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HIGH TEMPERATURE RANGE RECUPERATOR
PHASE II FINAL REPORT
PROTOTYPE DEMONSTRATION AND
MATERIAL AND ANALYTICAL STUDIES

MASTER

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ABSTRACT

Reported herein is a summary of the work performed to fully evaluate the commercial potential of a unique ceramic recuperator for use in recovering waste heat from high temperature furnace exhaust gases. The recuperator concept being developed consists of a vertical cylindrical heat exchange column formed from modular sections. Within the column, the gasketed modules form two helical flow passages - one for high temperature exhaust gases and one for preheating combustion air. The column is operated in a counterflow mode, with the exhaust gas entering at the bottom and the combustion air entering at the top of the column.

Activities included design and procurement of prototype recuperator modules, construction and testing of two prototype recuperator assemblies, exposure and mechanical properties testing of candidate materials, structural analysis of the modules, and assessment of the economic viability of the concept. The results of the project indicated that the proposed recuperator concept was feasible from a technical standpoint. Economic analysis based upon recuperator performance characteristics and module manufacturing costs defined during the program indicated that 3 to 10 years (depending upon pre-heat temperature) would be required to recover the capital cost of the system in combustion air preheat applications. At this stage in the development of the recuperator, many factors in the analysis had to be assumed. Significant changes in some of the assumptions could dramatically affect the economics. For example, utilizing \$2.85 per mcf for the natural gas price (as opposed to \$2.00 per mcf) could reduce the payback period by more than half in certain cases. In addition, future commercial application will depend upon ceramic component manufacturing technique advances and cost reduction.

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PROJECT SUMMARY

The heat carried away in the exhaust gases of direct heating industrial furnaces ranges from 20 percent (for low temperature operation) to 60-80 percent (for 2400°F operation) of the total available combustion energy (Reference 1). One method for recovering portions of this wasted energy is preheating of furnace combustion air. Millcreek Glass Corporation conceived a unique high temperature recuperator for combustion air preheat applications.

Millcreek Glass Corporation's concept consists of a vertical heat exchange column formed from modular sections. Within the column, gasketed modules form two helical flow passages - one for high temperature exhaust gases and one for preheating combustion air. The helical section forming the flow passages provides an interface for exchanging heat from the exhaust gases to the preheating combustion air. The column is operated in a counterflow mode, with the exhaust gas entering at the bottom and the combustion air entering at the top of the column.

The Department of Energy, Division of Industrial Energy Conservation, in recognizing the apparent utility of the recuperator design, funded a development program to determine commercial feasibility. Terra Tek, Inc. directed the work under DOE Contract No. EC-77-C-07-1660. Millcreek Glass assisted with the development under subcontract to Terra Tek, Inc.

The first phase of the project, conducted from April 1977 to May 1978, was devoted to the initial engineering of the recuperator and the identification of materials suitable for use in a high temperature exhaust environment. For high temperature applications, candidate ceramic and metallic materials were selected. Materials studies continued through Phase II. Candidate ceramic materials were evaluated for long term stability by exposing samples

to a strongly alkaline environment typical of glass furnace exhausts. In addition, the strength, modulus of elasticity, thermal expansion coefficient, and fracture toughness of alumina and KT-Silicon Carbide (two materials which have been fabricated into the required recuperator shape) were evaluated over a range of temperatures before and after exposure to a simulated glass furnace exhaust.

Other efforts during the second phase of the project, reported herein, were directed towards defining the recuperator hydraulic, thermal, and structural characteristics and evaluating the technical and economic feasibility of using such a system in high temperature heat recovery applications.

Two natural gas-fired recuperators were assembled - one using metallic (Inconel and stainless steel) modules and one using ceramic (KT Silicon Carbide and ECP-3) modules. Tests with the metallic column at temperatures below 800°F defined the hydraulic and the convective heat transfer characteristics of the recuperator and generated recuperator material temperature distribution data for a variety of steady state and transient thermal conditions. Operation of the ceramic column at temperatures of up to 2200°F verified hydraulic and thermal characteristics determined from low temperature tests and enabled evaluation of the effect of thermal radiation upon high temperature heat transfer performance.

Recuperator material temperature distribution data generated in the testing program were used in a finite element structural analysis of a prototype KT-Silicon Carbide module. KT-Silicon Carbide physical properties used in the analysis were obtained from the materials testing conducted during the project. The analysis indicated that the prototype modules will withstand the thermal loading conditions associated with recuperator startup, steady state, and upset operating events.

Using recuperator thermal/hydraulic characteristics determined during the program, recuperator system configurations for several heat recovery applications (covering a range of air preheat temperatures, furnace sizes, recuperator flow rates and recuperator module diameters) were defined. Capital costs associated with each of the configurations were evaluated based upon production module costs ascertained from prototype module manufacturing experiences and upon standard estimating techniques. Life cycle economic analysis determined the optimum recuperator system for a given service duty. High temperature recuperator systems sized to provide low air preheat temperatures (approximately 1000°F) were found to provide shorter payback periods than systems sized to provide high preheat temperatures (due to heat exchanger effectiveness versus area considerations). The payback period for an optimum 30 inch diameter recuperator system providing 1000°F air preheat was found to be 3 years; the payback period for the same diameter recuperator system providing 1700°F air preheat was found to be over 10 years. Manufacturing experience indicates the 30 inch diameter module to be the largest which can be fabricated using current technology.

CONCLUSIONS

Following are the major conclusions of the work conducted categorized by the major project tasks. It should be noted that the materials studies (and other related aspects) of the program were directed towards application of the recuperator in a glass furnace exhaust environment. Thus, module fabrication, materials selection and evaluation, and module structural analysis activities were based upon the use of prime recuperator material candidates for this application (cordierite, mullite, alumina, and several silicon carbide compositions) identified during Phase I of the project. Economic evaluation of the recuperator concept was based upon module costs representative of those of either KT-Silicon Carbide or AD995 Alumina modules. The thermal/ hydraulic performance analysis and systems comparison tasks were not dependent upon the materials aspects of the project. Thus, the applicability of the results of these studies is not limited to glass furnace recuperator installations.

Module Design and Fabrication

- 1) Prototype KT Silicon Carbide modules were successfully fabricated using a slip-cast technique.
- 2) A prototype KT Silicon Carbide module which had extended surfaces on the heat transfer interface was successfully fabricated.
- 3) Bonding of three KT Silicon Carbide modules at the mating interfaces was achieved using a welding technique.
- 4) Prototype ECP-3 modules were successfully fabricated using a slip-cast technique.
- 5) An AD995 alumina module was successfully fabricated using a green-ware machining technique.

- 6) Fabrication experience indicates that modules of up to 30 inches in diameter can be produced.

Thermal/Hydraulic Performance Analysis

- 1) A computer model which accurately simulates the thermal performance of the high temperature recuperator (both the prototype and larger diameter modules of geometrically similar configurations) was developed.
- 2) Thermal and hydraulic performance of the high temperature recuperator was experimentally characterized over a wide range of parameters. Test results indicated the overall heat transfer coefficient for the recuperator to be in excess of 5 Btu/hr ft² °F for high temperature, turbulent flow operation. The friction factor (f) for the recuperator was empirically determined to be:

$$f = 1.066 (\text{Reynolds Number})^{-0.196}$$

in the turbulent flow regime.

- 3) Analytical investigations indicate that addition of extended surfaces to one side or both sides of the recuperator heat transfer interface has the potential of enhancing recuperator heat transfer 20 and 25 percent respectively, during operation at optimum flow conditions. Additional factors such as fouling, pressure drops, and module manufacturing costs need to be investigated prior to recommending the use of extended surfaces in future recuperator module designs.
- 4) Analytical radiation heat transfer enhancement studies indicate gas stream doping has the potential of increasing recuperator heat transfer 23 percent (with 5 percent CO₂ doping) during high tempera-

ture operation. However, the viability of this means of enhancement is questionable due to CO₂ costs and furnace operation considerations.

Materials Selection and Evaluation

- 1) Young's Modulus, thermal expansion, flexure strength, and fracture toughness were measured as a function of temperature for pristine and exposed (in a glass furnace exhaust environment) samples of KT Silicon Carbide and AD995 Alumina.
- 2) Quantitative corrosion testing indicated that KT Silicon Carbide is not an acceptable material for use in a glass furnace exhaust environment. Severe corrosion of KT Silicon Carbide was observed in the short term exposure tests conducted.
- 3) AD995 Alumina was not affected by exposure to a glass furnace exhaust environment. In the short term quantitative corrosion tests conducted, no measurable dimensional changes or weight losses occurred.
- 4) Qualitative exposure testing of several candidate materials indicates that Vistal, ECP-1, ECP-2, silicon nitride, zirconia, and mullite have potential application in a glass furnace exhaust environment at temperatures of 2600°F, 2250°F, 2550°F, 1650°F, 1550°F, and 1550°F, respectively.

Structural Analysis

- 1) Stress analysis indicates that prototype KT Silicon Carbide recuperator modules will withstand the thermal loading conditions associated with startup, steady state, and upset operating events.

- 2) Future work should address the effect of fatigue and axial loading upon module structural integrity. Further stress analysis work will be required in designing larger diameter recuperator modules.

Comparison with Existing Systems

- 1) The high temperature recuperator requires considerably less heat exchange area (and corresponding unit volumes and material weights) than a checker regenerator system sized for the same heat recovery duty. At the high preheat temperature operating conditions at which the evaluation was made, the high temperature recuperator requires approximately a tenth of the heat exchanger area that a checker system requires. This is due to the following:
 - a) The recuperator can be operated at much higher velocities (with corresponding heat transfer enhancement) due to better leakage characteristics.
 - b) The geometry of the recuperator provides enhanced convective heat transfer.
 - c) A redundant regenerator unit is required for the checker systems due to the cyclic nature of its operation.
- 2) The high temperature recuperator provides an advantage over regenerator systems in that continuous, steady state operation is possible.
- 3) The high temperature recuperator is not a compact heat exchange unit. Several recuperator columns would be required to service a large industrial furnace. As a result, manifolding, distribution, and valving problems could be expected in using the high temperature recuperator in large scale operations. Considerable plant area

would be required for an installation due to stack access requirements.

Economic Analysis

- 1) High temperature recuperator systems sized to provide low air preheat temperatures (approximately 1000°F) were found to provide shorter payback periods than systems sized to provide high air preheat temperatures. The payback period for a 30 inch diameter recuperator system providing 1000°F air preheat is approximately 3 years; the payback period for the same diameter recuperator system providing 1700°F air preheat is over 10 years.
- 2) Literature review indicated that existing metallic recuperator systems provide payback periods of approximately 1.5 years in small, low temperature preheat applications.
- 3) The optimum flow rate operating points for individual recuperator columns (based upon economic considerations) were found to be approximately 10, 50, and 150 lbm/min for the prototype (8 inch), 15 inch, and 30 inch diameter recuperators, respectively.
- 4) Larger diameter recuperator systems were found to provide an economic advantage over the prototype (8 inch diameter) recuperator system.
- 5) With the assumptions used in the analysis, the payback period was found to be essentially independent of furnace size (for a given preheat temperature and recuperator diameter).

1. INTRODUCTION

1.1 Background

The heat carried away in the flue gases of direct heating industrial furnaces ranges from 20 percent of the total available combustion energy for low temperature operation (on the order of 600°F) up to 60-80 percent at temperatures of 2,400°F (Reference 1). The three principal methods for recovering some of this waste heat are:

- 1) Steam generation in waste heat boilers
- 2) Feed preheating
- 3) Combustion air preheating

Steam generation is used in applications with large process steam requirements. For example, in forging operations, waste heat boilers generate steam for the operation of steam hammers and presses.

The degree to which feed preheating can be accomplished is very process dependent. For example, in the cement industry, combustion gases are used to preheat the reactants resulting in furnace exhaust temperatures on the order of 900°F. In the glass industry, however, possible segregation of constituents in the feed batch precludes extensive preheating with existing furnace designs.

Combustion air preheating is employed in a wide range of industrial furnace applications. The two existing types of preheaters are defined as the recuperator and the regenerator. The recuperator is a continuous flow device wherein heat is exchanged between two streams across an interface. A regenerative heat exchanger contains heat storage elements which alternately absorb heat from the exhaust stream and then liberate heat to the preheat system. Intermittent gas flow stationary element regenerators are commonly used in

high temperature furnace operations. In these systems, hot flue gases and precombustion air are alternately cycled through the same flow passages. Presented in Figure 1.1 are curves of possible fuel savings versus combustion air preheat temperature over a range of process temperatures. This figure was generated assuming methane combustion with 10 percent excess air. As shown in the figure, high air preheat temperatures result in considerable fuel savings. Supporting calculations for the figure are included in Appendix 7.

Regenerators in high temperature furnace applications commonly consist of a channel filled with a checkerwork of brick or tile stacked in a pattern to provide a number of vertical flow passages. High preheat temperatures (up to 2000°F, Reference 2) can be obtained, although overall heat transfer coefficients are usually lower than for a recuperator. Regenerator systems, however, do have a number of practical limitations. These include high capital cost (large quantities of brick are required for heat storage and two separate units are needed per furnace) and potential operating problems due to the cyclic quality of the recovered heat.

Recuperators are often limited to moderate temperature service. This is due to the difficulty of fabricating large heat exchange units from ceramic materials which are structurally sound at high temperatures and which will not prematurely corrode, leak or plug. As a result of these materials problems, recuperators are commonly fabricated from metallic components. This limits the unit to applications where material temperatures can be kept below approximately 1650°F - 1800°F (References 2 and 30). In applications such as steel and glass melting, furnace exhaust temperatures can exceed 2400°F. Dilution of the exhaust gas or concurrent operation is necessary if metallic recuperators are to be used. This results in a loss of heat recovery potential.

FIGURE 1.1. Savings in Fuel versus Combustion Air Preheat Temperature

In an attempt to combine the advantages of both types of preheaters, Millcreek Glass has proposed a unique high temperature recuperator concept which features ceramic heat transfer surfaces and a counterflow design which permits continuous, steady-state operation. The recuperator consists of a furnace exhaust flue made up of a stack of ceramic tubular sections (modules) with a twisted helical interface running the entire length. The system geometry is readily apparent from inspection of Figure 1.2. Exhaust is directed up one channel of the stack from the exhaust port of the furnace. Ambient air enters the top of the opposite channel and absorbs heat transmitted from the exhaust through the helical interface as it flows down the stack. Gaskets are provided at each module joint to inhibit leakage between the passages and to the surroundings. The number of stacks and the length of each stack for a given application is a function of the exhaust flowrate and temperature and the required heat recovery performance.

