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State-of-the-Art Review of Heat-Recovery Processes

W. R. Gambill
W. R. Reed

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CHEMICAL TECHNOLOGY DIVISION

STATE-OF-THE-ART REVIEW OF HEAT-RECOVERY PROCESSES

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

*Engineering Division

Date Published: August 1978

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FOREWORD

This report describes a state-of-the-art review of processes and associated equipment for heat recovery in coal utilization applications. The review was conducted for and supported by the Major Facility Project Management (MFPM) Division (T. B. Simpson, Project Manager) of the Energy Research and Development Administration (ERDA).*

The authors express their appreciation to those in the process heat-exchanger industry and to those users of heat-recovery equipment who shared their views with us.

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

STATE-OF-THE-ART REVIEW OF HEAT-RECOVERY PROCESSES

W. R. Gambill and W. R. Reed

ABSTRACT

A review was conducted of industrial heat-recovery processes and related equipment of potential benefit to coal-conversion complexes. Major categories of heat-exchange processes and equipment investigated included special shell-and-tube; periodic-flow and rotary regenerators; heat pipe arrays; direct phase contactors; steam and organic Rankine cycles for power generation from waste heat; and heat-pump applications to distillation. The importance of heat recovery and utilization in coal-conversion plants is widely recognized, as are the general problems associated with the design of the exchangers.

A review of past experience in operating coal-conversion plants and of some proposed designs indicates that many flows will 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).

Discussions with individuals knowledgeable in the process heat-transfer field revealed a relatively high degree of unanimity in that design of these exchangers would be more difficult than would their fabrication. Designs are based on the process and property data provided and construction materials specified by the buyer. No warranties are made concerning material lifetime or effect of fouling on performance.

There is a clear need for more physical-property data (especially viscosities), improved heat-transfer correlations for multiphase streams (coal-oil and coal-water slurries, e.g.), and rates of fouling, corrosion, and erosion.

1. INTRODUCTION

Because the specific investment costs (dollars per million Btu of input coal) for converting coal or lignite to electricity, synthesis gas, SNG, methanol, and liquid fuels are all about the same, the product costs will vary significantly with the process thermal efficiencies. One factor obviously affecting these efficiencies is the degree of heat recovery and

use that can be attained reliably and economically in the plants. For this reason and because of the special problems posed by coal-conversion (e.g., slurries, dirty streams, coke formation, and laminar flow), a review of the processes and related equipment that are potentially applicable to coal conversion was desirable.

The survey was conducted by reviewing the general literature and the current catalogs and brochures of a number of companies specializing in the design and fabrication of large and/or severe-service heat-exchange equipment. Discussions were also held with a number of individuals conversant with the present state of the process heat-transfer field.

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 heat recovery, transport, and utilization; operations such as primary heat extraction from methanation catalyst beds and rejection of heat to the atmosphere or surface water are mentioned only in passing, if at all.

The problems of designing and fabricating reliable heat-exchange equipment for coal-conversion plants should be recognized as not fundamentally different from those affecting exchangers for other plants. The factors that currently assume the most importance in assessing capability in this area are those about which the least is known. Uncertainties include stream compositions, thermophysical properties, rates of fouling, corrosion and erosion, and the method of designing for heat interchange between process streams that may include unusually large amounts of solids and/or highly viscous liquids. These problems can be solved only by focused research and development, testing, and documented operating experience.

2. SCOPE

2.1 Process Streams

Heat recovery and use are possible in two general types of equipment. The first type is process vessels such as chemical reactors involving heat interchange because they contain exothermic reactions or because heat 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 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 using the flows between process vessels. This equipment type is the subject of this report.

The size and geometry of a process vessel often determine whether internal heat recovery surfaces or an external exchanger is preferable.

Most coal conversion plant designs include a section for gasification of coal and/or char. Some designs 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:

- coal treatment;
- gasification (in one or more stages), 900 to 3300°F;
- catalytic shift conversion (in one or two steps), 700 to 800°F;
- gas purification, -60 to 260°F; and
- methanation, 600 to 900°F.

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

Several auxiliary processes serve coal-conversion plants, some of which are partially or wholly proprietary in nature; they are not discussed in detail in this report. They include the power, sulfur recovery, and cryogenic air separation (O_2) plants. Cryogenic processes often use power recovered from the adjacent part of the plant as well as internal heat interchanging.

Coal liquefaction plants vary considerably, 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 for provision of utilities. Heat recovery is possible from hot streams in operations 2 through 8.

Types of streams from which heat may be recovered include:

1. solids — char, ash, and solidified slag, in some cases slurried with water or oil;
2. liquids — distillation bottoms and product streams;
3. gases — air, synthesis gas, SNG, and steam; and
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 ash 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; and
4. multiphase mixtures — reactor slurry feed composed of coal in oil with hydrogen gas, water for steam generation, and filter cakes.

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

Table 1. Principal coal-conversion reactions

Reaction	Typical temperature range		Typical pressure range psia	ΔH_r^a		Catalyst
	(°F)	(°C)		(Btu/lb mole)	(kcal/g mole)	
1. Devolatilization: Coal VM ^b → C + CH ₄	600-1000	315-540	15-300	Endothermic ^c		None
2. Carbon-steam:						
(a) C + H ₂ O ⇌ CO + H ₂	1600-1500	870-1040	200-600	58,300	32.4	None
(b) C + 2H ₂ O ⇌ CO ₂ + 2H ₂	1600-1500	870-1040	200-600	44,300	24.6	None
3. Water-gas shift: CO + H ₂ O ⇌ CO ₂ + H ₂	700-770	370-410	360-450	-16,600	-9.2	Iron oxide with Cr ₂ O activator, cobalt oxide
4. Hydrogasification: C + 2H ₂ ⇌ CH ₄	850-1450	455-790	1000-3000	-38,200	-21.2	Alkali carbonates (gas product); Co oxide and Mo oxide (liquid product)
5. Methanation: 3H ₂ + CO ⇌ CH ₄ + H ₂ O	600-900	315-480	300-700	-94,300	-52.4	Raney Ni (Ni and Al), Ni
6. Methane reforming: ^d CH ₄ + H ₂ O ⇌ 3H ₂ + CO	1250-1650	680-900	150-450	97,200	54.0	Ni, Ni oxide
7. Methanol synthesis: 2H ₂ + CO ⇌ CH ₃ OH	500-600	260-315	750-4500	-41,800	-23.2	ZnO, Cr ₂ O, CuO
8. Fischer-Tropsch synthesis: nCO + 2nH ₂ → (CH ₂) _n + nH ₂ O 2nCO + nH ₂ → (CH ₂) _n + nCO ₂	430-645	220-340	325-380	-70,900 -88,200	-39.4 -49.0	Fe or Fe with Cu and K activators
9. Carbon combustion:						
(a) C + O ₂ → CO ₂	1600-2600	870-1425	15-150	-170,100	-94.5	None
(b) 2C + O ₂ → 2CO	1600-2600	870-1425	15-150	-49,300	-27.4	None

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

^bVolatile matter.

^cTo devolatilize to coke, average value is ~1500 Btu/lb of coal.

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

2.2 Heat-Recovery Processes

This report summarizes the methods used to remove heat from a process stream and

1. interchange it with another process stream;
2. transfer it to combustion air;
3. transfer it to another stream, using the solid matrix of a thermal regenerator;
4. deliver it to another stream, using a heat pipe array;
5. interchange it with an intermediate transfer fluid;
6. deliver it to an intermediate fluid for power recovery; and
7. increase its value by the use of a heat-pump system.

Also included in the report is a discussion of several processes that have been applied to the cooling of hot solids.

2.3 Selecting Heat-Recovery Processes

The selection of heat-recovery processes for optimal efficiency in early coal-conversion plants encounters several obstacles. Some of the problems are discussed.

Evaluating fully the life-cycle costs of all promising alternative combinations of process conditions and heat-recovery equipment can be costly. The number of alternatives increases significantly if one is dealing with other than a specific plant site and is therefore obliged to consider a range of unit costs for electric power and water. Accurate trade-offs are impossible, and simplifying assumptions may not be justified.

