,574	
C2	30.07	453,218	
C3	44.1	291,876	
n-C4	58.12	163,579	
C5-400	105	89,229	
400-650	190	110,352	
650-850	320	728,544	
850-1050	440	1,333,332	
1050 +	600	2,538,000	
NAPHTHA	142.5		
LIGHT GAS OIL	205.5		
HEAVY GAS OIL	291.3		
RESIDUE	390		
RESIDUE	800	234,640	
CHAR			
ASH	200	556,540	
COAL (MAF)	800	2,008,640	
TOTALS		13,299,324	

NOTES:

Feed is a slurry (S.G. = 1.15) of 35 wt % coal and 65 wt % recycle oil along with a gas (80 vol % H₂) which is added to the slurry at an approximate rate of 52 scf gas/lb of slurry.

ITEM E2 - 350 psig Steam GeneratorTUBE SIDE - SteamHEAT LOAD - 500.8 MM Btu/hr P_1 ~365 psia P_2 ~365 psia T_1 436 °F T_2 436 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	
CO	28.01	
CO ₂	44.01	
H ₂ S	34.08	
SH	33.07	
NH ₃	17.02	
H ₂ O	18.02	633,900
N ₂	28.01	
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	
SO ₂	64.06	
C1	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		
ASH	200	
COAL (MAF)	800	
TOTALS		633,900

NOTES:

ITEM E2 - 350 psig Steam Generator

SHELL SIDE - Scrubbed Gas (with solids)

HEAT LOAD - 500.8 MM Btu/hr

P₁ 254 psia P₂ 252 psia

T₁ 900 °F T₂ 550 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	47,190
CO	28.01	1,061,083
CO ₂	44.01	689,844
H ₂ S	34.08	73,534
SH	33.07	27
NH ₃	17.02	22
H ₂ O	18.02	1,675,109
N ₂	28.01	13,016
Ar	39.95	5,673
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	6,896 ^a
SO ₂	64.06	
C1	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C3-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CARBON	12.01	663
ASH	200	38,260
COAL (MAF)	800	
TOTALS		3,572,394 Gas
		<u>38,923 Solids</u>
NOTES:		3,611,317 All

^a6896 was omitted on original specification.

ITEM F2 - 50 psig Steam Generator

TUBE SIDE - Steam

HEAT LOAD - 286.1 MM Btu/hr

P₁ ~65 psia P₂ ~65 psia

T₁ 298 °F T₂ 298 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	
CO	28.01	
CO ₂	44.01	
H ₂ S	34.08	
SH	33.07	
NH ₃	17.02	
H ₂ O	18.02	313,800
N ₂	28.01	
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	
SO ₂	64.06	
C1	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		
ASH	200	
COAL (MAF)	800	
TOTALS		313,800

NOTES:

ITEM F2 - 50 psig Steam Generator

SHELL SIDE - Scrubbed Gas (with solids)

HEAT LOAD - 286.1 MM Btu/hr

P₁ 252 psia P₂ 250 psia

T₁ 550 °F T₂ 350 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	47,190
CO	28.01	1,061,078
CO ₂	44.01	689,844
H ₂ S	34.08	73,534
CH ₄	33.07	26
NH ₃	17.02	22
H ₂ O	18.02	1,675,109
N ₂	28.01	13,016
Ar	39.95	5,673
N ₂ + Ar	30.81	
O ₂	32	^a
COS	60.07	6,896
SO ₂	64.06	
C1	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CARBON	12.01	663
ASH	200	38,260
COAL (MAF)	800	
TOTALS		3,572,394 Gas
		38,923 Solids
		<u>3,611,317 All</u>

NOTES:

^a42 lb/hr was deleted from original specification.

ITEM G2 - Raw Gas/Air Cooler

TUBE SIDE - Raw Gas (with solids)

HEAT LOAD - 1385.3 MM Btu/hr

P₁ 249 psia P₂ 246 psia

T₁ 350 °F T₂ 270 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	47,190
CO	28.01	1,061,078
CO ₂	44.01	689,844
H ₂ S	34.08	73,534
SH	33.07	26
NH ₃	17.02	22
H ₂ O	18.02	1,675,109
N ₂	28.01	13,016
Ar	39.95	5,673
N ₂ + Ar	30.81	
O ₂	32	^a
COS	60.07	6,896
SO ₂	64.06	
Cl	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CARBON	12.01	663
ASH	200	38,260
COAL (MAF)	800	
TOTALS		3,572,388 Gas
		38,923 Solid
		<u>3,611,311 All</u>

NOTES:

^a42 lb/hr was deleted from original specification.