The Department of Energy, Division of Industrial Energy Conservation in recognizing the apparent feasibility of this recuperator design has funded the present research and development program. Terra Tek, Inc., has been the prime contractor under DOE Contract No. EC-77-C-07-1660. Millcreek Glass has assisted with the development under subcontract to Terra Tek, Inc.

The first phase of the project was devoted to the initial engineering of the recuperator and the identification of a number of ceramic refractory materials suitable for use in a glass melting environment. From this engineering effort and a selection of the best candidate materials, prototype modules were made and qualitatively tested for thermal and mechanical stability on a small lab type glass furnace. A summary of the Phase I effort is presented in Reference 3.

FIGURE 1.2. Schematic of the Modular Counterflow Recuperator with a Helical Interface

Phase II, which is reported in detail herein, was devoted to developing and verifying engineering design procedures, establishing candidate material properties at high temperature, and verifying recuperator mechanical and thermal performance in a small scale prototype test. Phase II was concluded with a preliminary assessment of the potential cost effectiveness of such a recuperator system in a high temperature furnace application.

1.2 Outline of Phase II Program

The principal objectives of the Phase II program were to define the fundamental hydraulic, thermal and structural characteristics of the recuperator and to evaluate the technical and economic feasibility of using the recuperator in high temperature heat recovery applications.

Two natural gas fired prototype test recuperators were assembled - one using metallic (inconel and stainless steel) modules and one using ceramic (KT¹ Silicon Carbide and ECP-3²) modules. The metallic stack (warm flow test assembly) allowed considerable operating flexibility at temperatures up to 800°F and was used to define the hydraulic and convective heat transfer characteristics of the recuperator and to generate material temperature distribution data for a variety of steady state and transient thermal conditions. The material temperature distribution data formed the basis of a preliminary stress analysis effort designed to evaluate the structural soundness of the prototype design.

The ceramic stack could be operated at temperatures up to 2200°F. This assembly was used to verify the hydraulic and thermal characteristics obtained from the warm flow rig and to evaluate the added effects of thermal radiation upon high temperature heat transfer performance.

¹Carborundum Company Trademark

²Alpha Glass Company Trademark

A thermal model of the recuperator into which the experimentally defined heat transfer parameters were incorporated was developed. The model is used for given furnace exhaust conditions and preheat requirements. In addition, preliminary scoping calculations were carried out to evaluate methods for enhancing the convection and radiation heat transfer characteristics of the recuperator.

Materials studies continued through Phase II. Candidate ceramic materials were evaluated for long term stability by exposing samples to a strongly alkaline environment typical of glass furnace exhausts. In addition, the strength, modulus of elasticity, thermal expansion coefficient, and fracture toughness of alumina and KT-Silicon Carbide (two materials which have been fabricated into the required recuperator shape) were evaluated over a range of temperatures before and after exposure to the simulated glass furnace exhaust. The results of the materials characterization studies and the stress analysis were used to infer the structural adequacy of the prototype design. The feasibility of this concept in particular applications and the engineering related aspects of implementing the system on an industrial scale were addressed in the Alternate Applications Studies phase of the program.

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2. MODULE DESIGN AND FABRICATION

2.1 Module Design

During Phase I of the project, a preliminary module design was completed. In formulating the design, fabrication capabilities of candidate suppliers were considered. Six KT-Silicon Carbide modules of this preliminary design were fabricated by Carborundum. Qualitative testing of these modules during Phase I implied their mechanical stability under various thermal conditions. In view of Phase I experiences, the same basic module design (with minor modifications) was selected for use in Phase II prototype testing.

Presented in Figures 2.1 and 2.2 are the engineering design drawings for the prototype recuperator module. Photographs of completed individual and stacked modules are included as Figure 2.3. The design features of the module are as follows:

- 1) The module consists of a cylindrical section which has a twisted helical interface surface positioned within. The helical surface and the cylinder wall form passages for two flowstreams.
- 2) Each module forms one-quarter turn of the two helical flow passages. Thus, with the given nominal module dimensions (2 inch height by 8 inch diameter), the pitch to diameter ratio for the prototype is 1:1.
- 3) Male and female mating surfaces are provided on the bottom and top, respectively, of the module to facilitate recuperator stack assembly. A "V" groove slot and key are provided on the male and female interface mating surfaces, respectively, to provide an interpath gas seal and to enable angular alignment at assembly.

FIGURE 2.2. Prototype Recuperator Module Design (Auxiliary Views)

FIGURE 2.3. Photographs of Individual and Assembled Recuperator Modules

- 4) Castellations are provided on the upper module sidewall. This enables insertion of lifting tools at any module interface within a recuperator stack for lifting and separation operations.

2.2 Module Fabrication

Initial intentions in Phase II of the project were to obtain ceramic and metallic modules fabricated from several different materials. Investigation of supplier capabilities resulted in placing orders for the modules indicated in Table 2.1.

Table 2.1
Initial Module Orders

<u>Item</u>	<u>Supplier</u>	<u>Part Description</u>	<u>Quantity</u>
1	Carborundum	SiC-KT Module	90
2	Carborundum	SiC-S α Module	10
3	Carborundum	SiC-KT Finned Modules	(use 6 of item 1)
4	Carborundum	Join 4 KT Modules	(2 runs of 4 each item 1)
5	Coors	Alumina Modules	10
6	Coors	Mullite Modules	10
7	ECP (Alpha Glass)	ECP3 Modules	10
8	Star Brass	Stainless Steel	10
9	Star Brass	Inconel	50

Table 2.2
Phase II Module Delivery

Supplier	Item	Quantity of Acceptable Units
Carborundum	KT SiC	59
	KT SiC Finned	1
	KT SiC Joined	2 X 3 joined
Alpha Glass	ECP 3	10
Star Brass	Inconel	52
	Stainless Steel	10
	Mild Steel	1
Coors	Alumina AD 995	1

Problems encountered by suppliers during the course of module fabrication dictated changing the originally outlined orders. Presented in Table 2.2 is a listing of acceptable modules which were delivered during Phase II.

Carborundum was successful in fabricating 59 KT Silicon Carbide Modules of the prototype design. Carborundum also fabricated 1 KT-Silicon Carbide module (similar to the prototype) which had extended heat transfer surfaces (fins) on the module interface. All of the Carborundum modules were fabricated using a slip-cast technique. Carborundum was also successful in joining three KT-Silicon Carbide modules together using a welding technique.

Carborundum was unable to fabricate a module from $\text{Si}\alpha$ -Silicon Carbide due to shrinkage problems during firing. Also, it was necessary to relax the tolerance on module mating surface flatness for the KT-Silicon Carbide from 0.01 to 0.05 inches to enable successful module fabrication.

Due to the limited order size, Coors would not attempt to develop a module slip-cast fabrication technique. Coors was able to provide one acceptable alumina AD995 module within the allotted budget using a machining technique (the module was machined from greenware prior to firing). Fabrication of mullite modules was not attempted.

Alpha Glass was successful in providing 10 ECP-3 (chromium bearing alumina) modules. A slip-cast technique was used in fabrication of these modules.

Acceptable metallic modules were supplied by Star Brass. A sand cast technique was used in fabrication of these modules. Mating surfaces on the metallic modules were machined following casting to ensure a good seal upon recuperator assembly.

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3. THERMAL/HYDRAULIC PERFORMANCE ANALYSIS

3.1 Introduction

An analytical and experimental program was conducted to establish the leakage, hydraulic, and heat transfer characteristics of the prototype recuperator. Work conducted during the program provides a basis for predicting characteristics of industrial-scale recuperators.

The program consisted of:

- 1) Development of a numerical model which simulates recuperator steady state thermal performance over a range of operating conditions.
- 2) Conducting ambient temperature leakage and hydraulic characterization tests using a low temperature metallic recuperator test assembly (cold/warm flow test assembly).
- 3) Performing steady state heat transfer tests at low temperatures (below 600°F) using the cold/warm test assembly.
- 4) Conducting leakage, hydraulic, and steady state heat transfer tests at high temperatures (up to 2200°F) using a high temperature ceramic recuperator test assembly.
- 5) Validation of the numerical model via comparison of predicted versus experimentally determined recuperator performance.
- 6) Analytical investigation of possible recuperator heat transfer performance enhancement via finning, addition of heteropolar gases to recuperator air and combustion gas flows (to improve radiation heat transfer), or use of recuperator materials transparent to thermal radiation.

Steady state heat transfer tests were conducted at low temperatures to facilitate determination of recuperator convective heat transfer characteris-

tics. At temperatures below 600°F, radiation heat transfer effects in the recuperator are minimal; thus, convective phenomena were readily quantified. With a knowledge of convective characteristics, radiation heat transfer effects were established from the results of the high temperature tests.

3.2 Thermal Model

The developed thermal model simulates the steady-state thermal performance of the prototype recuperator. A detailed description of the model is included in Appendix 1. The input variables into the program are system geometry, material thermal properties, air and combustion gas mass flow rates, air and combustion gas inlet temperatures, and the partial pressures of CO₂ and H₂O in the combustion gas. From this input, the model predicts the fluid and material temperature distributions through the recuperator.

The formulation of the heat transfer model consisted of dividing the recuperator into a series of finite control volumes. Each control volume consisted of a one-half revolution of a flow passage. Within each control volume, four lumped temperatures zones were defined. These zones consisted of the bulk gas within the zone and of the three flow passage material boundaries within the zone. For each control volume, energy balance equations between the corresponding temperature zones within the volume were derived. Energy balance equations between adjacent control volumes within the recuperator were also generated. In derivation of the equations, convective heat transfer in the exhaust gas and air flows, conductive heat transfer through the module interface, and radiative heat transfer between recuperator surfaces and between the exhaust gas and the recuperator surfaces were considered. The convective heat transfer was based upon the results of the warm flow tests. Steady state recuperator performance was predicted by iterative solution of the governing energy balance equations.

The validity of the model was checked by comparing the model's results versus the hot flow test results. This comparison is shown in Figures 3.1 to 3.4. The temperature input to the model is the combustion gas temperature at module 6 and the air temperature at module 54. The hot flow assembly was made up of 62 modules. The true inlet temperatures were not used in the model. The model as formulated can only be used to predict the performance of recuperators having an integral number of turns of the two helical flow passages (i.e. recuperator columns having a number of modules which is a multiple of four). Thus, the central 48 modules of test column, for which required air and combustion gas flow temperature measurements were made, were used in the comparison. As shown in Figures 3.1 through 3.4, the thermal model accurately predicts recuperator performance.

The thermal model can also be used to predict steady-state performance of a geometrically similar, larger diameter recuperator by changing the system geometry input to the program. Upon experimentally determining convective characteristics of different recuperator module configurations, slight modification of the model will enable its use in predicting the performance of the new configuration.

3.3 Experimental Program

3.3.1 Cold/Warm Flow Tests

The objectives of the cold/warm flow testing were:

- 1) Determination of the leakage and hydraulic characteristics of the modular, gasketed recuperator over a range of ambient flow conditions.
- 2) Determination of recuperator convective heat transfer characteristics in the absence of significant thermal radiation effects.

FIGURE 3.1. Comparison of Model and Experimental Results, Test No. 22

FIGURE 3.2. Comparison of Model and Experimental Results, Test No. 24

FIGURE 3.3: Comparison of Model and Experimental Results, Test No. 32

FIGURE 3.4. Comparison of Model and Experimental Results, Test No. 48

- 3) Provision of design and operating experience for incorporation into the high temperature testing program.

3.3.1.1 Test Apparatus

3.3.1.1.1 Recuperator and Support Structure

The cold/warm flow test apparatus was constructed at the Millcreek Glass Co. facility in Salt Lake City, Utah. The recuperator assembly consisted of 52 metallic recuperator modules. This number of modules was selected to provide a flow passage length sufficient to ensure that entrance and exit effects had minimal influence upon the data obtained. During column assembly, each junction between recuperator modules was gasketed with three layers of 1/16 inch thick Fiberfrax¹ ceramic fiber paper material.

To minimize recuperator heat loss during operation, the column was insulated with 3 inches of Cerablanket² refractory fiber thermal insulation. The batt insulation was installed in 1-inch layers with overlapping joints. Each layer was banded to the recuperator.

The recuperator stack was supported by a steel manifold plate resting upon a steel structure. A plate was also situated at the top of the column to form a manifold. The column was compressed by threaded rods passing through the lower and upper manifold plates.

A photograph of the cold/warm flow test assembly, taken prior to insulation of the recuperator column, is included as Figure 3.5.

¹Carborundum Company Trademark

²Johns-Manville Company Trademark

Figure 3.5. Photograph of Cold/Warm Flow Test Apparatus

3.3.1.1.2 Piping and Auxiliary Equipment

The configuration of the entire apparatus is schematically presented in Figure 3.6. As shown in the schematic, the recuperator was operated in a counterflow mode with the combustion gas entering at the bottom of the recuperator and the air entering at the top of the recuperator.

Air was supplied to the recuperator by a centrifugal blower (rated for 10 in. H₂O pressure at 360 SCFM delivery). The blower was piped to the recuperator with standard 4 inch steel tubing components and a custom fabricated transition piece. An orifice meter run (including straightening vanes) fabricated from 3 inch pipe and orifice flanges was located in the blower suction line. Air exhausting from the recuperator was piped to the outside of the building with 4-inch steel tubing components. Butterfly dampers were positioned upstream and downstream of the recuperator for flow and pressure regulation.

Combustion gases were supplied to the recuperator from a natural gas-fired burner installed in a small combustor. Air was supplied to the burner by a centrifugal blower. A bypass line around the burner was provided in the combustion air supply line to allow precise control of temperature and flow conditions entering the recuperator (in conjunction with burner air and natural gas flow adjustments). Air and combustion gas lines were plumbed with standard 4-inch tubing components. Butterfly dampers were positioned in the air supply and bypass line for flow control. Natural gas was piped to the burner from the building natural gas supply with standard 1-inch piping components. Included in the natural gas line were a displacement-type meter and a regulating valve.

Combustion gas from the recuperator was exhausted to the outside of the building with a 4-inch tubing run.

FIGURE 3.6. Cold/Warm Flow Test Apparatus Schematic

3.3.1.1.3 Instrumentation

System instrumentation is outlined in Figure 3.7. Recuperator temperature measurement locations are outlined in Figure 3.8.

System and recuperator bulk fluid temperatures were measured with Type K thermocouples sheathed with 1/8 inch diameter stainless steel tubing. One-half inch diameter tubes, aligned parallel to the direction of the flow, were installed over the thermocouple junctions to mitigate the effect of thermal radiation upon temperature measurement accuracy.