Assuming that the plant site is known, the cost of specialized equipment is not easily determined. Trade-offs may be made in equipment design by manufacturers to yield differing designs for the same equipment item. Particularly vulnerable are high-temperature devices. Given the requirement to handle a hot process fluid of known composition, various manufacturers may differ as to the best way to compensate for

1. lower metal strength and higher corrosion rates at high temperatures,
2. thermal stresses caused by startup and shutdown or related to the exposure of various parts to different temperatures, and
3. high-heat fluxes often associated with recovery of large heat loads at high temperatures. Should cooled surfaces become fouled, metal temperatures would rise considerably. Fouling rates may vary with fluid temperature, and hotter surfaces generally are more susceptible to corrosion. Selection of the process operating temperature itself can thus involve trade-offs between higher product yields and increased equipment maintenance costs (or increased costs for maintainability of equipment).

One impediment to the optimal design of heat-recovery equipment in early coal-conversion plants might be the operation of a typical heat-exchanger manufacturing firm. Although the manufacturers may be very competent in their particular fields, they quite often specialize in one type of product. Both the engineering and the manufacturing firms may be inexperienced concerning fluid properties, corrosion effects on materials, and operating peculiarities for process streams in coal-conversion plants. The normal practice of designing several petrochemical heat exchangers and rating them against each other results in a minimization of costs. Unfortunately, complete ratings of exchanger designs for many coal conversion streams cannot be accomplished currently with sufficient accuracy to justify significant expenditures because the fouling, required maintainability features, corrosion, and other factors are not well enough known. The present intention is to build plants that will demonstrate the commercial feasibility of available technology in coal conversion. The intent is generally to scale up a pilot plant to a size smaller than commercial but large enough to demonstrate reliable integrated operation, especially of critical areas, in a commercial design. In fact, the experience gained in operating the demonstration plant should permit a more refined rating of heat-recovery equipment, which could result in a better design of the same type or a different approach to heat recovery.

It is likely that manufacturers will be attracted by the potential market in heat-recovery equipment for coal-conversion plants to the extent that they will begin to test their own designs for such services, perhaps in concert with each other or with government-supported test programs.

It is not likely that manufacturers will alter significantly the present policy of warranting equipment only for materials, workmanship, and performance in a clean condition.

Especially since 1970, the large and difficult problem of determining near-optimal configurations of interactive heat exchangers in trains has been addressed. A computer program, for example, has been developed^{2,3} to assist in optimizing the design of such heat-exchanger networks. The classic economic balance between total equipment cost and the energy saved from its installation is quantitatively incorporated. Interesting conclusions from work to date include

1. Complex exchanger networks with high heat-recovery levels are generally less sensitive to changes in process operating conditions than are systems with lower levels of heat recovery.
2. At high heat-recovery levels, the cost of an exchanger network is often little affected by its specific configuration.
3. For even a simple network involving two hot and two cold streams, it has been shown that more than 200 exchanger networks could meet the process requirements; for a system of nine hot and eight cold streams, more than 10^{100} configurations are possible.
4. The price of fuel has a greater influence on the final network configuration than other utility costs.
5. Quite small changes to heat-exchanger systems can result in significant fuel savings.

3. REVIEW OF HEAT-RECOVERY PROCESSES

3.1 Heat Interchange Between Process Fluids

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, which will usually be more attractive than alternate third-fluid systems when (1) conditions remain safe if feed and effluent inadvertently contact each other and (2) distances and elevations are such that the available pressures can maintain flow, and thermal losses from the lines are not excessive; that is, when adequate temperature differences can be maintained. It is important that process-fluid properties be known with sufficient exactness that an economical equipment size can be determined. Attempts to size equipment conservatively often cannot compensate for inadequate data.

3.1.1 Shell-and-tube exchangers

The shell-and-tube (S&T) exchanger is the most common type and has been applied in the chemical industry at high pressures and temperatures in large sizes. The principal problem of the S&T exchanger stems from the velocity changes of the baffled shell-side flow and of the tube-side flow at the tube sheets. The consequences of velocity changes are not so significant with gases or liquids, but multiphase flow can be a problem because phase separation can occur at turning points and at changes in cross-sectional areas. When solids are present, the particles can accumulate in a manner that will reduce the available heat transfer area, and they can cause erosion. Erosion has been countered in some S&T exchangers by using deflector plates and sacrificial shields as well as extended tubes or special ferrules. Uncertain viscosities are also a problem in the design of S&T exchangers because flows may be severely limited in parts of the exchanger.

A survey of costs for large heat exchangers was compiled from manufacturers' estimates and quotations obtained in early 1975 and has been published.⁴ Figure 1 summarizes these costs, which are in 1976

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Fig. 1. Approximate heat-exchanger costs as a function of shell- and-tube side pressures (1976 dollars).

dollars. The average costs varied from \$7 to \$50/ft² for S&T units of carbon steel construction with surface areas of 20,000 to 30,000 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 1 pass tube and shell sides. For induced-draft air-cooled units with aluminum fins and tube pressures of 100 to 4000 psia (other parameters the same as above), the average cost range was from \$15 to \$19/ft² for carbon steel construction and from \$22 to \$30/ft² for 304 stainless steel (costs based on bare tube area).

In a recent article,⁵ unit costs were compared for heat exchangers that were identical except for the materials used for the tubes and the 1/4-in. lining of the tube sheets. Although promoting the use of titanium, the article gave data for 14 metals of 40 that had been considered. Figure 2 shows the results for a range of surface areas for heat exchangers made of 3/4-in.-OD x 16-BWG 20-ft-long tubes unless otherwise noted. The heat exchangers involved all have mild steel shells, are of TEMA-type BEM design, and were designed according to ASME Code, Section VIII, Division I.

Shell-and-tube heat exchangers are now available with Union Carbide's proprietary High-Flux Tubing.⁶⁻⁸ The cost per unit of tubing weight is about five times higher than the cost for untreated carbon steel, but the cost ratio decreases with increasing base-metal cost. A metallic (porous layer 8 to 16 mils thick is bonded to copper alloys, to low-carbon steel, or to nickel steel and stainless alloys. This surface exhibits an overall heat-transfer coefficient for phase-change applications (nucleate boiling-film condensation) about four to five times higher than smooth tubing. It has been used successfully with cryogenics, light hydrocarbons, and aqueous solutions. For equal performance, the cost of a complete exchanger with High-Flux Tubing is apparently within $\pm 20\%$ of the cost of a smooth-tube unit.

3.1.2 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 provide high-reliability

Fig. 2. Cost comparison for S&T heat exchangers, mild steel shells, and tubing as shown.

service 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;⁹ they are fabricated by several companies with license rights. In this concept, 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 of about 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-segmental 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 compensates for the lower local rates. In addition, vortices shed downstream of each baffle rod enhance heat transfer at a low-energy-dissipation rate.

The 32 units mentioned above are or will be in service as waste-heat boilers; gas, oil, and water coolers; and partial condensers; some have operated as long as 6 years without vibration failures. Costs are within a few percent of comparable segmentally baffled S&T units.¹⁰ The RBE should be attractive for use in dirty services characteristic of many coal-conversion process streams.

3.1.3 Plate-type exchangers

An alternate type of heat exchanger which reportedly is efficient for viscous fluids is the plate-and-frame unit. The device is made of thin plates, which are stamped to produce various flow patterns when plates are clamped against each other. The compact units can be engineered to suit the fluid characteristics by varying corrugation patterns in the plates. The critical Reynolds number is reported to be low. The plate exchanger is limited to low pressures (~ 300 psig) because the broad plates are difficult to restrain at high pressure. The exchanger is also limited to low temperatures ($\sim 450^\circ\text{F}$ max) because gaskets are required to seal the plates to separate the fluids and to seal the fluids from the surroundings. Sizes are limited to about $15,000\text{ ft}^2$ per unit. Claims are made for designs capable of a 2°F approach and an attainable heat recovery of 90% or more. Costs are favorable compared to S&T exchangers when stainless steel or higher-grade materials are required.

3.1.4 Spiral-type exchangers

This type of exchanger employs a very deeply grooved plate or membrane that permits two opposing streams to flow in spiral paths counter-currently to each other. Cross-flow arrangements are also furnished as well as special models for overhead condenser service. Sizes as large as 2000 ft^2 and pressures as high as 150 psig (higher for special units) are available in standard models. The spiral-plate exchanger is somewhat limited, as is the plate-and-frame exchanger, although a welded closure can be provided to increase the temperature limits inherent with gaskets.