ITEM	G2	-	Raw Gas/Air Cooler
SHELL SIDE	a	-	Atmospheric Air ^b
HEAT LOAD	-	1385.3	MM Btu/hr
P ₁	AMB	psia	P ₂ AMB psia
T ₁	95	°F	T ₂ 200 °F
FLUID COMPONENT		MW	FLOW RATE (lb/hr)
H ₂		2.02	
CO		28.01	
CO ₂		44.01	
H ₂ S		34.08	
SH		33.07	
NH ₃		17.02	
H ₂ O		18.02	
N ₂		28.01	
Ar		39.95	
N ₂ + Ar		30.81	
O ₂		32	
COS		60.07	
SO ₂		64.06	
Cl		16.04	
C2		30.07	
C3		44.1	
n-C4		58.12	
C5-400		105	
400-650		190	
650-850		320	
850-1050		440	
1050 +		600	
NAPHTHA		142.5	
LIGHT GAS OIL		205.5	
HEAVY GAS OIL		291.3	
RESIDUE		390	
RESIDUE		800	
CHAR			
ASH		200	
COAL (MAF)		800	
TOTALS			54,972,222

NOTES:

^a Unit can be heat wheel, etc., rather than air-cooled finned-tube unit;
T₂ is variable at vendor-evaluator's discretion.

^b Design wet bulb = 76°F; Min temp = -15°F; Max wind = 50 mph.

ITEM	H3	-	Recycle Gas Precooler
TUBE SIDE		-	Recycle Gas (to be heated)
HEAT LOAD	115.6		MM Btu/hr
P ₁	183	psia	P ₂ 175 psia
T ₁	40	°F	T ₂ 79.4 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	199,293
CO	28.01	255,195
CO ₂	44.01	106,795
H ₂ S	34.08	9,223
SH	33.07	
NH ₃	17.02	
H ₂ O	18.02	4,058
N ₂	28.01	
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	
SO ₂	64.06	
C1	16.04	2,732,953
C2	30.07	155,503
C3	44.1	15,148
n-C4	58.12	11,806
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA ^a	142.5	5,459
LIGHT GAS OIL ^b	205.5	13
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		
ASH	200	
COAL (MAF)	800	
TOTALS		3,495,445

NOTES:

^aBoiling range = 32 to 450°F.^bBoiling range = 450 to 650°F.

ITEM	H3	-	Recycle Gas Precooler
SHELL SIDE	-	-	Recycle Gas (to be cooled)
HEAT LOAD	-	115.6	MM Btu/hr
P ₁	180	psia	P ₂ 175 psia
T ₁	120	°F	T ₂ 80 °F
FLUID COMPONENT	MW	FLOW RATE (lb/hr)	
H ₂	2.02	199,355	
CO	28.01	255,195	
CO ₂	44.01	106,813	
H ₂ S	34.08	9,223	
SH	33.07		
NH ₃	17.02		
H ₂ O	18.02	75,103	
N ₂	28.01		
Ar	39.95		
N ₂ + Ar	30.81		
O ₂	32		
COS	60.07		
SO ₂	64.06		
C1	16.04	2,733,090	
C2	30.07	155,540	
C3	44.1	15,111	
n-C4	58.12	11,840	
C5-400	105		
400-650	190		
650-850	320		
850-1050	440		
1050 +	600		
NAPHTHA ^a	142.5	19,587	
LIGHT GAS OIL ^b	205.5	2,403	
HEAVY GAS OIL	291.3		
RESIDUE	390		
RESIDUE	800		
CHAR			
ASH	200		
COAL (MAF)	800		
TOTALS		3,583,259	

NOTES:

^aBoiling range = 32 to 450°F.^bBoiling range = 450 to 650°F.

ITEM	T3	-	Feed Preheater
TUBE SIDE	-	-	Feed
HEAT LOAD	-	75.85	MM Btu/hr
P ₁	2500	psia	P ₂ ~2500 psia
T ₁	295	°F	T ₂ 518 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	40,623
CO	28.01	138
CO ₂	44.01	8
H ₂ S	34.08	3
SH	33.07	
NH ₃	17.02	2
H ₂ O	18.02	56
N ₂	28.01	
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	
SO ₂	64.06	
C1	16.04	51,023
C2	30.07	6,903
C3	44.1	1,334
n-C4	58.12	1,620
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA ^a	142.5	40,280
LIGHT GAS OIL ^b	205.5	64,227
HEAVY GAS OIL ^c	291.3	115,772
RESIDUE ^d	390	161,540
RESIDUE	800	
CHAR		
ASH	200	
COAL (MAF)	800	
TOTALS		483,529 Liquid & Gas 13,000 Solids 496,529 All

NOTES:

Fluid is a mixture of: 20,430 lb/hr gas with MW 2.51; 81,100 lb/hr gas with MW 5.25; and 395,000 lb/hr liquid and solids, S.G. 0.871.

^aB.R. = 32 to 450°F.

^bB.R. = 450 to 650°F.

^cB.R. = 650 to 900°F

^dB.R. = 900 + °F.

ITEM	I3	-	Feed Preheater
SHELL SIDE	-	-	Primary Flash Vapor
HEAT LOAD	-	75.85	MM Btu/hr
P ₁	~2490	psia	P ₂ ~2490 psia
T ₁	750	°F	T ₂ 585 °T

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	51,783
CO	28.01	7
CO ₂	44.01	15
H ₂ S	34.08	9,649
SH	33.07	
NH ₃	17.02	2,462
H ₂ O	18.02	25,339
N ₂	28.01	
Ar	39.95	
N ₂ + Ar	30.81	
O ₂	32	
COS	60.07	
SO ₂	64.06	
C1	16.04	98,267
C2	30.07	15,293
C3	44.1	3,293
n-C4	58.12	4,995
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA ^a	142.5	58,393
LIGHT GAS OIL ^b	205.5	79,274
HEAVY GAS OIL ^c	291.3	68,997
RESIDUE ^d	390	20,288
RESIDUE	800	
CHAR		
ASH	200	
COAL (MAF)	800	
TOTALS		438,055

NOTES:

a-d

Same as on preceding page.