Recuperator surface temperatures were measured with unsheathed, 24 gauge, Type K thermocouples bonded to the surface of the modules with an epoxy adhesive. All temperatures were monitored with a digital readout.

As shown in the system schematic, flow rates of the natural gas, combustion air supply, and recuperator air were measured during testing. Natural gas flow was measured with a displacement-type meter. The combustion air supply and recuperator air flows were measured with orifice metering assemblies. These assemblies, fabricated from 3-inch pipe and flange tap-type orifice flanges, were designed per ASME specifications (including straightening vanes). Orifice plates of several hole diameters were provided to maintain orifice pressure drops between 0.25 and 2 inches of water throughout the range of test flow rates.

Pressure taps were located in the sheet metal transition pieces located at the inlet and outlet of the recuperator air and combustion gas flow passages. Pressures at these taps were measured with inclined water manometers, as were differential pressures across flow measurement orifices.

FIGURE 3.7. Cold/Warm Flow Test Apparatus Instrumentation

FIGURE 3.8. Cold/Warm Flow Test Apparatus - Recuperator Temperature Measurement Schematic

3.3.1.2 Testing Procedure

3.3.1.2.1 Leakage Tests

To conduct the recuperator leakage tests, the configuration of the cold/warm flow assembly was modified slightly. An orifice meter assembly was added to the air exit line. The recuperator combustion gas inlet and outlet were sealed with blind flanges. A rotameter was plumbed into the combustion gas side of the recuperator to measure interpath leakage.

Leakage tests consisted of measuring recuperator air side inlet and outlet flows and the interpath leakage into the combustion gas side of the recuperator. Tests were conducted over a range of ambient temperature flows and pressures. Flow rate and pressure were independently varied by adjusting the dampers upstream and downstream of the recuperator. Testing was conducted with recuperator air side flow rates ranging from 0.2 to 9.4 lbs/min. Average air side recuperator pressures were varied between 0.1 and 7.7 inches of water.

3.3.1.2.2 Hydraulic Tests

Hydraulic data were obtained concurrently with the leakage tests. Over the same range of flow rates (corresponding to Reynolds numbers between 1,000 and 50,000), the pressure drop in the air flow stream through the recuperator was measured.

3.3.1.2.3 Heat Transfer Tests

Heat transfer tests were conducted over a wide range of recuperator operating conditions. Data were obtained during recuperator operation with combustion gas inlet temperatures ranging from 200°F to 600°F. At each temperature level, data were obtained with recuperator air side flow rates ranging

from 0.2 to 9.6 lb/min (corresponding to Reynolds numbers ranging from approximately 1,000 to 50,000). With the exception of the low and high flow rate test conditions, the recuperator combustion gas flow was adjusted such that a constant temperature difference existed between the flow streams throughout the recuperator.

Care was exercised to ensure that test data were obtained at steady state recuperator operating conditions. A data set was logged after critical system temperatures (recuperator inlet and outlet flows) were observed to change less than 2°F in a 15 minute time interval.

3.3.1.3 Results

3.3.1.3.1 Leakage Tests

Results of the leakage tests are presented in Appendix 2. A plot of total recuperator leakage (air flow passage) versus average flow passage pressure is presented in Figure 3.9. As shown in Figure 3.9, considerable variation of the total recuperator leakage rate versus pressure was observed during the testing. To conservatively present recuperator leakage characteristics, an equivalent orifice for recuperator leakage was defined based upon the worst-case leakage observed during testing. For an equivalent orifice:

$$\text{leakage} \propto \sqrt{P}$$

where: P is the average recuperator flow passage gauge pressure. The worst-case leakage observed during testing was:

$$\text{leakage} = .35 \text{ lb/min @ } P = 7.7 \text{ in H}_2\text{O}$$

which yields the following relationship for recuperator leakage (based upon a 52 module stack):

$$\text{leakage (lb/min)} = .126 \sqrt{P(\text{in H}_2\text{O})}$$

FIGURE 3.9. Recuperator Total Leakage versus Average Pressure - Cold Flow Tests

A plot of this relationship is included in Figure 3.9.

Based upon the previously outlined relationship, a curve of the percentage of flow leakage versus flow Reynolds number was generated. For this curve, a recuperator friction factor of 0.16 and a recuperator outlet pressure of 2 inches H₂O were assumed. This curve is presented in Figure 3.10.

Because of backpressure developed in the recuperator combustion flow passage/rotameter vent system during testing (caused by resistance to flow in the rotameter and connecting tubing), interpath leakages presented in Appendix 2 do not conservatively reflect interpath leakage characteristics. However, at each test condition, the recuperator combustion gas flow passage was totally isolated following measurement of interpath leakage using the rotameter. While isolated, the equilibrium pressure of the passage was measured.

Using the previously presented leakage relationship and the measured combustion gas flow passage pressure and average air flow passage pressure, a plot of interpath leakage versus the pressure difference across the interface was generated. This plot is included as Figure 3.11. Again using worst case data to define an equivalent orifice, recuperator interpath leakage is as follows (for a 52 module stack):

$$\text{leakage (lb/min)} = .0397 \sqrt{\Delta P}$$

where ΔP is the average pressure difference between the air and combustion gas flows in inches of water.

The preceding conservatively presents recuperator leakage characteristics. The data presented reflects the leakage of the entire test apparatus. A significant portion of the measured leakage could perhaps be attributed to system losses at blowers, pipe junctions, damper operator pipe penetrations, etc.

FIGURE 3.10. Percent Leakage versus Recuperator Reynolds Number (Based upon Cold/Warm Flow Results)

FIGURE 3.11. Interpath Leakage versus Average Pressure Differential Across Interface
(Based upon Cold/Warm Flow Results)

3.3.1.3.2 Hydraulic Tests

Presented in Appendix 2 are the recuperator air flow pressure drops measured over a range of flow rates. These data were nondimensionally correlated on a Stanton diagram which expresses the friction factor of the flow as a function of the Reynolds number. The friction factor was calculated from the Darcy-Weisbach equation:

$$f = \frac{\Delta P}{\rho \left(\frac{L}{D} \right) \left(\frac{V^2}{2g} \right)}$$

where:

f = Darcy friction factor (dimensionless)

ΔP = pressure drop through recuperator (lbf/ft²)

L = recuperator path length (ft)

D = recuperator hydraulic diameter (ft)

V = air velocity in recuperator (ft/sec)

g = gravitational constant (lbf ft/lbf sec²)

ρ = average air density (lbf/ft³)

The Stanton diagram of the test data is included as Figure 3.12. Included in the figure for comparison purposes are classical smooth pipe and rough pipe characteristics.

Considerable scatter in the test data exists for Reynolds numbers below 10,000. This was due to the inability to accurately measure the very small pressure decreases in the air flow stream present during operation at these low Reynolds numbers. Air side pressure drops at Reynolds numbers below 10,000 were on the order of 0.02 in H₂O.

FIGURE 3.12, Friction Factor versus Reynold's Number - Cold Flow Tests

3.3.1.3.3 Heat Transfer Tests

Convective heat transfer characteristics were determined from air side recuperator data. The following equations were used in correlating the test data:

$$q = M C_p (T_{air_{out}} - T_{air_{in}})$$

$$h = q / [A_i (T_{i_{ave}} - T_{air_{ave}}) + A_w (T_{w_{ave}} - T_{air_{ave}})]$$

$$T_F = 1/2 [T_{air_{ave}} + (A_w T_{w_{ave}} + A_i T_{i_{ave}})/(A_w + A_i)]$$

$$Re = \frac{V D_H \rho}{\mu}$$

$$Nu = h D_H / K$$

where:

A_i = interface heat transfer area (ft²)

A_w = wall heat transfer area (ft²)

C_p = heat capacity of air evaluated at $T_{air_{ave}}$ (BTU/lbm°F)

D_H = hydraulic diameter (ft)

h = convective heat transfer coefficient (BTU/hr·ft²·°F)

K = thermal conductivity of air evaluated at T_F (BTU/hr·ft²·°F)

M = air mass flow rate (lbm/hr)

Nu = Nusselt number (dimensionless)

q = heat transfer rate from interface and walls to air (BTU/hr)

Re = Reynolds number (dimensionless)

$T_{air_{ave}}$ = average air temperature (°F)

$T_{air_{in}}$ = inlet air temperature (°F)

$T_{air_{out}}$ = outlet air temperature (°F)

T_F = film temperature (°F)

$T_{i_{ave}}$ = average interface temperature (°F)

$T_{w_{ave}}$ = average wall temperature ($^{\circ}\text{F}$)

V = fluid velocity (ft/sec)

μ = air viscosity at T_f (lbm/ft $\cdot$ sec)

ρ = density (lbm/ft 3)

In calculation of the convective heat transfer coefficient, data from the central 38 modules of the recuperator column were used. Bulk air temperatures were averaged from measurements at modules 5, 25, and 45. Interface and wall recuperator temperatures were averaged from measurements on modules 14, 24, and 34 (and corresponding adjacent modules).

Included as Figure 3.13 is a plot of Nusselt number versus Reynolds number generated from the test data. All fluid properties used in data correlation were evaluated at the average film temperature (average of heat transfer surface temperature and bulk air temperature). Also included in the figure is the classical Nusselt/Reynolds number relationship for a straight pipe for comparison purposes. A least squares curve fit of the data in Figure 3.13 yielded the following relationship for recuperator Nusselt number:

$$Nu = 0.0486 Re^{0.786}$$

Variation in the average air Prandtl number was less than 2 percent over the range of test conditions and thus the effect of Prandtl number variation upon Nusselt number could not be determined. Assuming Prandtl number effects similar to those for a straight pipe, the Nusselt number correlation for the recuperator is:

$$Nu = 0.0542 Re^{0.786} Pr^{1/3}$$

FIGURE 3.13. Nusselt Number versus Reynolds Number - Warm Flow Tests

3.3.2 High Temperature Tests

3.3.2.1 Objectives

The objective of the high temperature testing program was to determine the heat transfer, leakage and hydraulic performance characteristics of the prototype ceramic recuperator over a wide range of conditions during high temperature, steady state operation. The results of the high temperature tests, in conjunction with convective data obtained during low temperature testing, enabled characterization of recuperator radiation heat transfer phenomena.

3.3.2.2 Test Apparatus

The test apparatus was constructed at the Millcreek Glass Company facility located in Salt Lake City, Utah.

3.3.2.2.1 Recuperator and Support Structure

The recuperator assembly consisted of 62 ceramic modules. Fifty-eight of the modules were Carborundum Type KT Silicon Carbide. The remaining four modules were Alpha Glass ECP-3 Alumina-Chrome-Plastic modules. The ECP-3 modules were situated at the top of the recuperator column.

Each interface between modules in the recuperator assembly was gasketed with two 1/16-inch thick pieces of Fiberfrax¹ ceramic fiber paper material. During assembly, each joint was custom fit (additional layers of gasket material were added as required) to minimize recuperator interpath and external leakage during operation. The gasket material was rated for continuous use at 2300°F.

To minimize recuperator heat loss during operation, the column was insulated with 6 inches of Cerablanket² refractory fiber thermal insulation. The

¹Carborundum Company Trademark

²Johns-Manville Company Trademark.

batt insulation was installed in 1-inch layers with overlapping joints. Each layer was banded to the recuperator.

The recuperator column was supported by the lower transition assembly. The structural portion of the assembly consisted of a horizontal Monofrax¹ plate supported by three vertical ceramic plates (one Monofrax¹ and two KT¹ Silicon Carbide). The area between the vertical plates was filled with a high temperature, castable refractory (Greencast 97²). Passages for the air and combustion gas flows were molded into the castable refractory during fabrication. Two holes were drilled in the horizontal Monofrax plate to form manifolds for the air and combustion gas flows.

A transition assembly was also situated on the top of the recuperator column. This assembly was fabricated similarly to the lower transition piece (without structural plates) from a castable refractory (Greencast 97²). In addition to acting as a manifold for air and combustion gas flows, the upper transition also served to compress the recuperator column to enable sealing of the gasketed joints.

A carbon steel structure supported the lower transition assembly, recuperator column, and upper transition assembly. This structure rested on the floor and was horizontally supported by the building. A 3/4-inch thick plate, supported by 3"X3" angle welded to the structure, formed a platform which supported the lower transition assembly. A 3/4-inch stainless steel plate was suspended from the top of the structure by four 3/4-inch threaded rods. This plate supported the upper transition assembly. The upper plate was unrestrained in the upward vertical direction to allow for thermal expansion of the recuperator column during operation.

¹Carborundum Company Trademark

²A. P. Green Company Trademark

A photograph of the test apparatus is included as Figure 3.14. Shown in the photograph are the insulated recuperator column, the column support structure, the lower transition assembly, and the recuperator air outlet piping.

3.3.2.2.2 Piping and Auxiliary Equipment

The configuration of the entire test apparatus is schematically presented in Figure 3.15. As shown in the schematic, the recuperator was operated in a counterflow mode with the combustion gas entering at the bottom of the recuperator and the air entering at the top of the recuperator.

Air was supplied to the inlet port of the upper transition assembly from a centrifugal blower (rated for 10 in. H_2O pressure at 360 SCFM delivery). The piping run from the blower to the transition assembly consisted of 3-inch diameter carbon steel tubing and of an orifice run (including straightening vanes) fabricated from 3-inch pipe. Final attachment to the transition was made with 3-inch diameter stainless steel tubing due to temperature considerations. A blast gate was situated in the piping run to regulate recuperator air flow.

Air was exhausted from the recuperator through the lower transition assembly. This heated air was piped from the transition exit port to the outside of the building. Initial attachment to the transition pipe was made with a 14-inch length of Coors AD-998, 5½ inch diameter alumina tubing. Attached to the alumina tubing was a run of 3 inch diameter stainless steel tubing components. Final piping to the building exterior was made with 3 inch carbon steel tubing components. Piping materials were graduated as indicated due to the temperatures present in the air exhaust stream during testing. An orifice run fabricated from 3 inch piping was situated in the carbon steel tubing run near the exit to the building.

Figure 3.14. Photograph of High Temperature Test Apparatus

FIGURE 3.15. High Temperature Test Apparatus Schematic

Combustion gases were supplied to the recuperator from a furnace designed and constructed for the testing by Millcreek Glass Co. The combustion gas inlet port of the lower transition assembly was piped to the exhaust manifold of the furnace with a section of 4 inch diameter alumina tubing.