Similar to the spiral-plate exchanger is the spiral-tube exchanger, which employs a number of tubes coiled side by side to form a wall that is coiled into a spiral shape and confined tightly 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 tube side and 300 psi shell side. Gaskets may be eliminated by welding the unit closed, but cleaning could then be a problem. With gaskets, the maximum temperature is limited to 450°F .

3.2 Heat Exchange Between Process Fluid and Combustion Air

The industrial furnace market currently uses both recuperators and regenerators for heat recovery. Thermal generators (described later) use internal thermal storage matrices that are alternately heated by hot flue gas and then cooled by atmospheric combustion air directed through them for preheating. Recuperators, on the other hand, use a metallic surface between the two continuously flowing streams. The surfaces may be constructed to maximize radiant or convective heat transfer, and two sections may be installed in series, a radiant section at the hotter (gas inlet) end and a convective section at the cooler end.

3.2.1 Recuperators

The radiant section of a stack-type recuperator consists of two concentric cylinders. When lateral pressure differentials are significant, the high-pressure stream is sent through the inner cylinder and the low-pressure stream through the annulus to maintain the cylindrical shape of the intervening wall. Two radiant sections can be arranged in series, the cooler-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. The basic radiant type is generally used for hot stream temperatures above 1600 to 1800°F, at which temperatures thermal radiation from CO₂ and steam is the principal transfer mode. Below ~1300°F, convection sections are used. Convective tubular exchangers, either vertical or horizontal and 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. H₂O), but higher pressure units (2 psig) can also be built. Both field-fabricated and shop-fabricated units are available. Manufacturers' literature suggests that radiant-type recuperators may be used with gas entering at temperatures as high as 2500 to 2600°F. Convective recuperators with flue gas through the tubes can handle dirty gas continuously at temperatures as high as 2000°F. Tubes of 2.5- to 4-in diam are used, and each is fitted with an expansion joint.

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

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

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

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

3.2.2 Regenerators

3.2.2.1 Batch type. Batch-type thermal regenerators (i.e., heat 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, of 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 a 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 summary¹¹ describes blast-furnace stoves using 9- by 4.5- by 2.5-in. firebrick assembled to form flues 3.25 in. square. An air flow of 100,000 scfm is preheated from ambient to 900 to 1200°F (1000°F time average) using four stoves, one unit heating air while three are being heated by hot gas. The shells for the stoves are 24 ft in diameter and 100 ft high; they contain 124,000 ft² of surface area. Bypassing air flow is used for control to maintain a reasonably constant preheat temperature. Open-hearth-furnace regenerators require larger flue openings because of particulates. A less expensive pebble-bed regenerator has been used successfully¹¹ to preheat air to 2800°F in a 25-ton-per-day blast furnace that produces pig iron and ferroalloys (Mn, Cr, and Si).

Work on the magnetohydrodynamics (MHD) process^{12,13} has favored batch-type regenerators with checker-work matrices and very high exit air temperatures (≈3000°F) at pressures as high as 8 atm.

Disadvantages of the batch regenerator include (1) the need for large switching valves, (2) a possible requirement for control valves if reasonably constant preheat temperatures are needed, (3) difficulty in cleaning, (4) cross contamination of streams unless purging between cycles is performed, and (5) sluggishness in bringing the regenerator to temperature. Nevertheless, such regenerators are relatively attractive for use with high-temperature-gas flows bearing particulates that might plug the smaller passages in equipment such as rotary regenerators. Batch regenerators are 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 likely require field fabrication.

Equations for the thermal design of batch regenerators have been summarized by Mueller¹⁴ and by Schack.¹⁵

3.2.2.2 Continuous type. Upstream of the furnace-boiler of an electric generating plant, combustion air is heated in an air preheater, which may be the "heat wheel" type (known also as a Ljungstrom or Lugat preheater). In these units, a thermal 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, as depicted by Fig. 3. The heat-wheel matrix is usually made of ceramic coated material in those sections that are to operate above 600 to 700°F. It may be made of two layers, one being a more corrosion-resistant material to resist acidic cold-end condensation. If the layer shows corrosion, the corroded matrix may be replaced; in some units it may be removed, reversed, and reinstalled to present a fresh surface to the cold inlet air. Leakage rates typically increase with corrosion of seals and matrix and often 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, though the added deposits can help thermally because of their heat capacity. Heat wheels are normally used in moderate temperature environments and at low pressures.

A device resembling the heat wheel is the "Rothemuhle air heater," depicted in Fig. 4, which uses a heavy stationary matrix for transiently storing heat. A sector of the frontal area of the disk-shaped matrix is subjected to a flow of cool air diverted through the disk by slots in two rotating hoods, one on either side of the disk. The hoods direct hot air through that portion of the disk 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 advantage of the Rothemuhle air heater is said to be the lighter duty involved in rotating the less massive hoods, which constitute ~20% of the device weight. In carbon steel it is used for temperatures as high as 800°F.

In both types of heat wheels, there is considerable area to be sealed if hot and cool gas streams are to be kept separate. Heat wheels in the utilities industry normally operate with pressure differences of less than 40 in. of water. Heat wheels have been reported¹⁶ with thermal efficiencies of as much as ~85% and diameters to ~70 ft. The same source

ORNL-DWG 77-19756

Fig. 3. Elements of a rotary regenerative air heater.

ORNL-DWG 77-19635

Fig. 4. Arrangement of counterflow regenerative air heater with stationary plates - Rothemuhle type. Adapted from: Babcock and Wilcox, *Steam/Its Generation and Use*, New York, 1975, p. 13-8.

also reports that purging has been used to limit contamination in a laminar heat wheel to as little as 0.04% and particle contamination to 0.2%. Single-heat wheels have been built to preheat as much as 9 to 10 million pounds per hour of air.

Equations and design curves for rotary air preheaters are available in several references.^{14,17} Smaller power plants rely on tubular air preheaters in addition to the rotary regenerative type.

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 currently range near 45 cents per pound per hour of flue gas ($\sim$ \$2/scfm of flue gas).

A recent inquiry concerning several heat exchangers in typical coal-conversion-plant evaluations resulted in a reply¹⁸ from one vendor which suggested studying the use of a Ljungstrom-type air preheater for conditions involving high-pressure differentials between the two flow streams. Similar work was reportedly undertaken in the 1950s by GE, Solar, Boeing, and C-E in a project for gas-turbine use. Conditions were to have been 1000°F and 4 atm pressure differential at the wheel. More recently (1974 to 1976), testing has been performed on ceramic heat wheels of small size for gas-turbine and Stirling engines.¹⁹ Equipment from several suppliers made of two materials (lithium aluminum silicate and aluminum silicate) was tested. Matrix geometry was also investigated. The tests were based on operation of a 28-in.-diam unit at 800 and 1000°C (1472 and 1832°F). Costs were roughly estimated for smaller (18-in.-diam) units for 170-horsepower Stirling engines.

The problems of using heat wheels are aggravated by high temperature (warping of many matrix materials makes sealing difficult), high pressure differentials (high sealing forces require the matrix to be sufficiently strong to support the torque that the drive must produce), and large heat loads (large size and large resultant deflections can cause leakage). In addition, the pressure drop across the heat wheel may require a fan to maintain flow. Controls and a bypass will be required in some cases²⁰ to ensure safety should the heat wheel stop.

Standard industrial heat wheels accommodate high temperatures and low pressure differentials in small sizes²¹ or moderate temperatures and low pressure differentials in large sizes.²²

3.2.3 Heat-pipe arrays

Heat pipes are high-axial-conductance, heat-transport devices that have undergone considerable improvement since their initial development about 12 years ago. Such a pipe is a closed structure 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 by gravity and/or capillary forces to the evaporator, as illustrated in Fig. 5.

Fig. 5. Schematic of heat pipe.