ITEM J3 - Recycle Gas Preheater I

TUBE SIDE - Preheater Gas (with solids)

HEAT LOAD - 393.03 MM Btu/hr

P_1 120 psia P_2 113 psia

T_1 348 °F T_2 220 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	
CO	28.01	
CO ₂	44.01	1,837
H ₂ S	34.08	
SH	33.07	
NH ₃	17.02	
H ₂ O	18.02	416,415
N ₂	28.01	3,043,438
Ar	39.95	52,353
N ₂ + Ar	30.81	
O ₂	32	246,591
COS	60.07	
SO ₂	64.06	
Cl	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		
ASH	200	68
COAL (MAF)	800	2,928
TOTALS		3,760,634 Gas 2,996 Solids 3,763,630 All

NOTES:

ITEM	J3	-	Recycle Gas Preheater I	
SHELL SIDE	-		Total Recycle Gas	
HEAT LOAD	-	393.03	MM Btu/hr	
P ₁	170	psia	P ₂	168 psia
T ₁	120	°F	T ₂	220 °F
FLUID COMPONENT		MW	FLOW RATE (lb/hr)	
H ₂		2.02	117,657	
CO		28.01	150,659	
CO ₂		44.01	16,818	
H ₂ S		34.08	290	
SH		33.07		
NH ₃		17.02		
H ₂ O		18.02	30,604	
N ₂		28.01		
Ar		39.95		
N ₂ + Ar		30.81		
O ₂		32		
COS		60.07		
SO ₂		64.06		
C1		16.04	1,613,448	
C2		30.07	91,803	
C3		44.1	8,943	
n-C4		58.12	6,970	
C5-400		105		
400-650		190		
650-850		320		
850-1050		440		
1050 +		600		
NAPHTHA		142.5	3,223 ^a	
LIGHT GAS OIL		205.5	92 ^a	
HEAVY GAS OIL		291.3		
RESIDUE		390		
RESIDUE		800		
CHAR				
ASH		200		
COAL (MAF)		800		
TOTALS			2,040,507	

NOTES:

^a Original specification included typographical errors showing 3,223 for C5-400 and 92 for 400-650.

ITEM K3 - Flow Dryer Air Preheater

TUBE SIDE - Pretreater Gas

HEAT LOAD - 201.64 MM Btu/hr

P_1 158 psia P_2 150 psia

T_1 1236 °F T_2 1000 °F

<u>FLUID COMPONENT</u>	<u>MW</u>	<u>FLOW RATE (lb/hr)</u>
H ₂	2.02	
CO	28.01	
CO ₂	44.01	1,638
H ₂ S	34.08	
SH	33.07	
NH ₃	17.02	
H ₂ O	18.02	45,623
N ₂	28.01	2,714,272
Ar	39.95	46,619
N ₂ + Ar	30.81	
O ₂	32	301,575
COS	60.07	
SO ₂	64.06	
C1	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		
ASH	200	
COAL (MAF)	800	
TOTALS		3,109,727

NOTES:

ITEM	K3	-	Flow Dryer Air Preheater	
SHELL SIDE	-	Air		
HEAT LOAD	-	201.64	MM Btu/hr	
P ₁	26	psia	P ₂	24 psia
T ₁	550	°F	T ₂	848 °F ^a
FLUID COMPONENT		MW	FLOW RATE (lb/hr)	
H ₂		2.02		
CO		28.01		
CO ₂		44.01		
H ₂ S		34.08		
SH		33.07		
NH ₃		17.02		
H ₂ O		18.02		
N ₂		28.01	2,294,360 ^b	
Ar		39.95		
N ₂ + Ar		30.81		
O ₂		32	700,640	
COS		60.07		
SO ₂		64.06		
C1		16.04		
C2		30.07		
C3		44.1		
n-C4		58.12		
C5-400		105		
400-650		190		
650-850		320		
850-1050		440		
1050 +		600		
NAPHTHA		142.5		
LIGHT GAS OIL		205.5		
HEAVY GAS OIL		291.3		
RESIDUE		390		
RESIDUE		800		
CHAR				
ASH		200		
COAL (MAF)		800		
TOTALS			2,995,000	

NOTES:

^aOriginal specification included typographical error showing 2,294,360 for H₂O instead of N₂.

^bOriginal specification showed 900°F for T₂.