Combustion air was supplied to the furnace burner from a centrifugal blower. The blower was piped to the furnace with 3 inch diameter carbon steel tubing. An orifice run was also located in the air supply line. A blast gate was situated in the line to regulate air flow.

Natural gas was supplied to the furnace burner from the building supply line. Standard steel piping components were used to plumb the system. Included within the line were a needle valve and a gas meter for regulating and measuring, respectively, of the natural gas flow to the furnace.

A draft was induced in the combustion gas side of the recuperator by a blower situated in the exhaust line running from the upper transition port to the building exterior. A bypass line was provided upstream of the blower to enable mixing of ambient temperature air with the combustion gases. This was required due to blower gas temperature limitations. The exhaust piping run was again graduated with ceramic, stainless steel, and carbon steel components due to temperature considerations.

3.3.2.2.3 Instrumentation

A schematic of system instrumentation is presented in Figure 3.16. Recuperator temperature measurement locations are outlined in Figure 3.17. As shown in the schematic, the flow rates of air and combustion gas entering and exiting the recuperator were measured. Combustion air entering the furnace and recuperator air side inlet and exhaust flows were measured with orifice metering assemblies. These assemblies, fabricated from 3 inch pipe and flange tap-type orifice flanges, were designed per ASME specifications. Orifice

- (T) Temperature Measurement
- (P) Static Pressure Measurement
- (ΔP) Differential Pressure Measurement
- (M) Metered Flow Measurement

FIGURE 3.16. System Instrumentation - High Temperature Test Apparatus

FIGURE 3.17. Recuperator Temperature Measurement Schematic - High Temperature Test Apparatus

plates of several hole diameters were provided to maintain orifice pressure drops between 0.25 and 2 inches of water throughout the range of specified flow rates.

Due to the temperature of the combustion gas exhaust present during testing, measurement of this flow rate with a standard orifice assembly was not possible. A high temperature pitot-static tube, fabricated from inconel, was selected for measuring the flow. This instrument was installed downstream of the upper transition assembly a sufficient number of pipe diameters to ensure that fully developed velocity profiles were present in the flow at the point of measurement.

Natural gas flowing into the furnace burner was measured with a displacement-type meter.

Pressure taps were located in each flow passage of both transition pieces to enable measurement of recuperator pressure drops during operation. These pressures were measured with inclined manometers, as were all differential pressures across flow measurement devices.

System temperatures were measured with Type S (platinum vs. platinum - 10 percent rhodium) thermocouples. The leads of these thermocouples were inserted within alumina protection tubes.

As shown in the recuperator temperature measurement schematic, several bulk fluid temperatures and module material temperatures were monitored during testing. Type S thermocouples were again used to measure recuperator temperatures. Some Type K thermocouples were also used in the lower temperature regions of the recuperator.

For thermocouple assemblies used for measurement of bulk air and combustion gas temperatures, the protection sheath and exposed thermocouple junction were inserted into a hole in the sidewall of a 1/4 inch diameter alumina tube.

Upon installation, the axis of the 1/4 inch tube was aligned parallel to the direction of flow in the recuperator, enabling aspiration of the shielded thermocouple during operation. The tube was required to mitigate temperature measurement errors due to radiation effects.

For thermocouples used to measure module surface temperatures, the exposed thermocouple junctions were attached to the module with A. P. Green No. T-36 high alumina refractory cement. Care was exercised during installation to ensure that thermocouple wire did not come in direct contact with silicon carbide modules due to the possibility of decomposition of the wires by free silicon or carbon.

During testing, all temperature data were recorded on paper tape with a data logger. Pressure readings were manually recorded.

3.3.2.3 Testing Procedure

3.3.2.3.1 Leakage Tests

Leakage tests were conducted on the air side of the recuperator with ambient temperature air flows prior to the initiation of the heat transfer tests. Testing consisted of measuring recuperator inlet and outlet flow rates over a range of recuperator flows (and a corresponding range of average pressures).

Air side leakage rates were also monitored throughout the heat transfer testing program. During all testing, the column was compressed by essentially the entire weight of the upper transition assembly (which weighed approximately 800 lbs).

3.3.2.3.2 Hydraulic Tests

Air side pressure drops were measured during the ambient temperature leakage tests. Both air and combustion gas side pressure drops were measured at each steady state heat transfer test condition.

3.3.2.3.3 Heat Transfer Tests

Heat transfer tests were conducted over a range of recuperator operating conditions. Combustion gas inlet temperatures to the recuperator ranged from 500°F to 2200°F. At each temperature level, data were obtained at combustion gas flow rates of approximately 1 and 3 lbs/min (corresponding to Reynolds Numbers ranging from approximately 5,000-10,000). At each combustion gas flow rate, data were obtained with recuperator air flows ranging from approximately 0.2 to 6.0 lbs/min (corresponding to Reynolds Numbers ranging from approximately 1,000-30,000). The matrix of actual test conditions is included as Table 3.1.

Care was exercised to ensure that test data were obtained at steady state recuperator operating conditions. A data set was logged after critical system temperatures (recuperator inlet and outlet flows) were observed to vary less than 3 percent during a 45 minute time interval.

3.3.2.4 Results

3.3.2.4.1 Leakage Tests

Results of the ambient temperature leakage tests are included in Appendix 3. Leakage rates present during high temperature heat transfer testing are also included in Appendix 3. A plot of total recuperator leakage versus flow passage pressure (divided by density) is presented in Figure 3.18. As shown in Figure 3.18, considerable variation of the total recuperator leakage rate versus pressure was observed during the testing. Also, ambient temperature leakage rates were found to be higher than leakage rates from the heated column. It is hypothesized that higher leakage rates during the ambient tests resulted because the column was uninsulated at this time.

Similarly to analysis of the leakage data from the cold/warm flow apparatus, an equivalent orifice for the ceramic recuperator leakage was defined

Table 3.1

Heat Transfer Test Matrix - High Temperature Tests

Test No.	Recuperator Combustion Gas Inlet Temperature (°F)	Combustion Gas		Air	
		Flow Rate (lb/min)	Reynolds No.	Flow Rate (lb/min)	Reynolds No.
1	625	1.13	3,809	0.24	896
2	637	1.04	3,680	1.12	4,413
3	576	1.24	4,568	2.02	8,675
4	470	1.28	5,117	3.14	14,684
5	493	1.29	5,157	6.16	29,829
6	557	4.04	13,758	0.22	863
7	517	4.20	14,933	1.16	4,617
8	512	4.27	15,524	1.94	8,028
9	529	4.35	15,916	3.40	14,669
10	506	4.50	16,985	6.23	28,436
11	1094	1.77	4,921	0.26	816
12	1030	1.91	5,646	1.28	4,349
13	1011	1.94	5,996	2.56	9,825
14	1035	1.85	5,769	3.37	13,581
15	1062	1.86	5,899	6.73	30,080
16	993	3.48	9,567	0.28	913
17	972	3.57	10,163	1.38	4,756
18	1063	3.33	9,787	3.08	11,652
19	992	3.12	9,506	3.66	14,342
20	995	3.33	10,401	7.22	30,844
21	1590	1.86	4,352	0.43	1,166
22	1461	1.73	4,608	1.82	6,063
22A	1566	1.99	4,942	1.56	4,692
22-32	1732	2.03	4,827	1.61	4,631
23	1580	2.02	5,223	2.78	9,336
24	1551	2.09	5,498	3.80	13,165
25	1507	2.06	5,484	4.72	15,442
26	1637	3.21	7,304	0.88	2,399
27	1609	3.40	7,870	1.65	4,761
28	1566	3.69	9,135	3.58	11,649
29	1610	3.81	9,382	4.50	15,019
29-30	1540	4.03	10,279	5.20	17,708
30	1589	4.17	10,529	6.43	21,897
32	1957	2.19	5,088	2.21	6,479
32A	1944	2.06	4,650	1.59	4,363
33	1921	2.73	6,345	2.81	8,450
33A	1996	2.76	6,301	2.69	7,954
33B	1937	3.42	7,779	3.37	10,028
34	1933	1.96	4,590	2.74	8,374
37	1962	3.10	6,660	1.90	5,131
37-38	1955	2.88	6,435	2.83	8,034
38	1949	3.22	7,109	2.86	8,467
39	1913	3.45	7,841	4.19	12,934
47	2217	3.09	6,338	2.05	5,286
47-48	2135	3.14	8,245	2.99	8,245
48	2126	3.23	7,053	3.39	9,786

FIGURE 3.18. Recuperator Leakage versus Average Recuperator Pressure - High Temperature Apparatus

based on worst-case data observed during heat transfer testing. Due to the range of temperatures at which data were obtained, density was included within the orifice correlation as follows:

$$\text{leakage} \propto \sqrt{\frac{P}{\rho}}$$

where P is the average recuperator flow passage gauge pressure and ρ is the average recuperator fluid density. The worst-case leakage observed during the heat transfer testing was:

$$\begin{aligned}\text{leakage} &= 1.04 \text{ lb/min @ } P = 6.16 \text{ in H}_2\text{O} \\ \rho &= 0.035 \text{ lb/ft}^3\end{aligned}$$

which yields the following relationship for recuperator leakage (based upon a 62 module stack):

$$\text{leakage (lb/min)} = .078 \sqrt{\frac{P \text{ (in H}_2\text{O)}}{\rho \text{ (lb/ft}^3\text{)}}}$$

A plot of this relationship is included in Figure 3.18.

Leakage tests in the cold/warm flow test assembly were conducted at ambient temperature and thus the air density did not vary. Because of this, the density term was included within the constant. For comparison purposes, the cold/warm flow apparatus correlation can be modified as follows:

$$\begin{aligned}\text{leakage (lb/min)} &= .126 \sqrt{P} = .032 \sqrt{\frac{P}{\rho}} \\ \text{for } \rho &= 0.064 \text{ lb/ft}^3.\end{aligned}$$

Normalizing to a 62 module stack:

$$\begin{aligned}\text{leakage (lb/min)} &= \left(\frac{62}{52}\right) (.032) \sqrt{\frac{P}{\rho}} \\ &= 0.038 \sqrt{\frac{P}{\rho}}\end{aligned}$$

In comparing the expressions for leakage for the two tests, it can be seen that leakage rates from the ceramic recuperator were approximately twice those of the metallic recuperator. This was to be expected because the mating surfaces on the metallic modules had been machined whereas the ceramic module surfaces were in the as-cast form.

3.3.2.4.2 Hydraulic Tests

Recuperator pressure drop data obtained throughout the high temperature testing program were correlated via the classical Stanton diagram which expresses the friction factor as a function of the Reynolds Number. Buoyancy effects in the recuperator were considered as follows in calculating the friction factors:

$$\Delta P = f \frac{L}{D_H} \frac{\rho V^2}{2g_c} - \text{Stack Draft}$$

or

$$f = (\Delta P + \text{Stack Draft}) \frac{2g_c}{\rho V^2} \left(\frac{D_H}{L} \right)$$

where:

$$\text{stack draft} = \pm H (\rho - \rho_{\text{ambient}})$$

(The positive sign applies to upward recuperator flow and the negative sign applies to downward recuperator flow.)

f = recuperator Darcy friction factor (dimensionless)

L = recuperator path length (ft)

D_H = recuperator hydraulic diameter (ft)

ρ = average density of recuperator flow (lbm/ft³)

V = average recuperator velocity (ft/sec)

g_c = gravitational constant (lbm·ft/lbf·sec²)

ΔP = recuperator pressure (lbf/ft²)

ρ_{ambient} = density of ambient air (lbm/ft³)

H = recuperator height (ft.)

Friction factor versus Reynolds Number plots generated from the air side and combustion gas side test data are included as Figures 3.19 and 3.20, respectively. As shown in the figures, there was less scatter in the air side test data. This can be attributed to better measurement accuracy of the air side flow rate (due to use of an orifice meter versus a pitot-static tube on the combustion gas side).

Although there was not enough data obtained to completely identify the laminar and transition flow regimes for the recuperator, the following can be concluded from the data:

- 1) A laminar flow regime existed in the recuperator for Reynolds Numbers up to at least 1200. A best fit of the data yielded the following expression for the friction factor in the laminar regime:

$$f = 6.161 \times 10^{-20} \text{ Re}^{-7.036}$$

The above was based upon data obtained during operation with Reynolds Numbers ranging from 860 to 1160.

- 2) A turbulent flow regime existed in the recuperator for Reynolds Numbers greater than 4000. A best fit of the data yielded the following expression for the friction factor in the turbulent regime:

$$f = 1.066 \text{ Re}^{-.196}$$

- 3) The turbulent correlation may be conservatively applied to determine friction factors for Reynolds Numbers ranging from 1200-4000.

FIGURE 3.19. Recuperator Air Side Friction Factor versus Reynolds Number - High Temperature Tests

FIGURE 3.20. Recuperator Combustion Gas Side Friction Factor versus Reynolds Number - High Temperature Tests

3.3.2.4.3 Heat Transfer Tests

Results of the heat transfer testing are tabulated in Appendix 3. Data obtained during testing were correlated via a recuperator overall heat transfer coefficient. To eliminate uncertainty in the test data caused by entrance and exit temperature effects in the recuperator inlet and outlet transition assemblies, the overall coefficient for each test condition was based upon the central portion of the recuperator (between modules 6 and 54). The overall heat transfer coefficient is defined as follows:

$$U = q/A \bar{\Delta T}$$

$$q = Ma C_{pa} (Ta_{in} - Ta_{out})$$

$$\Delta T = \frac{(Tg_{in} - Ta_{out}) - (Tg_{out} - Ta_{in})}{\ln [(Tg_{in} - Ta_{out})/(Tg_{out} - Ta_{in})]}$$

where:

A = recuperator heat transfer interface area (ft²)

C_{pa} = average air heat capacity (BTU/hr ft²°F)

Ma = air mass flow rate (lbm/hr)

q = heat transferred to air (BTU/hr)

T = temperature (°F)

$\bar{\Delta T}$ = log mean temperature difference (°F)

U = overall heat transfer coefficient (BTU/hr·ft²·°F)

subscripts:

a = air

g = combustion gas

in = inlet

out = outlet

The calculated overall coefficient for each test condition is presented in Table 3.2. Plots of recuperator overall coefficient versus average combustion gas temperature for test cases having similar Reynolds Numbers are presented in Figures 3.21 through 3.26.

