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**PLASTIC HEAT EXCHANGERS:
A STATE -OF-THE-ART REVIEW**

by

**David Miller, Robert E. Holtz, R. Nelson Koopman,
Thomas J. Marciniak, and Donald R. MacFarlane**

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ARGONNE NATIONAL LABORATORY, ARGONNE, ILLINOIS

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Components Technology Division

July 1979

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TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	9
I. INTRODUCTION.	10
EXECUTIVE SUMMARY	11
II. PROPERTIES OF RELEVANT PLASTIC MATERIALS.	17
A. Introduction to Properties of Plastics	17
B. Candidate Plastic Materials	26
C. Polytetrafluoroethylene	28
D. Polyphenylene Sulfide.	41
E. Polyvinylidene Fluoride	43
III. ADDITIVES AND TREATMENT FOR ENHANCING PROPERTIES	46
A. Description of Techniques.	46
1. Metallic Core.	47
2. Plastic Matrix with Filler	47
B. Methods for Predicting Properties of Composite Materials.	48
1. Mechanical Properties.	48
2. Thermal Conductivity	55
IV. FABRICATION TECHNIQUES AND CONFIGURATIONS	60
A. Shell-and-Tube Heat Exchangers	60
B. Plate-Type Heat Exchangers	63
C. Spiral-Type Heat Exchangers	67
V. APPLICATIONS AND ENVIRONMENTS	69
A. Diesel Engines	72
1. Exhaust Gas Heat	72
a. Operating Pressures.	73
b. Chemical Composition of Gases	74
c. Corrosion and Fouling.	76
2. Heat from Engine Cooling Water	77

TABLE OF CONTENTS

	<u>Page</u>
B. Gas Turbines	78
1. Exhaust Gas Heat	78
2. Operating Pressures	79
3. Chemical Composition of Gases	80
4. Corrosion and Fouling	80
C. Combustion of Fossil Fuels	80
VI. SPECIAL DESIGN CONSIDERATIONS AND ALLOWABLE COSTS OF PLASTIC HEAT EXCHANGERS	84
A. Design Considerations	84
B. Allowable Costs	85
1. Economic Analysis	85
2. Added Capital Cost	85
3. Annual Savings/Costs	85
a. Simple Payback	86
b. Example	87
VII. APPLICATION EXPERIENCE WITH COMMERCIALY AVAILABLE PLASTIC HEAT EXCHANGERS	90
VIII. COMMERCIAL VENDORS OF PLASTIC MATERIALS	93
A. Raw Materials	93
B. Fabricated Units	93
IX. RECOMMENDATIONS FOR FURTHER R&D EFFORTS	94
APPENDIX: Condensation and Boiling Heat Transfer	95
ACKNOWLEDGMENTS	103
BIBLIOGRAPHY	103

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1.	Configuration in Vinyl Polymers.	17
2.	Configurations Involving Carbon-Carbon Double Bonds	18
3.	T-M Diagram for Amorphous and Semicrystalline Polymers	18
4.	Behavior of Some Polymer Properties at Transition Temperatures	19
5.	Multicomponent Polymer Systems.	19
6.	Thermal Conductivity of Polyvinyl Chloride	20
7.	Relation between Thermal Conductivity and Degree of Crystallinity of Polytrifluorochloroethylene at 23°C	21
8.	Thermal Conductivity of Polytrifluorochloroethylene Specimens Listed in Table 1	22
9.	Effect of Heat Treatment on Thermal Conductivity of Polytetrafluoroethylene.	22
10.	Thermal Conductivity of Polyethylene	22
11.	Thermal Conductivity of Polystyrene.	23
12.	Thermal Conductivity of Polyvinyl Chloride	23
13.	Thermal Conductivity of Vinyl Chloride-Acetate Copolymer	23
14.	Thermal Conductivity of Polymethyl Methacrylate: Unstretched and Stretched to 375%.	24
15.	Real and Imaginary Young's Moduli of Oriented Polyethylene As a Function of Temperature	24
16.	Dynamic Young's Modulus of Epoxy Resins with Increasing Degree of Cross-linking	25
17.	Dynamic Shear Modulus, at ~1 Hz, of Polyvinyl Chloride Plasticized with Various Amounts of Diethylhexyl Succinate	25
18.	Relation of Crystallinity to Specific Gravity at 23°C	29
19.	Variation of Specific Volume with Temperature.	30
20.	Linear Thermal Expansion of Polytetrafluoroethylene	31
21.	Recommended Values for Linear Thermal Expansion of Polytetrafluoroethylene.	32
22.	Stress vs Strain in Tension and Compression for PTFE	33
23.	Effect of Temperature upon Tensile Stress/Strain Curves for PTFE	34

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
24.	Effect of Temperature upon Tensile Yield Stress of PTFE	35
25.	Creep in Compression at 25°C (77°F) for 'Fluon'G 163	36
26.	Isochronous Stress-Strain Curves in Compression at 25°C (77°F) for 'Floun'G 163	37
27.	Isometric Stress-Strain Curves in Compression at 25°C (77°F) and Various Strain Levels for 'Fluon'G 163	38
28.	Recovery from Creep in Compression at 25°C (77°F) for 'Fluon'G 163	39
29.	Plots of k , α , and C_p vs Temperature for Polytetrafluoroethylene.	39
30.	Thermal Conductivity of Polytetrafluoroethylene	40
31.	Estimated Specific Volume of Polyphenylene Sulfide vs Temperature.	41
32.	Tensile Strength vs Temperature for Ryton-6 Polyphenylene Sulfide	42
33.	Flexural Modulus vs Temperature for Ryton-6	42
34.	Creep (Apparent) Modulus of Ryton-6	42
35.	Estimated Thermal Conductivity of Polyphenylene Sulfide vs Temperature.	43
36.	PVDF Tensile Yield vs Temperature.	44
37.	Viscoelastic Spectrum of PVDF	44
38.	Tensile Creep/Time 254-508 μm (0.010-0.020 in.)	45
39.	Flexural Creep/Time and Temperature	45
40.	Modulus Enhancement Factor vs Filler Fraction	52
41.	Tensile Strength vs Temperature for 40% Glass-Filled Polyphenylene Sulfide	54
42.	Flexural Modulus vs Temperature for Ryton-4, a 40% Glass- Filled Polyphenylene Sulfide	54
43.	Creep (Apparent) Modulus of Ryton-4, a 40% Glass-Filled Polyphenylene Sulfide	54
44.	Theoretical Curves of Meridith-Tobias, Showing the Dependence of the Relative Thermal Conductivity, $k^* = k_c/k_m$, of a Com- posite on That of the Filler $\gamma = k_p/k_m$ for Various Volume Concentrations, ϕ	58

LIST OF FIGURES

<u>No.</u>	<u>Title</u>	<u>Page</u>
45.	Cost Comparison for S&T Heat Exchangers, Mild Steel Shells, and Tubing.	60
46.	Helical Tapes Space the Tubes	62
47.	Rod Baffle Configurations.	62
48.	Plate Elements for Plate-Type Heat Exchangers	64
49.	Assembled Plate-Type Heat Exchanger.	65
50.	Schematic of Water/Air Heat Exchanger.	68
51.	Configuration for Coextruded Heat Exchanger Channels.	68
52.	Comparison of Exhaust Temperatures from Diesel Engine and Spark-Ignited Gas Engine.	73
53.	Variation in SO_3 Content of Flue Gas with Sulfur Content of Fuel in Experimental Furnace.	81
54.	Variation in Percent Conversion of SO_2 to SO_3 with Sulfur Content of Fuel in Experimental Furnace	81
55.	Relationship of Acid Dew Point with H_2SO_4 Concentration in the Vapor.	82
56.	Recommended Operating Pressure for Teflon Heat Exchangers with 0.1-in. (2.5-mm) OD Tubes	91
A.1.	Calculated (a) Temperature Profiles, (b) Shell-Side and Tube-Side Coefficients, and (c) Their Effect on Overall Heat-Transfer Coefficients and Total Area Required.	95
A.2.	Flow in Horizontal Baffled Condenser with Vertical Baffled Cut; Schematic of Condensation on Horizontal Tubes in a Vertical Tube Row	97
A.3.	Schematic Relations for Condensation Outside Horizontal Tubes .	98
A.4.	Condensate-Retention Effect As Function of Fin Geometry and Fluid.	99
A.5.	Origin of Noncondensable Resistance	99

LIST OF TABLES

<u>No.</u>	<u>Title</u>	<u>Page</u>
1.	Density and Degree of Crystallinity of Polytrifluorochloro-ethylene Specimens	21
2.	Test Specimens	23
3.	Average Coefficients of Linear Thermal Expansion of PTFE in the Annealed Condition	32
4.	Thermal Conductivity of Polytetrafluoroethylene	40
5.	Fillers Used in Plastic Materials	48
6.	Materials--Commercially Available Barrier Coatings and Films .	66
7.	Environments Imposed by Various Applications.	70
8.	Estimated Heat Rejection for the Largest Fuel-Consuming Industries in the USA (1972-1973).	71
9.	Heat Recoverable from Typical Large Four-Cycle Diesel Engine at Full Rated Power	72
10.	Percent of Fuel Input Energy Recoverable from Exhaust Gas of Conventional Diesel Engine at Part Load	74
11.	Diesel Exhaust Analysis Summary	75
12.	Heat Recoverable from Exhaust Gas of Typical 800-kW Turbine-Generator Set	78
13.	Permissible Unit Cost of Exhaust Gas Heat Exchanger	87
14.	Plastic Heat Exchanger Size and Permissible Costs as a Function of Overall Heat Transfer Coefficient, U_p , and Payback Period at a Fixed $\$3/10^6$ -Btu Fuel Cost	89

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ABSTRACT

Significant increases in energy utilization efficiency can be achieved through the recovery of low-temperature rejected heat. This provides incentive for the development of heat exchangers which could be employed in applications where conventional units cannot be used. Some unique anticorrosion and nonstick characteristics of plastics make this material very attractive for heat recovery where condensation, especially sulfuric acid, and fouling occur.

The use of plastic materials in heat exchangers does not immediately suggest itself to traditionally oriented engineers. Some of the unique characteristics of plastics led to the commercial success of DuPont's heat exchangers utilizing polytetrafluoroethylene (trade name Teflon) tubes. Attributes which were exploited in this application were the extreme chemical inertness of the material and its flexibility, which enabled utilization in odd-shaped spaces. The wide variety of polymeric materials available ensures chemical inertness for almost any application. Lower cost, compoundability with fillers to improve thermal/mechanical properties, and versatile fabrication methods are incentives for many uses. Also, since many plastics resist corrosion, they can be employed in lower temperature applications (<436 K), where condensation can occur and metal units have been unable to function.

The primary motivation for this work comes from potential application to energy conservation. It is clear that if application and design can be merged to produce a cost-effective alternate to present methods of handling low-temperature rejected heat, then there is significant incentive for plastic heat exchangers, to replace traditional metallic heat exchangers or to be used in services where metals are totally unsuited.

I. INTRODUCTION

The use of plastic materials in heat exchangers does not immediately suggest itself to traditionally oriented engineers. Some of the unique characteristics of plastics led to the commercial success, in 1965,¹ of DuPont's heat exchangers utilizing polytetrafluoroethylene tubes. Attributes exploited in this application were the extreme chemical inertness of the material, and its flexibility which enabled fabrication for installation in odd-shaped spaces. As long as care is taken to limit the upper temperature and the pressure differences across the tube wall, the devices operate reliably and are competitive in price to alternative construction materials. Cost and versatile fabrication methods are incentives for many uses. Also, since plastics are moisture resistant, they can be employed in lower temperature applications (<325°F) of gaseous waste heat recovery where there are condensation effects and metal heat exchangers have been unable to function.

Plastics have been used or suggested as cost-saving, chemically resistant structural materials for baffles to avoid fretting of vibrating tubes; in conjunction with low-temperature heat sources;² Ocean Thermal Energy Conversion (OTEC);³ and for materials dry cooling towers.^{4,5,6} Even the natural polymer, paper, has been used for heat transfer in space-conditioning applications.⁷

The wide variety of polymeric materials available ensures chemical inertness for almost any application. Many of these materials have low cost, light weight, and lend themselves to inexpensive fabrication methods such as gluing, ultrasonic welding, snap-fitting, extrusion, molding, pressing, and rolling.

While the primary motivation for the present study is the potential for plastic heat exchangers in energy conservation, examination of data required to realize this potential emphasizes the need for resolving unique problems posed by candidate applications. It is immediately clear that if application and design of plastics can minimize their disadvantages (low structural strength and operating temperature), then their advantages (low cost, chemical inertness, flexibility, toughness, and unique surface properties in promoting boiling, condensation and resistance to fouling) are significant incentives to replace traditional metallic devices or to employ plastic heat exchangers where metal units cannot be used.

This report provides a data base for the initiation of R&D activities on plastic heat exchangers. Included are an assessment of plastic materials, fabrication techniques, listing of applications, cost studies, vendor information, and recommendations for further efforts.

EXECUTIVE SUMMARY

Introduction

Some of the unique anticorrosion and nonstick characteristics of plastics prompted DuPont to develop and market heat exchangers made of polytetrafluoroethylene (Teflon*) tubes. Since the introduction of the DuPont units in 1965, plastics have received increased interest for equipment in adverse environments.

The primary motivation for this review is the potential for plastic heat exchangers to recover low-temperature heat rejected from systems that are prone to corrosive condensation and fouling since significant increases in energy utilization efficiency can be achieved. The review also provides incentive for research and development of enhanced-property materials, fabrication techniques, and innovative designs of plastic heat exchangers for applications where conventional units cannot be used.

This report provides a data base for initiation of these R&D activities. Included are an assessment of plastic material properties, fabrication techniques, proposed applications, cost studies, vendor information, and recommendations for further efforts.

Properties of Relevant Plastic Materials

Plastic material properties relevant to the design and fabrication of heat exchangers are numerous, complicated, and, often, difficult to predict. For example, the chemical nature and molecular weight distribution of a polymer depends, to some extent, upon the method of fabrication and post-fabrication treatment, which, in turn, determines the thermal and mechanical properties of the material. Also pertinent are the presence of flaws, crystallinity, and degree of orientation resulting from manufacture, fabrication, and use. Plastics undergo phase changes which further complicate determination of their physical characteristics.

Many polymers were examined in this study, and three candidates were selected for extended study: polytetrafluoroethylene, polyphenylene sulfide, and polyvinylidene fluoride. Selection criteria were strength, known properties and fabrication techniques, chemical compatibility with proposed applications, compatibility with prospective additives (or fillers), and commercial availability.

Polytetrafluoroethylene (commonly Teflon) is a well-known, widely used plastic for which there is considerable, but not exhaustive, property data available. A summary review of these data is presented in the text. Salient

*DuPont trade name.

features of Teflon, the leading candidate for plastic heat exchangers, are unique chemical inertness, resistance to fouling, general availability and ease of fabrication, utility as a boiling/condensation promoter, and relatively good thermal and mechanical properties which can be improved by using additives or fillers.

Polyphenylene sulfide (best known as Ryton) has chemical inertness, high strength, and can be used at higher temperatures ($\sim 260^{\circ}\text{C}$). However, compared to available design data for polytetrafluoroethylene, little is available for polyphenylene sulfide. A summary of these data is presented in the text, with extrapolations where fundamentally sound.

Polyvinylidene fluoride (best known as Kynar) was chosen because of its chemical inertness, high strength, resistance to creep, flexibility, and ease of fabrication (especially in thin sections); however, it is temperature limited (upper bound $\sim 140^{\circ}\text{C}$). Polyvinylidene fluoride represents a trade-off of somewhat poorer physical and chemical properties for reduced cost compared to the other two candidate materials. Limited design and property data available are reviewed in the text.

To improve certain characteristics of homogeneous polymer materials, additives are introduced during the manufacturing process to produce a composite material. These additives (e.g., metals, fibers, and/or plasticizers) can be introduced in an isotropic or anisotropic manner, or can be made anisotropic through fabrication processes.

Additives and Treatments for Enhancing Properties

The objectives of producing a composite material are to: increase mechanical properties, thermal conductivity, and the heat-distortion temperature; and, to reduce fouling, gas/liquid permeability, and overall costs. Composite materials may be divided into three classes: (1) particulate-filled, continuous-polymer matrix phase; (2) fiber-filled, continuous-polymer matrix phase; and (3) networks consisting of two or more continuous phases. In this study, plastic-coated metal cores are also considered composite materials.

A network of two or more continuous phases may be sufficient to enhance thermal and mechanical properties of a composite, but, in some cases, adverse side-effects result from the introduction of a filler material. These adverse effects may be counteracted by an additive, such as a plasticizer.

Clearly, the physical characteristics of composite materials are complex. They are complicated by uncertainties in the properties of the constituents, the morphology of the system, the shape of the filler phase, the phase interfaces, and the orientation of the filler phase. An introduction to the state-of-the-art prediction methods for designing with composite materials is presented in the text.

In support of plastic heat exchangers, future work will probably focus on the formation of composite materials which are basically polymer matrices with fiber or flake fillers. As described in the text, initial screening of materials reduced the choice to aluminum or graphite, with graphite being most favored due to its inertness and fabrication ease.

Fabrication Techniques

Conventional heat exchanger configurations probably will be used for most plastic units, but the use of plastics will require innovative fabrication procedures. The increased interdependence of plastic material properties and fabrication techniques is of prime importance. The text reviews fabrication techniques for plastic heat exchangers, associated problems, and possible solutions.

The shell-and-tube configuration is the most common type of heat exchanger fabricated with plastic or composite polymer materials. At present, very thin wall tubes are used to compensate for the lower thermal conductivity of plastics. Fabrication problems include tube spacing, vibration, clogging due to the small diameter tubes used, and fastening to the tubesheet.

An interesting alternative is the plate-and-frame heat exchanger. This unit features thin plates which are corrugated to provide rigidity. These units are generally limited to liquid-liquid duty at relatively low temperatures and pressures (260°C, 2 MPa). The plate-and-frame type exchanger concept is amenable to use of plastics either as plates or as a coating on inexpensive metal plates.

Another class of heat exchangers is the spiral or helical type employing either rods or plates. Unique features are compactness and accommodation of differential expansion. Plastics have been used in the spiral-rod version and should be considered for the plate version.

To summarize, plastics can be used as a single phase, composite or coating in heat exchanger fabrication. Specialized component fabrication development may be required to ensure the competitive performance of plastic heat exchangers.

Applications and Environments

In addition to the foregoing generic problems of plastic heat exchangers is the identification of potential application areas and the environments present. The remaining efforts will be to demonstrate that plastic heat exchangers are a cost-effective alternative to present methods of handling low-temperature rejected heat.

High energy costs have generated increased interest in recovery and application of low-temperature waste heat, for which plastic heat exchangers are especially suited. Among others, these include production of low-pressure steam and power generation, air and water heating, water purification, cryogenic gas expansion, and agriculture.

Interest in recovery of low-temperature rejected heat is well-founded. A 1972-1973 survey of the largest fuel-consuming industries in the U. S. showed that almost half (48%) of the total heat rejected was at a temperature below 100°C, and that 74% of the total was below 250°C.

A common heat source for application of waste heat recovery is exhaust gas from internal combustion engines, fossil fuel-fired boilers, and gas turbines. Significant increases in energy efficiency can be obtained from the recovery of this heat when lower unit outlet temperatures are achieved. Lower outlet temperatures permit not only larger, sensible heat recovery, but further increases due to latent heat recovery from condensation. Some constituents of the exhaust gas are potentially very corrosive if allowed to condense, especially sulfuric acid. The chemical compatibility of polymers is good with these corrosive environments and plastic heat exchangers would have longer lifetimes than conventional units in the same service.

Performance of plastic heat exchangers also should be improved over conventional units since fouling, caused by corrosion as well as particulates, is expected to be diminished. Details on heat sources, potential energy efficiency gains from rejected heat recovery, and environments in which the exchangers must operate are presented in the text.

Special Design Considerations and Costs of Plastic Heat Exchangers

Plastic heat exchangers have some unique characteristics which enable their application where the use of metal counterparts is prohibitive. Because of this, special design and cost considerations of plastic heat exchangers become apparent. To best address these aspects, recovery of heat rejected from a diesel operated-electric generator set is used as an example.

With respect to design considerations, plastic heat exchangers must not only be able to handle the exhaust gas flow, temperature, and composition, but also cyclic load variations and coolant freezing or overheating. The prediction of performance is more complicated due to condensation, which is expected to occur in the equipment. Fouling and its effect on performance is expected to be reduced by using plastics in the heat exchanger. Because plastics are generally easily formable, nonconventional exchanger shapes could be employed to match available space requirements in particular applications. However, this could render prediction of pressure drop more difficult since standard prediction techniques might not be applicable. Lastly, the flexibility of plastic tubes focuses increased attention on vibration, fabrication, and fatigue design problems.

Permissible costs of plastic heat exchangers are explored using a simple payback analysis (investment divided by the annual return, or savings). In the text, the permissible plastic heat exchanger capital costs are developed as a function of fuel costs and payback period. The specific example of the diesel-generator set is examined. The dominating factor in the analysis is the cost of fuel saved. The analysis showed that significant economic and energy efficiency gains can be realized if waste heat is recovered to provide useful thermal energy. There are, of course, many conditions of a specific application which affect the economics of the plastic heat exchanger; thus, the merits of each application need specific examination.

Experience with Commercially Available Plastic Heat Exchangers

In May 1965, the DuPont Company introduced shell-and-tube heat exchangers made of Teflon tubing. These exchangers have performed well in highly corrosive applications with the virtual elimination of sulfuric acid corrosion in several cases being reported. Applications of these units include heating of steel-pickling baths and agitated reactor vessels, and temperature control of high-purity distilled water processes.

AB Atomenergi-Studsvik manufactures plastic heat exchangers in Sweden which are essentially limited to liquid/gas applications (liquids inside the tubes). These units are made generally from polyethylene and polybuten, and applications include air conditioning using sea water and combustion waste heat recovery.

Both manufacturers use many small-diameter, thin-walled tubes to overcome the low thermal conductivity of plastics. No indication is given that any enhancement techniques are being pursued for either thermal or structural considerations.

Commercial Vendors of Raw Plastic Materials

Polyphenylene Sulfide

Phillips Chemical Company (sole supplier); trade name: Ryton.

Polyvinylidene Fluoride Resin

Fluorchemicals Division, Pennwalt Corporation; trade name: Kynar.
Kreha Corporation of America.