The relevant literature is abundant; a good overall treatment may be found in Chi's book.²³ A lucid summary of the several operating limits imposed by fluid dynamics considerations (limits relating to boiling, capillary pumping, sonic flow, and entrainment) is given in Tien's article.²⁴ The potential of the heat pipe in coal-conversion processes has been addressed by Ranken in his overview.²⁵ 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, especially in first-generation plants where the shift reactor effluent gas is cooled to a near-ambient temperature for sulfur removal and then reheated for methanation. Also included²² are descriptions of Biery's heat-pipe recuperator concept,²⁶ 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.

Strimbeck and co-workers²⁷ reported earlier results from a MERC/LASL experimental test program, which used three heat pipes to couple thermally a fluid-bed, coal-air combustor at $\sim 2200^{\circ}\text{F}$ with a fluid-bed, coal-steam gasifier at $\sim 1800^{\circ}\text{F}$. In the subject 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 units²⁸ produced by Hughes Aircraft Company (Electron Dynamics Division) in Torrance, California. This type of unit is depicted in Fig. 6. Their "Heatbank" exchangers currently use aluminum or copper tubes and fins and Freons 11, 12, or 113 as internal working fluids. The maximum hot-gas temperature is $\sim 450^{\circ}\text{F}$, though tests are being conducted with more expensive carbon

Fig. 6. Heat pipe bundle incorporated in gas-to-gas recuperator.

steel heat pipes containing Freon 112 or water for use at temperatures as high as $\sim 600^{\circ}\text{F}$. Current assemblies, which have seen use primarily in heating, ventilating, and air conditioning (HVAC) applications, are available with heat-pipe lengths as long as 11.5 ft., assembly heights and depths (8 rows deep) of as much as ~ 4.5 and 1.7 ft, respectively, and with 4 to 14 fins per inch. A central partition plate is used to separate the cool and hot gas streams. The pipes are usually inclined slightly (tilt angle, $\sim 6^{\circ}$; evaporator end, low) to augment condensate return by gravity. Thermal effectiveness normally ranges 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,²⁹ nickel plating over copper tubes and fins was used. The maximum gas-flow rate (each side) is limited at present to $\sim 30,000$ scfm. Approximate exchanger costs vary from \$0.50 per scfm at high-flow rates to \$0.95 per scfm in

the low-flow-rate range. 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 Corporation of Dallas, Texas. Their "Q-Pipe" exchangers are quite 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 from -60 to 900°F . Flow partitions are fabricated from 14- or 16-gage aluminum or 304 stainless steel. Fins, spaced at 8 to 14 per inch, are 7- to 9-mil-thick aluminum or copper. The maximum gas-flow rate in operating units 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 one year. A summary of Q-Pipe exchanger applications³⁰ 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 per inch, arrays 2 to 16 rows deep, face velocities of 400 to 670 ft/min, and have been built for one-side flow rates as high as $\sim 30,000$ scfm. Special systems are said to be deliverable for flows as high as 10^6 scfm, temperatures as high as 1500°F , and thermal effectivenesses to $\sim 90\%$. Such custom units employ carbon steel or stainless steel pipes and fins with plate or welded fin configurations.

Some design considerations and correlations applicable to heat-pipe heat exchangers have been presented in two recent papers.^{31,32}

3.3 Heat Interchange Using Intermediate Heat-Transfer Fluids

3.3.1 Waste-heat boilers and transfer-line exchangers

Waste-heat or heat-recovery boilers may be used to generate steam for utilities (e.g., to drive turbines) or for process use as illustrated by Fig. 7. They may also be used to vaporize other liquids such as organic heat-transport fluids at temperatures generally $<750^{\circ}\text{F}$. Waste-heat boilers have been built for flue-gas flow rates as high as 1 million scfm. In the Claus sulfur-production 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.

Fig. 7. Schematic of waste-heat boiler for heat recovery from hot gas.

In retrospect, it might seem that current waste-heat boilers have been developed in service in the users' plants, an evolution resulting to some degree because both manufacturer and user are in competitive industries. Boilers have operated at varying conditions because both feedstocks and products have been varied for extended periods. Coal-conversion plants using waste-heat boilers have been built overseas, and recent U.S. plant designs incorporate such exchangers. The approach is attractive for several reasons. It permits some degree of corrective action for a poorly performing unit. Use of a waste heat boiler may also allow the designer to deal with fewer unknowns in a given heat exchanger, provided the water is treated to a high quality.

The current state of the art in waste-heat boilers (WHB) as described³³ indicates that process steam is usually generated at 125 to 650 psig. For power generation, pressures of 650 to 1650 psig are commonly used. In the case of coal-conversion processes, the higher pressure steam may be used as a process feed material. Either natural or forced convection units are available, depending on space limitations.

Designs for heat-recovery steam generators have been developed for several industries. Boiler manufacturers, fired heater companies, vendors of shell-and-tube heat exchangers, suppliers of transfer-line exchangers (for ethylene furnace product quenching), and fabricators of reactor vessels have all shown originality in their approaches. Articles on waste-heat boilers³³⁻³⁶ summarize the pros and cons of various traditional designs, which are discussed below.

Flue-gas boilers normally handle flue gas at near-atmospheric pressure, and for reasons of economy generate steam within the tubes. One author³⁴ indicated that "if the boiler is designed by a boiler manufacturer, he will typically use 1-1/2 inch nominal bore (NB) plain tubing, whereas a fired heater contractor is more likely to use 4 inch NB tubing, frequently with extended surface heat transfer fins." Such boilers of a vertical configuration typically use natural circulation; but, if horizontal, the boilers require forced circulation to lessen the chance of dry-out in the tubes. In either case, a mixture of water and steam cools each tube and tubesheet or tube header before being

directed to a steam drum for separation of the steam. Inlet gas temperatures are generally limited to about 1800°F, and gas pressures must be low (typically below 2 psig) unless a shell-and-tube-type exchanger is used.

Process gas waste-heat boilers are of either firetube or watertube design. Process gas may be at elevated pressures. If the gas is at high pressure and is hot and dirty, it is common to route the gas on the tube side where velocities are better managed and the surface can be more easily cleaned.

Some firetube boiler designs have used an internal by-pass pipe with a valve for control. Refinery operators have reported trouble with such valves. Thermal stresses in the tubesheet of the firetube boiler can be a problem, particularly if thick tubesheets are used. One method used to sidestep this is to design the tubesheet for only a modest differential pressure, with the proviso that the pressures of both fluids are to be closely and continuously matched. One manufacturer also points out³⁵ that firetube boilers are somewhat different from conventional shell-and-tube exchangers because the former are often fixed tubesheet designs with the tubes, tubesheets, and shell kept closer to the same temperature by the water boiling in the shell. This factor permits some reduction in thermal stress in a boiler as compared with a sensible heat shell-and-tube exchanger.

The inlet tubesheet in a firetube boiler is particularly vulnerable. Protection may be provided by insulating the outer tubesheet face with a refractory material (also the inlet channel and perhaps the outlet channel) and effectively insulating the tubesheet at each tube penetration by using an insulating layer between a tube ferrule and the tubesheet or tube wall. Tube ferrules are used to direct the hot gas from the inlet channel through the refractory lining and the tubesheet into the tube at a safe distance from the tubesheet. The ferrule also impedes refractory particles from entering the tubes if they are eroded from the refractory face. Insulating the tubesheet permits it to be thinner (because the material is stronger at lower temperatures) and also permits the tubesheet to operate nearer the temperature of the shell and

thus reduces thermal stresses. Proper circulation of water is important to carry off steam, prevent dry-out, and disperse particulates to reduce fouling. Thick tubesheets are unavoidable in some cases (such as a U-tube high-pressure design), for which shell diameters may be minimized to reduce the total load on the tubesheets.

An interesting double-tube steam generator is used for rapidly quenching low-pressure effluent gases from ethylene furnaces in the United States as well as overseas. One can think of the unit as a fire-tube boiler with a small concentric shell around each tube. A German manufacturer makes the units as well as other heat-recovery devices (NH_3 synthesis gas coolers, for example) with a similar double-tube arrangement. The ethylene transfer-line exchanger (TLX) consists of several water-jacketed firetube elements. The central pipe of each element extends at both ends through oval cross-section headers and is welded to the headers. The jacket does not extend through the headers but is so attached that each oval header feeds the annular jacketed space. Several elements are installed in parallel between the two oval headers and form a register. Several registers are combined to form a vertical tube bundle. The exterior surfaces of oval headers in adjacent registers are welded to each other to form a leak-tight "tubesheet," somewhat similar in concept to a membrane wall.