ITEM L3 - Combustion Air Preheater

TUBE SIDE - Flue Gas

HEAT LOAD - 1074 MM Btu/hr

P_1 _____ psia P_2 _____ psia (near atmospheric)

T_1 720 °F T_2 295 °F

FLUID COMPONENT	MW	FLOW RATE (lb/hr)
H ₂	2.02	
CO	28.01	
CO ₂	44.01	1,941,074
H ₂ S	34.08	
SH	33.07	
NH ₃	17.02	
H ₂ O	18.02	605,280
N ₂	28.01	7,381,219
Ar	39.95	128,613
N ₂ + Ar	30.81	
O ₂	32	683,322
COS	60.07	
SO ₂	64.06	6,116
Cl	16.04	
C2	30.07	
C3	44.1	
n-C4	58.12	
C5-400	105	
400-650	190	
650-850	320	
850-1050	440	
1050 +	600	
NAPHTHA	142.5	
LIGHT GAS OIL	205.5	
HEAVY GAS OIL	291.3	
RESIDUE	390	
RESIDUE	800	
CHAR		
ASH	200	523
COAL (MAF)	800	48
Miscellaneous solids		200
TOTALS		10,745,624 Gas
		771 Solids
		10,746,395 All ^a

NOTES:

Mass flows shown for heat wheel case; T_2 is corrected to reflect decrease resulting from air leakage into gas stream.

^a10,746,395 includes 1,179,700 lb/hr of air leakage into flue gas stream inside preheater.

ITEM	L3	-	Combustion Air Preheater
SHELL SIDE	-		Combustion Air
HEAT LOAD	-	1074	MM Btu/hr
P ₁	psia	(Atmospheric)	P ₂ psia
T ₁	105	°F	T ₂ 647 °F
FLUID COMPONENT	MW	FLOW RATE (lb/hr)	
H ₂	2.02		
CO	28.01		
CO ₂	44.01	3,816	
H ₂ S	34.08		
SH	33.07		
NH ₃	17.02		
H ₂ O	18.02	131,041	
N ₂	28.01	6,927,578	
Ar	39.95	120,898	
N ₂ + Ar	30.81		
O ₂	32	2,123,551	
COS	60.07		
SO ₂	64.06		
C1	16.04		
C2	30.07		
C3	44.1		
n-C4	58.12		
C5-400	105		
400-650	190		
650-850	320		
850-1050	440		
1050 +	600		
NAPHTHA	142.5		
LIGHT GAS OIL	205.5		
HEAVY GAS OIL	291.3		
RESIDUE	390		
RESIDUE	800		
CHAR			
ASH	200		
COAL (MAF)	800		
TOTALS		9,306,884 ^a	

NOTES:

Mass flows shown for heat wheel case.

^a 9,306,884 includes 1,179,700 lb/hr of air which is lost to the flue gas inside the preheater.

Appendix D: Summary of Responses to Questionnaire
for Severe-Service Heat Recovery Equipment

Summary of responses to questionnaire for severe-service heat recovery equipment

Questionnaire equipment identifier	Service	Heat load (Btu/hr)	Source of estimate (vendor No.)	Equipment type ^a	Flow arrangement	Heat transfer area per shell (ft ²)	No. Shells	Total area (ft ²)
A1	Cool raw syngas and char, and superheat 500 psig steam	148.5 x 10 ⁶	b	S&T, vertical	Syngas-tube Steam-shell	12,600	1	12,600
			4	S&T	Same	N/A	2	N/A
			6	S&T, vertical, FTS	Same	15,000	1	15,000
			9	Recommended steam generation				
			28	S&T, horizontal, double tube sheet at inlet	Syngas-tube Steam-shell	5,900	1	5,900
			38	S&T, floating head	Same	2,170	2 trains of 2 each	8,680
B1	Cool raw syngas and char, and generate 1200 psig steam	231.7 x 10 ⁶	b	S&T, vertical, WHB	Syngas-tube Steam-shell	3,650	1	3,650
			4	S&T, WHB, with drum	Same	4,200	2	8,400
			6	S&T, horizontal, with drum	Same	5,000	1	5,000
			9	S&T, FTS, 1-pass shell	Same	12,500	1	12,500
			28	S&T, horizontal, double tube sheet at inlet	Same	3,600	1	3,600
			38	S&T, floating head	Same	2,000	2	4,000
C1	Cool raw syngas and char, and superheat 1200 psig steam	931.8 x 10 ⁶	b	S&T, vertical	Syngas-tube Steam-shell	15,270	3	45,700
			4	S&T	Same	N/A	10-12	N/A
			6	S&T, FTS, vertical	Same	21,670	3	65,000
			9	Recommended steam generation				
			28	S&T, horizontal, double tube sheet on inlet	Same	6,170	6	37,000
			38	S&T, floating head	Same	11,400	4	45,600