To depict the effect of radiation heat transfer upon recuperator performance, overall coefficients determined from test data were compared to coefficients calculated assuming only convection heat transfer. The convective relationship used was that determined from warm flow testing. The following were used to calculate the convective overall coefficient:

$$Nu = 0.0542 Re^{.786} Pr^{1/3}$$

$$h = \frac{Nu \cdot k}{D_H}$$

$$U = (1/h_a + 1/h_g)^{-1}$$

where:

D_H = hydraulic diameter

h = convective heat transfer coefficient (BTU/hr·ft²·°F)

k = fluid thermal conductivity evaluated at the film temperature (BTU/hr·ft·°F)

Nu = Nusselt Number (dimensionless)

Pr = Prandtl Number (dimensionless)

Re = Reynolds Number (dimensionless)

U = overall heat transfer coefficient (BTU/hr·ft²·°F)

Curves of the calculated convective overall coefficients versus average combustion gas temperature are also presented in Figures 3.21 through 3.26.

Comparison of the actual and convective overall coefficient curves in Figures 3.21 through 3.26 yield the following conclusions:

- 1) The actual and convective overall coefficient curves coincide at low

Table 3.2

Overall Heat Transfer Coefficient from Data
(High Temperature Tests)

Test No.	Reynolds No.		Gas Inlet Temperature (F)	Overall Heat Transfer Coefficient (U, BTU/hrft ² F)
	Air	Gas		
1	828	3,691	625	.98
2	4,507	3,662	617	2.27
3	8,877	4,647	540	2.62
4	14,825	5,217	424	2.73
5	30,327	5,367	412	2.83
6	780	13,500	557	.69
7	4,560	14,600	522	2.71
8	8,020	15,397	508	3.21
9	14,738	15,923	512	3.86
10	28,636	17,163	476	4.35
11	708	4,617	1094	1.01
12	4,335	5,462	1020	3.13
13	10,026	5,974	943	3.47
14	14,053	5,854	929	3.43
15	31,128	6,176	871	3.46
16	795	9,128	993	1.07
17	4,585	9,827	972	3.57
18	11,668	9,534	1007	4.20
19	14,451	9,336	913	4.30
20	31,695	10,510	855	4.81
21	1,019	4,056	1590	1.95
22	5,957	4,430	1362	3.86
22A	4,641	4,758	1513	4.11
22-32	4,571	4,656	1667	4.51
23	9,397	5,135	1408	4.56
24	13,960	5,507	1348	4.68
25	18,246	5,558	1310	4.74
26	2,175	6,918	1637	3.44
27	4,477	7,557	1604	4.86
28	11,503	8,796	1495	5.79
29	15,125	9,207	1477	6.22
29-30	18,247	9,893	1449	6.22
30	23,443	10,376	1431	6.65
32	6,326	4,886	1836	5.21
32A	4,281	4,507	1862	4.92
33	8,251	6,093	1805	5.75
33A	7,755	6,060	1879	5.67
33B	9,884	7,511	1835	6.29
34	8,381	4,549	1748	5.55
37	4,860	6,489	1908	5.73
37-38	8,035	6,555	1857	7.52
38	7,975	6,930	1875	6.42
39	12,912	7,716	1784	6.73
47	5,038	6,222	2119	6.40
47-48	8,014	6,544	2031	7.08
48	9,531	6,839	2013	6.77

FIGURE 3.21: Overall Heat Transfer Coefficient versus Combustion Gas Temperature
($Re_{air} = 5,125$, $Re_{cp} = 4,853$)

FIGURE 3.22. Overall Heat Transfer Coefficient versus Combustion Gas Temperature.
 ($Re_{air} = 4,704$, $Re_{cp} = 8,939$)

FIGURE 3.23. Overall Heat Transfer Coefficient versus Combustion Gas Temperature
($Re_{air} \approx 14,307$, $Re_{cp} \approx 10,550$)

FIGURE 3.24. Overall Heat Transfer Coefficient versus Combustion Gas Temperature
($Re_{air} = 14,000$, $Re_{cp} = 6,074$)

FIGURE 3.25: Overall Heat Transfer Coefficient versus Combustion Gas Temperature
($Re_{air} = 9,200$, $Re_{cp} = 5,734$)

FIGURE 3.26. Overall Heat Transfer Coefficient versus Combustion Gas Temperature
 ($Re_{air} = 850$, $Re_{cp} = 4,120$)

temperatures, thus validating the previously determined convective relationship.

- 2) At high temperature test conditions (average recuperator combustion gas temperatures of approximately 1750°F), 40-50 percent of recuperator heat transfer was due to thermal radiation phenomena.

From Table 3.2 and Figures 3.21 through 3.26, it can be seen that overall recuperator heat transfer coefficients for high temperature, turbulent flow test conditions ranged from between 5 to 7.5 BTU/hr·ft²·°F.

3.4 Characterization of Performance

3.4.1 Thermal Performance

Using the previously outlined computer model (Section 3.2), the thermal performance of the prototype recuperator (8 inch diameter modules) and of two additional recuperator sizes (15 inch and 30 inch diameter modules) was characterized over a range of flow rates and column heights (and corresponding air preheat temperatures). The range of recuperator module sizes considered in the characterization was selected on the basis of anticipated supplier fabrication capabilities in a production situation.

The following recuperator parameters were maintained constant in the analysis:

Combustion gas inlet temperature	= 2400°F
Air inlet temperature	= 100°F
Module material thermal conductivity	= 10 BTU/hr·ft·°F
Combustion gas carbon dioxide partial pressure	= 0.087 ATM
Combustion gas water vapor partial pressure	= 0.174 ATM

The analysis was based upon the following:

- 1) 15 and 30 inch recuperator modules were scaled up geometrically from the prototype size (module pitch to diameter ratio was maintained at 1:1).
- 2) Equal air and combustion gas flow rates were assumed.
- 3) Methane combustion with 10 percent excess air was assumed in determining combustion gas carbon dioxide and water vapor partial pressures.
- 4) The recuperator column was assumed to be well insulated, corresponding to 1 foot of insulation for the prototype, 2 feet of insulation for the 15 inch diameter recuperator, and 4 feet of insulation for the 30 inch diameter recuperator.

The results of the analysis are presented graphically in Figures 3.27, 3.28, and 3.29 for the 8 inch, 15 inch, and 30 inch diameter modules, respectively. The figures portray the air preheat temperature (recuperator air outlet temperature) as a function of recuperator height and mass flow rate. As shown in the figures, the air preheat temperature is not a linear function of recuperator height (or corresponding heat transfer area). For example, in the case of the prototype recuperator operating with air and combustion gas flow rates of 5.0 lbm/min, 33 modules are required to provide a 50% efficient recuperator yielding 1250°F preheated air. Ninety modules are required for the same prototype recuperator flow rates to provide a 70 percent recovery efficiency yielding 1710°F preheated air.

3.4.2 Hydraulic Performance

The hydraulic performance was characterized by calculating the pressure drop across the air and combustion gas sides of the recuperator for a given

FIGURE 3.27. Prototype Recuperator, Air Preheat Temperature Versus Size

FIGURE 3.28, 15 Inch Diameter Recuperator - Air Preheat Temperature versus Size

FIGURE 3.29: 30 Inch Diameter Recuperator - Air Preheat Temperature versus Size

system geometry, mass flow rate, and average air and combustion gas temperatures. The conditions used are the same as those in Section 3.4.1.

The governing equations are the following:

$$\Delta P = f \frac{L}{D_H} \frac{\rho V^2}{2g_c} \pm H (\rho_{atm} - \rho) \frac{g}{g_c}$$

with

+ = downward recuperator flow

- = upward recuperator flow.

$$f = \begin{array}{ll} 720/Re & Re \leq 1000 \\ .250 & 1000 \leq Re \leq 2000 \\ 1.066 Re^{-.196} & Re > 2000 \end{array}$$

where:

D_H = hydraulic diameter (ft)

f = darcy friction factor

g = gravitational acceleration (ft/sec²)

g_c = gravitational constant (lbm.ft/lbf sec²)

H = recuperator height (ft)

L = path length (ft)

Re = fluid Reynolds Number

V = fluid velocity (ft/sec)

ρ = fluid density at average temperature and pressure (lbm/ft³)

ρ_{atm} = atmospheric air density (lbm/ft³)

The results are shown in Figures 3.30, 3.31, and 3.32.

FIGURE 3.30. Recuperator Total Pressure Drop versus Number of Modules - Prototype Modules.

FIGURE 3.31. Recuperator Total Pressure Drop versus Number of Modules - 15 Inch Diameter Modules

FIGURE 3.32. Recuperator Total Pressure Drop versus Number of Modules- 30 Inch Diameter Modules

3.5 Performance Enhancement Studies

3.5.1 Introduction

Two means of enhancing the heat transfer performance of the recuperator were analytically investigated. These were:

- 1) Improvement of recuperator radiation heat transfer via use of transparent materials in the module heat transfer interface, via the addition of radiation-participating gases (CO_2 , H_2O , etc.) to the recuperator flows, and via use of module materials having high emissivities.
- 2) Enhancement of recuperator convective heat transfer by the addition of extended surfaces to the module interface.

Following is a quantitative assessment of the possible recuperator heat transfer performance improvement via these two means.

3.5.2 Enhancement of Radiation Heat Transfer

3.5.2.1 Introduction

The prototype recuperator was designed to operate with combustion gas inlet temperatures ranging from 800°F to 2800°F. To quantify the relative contribution of radiative and convective phenomena to total recuperator heat transfer over this temperature range, a simple mathematical model was developed. Included as Appendix 4 is a description of the simplified model.

Figure 3.33, which depicts the relative contribution of the convective and radiative heat transfer modes to total recuperator heat transfer over a range of temperatures, was generated using this model. As shown in Figure 3.33, the relative contribution of radiative heat transfer increases with increasing average interface temperature. At an average recuperator interface

FIGURE 3.33. Relative Contribution of Radiation and Convection to Total Recuperator Heat Transfer versus Average Interface Temperature

temperature of 1800°F (corresponding to a recuperator recovering 70 percent of the available energy for a 2800°F combustion gas flow), the radiative contribution to total recuperator heat transfer approaches 50 percent. In view of the preceding, enhancement of radiation heat transfer has the potential of significantly improving overall recuperator heat transfer performance.

3.5.2.2 Infrared Transparent Interface

Use of infrared transparent materials in the recuperator module heat transfer interface was proposed as a means of enhancing radiation heat transfer. A review of candidate high temperature materials indicated that several had high transmissivities in the infrared region and thus the concept was viable from a materials standpoint.

The evaluation of the effect of interface transmittance was made using the simple mathematical model. Module interface transmittances of 0.18 and 0.75 were used in the evaluation. For each transmittance, recuperator performance was assessed at average combustion gas temperatures of 1836°F and 2436°F. In the analysis, recuperator wall surfaces were assumed to have an emissivity of 1.0

Presented in Figure 3.34 is a comparison of total recuperator heat transfer for the four transparent interface cases with baseline zero interface transparency cases. As shown in the figure, the net effect of interface transparency is a decrease of up to 3 percent at intermediate temperatures and of up to 17 percent at high temperatures in total heat transfer. This is due to a lower interface temperature (as a result of transparency) which then results in less effective heating of the air by convection. This is balanced somewhat by hotter confining walls but not entirely. Thus, enhancement of radiation heat transfer via use of a transparent interface adversely affects recuperator performance due to convective effects.

RELATIVE EFFECTIVENESS (Q/Q_{Baseline})

FIGURE 3.34. Relative Effects of Interface Transmissivity

3.5.2.3 Surface Emissivity

Exact information on the emissivity of refractory materials at various temperatures is not readily available. Suppliers report that silicon carbide materials have emissivities on the order of 0.9 or higher over the temperature range of interest (Reference 4). Tooley (Reference 5) gives an emissivity of 0.91 for clean brick and tile. However, Tooley indicates that glazed refractories have emissivities on the order of 0.7. It is expected that the emissivity of a clean, white alumina surface would be lower than that of clean silicon carbide, but after glazing and discoloration from use, there would be less difference between the two.

To define the effect of surface emissivity upon recuperator heat transfer, test cases were run for emissivities of 0.7, 0.9, and 1.0 using the mathematical model. As shown in Figure 3.35, surface emissivity over the range considered has little effect upon recuperator heat transfer. A 2 percent decrease in heat transfer resulted from changing the emissivity from 1.0 to 0.7.

3.5.2.4 Gas Stream Doping

Radiant heat transfer from hot gases to confining walls depends upon the gas and wall temperatures, the geometry of the enclosure, and the partial pressure of the emitting gases. If more than one gas emitter is present, the total emissivity is the sum of the individual parts.

Gas emissivity enhancement can be accomplished in two ways: An effective absorber-emitter such as H_2O or CO_2 can be injected directly into the gas stream (doping) or the ratio of O_2 to N_2 in the combustion air can be altered so that after combustion the inert N_2 contributes less to the total pressure thus permitting greater radiation by H_2O and CO_2 (oxygen enrichment). For cases in which recuperated air is used for furnace combustion, addition of a

RELATIVE EFFECTIVENESS (Q/Q_{Baseline})

FIGURE 3.35. Relative Effects of Surface Emissivity

doping agent to the air side also results in enhancement of combustion gas side emissivity. Oxygen enrichment also results in higher flame temperatures because of the reduced relative amount of N_2 . Since the oxygen enrichment method does not affect the absorptivity of the recuperated gas, the overall effectiveness is less than when the cooling gas is doped directly.

To determine the effect of doping upon heat transfer, air and combustion gas emissivities and absorptivities were calculated for CO_2 doping concentrations ranging from 0 to 5 percent. The emissivities and absorptivities were calculated using average recuperator temperature conditions per the procedure presented in Kreith (Reference 6). Calculated values are presented in Table 3.3.

Using the model, recuperator heat transfer for CO_2 doping concentrations ranging from 0 to 5 percent was calculated. The results are presented in Figure 3.34. As shown in the figure, recuperator heat transfer was enhanced by 23 percent via 5 percent CO_2 doping.

Because of the large quantities of N_2 present in air, effective doping requires large volumes of doping agent. The formulas for the undoped and 5% doped cases are:

and

and

Thus, a 5 percent CO_2 doping results in about a 50 percent increase in exhaust CO_2 .