Each oval header connects outside the TLX to a piping manifold. The net result is that the exterior of each double-tube element can be at atmospheric pressure, surrounded only by insulation and a weather-proof jacket. The concept is illustrated by the two sections shown in Fig. 8. The water-steam jacket is usually heated only from within. If the hot gas is clean, it could be routed over the external surface of the jacket, but a refractory-lined shell would be required. It has been reported³⁶ that the added cost of the refractory lining normally offsets the saving from the deletion of some of the double-tube elements.

Double-tube-type boilers need not be used only as ethylene transfer-line exchangers. One manufacturer³⁷ has used a double-tube arrangement with two banks of serpentine coils in parallel. Each bank consists of several S-shaped, double-tube elements mounted in a vertical plane, side

Fig. 8. Ethylene transfer line exchanger of the double-tube type: (a) cutaway section of a TLX having ten registers; (b) cutaway of part of a register. G, Hot gas; W, water; 1, inner tube; 2, outer concentric tube; 3, oval leader tube; 4, welded joint; identifiers 5 through 10 not germane. Source: Schmidt'sche Heissdampf GmbH, *Waste Heat Boilers for Chemical, Metallurgical, and Other Processes*, Kassel-Bettenhausen, W. Germany (n.d.). Reprinted by permission of author.

by side. Four headers serve the two banks; no tubesheets are used. Each element includes one forged distributor at each end to accommodate the entries to the tube and the jacket. The forced-circulation unit described serves to cool ammonia synthesis gas (3278 psia, 716/471°F) and heat 262 ton/hr of feedwater (2480 psia, 435/572°F). The same firm describes a double-tube, natural-circulation, medium-pressure steam generator using two banks of vertical U-bend, double-tube elements with two gas headers and a single steam/condensate drum. This device also uses forged distributors with each element. Both the process headers and the steam drum are subjected to only one flow stream. The unit is rather tall and is supported at the top. It cools ammonia synthesis gas (3350 psia, 667/505°F) and generates 54.6 ton/hr of steam from feedwater at 595 psia.

The double-tube boiler seems to have considerable merit in permitting the use of simplified shop construction (small welds might be automated), and it eliminates the requirements for thick tubesheets or the alternate operational restrictions imposed by a "differential pressure design." With some double-tube designs, the firetubes are straight and easily decoked. A possible application of transfer-line exchangers in coal conversion is the rapid cooling of the stream from a flash hydropyrolysis (FHP) reactor. In this process, pulverized coal is mixed with hydrogen at elevated pressures and heated rapidly. A mixture is formed that, if quenched rapidly, yields valuable hydrocarbon liquids and gases. Typically, an FHP stream would be cooled from 1600 to 1800°F down to 600 to 800°F, a temperature level low enough to freeze the chemical reactions but high enough that the light oils remain in the vapor phase. Currently, direct-contact quenching seems to be the concept of choice, using a water spray, low-temperature steam, higher-temperature liquid metals, or molten salts. A liquid metal or molten salt would be recovered and typically used to generate steam.

As stated earlier, process-gas boilers may be either of the fire-tube or watertube variety. Watertube boilers for waste-heat recovery are often based on a patented or proprietary design. Generally, a vertical tube orientation is preferred to minimize the potential for dry-out in the tube. Any impurities in the feedwater can concentrate

at points of dry-out, and accelerated corrosion can occur in such zones. A recent article summarizes operating problems with waste-heat boilers of the most common designs.³⁴ Failure mechanisms in watertube boilers have included:

1. Creep rupture of individual tubes caused by overheating following complete or partial flow blockage. Debris and magnetite scale dislodged during plant upsets have caused such blockages.
2. Rapid tube corrosion following massive acid breakthrough from the water-treatment plant.
3. Accelerated corrosion beneath deposits in regions of high heat flux.
4. Creep rupture of the shell because of failure of the refractory caused by a combination of poor design, workmanship, and quality control.
5. Extended tubewall dry-out.

Variations of watertube boilers include the double drum for dirty process gas at low pressures (~ 3 psig), bayonet-tube arrangements, and the U-tube design. The last two types require especially clean water. Circulation can be either natural or forced. Another type of watertube boiler³⁷ is used to cool gas after a secondary reformer in an ammonia plant. This unit is a vertically mounted cylindrical vessel with a removable top closure through which the entire complex "tube bundle" is fed by multiple water and steam connections. A valve controls the flow to each coil. The coils occupy an annular space within the vessel. Flows through the exchanger are given³⁷ for the case of generating 151 ton/hr of saturated steam at 2248 psia using process gas (435 psia, 1742/694°F). Flow is forced circulation, and externally mounted steam separators are used. The multiple pass hairpins that make up one coil are flexible and can be inspected by flexing the adjacent coils to permit access. The entire assembly is built for gas to enter at the bottom, pass axially over the annular bundle (some flow being bypassed for control purposes through a valve-regulated central bypass pipe), and then

make a final downward pass through a narrow passageway immediately against the vessel wall. The outer flow passage is formed by an outer row of tubes that are closely spaced but not welded. Advantages claimed for the design include small thicknesses for the elements, a small vessel diameter, fewer external piping runs, and easy bundle cleaning. The shell side, however, would seem to be subject to fouling.

3.3.2 Experience with waste-heat boilers in coal gasification and similar services

The reported experience with heat-recovery steam generators used to cool raw-gasifier gas is limited. Franzen, in responding to several questions concerning the Oschatz-Schack waste-heat boilers used with Koppers-Totzek coal gasifiers, stated³⁸ that the steam-generation pressure is from 40 to above 100 atm, that the gas space velocity is "high," and that tube life with respect to erosion is "satisfactory." In his review of ammonia production from coal,³⁹ Waitzman cites high-pressure steam generation "in a commercially proven waste-heat boiler" as a benefit of the Koppers-Totzek process. A review of the first 18 months of operating experience with the coal-based ammonia plant at Modderfontein, South Africa,⁴⁰ however, indicates that ash-impingement erosion did occur in a region of the waste-heat boilers above the Koppers-Totzek gasifiers, which operate at about 2900°F. Quench water is injected into the hot gas and molten ash leaving the gasifier top outlet, resolidifying the ash, and cooling the gas to an unspecified temperature. The gas then passes sequentially through a radiant boiler and two parallel tubular (convective) boilers which are connected by three tangent-tube boilers, all of which generate steam at ~800 psia. The gas changes direction in passing through the tangent-tube boilers, and the abrasive (high silica and alumina) ash particles impinge on the inner surface of the boiler tubes. The worst-affected areas were initially eroded at a rate of about 3/16 in. per 1000 operating hours, corresponding to a maximum local wall thinning rate of 1.644 in. per year. The localized erosion was arrested by installing protection plates made of 310 stainless steel. As an experiment, all tangent-tube inner surfaces on one set of boilers were coated with "an abrasion-resistant alumina material."

The SHG (Schmidt'sche) Company of West Germany offers a waste-heat boiler design for cooling process gas from partial-oxidation gasifiers.⁴¹ Gas, oil, and coal feedstocks are mentioned, as are inlet gas temperatures of 1300 to 1500°C (2372 to 2732°F) and pressures up to 80 bars (1160 psia). The design consists of a vertically suspended bundle of bayonet tubes cooled internally by boiling water in either natural or forced circulation. Though "ensured good control" over the extreme operating conditions — which include high inlet heat fluxes [as high as 1 MW(t)/m², i.e., 317,160 Btu/hr·ft²], soot and erosive ash entrainment, and corrosion by H₂S — is mentioned, no specific operating experience is cited.