Polytetrafluoroethylene

Montedison USA; trade name: Algaflon.
ICI-U.S. of Imperial Chemical Industries; trade name: Fluon.
Polymer Corporation; trade name: Fluorosint.
Specialty Chemicals Division of Allied Chemical Corporation; trade name: Halon.
Foster Grant Company; trade name: Hostaflon.
E. I. duPont de Nemours and Company; trade name: Teflon.

Although there are many commercial vendors of fabricated components, only DuPont and AB Atomenergi-Studsvik are known to fabricate and market complete plastic heat exchangers.

Recommendations for Further R&D

Detailed recommendations are presented in the text. In summary, the primary thrust of the plastic heat exchanger development program should be twofold:

(1) Perform studies of (a) plastic or composite plastic mechanical and thermal properties; (b) properties-enhancement techniques; and (c) fabrication techniques.

(2) Identify commercial areas where (a) significant quantities of low-temperature heat are being rejected; (b) needs exist on-site or a local market is available for the recovered heat; and (c) particular environments make plastic heat exchangers applicable and better suited than conventional units.

II. PROPERTIES OF RELEVANT PLASTIC MATERIALS

A. Introduction to Properties of Plastics

Plastics are complex materials. Therefore, to correctly and efficiently design devices made of plastic materials of interest, sufficient information must be available to characterize the plastics so that the corresponding properties will be known as functions of environmental conditions.

The complexity arises from the chemical and physical nature of the macromolecules which are characterized by the chemical structure, the nature of the repeating units and the end groups, the composition of cross-links and possible branches, and the nature of defects in the structural sequence. The molecular-weight distribution describes the average molecular size and regularity.

Chemical nature and molecular weight distribution depend on the method of production and treatment of the polymer, which determines the properties and behavior of the material. This microscopic behavior is further complicated by such macroscopic characteristics as the presence of flaws, crystallinity, and degree of orientation resulting from manufacture, fabrication, and use.

While a discussion of the chemical nature of monomers and resulting polymers is not pertinent to this review, the number of alternative structural permutations is large, with drastic alteration of the properties of chemically similar substances. Some of this variability is depicted in Figs. 1 and 2.

A further complication of the material properties results from the existence of phase changes, in which the semicrystalline, glassy, and rubbery, thermodynamically metastable states undergo transitions to the "normal" state. These states depend, in turn, on the amount of short- and long-range

Fig. 1

Configuration in Vinyl Polymers. The main carbon-homochain is depicted in the fully extended planar zigzag configuration. For clarity, hydrogen atoms are omitted. Reproduced from *Properties of Polymers*, D. W. van Krevelen, Elsevier Scientific Publishing Company, 1976.

Fig. 2. Configurations Involving Carbon-Carbon Double Bonds. Reproduced from Properties of Polymers, D. W. van Krevelen, Elsevier Scientific Publishing Company, 1976.

one component. These include polymer blends, copolymers, and the fiber- or particulate-reinforced composites discussed in Section III. A classification of these systems is indicated in Fig. 5.

Fig. 3

T-M Diagram for Amorphous and Semicrystalline Polymers. Reproduced from Properties of Polymers, D. W. van Krevelen, Elsevier Scientific Publishing Company, 1976.

order and short- and long-time stiffness due to structure and molecular weight distribution. Graphical representations of these phases and their dependence on temperature and molecular weight are shown in Fig. 3. The effect of an increase in solvents or plasticizers is similar to a decrease in molecular weight.⁸

At a given molecular weight, the glass transition temperature T_g is a primary determinant of polymer properties, as is the melting point T_m . The effect of these transitions on some properties is indicated in Fig. 4. The strong effects of these two transitions on semicrystalline materials make it imperative that the materials be well-characterized so that prior measured or calculated properties are applicable.

Modern technology has shifted increasingly to materials of more than

Fig. 4

Behavior of Some Polymer Properties at Transition Temperatures. Reproduced from *Properties of Polymers*, D. W. van Krevelen, Elsevier Scientific Publishing Company, 1976.

Fig. 5

Multicomponent Polymer Systems. Reproduced from *Properties of Polymers*, D. W. van Krevelen, Elsevier Scientific Publishing Company, 1976.

Properties of various multicomponent polymer systems are even more complex. Their relation to the properties of the homopolymer and additives range from simple additivity for some properties to complex tensorial relations for others.

The great scatter in available information is illustrated by Fig. 6, where only a small portion of carefully selected data for thermal conductivity of polychlorethene (also known as polyvinyl chloride (PVC) and polychloroethylene) are plotted on log-log coordinates.⁹

Fig 6 Thermal Conductivity of Polyvinyl Chloride. For the sake of clarity only part of the 69 available data sets is presented. Reproduced from "Thermophysical Properties of Polystyrene and Poly (vinyl chloride)," C. Y. Ho et al., Proceedings of the 7th Symposium on Thermophysical Properties, ASME, 1977.

This study must be contrasted against the single data points provided by manufacturers in technical literature describing a product of which they produce over a million kg/yr (with worldwide production of several billion kg/yr).

The unique characteristics of plastic materials can be illustrated by referring to various properties and their variations.¹⁰ The examples used for these illustrations are chosen to use available data, since the data needed are generally not available for the candidate materials.

The degree of crystallinity affects the density of polymers containing both crystalline and amorphous forms, the amount of which will, in turn, depend on the method of preparation or prior history of the material. This is demonstrated in Table 1.

TABLE 1. Density and Degree of Crystallinity of Polytrifluorochloroethylene Specimens¹⁰

Specimen	Density, g/cm ³	Degree of Crystallinity ^a
PTFCE-I	2.116	0.286
PTFCE-II	2.140	0.716
PTFCE-III	2.147	0.841
PTFCE-IV	2.151	0.913

^aFraction of the bulk polymer which is in crystalline form.

In turn, the degree of crystallinity (or density) affects the value of the thermal conductivity at a single temperature, as shown in Fig. 7.

Fig. 7

Relation between Thermal Conductivity and Degree of Crystallinity of Polytrifluorochloroethylene at 23°C. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, *Chemical Reviews*, Vol. 66, 1966. Copyright by the American Chemical Society.

For the four specimens of Table 1, it can be seen that both the thermal conductivity and its temperature dependence will be functions of crystallinity, as shown in Fig. 8.

Fig. 8

Thermal Conductivity of Polytrifluorochloroethylene Specimens Listed in Table 1. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, Chemical Reviews, Vol. 66, 1966. Copyright by the American Chemical Society.

Heat treatment of a specimen also will determine its thermal conductivity, as shown in Fig. 9 for quenched and annealed specimens of candidate material PTFE.

Fig. 9

Effect of Heat Treatment on Thermal Conductivity of Polytetrafluoroethylene. Reproduced from "Thermal Conductivity of Polymers, D. R. Anderson, Chemical Reviews, Vol. 66, 1966. Copyright by the American Chemical Society.

Figures 10 to 13 show the effect of average molecular weight on thermal conductivity of polymer specimens listed in Table 2.

Fig. 10

Thermal Conductivity of Polyethylene. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, Chemical Reviews, Vol. 66, 1966. Copyright by the American Chemical Society.

Fig. 11

Thermal Conductivity of Polystyrene. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, Chemical Reviews, Vol. 66, 1966, Copyright by the American Chemical Society.

Fig. 12. Thermal Conductivity of Polyvinyl Chloride. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, Chemical Reviews, Vol. 66, 1966. Copyright by the American Chemical Society.

Fig. 13. Thermal Conductivity of Vinyl Chloride-Acetate Copolymer. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, Chemical Reviews, Vol. 66, 1966. Copyright by the American Chemical Society.

TABLE 2. Test Specimens

Sample	Polymer	Degree of Polymerization
PE-I	Polyethylene	MW = 21,000
PE-II	Polyethylene	
PE-III	Polyethylene	70,000-80,000
PE-IV	Polyethylene	
PVC-I	Polyvinyl chloride	1,000
PVC-II	Polyvinyl chloride	1,300
PVC-III	Copolymer, ca. 95:5	1,500
PVC-IV	Copolymer, ca. 85:15	800
PS-I	Polystyrene	770
PS-II	Polystyrene	1,000

If a polymer is deformed, its properties will change from isotropic to anisotropic. This is illustrated for thermal conductivity in Fig. 14.

Fig. 14

Thermal Conductivity of Polymethyl Methacrylate: Unstretched and Stretched to 375%. Reproduced from "Thermal Conductivity of Polymers," D. R. Anderson, *Chemical Reviews*, Vol. 66, 1966. Copyright by the American Chemical Society.

The effect on Young's modulus is illustrated in Fig. 15.

Fig. 15

Real and Imaginary Young's Moduli of Oriented Polyethylene as a Function of Temperature. Measurements are shown both parallel $\parallel$ and perpendicular $\perp$ to the direction of orientation. Reproduced from Takayanagi, Imada, and Kajiyama, J. Polymer Sci. Part C, No. 15, 263-281 (1966).

Structure of a polymer strongly affects its properties. In Fig. 16, both the effect of temperature and cross-linking are illustrated qualitatively. Note that the final Young's modulus and the temperature at which the rapid change occurs depend on the composition of the resin-hardener mixture and the amount of cross-linking of polymers used.

The plastic materials considered are often not pure, but contain significant amounts of substances (e.g., plasticizers) which give them certain desired properties. As shown in Fig. 17, the temperature dependence of the shear modulus varies markedly with the amount of plasticizer. Even with pure PVC, a designer must cope with a temperature dependence which causes this property to change almost two orders of magnitude in about 10°C!

Careful consideration of ideas related above should sensitize the reader to the variation in properties of plastics having the same chemical composition but different structural characteristics and molecular weight. Problems are also posed by the great diversity of polymers having the same generic (i.e., polyester, polyamide, polycarbonate, etc.) or trade names (Teflon, Delrin, Nylon, etc.).

Fig. 16. Dynamic Young's Modulus of Epoxy Resins with Increasing Degree of Cross-Linking. Cross-linking increases as the ratio of the components, MDA and epoxy, approach the stoichiometric value. Reproduced from Murayama and Bell, J. Polymer Sci. Part A-2 8, 437-445 (1970).

Fig. 17

Dynamic Shear Modulus, at ~ 1 Hz, of Polyvinyl Chloride Plasticized with Various Amounts of Diethylhexyl Succinate. Numbers refer to ratio of polyvinyl chloride to plasticizer. Reproduced from Schmieder and Wolf, *Kolloid Zeit.* 127, 65 (1952).

B. Candidate Plastic Materials

A preliminary screening was made of plastics based on wide availability, strength, known properties and fabrication techniques, cost, and chemical compatibility with proposed applications. Because of the limitations of time and manpower this list of candidate materials has been drastically pared to three for greater emphasis. The reasons for initial selection and rejection are given as follows:

(1) Polyethene (PE, commonly known as polyethylene) is the least expensive plastic, having good chemical resistance and excellent thermal conductivity, but low melting point and structural strength. Although the structural properties (and conductivity) can be augmented by additives such as metals, graphite, or glass fibre, the deflection temperature is raised only to 120°C.

(2) Polypropene (PP, commonly known as polypropylene) is also inexpensive, with good chemical resistance, higher melting point and structural strength than PE, but lower thermal conductivity. The melting temperature is 176°C, but without additives, the material is useful to only about 125°C.

(3) Polytetrafluoroethylene (PTFE, Teflon, Fluon, or Kel-F) is the leading candidate for application to plastic heat exchangers because of its unique chemical inertness, resistance to fouling, general availability in forms of interest, ease of fabrication, utility as a promoter in boiling and condensation, and relatively good thermal and mechanical properties, as pure resin, up to 260°C (500°F). This temperature range can be extended somewhat by the use of fillers (see Section II). This material also has been used for twelve years by the DuPont Company as a commercial material in heat exchangers, with 10,000 units in operation. These are available in traditional shell-and-tube configurations, and immersion form of both coil and bundle types. Limitations of this material are its modest upper temperature range, loss of structural strength, and high cost. A detailed description of the physical properties is given in Section III.B.

(4) Polyamide (PA, broad family of Nylon-type polymers) is chemically inert, has a melting point which varies from 216 to 275°C, depending on the type, high structural strength, but some water pickup.

(5) Polyester (PET, Dacron-type polyethylene terephthalate is the best known of this versatile family of polymers) is often used in large-size fiber-reinforced units. It is attacked by halogenated hydrocarbons but it can be useful to about 200°C, with the more expensive aromatic polyester useful to 300°C.

(6) Polycarbonate (PC, Lexan or Arylon T-type, polybisphenol carbonate is the best known of this family) is transparent, hard, and strong, but limited to about 130°C and not likely to be available in tubular form.

(7) Phenolic (Bakelite-type phenol-formaldehyde is best known) is a family of thermosetting materials widely used in filled form in electrical applications because of low cost, strength, and hardness. Although it can be used to almost 400°C, it is subject to brittleness which limits many applications.

(8) Epoxy (BPA-ECH, bisphenol-A epichlorhydrin resin is the most widely used) plastics, in fiber-reinforced form are very widely used for large shapes, such as tanks and structural shapes, because of low cost, high strength-to-weight ratio, and inertness to most chemicals, with the exception of chlorinated hydrocarbons. It is useful to only 90-120°C.

(9) Acetal (Delrin-type is homopolymer of formaldehyde, Celcon is copolymer with trioxane) homopolymers have excellent strength and molding properties, and good chemical resistance. Their useful upper temperature is limited to about 165°C.

(10) Polychlorethene (polyvinyl chloride, PVC, and copolymers with other materials) is available in many forms, and is very widely used in industrial and home plumbing, and as a chemically inert structural component. It is of limited use in very hot water, only up to about 80°C.

(11) Melamine (Formica-type, condensation between melamine and urea) thermosetting resins are generally augmented with mineral fillers, have good thermal conductivity, and are strong, though brittle. They are useful in thin sections to about 200°C.

In support of future R&D, the following plastics are more exotic and expensive high-temperature resistant polymers which represent the state-of-the-art. Included are names of manufacturers and corresponding trade names.

(12) Polyphenylene Sulfide (PPS; Ryton--Phillips Petroleum Company) was chosen as a candidate material because of its chemical inertness, modest cost, high strength, and capability for use up to 260°C. A detailed listing of available properties is given in Section III.D.

(13) Polyvinylidene Fluoride (PVDF; Kynar--Pennwalt Company) was chosen as a candidate material because of its chemical inertness, high strength, resistance to creep, flexibility, and ease of fabrication in thin sections. It is limited to only about 140°C upper temperature in application. A detailed discussion of this material is given in Section II.E.

(14) Polyimide (PI; Kapton and Vespon--DuPont; Kerimid and Kinel--Rhone Poulenc) thermosetting plastics are expensive, but have a useful upper temperature limit of 360°C, favorable properties; chemical resistance is slightly lower than competitive plastics. The material is somewhat limited in availability.

(15) Polysulfone (Udel--Union Carbide) plastics are strong and rigid, with good water and chemical resistance, except for polar organic solvents, and useful to about 175°C.

(16) Polyphenyl Sulfone (Radel--Union Carbide) plastics are expensive, but strong, tough, and easily fabricated, have good steam and oxidation resistance, can withstand all but highly polar solvents, and have a useful temperature limit of 200°C.

(17) Aromatic Polyester (Ekcel, Ekonol--Carborundum) combines good mechanical and high-temperature properties with good thermal stability in air to 400°C.

(18) Phenolic-Aralkyl (Xylox--Ciba Geigy) plastics are thermosetting, easily processed, have good mechanical strength and solvent resistance, but are limited to 200°C.

(19) Polyamide-Imide (Torlon--Amoco) copolymers offer many of the very favorable properties of polyimides at lower cost and greater ease in fabrication. These are offset by loss of chemical inertness, particularly to water and acids, and a lower temperature limit of about 250°C.

(20) Hexafluoroisobutylene-Vinylidene Fluoride (HFIB-VF₂; CM-1--Allied Chemical) copolymer has a maximum continuous service temperature of about 290°C, with high creep resistance and excellent chemical resistance. No detailed information is available on this material.

These polymers could be subdivided into thermoplastic and thermosetting materials, but this classification is not as convenient as an upper useful temperature limit since some of the chemical types exist in both forms or with overlapping properties.

C. Polytetrafluoroethylene

Physical properties of even a well-known, highly used, plastic as polytetrafluoroethylene (PTFE) can only rarely be expressed in simple terms. PTFE is not processed by techniques associated with most thermoplastic polymers, but by a method akin to powder metallurgy. One result of the fabrication methods used with PTFE is that the characteristics of the fabricated polymer are unusually dependent on the details of the fabrication process. Any table or graph purporting to express the physical properties of PTFE--including those in this report-- must be regarded as a simple summary of a very complex subject.^{11,12}

Some of the complexities of this material are illustrated in Fig. 18 by the interrelation of specific gravity with crystallinity as affected by voids from fabrication.

Fig. 18. Relation of Crystallinity to Specific Gravity at 23°C

The amounts of crystallinity and voids in fabricated PTFE are matters of considerable practical importance. Fabricators, in general, attempt to maximize the amorphous content for greater ultimate tensile strength and flex life. However, where permeability must be minimized, higher crystallinities and low voids are desirable.

Physical properties of PTFE are strongly dependent on two, low-temperature, reversible, first-order transitions at 19 and 30°C, which together involve a 1% change in density. The glass transition temperature is 123°C (253°F) with a crystalline melting point of 327°C (620°F). In practice, the melting point for unsintered material has a range of 332-346°C, from low to high molecular weight materials. These temperatures, determined by heating at 16°C/min, may be as much as 10°C higher than at very slow heating rates.

As a result of expansion, the specific volume, shown in Fig. 19, or its inverse, the density, increases with temperature. Linear expansion is an alternative method of depicting the same information, as shown in Fig. 20.¹⁴ The specific volume and thermal expansion are related by the equation:

$$V = V_0[1 + \alpha_L(T - T_0)]^3, \quad (1)$$

where

V = specific volume, m^3/kg ,

V_0 = specific volume at the reference temperature,

α_L = linear thermal expansion, $\left(\frac{1}{L} \frac{dL}{dT}\right)$, $^{\circ}\text{C}^{-1}$,

T = temperature, $^{\circ}\text{C}$,

and

T_0 = reference temperature.

The wide range of experimental variation shown in Fig. 20 emphasizes the necessity for carefully characterizing the sample material. Recommended values for annealed, high-density (2.2 g/cm^3) material are shown in Fig. 21. Equations fit to the recommended data¹³ to within about $\pm 5\%$, with greater uncertainty in the transition region, are:

$$L/L_0 = -2.52 + 1.37 \times 10^{-2}T - 6.06 \times 10^{-5}T^2 + 1.46 \times 10^{-7}T^3 \quad (25-293 \text{ K}) \quad (2a)$$

$$L/L_0 = -9.60 + 1.51 \times 10^{-2}T + 1.64 \times 10^{-4}T^2 - 3.52 \times 10^{-7}T^3 \quad (293-340 \text{ K}) \quad (2b)$$

$$L/L_0 = -17.07 + 0.12T - 2.75 \times 10^{-4}T^2 + 2.45 \times 10^{-7}T^3 \quad (340-570 \text{ K}) \quad (2c)$$

Fig. 19. Variation of Specific Volume with Temperature. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12, 3rd Edition, Imperial Chemical Industries Limited, 1972.

Fig. 20. Linear Thermal Expansion of Polytetrafluoroethylene. Reproduced with permission from "Thermophysical Properties of Matter," Y. S. Youlounkian, ed., Vol. 13, 1977. Copyright by Purdue University.

Fig. 21. Recommended Values for Linear Thermal Expansion of Polytetrafluoroethylene. Reproduced from Advances in Thermophysical Properties at Extreme Temperatures and Pressures, S. Gratch, ed., "A Method for the Simultaneous Determination of All Thermal Properties of Poor Heat Conductors over the Temperature Range 80 to 500 K," Luikov, Vasiliev, and Shashkov, ASME, 1965.

For design purposes, it is often convenient to specify an average value of the linear expansion coefficient for a given temperature range. These are listed in Table 3, using room temperature as a base. There is no perceptible effect of the glass transition temperature T_g in these data.

TABLE 3. Average Coefficients of Linear Thermal Expansion of PTFE in the Annealed Condition

Temperature Range		Coefficient of Linear Thermal Expansion ($\times 10^{-6}$)	
$^{\circ}\text{C}$	$^{\circ}\text{F}$	per deg C	per deg F
25 to -190	77 to -310	86	48
25 to -150	77 to -238	96	53
25 to -100	77 to -148	112	62
25 to -50	77 to -58	135	75
25 to 0	77 to 32	200	111
25 to 50	77 to 122	124	69
25 to 100	77 to 212	124	69
25 to 150	77 to 302	135	75
25 to 200	77 to 392	151	84
25 to 250	77 to 482	174	96
25 to 300	77 to 572	218	121

Basic mechanical design information includes stress-strain relations and their temperature dependence. These are shown in Fig. 22 for tension and compression for several temperatures. Although the slopes for compression and tension (or elastic moduli) at any temperature are identical for small strains, the lack of symmetry for large deformations should be noted.

Fig. 22. Stress vs Strain in Tension and Compression for PTFE¹¹

In Fig. 23, deformation characteristics are shown for even larger strains, with temperature as a variable. The irregularity of these curves is due to reorienting of the molecular chains at these large deformations, with some hysteresis. The end of the curve indicates fracture or tensile yield stress. These tensile load-extension curves depend on crystallinity, molecular weight, size and shape of the original particles, severity of faults remaining after fabrication, and the specimen shape.

Fig. 23. Effect of Temperature upon Tensile Stress/Strain Curves for PTFE. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12, 3rd Edition, Imperial Chemical Industries Limited, 1972.

For design purposes, the effect of temperature on yield stress, for failure in less than one minute, is shown in Fig. 24. If the material is to be used under load for any considerable length of time, it should not be stressed beyond a small fraction of the yield stress shown.

Fig. 24. Effect of Temperature upon Tensile Yield Stress of PTFE. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12. 3rd Edition. Imperial Chemical Industries Limited, 1972.

If the longer time behavior under load is needed, the creep characteristics must be considered. This information can be displayed in several ways. For example, in Fig. 25, deformation in percent strain is plotted

Fig. 25. Creep in Compression at 25°C (77°F) for 'Fluon'G 163. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12, 3rd Edition, Imperial Chemical Industries Limited, 1972.

versus the logarithm of time for a family of constant stress levels. Alternatively, the information can be replotted as isochronous stress-strain curves (see Fig. 26), or as isometric stress-strain (see Fig. 27). Each of these plots

Fig. 26. Isochronous Stress-Strain Curves in Compression at 25°C (77°F) for 'Fluon' G 163. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12, 3rd Edition Imperial Chemical Industries Limited, 1972.