To achieve air side emissivities comparable to 5% CO_2 doping, a concentration of approximately 22 percent H_2O must be used. If the air stream is to

Table 3.3
Partial Pressures and Total Gas Emissivities
for CO₂ and H₂O Doping

	% CO ₂ Doping				% H ₂ O Doping			
	0.0	0.3	1.0	5.0	0.0	1.0	5.0	22.0
Air Side								
P _{H₂O}	0.0	0.0	0.0	0.0	0.0	0.010	0.050	0.220
P _{CO₂}	0.0	0.003	0.01	0.05	0.0	0.0	0.0	0.0
Exhaust Side								
P _{H₂O}	0.1667	0.1667	0.1667	0.1667	0.1667	0.1752	0.2091	0.5732
P _{CO₂}	0.0833	0.0863	0.0933	0.1333	0.0833	0.0825	0.0795	0.0666
ε _{Air @ 1696°R} (bulk air temp)	0.0	0.0063	0.028	0.060	0.0	0.006	0.019	0.069
α _{Air @ 1886°R} (wall temp)	0.0	0.0058	0.018	0.062	0.0	0.006	0.019	0.061
α _{Air @ 1953°R} (interface temp)	0.0	0.0055	0.027	0.063	0.0	0.005	0.022	0.058
α _{Gas @ 1965°R} (interface temp)	0.103	0.104	0.105	0.117	0.103	0.106	0.113	0.159
α _{Gas @ 1988°R} (wall temp)	0.101	0.102	0.103	0.115	0.101	0.105	0.111	0.159
ε _{Gas @ 2296°R} (bulk gas temp)	0.087	0.087	0.090	0.099	0.087	0.090	0.096	0.134

FIGURE 3.36. Relative Effects of CO₂ Doping

be used as preheated combustion air, 22% H_2O doping represents a major dilution of the combustion reactants and is not considered a realistic alternative. For the case of 5 percent H_2O doping, the increase in emissivity on the air side is approximately one-third that of 5 percent CO_2 doping.

Before adding doping agents to the air side can be considered, the effect of doping on the furnace operation must be examined. Doping can add to the effect of dissociation of combustion products and lengthen the flame. It can also reduce the adiabatic flame temperature. It is predicted that 5 percent CO_2 doping would reduce the temperature 220°F and that 5 percent H_2O doping would reduce the temperature 240°F.

3.5.3 Effect of Extended Surfaces (Fins)

The addition of extended surfaces (fins) to convective heat transfer surfaces increases the total heat transferred across the surfaces. The increase in heat transferred from a flat surface and a finned surface is not only dependent upon the additional surface area, but also upon the fluid convective heat transfer coefficient and the thermal conductivity of the fin.

To assess possible recuperator performance enhancement via the addition of extended surfaces to the module heat transfer interfaces, a parametric study was conducted. In the study, recuperator heat transfer performance was calculated for different fin configurations over a range of convective conditions. A parametric study was dictated due to the uncertainty of the effect of extended surfaces upon convective heat transfer coefficients.

Four module configurations were considered: 3 fins per interface surface on both air and gas sides, 3 fins per interface surface on only the air side (Figure 3.37), 5 fins per interface surface on both air and gas sides, and 5 fins per interface surface on only the air side (Figure 3.38). The effect of variation of the convective heat transfer coefficient with equal coefficients

FIGURE 3.37. Three Fin Configuration

SECTION A-A

FIGURE 3.38.. Five Fin Configuration

on the gas and air sides was studied for all four (4) fin configurations. The effect of different coefficients on the air and gas sides was studied for one fin configuration only (3 fins per interface surface on the air side only). For this portion of the study, the gas convective heat transfer coefficient was varied from 1/2 to 4 times the air side coefficient.

3.5.3.1 Analysis

The effect of fins on the heat transfer characteristics of the recuperator were analyzed as the increase in the overall heat transfer coefficient, U , with fins over that without fins. The overall heat transfer coefficient for the recuperator without fins is given by:

$$U_{w/o} = \left(\frac{1}{h_g} + \frac{1}{h_a} \right)^{-1}$$

And, for fins on both sides

$$U_w = \left(\frac{1}{H_g} + \frac{1}{H_a} \right)^{-1} \quad \frac{1}{A}$$

where,

$$H = h[\eta_f A_f + (A - A_B)]$$

which reduces to

$$H = hA$$

when fins are removed from one surface

where:

$U_{w/o}$ = Recuperator overall heat transfer coefficient without fins
(BTU/hr ft² °F)

U_w = Recuperator overall heat transfer coefficient with fins (BTU/
hr ft² °F)

h_g = Combustion gas side convective heat transfer coefficient (BTU/hr ft² °F)

H_g = Combustion gas side overall convective heat transfer coefficient (BTU/hr ft² °F) (Based upon area and fin efficiency)

H_a = Air side overall convective heat transfer coefficient (BTU/hr ft² °F) (Based upon area and fin efficiency)

η_f = Fin efficiency (dimensionless)

A_f = Fin surface area (ft²)

A_B = Base area of fin (ft²)

A = Total heat transfer area (ft²)

The fin efficiency, η_f is defined by Holman (Reference 7). The percent increase is defined as,

$$A\% = \frac{U_w - U_{w/o}}{U_{w/o}} \times 100$$

Four (4) fin configurations were considered: two (2) configurations with both air and gas sides finned, and two (2) configurations with only the air side finned. A three (3) fin and a five (5) fin module were the two (2) configurations considered for the above cases (see Figures 3.39 and 3.40).

For the configuration of three (3) fins on the air side, five (5) convective conditions were considered. The overall gas convective heat transfer coefficient was analyzed for $h_g = 1/2 h_a$ to $h_g = 4 h_a$ for multiples of the air coefficient of 1/2, 1, 2, 3, and 4. The air convective coefficient was varied over the range from 1 BTU/hr ft² °F to 30 BTU/hr ft² °F. The results are shown in Figures 3.41 to 3.44. For the other three (3) configurations the air and gas convective heat transfer coefficients were considered equal, over the range from 1 to 30 BTU/hr ft² °F, Figures 3.39 and 3.40.

FIGURE 3.39. Recuperator Performance, $h_a = h_g$, Finned Air Side Only, 3 Fin and 5 Fin Configuration

FIGURE 3.40. Recuperator Performance, $h_a = h_g$, Finned Air and Combustion Gas Side, 3 Fin and 5 Fin Configuration

FIGURE 3.41. Recuperator Performance, $h_g = \frac{1}{2}h_a$, Finned Air Side Only, 3 Fin Configuration

FIGURE 3.42. Recuperator Performance, $h_g = 2h_a$, Finned Air Side Only, 3 Fin Configuration

FIGURE 3.43. Recuperator Performance, $h_g = 3h_a$, Finned Air Side Only, 3 Fin Configuration

FIGURE 3.44. Recuperator Performance, $h_g = 4h_a$, Finned Air Side Only, 3 Fin Configuration

3.5.3.2 Results

The addition of fins to the recuperator interface surfaces increases the overall heat transfer coefficient of the recuperator. This means that for a given air preheat condition the required heat transfer surface is reduced. This follows from the heat balance equation:

$$\dot{M} C_p \Delta T = UA \overline{\Delta T}$$

The analysis shows that the greatest increase is achieved by using a 5 fin configuration with both the air and gas sides finned. An 85 percent increase was calculated for the case of equal convective heat transfer coefficients ($h_g = h_a$) of 1 BTU/hr ft² °F. This decreased to about a 30 percent increase for convective heat transfer coefficients of 30 BTU/hr ft² °F. As shown in Figure 3.37, finning of only the air side of the recuperator heat transfer interface (required when fouling is of concern on the combustion gas side), has the potential of enhancing recuperator performance on the order of 25-40 percent.

3.5.4 Conclusions

3.5.4.1 Radiation Heat Transfer Enhancement

The following was concluded from the radiation heat transfer enhancement studies:

- 1) Doping of the recuperator flow streams with 5 percent CO₂ has the potential of enhancing heat transfer performance 23 percent. However, the viability of this means of enhancement is questionable due to economic and furnace operation considerations.
- 2) Variation of recuperator module material surface emissivities in the range of 0.7 to 1.0 has little effect upon overall recuperator performance.

- 3) Use of infrared-transparent materials in recuperator module interfaces is detrimental to recuperator heat transfer performance due to convective effects. For high temperature operation, a decrease of 17 percent in recuperator performance was determined.

3.5.4.2 Convective Heat Transfer Enhancement

The following were concluded from the convection heat transfer enhancement studies:

- 1) The addition of extended surfaces to both sides of the recuperator module heat transfer interface has the potential of increasing recuperator heat transfer performance from 30 to 85 percent (depending upon recuperator convective conditions).
- 2) The addition of extended surfaces to one side of the recuperator interface has the potential of increasing recuperator heat transfer performance from 25 to 40 percent (depending upon recuperator convective conditions).
- 3) Additional factors such as fouling, pressure drops, and module manufacturing costs need to be investigated prior to recommending the use of extended surfaces in future recuperator module designs.

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4.0 MATERIALS SELECTION AND EVALUTION

4.1 Corrosion Tests

Exposure of candidate materials specimens to a simulated glass furnace exhaust continued in Phase II. The test equipment and test procedures were the same as in Phase I and are fully described in the Phase I final report (Reference 3).

The furnace was fired with natural gas at a rate of approximately 250,000 BTU/hr. and charged with a blend of normally encountered glass furnace volatiles at a rate of $2\frac{1}{2}$ pounds/day. At the recommendation of Brockway Glass, a blend of 8 parts (by weight) sodium sulphate, 1 part sodium carbonate and 1 part carbon was used in the furnace charge. An analysis of flue deposits, shown in Table 4.1, indicates the principal constituent is sodium sulphate (approximately 90% by weight).

The results of the tests are displayed in Table 4.2. It should be noted that many material suppliers were unwilling to reveal their material compositions and manufacturing techniques. As a result only trade names appear in the table.

4.2 Materials Properties Tests

4.2.1 Introduction

The purpose of the materials properties tests was to define the mechanical characteristics of candidate recuperator materials as functions of temperature and to identify any degradation caused by exposure to a corrosive exhaust environment. The data obtained permits optimum material selection for a given service. Tests were conducted for:

TABLE 4.1

Analysis from Flue of Materials Test Furnace
(Analysis conducted by Ford Chemical Lab, SLC)

Material	Percent by Weight	
	Flue Temperature = 1400°F	Flue Temperature = 1680°F
Aluminum	0.185	0.153
Calcium	1.494	0.985
Sodium	30.160	31.767
Chloride	0.040	0.012
Flouride	1.980	2.842
Sulfur	19.950	20.650
Silica	3.158	5.210

TABLE 4.2
RESULTS OF FLUE ENVIRONMENT MATERIALS TEST

Temperature (°F)	Material	Supplier	Port #	Dates	Observations
2600	SKT SiC	Carborundum	1	1/22-2/19	Slight warping. No other apparent deterioration.
	Vistal	Coors	1	12/1-1/22	Longitudinal crack. No other apparent deterioration.
	Alundum	Norton	1	10/18-10/25	Material softened and slumped.
2530	GTE Zirconia	GTE	1	10/18-11/21	Cracking increasing until failure.
	Alcoa 820A	Alcoa	4	12/21-1/15	Sample melted off.
	ECP 3	Alpha Glass	4	1/29-2/19	Material slumped.
2510	ECP 1		2	10/18-2/19	Material considerably corroded & weakened; binder washing away.
	ECP 2		3	10/18-2/19	Slight rounding on edges.
2270	ECP1		5	10/18-2/19	Slight etching of surface. No other apparent deterioration.
	ECP 2		6	10/18-2/19	Sample broken on 12/12. Interior considerably changed in appearance. By termination date some expansion of bottom surface evident.
	GTE Zirconia	GTE	7	10/18-12/21	Cracking increasing to failure. Some corrosion attack.
	Alcoa 620	Alcoa	7	1/15-2/19	Material warped upward.
2150	Alundum	Norton	9	10/18-11/21	Material warped upward.
1830	Inconel 600	Commercial	13	11/21-12/12	Considerable oxidation. Material curled upward.
1800	ECP 2	Alpha Glass	12	12/1-1/15	Considerable expansion of material. Sample broken in port.

Table 4.2 Cont.

Temperature (°F)	Material	Supplier	Port #	Dates	Observations
1740	Inconel 600	Commercial	17	10/23-12/1	Considerable oxidation.
	Stainless 309		16	11/1-11/21	Severe oxidation.
	Stainless 309		20	12/1-12/12	Severe oxidation.
1670	Coors Cordierite	Coors	21	12/1-2/19	Surface of material attacked, peeled back & cracked underneath.
1650	Garrett Airesearch Silicon Nitride	Garrett Airesearch	19	12/1-2/19	Heavily coated. No apparent deterioration.
1550	SKT SiC	Carborundum	23	12/1-2/19	Heavily coated. No apparent deterioration.
1550	Coors Zirconia	Coors	24	12/1-2/19	Glazed. No deterioration observable through glaze.
1460	ECP 1	Alpha Glass	26	12/1-2/19	Dry deposition. No apparent deterioration.
	ECP 2		25	12/1-2/19	Dry deposition. No apparent deterioration.
1440	Coors Mullite	Coors	28	12/1-2/19	Dry deposition. No apparent deterioration.
	Inconel 600	Commercial	27	12/1-2/19	Dry deposition. Superficial oxidation.

Corrosion in glass furnace exhaust
Young's Modulus (elasticity)
Thermal Expansion
Flexure Strength
Fracture Toughness

The materials to be tested as identified in Phase I (Reference 3) included alumina (AD995), two forms of silicon carbide (KT and Sintered Alpha) and mullite. During the Phase II effort Sintered Alpha and mullite were dropped from the program because fabrication of recuperator sections from these materials could not be achieved within the time and budget limitations.

4.2.2 Equipment and Testing Procedures

4.2.2.1 Exposure Furnace

Samples were exposed to a glass furnace exhaust by inserting them in the flue stream of a day tank furnace melting glass volatiles. The small samples were supported on kiln furniture constructed from alumina rods. This permitted exposure. In this manner test results were related to the operational material surface conditions.

The furniture consisted of 1/4 inch diameter, 998 alumina tubes (or, in some cases, rods) 3½ inches long. These tubes extended across the 2½ inch dimension of the flue and into holes drilled into the facing bricks on either side of the flue. Thus the tubes were supported from each end. Notches were made in these tubes to locate the round materials samples and prevent their rolling. Each level held all of the samples of one material to be exposed at that nominal temperature range in that exposure run.

Figure 4.1 shows a drawing of the furnace; the sample distribution is depicted in Figure 4.2. During the steady exposure, sample temperatures were verified periodically with Type S (Pt vs Pt 10% Rh) thermocouples. Glass exhaust atmospheres were maintained by the measured, periodic addition of glass

FIGURE 4.1. Cutaway View of the Day Tank Furnace Used in Exposing Test Samples to Glass Furnace Exhaust

FIGURE 4.2. Arrangement of Exposure Samples in the Furnace Flue

volatiles (1 part Na_2CO_3 : 8 parts Na_4SO_4 : 1 part C) to a bed of molten glass.