During the 1950s, du Pont operated a coal-based ammonia plant at Belle, West Virginia, using two Babcock & Wilcox (B&W) gasifiers, a smaller unit of semiplant scale, and a later larger one of commercial scale. The operating experience with the heat-exchange equipment in this plant is probably reflected in statements made in a B&W preliminary evaluation⁴² of the technical and economic aspects of producing fuel-grade methanol from coal (September 1973). Prior to methanol synthesis, the gas leaving the B&W gasifier (operated at 330 psia) was cooled to 1200°F by extracting heat in steam superheater banks, following which particulates were removed in multiclones and process steam generated in waste-heat boilers. The gas then flowed (at 300°F) to a Venturi scrubber and to a water-scrub tower. As stated in the evaluation, prior to water scrubbing of the synthesis gas, "the high temperature sour reducing environment within the gasifier represents a high potential corrosion problem for all heat exchange surfaces, and the ability to maintain high metal temperatures in a corrosive gasifier environment must be demonstrated."⁴²

Analogous waste-heat boiler experience has been cited by Ruhe,⁴³ who is aware of two successful applications involving an inlet-gas temperature of 2700°F. In the first, 12 units were installed on a nickel smelter in the Dominican Republic to cool very dusty off-gas at ~2 psig to 1800°F. Gas enters the 3-in.-diam tubes at 500 to 600 fps, generating steam at pressures as high as 150 psig. The units reportedly

have worked well since their installation about 2 years ago. In the second application, a boiler was installed on an oil-fed Texaco partial-oxidation unit in Italy about 15 years ago. Gas enters at $\sim 2700^{\circ}\text{F}$ and 400 psig, laden with soot and bearing H_2S . Steam at 600 psig is generated as the gas is cooled to about 500°F . Initial corrosion problems with hot sulfidation of the tube walls presumably have been solved.

In summary, waste-heat boilers that cool raw coal-gasifier off-gas are subjected to extreme conditions that can be handled by special designs, though apparently not without problems. A recent program recommendation⁴⁴ stated that development work and plant experience are needed to complete and confirm designs of gasifier waste-heat boilers that are exposed simultaneously to high temperature and high pressure. It was suggested that such boilers be designed and installed in selected pilot plants.

3.3.3 In-plant thermal-energy transport

Thermal transport systems are designed to transfer thermal energy from a central source such as a fired heater to a series of sometimes distant energy users by means of a heat-transfer fluid. The American Schack Company, Inc., Pittsburgh, Pennsylvania, for example, offers a series of such "Heat-Energy Systems" ranging in capacity from 2×10^6 to about 150×10^6 Btu/hr.⁴⁵ Their systems encompass liquid- and vapor-transport fluids in both atmospheric and pressurized loops.

Shell Oil Company is installing a 1.5-mile-long closed waste-heat loop at its large (240,000 bbl/day) refinery in Norco, Louisiana.⁴⁶ Refined kerosene will be used to absorb heat from 11 sources (mainly 800°F flue gas) and transfer it in heat exchangers to preheat boiler feedwater and to provide reboiler heat to a butylene fractionating unit. The kerosene will be circulated at a rate of 3940 gpm between high and low temperatures of 417 and 210°F . Heat savings, which total ~ 4.8 billion Btu/day, are valued at \$400/hr based on a replaced fuel cost of \$2 per million Btu (intrastate natural gas).

Fried has summarized the characteristics, physical properties, and heat-transfer efficiency factors of 14 high-temperature heat-transfer

fluids, with emphasis on the temperature range from 350 to 1000°F.⁴⁷ Both liquid-phase and vapor-phase systems are treated. The fluids include water, heat-transfer salt, and a variety of heat-transfer oils.

The use of thermal transport can be extended conceptually to include thermal energy storage, which enables retention of available surplus or waste heat, for recovery and use later when it is needed. A significantly variable capacity factor in a coal-conversion plant might make the use of thermal storage attractive. Primary emphasis to date has been on electric-power systems.^{48,49}

The Rocketdyne Division of Rockwell International has developed an industrial thermal-storage system that uses a high-temperature heat-transfer fluid flowing through a storage unit that contains rock. The stored heat is usable for generating process steam or electrical power, process heating and drying operations, and space and water heating. Information received on the Rocketdyne Dual-Medium Thermal Storage System⁵⁰ indicates that a 5-MWhr subsystem that employs 8000 gal of Caloria HT43 organic heat transfer fluid and 210 tons of granite rock in a steel tank has been successfully tested. The maximum fluid-operating temperature was 575°F (302°C). Rocketdyne is also evaluating techniques for storing thermal energy at temperatures as high as 1000°F (538°C).

3.4 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 Corporation, Sarasota, Florida. A waste-heat stream is used to generate steam, which condenses in a shell-and-tube exchanger and generates Freon vapor to drive a rotary positive-displacement expansion engine. The expander isentropic efficiency with Freon 12, at a rated ΔP 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 onstream availabilities exceeding 99%.

The system is presumably applicable to any waste-heat source temperature of 150°F or higher; with a difference between heat-source and heat-sink temperatures of 100°F or above, typical payout times of 3 years or less are claimed. Conservative approximations of output power under usual circumstances can be made with the following simplified equations taken from a descriptive 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 where it is assumed that the cooling (condensing) water is in the normal ambient temperature range. A system that generates 10 kWe costs about \$12,500, or \$1250/kWe (April 1977), for a 185°F hot-water stream and cooling water at 85°F. A 10-MWe system is projected to cost ~4 million or \$400/kWe.

High-temperature cycles that use steam as the working fluid are exemplified by the waste-heat steam-power systems provided by Thermo Electron Corporation (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 are offered that generate 500 kW to 15 MW and that operate as either condensing or backpressure systems. 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. Typically, these units cool fairly clean waste gases from temperatures of 1000 to 1500°F down to about 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.

The installed cost, excluding foundations, is about \$650/kWe (June 1977). Thermo Electron is also testing prototype units, using organic working fluids such as thiophene.

3.5 Heat Pumps for Increasing the Value of Waste Heat

Heat pumps may be used in several ways to accomplish heat recovery. Several articles have outlined the economics of heat pumps as applied to distillation processes.⁵¹⁻⁵³ Savings in utilities may be effected by using a heat-pump system to condense the vapor at the top and to boil the liquid at the bottom of a column. The approximate economics are given both for new installations and for revisions to existing facilities. The trade-offs are many, involving the costs of utilities, capital, and equipment for various degrees of corrosiveness. One article⁵¹ concluded:

Heat pumps are easily applicable to distillation schemes when refrigeration or chilled water are required for condensation. Heat pumps have limited, but very real ranges of applicability when reboiler temperatures exceed 300°F.... Range is narrowest when conventional design (condenser) could produce low-pressure steam for other applications. Heat pumps are not justified when condenser temperatures are in the range of 100 to 250°F if the plant steam system is used to provide both power and heating.

The situation is different when steam is generated only for process heating because low-pressure process steam then becomes relatively more costly; against the higher priced steam, a heat pump has a wider range of application.

It appears that little use could be made of conventional industrial heat pumps for distillation in coal-conversion plants. The exceptions might be in cases needing some degree of refrigeration (Rectisol process, e.g.). Generally, however, steam will be needed for heating and power

generation, and condensers will not require chilled-water cooling or refrigeration, which closes the two paths most often available for heat pumps.

Heat pumps may be useful in drying applications. Drying is usually thermally inefficient; moisture is rejected, carrying with it a large amount of energy. In addition, the drying medium is seldom used to its full capacity for moisture absorption. One author has discussed the use of heat pumps to improve the energy use in drying processes (chamber type dryers particularly).⁵⁴

In England, an effort is under way⁵⁴ to increase the operating temperature of heat pumps with an air heat sink from 117 to 171°F and thereby improve their efficiency. By using drying air at 171°F and recirculating and dehumidifying it with a heat pump, the author concludes that the total energy consumption would be less than that required in either conventional direct-fired dryers (if 58% efficient) or in steam-heated dryers (if 75% efficient).⁵⁴

The maximum operating temperature for a heat-pump system is limited by the chemical stability of the fluid used as the working medium.⁵⁵ American heat pumps for industrial use normally utilize water heat sinks at both low- and high-temperature ends (230°F maximum).⁵⁵

3.6 Solids Coolers

Coal-conversion process equipment must deal with hot solids such as ash, char, catalysts, and thermal transport media or with mixtures of these materials.