Questionnaire equipment identifier	Service	Heat load (3tu/hr)	Source of estimate (vendor No.)	Equipment type ^a	Flow arrangement	Heat transfer area per shell (ft ²)	No. Shells	Total area (ft ²)
D2	Reactor feed/effluent exchanger	2919.5 x 10 ⁶	c	S&T	Effluent-tube Feed-shell	7,350	32	235,200
			4	S&T	Same	16,000	4	64,000
			6	S&T, U-T	Effluent-shell Feed-tube	11,500	10	115,000
			9	S&T, floating head	Same	15,700	20	314,000
			28	S&T, U-T	Same	8,000	10	80,000
			38	S&T, U-T	Effluent-tube Feed-shell	8,300	8	66,400
E2	Cool gas and generate 350 psig steam	500.8 x 10 ⁶	c	S&T, WHB with drum	Steam-tube Gas-shell	6,700	2	13,400
			4	S&T, U-T with common drum	Same	25,000	2	50,000
			6	S&T, FTS, horizontal	Gas-tube Steam-shell	30,000	1	30,000
			9	S&T, transfer line exchanger	Gas-tube Steam-shell	7,440	10	74,400
			28	S&T, U-T	Steam-tube Gas-shell	24,000	1	24,000
			30	S&T, FTS, vertical	Gas-tube Steam-shell	8,810	4	35,240
F2	Cool scrubbed gas and generate 50 psig steam	286.1 x 10 ⁶	c	S&T, WHB, with drum	Gas-shell Steam-tube	8,065	2	16,130
			4	S&T, U-T, with common drum	Same	28,500	2	57,000
			6	S&T, FTS, vertical	Steam-shell Gas-tube	30,000	1	30,000
			9	S&T, FTS	Gas-shell Steam-tube	8,710	10	87,100
			28	S&T, U-T	Gas-shell Steam-tube	27,500	1	27,500
			30	S&T, FTS vertical	Steam-shell Gas: tube, downward	10,410	4	41,640
			38	S&T, U-T	Gas-shell Steam-tube	12,000	3	36,000

Questionnaire equipment identifier	Service	Heat Load (Btu/hr)	Source of estimate (vendor No.)	Equipment type ^a	Flow arrangement	Heat transfer area per shell (ft ²)	No. Shells	Total area (ft ²)
G2	Cool raw gas with air	1385.3 x 10 ⁶	c	Air cooler; 64 fans	Gas: tube Air: outside	53,444	32	1,710,208
			5	Heat wheel, develop- ment to be required	N/A	N/A	N/A	N/A
			6	Tubes in duct	Gas: tube Air: duct	43,000	1-unit (5 assemblies)	215,000
H3	Cool recycle gas and heat recycle gas	115.6 x 10 ⁶	d	S&T	Cold gas-tube Hot gas-shell	8,440	18	151,926
			5	Compact, Finned plate unit	Counter-current	N/A	8 modules	N/A size 88' x 17' x 8'
			6	S&T, FTS, vertical or horizontal	Cold Gas-Tube Hot Gas-shell	40,000	1	40,000
			9	S&T, floating head	Same	8,180	10	81,800
			28	S&T, FTS	Same	30,000	1	30,000
			30	S&T, FTS, horizontal	Same	10,610	4	42,440
I3	Cool primary flash vapor and heat mixture of oil and hydrogen feed	75.85 x 10 ⁶	d	S&T	Feed-tube Vapor shell	2,778	3	8,335
			4	S&T	Same	1,700	1	1,700
			5	Compact, Finned plate unit, develop- ment required	Counter-current	N/A	N/A	N/A
			6	S&T, U-T, horizontal	Feed-tube Vapor-shell	5,000	1	5,000
			9	S&T, floating head	Same	3,900	1	3,900
			28	S&T, U-T or floating head, 2-pass	Same	1,800	1	1,800
			38	S&T, U-T	Same	3,080	1	3,080

Questionnaire equipment identifier	Service	Heat load (Btu/hr)	Source of estimate (vendor No.)	Equipment type ^a	Flow arrangement	Heat transfer area per shell (ft ²)	No. Shells	Total area (ft ²)
J3	Cool preheater off-gas and heat recycle gas	393.03 x 10 ⁶	d	S&T	Preheater off-gas with solids-tube, recycle gas-shell	4,251	8	34,008
			4	S&T	Same	25,000	1	25,000
			5	Compact finned plate unit	Counter-current	N/A	8 modules	N/A Size 88' x 17' x 8'
			6	S&T, FTS, vertical	Preheater gas- tube, recycle gas-shell	35,000	2	70,000
			9	S&T, floating head	Same	5,820	20	116,400
			28	S&T, U-T, or floating head	Same	42,000	1	42,000
			30	S&T, FTS, horizontal; pressure drop over spec.	Same	15,350	4	61,400
			38	S&T, FTS	Same	18,900	2	37,800
K3	Coal pretreater gas (for coal) and heat dryer heating air (for coal)	201.64 x 10 ⁶	d	S&T	Pretreater gas- tube, air-shell	2,036	18	36,648
			2	S&T, FTS, vertical	Same	10,820	2	21,635
			5	Compact finned plate unit, requires development	Counter-current	N/A	N/A	N/A
			6	S&T, FTS horizontal	Pretreater gas- tube, air-shell	15,000	1	15,000
			9	Special study would be required				
			38	S&T, U-T	Same	3,860	10	38,600