The environment to which the materials samples were exposed was created by the volatilization of a simulated glass batch in the natural gas fired furnace. The simulated glass batch was charged at a rate of 2.5 lb. per day using an automatic hourly charger for the 200 and 800 hour runs and manually for the 50 hour run. The furnace burned approximately 250 cubic feet per hour of natural gas. These conditions simulated the density of these volatiles that would be seen in the exhaust stream of a large industrial furnace melting soda lime glass (Reference 8).

There are two (2) types of corrosion effects to be considered; erosive and alterational. Erosive corrosion implies a loss of material and alteration corrosion implies a change in material properties.

Changes in material properties (macrostructural changes) could be the result of microstructural changes in a material. X-ray analyses performed by the Oak Ridge National Lab were used to detect any changes of this type. The Oak Ridge results are reported in Appendix 5.

4.2.2.2 Test Furnace and Load Frame

All tests for elasticity, thermal expansion, flexure strength and fracture toughness were performed in a C-M model 1500 M electric furnace mounted on a hydraulically controlled 150,000 pound load frame. This load unit provided an extremely rigid frame for the tests.

The 1500-M furnace was modified to permit insertion of alumina ceramic load rods top and bottom and alumina strain rods in the bottom. Figure 4.3 shows the laboratory setup for these tests. The furnace load rod entry ports were sized to a minimum to reduce convection. Test parts were heated principally by radiation. Convection did not affect the test specimen but it did

FIGURE 4.3. The Test Furnace-Load Frame Setup Used to Measure Materials Properties at Elevated Temperatures

transport heat upward toward the sensitive load cell. Conduction of heat up the load rod also contributed to a heating of the load cell. Load cell cooling was provided by a directed-air-blower.

Load cells were calibrated using a standard Morehouse Ring Dynamometer for loads above 100 pounds and with calibrated weights for loads below 100 pounds. Electronic amplifiers and recorders were calibrated using standard techniques. All measurements were recorded on an H.P. X-Y1-Y2 recorder. Displacements for elasticity and expansion were measured using Shaevitz $\pm .010$ LVDT's which yielded reproducible resolutions of 10^{-5} inches at elevated temperatures. LVDT's were referenced directly to sample ends. A backup displacement gage consisting of dial indicators on the load piston provided reproducible displacements of about 10^{-4} inches. This backup system had to be calibrated against the LVDT system and was only reliable for tests that followed a precise heating schedule, since the conductive heating of long load rods entered into the total measured displacement.

4.2.2.3 Elasticity and Thermal Expansion Tests

The configuration used for the compression type elasticity tests and thermal expansion tests is shown in Figure 4.4. The concentric rods and tubes provided a reliable measure of strain across the sample. The system used two LVDT's, one on each side of the sample to average out for any tilt that may have resulted from nonuniformity of sample material or departure from parallelness at the ends.

At low loads, the stress-strain curves were nonlinear because of small nonconformances in the loading surfaces. The porous, granular nature of the test materials also contributed to the non-linearity of the stress-strain relationship. In order to obtain the most consistent results, all elasticity

FIGURE 4.4a. View of a Test for Elasticity and Thermal Expansion

(b) SAMPLE IN PLACE

FIGURE 4.4b. View of a Test for Elasticity and Thermal Expansion

(c) LVDT'S IN PLACE

FIGURE 4.4c. View of a Test for Elasticity and Thermal Expansion

measurements were made between 5,000 and 10,000 psi with the modulus defined as the average slope between these loads. Thus, the emphasis is placed upon the relative differences between sample sets to more accurately define exposure related changes.

Typically, the stress-strain curve for porous materials such as rocks is different on the loading and unloading sides. A similar result was expected for these ceramics and thus both loading and unloading values were obtained.

After assembling the test, the modulus of elasticity at room temperature was measured by loading to 10,000 psi and then unloading to 100 psi. During all subsequent heating, the load was maintained at about 100 psi in order to maintain alignment. After stabilizing at a preset temperature, values for thermal expansion and elasticity were obtained. The small (100 psi) static load resulted in a small displacement of less than 2×10^{-4} inches. At 600°C the thermal expansion of SiC results in a strain of about 5×10^{-3} so the error due to load resolution is less than 1% at 600°C and less than 0.5% at 1200°C.

The strain system was calibrated using a fused quartz sample whose properties are well known and for which the coefficient of thermal expansion is an order of magnitude smaller than the materials under test. To minimize the error due to thermal transients, all tests were begun with all components at room temperature. Heating and testing followed a rigid time schedule.

Test samples were 0.5 inches in diameter by 1.5 inches long and were cut from materials fabricated similarly to the recuperator modules. Exposed samples were cleaned or ground as necessary on the ends only to permit alignment.

4.2.2.4 Flexure Strength Tests

Flexure strength was determined from 4-point bend tests with load rates and dimensions meeting ASTM standards. In order to obtain values at very high temperatures a special 4-point bend fixture was constructed from sintered alpha SiC. The fixture is shown in Figure 4.5. Bend points were 0.5 inches apart for the load and 1.5 inches apart for support.

All alumina samples were cut from a single plate. KT samples came from two (2) sources. The samples used for the pristine (before exposure) tests were cut from a cast plate which later proved to be of lower density than the recuperator parts. Exposure samples and some second run, room temperature, pristine samples were cut from recuperator parts. Exposed samples were tested in the as-exposed condition.

4.2.2.5 Fracture Toughness Tests

There are several ways of making fracture toughness measurements. Due to the brittle nature of ceramics, and due to furnace size and sample fabrication limitations, use of the short rod specimen technique discussed by Barker was dictated (References 9, 10, and 11). Figure 4.6 shows the test assembly. Specimens were cut from stock fabricated similarly to module material. Specimen dimensions were 0.5 inches in diameter by 0.75 inches long. The sample was slotted from two sides on a diameter so as to leave an uncut wedge oriented lengthwise to the sample. For a given geometry, the value of $K_{I\text{CRS}}$ is directly proportional to the maximum applied load for a linear elastic material in which the non-elastic zone at the crack tip is very small compared to specimen dimensions.

For testing at elevated temperatures, grooves were ground into the load end face to accommodate high temperature grips. Samples were loaded (with a uniform 0.5 lb load) during heating to maintain assembly integrity.

FIGURE 4.5. The 4-Point Bend Fixture Shown Exploded (Top) and Assembled (Below) Used for Obtaining Flexure Strengths

FIGURE 4.6. The Test Assembly for Short Rod Fracture Toughness Tests

4.2.2.6 Creep and Thermal Conductivity

Thermal conductivity and creep tests were not included in the experimental program. However, suppliers were surveyed for pertinent data and a review of the applicable literature was made to define thermal conductivity as a function of temperature for aluminas and silicon carbides.

Creep is expected to be a significant parameter at these high temperature load bearing conditions. During Phase II, budget, supply and time considerations limited the creep studies to a literature search and a review of supplier data.

4.2.3 Test Matrix

Properties were defined as functions of temperature over a range of temperatures to 1500°C, or maximum use temperature, whichever was less. Properties were defined as a function of temperature for the pristine state and after exposure at temperature to a glass furnace operational environment. Exposure periods of 50, 200 and 800 hours were used. The test program is given in Tables 4.3 and 4.4 below. The test plan called for only two or three data samples at each data point because of time and budget constraints. This is considered an absolute minimum for statistical significance. Ceramics generally exhibit a large standard deviation for material properties. (Many researchers prefer to define destructive test values in terms of Weibull statistics and consider five or six samples to be minimal.) It is expected that since samples of a given material were fabricated in one or two batches, the variations will be smaller and fewer samples required. In the case of flexure tests for SiC, a need for more samples was apparent and these were added as necessary.

TABLE 4.3
HIGH TEMPERATURE TEST MATRIX

Test	Sample Size	Maximum Load Required	Approx. Load/Area	Type of Load	Instrumentation
E Young's Modulus	1/2" dia.	2,000 lbs.	10,000 psi	Comp	LVDT-Strain, Load, temp.
α Thermal Expansion	1/2" dia. 1.5" long	0	0	None	LVDT-Strain, temperature
IC Fracture Toughness	1/2" dia. 3/4" long at right angle to load	130 lbs.	NA	Tension	Load, temperature
Flexure Strength	1/8"X1/4" X2" bar	130 lbs.	50,000 psi (fiber stress)	Comp	Load at set rate, temperature

TABLE 4.4
ENVIRONMENT EXPOSURE MATRIX

<u>Test</u>	<u>Exposure Time, Hrs.</u>	<u>Exposure Temperature °C</u>	<u>Test Temperature °C</u>	<u>Samples Per Material</u>
Young's Modulus (E) and Thermal Expansion (α)	0	(RT)	RT-1500	3
	50	982	1000	2
	200	982	1000	2
	800	982	1000	2
	50	1150	1200	2
	200	1150	1200	2
	800	1150	1200	2
	50	1315	1315	2
	200	1315	1315	2
	800	1315	1315	2
Fracture Toughness (K _{IC})	0	(RT)	RT-1500	12
	50	982	1000	2
	200	982	1000	2
	800	982	1000	2
	50	1150	1200	2
	200	1150	1200	2
	800	1150	1200	2
	50	1315	1315	2
	200	1315	1315	2
	800	1315	1315	2
Flexure Strength	0	(RT)	RT-1500	18
	50	982	1000	3
	200	982	1000	3
	800	982	1000	3
	50	1150	1200	3
	200	1150	1200	3
	800	1150	1200	3
	50	1315	1315	3
	200	1315	1315	3
	800	1315	1315	3

4.2.4 Results and Discussion

4.2.4.1 Furnace Exposure Observations

Exposure tests were conducted in three (3) runs between mid-October, 1979 and mid-January, 1980, on prepared samples of Carborundum's KT silicon carbide and Coors' AD998 alumina. The test samples had been machined to proper dimensions for testing prior to exposure. The exposure periods were 50 hours, 200 hours and 800 hours. The samples were exposed (in each run) at three nominal temperatures ranges: 1315°C, 1150°C, and 982°C.

Flue exhaust gas temperatures were measured by thermocouples inserted through ports in the furnace flue. Average exhaust gas temperature profiles present in the furnace flue during testing are included as Figure 4.7. Denoted on the figure are sample locations within the flue. Exhaust gas temperature variations at any given flue location of approximately $\pm 100^\circ\text{C}$ were observed during the course of the exposure tests.

The 800 hour exposure was started normally and proceeded normally for 318 hours of exposure, at which point one wall of the furnace lining collapsed. The furnace was turned down so its condition could be evaluated. It was determined that a cold repair was needed, and thus the furnace was shut down. The inside of the furnace was repaired and the furnace reheated. Exposure was continued without complication.

A number of material samples were lost in the exposure runs. In some cases, the support furniture broke and in other it appears the sample may have simply been dislodged.

4.2.4.2 Corrosion

During the exposure tests, severe corrosion of the silicon carbide samples occurred. A number of silicon carbide samples disappeared, leaving the furniture standing. It is suspected that on the 800 hour run, some of the

FIGURE 4.7. Temperature Observations in the Exposure Furnace Flue

silicon carbide may have been corroded away until there was not enough left for the furniture to hold. The only three SiC samples remaining at the end of that run were the fracture toughness samples at the 982°C level. But, in any event, there was so little material left that it is easy to imagine that many other samples of this material may have become too small or misshapen to stay on their furniture. A summary of SiC weight loss and erosion rate measurements is provided in Table 4.5.

Some scatter in the measured SiC erosion rates presented in Table 4.5 is evident. The extraordinarily high erosion rates for the 50 hour tests are suspected to be related to the charging mode. During the 50 hour tests, the day's volatile charge was loaded in four separate batches during the 10 hour working day. Excessive carryover was observed in the flue during this manual loading. During the 200 and 800 hour tests the day's charge was loaded hourly in 24 smaller batches. Thus, it is likely that flue concentrations were at least 6 times greater during the short periods in the 50 hour test, causing higher erosion due to concentration effects. Flow disturbances in the test section (caused by sample supports) also caused scatter in the test data.

Erosion of alumina at all temperature levels was limited to slight rounding of corners; no significant dimensional changes occurred. No measurable weight losses occurred in any of the alumina samples. In fact, the uncleanable flue deposits generally resulted in a 0.5 to 1.0% weight increase in the alumina samples.

A visual appreciation for the corrosion of alumina and silicon carbide in a glass furnace exhaust is readily obtained from Figure 4.8 which shows samples from 50 and 800 hour exposures. These pictures reveal various types of condensate characteristic of temperature. For the most part, these condensates chip and flake off easily from exterior surfaces but have to be ground

TABLE 4.5

EROSION OF SiC IN GLASS EXHAUST

<u>Material</u>	<u>Exposure Hrs@°C</u>	<u>Wt. Loss Gms/Samples</u>	<u>Loss/Surface Area Gm/Cm²</u>	<u>Erosion Rate Cm/Year</u>
SiC-F.T.	50@1315	2.41/3	0.079	4.61
	50@1150	4.06/3	0.133	7.79
	50@ 982	1.14/3	0.037	2.18
	200@1315	0.336/2	0.017	0.24
	200@1150	2.44/3	0.080	1.17
	200@ 982	2.58/3	0.085	1.24
	800@ 982	14.37	0.472	1.72
SiC-E	50@1315	3.89/2	0.133	7.77
	50@1150	5.40/2	0.185	10.79
	50@ 982	2.19/2	0.075	4.38
	200@1150	3.94/2	0.135	1.97
	200@ 982	3.52/2	0.120	1.76
SiC-F.S.	50@1315	4.16/3	0.143	8.33
	50@ 982	1.51/1	0.155	9.06
	200@1315	0.739/2	0.038	0.555
	200@1150	0.536/1	0.055	0.804
	200@ 982	4.48/3	0.154	2.24

FIGURE 4.8a. Exposed Samples Taken From the Exhaust of a Glass Furnace After 50 Hours of Exposure

FIGURE 4.8b. Exposed Samples Taken From the Exhaust of a Glass Furnace After 800 Hours of Exposure

away from the interior slots in the fracture toughness samples.

Based on the exposure tests, it is concluded that silicon carbide is not suitable in glass furnace applications. Alumina AD995, however, does appear to be a potential candidate.