For waste materials posing no contamination problem, a fluid having good heat-transfer properties (water, e.g.) can be mixed with the solid. One method for recovering heat from hot ash or slag is to quench it in a water bath, flashing steam from the water in a special vessel. In a proposed gasifier section for a Synthoil plant, for example, 50 psig steam is generated in a quench 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 that can be conveyed easily 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. If pressure differences between fluids could cause leakage through the heat-transfer walls, the process fluids should be mutually compatible.

For cases in which contamination of the solid by a bath liquid would seriously decrease its value, a fluidized-bed heat exchanger could be used. The fluidized bed requires a gaseous fluidizing medium. A compatible fluid can be separated from the carry-over solids downstream of the cooler. An ERDA-sponsored study by the TRW Systems Group⁵⁶ proposed a fluidized-bed char cooler to receive char at 555 psia and 1600°F. With the recovered heat, 600 psia steam was generated in a tube bundle immersed in a bed of char fluidized with steam (see Fig. 9). Although such fluidized-bed coolers are not standard commercial equipment, they have been built for special applications, such as for cooling alumina.

Several configurations of air-cooled clinker heat-recovery units are used in the cement industry,⁵⁷ as illustrated by Fig. 10. These units require large flows of air, which are used to heat incoming feed material. The air flow is cleaned of carry-over solids between stages. The equipment is currently limited to operation at near atmospheric pressure.

Heat recovery from solids in coal-conversion plants does not appear attractive with the metallic conveyor-belt coolers used, for example, in the food industry. The belts may be made of thin stainless steel but require cooling to prevent annealing. Although hot solids may be handled, the cooling medium (generally water) is allowed only a small temperature rise of ~2°F. The coolant is not useful as a heat source in other parts of the plant, although it is uncontaminated. Metallic conveyor belts are not currently made for operation at pressures significantly above atmospheric.

One commercially available process for recovering heat from solids is the "Waagner-Biro Dry Coke Quenching System."⁵⁸ The Austrian-developed process uses an essentially inert-gas mixture as a quenching fluid and

ORNL-DWG 77-19645

Fig. 9. Conceptual fluidized-bed char cooler. Adapted from: D. Kunii and O. Levenspiel, *Fluidization Engineering*, John Wiley, New York, 1969, p. 25.

ORNL-DWG 77-19646

Fig. 10. Shaft-type clinker cooler, Walther-Beratherm system.

heat carrier. The system is intended to compete with wet quench systems now in use for coke production in the U.S. steel industry. Hot coke at 1742 to 2102°F is lifted in a closed transfer basket to the top of a heat exchanger, which is actually a refractory-lined bunker used as a vertical-shaft cooler. The coke slowly passes through the bunker in 3 to 4 hr and is cooled to less than 392°F. The cooling medium (73% N₂, 2% H₂, 10.5% CO, 14% CO₂, and 0.5% O₂) is circulated by a forced-draft fan through the bunker where it is heated from 302 to >1200°F. The coke cooling medium is cleaned of fines, and a portion of it is used as a bypass stream to level the heat-output rate of the system. In the proposed scheme, the medium is used first for generating steam and then for heating air that serves to dry the feed coal. The description proposes first to heat the feedwater to the waste-heat boiler for deaeration and then to cool the air in a recuperator so that it can be used at a low temperature that will ensure the cooling of the coke to at least 392°F. The recovered heat amounts to 1.2 to 1.5 million Btu per ton of coke charged.

3.7 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 the partial cooling of an ethylene-furnace effluent process stream. Such quenching cools the stream rapidly through a critical temperature range and prevents an undesired continuation of the reaction (i.e., freezing of the chemical equilibrium). Direct-contact cooling is also employed while entrained dust is simultaneously scrubbed from dirty hot gases, in barometric steam condensers, and routinely in wet cooling towers.

This section deals with cool-liquid-hot-gas contactors; those that cool a hot solid with a cooler liquid are treated in Sect. 3.6. 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 CO 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.

Fair's review article⁵⁹ summarizes briefly 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 (with, e.g., 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 with total gas condensation. The design relations are quite approximate, many are derived by analogy from mass-transfer data or correlations, and none are applicable directly to tower diameters exceeding 6 ft (usually restricted to $D \leq 2$ ft). At best, the equations are based on data obtained with air/water, air/oil, and H_2 -light hydrocarbon/oil. To account better for all the influential variables, more comprehensive correlations are needed. Areas addressed should include the degree of back-mixing 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 quite important because thermal effectiveness decreases rapidly with nonuniformity of dispersal.

A summary 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) has been published;⁶⁰ ranges of tray thermal efficiency values are included.

The Ozaki quench cooler (patented in Japan) is soon to undergo prototype testing.⁶¹ Employed in Union Carbide's light-crude-oil cracking process, the cooler is claimed to prevent coking on the tube

side of firetube waste-heat boilers by causing quenching oil to flow concurrently with the gas on the inner wall surfaces. Gas from the cracking process typically might be cooled (tubeside) from 1650 to 750°F while generating steam (shell side) at 600 to 1400 psia. A thin oil film is maintained by high hot-gas velocities (65 to 165 fps). Other advantages claimed include low quenching-oil degradation and minimal wall corrosion by gases such as hydrogen sulfide because of the protective quench-oil coating.

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

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4. BACKGROUND DATA AND PROBLEM AREAS

4.1 Heat-Exchanger 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,⁶² who tested several hundred heaters, coolers, and condensers in the Imperial Chemical Industries Billingham petrochemical complex. Flows (always countercurrent) included pure and mixed liquids and gases at pressures from 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.

As outlined in Sect. 4.2, 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 corresponding low heat-transfer rates have been documented in a report⁶³ prepared by Fluor for the Electric Power Research Institute, which indicates that overall heat-transfer coefficients for operating coal-liquefaction plant heat exchangers have varied over the following typical ranges:

<u>Exchanger type</u>	<u>Coefficient range (Btu/hr·ft²·°F)</u>
Fired coal slurry preheaters	1 to 3
Recycle gas coolers	30 to 100
Heavy vapor condensers	40 to 80
Reactor effluent coolers, preheating coal-paste feed	10 (dirty) to 50 (clean)

Film coefficients for process streams in German coal-hydrogenation plants were reported by Laughrey et al.⁶⁴ For heating coal pastes at a velocity of 8 fps, h varied from 80 to 120 Btu/hr·ft²·°F; for heating

mixtures of middle oil and hydrogen (15 to 30 ft³ of H₂/lb of oil) at 6 to 15 fps, h ranged from 130 to 160 Btu/hr·ft²·°F; and for cooling heavy-oil letdown (HOLD) streams, the average coefficient was 90 Btu/hr·ft²·°F at bulk 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 tar-distillation plants in England also indicate quite low heat-transfer rates.⁶⁵ 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 from 160 to 3000 Btu/hr·ft² and that of overall heat-transfer coefficient from 0.4 to 26 Btu/hr·ft²·°F.

Information concerning the shell-and-tube exchanger used to preheat the coal/water slurry before it enters the centrifuges at the Mohave generating station was obtained from M. I. Dina, plant engineer at Mohave.⁶⁶ The unit is horizontal with 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 indicates that 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. Though no fouling has occurred, tube-inlet erosion has been severe. Stainless steel ferrules at the tube inlets had 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 quite viscous and in laminar or transition flow, and that overall heat-transfer coefficients are relatively low (≤ 100 Btu/hr·ft²·°F).

4.2 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.⁶³ Except for the CSF-fired preheaters for which the N_{Re} was apparently 10,000 to 15,000, the minimum

moduli, based on maximum viscosities,^{63,64} 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. A comprehensive review of industrial laminar-flow heat transfer by Porter⁶⁷ provides an informative summary, including 263 references.

Augmentation of laminar-flow heat transfer in tubes would obviously be desirable. Bergles has reviewed⁶⁸ past studies in which curved tubes, rotating tubes, displaced promoters, surface roughness, internal extended surfaces, and swirl flow have been used. In the same report⁶⁸ and in a later one,⁶⁹ he described the results of tests using twisted-tape inserts, Noranda Forge-Fin (internally finned) tubing, and both Koch and Kenics static-mixer inserts. 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 basis of the ratio of heat transfer to pressure drop, 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 increased by a factor of 2 to 3. The small diameter, constant wall heat-flux tests reported in references 68 and 69 are being extended to both larger diameters and a constant-wall-temperature boundary condition at Heat Transfer Research, Inc., using viscous liquids such as ethylene glycol. A recent application of spirally twisted cupro-nickel tubing in outboard keel coolers for cooling engines and auxiliary equipment on very large boats and small ships has been reported.⁷⁰ The increased surface area and turbulent helical flow combine to double the heat-transfer rate of that in straight tubing with normally laminar flow.