Fig. 27. Isometric Stress-Strain Curves in Compression at 25°C (77°F) and Various Strain Levels for 'Fluon' G 163. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12, 3rd Edition, Imperial Chemical Industries Limited, 1972.

is for a constant temperature and represents results for a single, well-characterized material. If the applied stress causing creep is removed, the material will recover part of the deformation. Data on recovery is plotted in Fig. 28 in the convenient coordinates of fraction of recovered strain (strain recovered divided by creep strain) versus reduced time (time of recovery divided by time of creep). The asymptotic behavior for long reduced times gives an indication of the permanent set produced by creep.

Fig. 28. Recovery from Creep in Compression at 25°C (77°F) for 'Fluon' G 163. Reproduced from "Physical Properties of Polytetrafluoroethylene," 'Fluon' Tech. Service Note F12, 3rd Edition, Imperial Chemical Industries Limited, 1972.

The important thermal variables of thermal conductivity k , heat capacity C_p , and thermal diffusivity $\alpha = \frac{k}{\rho C_p}$ (where ρ = density) are plotted in Fig. 29.¹³ Here, anomalies in the three variables are due to the 19°C transition, and 30°C transitions, where CF_2 units of the polymer chain go from a fixed helical configuration to free movement about the carbon chain.

Fig. 29

Plots of k (thermal conductivity $\sim \lambda$ in figure), α , and C_p vs Temperature for Polytetrafluoroethylene. Reproduced from Advances in Thermophysical Properties at Extreme Temperatures and Pressures, S. Gratch, ed., "A Method for the Simultaneous Determination of All Thermal Properties of Poor Heat Conductors over the Temperature Range 80 to 500 K," Luikov, Vasiliev, and Shashkov, ASME, 1965.

There are some additional data in the range 166 to 299°C which have large scatter;¹⁴ for convenience, these are listed in Table 4 and plotted in Fig. 30. Additional measurements at higher temperatures are needed.

TABLE 4. Thermal Conductivity
of Polytetrafluoroethylene

Temp, °C	k, W/m·K	Temp, °C	k, W/m·K
166.1	0.401	258.9	0.397
197.7	0.317	280.0	0.375
208.3	0.346	280.0	0.420
212.2	0.346	298.9	0.404
254.4	0.352

Fig. 30. Thermal Conductivity of Polytetrafluoroethylene

Polytetrafluoroethylene is very stable toward thermal decomposition, and, for this reason, is used as a "nonstick" coating in cooking utensils. Decomposition at processing temperatures of about 380°C (716°F), is very low, and only 0.00152%/min at 423.5°C. Products of decomposition are C_2F_4 , C_3F_6 , and C_4H_8 , all non-toxic. This decomposition is little affected by vacuum or oxidizing conditions.

D. Polyphenylene Sulfide

Polyphenylene sulfide is a crystalline, aromatic polymer with a rigid, symmetrical backbone chain consisting of para-substituted benzene rings connected by single sulfur atoms. This polymer has a high melting point, 290°C (550°F), and has outstanding chemical inertness and fire resistance (as established by industry flammability tests). It can be used for prolonged periods of time at temperatures as high as 230°C (450°F). Below 190-205°C (375-400°F), there are no known solvents for the polymer. In air, degradation is complete only at 715°C (1320°F), but in an inert atmosphere, approximately 40% of the polymer weight remains at 1015°C (1860°F). These properties and the strong affinity for a variety of fillers, which improve properties and lower cost, make this polymer attractive for potential application.

Compared to the large amount of design data available for PTFE, there is a relatively small amount available for polyphenylene sulfide.¹⁵

A single data point of 1.34 is a handbook value for the specific gravity (manufacturers literature gives the specific gravity as 1.3). This must be presumed to be at room temperature, 25°C (77°F), and leads to a value of $1.34 \times 10^3 \text{ kg/m}^3$. Temperature dependency of the specific volume, and its reciprocal density, can be estimated over the range to the melting point, using Eq. (1) and substituting the handbook value of $5.5 \times 10^{-5} \text{ } ^\circ\text{C}^{-1}$ for the linear thermal expansion coefficient. The results are given in Fig. 31.

Fig. 31

Estimated Specific Volume of Polyphenylene Sulfide vs Temperature

Fig. 32. Tensile Strength vs Temperature for Ryton-6 Polyphenylene Sulfide¹⁵

Fig. 33. Flexural Modulus vs Temperature for Ryton-6¹⁵

Mechanical design information is more detailed. Room-temperature tensile strength is 66 MPa (9500 psi) in the as-molded condition. Temperature dependence of the tensile strength is given in Fig. 32. The low- and high-temperature strength decreases with annealing. The flexural modulus of the as-molded material is 3.8 GPa (5.5×10^5 psi) at room temperature; temperature dependence is given in Fig. 33. The annealed material has improved flexural modulus, being significantly higher in the range 80-135°C (175-275°F), where the property changes rapidly. The room temperature compressive strength is 16,000 psi. No temperature dependence is available. Creep behavior is depicted in Fig. 34 by the apparent modulus as a function of time for three temperatures and associated applied stresses.

Fig. 34. Creep (apparent) Modulus of Ryton-6¹⁵

The thermal data available is very limited. The specific heat is 1.1 kJ/kg·°C at 25°C, and the thermal conductivity is given as 0.286 W/m·°C, with no temperature dependence. Since the temperature dependence of purely crystalline polymers is given as inversely proportional to the absolute temperature,⁸ this could be used to estimate thermal conductivity at other temperatures. The results are shown in Fig. 35.

No information is available about any hazards from decomposition products of polyphenylene sulfide.

Fig. 35

Estimated Thermal Conductivity of Polyphenylene Sulfide vs Temperature

E. Polyvinylidene Fluoride

Polyvinylidene fluoride (PVDF) represents a compromise of lower cost, but somewhat poorer physical and chemical properties of its structural and chemical relative PTFE. It can be thought of as a hybrid between PTFE and polyethene (PE), having a chemical formula $(-\text{CH}_2 - \text{CF}_2-)_n$, midway between these two and using half the expensive fluorine of PTFE.

Like PTFE and PE, this material has a structure which is part crystalline and part amorphous, depending on the production technique and prior history of finished products. Both emulsion and suspension techniques are used to manufacture the basic polymer. The emulsion technique produces a polymer characterized by very small particles and a broad molecular weight distribution, whereas the suspension technique produces larger particles, narrower molecular weight distribution, and higher crystallinity, and, therefore, higher melting point. These general distinctions lead to important differences in physical properties and are, in part, responsible for the spread in property values reported in the literature without proper characterization of the sample. In addition, these materials exist in syndiotactic and isotactic forms.

Engineering design data are limited and exhibit considerable scatter for this substance.¹⁶

The density of nonporous samples ranges from 1.75 to 1.78 kg/m³ (109-111 lb/ft³). No temperature dependence of density or specific volume with temperature was found. Values could be calculated from the density and the linear thermal expansion coefficient, as was done for other substances, if the properties were known for a well characterized material. Handbook values give the thermal expansion coefficient as $8.5 \times 10^{-5}/^\circ\text{C}$, whereas the manufacturer gives $7.9\text{-}14.1 \times 10^{-5}/^\circ\text{C}$ in the temperature range 25-60°C (77-140°F). Since the melting point range is given as 156-168°C (312-334°F),

this implies about a 3% increase in length from room temperature to the melting point, and about a 9% increase in volume over this range.

The room-temperature tensile strength at breaking is given as 36-51 MPa (5200-7400 psi) with an elongation of 25 to 500%, indicating toughness as well as strength. The temperature dependence of the tensile yield strength is given in Fig. 36. The compressive strength is given as 55-69 MPa (8000-10,000 psi). Room-temperature flexural strength is 59-75 MPa (8600-10,800 psi); temperature dependence of compressive and flexural strengths is expected to be similar to that of the tensile strength.

Fig. 36. PVDF Tensile Yield vs Temperature (ASTM D638)¹⁶

Elastic moduli include the tensile and compressive moduli of 830 MPa (120,000 psi); the flexural modulus is 1380 MPa (200,000 psi). The temperature dependence of

Young's modulus E is given in Fig. 37. Effects of temperature on the shear and bulk moduli, which can be calculated from Young's modulus by the usual relations, should be similar.

Fig. 37. Viscoelastic Spectrum of PVDF¹⁶

Creep behavior of PDVF is shown for tensile loads in Fig. 38, and for flexural loading in Fig. 39. No information was found for compressive creep.

Fig. 38
Tensile Creep/Time 254-508 μm (0.010-0.020 in.). Film at room temperature.

Fig. 39. Flexural Creep/Time and Temperature

The thermal conductivity at room temperature is 0.100-0.126 W/m·K (0.058-0.073 Btu/h-ft·°F), and the heat capacity is 1.26-1.42 kJ/kg·K (0.30-0.34 Btu/lb·°F). No information on temperature dependence was found. Estimates could be made for assumed temperature dependence of the conductivity as was done for polyphenylene sulfide.

Polyvinylidene fluoride will not support combustion at oxygen concentrations less than 44%. Decomposition begins in air at 375°C (707°F) and in nitrogen at 410°C (770°F), with the evolution of corrosive and toxic hydrogen fluoride gas. This may limit use of this material under conditions where loss of coolant or impediments to flow could lead to such elevated temperatures.

III. ADDITIVES AND TREATMENT FOR ENHANCING PROPERTIES

A. Description of Techniques

Traditionally, metallic materials have been used in heat exchangers because of their low cost, availability, fabricability, and good thermal and structural properties. Plastic materials are now being considered for replacing metals in some applications because of their chemical inertness, unique fabricability, and potential for low initial and operating cost because of the lack of fouling,¹ and enhanced boiling and condensation behavior.¹⁷

Composite materials may be divided into three general classes:

(1) Particulate-filled materials consisting of a continuous matrix phase; in our case, the plastic, and a discontinuous filler phase made up of discrete particles with no special shape or orientation effects.

(2) Fiber-filled materials with shape and orientation effects.

(3) Skeletal or interpenetrating network composites consisting of two or more continuous phases.

Reasons for using composite materials rather than simpler homogeneous polymers are: (1) reduced cost; (2) increased stiffness, strength, toughness and dimensional stability; (3) increased heat distortion temperature; (4) increased thermal conductivity; and (5) reduced permeability to gases and liquids.

The properties of composite materials are determined by the properties of the components, by the shape of the filler phase, by the morphology of the system, and by the nature of the interface between the phases, such as the strength of an adhesive bond between phases. In the case of fiber-reinforced polymers, orientation effects can lead to complex tensorial characteristics of properties.

It is logical, therefore, to consider combining the best features of each material into a composite which is optimal from an economic standpoint: each material enhancing the other. This consideration leads to two asymptotic composite forms:

(1) A metallic core with a protective coating of plastic material to inhibit corrosion and fouling, and enhance boiling and condensation heat transfer.

(2) A plastic matrix with a dispersion of metal or other material to increase thermal conductivity, and structural properties for improved performance at higher temperatures.

The advantages and disadvantages of each material combination will be discussed separately, along with a review of materials properties available and recommendations for design and estimation procedures.

1. Metallic Core

The use of an inert coating to protect a metal from corrosion has its roots in antiquity with the development of paint and the use of gold, silver and copper leaf to protect iron. Cost considerations require a thin, tough, adherent coating which, if damaged, can be easily repaired.

Both PTFE and PVDF have demonstrated satisfactory applicability. Use of PTFE in cooking utensils, oven linings, and some chemical process equipment is well-known. A principal use of PVDF, stressed by the vendor, is the application of protective coatings, which is made simpler by the solubility of the polymer in selected solvents.

In addition, polyphenylene sulfide has been accepted as a nonstick coating for food applications and is promoted by the vendor for coating applications in the form of a slurry. No information was located about the compatibility of the PPS coating and common metals.

Because the polymers would be used as thin films, design would be relatively simple: the coating would be a thermal resistance in series with the metallic core. Mechanical and structural properties would be determined principally by the core properties, with the neglect of the structural properties of the polymer generally giving conservative results. Loss of polymer strength at higher temperatures could be acceptable. Sufficient information probably exists about the thermal conductivity of the polymers to enable design at present.

If a reliably coated surface can operate so the value of the heat recovered more than compensates for the additional cost of extra surface, due to an increase in overall thermal resistances, then this class of composite materials has potential for capturing a share of the heat exchanger market.

The prospects of reliable techniques for coating prefabricated heat exchangers or for fabricating exchangers of coated metal sheet or tubes are sufficiently promising to merit development to the point where costs can be reliably estimated.

2. Plastic Matrix with Filler

Use of metallic or other substances dispersed in a matrix of plastic can increase thermal conductivity and improve structural properties at lower cost since many fillers are less expensive than the candidate plastics. Fillers in general use in plastics are given in Table 5.

TABLE 5. Fillers Used in Plastic Materials

Nonmetals	{	Alumina	Metals	{	Aluminum
		Carbon black			Brass
		Glass			Copper
		Graphite			Lead
		Mica			Magnesium
		Silica			Steel
		Other (MgO, TiO ₂ , etc.)			Zinc

Candidate filler materials for increasing thermal conductivity should satisfy economic, mechanical, and fabrication needs as well as thermal considerations. A filler should have a low cost per unit volume and high conductivity. Fibers are preferred since effectiveness in increasing the conductivity of the composite is a function of the volumetric concentration, shape, and conductivity of the filler. To improve the mechanical properties of the composite, the filler should be available in small particle size to increase the surface-to-volume ratio for good compatibility. Fiber materials of greater strength than the matrix lead to large improvements in properties. Excessive hardness of some ceramic materials could cause abrasion of processing equipment. Compared to plastics, all fillers are sufficiently rigid to be treated as infinitely rigid inclusions in the calculation of mechanical properties.

Consideration of cost, density, conductivity, and availability probably reduces the choice to aluminum and graphite particles. Although glass fibers are widely used as commercial fillers, their conductivity is not much higher than the plastics and benefits to structural properties are probably much greater than the increase in thermal conductivity (see Fig. 43). In general, metallic particles are more compatible with thermosetting plastics than thermoplastic materials because metal-plastic adhesion has proven to be a problem.

B. Methods for Predicting Properties of Composite Materials

This section is a brief introduction to the state-of-the-art of predicting properties of composite materials, with emphasis on plastic systems. Because of limitation of time, no claim is made for completeness and no evaluation of the relative merits of various theories or correlations was made for comparison with existing data.

1. Mechanical Properties

The text by Manson and Sperling¹⁸ is highly recommended as background, since it is relatively comprehensive and complete to the date of publication. In addition, Nielsen¹⁹ supplements and augments the material by Manson and Sperling, and gives special attention to creep and stress relaxation and the dynamic mechanical properties, so important to vibration damping,

which will be a major design characteristic of these flexible materials. The effect of voids and reinforcement by fibers is reviewed by Bishop.²⁰

The following discussion presumes a composite material consisting of a filler dispersed in a plastic matrix.

Most theories for predicting properties of composite materials use the volumetric concentration of filler as a primary variable, assuming uniformly dispersed spherical particles and perfect adhesion between particle and matrix, unless these conditions are specifically relaxed. The effect of voids can be important, but is not included here.

Neglecting voids, the effective density $\bar{\rho}$ of a composite with volumetric fraction φ of particle of density ρ_p and matrix of density ρ_m is given by simple additivity

$$\bar{\rho} = \varphi \rho_p + (1 - \varphi) \rho_m. \quad (3)$$

Hereafter, the superscript bar will denote the effective value, and the subscripts p and m the particle and matrix, respectively. The effective specific volume is given as the inverse of $\bar{\rho}$:

$$\bar{v} = 1/\bar{\rho}. \quad (4)$$

The temperature dependence of density or volume can be determined by using the appropriate temperature dependence of the particle and matrix densities. In the case where thermal expansion will be constrained by elastic interaction between filler and matrix, the problem becomes more complex. Rogers et al.²¹ give experimental data and a comparison with current theories for thermal expansion of fiber-reinforced plastics.

Manson and Sperling give a complex relation for thermal expansion $\bar{\alpha}$ due to Kerner:

$$\begin{aligned} \bar{\alpha} = & \alpha_p \varphi + \alpha_m (1 - \varphi) - (\alpha_p - \alpha_m) \varphi (1 - \varphi) (1/K_p - 1/K_m) \\ & \times [(1 - \varphi)/K_p + \varphi/K_m + \frac{3}{4} G_m]^{-1}, \end{aligned} \quad (5)$$

where α is the thermal expansion coefficient, K is the bulk modulus and G is the shear modulus with the usual subscripts. Equation (5) indicates significant deviations from simple volumetric additivity will occur, with a maximum at $\varphi = 0.5$. This is close to the densest packing of spheres in a simple cubic array of $\varphi = \pi/6 = 0.5236$. Possible deviations from simple volumetric additivity are illustrated by several examples in Ref. 18, and the error in thermal expansion coefficient may be as much as 200% high compared to some equations given there.

The prediction of elastic moduli is complex in cases where the particles restrict deformability of the matrix. The simplest bounds for a generalized modulus M (e.g., elastic modulus E , shear modulus G , bulk modulus K), are given by the simple relations¹⁸

$$\text{upper bound } \bar{M}_U = \varphi M_p + (1 - \varphi)M_m \text{ (case of equal strains),} \quad (6a)$$

and

$$\text{lower bound } \bar{M}_L = \frac{M_p M_m}{\varphi M_p + (1 - \varphi)M_m} \text{ (case of equal strains).} \quad (6b)$$

If any two of the elastic moduli, or one and Poisson's ratio ν , are known, the remainder can be calculated. For isotropic materials,

$$E = 2G(1 + \nu),$$

$$K = E/3(1 - 2\nu),$$

or

$$\nu = (E - 2G)/2G = (3K - E)/6K.$$

For anisotropic substances, the number of elastic moduli and their relationships becomes very complex.¹⁹

These results can be cast into another form by defining the augmentation (or decrease) in the matrix modulus M^* (henceforth the superscript asterisk will denote the ratio with the unmodified matrix) as

$$M^* = \bar{M}/M_m;$$

therefore, the upper bound is

$$M_U^* = \varphi M_p/M_m + 1 - \varphi = 1 + \varphi(M_p/M_m - 1), \quad (7a)$$

and similarly, the lower bound is

$$M_L^* = (M_p/M_m)/1 + \varphi(M_p/M_m - 1). \quad (7b)$$

In many cases, observed values fall between these bounds, although when the interaction between particle and matrix is large, the bounds may be exceeded. The prediction of improved bounds and comparisons with the data is an active field of research, and several current developments are cited briefly below and referenced for the interested reader.

In general, particle size per se does not appear as a variable in most equations. If the modulus of the filler is very large compared to that of

the matrix, the polymer modulus becomes unimportant, leaving a simplified expression depending only on the volume fraction of the filler. However, particle size, shape, and agglomeration may be influences, as well as whether a high-modulus filler is continuous or dispersed. Also, fibrous or flake fillers increase the modulus to a larger extent than do spherical fillers.

A number of equations of increasing complexity can be used. The Guth-Smallwood equation, an extension of one due to Einstein for the case of incompressible filler, is

$$G^* = 1 + 2.5\varphi + 14.1\varphi^2 \quad (8)$$

and is valid for $\varphi < 0.30$. This has been recently extended to the more general elastic case by Chen and Acrivos.²²

Randomly oriented, plate-like particles such as graphite or aluminum flakes, are theoretically expected to yield a greater increase in the moduli than spherical particles for the same value of φ . An equation due to Mooney is recommended by Nielsen¹⁹ for this case, which also includes incomplete dispersion (agglomeration) of particles:

$$\ln M^* = [2.5 + (32/15\pi)(1/P - 1)\varphi - 0.628(1/P - 1)\varphi/(1/P - 0.075)]/(1 - S\varphi). \quad (9)$$

Here, S is the agglomeration factor defined as the ratio of apparent filler volume to true volume, and P is the axial ratio of the plate-like filler particles ($P = 1$ for a sphere). Equation (9) predicts a much larger enhancement than Eq. (8), but its validity is still unproven.

In addition to theoretically derived relations, empirical representations are available. Boehme,²³ studying the enhancement of the strength of high-density polyethylene (HDPE) filled with aluminum powder, fit his data to

$$G^* = e^{A\varphi}. \quad (10)$$

Values of the exponent A were 2.6 for pure HDPE, and 3.0 for HDPE with an 8% acrylic acid graft copolymer, HDPE-AA. He attributed the difference in augmentation to improved peel strength by a factor of 8 at the aluminum-plastic interface with the copolymer, since the elastic moduli of the two unfilled plastics was identical.

The enhancement of the modulus is illustrated in Fig. 40 for HDPE only as a function of the filler (graphite flake and aluminum powder) fraction. No data were found on the elastic moduli of PTFE, PPS, or PDVF.

Fig. 40
Modulus Enhancement
Factor vs Filler Fraction

The dimensional stability of a material under load is characterized by the related concepts of creep and stress relaxation, which are essentially inverse behaviors. As a first approximation,

$$\left(\frac{\epsilon(t)}{\epsilon_0} \right)_{\text{creep}} \cong \left(\frac{\sigma_0}{\sigma(t)} \right)_{\text{relax}}, \quad (11)$$

where ϵ_0 is the initial strain, $\epsilon(t)$ is creep strain after time t , σ_0 is the initial stress, and $\sigma(t)$ is the stress after time. Similarly, the creep elongation could be estimated in terms of the shear or tensile moduli. For the effective behavior of the composite compared to the matrix, this yields

$$\frac{\bar{\epsilon}(t)}{\epsilon_{III}(t)} \approx \frac{G_m}{G} = \frac{1}{G^*}. \quad (12)$$

However, at long times, the creep is slightly less than predicted by Eq. (12) due to filler-polymer interactions.¹⁸

Because plastic materials exhibit typically high creep rates, even at low temperatures and stresses, the creep characteristics are more important than instantaneous mechanical properties when designing for durability. Since creep strain is a function of time, temperature, material, filler character, and stress level, a considerable body of experimental data is required to completely describe the creep behavior of a specific combination of materials. The behavior of filled plastics is very sensitive to the magnitude of the imposed deformation, with nonlinear effects observed at lower strains than in unfilled systems.

Design lifetimes of the order of 100,000 h are appropriate to plastic heat exchangers, so actual creep strain and failure test data are not likely to be available. It is therefore necessary to use data from shorter tests and appropriate models for extrapolation in time.