Questionnaire equipment identifier	Service	Heat load (Btu/hr)	Source of estimate (vendor No.)	Equipment type ^a	Flow arrangement	Heat transfer area per shell (ft ²)	No. Shells	Total area (ft ²)
L3	Cool flue gases and preheat combustion air	1.07 x 10 ⁶	d	Rotating-plate type heat exchanger	Horizontal disc, counterflow	274,000	6	1,644,000
			2	Rectangular shell, with 10 vertically mounted rectangular tube bundles; cross flow	Flue gas-tube, air shell	654,500	1	654,500
			5	Rotating plate type exchanger	Horizontal disc (vertical shaft)	N/A	2	N/A
			9	S&T, FTS	Flue gas-tube air shell, 100 trains, two units per train	14,820	200	2,964,000
			38	S&T, U-T	Flue gas-tube, air-shell; double divided flow on shell	75,900	10	759,000

^a S&T = shell and tube; FTS = fixed tube sheet; WHB = waste-heat boiler; U-T = U-tube bundle.

^b See ref. 2 in this report.

^c See ref. 3 in this report.

^d See ref. 4 in this report.

Appendix E: Correspondence

1. Report of Houston Trip, March 1977
2. Letter Concerning H-Coal Pilot Plant
Heat Exchangers, April 1977

Summary of Information on Waste Heat Recovery and
Process Heat Exchangers Obtained During Houston
Trip, March 19-24, 1977 (Issued May 5, 1977)

W. R. Gambill

During the subject visit, I participated in an A.I.Ch.E. Today course titled "Process Design for Energy Conservation", attended the 83rd National A.I.Ch.E. Meeting and the 9th Petrochemical & Refining Exposition, and met with several individuals conversant with the current state of the process heat transfer field.

- I. Energy Conservation Course: The two-day course was conducted by Mr. D. E. Steinmeyer and Dr. H. R. Null of Monsanto's Corporate Engineering Dept. (St. Louis). They did an excellent job within the time available. A significant portion of the course material related to heat exchangers, primarily in terms of the economically optimal pressure drop (ΔP_o) and approach temperature difference (ΔT_o).

ΔP_o : A guideline highly regarded by Steinmeyer is that ΔP_o corresponds to an annualized ratio of (Power Costs/Surface Costs) $\approx 1/3$. This result, which is based on a derivation by S. K. Jenssen published in 1969, should apply well to nonfouling fluids in turbulent flow in large exchangers. Restated, $\sim 1/4$ the total costs of an exchanger operation should be power charges. ΔP_o (or V_o) depends on the fluid density and viscosity, but not the specific heat and thermal conductivity. An extended treatment indicates that the power-to-surface cost ratio is ~ 0.40 tubeside and ~ 0.27 shellside — i.e., G_o (optimal mass velocity) is lower on the shellside. In general, ΔP_o is proportional to the number of heat transfer units [$\Delta P_o \propto (NTU)_{ht.} = \frac{UA}{WC} = \frac{T_1 - T_2}{\Delta T_m}$], and the level of ΔP_o increases with fouling and with construction-material price. The economic optimum is broad — e.g., for $\Delta P = \frac{\Delta P_o}{2}$ or $2 \Delta P_o$, the yearly total costs exceed minimum costs by $\sim 20\%$. Associated rules of thumb include the following: V_o for a clean, relatively nonviscous liquid (tubeside) is ~ 6 fps in a carbon steel HX (heat exchanger) and somewhat higher in HXs fabricated

from more expensive materials; a good design ΔP for liquid/liquid HXs is 1 - 10 psi per transfer unit (often ~ 5 psi total); and ΔP_o for low-pressure gases is lower, falling in the range of 0.03 to 0.3 psi per transfer unit. The general conclusion was that power costs rising relative to capital costs will lead to lower velocities and slightly larger HXs. For a tripled power cost, e.g., and other factors constant, the optimal HX area for sensible transfer increases $\sim 31\%$.

ΔT_o : Four cases were considered in some detail: no sensible heat transfer (SHT), 1 fluid SHT, 2 fluids SHT and balanced (i.e., identical temperature ranges), and 2 fluids SHT and unbalanced. Approximate results based on a 95% capacity factor, $1/(R_{\text{metal}} + R_{\text{fouling}}) = 695 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, a utility h of $1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, an annual incremental surface cost of $\$1.80/\text{ft}^2\text{-yr}$ ($\$5.40/\text{ft}^2$), and saving a utility valued at $\$1/10^6 \text{ Btu}$ are given below.

<u>Fluid</u>	ΔT_o ($^\circ\text{F}$)	
	<u>1 Fluid Sensible</u>	<u>Balanced Sensible Case</u>
Water	1	$(T_1 - T_2)_{\text{max.}}^{1/2}$
Air (1 atm)	30	$8 (T_1 - T_2)_{\text{max.}}^{1/2}$
Oil (5 csk.)	2	$2 (T_1 - T_2)_{\text{max.}}^{1/2}$

The quantity $(T_1 - T_2)_{\text{max.}}$ is the temperature difference between incoming streams. As indicated, ΔT_o is smaller for liquids than for gases. Fluid properties effect ΔT_o by their influence on the U value. For a HX with $A = A_o/2$, the total cost is $\sim 20\%$ higher than the minimum cost; an exchanger with $A = 2 A_o$ is slightly more uneconomic.