An extension of these efforts to enhance laminar-flow heat-transfer rates in a nonfouling fashion with heavy oils and coal-oil slurries should be worthwhile.

4.3 Property Data and Coefficient Correlations

Relevant physical-property data or methods for estimating the properties are comparatively available for pure and mixed gases and for

pure light liquids.^{71,72} For heavy oils, many liquid mixtures, slurries, slags, refractories, and some construction materials, however, it is clear that many more data are needed.^{73,74} 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 reduces largely to calculating the heat-transfer coefficient. As an illustration of the relative importance of the various physical properties, consider the heat-transfer coefficient for turbulent, single-phase tube flow, for which the coefficient h is proportional to $\left(k^{2/3} C_p^{1/3} \rho^{0.8} / \mu^{0.467} \right)$. Densities (ρ) and heat capacities (C_p) are usually available or can be reasonably estimated, but a threefold error in the thermal conductivity (k) results in a twofold error in h ; likewise, an error in viscosity (μ) of 4.4-fold leads to a twofold error in h .

Knowledge of heat-transfer coefficients is the primary requirement in designing heat-transfer equipment. Adequate correlations for predicting coefficients are available for some coal-conversion applications, but are not for others. The need for reliable correlations for direct-contact coolers and condensers, for coal slurry heaters and slurry-to-slurry interchangers, and for surfaces in fluidized-bed combustors and gasifiers has been addressed by Kasper et al.⁷⁴

A study of heat transfer to and from nonflocculated coal-water slurries in turbulent tube flow has been described.⁷⁵ All the coal was pulverized to -250 mesh (maximum particle diameter $\sim 30 \mu\text{m}$) and slurries 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 not ordinarily available are required to apply the correlation.

4.4 Fouling and Other Process-Fluid Effects

A recent Westinghouse report⁷⁶ is explicit about the ways in which fouling, corrosion, and erosion render current heat-recovery system design for hot coal gas both difficult and uncertain. Problems addressed

include tar and slag fouling, residual hydrocarbon deposition, carbon laydown, hydrogen and H_2S attack, nitriding, and aqueous and chloride-salt corrosion. Statements in the report 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 dirty-service heat 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 data needed for material selection and the erosion data for application of specific mechanical design techniques.

4.5 Summary of General Vendor Comments

In replying to an earlier (1977) questionnaire concerning 12 severe-service exchangers included in three coal-conversion plant evaluations, many vendors prefaced their specific ideas and recommendations with general comments applicable collectively; such comments, paraphrased in summary below, apply in most cases to shell-and-tube units.

The preliminary and tentative nature of the estimates in their replies was stressed by several of the vendors, and a lack of experience in designing exchangers for streams with high-solids content was specifically cited by three.

Major problems were perceived by several companies to be those associated with the stream solids: whether char will stick, plug, and

erode; what velocities to use; characterization of the solids; and probable deposition, fouling, plugging, and erosion, compounded by heavy carrier liquids.

Addressing onstream times, the responses varied from relatively low to perhaps 2 years if materials selection and exchanger design are proper and there is no flow-path plugging by solids. One company commented that onstream availability is not guaranteed, and an interval of about 1 year was estimated by another.

No vendor stated a required development time for adequate design, although 2 years was implied by one company, which 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 Heat Transfer Research, Inc., and Particulate Solid Research, Inc. The acquisition of operational data in demonstration-scale units and the ability to select materials resistant to char abrasion and trace-compound corrosion were emphasized by some.

Exchanger warranties uniformly follow that in the Tubular Exchanger Manufacturers Association Standards and typically extend for a 1-year interval for workmanship and materials, but vendor materials recommendations are excluded from liability.

Delivery times for large exchangers were estimated to vary from a minimum of about 4 months to a maximum of about 30 months, depending on market activity and timely availability of materials such as nickel and chromium alloys. The average of the various fast and slow delivery times was about 1 year.

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

1. Shell and U-tube-bundle exchangers would probably be the predominant type.
2. Higher allowable pressure drops than used in some coal-conversion evaluation reports would be desirable.
3. The design should reflect the expected cleaning procedure.

4. 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.
5. A transient (startup/shutdown) stress-and-fatigue analysis of units operating at high temperatures and pressures is desirable and can be performed.
6. The optimal number of shells is best determined by an economic balance involving the costs, mechanical properties, and environmental resistances of several construction materials, especially when high-pressure fluids are on the shell side.
7. Unit weight limits in several U.S. shops extend to the 150- to 200-ton range.
8. A capability exists for gun drilling tubesheets up to 42 in. thick.

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Appendix A

LISTING OF VENDORS CONTACTED

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Appendix A

LISTING OF VENDORS CONTACTED

Vendors of special equipment items were contacted by telephone as follows:

Solids coolers:

Sandvik Conveyor, Inc., J. Michael Vaccaro, Southern Regional Manager,
Atlanta, Georgia

Fuller Company, Catasqua, Pennsylvania:
W. W. Kramer, Chief Engineer, Fluidized Bed Equipment;
Reuben J. Aldrich, Sales Manager, Fluidized Bed Equipment

Exchangers ordered for H-Coal Pilot Plant:

Brown Fintube Co., Therma Technology, Inc., B. Farnsworth, Senior
Thermal Design Engineer, Tulsa, Oklahoma.

The following companies were contacted by letter:

American Heat Reclaiming Corporation, 1270 Avenue of the Americas,
New York, N.Y. 10020, Attn: Chief Executive Officer

American Schack Company, Inc., The, P.O. Box 11006, Pittsburgh,
Pennsylvania 15237, Attn: Mr. D. R. Hirshberg, Chief Engineer

C-E Air Preheater, Combustion Engineering, Inc., P.O. Box 372,
Wellsville, N.Y. 14895, Attn: Chief Executive Officer

C-E Lummus, Heat Transfer Division of Lummus, 1515 Broad Street,
Bloomfield, N.J. 07003, Attn: W. Schoonman, Vice President
and General Manager

Delta Southern Company, Inc., 1435 Choctaw Drive, Baton Rouge,
Louisiana 70805

Engineers and Fabricators Co., A Marley Company, 3501 W. 11th Street,
Houston, Texas 77008, Attn: Edward P. Hansen, President and
General Manager

Fallon Engineering Company, National Headquarters, 8040 San Fernando
Road, Sun Valley, California 91352, Attn: Chief Executive Officer

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

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

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

Hunt Tool Company, 1945 Cullen, Houston, Texas 77023

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

Nooter Corporation, P.O. Box 451, St. Louis, Missouri 63166, Attn: Mr. Gene R. Smith, General Sales Manager, Shop Fabrication

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

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

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 Department

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

Thermal Transfer, Division of Kleinewefers, 1100 Rico Road, Monroeville, Pennsylvania 15146, Attn: Bernard W. Savko, Assistant Manager—Project Engineering

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

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

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

Yuba Heat Transfer Corporation, P.O. Box 3158, Tulsa, Oklahoma 74101, Attn: A. J. Cunningham, Sales Manager—Process Exchangers

Appendix B

LETTER REPORT OF MARCH 1977 HOUSTON TRIP

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Appendix B

LETTER REPORT OF MARCH 1977 HOUSTON TRIP

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 coal-conversion-related problems and equipment are summarized in a letter report that is reproduced as this Appendix.

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.

Fluid	ΔT_o ($^\circ\text{F}$)	
	1 Fluid Sensible	Balanced Sensible Case
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 $\sim 300^{\circ}\text{F}$, 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, $5 - 10^{\circ}\text{F}$ 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 ($20,000 \text{ ft}^2$ 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 (H_2SO_4) 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 $(U/\Delta P_{\text{shell}})$, 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 ~ 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, $(Re_c)_{shell} = \frac{D_o G_m}{\mu_b} = 50$, where D_o is tube OD, G_m is mean mass velocity based on the net centerplane 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.

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