The effect of filler volume fraction is given by the empirical equation

$$\bar{\epsilon} = \epsilon_m(1 - \varphi^{1/3}). \quad (13)$$

If it can be assumed that the polymer in the composite breaks at the same strain as the unfilled polymer, Eq. (13) can be used to predict the effect of the filler on elongation.

Experience has shown that at high filler concentrations ($\varphi = 0.5$), voids may appear in the composite as a result of adhesion failure of the filler, or its aggregates, breaking up. This phenomena, known as dewetting, leads to early failure of the composite from a drastic deterioration of the creep modulus.

Rusch²⁴ found that the tensile yield strength of a 50% aluminum-HDPE composite was lower than the value for unfilled plastic. However, similar to the enhancement of the shear stress found by Boehme,²² the grafting of just 0.5% acrylic acid copolymer increased the tensile yield strength by about 20% above that of the unfilled plastic.

Composite tensile and yield strengths are known to be influenced by filler particle size as is the strain at yield and breaking points. Interestingly, some empirical relationships indicate that the yield strength varies directly with the particle diameter while others use the inverse or inverse square root of the particle diameter.

Other investigators²⁵ have found that, in isotropic materials with low adhesions, the tensile yield strength of the composite $\bar{\sigma}_y$ is described by a linear relation with the diameter and a complex dependence on volume fraction of filler:

$$\bar{\sigma}_y = \sigma_{ym}(1 - \varphi) - k(\varphi)d, \quad (14)$$

where

σ_{ym} = yield strength of the matrix,

$k(\varphi)$ = function of φ experimentally determined as the slope of the plot of tensile strength versus particle diameter,

and

d = average particle diameter.

For orthotropic materials, the problem includes additional variables. The reader is referred to the review of Dharan,²⁶ which assesses the various failure modes observed in composites and reviews fracture mechanics theories, including semiempirical parameters.

Temperature dependence of tensile strength is shown in Fig. 41, and the flexural modulus in Fig. 42, for as-molded and annealed glass-filled PPS. The annealed material has superior high-temperature properties. The creep modulus is shown in Fig. 43 for several temperatures and applied stress levels. The excellent chemical and physical properties of this material are presently available only in injection molded shapes.

Fig. 41. Tensile Strength vs Temperature for 40% Glass-Filled Polyphenylene Sulfide¹⁵

Fig. 42. Flexural Modulus vs Temperature for Ryton-4, a 40% Glass-Filled Polyphenylene Sulfide¹⁵

Fig. 43
Creep (Apparent) Modulus of Ryton-4, a 40% Glass-Filled Polyphenylene Sulfide¹⁵

It is conjectured that filler particles and interfacial voids may act as fracture initiators. With plastic heat exchanger wall thicknesses of 0.25 mm (0.01 in.), the particle diameters for fillers may be only an order of magnitude smaller.

There are single data ranges for PTFE filled with 15 wt % each of glass fiber and graphite for tensile strength, flexural strength, and flexural modulus.¹¹ Enhancements of 2-3 on the strengths and 4-6 on the modulus are indicated.

For PPS, data are available for a 40 wt % glass-filled, and an unspecified glass- and mineral-filled composition.¹⁵ Strength enhancements are about 1.5-2 and modulus enhancement is about 3 at room temperature. The temperature dependences are similar to about 90°C, and the enhancement decreases somewhat with temperature. High-temperature creep behavior is improved by a factor of about 5.

An additional method of augmenting mechanical properties of polymeric substances is the production of oriented polymers. Oriented specimens can be prepared by:

- (1) Stretching the material while it is in the fluid state and then quenching it.
- (2) Stretching a material that cold-draws.
- (3) Cold rolling of a ductile material.

The longitudinal elastic moduli $E_{||}$ in the stretched direction is generally greater than the modulus E_0 of the unstretched and therefore, un-oriented polymer. The transverse elastic modulus $E_{\perp}$ is generally less than E_0 . Augmentation by a factor of about 3 has been achieved for polyethylene.¹⁹ Readers are referred to Ref. 19 for the effects of orientation on other plastic properties such as creep.

2. Thermal Conductivity

In the same fashion that rigid fillers can be used to augment the elastic and plastic properties of composite materials, materials having higher thermal conductivity can be used to augment plastics having relatively poorer thermal conductivity. Just as fibrous materials provide special augmentation of the structural properties by distributing the load over a greater area or volume, fibrous conducting materials can "short circuit" the plastic materials, reduce temperature gradients, and, thereby, improve the effective thermal conductivity of the composite.

The phenomenon of thermal conductivity (its physical and mathematical analogs, the electrical conductivity, susceptibility, permittivity, permeability, refractivity, or transport property, such as particle diffusivity) represents one of the oldest boundary-value problems in physics. An exact solution for arbitrary concentrations has never been obtained. Even today, such properties of composite media are estimated by one or another of a myriad of approximate formulas,²⁷⁻³⁹ or sometimes measured by building model physical systems.²⁸ Neither the approximate formulas nor the experiments are known to be highly accurate, especially in the limit of infinitely conducting or unusually shaped and distributed particles.

Because the effective thermal conductivity of a composite depends on details of the distribution of a random array of irregularly shaped phases with varying amounts of solid-solid contact, the problem of determining heat-flux lines and temperature gradients cannot be specified exactly. In order to obtain analytical or correlational solutions of the problem, certain simplifications are necessary which may introduce errors attributable to one or more of the following:

- (1) Distorted or unreal geometry.
- (2) Failure to properly consider solid-solid contact including interfacial resistance.
- (3) Idealized heat flow.

The most comprehensive review of this problem was done by Crane, Vachon, and Khader,²⁷ who subdivided the solutions into flux-law models and Ohm's law models. The latter, using thermal analogies, would correctly be called Fourier's law models. (The reader is referred to Ref. 27 for references and equations.)

By considering materials as conducting in parallel and in series, a set of upper and lower bounds was obtained by Hashin and Shtrikman,²⁹ along with closer bounds by using variational techniques for approximate solution of the equations. If the particle conductivity k_p is taken as greater than that of the matrix k_m , then the upper bound for augmentation of the thermal conductivity is

$$\begin{aligned}
 k_U^* &= \bar{k}/k_m + \frac{2(1 - \phi)k_p/k_m(1 - k_p/k_m)}{2k_p/k_m + (1 - k_p/k_m)} \\
 &= \gamma + 2(1 - \phi)\gamma(\gamma - 1)/[\phi(\gamma - 1) - 2\gamma],
 \end{aligned}
 \tag{15a}$$

and the lower bound will be

$$\begin{aligned}
 k_L^* &= 1 + 2\phi(k_p/k_m - 1)/[\phi(k_p/k_m - 1) - 2k_p/k_m] \\
 &= 1 + 2\phi(\gamma - 1)/[2 + (1 - \phi)(\gamma - 1)].
 \end{aligned}
 \tag{15b}$$

Since these represent bounds, various averages will be closer to the true behavior. These are:

$$(1) \text{ arithmetic average, } k^* = (k_U^* + k_L^*)/2;$$

$$(2) \text{ geometric mean, } k^* = \sqrt{k_U^* k_L^*};$$

and

$$(3) \text{ harmonic mean, } 1/k^* + 1/k_U^* + 1/k_L^*.$$

These simple bounds and averages have not been properly tested for composites.

Correlations based on flux-law models are sometimes referred to as exact solutions in that the heat-conduction equation, or equivalent, is solved exactly for a given, highly idealized geometry. The flux-law models are developed from the early Maxwell analysis of loose suspensions, which analyzes a cubic array of particles by considering a single cube containing one particle. The resulting solution of Laplace's equation for the homogeneous and isotropic phases in a simple cubic array of particles led to the expression for thermal conductivity augmentation:

$$\begin{aligned} k^* &= k/\bar{k}_m = \frac{k_p + 2k_m + 2\phi(k_p - k_m)}{k_p + 2k_m - \phi(k_p - k_m)} = \frac{(\gamma + 2) + 2\phi(\gamma - 1)}{(\gamma + 2) - (\gamma - 1)} \\ &= 1 + 3\phi \left/ \left(\frac{\gamma + 2}{\gamma - 1} - \phi \right) \right., \end{aligned} \quad (16)$$

where k is the thermal conductivity, with the same previous meanings for the volume fraction of filler and for the superscripts and subscripts, and $\gamma = k_p/k_m$ is the ratio of filler-to-matrix conductivities. It follows from this solution that the particle size and geometry do not enter into this solution, contrary to experimental evidence. There is generally good agreement with this equation for low-volume fractions.

The solution can be put in the form

$$k^* = k^*(\phi, \gamma, \text{geometry}), \quad (17)$$

and extensions of the Maxwell theory are of this form.

Lord Rayleigh extended the Maxwell solution by considering the thermal interaction between particles in a simple cubic array, producing a solution (slightly corrected by Runge and further corrected by Doyle) which was useful to a filler fraction of about 25%, and which reduced to the Maxwell solution at low filler fractions.

Meredith and Tobias²⁸ followed the Rayleigh development, including a larger number of terms for the infinite series defining the temperature field. Their result (with only the Runge correction to Rayleigh equation) is:

$$k^* = 1 - 3\phi \left/ \left\{ \frac{2 + \gamma}{1 - \gamma} + \phi - \alpha\phi^{10/3} \right/ \left[\frac{\gamma + 4/3}{(\gamma - 1)} + C\alpha^{1/2}\phi^{7/3} \right] + \beta(\gamma - 1)/(\gamma + 6/5)\phi^{14/3} + \dots \right\} \right., \quad (18)$$

where α is a constant with the values 1.315, 0.0752, and 0.129 for simple cubic (s.c.), face-centered cubic (f.c.c.), and body-centered cubic (b.c.c.) arrays, respectively, and $C = 0.357$. For the simple cubic case, $\beta = 0.016$, leading to a negligible contribution for all values of ϕ less than $\pi/6 = 0.5236$ for touching spheres. The Meredith-Tobias solution reduces to the Rayleigh and Maxwell solutions for small volume fractions of filler, agrees well with data for $\phi < 0.40$ and $\gamma < 10$,^{27,36} and is widely used.

Results of the Meredith-Tobias calculations are shown in Fig. 44.³⁶ It is seen that augmentations of 5 are about the maximum expected.

Fig. 44

Theoretical Curves of Meredith-Tobias, Showing the Dependence of the Relative Thermal Conductivity, $k^* = k_0/k_m$, of a Composite on That of the Filler $\gamma = k_p/k_m$ for Various Volume Concentrations, ϕ . k_m is the conductivity of the matrix. Note that for high values of γ , the conductivity is dependent only on ϕ .³⁶

Zuzovsky and Brenner³¹ have criticized the results reported by Meredith and Tobias because they used an axially symmetric temperature in the direction of the microscopic temperature gradient instead of a potential with the fourfold symmetry of the cubic array. Zuzovsky corrected this error, extended the work on s.c., b.c.c., and f.c.c. arrays to the order $\phi^{14/3}$, where the results agree quite well with experimental data of Meredith and Tobias to $\phi = 0.45$. (The reader is referred to Ref. 31 for details.)

An additional error in the original Rayleigh derivation was corrected recently by Doyle.³² Using a numerical technique, he also corrected the Meredith-Tobias work to the $\phi^{10/3}$ term, and published his results in graphical and tabular form for the case of $\gamma = \infty$ for the three cubic arrays. These results are in excellent agreement with s.c. data to $\phi = 0.51$, and the simple interpolation formula used by Doyle for the s.c. array:

$$k^* = 1 - \frac{\pi}{2} \ln(1 - 6\phi/\pi),$$

is found to give an accurate fit over the entire range of ϕ . Results for other values of γ are alluded to, but not yet published.

An empirical correlation, due to Krupiczka,³⁷ is available for high packing fractions:

$$k^* = \gamma [0.280 - 0.757 \ln(1 - \phi) - 0.057 \ln(\gamma)] \quad (19)$$

Equation (19) fits the data well, but is limited in applicability for large values of γ .

The review of Crane et al.²⁷ also includes results of Ohm's law models. Although some of these fit the data well for dilute or dense filler fractions, the particle geometry assumed is unrealistic for many models (i.e., cubes in series and parallel). Of these models, only those of Willhite, Kunii, and Smith,³⁸ and Hengst-Kaganer³⁹ are sufficiently accurate to be used, but are best for $\gamma < 15$. The parallel isotherm model of Crane and Vachon³³ provides comparable accuracy and is superior for $\gamma > 15$, which is typical of the composites of interest for heat exchangers. Since experimental data verifying these models is based on conductivity of fluids in packed beds, care must be taken to exclude convective or radiation effects and appropriate boundary conditions. Limited miscellaneous data on metal-filled plastics can be found in Ref. 40.

Only Hamilton and Crosser³⁵ address the effect of shape on composite conductivity in any detail. They show that the conductivity of the composite is very strongly affected if the conductivity of the filler material is much greater (say, >100 times) than that of the matrix. The thermal conductivity is always higher for composites containing nonspherical randomly oriented particles than for those containing an equal volume concentration of spheres. Any nonspherical particle will be less compact than a sphere and, therefore, will tend to "short circuit" the poorer conducting matrix.

Consideration of candidate fillers for augmenting thermal conductivity of the candidate plastics again requires low cost, low density, but now, high conductivity as well as compatibility and availability in flake or fibrous form. Again, aluminum and graphite appear to be the best materials. Both are available in flake and fiber form at increased cost (very increased for aluminum, about \$40/lb, but rapidly decreasing, for graphite fiber). If exposed in the matrix, aluminum must be coated, whereas graphite is sufficiently inert chemically that it requires no coating, provided surface permeability does not suffer.

Availability of natural graphite at less than \$0.20/lb gives great incentive to use this material at near-maximum volume fractions, with large augmentation of both mechanical and thermal properties. Appropriate consideration must be given to effects on fabricability and use.

IV. FABRICATION TECHNIQUES AND CONFIGURATIONS

If the chemical, structural, and thermal properties of plastic materials can be matched to a proposed use, the next consideration is the economic fabrication of the proposed configuration. This section is a review of conventional heat exchanger configurations, existing and proposed plastic heat exchangers, related problems, and possible techniques for solving them. The interplay between material properties, configurations and their fabrication, and the design process is emphasized.

A. Shell-and-Tube Heat Exchangers

The most common heat exchanger configuration is the shell-and-tube type (S&T); it is widely used and is available in large sizes (e.g., 13.5 ft in diameter, 38 ft long and up to 40,000 ft² per shell). A comparison of heat exchanger cost versus area and material is given in Fig. 45 for corrosion-resistant materials.⁴¹

Fig. 45. Cost Comparison for S&T Exchangers, Mild Steel Shells, and Tubing⁴¹

Design problems peculiar to the S&T heat exchanger are: fastening the tubes in tubesheets; spacing and supporting the tubes to prevent vibration; leakage and wear; accommodating pressure differences and thermal expansion; and reducing erosion, corrosion, and fouling. Velocity changes of baffled shell-side flow and of tube-side flow at the tubesheets are important in the economic trade-off between capital cost of the exchanger and operating cost (maintaining flow) and maintenance of the exchanger versus the value of the energy transferred.

Guidance in the fabrication of shell-and-tube exchangers from plastic materials can be obtained by considering the commercially available DuPont equipment.⁴² To minimize thermal resistance of the PTFE material, thin-walled tubes (0.010 and 0.025 in.) are used. To maximize allowable pressure difference between tube- and shell-sides, the tube diameter is limited to ten times the wall thickness, leading to 0.100- and 0.250-in. outer diameters, respectively.

The small tube diameter in plastic heat exchangers compensates in part for higher thermal resistance in the tube wall by allowing for higher surface area per unit volume (up to $240 \text{ ft}^2/\text{ft}^3$) than in conventional metal exchangers (less than $50 \text{ ft}^2/\text{ft}^3$). The smaller tube diameter requires straining of tube-side fluid to avoid clogging. A common problem is fastening the tubes into the tubesheet and spacing the tubes to preserve the shell-side geometry at optimal spacings.

DuPont has solved the problem of fastening a tube into a tubesheet by fusing the ends into a rigid honeycomb structure.^{43,44} Depending on the nature of the material, fusion welding, ultrasonic welding, or solvent welding are possibilities. Specialized material surface treatments are available which permit the use of conventional adhesives.⁴⁵

Other techniques include expanding the tube over a ferrule, or using a rigid insert to constrain the tube in a shaped hole in the tubesheet (see illustration).

The problems of spacing the tubes can be resolved in a variety of ways. The DuPont exchangers feature a number of solutions. For example, plastic tape is wrapped around the tubes to give the correct spacing, and the bundle is rolled up, as shown in Fig. 46; or the flexible thin tubes are braided and

Fig. 46. Helical Tapes Space the Tubes

flexible plastic rods (or flattened tubes) are used as ties to position the bundle. A unique plastic spacer comb arrangement is described in the patent literature.^{46,47}

Also available are the traditional segmentally baffled arrangements made of plastic or plastic-coated materials, and the rod-baffled exchanger (RBE) concept.⁴⁸ The latter concept was developed by Phillips Petroleum Company to replace three, very large, S&T waste-heat boilers which failed because of excess tube vibration. In this concept, a matrix of rods, comprising a baffle set, is arranged to contact the tubes. Several configurations are shown in Fig. 47.

Fig. 47. Rod Baffle Configurations

Although the rods can be at an angle to the tubes and a triangular tube pitch can be accommodated, experience in fabrication and use supports the

preferred perpendicular arrangement on a square pitch. Here, the rod diameter is equal to the spacing between adjacent tubes, and the baffle set touches and constrains the tubes from four directions. While close tolerance fabrication is important to minimize the clearance between the baffle ring (to which the rods are fastened) and the heat exchanger shell, this configuration appears uniquely desirable to rigidly support the very flexible plastic tubes. Virtues claimed for the configuration include minimal tube vibration, low shell-side pressure drop and fouling-rate, and ease of cleaning.

Based on experimental tests, with confirmation by field data, the shell-side pressure drop of the RBE can be about half that of the conventional exchanger with a segmental plate baffle. Because there are no flow stagnation zones, heat transfer is effective over the entire tube surface, thereby more than compensating for the higher, local crossflow heat transfer rates in a traditional exchanger. The improved heat transfer per unit pressure drop, about 1.8 times higher for the RBE, is attributed to vortices shed downstream of each baffle rod, and has been confirmed for water, air, and oil.

RBE units, with shell diameters to 6 ft and lengths to 65 ft, have operated as long as six years without vibration failure. Initial costs are within a few percent of comparable segmentally baffled S&T units. It is of interest to note that the prior work on the RBE was developed by Oak Ridge National Laboratory, under the auspices of the U. S. Atomic Energy Commission, for a molten salt-to-NaK exchanger as part of the Aircraft Nuclear Propulsion program.⁴⁹

Fabrication of tubes from plastics is generally done by extrusion. Other methods include forming and welding of strip into tubular forms, injection molding (limited to shorter lengths at greater expense), coextrusion of several materials, and extrusion of forms other than circular cylinders. Desirable configurations include channels for countercurrent or crossflows in compact exchanger configurations.⁵⁰

In addition, shape⁵¹ or surface^{17,51} configurations can be extruded or embossed to augment heat transfer by inducing turbulence. Plastics which evidence sufficient flexibility can be extruded to enable all methods of assembly. Standard design and fabrication procedures are available for guidance.^{52,53}

B. Plate-Type Heat Exchangers

An interesting alternative to shell-and-tube heat exchangers is the plate-and-frame unit introduced in 1923, and marketed by A.P.V. Company and by American Heat.^{54,55} This device is made of thin plates, which have various corrugations formed into the surface (see Fig. 48) to provide flow paths and rigidity against deflection due to pressure differences when the plates are clamped together (see Fig. 49).

Fig. 48. Plate Elements for Plate-Type Heat Exchangers. Reproduced from the A.P.V. Company Limited Publication A340v, 1978.

Fig. 49. Assembled Plate-Type Heat Exchanger. Reproduced from the A.P.V. Company Limited Publication A340v, 1978.

These units are generally limited to liquid-liquid duty and maximum pressure of about 2 MPa (300 psi) and 260°C (500°F), because of the distortion of large plates and the gaskets necessary to separate and seal the fluids from the atmosphere. The units are compact, compared to S&T exchangers, and can be engineered to suit the fluid characteristics by choice of the material and varying of the pattern of corrugations embossed in the plates. The critical Reynolds number for turbulent flow is low for the tortuous path. These results combine to give surface area-to-exchanger volumes between 50 and 100 ft²/ft³. The fluid holdup in plate-type exchangers is small compared to conventional shell-and-tube types.

Additional claims made for the plate-type exchanger are lighter weight, low fouling, ease of inspection and maintenance, ease of rearrangement of surface for expansion or decrease of duty, and ease of having several heating or cooling duties performed in the same frame. Since the fluids are in true countercurrent flow, close approach temperatures (as low as 2°F) and a high heat recovery efficiency (up to 90%) are claimed. Where stainless steel or other high-grade materials are required, costs are favorable compared to S&T exchangers. Current size limitations are about 15,000 ft² per unit.

The plate-type exchanger concept lends itself well to the use of plastics. Exotic construction materials can be avoided by coating cheaper metals with a corrosion-resistant plastic film on one or both sides.

As previously identified in Section III, a thin film of specialized plastic on a heat transfer surface can enhance the properties of the entire composite structure, and at relatively low cost. This film can be used to protect a metallic surface layer from corrosion and enhance boiling and condensation. Films can also be used on plastic cores as barriers against abrasion or corrosion of exposed metal filler.

Table 6 lists some typical commercially available barrier films and their costs and permeability.³ Also included is epoxy, which is not an extrusion, as are the others, but rather a coating used extensively for its chemical resistance, toughness, and outstanding adhesive properties.