Other Conclusions: Waste heat boilers will be used increasingly for heat recovery, at the expense of air coolers. Generated steam will be used for preheating feedwater, reboiling, and electricity generation. At some temperature below 300°F , waste heat recovery per se gives no economic return (exit flue gas temperatures in

current operations in the range of 200 to 275°F were mentioned by 3 - 4 participants). Often overlooked, but noted during the course, is the fourfold increase in the time value of effective engineering applied to reduction of energy consumption.

- II. AIChE Meeting: Of a dozen papers I heard presented, two were especially germane. Kasper, et al., of Dravo Corp., in their paper titled "Heat Transfer Problems in Coal Conversion", correctly pointed out the serious present lack of physical property data for materials of construction, liquid products, slurries, slags, and refractories; of heat-transfer correlations for laminar and multiphase flows and for direct contact exchangers; and of fouling rate data. Ruhe of Foster Wheeler Canada (Ontario), in a paper titled "Steam and Water Heat Recovery Methods," outlined typical equipment and process arrangements applicable to flue gas heat recovery. Many of the units are cooling flue gas to ~ 300°F, but one has been operating for 5 years with a 140°F exit temperature and cold-end condensation. Another unit, a waste heat boiler apparently mounted above a K-T gasifier, cools a "very dusty" gas stream at inlet conditions of 2700°F and 600 fps (about 1/2 the acoustic velocity).
- III. Heat Exchanger Panel: On the night of March 22, the Houston Process Heat Exchanger Society held a meeting and panel discussion focused on "Heat Exchangers and Energy Conservation in 1977". The panel moderator was Frank Rubin of Davy Powergas; the panelists were K. Beyer (Exxon USA), E. H. Mergens (Shell Oil), F. Moore (Texaco), C. W. Conklin (Sun Tech Inc.), D. E. Steinmeyer (Monsanto), and J. R. Styslinger (Pullman-Kellogg). Capsule comments follow:
- Beyer — Computerized HX monitoring has been implemented and has led to optimized cleaning periods about half those of prior practice. More paralleled HXs are being used, as is more thermal insulation for shells and heads. Hydro and chemical cleaning are displacing mechanical methods.
- Mergens — Efforts are being made to supply petrochemical plant heat requirements with heat rejected from adjacent refineries and to

apply direct-contact transfer line coolers to units other than ethylene furnaces.

Moore — To maintain heat transfer rates, low concentrations of fouling and corrosion inhibitors and neutralizing agents are being used and investigated. Control of fouling is being achieved in some units by using higher velocities shellside (smaller baffle pitch) and tubeside (more passes).

Conklin — Baffles unresponsive to acoustic excitation are being developed and used. More finned tubing in the convection sections of fired heaters is anticipated. Low fouling rates have been experienced with titanium tubing (some finned) in foul water service.

Steinmeyer — A recapitulation of a portion of the HX material from the energy conservation course.

Styslinger — Heat recovery in feed/effluent exchangers being installed is as high as 95% maximum. For liquids, 5 - 10°F approaches to the heat sink are becoming common; and optimal pressure drops are declining, especially for vapor streams, leading generally to somewhat higher justifiable HX costs than 5 to 10 years ago. There is a trend to larger shells (20,000 ft² and up) and to longer tubes.

General Discussion — One instance was noted of cooling flue gas to 140 - 150°F and recovering the latent heat of condensation, corrosive (H₂SO₄) condensation being managed with a cold-end section of glass tubing. Heat pipes and rotary regenerators (heat wheels) are apparently seeing increasing use for preheating combustion air with flue gas. The Exxon refinery at Baytown has effected a saving of 400×10^6 Btu/hr (5%) with very little capital expenditure. Some research on coal-fired process heaters is underway in the Ohio Valley, funded jointly by ERDA and industry.

- IV. Petro Expo: I visited ~ 40 booths and obtained brochures; most of the wares displayed were fairly standard, but the interesting rod baffle exchanger (RBE) developed by Phillips Petroleum and licensed to several companies is apparently gaining acceptance. Units to 5 ft diameter and 60 ft length have been installed over the past

five years; virtues include a shell-side ΔP about half that of a standard segmentally baffled exchanger, reduced shellside fouling, near-zero tube bundle vibration, and a ratio of heat transfer to ΔP , as measured by $(\bar{U}/\Delta P_{\text{shell}})$, about 1.8 times that of double segmental baffled HXs.