TABLE 6. Materials--Commercially Available Barrier Coatings and Films

	Cost, \$/ft ²	Cost, 1 mil Equiv	WVTR ^a	CO ₂ ^b
1-mil Epoxy	0.0062	0.0062	0.10	0.4
3/4-mil Polyester ^c	0.0091	0.0091	0.59	8.6
1/4-mil Saran				
1/2-mil Polyethylene ^c				
1/3-mil Aluminum Foil	0.0140	0.0076	0.05	0.39
1-mil Polyethylene				
1-mil Cellophane ^c	0.0089	0.0067	0.24	0.39
1-mil Saran				
5/8-mil Oriented Polypropylene ^c	0.0048	0.0059	0.25	0.5
3/16-mil Saran				
1-mil Polyethylene ^c	0.0128	0.0064	0.17	0.4
1-mil Cellophane				
1 1/2-mil Polyethylene	0.0147	0.0074	0.18	27.8
1/2-mil Polyester				
10-mil HD Polyethylene			0.03	64.4
1-mil HD Polyethylene	0.0019	0.0019	0.30	580.0
1-mil Saran	0.0099	0.0099	0.20	3.8
1-mil Polyester	0.0084	0.0084	0.70	15.0
1-mil Aluminum Foil	0.0065	0.0065	0.025	0.15
1-mil Cellophane	0.0053	0.0053	0.40	0.4
1-mil Oriented Polypropylene	0.0040	0.0040	0.31	0.5
1-mil Vinyl	0.0040	0.0040	0.70	20.0
1-mil LDPE		0.0013		

^aWater vapor transmission gm/100 in.²/day/mil

^bcc/100 in.²/day/mil thk/atm at 25°C.

^cComposite costings.

Epoxy resins are used to coat pipes, structural steel (such as heat exchanger shells and supports) and concrete, where resistance to water and corrosion is desired. Coatings of epoxy are applied on interiors of high-density polyethylene gasoline tanks and pipes to prevent permeation. Water-based epoxy systems are being used to replace conventional solvent-based epoxy coatings. Curing of these materials occurs without production of volatile by-products.

Among the techniques available for coating sheets or fabricated surfaces are dipping, spraying, and fluidized-bed coating. Electrostatic techniques can be used to remove excess material or to build up additional amounts on corners or edges.

As an alternative to laminating metallic cores for plate-type exchangers, the plates could be constructed of a plastic matrix containing additives or fillers for enhancing structural and thermal properties. In this application, many unique fabrication properties of plastics are apparent. For example, corrugated plates can be fabricated by injection, blow, compression, and transfer molding; pressing, hot stamping, embossing, and vacuum forming; and rolling, punching, and thermoforming. Additionally, the plates can be designed and fabricated to implement self-gasketing features of plastic materials.

It is of interest to consider the limit of the plate exchanger as a thin film and note the early application of film materials to low temperature-low pressure applications, such as in desalination.^{56,57} Such thin-film exchangers can be easily and inexpensively fabricated by gluing or welding, using several possible processes.

Examples of processes for fabrication of heat exchangers from thin plastic sheets and tubes is given in Refs. 56 and 57 (historically among the first references) as well as Refs. 3, 5, 46, and 47. Some estimates of materials, fabrication and assembly costs are given in some of these references.

C. Spiral-Type Heat Exchangers

The spiral- or helical-type heat exchanger permits two streams to flow countercurrent in concentric curved paths. There are two general configurations: the first features spirally wound tubes, generally connected by manifolds (e.g., Graham Manufacturing Co.), designed to ensure constant velocities consistent with compactness and accommodation of differential expansion. Current sizes are limited to about 500 ft² and 300 psig (shell-side). In some cryogenic exchanger applications, a combination of countercurrent flow and crossflow is achieved, yielding small differences in temperatures; for example, the Studsvik plastic exchanger² shown in Fig. 50.

The second configuration features spirally wound plates with relatively long passages (e.g., American Heat Co.). Turbulence occurs at a lower Reynolds number than in straight tubes. The compact spiral-plate design can provide 1000 ft² of surface in a unit 42 in. in diameter and 60 in. long, or about 20 ft²/ft³. Each path of the exchanger is accessible by removing the respective cover. Limitations are currently 2000 ft² in a single unit, and 1 MPa (150 psig) and 232°C (450°F) due to gasket limitations.

Both spiral-tube and spiral-plate heat exchangers lend themselves to adaptation in plastic materials. As shown in Fig. 51, the counter-flow tubes can be coextruded (or otherwise fabricated), using an augmented material for the heat-transfer wall, and a less-expensive material for the remainder of

the channel. The spiral-plate exchanger could be made easily of coated metal or solid plastic, and, where required, openings for inspection or cleaning could be self-gasketed.

Fig. 50

Schematic of Water/Air Heat Exchanger
(Reverse Flow Directions Possible)

Fig. 51. Configuration for Coextruded Heat Exchanger Channels

Special considerations are imposed on plastic heat exchangers for automotive applications, which include exposure to atmospheric conditions; for example, sensitivity of some plastic materials to ultraviolet light and to ozone degradation. Additives are currently available to resist these causes of degradation in many common plastics.

A final configuration for application of plastic materials to recover low-temperature waste heat is the heat pipe.^{58,59} These high-conductivity structures are subject to the same corrosion and fouling problems as are conventional exchangers. A thin plastic coating on exposed surfaces or a shell of plastic, with augmentation for strength and conductivity, could effect a low-cost extension of heat pipe technology.

V. APPLICATIONS AND ENVIRONMENTS

This section provides a brief description of the many candidate heat sources and potential applications for recovery of low-temperature rejected heat by plastic heat exchangers. Some of the more promising source/application combinations are discussed in detail.

For most applications of interest, the nature of the plastic heat exchanger would fall into one of the following four categories:

(1) Gas/Gas Heat Exchanger--This would be an extended surface-type heat exchanger, regenerator, or recuperator, with a gas on both sides. Applications include (1) direct preheating of incoming combustion air for a boiler by cooling hot flue gases (Ljungstrom heaters); and (2) direct air-space heating by exhaust gases from an internal combustion engine.

(2) Liquid/Liquid Heat Exchanger--This would probably take the form of a typical shell-and-tube heat exchanger. Applications include (1) transport of heat between stages of a chemical process involving liquid streams; (2) recovery of heat from a boiler blowdown stream; and (3) low-temperature power-converting bottoming cycles using a variety of working fluids. Common practice is to circulate the more viscous fluid on the shell-side, and the more fouling fluid on the tube-side for ease of cleaning.

(3) Gas/Liquid Heat Exchanger--This would be a heat exchanger with extended surface on the gas (or heat source) side, which is usually the shell-side for lower pressure drop. The form of the extended surface probably would be different from a conventional fin-tube made of metal because of the relatively poor thermal conductivity of plastics. Applications include generation of low-pressure steam or hot water, using (1) boiler flue gases, or (2) internal combustion engine exhaust streams.

(4) Liquid/Gas Heat Exchanger--This would also be an extended surface heat exchanger, but with the liquid stream (heat source) inside the tubes and the heat transferred to a gaseous fluid such as air (similar to an auto radiator). Applications include (1) direct space heating from the water or lube oil coolant of an internal combustion engine, or (2) dry cooling tower elements.

In any specific application, the environment for the heat exchanger will be comprised of two sets of parameters; one for the heating fluid, and one for the heated fluid. The environment may be much more severe on one side of the heat exchanger than the other, in which case the corresponding set of parameters will control applicability and, hence, limit the life of the plastic material selected. Some examples of heat sources and of waste heat applications are presented in Table 7. Estimates of heat rejection versus temperature level for various industries are given in Table 8.⁶⁰

TABLE 7. Environments Imposed by Various Applications

Heat Sources	Heat Applications
<u>Engine Exhaust Gases and Cooling Systems</u>	<u>Low Pressure Steam Production*</u>
1. Gasoline	<u>Water Heating*</u>
2. Diesel*	<u>Air Heating*</u>
3. Dual-fuel	1. Space heating*
4. Gas turbine*	2. Air conditioning*
<u>Combustion of Fuels</u>	<u>Expansion of Cryogenic Gases</u>
1. Natural gas*	1. Ammonia
2. Coal combustion*	2. Methane
3. Coal-derived gases	3. L.N.G.
4. Diesel oils	<u>Water Purification</u>
5. Residual oils*	1. Desalinization
6. Refinery gases	(a) Evaporation
7. Heat-derived gases	(b) Freezing
8. Wood and other cellulosic products	2. Deionization
9. Waste incineration, including sewage sludge	<u>Agriculture</u>
10. Boiler blow-down	1. Algae production
<u>Dry Cooling Towers</u>	2. Greenhouse
1. Power plants	3. Soil-heating
2. Other environments	4. Aquaculture
<u>Primary Metals Industry</u>	<u>Low Temperature Power Generating Bottoming Cycles</u>
1. Steel-making	<u>Ocean Thermal Energy Conversion (OTEC) Cycles</u>
2. Aluminum	
<u>Chemical Industry</u>	
1. Cooling water	
2. Cooling tail-gases	
3. Cooling and/or heating reactors, ovens or furnaces	
4. Neutralization tanks	
<u>Food Industry and Pharmaceuticals</u>	
1. Brewing	
<u>Paper and Pulp Industries</u>	
1. Digestors	
2. Draft liquor processing	

*Discussed in further detail in this chapter.

TABLE 8. Estimated Heat Rejection for the Largest Fuel-Consuming Industries in the USA (1972-1973). Reproduced from J. Taborek, "Heat Exchanger Design," Proceedings of 6th International Heat Transfer Conference, Vol. 6, 1978.

Industry (or Process) and Operation	Rejected Heat, 10 ¹⁵ J/year (Estimated Accuracy Is ±20%)				Total All Ranges
	Below 100 C	100- 250 C	250- 800 C	800- 1800 C	
Chemical and Petroleum					
Petroleum	994	497	695	-	
Chlorine/caustic soda	254	51	-		
Ethylene/propylene	151	79	40	-	
Ammonia	115	139	40	-	
Oxygen/nitrogen	171	147	-	-	
Other	700	378	79	-	4530
Primary Metals					
Steel	79	397	596	795	
Aluminum	378	79	20	79	
Other	397	119	40	79	3058
Miscellaneous					
Cement	79	40	219	-	
Glass	60	12	87	36	
Paper	1391	755	119	-	
Food and Other	874	199	278	20	4169
GRAND TOTALS	5643	2892	2213	1009	11757
Ratio to Grand Total Σ _T	0.48	0.25	0.19	0.09	

Specific applications discussed here are characterized in terms of (1) pressure, (2) temperature, and (3) chemical composition and reactivity of the fluids. Because of the generally low strength of plastics at elevated temperatures, practical applications are limited to cases where the pressure is no more than a few atmospheres and temperatures are below 177°C (350°F).

For example, an inlet temperature of 177°C (350°F) for the heating (source) fluid and a practical heat exchanger ΔT of 38°C (100°F) would limit the exit temperature of the heated fluid to $\sim 120^{\circ}\text{C}$ (250°F). Under these conditions, low-pressure steam at ~ 0.1 MPa (15 psig) could be generated in a plastic heat exchanger recovery system. In a counter-flow heat exchanger with inlet water at 21°C (70°F), the exit temperature of the source fluid could be as low as 77°C (170°F).

The chemical compatibility of plastics is good with a wide variety of fluids and chemical agents and is, in general, superior to metals like carbon steel or aluminum. This is one of the compensating advantages for plastics in the present context: they can be used in corrosive environments which would have deleterious effects on performance and design lifetime of carbon steel components. Accordingly, the environments discussed herein are defined in terms of constituents in the fluids which would adversely affect conventional metals used for construction of heat exchangers.

The following sections characterize some of the more common heat-source environments, such as internal combustion engines and fossil-fuel-fired boilers, as applied to the generation of low-pressure steam, hot water heating, and air conditioning (heating and cooling). Some of the less common environments listed in Table 7 require special considerations and detailed discussions which are beyond the scope of this report.

A. Diesel Engines

Table 9 gives a breakdown of the heat recoverable from a typical large four-cycle diesel engine operating at full power. Practically all heat rejected to the cooling water and the lube oil of these engines can be recovered in suitable heat exchangers. However, only a portion of the available heat in the exhaust gas is usually recoverable because the temperature of the exhaust gas from the heat-recovery unit must be maintained above $\sim 160^{\circ}\text{C}$ (325°F) to provide an adequate margin of safety from corrosive effects at temperatures below the sulfuric acid dew point.

TABLE 9. Heat Recoverable from Typical Large Four-Cycle Diesel Engine at Full Rated Power
(Reproduced from Ref. 61)

	Heat Available		Heat Recoverable	
	Btu/Bhp-hr	Percent	Btu/Bhp-hr	Percent
Brake work	2545	37.9	2545	37.9
Jacket water heat	1125	16.8	1125	16.8
Lube oil heat	270	4.0	270	4.0
Exhaust gas heat	2270	33.8	1638	24.4
Radiation, friction, etc.	505	7.5	0	0
Heat rate	6715	100.0	5578	83.1

Table 9 indicates that if exhaust gas, jacket water, and lube oil heat is recovered, the useful energy will be increased from 37.9 to 83.1% of the total heat input value of the fuel. If the lube oil heat is not recovered, the overall utilization factor will be about 79.1%.⁶¹

1. Exhaust Gas Heat

The exhaust gas heat recovery for the engine heat balance given in Table 9 was based on a mass flow rate of 12 pounds per brake horsepower-hour and an exhaust gas temperature of 457°C (855°F). Although the temperature of the exhaust gas following the supercharger is variable, depending on the engine make, it generally would be from about 340 to 480°C (650 to 900°F) at full load. At part load, the exhaust gas temperature and the gas mass flow rate decrease, and the heat recoverable can be estimated, using the equation:

$$Q = W_{ex} C_p (T_1 - T_2), \quad (20)$$

where

Q = heat recoverable, Btu/hp-h (kW/s),

W_{ex} = exhaust gas flow rate, lb/hp-h (kg/kJ),

C_p = average specific heat of gas, Btu/lb-°F (kJ/kg·°C),

T_1 = inlet exhaust gas temperature, °F (°C),

T_2 = outlet exhaust gas temperature, °F (°C).

Figure 52 gives exhaust temperatures from conventional four-cycle diesel and spark-ignited gasoline engines. Using these data, Eq. (20) and lower limit temperatures of 160°C (325°F) and 77°C (170°F), Segaser⁶¹ calculated the percentage of total fuel input heat recoverable from the exhaust gas of a conventional diesel engine. The results are presented in Table 10. At full load, 24.4 to 31.6% of the fuel input heat may be recovered from the exhaust gas; this fraction decreases with decreasing engine load.

a. Operating Pressures

Typically, the heat exchanger pressure on the exhaust gas side would be in the range 4-15 in. H₂O, whereas the pressure on the liquid side could be up to 15 psig, if low-pressure steam is generated.

Fig. 52

Comparison of Exhaust Temperatures from Diesel Engine and Spark-Ignited Gas Engine. Reproduced from C. L. Segaser, "Heat Recovery Equipment for Engines," ANL/CES/TE 77-4, 1977.

TABLE 10. Percent of Fuel Input Energy Recoverable from Exhaust Gas of Conventional Diesel Engine at Part Load⁶¹

% Load	(1) Thermal Eff., %	(2) Heat Rate, Btu/Bhp-h	(3) W _{ex} , lb/Bhp-h	(4) T ₂ , °F	(5) T ₁ , °F		(6) C _p , Btu/lb-°F	(7) Heat Recovered, Btu/Bhp-h	(8) Heat Recovered, %	
									325°F	170°F
100	37.9	6715	12	850	325	170	0.260	1638	24.4	31.6
75	37.0	6878	12	733	325	170	0.258	1263	18.4	25.4
50	35.0	7271	12	617	325	170	0.256	897	12.3	18.8
25	25.0	10,180	12	500	325	170	0.255	536	5.3	10.0

(1) Thermal efficiency, E_{th} , from Fig. 53.(2) Heat rate, H.R. = $2545/E_{th}$ = $2545/(1)$.(3) W_{ex} = exhaust gas flow rate.

(4) Exhaust gas temperature from Fig. 53.

(5) Exit gas temperature from recovery boiler.

(6) Average specific heat of gas at $T_2 + T_1/2$.(7) Heat recovered = $W_{ex}C_p(T_2 - T_1)$ = (3) x (6) [(4) - (5)].(8) Percent of fuel input heat recovered = $(7)/(2)$ x 100.

b. Chemical Composition of Gases

The chemical composition of the exhaust gas from the diesel engine is a strong function of the following parameters:

- (1) Sulfur content of fuel.
- (2) Type of engine and amount of excess air.
- (3) Fractional engine load during operation.

A recent report⁶³ has presented results of detailed studies on the chemical composition of exhaust gases from diesel engines. These results are summarized in Table 11. Basically, they show that a reasonably accurate prediction of the total sulfur oxides in diesel exhaust can be obtained from a calculation based on the initial sulfur content of the fuel. Some difficulty was experienced in measuring the sulfur oxide content of gases from very low sulfur fuels (0.08%); therefore, these measurements are not reliable. An important parameter in measuring the corrosion potential for sulfur oxides in the engine exhaust gas is the fraction of SO_2 converted to SO_3 . However, the SO_2 can be dissolved in condensed moisture, especially in the presence of sulfuric acid, and form sulfurous acid which is corrosive as well as oxidizable to sulfuric acid by the excess oxygen in the exhaust gas.

As shown by the data in Table 11, this fraction of the SO_2 converted to SO_3 varies between approximately 2 and 8%; hence, the sulfuric acid concentration would be based on this fractional conversion of the sulfur oxides. It was found that the corresponding sulfuric acid dew points ranged between 112 and 117°C (234 and 242°F); also, that these temperatures could be predicted reasonably well by a calculational model. Measured and calculated dew points are summarized in Table 11.

TABLE 11. Diesel Exhaust Analysis Summary. Reproduced from R. Silvestrini,
 "Phase I Topical Report Exhibit III-B Heat Exchanger Fouling and Corrosion
 Evaluation," Garrett AiResearch Manufacturing Company of California,
 Report 78-15161(1), 1978

Site	Unit No.	Engine	Cyl.	Load [†] kW	Exhaust [†] Temp. °C (°F)	O ₂ [†] (%)	CO ₂ [†] (%)	H ₂ O [†] (%)	DuPont	Wet Chemistry			Predicted SO _x Dry at 3% O ₂	Dew Point		P.R.T. Meas. [†] °C (°F)	Chlorides [†] ppm	Smoke [†]	Solid Part. [†] ng/J (lb/MMBtu)
									SO ₂ ppm Dry at 3% O ₂	SO ₂ ppm Dry at 3% O ₂	SO ₃ ppm Dry at 3% O ₂	Calc. °C (°F)		Meas. [†] °C (°F)					
1. Fuel No. 2 Ash ≤0.001% Sulfur 0.40%	12	EMD	12	950	327 (620)	14.8	4.3	5.5	605	(172)	2.8	8.	236	117 (243)			0.16	4.5	12.6 (0.029)
				1450	382 (720)	12.6	5.7	5.2	394	(182)	2.7	5.6	237	(242)			0.91	6.5	18.2 (0.042)
	8	EMD	15	1378	360 (680)	9.4	7.5	7.0	315	-	0.89	2.9	236	113 (236)			0.80 1.0 ^{††}	9.5	39.7 (0.091)
2. Fuel No. 6 Ash 0.002% Sulfur 0.80%	6	DE	16	5830	462 (845)	11.0	6.8	5.2 (4.2°)	483	458	5.3	13.4	472	120 (249)	177 (242)	104 (220)		5.5	59.0 (0.14)
				3000	421 (790)	12.4	5.6	4.5 (3.9°)	487	494	3.5	9.1	467	116 (241)	112 (234)	99-102 (200- 215)	5.4	8.0	73.8 (0.17)
	5	DE	16	5870	416 (780)	11.8	6.3	4.9 (4.2°)	449	444	3.7	8.8	471	117 (242)	114 (238)	102 (215)	2.9	5.0	80.6 (0.19)
				5870	241 (465)	12.2	5.9	4.8	503	438	3.1	6.6	470	117 (242)	116 (240)	102 (215)		5.0	28.4 (0.07)
																	1.2 ^{††}		
3. Fuel No. 2 Ash ≤0.001% Sulfur 0.08%	7	CB	20	5600	442 (830)	11.3	6.4	7.2	201	38	0.98	1.85	44	113 (235)	98 (209)	-	4.3§	5.0	9.8 (0.023)
				3500	407 (765)	12.3	5.7	5.3	220	* 43	1.7	3.6	44	113 (236)	97 (207)	-	1.6§	4.5	7.4 (0.017)

() Shell-Emeryville Method.

[†] Measured values.

[§] Corrected for fuel ambient chloride levels.

^{*} Corrected moisture

^{††} Ambient Chloride level (apparently not affecting measured exhaust chloride levels).

Measured data on total particulates in the exhaust gas are also summarized in Table 11. The solid particulates varied from 0.017 to 0.19 lb/10⁶ Btu. It was found that the fuel-ash content accounts for less than 2% of the particulate matter in the moderate sulfur fuel; the major source is believed to be carbon from incomplete combustion. In most cases, because of more efficient combustion, the total particulate loading decreased with engine load. The majority of particles (72-78%) were smaller than one micrometer.

Chlorides in the exhaust gas can originate from the fuel (as a contaminant) and from the combustion air. The range of measured chloride concentration in the exhaust was 0.16 to 9.0 ppm, and was primarily influenced by varying levels of chloride in the combustion air. These chlorides would pose a stress-corrosion problem for stainless steel components in heat exchangers.

Measured data on CO₂, O₂, and moisture content of diesel exhaust gas are also presented in Table 11. The oxygen decreases while carbon dioxide and moisture increase with increasing load. These observations are consistent with the normal combustion characteristics of diesel engines: oxygen is consumed in the combustion of more fuel to produce more carbon dioxide and water.

c. Corrosion and Fouling

As stated, the main source of corrosion in diesel exhaust is sulfuric and other acids from oxidation of sulfur in the fuel. For this reason, (1) metallic heat exchangers have not performed well; and (2) it is anticipated that the unique corrosion resistance of plastic materials can be used in an exchanger to successfully recover the heat rejected in these adverse environments.

The severity of heat exchanger corrosion and fouling depends upon the design and operating characteristics of the waste heat recovery system and the diesel engine. In a normal constant-speed engine, the following factors are involved in considering whether the operating conditions are better or poorer as a function of load:⁶²

- (1) Excess oxygen (varies inversely with load).
- (2) Water content (varies directly with load).
- (3) Gas temperature (varies directly with load).
- (4) Particulates (engine specific).
- (5) Heat removal (should vary with load).

As the load decreases, the SO₃ concentration appears to increase slightly but the water decreases, with the net result that the sulfuric

acid dew point remains constant or is slightly reduced. Therefore, if the heat-removal rate through the heat exchanger can be regulated to ensure a controlled exit temperature, the sulfuric acid condensation can be maintained constant by varying the engine operating conditions. However, if the heat-removal rate must be constant for external process or heat-utilization reasons, the sulfuric acid condensation rate will increase at reduced engine loads, primarily due to the lower exit gas temperatures. These observations emphasize the need to concurrently consider engine load variations, acid dew point, and heat exchanger configuration in a low-temperature heat recovery system.

Fouling problems caused by particulate matter are of equal concern in heat exchanger design; therefore, there is considerable interest in the dependence of fouling on the type and sulfur content of diesel fuel. The findings of exhaust gas emission studies indicate a general range of particulate emissions varying with sulfur content, and a large portion of particulates that are less than one micrometer in size. These emissions appear to depend on sulfur content and engine load, but more likely depend on the engine age and state of repair. Emissions from a naturally aspirated engine increase with load because the air/fuel ratio decreases; in a turbocharged engine, the reverse may be true.

In the present application, a low final exhaust gas exit temperature of 77°C (170°F) implies there will be considerable condensation and sulfuric acid in the plastic heat exchanger. Therefore, design features will have to be incorporated to inhibit formation of such material or to permit periodic removal to preserve the heat transfer capabilities of the unit.

2. Heat from Engine Cooling Water

From about 15% to as high as 30% of the total heat supplied to a diesel engine is rejected to the jacket cooling water, and essentially all of this heat can be recovered.⁶¹ In the full-load heat balance given in Table 9, 16.8% of the heat supplied is rejected to the cooling water. In turbocharged engines, heat rejection to cooling water can be as high as about 30%. If the engine is cooled by a forced-convection heat exchanger system, the jacket water is normally set at 82°C (180°F). Under these conditions, a plastic heat exchanger with an overall ΔT of 17°C (30°F) would permit secondary hot water production at about 65°C (150°F).

One of the most popular heat-recovery systems for internal combustion engines utilizes the principle of ebullient cooling for the engine. This system involves natural circulation of the jacket water at or near saturation pressure, with engine cooling accomplished by vaporization of the water. Some benefits of ebullient cooling are:

- (1) Elimination of the jacket water circulation pump.

- (2) Extended engine life (A 2-3°F differential between inlet and outlet promotes uniform temperature throughout the engine).
- (3) Heat is recovered directly in the form of low-pressure steam (up to 15 psig).

If such a system is used in conjunction with an internal combustion engine, the plastic components would be in the external steam separator which is required to produce reasonably dry process steam. This would appear to be a relatively straightforward and advantageous use of plastics and, as stated earlier, practically all of the engine cooling-jacket heat can be recovered. Plastics would probably have an advantage over metals because of their better corrosion resistance to the engine coolants and the generally better properties of plastic surfaces with regard to fouling and adherence of boiler deposits.

B. Gas Turbines

The type of gas turbine prime mover most commonly used is a simple open-cycle, single-shaft machine (i.e., the compressor and turbine wheels are mounted on a common shaft). Shaft power to drive the generator is delivered through a combined output speed reducer and accessory drive section that reduces the relatively high turbine speed (20,000 to 40,000 rpm) to a value compatible with presently available generators. Exhaust gas is discharged at temperatures ranging from 260 to 650°C (500 to 1200°F).⁶¹

The open-cycle gas turbine is relatively inefficient, with full load thermal efficiencies varying from 10-25%, depending on its size and design. Theoretical efficiencies approaching 80% may be obtained by utilizing all of the recoverable exhaust energy as well as the shaft energy produced at a specific fuel heat rate.

1. Exhaust Gas Heat

Table 12 gives the estimated part-load heat recoverable from the exhaust gas of a typical 800-kW open-cycle gas turbine-generator set operating

TABLE 12. Heat Recoverable from Exhaust Gas of Typical 800-kW Turbine-Generator Set⁶¹

% Load	Net Load, kW	E _{th} , %	Heat Rate, Btu/kW-h	Input, Btu/h (x 10 ⁶)	T ₁ , °F	C _p Btu (lb)(°F)	W _{ex} /t, lb/h	Heat Recoverable			
								T ₂ = 325		T ₂ = 170	
								Btu/h (x 10 ⁶)	%	Btu/h (x 10 ⁶)	%
100	800	22	15,514	12.41	900	0.251	47,500	6.86	55	8.70	70
75	600	20	17,065	10.24	780	0.250	47,800	5.44	53	7.29	71
50	400	16	21,331	8.53	660	0.248	48,000	3.99	47	6.02	71
25	200	10	34,130	6.83	570	0.247	48,200	2.92	43	4.76	70

at sea level with compressor input air at 80°F. The available exhaust heat was calculated using the equation

$$Q = \frac{W_{ex}}{\delta} C_p (T_1 - T_2), \quad (21)$$

where

T_1 = turbine exhaust temperature, °F

T_2 = exit temperature from heat-recovery heat exchanger, °F (2 cases: 325°F and 170°F),

C_p = average heat capacity of gas, Btu/(lb)(°F),

and

$$\frac{W_{ex}}{\delta} = \text{flow rate of exhaust gas, lb/h.}$$

Table 12 shows that the recoverable heat at full load is increased from 55 to 70% of the full input heat value by reducing the exhaust discharge temperature from 325 to 170°F; at 25% load, the increase is from 43 to 70%. Thus, there is considerable incentive to recover this low-temperature heat.

2. Operating Pressures

One of the problems with designing a gas turbine exhaust heat-recovery heat exchanger is that it must handle a relatively high gas flow rate with a small temperature difference. Since turbine exhaust back-pressure increases fuel consumption about 1% for each additional four inches of water pressure at the turbine outlet, it is desirable to maintain as low a pressure drop through the heat exchanger as possible to avoid this penalty.⁶³ In practice, the range of exhaust pressures is between 1 kPa and 3.74 kPa (4 and 15 in. H₂O). Pressures below 1 kPa place too great a penalty on heat-recovery equipment, and pressures above 3.74 kPa can be reduced economically by designing the heat recovery system more liberally.

These requirements usually dictate high-profile, externally extended heating surfaces. Fins may be continuously fused or spot-welded to the base tube, but connections should be positive because they are subject to wide temperature gradients from fin tip to root attachment. Crossflow of gases across the tubes provides good heat transfer, and most gas turbine recovery systems have adopted this flow pattern. With fins perpendicular to the tube axis, a design is derived that has a single pass on the fin-side and multiple passes on the tube- or water-side. This produces good heat transfer rates with low resistance to the exhaust gas flow. A thin coating of plastic on such an extended surface would be acceptable, but it is doubtful that thermal conductivity could be sufficiently augmented in a plastic material to provide an efficient high-fin configuration.

As with the diesel engine exhaust heat recovery system, water pressure on the liquid side could be up to 0.1 to 0.14 MPa (15 to 20 psig), if low-pressure steam is generated.

3. Chemical Composition of Gases

Although specific detailed information on the composition of gas turbine exhaust gases has not been located, it is highly likely that the characteristics would be similar to diesel engine exhaust gases. However, because of the larger amounts of combustion air (excess air) used in a gas turbine, the sulfuric acid dew point would be lower (for the same fuel sulfur content and fractional conversion of SO_2 to SO_3).

4. Corrosion and Fouling

Basically, the same types of problems discussed previously for diesel engines would be expected in the plastic low-temperature heat exchanger for a gas turbine energy recovery system.

C. Combustion of Fossil Fuels

The general field of heat recovery from flue (stack) gases is broad. Applications in power plants for low-temperature recovery of feed water heat and combustion air preheat have long been established. Cut-off temperatures on the low side for the flue gases have been determined by combined considerations of economics and sulfur-related corrosion in the heat exchangers. Because the characteristics of stack gas from a fossil-fueled boiler are highly variable, depending on whether the fuel is natural gas, oil, or coal, the design of stack-gas heat recovery equipment poses a different set of problems and must be accomplished within a different set of constraining parameters for each type of fuel. To limit the scope of the discussion in this Section, attention is focused primarily on oil-fired boilers.

A chronic problem in oil-fired boilers is associated with corrosion and plugging of air heaters and other low-temperature surfaces, frequently limiting capacity and efficiency. In severe cases, this problem causes unscheduled outages which can be very serious and costly, especially in the electric utility industry.

The basic causes of low-temperature corrosion are well-known.⁶⁴ Briefly, almost all the sulfur in the fuel is oxidized to SO_2 during combustion and a small part, usually less than 1%, is further oxidized to SO_3 . As the flue gases cool in passing through the boiler, the SO_3 reacts with water vapor to form sulfuric acid. Flue gases containing even traces of SO_2 have high acid dew point temperatures, up to about 350°F, and liquid that condenses below the dew point contains significant concentrations of sulfuric acid. Thus, when air heater and economizer metal surface temperatures are below the acid

dew point, a film of sulfuric acid forms and causes rapid corrosion. Further, gas passages become plugged by fly ash adhering to the wetted metal surfaces.

Fig. 53. Variation in SO_3 Content of Flue Gas with Sulfur Content of Fuel in Experimental Furnace.⁶⁵

The practice is to design and operate air heaters and economizers with metal temperatures high enough to minimize condensation and subsequent corrosion and ash deposition. The result, however, is loss of a portion of the sensible heat in the flue gases which otherwise would be available. For this reason, other means of controlling corrosion are continually being sought to permit operation at lower stack outlet temperatures to improve thermal efficiency. Figures 53 and 54 show the measured variations in SO_3 content of flue gas with sulfur content of the fuel, and the variation in percent conversion of SO_2 to SO_3 with sulfur content of the fuel, respectively.⁶⁵ As

evidenced by the data in Fig. 53, even with a relatively low fuel sulfur content (0.5%), a significant quantity of SO_3 is produced in the flue gas. Several west coast utilities burning oil containing 0.3-0.5% sulfur are encountering both acid-smut fallout and air-heater corrosion. The severity of the acid problem, even with this relatively high-quality fuel, is indicated by the fact that one utility has to replace the cold-end basket in their Ljungstrom air heaters every two years. Admittedly, the severity of the acid problem is compounded by the fact that this utility deliberately operates with low stack temperatures to improve unit efficiency. As shown by the data in Fig. 55, an acid dew point temperature in the range of 260 to 300°F is indicated for a 2% sulfur fuel being fired under the conditions indicated in Figs. 53 and 54.

Fig. 54
Variation in Percent Conversion of SO_2 to SO_3 with Sulfur Content of Fuel in Experimental Furnace

Fig. 55

Relationship of Acid Dew Point with H₂SO₄ Concentration in the Vapor. Reproduced from "A Practical Review of the Cause and Control of Cold End Corrosion and Acidic Stack Emissions in Oil-Fired Boilers, J. E. Radway and L. M. Exley, ASME Paper No. 75WA/CD-8, 1975.

To alleviate the problems with acid and corrosion, it is common practice to operate a Ljungstrom air heater with exit temperatures no lower than 300°F. If such a system heats incoming air at 70°F, the average cold-end metal temperature is $300 + 70/2$ or 185°F, which is well below the acid dew point in many cases. A 40°F reduction in exit gas temperature from the heater is roughly equal to a 1% increase in fuel efficiency. Therefore, there is considerable incentive for reducing the stack gas temperature exiting from the heater. For example, a reduction from 300 to 150°F would yield a 4% increase in overall fuel efficiency.

Various mechanical techniques for preventing acid corrosion include insulating the ducts and fan housing, preheating the air to the heater bypass ducts, and tailoring boiler operating schedules to avoid low-load situations. Thus, the most likely application for plastic heat exchangers in the present context is to design the air preheater cold-end sections for fabrication with plastic materials. Here, as in applications discussed previously, the primary advantage offered by plastics is improved corrosion resistance. This would eliminate the need to periodically replace portions of the air preheater, and permit even lower exit flue gas temperatures than are currently practical.

An associated problem is that such cold surfaces can lead to "acid smut," even if acid-resistant materials eliminate corrosion problems.⁶⁵ Acid smut fallout is caused by condensation of sulfuric acid present in flue gases in cooler sections of the air heater ducts, collectors, fans, and stack. The condensed acid causes a conglomeration of oil, ash, and unburned carbon normally present in flue gases. Some of this conglomerate deposits on cold-end surfaces and may be blown out of the stack when the unit reverses and temperatures and gas velocities change. This could be a troublesome environmental problem, and means would have to be developed to cope with it in any plastic low-temperature air preheater design.

Some of the foregoing considerations would apply to boilers fired with natural gas and coal. Since natural gas generally contains no sulfur, the cold-end acid-corrosion problem is essentially eliminated. Coals, and particularly western coals, have high ash contents, which impose additional constraints and considerations involving operation of electrostatic precipitators. These factors must be considered in the design of any low-temperature stack gas heat recovery system for a coal-fired boiler.

Historically, heat recovery in industry has been practiced largely as a convenience when two streams, well-balanced in enthalpy change and temperature levels, could be logically paired. Operational problems, such as ease of startup and flexibility of operation are the primary objectives. High energy costs are now forcing designers to look closer at new heat-recovery applications. Increased amounts of heat are being recovered at smaller driving forces by using more heat exchanger surface and lower outlet temperatures. Since greater capital costs are required to fabricate heat exchangers with larger surface area and of more exotic metals to combat corrosion, the use of plastics could effect significant cost savings in manufacture and maintenance as well as improved performance. Plastic heat exchangers could well be the only equipment for heat-recovery applications in very adverse environments.

If savings from heat recovery are to be realized, applications must be able to "bootstrap" themselves up to operating conditions because of the dependence of one stream on heat recovered from another. This leads to greater feedback and less operational flexibility in exterior control of heat addition or rejection. The resulting system is less stable and harder to control. Transient behavior of such coupled systems must be studied carefully and, if serious problems are identified, the thermal networks may have to be partially decoupled by provision of external heat sources or sinks.

VI. SPECIAL DESIGN CONSIDERATIONS AND ALLOWABLE COSTS OF PLASTIC HEAT EXCHANGERS

Plastic heat exchangers have some unique characteristics which enable their application in environments which inhibit or prohibit the use of conventional metallic heat exchangers. However, these characteristics reflect special design and cost considerations. To best address these considerations, a realistic application of plastic heat exchangers to recover waste heat from a diesel-operated electric generator set is used as an example.

A. Design Considerations

The recovery of waste heat from the exhaust, cooling jacket, and lubrication system of a diesel-operated electric generator to produce pressurized hot water or steam for a power system in either the residential/commercial or industrial sector is specially suited to the use of plastic heat exchangers. The corrosion effects of acid condensate and fouling by unburned fuel and particulates presently inhibits the use of conventional metallic exchangers.

This type of a cogeneration application provides an assessment of the desirability of recovering the latent heat in the moisture produced in the combustion of sulfur-bearing fuels, as well as the sensible heat. Available data on diesel exhaust⁶² indicate that, if all the water vapor is condensed, 20-25% of the rejected energy could be recovered. This means that the unit exit temperature would be approximately 32°C (90°F).

Design constraints on the plastic heat exchanger would be the capability to handle the exhaust gas volume flow, temperature, and composition. Cyclic variations to meet the electrical demand and flow/mechanical-induced vibration must be considered. Additionally, plastic heat exchangers must be able to accommodate operational problems such as freezing, overheating due to loss of coolant, and maintainability.

One of the more significant constraints in utilizing diesel exhaust is the low pressure head available for driving the gas through the exchanger, typically about 0.3 lb/in.² (2 kPa). Since plastic heat exchangers require more surface area to offset the lower thermal conductivity, a fan also may be required to increase the pressure head.

As previously stated, condensation of potentially very corrosive liquids in the plastic heat exchanger is advantageous to increased diesel/generator efficiency. However, this corrosive condensate presents a problem in predicting the overall thermal/hydraulic performance. For example, the unique chemical nature of plastics usually encourages vapor to condense as drops rather than a continuous film; hence heat transfer coefficients may be up to twenty times higher than the film-wise condensation coefficients. A treatment of condensation heat transfer pertinent to heat exchangers is presented in the Appendix.

For the same reason, long-term fouling is not expected to be as extensive as in conventional metallic exchangers. Therefore heat transfer coefficients would be more likely to remain fairly constant throughout the design life of the equipment.

Pressure drop in the presence of all three phases (gas, condensate, and particulate matter) will be difficult to predict; and even more so in the special configurations required to enhance thermal performance. Conventional exchanger pressure drop calculation techniques may not be applicable.

Lastly, the unique flexibility of plastic tubes focuses increased attention on fabrication, vibration, and fatigue design problems, since tube oscillations will affect mechanical and thermal-hydraulic performance. Extreme flexibility of plastic tubes may invalidate analyses based upon small displacements.

B. Allowable Costs

One problem in developing any new technology is estimating whether it will be sufficiently cost-competitive that a market will, in fact, exist. More specifically, given that an acid-resistant, plastic heat exchanger is available to increase heat recovery from the exhaust gas stream of a diesel engine, how much extra would someone pay to have it installed to reduce exhaust gas temperatures to less than 150°C (300°F)?

1. Economic Analysis

Simple payback is one way of estimating the allowable cost. Simple payback is the amount of time, in years, required to retire an investment with returns from the investment; i.e., the investment divided by the annual return, or savings.

2. Added Capital Cost

Added capital cost associated with the investment is the amount paid for the heat-recovery boiler minus the cost of the displaced boiler capacity. Assume that the heat-recovery rate is given as ΔQ in 10^6 Btu/h, and C_{HX} and C_B are the installed cost of the heat exchanger and boiler, respectively, in units of $\$10^6$ Btu/h; thus

$$\text{Added Capital Cost} = (C_{HX} - C_B)\Delta Q.$$

3. Annual Savings/Costs

Annual savings and costs for the system include: (1) the fuel saved; (2) the boiler maintenance not needed; and (3) the cost of heat exchanger maintenance. Define the annual fuel savings as follows:

$$\Delta Q_A = 8760 L \Delta Q,$$

where ΔQ_A is the annual fuel savings (10^6 Btu) and L is the system load factor. If C_F is the fuel cost, $\$/10^6$ Btu and η_B is the boiler efficiency, the fuel saved per year is $(\Delta Q_A C_F)/\eta_B$.

Maintenance cost of the heat exchanger and boiler is given as either some fraction of the initial capital cost or some fixed amount per unit of capacity per year:

$$\text{Boiler maintenance} = C_{MB} \Delta Q,$$

and

$$\text{Heat exchanger maintenance} = \alpha_{HX} C_{HX} \Delta Q,$$

where

$$C_{MB} = \text{cost of boiler maintenance } (\$/10^6 \text{ Btu/h/yr}),$$

$$C_{HX} = \text{capital cost of heat exchanger } (\$/10^6 \text{ Btu}),$$

$$\alpha_{HX} = \text{fraction of capital cost for annual maintenance.}$$

Thus,

$$\text{Total saving} = \frac{\Delta Q_A C_f}{\eta_B} + C_{MB} \Delta Q - \alpha_{HX} C_{HX} \Delta Q,$$

or substituting for ΔQ_A ,

$$\text{Total saving} = \left(\frac{8760 L C_f}{\eta_B} + C_{MB} - \alpha_{HX} C_{HX} \right) \Delta Q.$$

a. Simple Payback

The simple payback period is then given by the equation

$$P = \frac{C_{HX} - C_B}{\frac{8760 L C_f}{\eta_B} + C_{MB} - \alpha_{HX} C_{HX}}.$$

This equation can be arranged in a form which is directly usable here. That is, given the payback period, what is the value of C_{HX} ?

Accordingly,

$$C_{HX} = \frac{\left(\frac{8760 LC_f}{\eta_B} + C_{MB} \right) P + C_B}{(1 + \alpha_{HX} P)}$$

b. Example

Assume the following constants:

$$L = 0.6,$$

$$C_{MB} = \$75/10^6 \text{ Btu/h/yr},$$

$$C_B = \$1870/10^6 \text{ Btu/h},$$

$$\eta_B = 0.75,$$

$$\alpha_{HX} = 0.03.$$

These are, in general, the values used in Ref. 66. For this example, both the payback period (P) and fuel cost (C_f) are considered parameters. Using the above constants the permissible costs per unit of capacity of an exhaust gas plastic heat exchanger can be calculated. These costs are shown in Table 13.

TABLE 13. Permissible Unit Cost of Exhaust Gas Heat Exchanger (C_{HX}) in $\$10^6$ Btu/h

P (yr)	C_f (\$/10 ⁶ Btu)	
	1.0	3.0
1	8692	22914
2	15128	41573
3	21210	59786
4	26966	77023

In this example, the plastic heat exchanger will be used in series with a metal unit due to temperature limitations. The capital cost (based upon 1977 dollars) for a conventional heat exchanger recovery unit on a 4000-Bhp (2983-KW) diesel engine is given as⁶¹

$$C (\$/kW) = 580 W^{-0.555},$$

where

W = Engine rating (kW).

Thus, the cost of the exhaust gas heat recovery equipment is \$6.839/kW of installed engine capacity, or \$20,400 total.

Now assuming

$\dot{m}$ = Mass flow rate of exhaust gas = 47817 lb/h,

C_p = Specific heat of exhaust gas = $0.25 \frac{\text{Btu}}{(\text{lb})(^\circ\text{F})}$,

T_{in} = 855°F,

and

T_{out} = 325°F,

the heat recovered from the exhaust gas by a conventional unit is

$$\begin{aligned} Q_m &= \dot{m}C_p(T_{\text{in}} - T_{\text{out}}) \\ &= 47817(0.25)(853-325) \\ &= 6.312 \times 10^6 \text{ Btu/h.} \end{aligned}$$

The plastic unit further reduces the exhaust gas temperature to 150°F; thus

$$\begin{aligned} Q_p &= \dot{m}C_p(T_{\text{in}} - T_{\text{out}}), \\ &= 47817(0.25)(325-150), \\ &= 2.092 \times 10^6 \text{ Btu/h,} \end{aligned}$$

which represents approximately a 33% increase in energy recovered. This unit will not recover appreciable latent heat from condensation, but will be subjected to liquid sulfuric acid attack.

If the heat recovered by the plastic unit is to produce hot water at 57°C (135°F) from a 13°C (55°F) supply, the total area A_p required for the heat exchanger can be estimated by

$$A_p = \frac{Q_p}{U_p \Delta T_m},$$

where $\Delta T_m \equiv$ log-mean temperature difference, and $U_p \equiv$ the overall heat transfer coefficient.