V. Individual Discussions:

- a) J. T. Taborek — Jerry is Technical Director of Heat Transfer Research, Inc. (HTRI), which is now supported by ~ 130 member companies. He discussed three studies they are conducting. Tests are continuing on laminar swirl-flow heat transfer (at $Re = 10 - 200$ with viscous liquids in large tubes operated at a constant wall temperature; heat transfer coefficients are at least doubled. Highly instrumented tests with banks of 6-in. diameter heated finned tubes in a subsonic wind tunnel (normal air flow) have shown that with an inline array, the bypass flow becomes very high when the gap between the fin tips exceeds $\sim 1/4$ in.; and for equal heat transfer and pressure drop, only half the number of tubes is required for triangular pitch compared with square pitch. The data generally show a greater advantage for triangular arrays than predictions based on traditional tube-bank correlations. HTRI has studied fouling quantitatively only with cooling-tower water, sea water, and (to a limited degree) crude oil. With foul water, they have found Ti tubes excellent at higher velocities (8-10 fps), but poor at low velocities of ~ 1 fps, in which range there is little turbulent scrubbing of the surface to compensate for the absence of biocidal ions.
- b) A. Docherty — Tony is president of Thermex, Inc. (Houston), which makes shell-and-tube units. He thinks that all of the HXs in the ORNL and Parsons Co. draft evaluation reports for the Synthoil, Hydrocarbonization, and Fischer-Tropsch processes are within present industrial capability. No material life warranties are made, and guarantees are only for thermal and mechanical performance based on the process and property data

provided and materials specified. He recommended eleven U.S. fabricators of large, severe-service exchangers for further views concerning industrial exchanger capability. He cited direct experience with units with up to 40,000 ft²/shell, 28 tube passes, a tube length of 80 ft, and shells to 12 ft diameter by 38 ft long.

- c) E. P. Hansen — Ed is president of Engineers and Fabricators Co. (EFCO) in Houston. I visited their facility and found it large and active. Along with Struthers Wells Corp. (Warren, Pa.) and Industrial Fabricating Co. (Tulsa), EFCO is one of the "big three" in the industry. It has fabricated shells to 13.5 ft diameter. Hansen also expects that many more waste heat boilers will be ordered. He said that all companies use TEMA standards for mechanical design; and during a brief discussion of costs, noted that for shell-and-tube HXs, the unit (per sq ft) cost decreases with increasing shell L/D ratio, and that the cost exponent, ~ 0.6 for small shells, increases to 1.0 at a shell diameter of ~ 5 ft.

I discussed tube vibration problems with Hansen and with C. J. Hammack, EFCO's chief thermal designer. They feel the problem is largely solved, though some experimental and correlative work is continuing. Mechanisms mentioned included buffeting from vortex shedding, acoustical resonance, and fluid elastic whirling. The design ratio of exciting to natural frequencies (or the inverse) is 2:1 minimum and usually 3:1; Combustion Engineering sometimes requires a ratio larger than 3:1. Hammack also thought the HXs listed in the evaluation reports (see Vb) are within present industrial fabrication capabilities.

Many data, including HTRI's, indicate that in a segmentally baffled shell-and-tube HX, the maximum shellside critical Re is 50 (range = 5 - 50). Restated, $(Re)_{c, shell}^{max} = \frac{D_o G_m}{\mu_b} = 50$, where D_o is tube OD, G_m is mean mass velocity based on the net center-plane free flow area, and μ_b is bulk fluid viscosity. In the past, this value was often taken to be ~ 300 .

VI. Summation: The multi-function visit to Houston was quite productive in terms of gathering relevant information and viewpoints.

H-Coal Pilot Plant Heat Exchangers
(Issued April 26, 1977)

W. R. Gambill

As noted in a recent issue of the OGJ (April 18, 1977), three divisions of Therma Technology, Inc. (Tulsa) have received contracts totaling \$750,000 to fabricate heat exchangers for the H-coal liquefaction pilot plant to be built in Kentucky. The Western Supply Division will provide shell-and-tube exchangers, Happy Division air-cooled units, and Brown Fintube Co. double-pipe hairpin exchangers.

I spoke this morning with B. Farnsworth, Senior Thermal Design Engineer for Brown Fintube. My inquiry was related to the survey of industrial capability for waste heat recovery equipment in coal conversion plants.

Mechanical designs will be done by Therma Tech, using appropriate ASME and TEMA codes. They will also provide the thermal designs based on information obtained from HRI, the engineering contractor. The materials of construction, all ASME qualified, have been specified by the process group, as have all relevant physical properties. Guarantees made by Therma Tech will be for workmanship, materials as specified, mechanical design, and thermal design based on the fluid properties and other data provided them. In accord with industry practice, no warranty is made regarding in-service materials integrity, corrosion, or erosion. In some cases, guarantees of no tube-vibration failure at the design point are made.

Brown Fintube is working on liquid coolers and heaters and process fluid interchangers. The maximum solids concentration in liquid streams is ~ 25 wt %. Many of the viscosities provided are high and uncertain, and most heat-exchanger process flows will be in the laminar or transition regimes.

Since a number of heat exchanger companies are located in the Tulsa area, a trip there should be worthwhile. The subject concerns, other than Therma Technology, include Escoa Fintube Corp., Essex Inc., Heat/Fluid Engineering Corp., Hughes-Anderson Engineering Corp., Industrial Fabricating

Company, Con-Rad Division of U. S. Industries, Inc., Waner Mfg. Corp., Yuba Heat Transfer Corp., and Zurn Industries, Inc. The companies whose products, scale of operation, and experience are most relevant to future coal conversion exchangers are probably Therma Technology, Industrial Fabricating, Con-Rad, Yuba, and Zurn.

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