Specifically for counterflow in a double-pipe heat exchanger,

$$T_m = \frac{(T_{\text{hot out}} - T_{\text{cold in}}) - (T_{\text{hot in}} - T_{\text{cold out}})}{\ln \left[\frac{(T_{\text{hot out}} - T_{\text{cold in}})}{(T_{\text{hot in}} - T_{\text{cold out}})} \right]}$$

in this case,

$$\Delta T_m = \frac{(150 - 55) - (325 - 135)}{\ln \left[\frac{(150 - 55)}{(325 - 135)} \right]} = 137^\circ\text{F};$$

thus

$$A_p U_p = \frac{Q_p}{\Delta T_m} = \frac{2.092 \times 10^6}{137} = 15270 \frac{\text{Btu}}{(\text{h})(^\circ\text{F})}.$$

Table 14 lists values for U_p (these values are parameterized since specific values are difficult to predict at this stage of the development), and the resultant plastic heat exchanger costs. The plastic heat exchanger costs were determined, using values listed in Table 13 for $\$3/10^6$ Btu-fuel and the calculated Q_p . The resultant unit costs were then simply divided by the calculated exchanger area A_p .

TABLE 14. Plastic Heat Exchanger Size and Permissible Costs as a Function of Overall Heat Transfer Coefficient, U_p , and Payback Period at a Fixed $\$3/10^6$ -Btu Fuel Cost

$U_p \left(\frac{\text{Btu}}{\text{hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}} \right)$	$A_p \text{ (ft}^2\text{)}$	Permissible Cost ($\$/\text{ft}^2$) and Payback Period (yr)			
		1	2	3	4
5	3054	15.70	28.48	40.95	52.76
10	1527	31.39	56.96	81.91	105.52
20	763.5	62.78	113.91	163.81	211.04
60	254.5	188.35	341.73	491.44	633.13

Inspection of Table 14 shows that even for conservative values of U_p , permissible costs appear high, with reasonably short payback periods. With fuel costs expected to escalate above $\$3/10^6$ -Btu, and improvements in plastic heat exchanger performance, greater fuel savings and reduced unit capital costs will be realized. In any event, it is reasonable to expect high economic gain from the recovery of rejected heat with significant increases in energy efficiencies using plastic heat exchangers.

The foregoing analysis assumes that all of the waste heat recovered from the engine exhaust gases can be used. This assumption may hold in industrial applications, with an essentially base-loaded engine and a continuing high demand for both electrical and thermal energy, but not in residential or commercial applications with time-varying electrical and thermal demands. For example, high electrical and thermal demands may be noncoincident, thus changing the kind of equipment needed (it may be possible to eliminate a fuel-fired boiler). Such systems must be examined on a case-by-case basis.

Admittedly, this analysis is quite simple; however, the dominating factor is the cost of fuel saved, which renders heat recovery from diesel engines a very attractive option.

VII. APPLICATION EXPERIENCE WITH COMMERCIALY AVAILABLE PLASTIC HEAT EXCHANGERS

Presently, there are only two commercial vendors of plastic heat exchangers: DuPont and Studsvik. Following is a summary of their experience with plastic units in various applications.

The DuPont Company introduced shell-and-tube heat exchangers made with Teflon tubing in May, 1965, following about two years of testing and evaluation. Teflon, which has superior nonstick qualities, eliminates fouling factors in the calculation of overall heat transfer coefficients in most applications. However, Teflon fluorocarbon resin has a low thermal conductivity. DuPont overcomes this problem by using thin-wall, small-diameter tubes.

The maximum recommended operating conditions for the DuPont units are shown in Fig. 56. Problems with these units in commercial service are claimed to be minimal and consist mostly of plugging of the small tubes by particulate matter, tube leaks, and O-ring failures. Plugging can be eliminated by filtering or by occasional backflushing. Tube leaks are generally attributed to excessive shell-side flow rates subjecting the tubing to inordinate hydraulic forces, or to cuts by large suspended solids, usually on the shell-side.⁷⁸

These exchangers can be returned to service by heat-sealing a short length of PTFE filler rod into the tube opening at each tubesheet using hot air from a plastic weld gun. Removal of the exchanger heads for access to the honeycomb tubesheets is the only disassembly required. Failed O-rings are usually caused by excessive deformation or chemical attack, but are easily replaced.

The unique flexibility of plastic tubes introduces a new variable in the design of tubular heat exchangers. Here, the degree to which flexible bundles

of heat exchanger tubes are redistributed by the forces of tube-side and shell-side flows is a function of the nature of containment and temperature feedback on tube properties. This interaction of flow and geometry introduces additional problems in the prediction of pressure drop and thermal performance. Incoming shell-side fluid tends to compress the tube bundle inward, whereas the radial forces of disengaging fluid at the exit are in an outward direction, providing an adverse feedback effect where the tubes compact and increase the pressure drop, causing further deformation, etc. Design equations and figures are given by Smith⁸¹ for these problems.

Fig. 56
Recommended Operating Pressure
for Teflon Heat Exchangers with
0.1-in. (2.5-mm) OD Tubes⁷⁸

For large numbers of tubes, the tube matrix is reinforced by diagonal laces of flattened PTFE tubes which restrain the tubes in smaller groups. The dominant pressure loss term is due to the tapes separating individual tubes, and includes a term to account for "bucketing" or cupping of the tapes.

An alternative method of spacing large numbers of tubes (up to 3100), is to use perforated spacer plates of PTFE positioned by fluorocarbon film-covered, stainless steel rods. The characteristics of such an assembly in crossflow are generally predicted by standard procedures, although sensitivity to tube flexibility and tube slack are noted.⁸⁰

The experience with DuPont plastic heat exchangers in over thirteen years of successful application to a wide variety of industrial problems is documented in Refs. 1, 42-44, and 77-81. Some of these applications are highlighted below.

(1) Pickling in steel manufacturing. In this application, the heat exchangers are subject to a 2-4% HF and 10-15% HNO₃ solution at 140-180°F. Plastic heat exchangers effect significant improvements over conventional units in maintenance and downtime. Major operating benefits are a more stable bath concentration, and up to 50% reduction in acid consumption.

(2) Heating of agitated reactor vessels. This application can include environments of high acid concentrations and crystalization. The flexibility and nonstick characteristics of plastic tubes under vibration induced by agitation essentially eliminated crystal formation and fouling.

(3) Heating and cooling of distilled water. Because of their adaptability to space limitations, plastic heat exchangers were selected to eliminate contamination of the distilled water during heating and cooling at temperatures up to 82°C (180°F) and pressure of 0.1 MPa (15 psig). A minimum of four years of maintenance-free operation is reported.

(4) Elimination of sulfuric acid corrosion. At least 2-4 years of maintenance-free operation has been reported by producers of chlorine and caustic soda. Due to the presence of sulfuric acid in these applications, the average life expectancy of a stainless steel shell-and-tube heat exchanger was 2½ years. The plastic heat exchangers have reduced overall capital costs as well as downtime.

Plastic heat exchangers manufactured by Studsvik² feature the spiral tube configuration and are essentially limited to liquid/gas applications. These units are fabricated from high-density polyethylene, cross-linked polyethylene, or polybutene, depending upon operating temperature requirements. Many thin-walled tubes are used to reduce the effect of increased plastic thermal resistance on overall performance. They present cost-effective alternatives to metallic exchangers operating in adverse environments.

Some applications presently using the Studsvik plastic heat exchangers are:

(1) Air conditioning using seawater. Salt water corrosion prohibited the use of metal heat exchangers, whereas plastic units have given good long-term service.

(2) Waste heat recovery. Heat is recovered from exhaust air containing corrosive vapors. Corrosion resistance and adaptability to space requirements made the selection of the Studsvik unit more favorable.

VIII. COMMERCIAL VENDORS OF PLASTIC MATERIALS

Any serious consideration of plastic materials in heat exchangers must include sources of raw materials and manufacturers to convert these materials into a product consistent with, or better than, design specifications.

A. Raw Materials

The following listing is limited to materials (and selected suppliers) which best suit the broad requirements of plastic heat exchangers. The selection criteria were good thermal/mechanical properties, availability, ease of fabrication, and compatibility with fillers to produce enhanced property composites.

For polyphenylene sulfide, the sole supplier is Phillips Chemical Company, a division of Phillips Petroleum Company. The trade name is Ryton, and the material is available in a form for molding and for coatings. No tubes are produced, but it is believed that thin-walled tubes could be produced.⁸²

The basic polyvinylidene fluoride resin is marketed principally under the trade name Kynar by the Fluorochemicals Division of the Pennwalt Corporation. The sourcebook for the plastic industry, Modern Plastics Encyclopedia⁸³ also lists the Kreha Corporation of America, and a number of fabricators from whom the material is available in sheet and tubular form, and for coatings.

Polytetrafluoroethylene (PTFE) is much more widely manufactured and marketed. A sampling of trade names and producers includes: Algaflon, by Montedison USA of Montecatini Edison; Fluon, by ICI United States of Imperial Chemical Industries; Fluorosint, by the Polymer Corporation of Reading, PA; Halon, by the Specialty Chemicals Division of Allied Chemical Corporation; Hostaflon, by the Foster Grant Company, a subsidiary of the Plastics Division of American Hoechst Corporation; and the largest-selling Teflon, by E. I. duPont de Nemours and Company.

B. Fabricated Units

Of the many fabricators listed in the Modern Plastics Encyclopedia, only two are known to fabricate and market plastic heat exchangers: Heat Transfer Products Division of the Industrial Chemicals Department of DuPont,⁴² and AB Atomenergi-Studsvik² in Sweden. Also, the assignees of Carlo Roma's patents^{4,46,47} are expected to manufacture or license their designs.

Phillips Petroleum Company, a producer of PPS plastic, and designer and fabricator of industrial heat exchangers, might be a logical organization to consider as a potential fabricator of plastic heat exchangers.

IX. RECOMMENDATIONS FOR FURTHER R&D EFFORTS

As this document was prepared, it became obvious that the advent of viable, commercial, low-temperature plastic heat exchangers posts a challenging research and development program. Accordingly, the following areas of specific activity are recommended and listed in the desired order of earliest implementation.

Initially,

(1) Develop techniques for compounding composite materials, and fillers in a plastic matrix. This will involve an assessment of thermal/mechanical properties as well as costs. A significant portion of this effort shall be accomplished under subcontract by or in collaboration with industry.

Concurrently,

(2) Establish applications of low-temperature heat recovery best suited for plastic heat exchangers. This will involve predicting operating environments, rendering conceptual designs, and performing corresponding costs analyses.

With a good understanding derived from the foregoing efforts,

(3) Improve existing, or develop new, techniques for fabricating plastic components with composite materials.

(4) Develop innovative designs for plastic units which take special account of tube or plate vibration, fastening or sealing methods, and nonconventional shapes. Again, industrial involvement shall be sought.

Lastly,

(5) Assess the environmental impact of low-temperature heat recovery using plastic units. This would probably be done for a specific design or series of designs evolved from the previous work. [For example, exhaust gas pollutants may be decreased by condensation in the recovery unit, but the corrosive chemical condensates could pose a disposal problem.]

APPENDIX

Condensation and Boiling Heat Transfer

The special treatment of condensation and boiling heat transfer deserves more attention, especially where plastic exchangers are employed for low-temperature heat recovery. This appendix presents a detailed study of prediction methods and available experimental data.

The interaction of temperature variation with individual heat transfer coefficients due to boiling and condensation, and their effect on the overall heat transfer coefficient and area is illustrated in Fig. A.1 for the case of condensation of a pure vapor.⁶⁰

Fig. A.1

Calculated (a) Temperature Profiles, (b) Shell-Side and Tube-Side Coefficients, and (c) Their Effect on Overall Heat-Transfer Coefficients and Total Area Required. Reproduced from J. Taborek, "Heat Exchanger Design," Proceedings of 6th International Heat Transfer Conference, Vol. 6, 1978.

Traditionally, condenser designs have been based on the heat transfer coefficient for condensation calculated by assuming heat conduction through a condensate film.⁶⁸⁻⁷⁰ The film thickness is determined by a force balance, where the shear force at the heat exchanger surface is equated to the sum of the gravity force on the liquid and the vapor shear force at the vapor-liquid interface. If the surface is curved or at an angle to the gravity force, appropriate components must be used, and the local coefficients must be averaged over the configuration.

The simplest solution is that of Nusselt⁷⁰ for laminar condensation of a pure vapor on a vertical wall, with negligible shear. This leads to an average heat transfer coefficient h_m , in terms of the film thickness δ , and liquid thermal conductivity k , in consistent units:

$$Nu_c = \frac{h_m \delta}{k} = \psi.$$

Here, Nu_c is the Nusselt group for condensation, and ψ is a function which depends on the condensate flow regime and the physical properties of the liquid. For the cited case of laminar flow with gravity dominating (negligible shear force):

$$\delta = C_1 (Re_c)^{1/3} [\mu_l^2 / \rho_l (\rho_l - \rho_v) g]^{1/3},$$

where

C_1 = a constant depending on the geometry, i.e., vertical or horizontal tubes. (Readers are referred to textbooks or handbooks for typical values.⁶⁸⁻⁷⁰)

Re_c = condensate Reynolds number, defined at $4\Gamma_c/\mu$, where Γ_c is the condensation rate at the bottom of the surface in kg/s·m (lb/hr·ft)

μ = viscosity

ρ = density

g = acceleration due to gravity,

and the subscripts l, v, and c denote liquid, vapor, and condensate, respectively.

The physical properties k , ρ , and μ for the liquid are usually evaluated at a suitable film temperature T_f ($T_f = T_{sat} - 0.75 \Delta T$, where T_{sat} is the saturation temperature at the local pressure, and ΔT is the temperature difference between the bulk vapor temperature and the condenser surface temperature).

Figure A.2 shows the geometry for condensation on the outside of horizontal tube bundles.⁷¹ In the laminar regime, the heat transfer on lower tubes depends on the condensate flow from above. The Nusselt derivation for the heat transfer coefficient for N tubes (h_N) in terms of a single tube (h_1) is

$$h_N = h_1 / N^{1/4}.$$

Fig. A.2. Flow in Horizontal Baffled Condenser with Vertical Baffled Cut; Schematic of Condensation on Horizontal Tubes in a Vertical Tube Row. Reproduced from *Heat Exchangers: Design and Theory Sourcebook*, Chapter 3, by J. Taborek, Afgan and Schlönder, eds., Scripta Publishing Company, 1974. (Copyright now held by Hemisphere Publishing Corp., Washington, D.C.)

Uncertainties in the design of shell-side condensers are introduced for large shell diameters by the effect of the number N of vertical tube rows in the horizontal laminar condensing regime assumed in the Nusselt model. This phenomena is poorly understood, even for moderate-sized exchangers, and the penalties compared to a single tube range from practically zero to the theoretical $N^{1/4}$ (Refs. 68-71).

For large-diameter exchangers, which might have as many as 100 vertical tube rows, the difference between the extremes is 70%. Even less experimental information, and no theoretical guidance is available for finned tubes. Further, the effect of baffle windows, in which the vapor flow becomes essentially parallel to the tube axis is known to the extent that it is dependent on the vapor velocity; dependable data on larger sizes of bundles are presently proprietary. Despite the high industrial importance of shell-side condensation, the conservative Nusselt solution is still generally used.

As the condensate flow increases, the film becomes turbulent ($4\Gamma/\mu > 2100$) and

$$\delta = [\mu_1^2/k_1^3\rho_1(\rho_1 - \rho_v)g]^{1/3},$$

and

$$\psi = C(Re_c)^{0.4}.$$

Based on experimental information from large-scale heat exchangers, Taborek⁷¹ recommends the inclusion of the Prandtl number Pr_1 for the liquid:

$$\psi = C Re_C^{0.47} Pr_1^{0.4},$$

where

$$Pr = \frac{C_p \mu}{k}.$$

When the vapor shear is sufficiently large, as in the entrance region of the exchanger, it may control the film thickness by producing surface waves, tearing off and entraining the condensate as a spray. The operating conditions for this regime are indicated in Fig. A.3 for a typical horizontal condenser with vertical baffle cuts to facilitate condensate drainage. Here, the Nusselt solution is clearly invalid; measured coefficients are 20 times higher. Taborek recommends the equation

$$Nu = 0.6 Re^{0.6} Pr^{0.4} \xi$$

where

$$\xi = 0.5(\rho_e/\rho_v + 1)^{1.2}.$$

Fig. A.3

Schematic Relations for Condensation Outside Horizontal Tubes. Reproduced from *Heat Exchangers: Design and Theory Sourcebook*, Chapter 3, by J. Taborek, Afgan and Schlünder, eds., Scripta Publishing Company, 1974. (Copyright now held by Hemisphere Publishing Corp., Washington, D.C.)

In an actual exchanger, the effects of vapor bypassing in the clearances between bundle and shell must be accounted for, by adjusting the velocity in Re and the coefficient 0.6, multiplying the $Re^{0.6} Pr^{0.4} \xi$ term, by the use of experimental data.

Figure A.3 shows, qualitatively, the relative effects of increasing condensate flow rate W_c , pressure P , number of tube rows N , and Prandtl number Pr on the condensation coefficient h_c . The regimes depicted are:

- (A) gravity controlled laminar flow, (B) gravity induced turbulent flow, and
- (C) vapor shear controlled turbulent flow.

The Nusselt solution does not include surface tractive forces generated by surface tension. As illustrated in Fig. A.4, these forces manifest themselves in terms of condensate retention in the gaps between fins, and render the surface ineffective.

Fig. A.4

Condensate-Retention Effect As Function of Fin Geometry and Fluid. Reproduced from *Heat Exchangers: Design and Theory Sourcebook*, Chapter 3, by J. Taborek, Afgan and Schlönder, eds., Scripta Publishing Company, 1974. (Copyright now held by Hemisphere Publishing Corp., Washington, D.C.)

The relevant parameters, experimentally confirmed, are surface tension, density, and fin height-to-spacing ratio. The effect can be very significant for water and other fluids with a large surface tension-to-density ratio, and for tubes with high fin density. The same phenomena occur on bare tubes, but to a lesser extent.

Although the previous discussion has assumed film condensation, the unique chemical nature of most plastic materials encourages the vapor to condense as drops. Consequently, a large part of the surface remains uncovered by the insulating liquid film and the heat transfer coefficient can be twenty times higher. Use of film condensation coefficients is recommended as a conservative design basis. (Ref. 72 is recommended as a current review of the subject.)

Effects of Noncondensable Gases

A most serious design problem for the selected application is the presence of large amounts of inert gases--principally oxygen, nitrogen, and carbon dioxide--from the combustion process and excess air. In condensation of water vapor with only 2.89% air by volume, the heat transfer coefficient was reduced by a factor of >3 . This effect can be ascribed to the introduction of a diffusional or mass-transfer resistance to the transport of heat to the cool surface.⁷³

This can be understood by considering Fig. A.5. At a finite heat transfer rate, a temperature profile is established across the condensate film with the vapor-liquid interface at some temperature $T_{if} \geq T_s$, the saturation temperature at the local pressure. In the absence of inert gas, the interface is at the saturation temperature.

Fig. A.5. Origin of Noncondensable Resistance

The presence of a noncondensable gas lowers the partial pressure of the vapor and, therefore, the saturation temperature of the vapor is in equilibrium with the condensate. The motion of the vapor toward the condenser surface implies a similar bulk motion of the inert gas. At the condensing interface, the vapor is condensed at temperature T_{if} and removed, by gravity, vapor shear or interfacial tension, as a liquid. The gas concentration will increase at the interface until it is sufficiently high, at equilibrium, to diffuse away at the same rate that it is convected to the surface.

If the concentration (mole fraction) of the gas is Y_g and the local pressure is P , the partial pressure of the bulk gas is

$$P_{g\infty} = Y_{g\infty}P,$$

and the partial pressure of the bulk vapor is

$$P_{v\infty} = (1 - Y_{g\infty})P - Y_{v\infty}P,$$

where the subscripts g , v , and ∞ represent gas, vapor, and the bulk stream, respectively.

Because of the convective flux, the partial pressure of the gas at the condensate film interface (denoted by subscript if) is $P_{gif} > P_{g\infty}$. From Dalton's law of partial pressure

$$P_{gif} + P_{vif} = P;$$

therefore, $P_{vif} < P_{v\infty}$.

If thermodynamic equilibrium exists at the interface,

$$P_{vif} = P_{sat}(T_{if}),$$

where $P_{sat}(T)$ is the saturation temperature of the vapor at temperature T . (This nonlinear relationship is more complicated in the case of sulfuric acid condensates.⁶²) Because of the inert gas, the available temperature drop across the condensate film is reduced from $(T_\infty - T_s)$ to $(T_{if} - T_s)$, where T_∞ is the bulk temperature of the condensing vapor-gas mixture.

The total heat transferred is, therefore, the sum of the sensible heat transfer through the inert gas film and the latent heat transfer via condensation. In equation form, referring to Fig. A.5,

$$h_c(T_{if} - T_s) = U(T_{if} - T_c) = h_g(T_\infty - T_{if}) + KM_v L(P_{v\infty} - P_{vif}),$$

where h_c is the appropriate condensation coefficient, U is the usual overall heat transfer coefficient,⁶⁸⁻⁷¹ h_g is the convective heat transfer coefficient for

the inert gas, K is the mass transfer coefficient, L is the latent heat of condensation, and M_v is the molecular weight of the vapor.

The convective heat transfer coefficient and mass transfer coefficient are usually known for the geometry and flow conditions as Colburn j factors^{68-70,73} for heat transfer:

$$\frac{h_g}{(C_p)G_g} \left[\frac{(C_p)_g \mu_{gv}}{k_g} \right]^{2/3} = j_H \left(Re = \frac{DG}{\mu_{gv}} \right),$$

where G is the mass velocity, and D is the characteristic diameter or length ($4 \times$ flow area/wetted perimeter) and for mass transfer:

$$\frac{K}{M_m} \left[(P_{g\infty} - P_{gif}) / \ln(P_{g\infty}/P_{gif}) \right] \left(\frac{\mu_{gv}}{\rho_g \Delta} \right)^{2/3} = j_M(Re),$$

where M_m is the mean molecular weight of the gas-vapor mixture, and Δ is the diffusivity of the gas-vapor mixture.

The calculational procedure requires a double-trial and error procedure.⁷³ At each station along the heat exchanger, the values of the physical properties M_m , G , Re , j_H , j_M , M_m , $P_{g\infty}$, h_g , and h_c (or U) are known or can be calculated using an assumed interface temperature T_{if} . From the assumed value of T_{if} , $P_{vif} = P_{sat}(T_{if})$ also can be calculated and, from this, P_{gif} . The mass transfer coefficient can be calculated and the total heat balance checked. Values of T_{if} are iterated until the heat balance checks; and the heat transferred to that point, coolant temperature rise and surface area needed can all be calculated from the usual heat balances.

For example, the area can be obtained by graphical or numerical integration of

$$A = \int_0^q \frac{dq}{U(T_{if} - T_c)},$$

where q is the heat transferred. If there is a phase change in the coolant (boiling), this must be accounted for in the coolant heat balance.

It should be apparent that a computer program to perform such calculations must be sophisticated and should be combined with a comparable physical properties program and calibrated against the performance of similar exchangers. A number of commercially and academically based programs is given in Ref. 67. The most sophisticated programs are believed to be those from Heat Transfer Research, Inc. and from Heat Transfer and Fluid Flow Services in England.

In calculating the condensation behavior, it is generally assumed that the vapor and liquid flow colinearly and in intimate contact so that equilibrium is maintained at all points between contacting streams. If, however, as shown in Fig. A.2 for a horizontal shell-side condenser, the condensate drops out of the vapor and flows to the exit without further interaction, the remaining vapor is depleted sooner and less surface is needed. Lack of equilibrium at the vapor-liquid interface is a problem only at subatmospheric pressures or with liquid metals.

Two-dimensional computer programs for the prediction of flow and heat transfer in condensation in the presence of inert gas have been developed at the National Engineering Laboratory in England.⁷⁴ Results are given which indicate that these computer models are now capable of predicting many of the effects of inert gas concentrations found in unbaffled condensers,⁷⁵ but the large computer times involved preclude their use for practical design. Some of the results of the English work on condenser design are available in computer design codes now commercially available through Heat Transfer and Fluid Flow Service (HTFFS), Harwell, England.⁶⁷

An interesting side-effect of noncondensable gases is described by Wild.⁷⁶ In the absence of noncondensables and with a layer of subcooled liquid in the bottom of a heat exchanger shell, Wild found liquid oscillations which led to mechanical damage. These were ascribed to a small surge in vapor velocity depressing liquid downward and making submerged surface available for condensation. This leads to a rapid depressurization and a rise in liquid level reducing condensation area and initiating repeat of the oscillations. Small amounts of noncondensable gas completely eliminated this effect.

Pressure drop on the condensing side reduces the final condensing temperature and the mean temperature difference, leading to a larger unit, and should always be checked. In designs requiring a close temperature approach between the entering coolant and exiting condensate (subcooled or not) underestimation of pressure drop on the condensing side may lead to a heat exchanger which cannot achieve its specified terminal temperatures. As pressure drop in condensing two-phase flows is relatively inaccurate, the designer must consider carefully the consequences of underestimating the pressure drop.

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BIBLIOGRAPHY

1. R. E. Githens, W. R. Minor, and V. J. Tomsic, "Flexible Tube Heat Exchangers," *Chem. Eng. Prog.* 61(7), 55-62 (July 1965).
2. AB Atomenergi, Studsvik, Sweden, "Plastic Heat Exchangers for Waste Heat Recovery and Cooling Applications," Preliminary Information Sheet (Aug 1975).
3. W. B. Suratt, G. H. Hart, and E. N. Seider, "Development of Plastic Heat Exchangers for Sea Solar Power Plants - Final Report," DSS Engineers, Inc., Fort Lauderdale, FL, NSF/RA/N75-026(PB-242155) (Mar 1975).
4. C. Roma, "An Advanced Dry Cooling System for Water from Large Power Station Condensers," *Proc. Amer. Power Conf.* 35, 626-637 (1973).
5. B. C. Fryer et al., "Engineering and Cost Comparison of Three Different All-Dry Cooling Systems," BNWL-2121 (Sept 1976).
6. J. G. DeSteele and K. Simhan, "European Dry Cooling Tower Operating Experience," BNWL-1995 (Mar 1976).
7. "Mitsubishi Enthalpy Exchanger--Lossnay," Mitsubishi Electric Company Report N-7807 LO-02 (July 1978).
8. D. W. van Krevelen, "Properties of Polymers: Their Estimation and Correlation with Chemical Structure," 2nd Ed., Elsevier, Amsterdam (1976).
9. C. Y. Ho et al., "Thermophysical Properties of Polystyrene and Polyvinylchloride," *Proc. 7th Symp. on Thermophysical Properties*, A. Cezairliyan, Ed., ASME, New York, 198-218 (1977).
10. D. R. Anderson, "Thermal Conductivity of Polymers," *Chem. Rev.* 66, 677-690 (1966).
11. "Teflon Fluorocarbon Resins--Mechanical Design Data," E. I. duPont de Nemours and Co. Tech. Publication E-05561.
12. "Physical Properties of Polytetrafluoroethylene," Imperial Chemical Industries, Ltd. Tech. Service Note FI2, 3rd Ed. (Aug 1972).
13. A. V. Luikov, L. L. Vasiliev, and A. G. Shashkov, "A Method for Simultaneous Determination of All Thermal Properties of Poor Heat Conductors Over the Temperature Range 80 to 500 K," *Adv. in Thermophysical Properties at Extreme Temperatures and Pressures*, 3rd Symp. on Thermophysical Properties, S. Gratch, Ed., ASME, New York, 314-319 (1965).
14. Y. S. Touloukian et al., "Thermophysical Properties of Matter," *Thermal Conductivity - Nonmetallic Solids* (Vol. 2, 1970), and *Thermal Expansion - Nonmetallic Solids* (Vol. 13, 1977), IFI/Plenum Press, New York.

15. "Ryton Polyphenylene Sulfide for Injection Moulding," Phillips Chemical Company Tech. Service Memorandum 258 (Nov 1976).
16. "Kynar," Pennwalt Corporation Tech. Publication PL-148-1177-10M-WI.
17. A. E. Bergles, "Enhancement of Heat Transfer," *Proc. Sixth International Heat Transfer Conf.* 6, 89-108 (1978).
18. J. A. Manson and L. H. Sperling, "Polymer Blends and Composites," Plenum Press, New York (1976).
19. L. E. Nielsen, "Mechanical Properties of Polymers and Composites - Vols. 1 and 2," Marcel Dekker, New York (1974).
20. P. H. H. Bishop, "Predicting Mechanical Properties of Fibre Composites," B. Parkyn, Ed., *Glass Reinforced Plastics*, Iliffe Books, London (1972).
21. K. F. Rogers et al., "The Thermal Expansion of Carbon Fibre-Reinforced Plastics," *J. Mater. Sci.* 12, 718-734 (1977).
22. H. S. Chen and A. Acrivos, "The Effective Elastic Moduli of Composite Materials Containing Spherical Inclusions at Non-Dilute Concentrations," *Int. J. Solids Struct.* 14(5), 349-364 (1978).
23. R. D. Boehme, "Aluminum Reinforcement of Some Polyethylenes," *J. Appl. Polymer Sci.* 12, 1097-1108 (1968).
24. K. C. Rusch and J. K. Rieke, "Strength Properties of Aluminum-Filled Polyethylene-Acrylic Acid Graft Copolymers," *J. Polymer Sci. Part C, Polymer Symp.* No. 30, pp. 453-457 (1970).
25. G. Landon, G. Lewis, and G. F. Boden, "The Influence of Particle Size on the Tensile Strength of Particulate-Filled Polymers," *J. Mater. Sci.* 12, 1605-1613 (1977).
26. C. K. H. Dharan, "Fracture Mechanics of Composite Materials," *J. Eng. Materials and Technology, Trans. ASME* 100(3), 233-247 (1978).
27. R. A. Crane, R. I. Vachon, and M. S. Knader, "Thermal Conductivity of Granular Materials - A Review," *Proc. 7th Symp. on Thermophysical Properties*, A. Cezairliyan, Ed., ASME, New York, 109-123 (1977).
28. R. E. Meredith and C. W. Tobias, "Resistance to Potential Flow Through a Cubical Array of Spheres," *J. Appl. Phys.* 31, 1270-1273 (1960).
29. Z. Hashin and S. Shtrikman, "A Variational Approach to the Theory of the Effective Magnetic Permeability to Multiphase Materials," *J. Appl. Phys.* 33, 3125-3131 (1962).
30. J. B. Keller, "Conductivity of a Medium Containing a Dense Array of Perfectly Conducting Spheres or Cylinders or Nonconducting Cylinders," *J. Appl. Phys.* 34, 991-993 (1963).
31. M. Zuzovsky and H. Brenner, "Effective Conductivities of Composite Materials Composed of Cubic Arrangements of Spherical Particles Embedded in an Isotropic Matrix," *Z. Ang. Math. Phys.* 28, 979-992 (1977).
32. W. T. Doyle, "The Clausius-Mosotti Problem for Cubic Arrays of Spheres," *J. Appl. Phys.* 49, 795-797 (1978).
33. R. A. Crane and R. I. Vachon, "A Prediction of the Bounds on the Effective Thermal Conductivity of Granular Materials," *Int. J. Heat Mass Transfer* 20, 711-723 (1977).

34. K. Schulgasser, "Concerning the Effective Transverse Conductivity of a Two-Dimensional Two-Phase Material," *Int. J. Heat Mass Transfer* 20, 1273-1280 (1977).
35. R. L. Hamilton and O. K. Crosser, "Thermal Conductivity of Heterogeneous Two-Component Systems," *Ind. Eng. Chem. Fundam.* 1, 187-191 (1962).
36. F. F. T. deAraujo and H. M. Rosenberg, "The Thermal Conductivity of Epoxy-Resin/Metal-Powder Composite Materials from 1.7 to 300 K," *J. Phys. D. Appl. Phys.* 9, 665-675 (1976).
37. R. Krupiczka, "Analysis of Thermal Conductivity in Granular Materials," *Int. Chem. Eng.* 7(1), 122-124 (1967).
38. G. P. Willhite, D. Kunii, and J. M. Smith, "Heat Transfer in Beds of Fine Particles (Heat Transfer Perpendicular to Flow)," *Amer. Inst. of Chem. Engr. J.* 8, 340-345 (1962).
39. M. G. Kaganer, "Thermal Insulation in Cryogenic Engineering," *Izd. Mashinostromenie* (1966).
40. J. Delmonte, "Metal-Filled Plastics," Reinhold Publ. Co., New York (1961).
41. W. R. Gambill and W. R. Reed, "State-of-the-Art Review of Heat-Recovery Processes," ORNL/TM-6222 (Aug 1978).
42. "Heat Exchangers - Shell-and-Tube Type," E. I. duPont de Nemours and Co. Catalog E-09547 (May 1976).
43. R. D. Smith, "Heat Exchanger Apparatus with a Novel Bypassing Arrangement for Shellside Flow," U. S. Patent 3,417,812 (1968).
44. R. D. Smith, "Subdivided Heat Exchanger Tube Bundle Assembly Providing Longitudinally Distributed Fluid Bypass and Distributing Channels," U. S. Patent 3,438,434 (1969).
45. "Here Is the Only Sure Way to Make Other Materials Bond to Fluorocarbon Plastics," Matheson Chemical Company, Matheson Update 3(5), 22 (Fall 1978).
46. C. Roma, "Tube Nest for Heat Exchangers, and Modular Elements for Said Nest Consisting of Thermoplastic Tubes, and Process Manufacturing the Modular Elements and the Tube Nests," U. S. Patent 4,030,540 (1977). (See also Italian Patent Specification 1408823 (8 Oct 1975)).
47. C. Roma, "Process for Manufacturing Modular Elements and a Tube Nest for Heat Exchangers," U. S. Patent 4,054,980 (1977).
48. W. M. Small and R. K. Young, "Exchanger Design Cuts Tube Vibration Failure," *Oil & Gas J.* 75, 77-80 (Sept 1977).
49. A. P. Fraas and M. N. Oezisik, "Heat Exchanger Design," John Wiley & Sons, New York (1965).
50. W. M. Kays and A. L. London, "Compact Heat Exchangers," 2nd Ed., McGraw-Hill, New York (1964).
51. "High Flux Boiling and Condensation Heat Transfer Tube - Hitachi Thermoexel," Hitachi Cable, Ltd. Catalog EA-501 (1978).
52. "Engineering Design for Plastics," E. Baer, Ed., Reinhold Publishing Co., New York (1964).

53. S. Levy and J. H. DuBois, "Plastics Product Design Engineering Handbook," van Nostrand Reinhold Co., New York (1977).
54. "Modern Designs for Effective Heat Transfer," American Heat Catalog S.A1667, H.R. (Apr 1976).
55. "Processing Equipment," APV Company Catalog CP-45.
56. M. Telkes, U. S. Dept. of Interior Research and Development Report No. 13 (Dec 1956).
57. J. M. Weaver, "Plastic Film Heat Exchangers," Chem. Eng. Prog. 56(7), 49-51 (July 1960).
58. W. H. Thielbahr, "Heat Exchanger Technology Needs for Conservation Research and Technology, ERDA," Naval Weapons Center Report NWC-TM-2930 (Dec 1976).
59. J. D. Parker, "Heat Pipes Gain Use in Heat Transfer," Oil & Gas J. 75(37), 89-92 (Sept 1977).
60. J. Taborck, "Heat Exchanger Design," Proc, 6th International Heat Transfer Conf. Vol. 6, 269-293 (1978).
61. C. L. Segaser, "Heat Recovery Equipment for Engines," ANL/CES/TE 77-4 (Apr 1977).
62. R. Silvestrini, "Phase I Topical Report Exhibit III-B Heat Exchanger Fouling and Corrosion Evaluation," Garrett AiResearch Manufacturing Co. of California Report 78-15161(1), (1978).
63. W. V. Hambleton, "General Design Considerations for Gas Turbine Waste Heat Steam Generators," Actual Specifying Engineer, 95-101 (Apr 1970).
64. R. C. Attig and P. Sedor, "A Pilot-Plant Investigation of Factors Affecting Low-Temperature Corrosion in Oil Fired Boilers," *Proc. of American Power Conference*, Vol. XXVI, 553-562 (1964).
65. J. E. Radway and L. M. Exley, "A Practical Review of the Cause and Control of Cold End Corrosion and Acidic Stack Emissions in Oil-Fired Boilers," ASME Paper #75-WA/CD-8 (July 1975).
66. T. J. Marciniak, J. C. Bratis, A. Davis, and C. Lee, "An Assessment of Stirling Engine Potential in Total and Integrated Energy Systems," ANL/ES-76 (Feb 1979).
67. J. N. Peterson et al., "Computer Programs for Chemical Engineers - Part 2," Chem. Eng. 85(15), 69-82 (July 1978).
68. D. Q. Kern, "Process Heat Transfer," McGraw-Hill, New York (1950).
69. D. Q. Kern and A. D. Kraus, "Extended Surface Heat Transfer," McGraw-Hill, New York (1972).
70. W. H. McAdams, "Heat Transmission," 3rd Ed., McGraw-Hill, New York (1954).
71. J. Taborek, "Design Methods for Heat Transfer Equipment - A Critical Survey of the State-of-the-Art," *Heat Exchangers: Design and Theory Sourcebook*, N. H. Afgan and E. U. Schlünder, Eds., Scripta Book Co., Washington, D.C., 45-74 (1974).
72. I. Tanasawa, "Dropwise Condensation - The Way to Practical Applications," *Proc. 6th Int. Heat Transfer Conf.*, Vol. 6, 393-405 (1978).

73. A. P. Colburn, "Problems in Design and Research on Condensers of Vapors and Vapor Mixtures," *Proc. General Discussion on Heat Transfer*, Inst. Mech. Eng., London and ASME, New York, 1-11 (1951).
74. J. L. Wilson, "The Design of Condensers by Digital Computers," *Symp. on Design, Decision and the Computer*, Inst. of Chem. Engineers, London (1972).
75. I. D. R. Grant and B. G. Purser, "Measurement of Air Distribution in a Condenser," National Engineering Laboratory Report NEL-477 (1971).
76. N. H. Wild, "Non-condensable Gas Eliminates Hammering in Heat Exchanger," *Chem. Eng.* 76(8), 132-134 (Apr 1969).
77. R. R. Hood, "Designing Heat Exchangers in Teflon," *Chem. Eng.* 74(11), 181 (May 1967).
78. F. J. Czeiner, "PTFE Heat Exchangers," *Chemical Process Engineering (Heat Transfer Survey)* 50(8), 60-61 (Aug 1969).
79. F. T. Smith, "Two-Phase Condensing Flow (Steam-Water) through Small Bore Tubes of Teflon," *Chem. Eng. Prog. Symp. Ser.* 92, Vol. 65, *Heat Transfer*, W. R. Martini, Ed., AIChE, New York, 46-52 (1969).
80. F. T. Smith, "Cross-Flow Heat Exchange Through Bundles of Fluorocarbon Resin Tubes," *5th Int. Heat Transfer Conf.*, Tokyo, Vol. 5: *Heat Transfer*, 228-232 (1974).
81. R. D. Smith, "Shellside Performance of Heat Exchangers Having Tubes of Teflon," *4th Int. Heat Transfer Conf.*, Paris, Vol. 1, Paper HE2.9 (1970).
82. G. S. Roush, Research and Development, Phillips Petroleum Company, personal communication (Aug 1978).
83. "Modern Plastics Encyclopedia 77-78," J. Agranoff, Ed., *Modern Plastics* 54(10A), 1-862 (Oct 1977).

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M. Greeven, Garrett AiResearch Mfg. Co., Torrance
 L. Hays, Biphase Energy System, Inc., Santa Monica
 J. Taborek, Heat Transfer Research, Inc., Alhambra
 J. Yampolski, General Atomic
 A. Metcalf, Solar Turbines International, San Diego
 L. Trimble, Lockheed Missiles and Space Co., Sunnyvale
 M. Suo, United Technologies Research Lab., East Hartford

J. Chen, Lehigh U.
 M. I. Rudnicki, Aerojet Energy Conversion Co., Sacramento
 A. C. Thekdi, Midland-Ross Corp., Toledo
 R. Kulvicki, Southern California Edison Co., Rosemead
 G. Higgins, The Dow Chemical Co., Midland, Mich.
 D. Wright, Rocketdyne Div. Rockwell International, Canoga Park
 C. Moyer, Aerotherm/Acurex Corp., Mountain View, Calif.
 J. Michel, Oak Ridge National Lab.
 W. Ranken, Los Alamos Scientific Lab.
 R. L. Fulton, Lawrence Berkeley Lab.
 J. L. Frame, Fluidyne Engr. Co., Minneapolis
 G. M. Benson, Energy Research and Generation, Inc. Oakland, Calif.
 J. Boyen, Emeryville, Calif.
 J. P. Rowe, Babcock and Wilcox, Alliance, Ohio
 E. W. Ungar, Battelle Columbus Labs.
 G. K. Hart, DSS Engineers, Inc., Ft. Lauderdale
 F. Arcella, Westinghouse Electric R&D Center, Pittsburgh
 G. Hanson, Thermal Transfer Corp., Monroeville, Pa.
 Black and Veatch Co., Kansas City, Mo.
 B. Rose, Mechanical Technology, Inc., Latham, N. Y.
 D. Niggeman, Sundstrand Corp., Rockford, Ill.
 R. E. Barber, Barber-Nicholas Engr. Co., Arvada, Colo.
 L. C. Hoagland, Amtech, Inc., Newton, Mass.
 J. R. Schorr, Battelle Columbus Labs.
 H. F. Poppendiek, Geoscience, Ltd., Solana Beach, Calif.
 R. Scholtz, United Aircraft Products, Inc., Dayton, Ohio
 B. Sobers, Navy Energy & Natural Resource R&D Office, Washington
 O. M. Griffen, Naval Research Lab., Washington
 B. Haslett, Grumman Aerospace Corp., Bethpage, N. Y.
 J. Clark, Jr., Arthur D. Little, Inc., Cambridge, Mass.
 J. E. Serrine Co., Raleigh, N. C.
 Bechtel National, Inc., San Francisco, Calif.
 L. R. Lawrence, Jr., Gas Research Inst., Chicago
 R. L. Hershey, Booz-Allen Applied Research, Bethesda
 A. A. Hochrein, Jr., Daedalean Assoc., Inc., Woodbine, Md.
 W. Deutsch, Santa Clara, Calif.
 N. Jarrett, Automotive Industrial Div., Modine Manufacturing Co., Racine, Wis.
 T. W. Meyers, Deere & Co., Moline, Ill.
 F. Notaro, Linde Div. of Union Carbide, Tonawanda
 H. H. Osborn, The Air Preheater Co., Inc., Wellsville, N. Y.
 F. W. Staub, General Electric Co., Schenectady
 J. Radway, EnerChem Inc., Cleveland
 C. Benham, Solar Energy Research Inst., Golden
 A. Sims, The Ben Holt Co., Pasadena
 Patterson-Kelly Co., East Stroudsburg, Pa.
 Young Radiator Co., Racine, Wis.
 Graham Manufacturing Co., Inc., Great Neck, N. Y.,
 American Heat Reclaiming Corp., New York City
 APV Company, Inc., Tonawanda, N. Y.
 W. Aung, National Science Foundation
 N. Z. Azer, Kansas State U.
 A. Bejan, U. Colorado
 K. J. Bell, Oklahoma State U.
 A. E. Bergles, Iowa State U.
 W. Bessler, General Electric Co., Schenectady

G. Borushko, Exxon Research and Engineering Co., Florham Park, N. J.
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 H. D. Foust, The Trane Co., La Crosse, Wis.
 W. J. Greenlee, Sundstrand Corp., Rockford, Ill.
 F. Hayes, The Trane Co., La Crosse, Wis.
 B. R. Hollworth, Clarkson College
 R. E. Holmes, Texas A&M U.
 S. Hu, Excello Corp., Highland Park, Mich.
 M. K. Jensen, Iowa State U.
 G. H. Junkhan, Iowa State U.
 E. G. Keshook, U. Tennessee
 J. Keyes, High Performance Tube, Inc., Irvington, N. J.
 H. C. Knapp, Fuel Efficiency, Inc., Newark, N. Y.
 J. G. Knudsen, Oregon State U.
 J. S. Lee, Borg-Warner Research, Des Plaines, Ill.
 W. Leidenfrost, Purdue U.
 E. Levy, Lehigh U.
 R. Loehrke, Colorado State U.
 A. L. London, Stanford U.
 G. S. Mailen, Union Carbide Corp., Oak Ridge
 W. J. Marner, Heat Transfer Research Inc., Alhambra
 D. E. Metzger, U. Arizona
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 M. Nakazatomi, Iowa State U.
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 R. W. Perkins, Turbotec Products, Inc., New Britain, Conn.
 T. L. Rabas, Westinghouse Electric Corp., Philadelphia
 L. W. Rodgers, KVB, Inc., Columbus
 W. M. Rohsenow, Massachusetts Inst. Technology
 R. R. Rothfus, Carnegie-Mellon U.
 R. K. Sakhuja, Thermo Electron Corp., Waltham, Mass.
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