4.2.4.3 Elasticity

The pristine sample results are shown in Figure 4.9 for AD995 and Figure 4.10 for SiC-KT. As expected, the unloading slopes between 5,000 and 10,000 psi are quite different from the loading slopes for both materials. The flags on the data points in Figures 4.9 and 4.10 represent the sample standard deviation. The solid curves are linear regression fits to the data. The ratio of E_{load}/E_{unload} was quite different for the two materials. For AD995, E_{load}/E_{unload} was about constant up to 1000°C and then rapidly increased. As noted later, 800 to 1000°C is the region at which flexure strength begins to decrease in AD995. For this reason and because of the obvious change in the slope of E vs. T , separate fits to the AD995 data are shown above and below 1000°C. For SiC-KT the ratio of E_{load}/E_{unload} increased gradually between RT and 132°C.

The post exposure data for E is shown in Figures 4.11-4.14 compared to the pristine data fit. The flags again denote the sample standard deviation. For AD995 the loading and unloading data are within a standard deviation of the pristine data fit (Figures 4.11 and 4.12). In both the load and unload cases there is a small decrease in the mean values after exposure. However, the magnitude of change compared to the standard deviation does not justify a firm conclusion that exposure has affected this material property for AD995.

The post exposure data for SiC-KT in Figures 4.13 and 4.14 clearly demonstrates a change in the measured value of E for both loading and unloading. Cross sections of samples were cut and revealed that density and structural

FIGURE 4.9. AD 995 Elasticity Versus Temperature (Pristine Samples)

FIGURE 4.10. SiC-KT Elasticity Versus Temperature (Pristine Samples)

FIGURE 4.11. AD 995 Elasticity Versus Temperature for Relaxation (Exposed Samples)

FIGURE 4.12. AD 995 Elasticity Versus Temperature for Compression (Exposed Samples)

FIGURE 4.13. SiC-KT Elasticity Versus Temperature for Relaxation (Exposed Samples)

Figure 4.14. SiC-KT Elasticity Versus Temperature for Compression (Exposed Samples)

alterations in exposed samples were limited to a well defined "skin." The thickness of skin did not vary with exposure time but total dimensional change did. Thus it appears that exposure effects totally erode the outer surface at about the same rate as penetration to the inner material occurs.

4.2.4.4 Thermal Expansion

The thermal expansion data obtained with the precision LVDT's showed a good fit to the literature data (References 12, 13, and 14), a very small standard deviation, and no change as a result of exposure to glass furnace exhaust. This data is shown in Figure 4.15. The SiC-KT pristine data from LVDT's was not obtained, so the comparison is made to the dial gage data for pristine samples shown by the dash line.

The exposed results for AD995 were expected to be the same as the pristine since there were no apparent physical changes and E was essentially unchanged. The constancy with exposure for SiC-KT is a further indication that the main portion of samples was unaffected by the exposure.

The independent dial gage data is shown in Figure 4.16 and shows the somewhat larger spread or standard deviations inherent in this less accurate system (10^{-4} vs. 10^{-5} inches per resolution). Figure 4.17 shows the mean values for exposed LVDT data, pristine dial gage data and pristine LVDT data. The flags on the data indicate the sample standard deviations.

4.2.4.5 Flexure Strength

Figure 4.18 shows the pristine and exposed data for AD995 flexure strengths. The curve is fitted to the pristine data. Between 1300°C and 1400°C two curves are shown. The lower curve denotes a stress level at which the onset of a rapid creep is evident. Continued strain results in further creep and eventually a fracture. The ultimate fracture strength is shown by

FIGURE 4.15. Coefficient of Thermal Expansion Versus Temperature for AD 995 and SiC-KT (LVDT Data)

FIGURE 4.16. Coefficient of Thermal Expansion Versus Temperature for AD 995 and SiC-KT (Dial Gage Data)

FIGURE 4.17. Summary of Pristine and Exposed Thermal Expansion Data for AD 995 and SiC-KT

FIGURE 4.18: AD 995 Flexure Strength Versus Temperature (Exposed and Pristine Samples)

the upper curve. As before, the flags denote the sample standard deviations. For the few points without flags, the sample population was one.

Post exposure strengths for the 50 and 200 hour sets were significantly higher than pristine at 982°C and 1150°C. At 1320°C the data was ambiguous. Two samples showed a marked increase in strength of nearly 100 percent with no evidence of rapid creep while two other samples showed a 30 percent decrease in strength and definite pre-fracture creep.

Since the flexure strength is susceptible to changes in surface conditions, it is suspected that glazing may have filled surface defects and through capillary and cohesive means improved the surface characteristics of the samples.

The flexure strength data for SiC-KT is more difficult to interpret because of dimensional changes after exposure and the aforementioned differences in sample characteristics for exposed and pristine sample sets. All the KT data is shown in Figure 4.19. Again the flags denote the sample standard deviation. The solid curve and open circles are the pristine data taken on uniform low density samples. A subset of very low density samples indicates a density-strength relationship. This is confirmed by room temperature tests on high density samples from recuperator parts. Carborundum performed 30 such tests on samples taken from several parts (Reference 15) and samples were taken from one part and tested at Terra Tek. These results are shown in Figure 4.19. ORNL also tested recuperator module samples (see Appendix 5). The results agree very well with TTI and Carborundum data. The results indicate a large effect of porosity upon strength.

Since exposed samples were also taken from recuperator parts, a proper comparison to positive tests requires extrapolating the room temperature recuperator part, pristine data to 1500°C. This extrapolation is assumed to

FIGURE 4.19. SiC-KT Flexure Strength Versus Temperature (Exposed and Pristine Samples).

have a constant ratio to the low density set data and is shown by the dashed curve in Figure 4.19.

The exposed samples of SiC-KT were eroded sufficiently to alter their cross sectional shapes. Original rectangular cross sections had degraded to nearly elliptical shapes. The data was analyzed for both cases where measured dimensions were used as the length for rectangles (shown by the open data points) and ellipses (shown by the data points).

A further complication arises in the analysis as the effect of the corroded skin must be accounted for. If this skin has no tensile strength but does have a compressive strength to support the sample at the bend points, then the ratio of true strength to measured strength will depend upon the skin thickness, s , as follows:

$$FS = \frac{(\frac{1}{2} \text{ Load})(y)}{I} = (1.5 \text{ Load})/bh^2$$

$$\frac{FS \text{ true}}{FS \text{ measured}} = \frac{bh^2}{(b-2s)(h-2s)^2}$$

Thus a skin of weak material would result in a measured strength less than the actual flexure strength of uncorroded material. A skin thickness of 0.030 inches gives a ratio of FS_T/FS_M of nearly 5 which obviously indicates that in actuality either the skin is less than 0.030 inches or that it retains significant tensile strength even though it has lost some density and is quite porous. A skin thickness of 0.010 inches of zero tensile strength would give a ratio of 1.54 which is about the difference between the exposed rectangular measured and theoretical unexposed values. Since some of this difference can be accounted for by elliptical or nonrectangular specimen cross sections, the skin is either thinner than 0.010 inches or it retains some strength. It does

not appear that the strength of the material inside the skin is appreciably affected.

4.2.4.6 Fracture Toughness

Three difficulties arose in obtaining a complete set of fracture toughness data. The original test grips were of alumina which did not survive at temperatures above 1200°C. The reasons for this became obvious when the decrease in flexure strength above 800°C became known. A second difficulty was spurious grip breakage at low temperatures of both alumina and SiC grips. It is hypothesized because of the shape of the breaks in the grip teeth that the brittle nature of ceramics combined with the small but unavoidable irregularities in ground surfaces resulted in stress concentrations. The data for pristine conditions is not as complete as desirable at elevated temperatures but it is sufficient to support some very important conclusions. The test data and literature values for Refel SiC are shown in Figure 4.20. Also shown are two points where the grips broke without breaking the sample (solid points) indicating that fracture toughness exceeds this value.

From Figure 4.20 it can be seen that fracture toughness for KT is relatively constant at about $6 \text{ Ksi}\sqrt{\text{in}}$ up to at least 1100°C. An indication of the likely behavior of KT above 100°C is provided by work on Refel SiC, a similar material. Data from Refel G_{IC} measurements (References 16 and 17) have been converted to K_{IC} . These data are shown in Figure 20. The increase in K_{IC} for Refel between 1100°C and 1400°C is ascribed (Reference 18) to an increase in the plasticity of the free Si as temperature increases, which leads to an energy dissipation. The rapid fall at 1400°C results when the Si melts. It is likely that the value of AD995 is at least 4.5 to $5 \text{ KSi}\sqrt{\text{in}}$ up to about 800°C. Because of the change in AD995 flexure strength above 800°C one might also expect a change in fracture toughness in this range.

FIGURE 4.20. Fracture Toughness Versus Temperature - SiC-KT, SiC-Refel, and AD 995

The third difficulty encountered in testing was the heavy penetration of condensed matter into the pre-cut slots of samples. The only way to conduct a test was to remove these condensates with a thin diamond saw blade. This process resulted in second cuts into the base material so it was next to impossible to realign the samples properly while cutting. The process of cutting also violated the constraint that surfaces critical to a test result should not be altered after exposure.

4.2.4.7 Thermal Conductivity

Literature values of thermal conductivity for the reaction bonded SiC varied considerably as seen in Figure 4.21. The values for the early form of KT are quite low (Reference 12). An improved, fine-grained material called Super KT has considerably higher conductivity values (Reference 19). More recent data on KT material comparable to that used in making recuperator parts indicates thermal conductivities on the order of 33-42 Btu/hrft²°F for temperatures above 1000°F (Reference 20). Figure 4.22 shows values for several alumina compositions (Reference 14).

4.2.4.8 Creep

Very little specific data was located on creep for either AD995 or SiC-KT. The most likely source for KT data is Carborundum, and they indicated they have not made these measurements on KT but plan to do so about mid-1980 (Reference 21). Coors has not measured creep in alumina and has no plans to do so (Reference 22). Other likely sources including Art Heuer at Case Western Institute and Sheldon Wiederhorn at N.B.A. were not aware of any specific data on AD995 or KT. In lieu of specific data, an indication of probable SiC creep rates can be inferred from sparse data for Refel SiC (a reaction bonded or sintered SiC product with free Si similar to KT). Table 4.6 shows data

FIGURE 4.21. Silicon Carbide Thermal Conductivity Versus Temperature

FIGURE 4.22. Alumina Thermal Conductivity Versus Temperature.

TABLE 4.6
CREEP OF SiC CERAMICS

<u>Material</u>	<u>Test Temperature °C</u>	<u>Stress* KSI</u>	<u>Strain %</u>	<u>Time Hours</u>	<u>Reference</u>
Refel	1100	42,000	0.28	100	24
Refel	1100	42,000	0.73	100	24
Refel	1230	42,000	0.97	70	24
Refel	1000	43,900	0.01	100	21
Refel	1000	31,200	0.018	100	21
Refel	1000	68,000	0.028	100	21
Refel	1100	43,900	0.06	100	21
Refel	1100	43,900	0.105	100	21
Refel	1100	30,000	0.038	100	21
Refel	1200	30,000	0.12	100	21
Refel	1200	43,900	0.06	1	21
Refel	1200	33,600T	0.11	100	22
Refel	1200	22,500T	0.11	100	22
Refel	1100	10,200T	0.01	100	23
Refel	1100	20,300T	0.017	100	23
Refel	1100	25,400T	0.022	100	23
Refel	1200	10,200T	0.018	100	23
Refel	1200	20,300T	0.025	100	23
Refel	1200	25,400T	0.04	100	23
Refel	1300	10,200T	0.011	100	23
Refel	1300	20,300T	0.045	100	23
Refel	1300	25,400T	0.07	100	23
KT	1200	33,600	0.18	100	22
KT	1200	22,500	0.38	100	22

*Compressive or bending except where noted by T to indicate tensile.

(References 23, 24, 25, and 26) compiled by Edington et al. (Reference 18).

Meaningful data for alumina could not be located. Generally, the literature treats the case for isostatic pressing conditions for shape fabrication which obviously is not the desirable condition for an operating recuperator. For example, Fryer and Thompson (Reference 27) discuss the compressive creep at 1100°C to 1250°C of polycrystalline alumina pellets of 0.7 to 0.95 relative density. Lessing and Gordon (Reference 28) discuss polycrystalline alumina creep at 1400°C to 1500°C. Neither of these is applicable to the specific use conditions or alumina form.

5.0 STRUCTURAL ANALYSIS

5.1 Introduction

A stress analysis was performed on the subscale recuperator to determine the structural integrity of the module design. A three dimensional finite element stress analysis of the thermal load was performed by the Acurex Corp. Acurex analyzed a single silicon carbide module at three thermal loading conditions. Acurex's final report is included as Appendix 6.

The temperature distributions used for the analysis were determined experimentally using data from the warm flow test apparatus. These temperature distributions were scaled up from the medium temperature tests to the temperatures expected during high temperature operation.

The material properties used were determined experimentally as detailed in Section 4.0, Materials Selection and Evaluation.

5.2 Stress Model

Acurex developed a three dimensional finite element model of the subscale recuperator module. The ANSYS computer code was used to analyze the stresses within the module.

The subscale recuperator module was divided into 204 finite elements with 438 nodal points, Figure 5.1. The basic finite element used in the analysis was STIF 45. This is an 8 node solid brick. Each node has three degrees of freedom, so the brick can deform in any direction. The program calculated the displacement of each node based on the material properties and the thermal loading using a simultaneous solution.

It was assumed that the module was stress free at room temperature. The only boundary condition applied was the temperature distribution. The axial

FIGURE 5.1. Finite Element Model of Subscale Recuperator Module (Isometric View)

load was not considered since it makes the problem non-linear which was beyond the work scope for Acurex.

5.3 Experimental Program

5.3.1 Introduction

An experimental program was conducted to provide module temperature gradient data for use in the stress analysis work. Testing consisted of subjecting the cold/warm flow metallic recuperator assembly (with additional temperature instrumentation) to a series of steady state and thermal transient operating events (e.g. normal startup and shutdown, upset conditions, etc.). Throughout each transient event, temperature profiles on the upper and lower modules within the recuperator were measured at appropriate time intervals.

Project schedules dictated the use of the cold/warm flow assembly to provide the required data. Due to the temperature limitations of the apparatus, recuperator combustion gas inlet temperatures did not exceed 800°F during testing. Thus, it was necessary to extrapolate data obtained from the intermediate temperature tests to predict the temperature profiles that would be present during operation at high temperature. Later high temperature testing verified the conservatism of the approach used to extrapolate the intermediate temperature data.

5.3.2 Test Apparatus

The cold/warm flow test apparatus (previously described in Section 3.3 of this report) was used to conduct the thermal transient tests. In addition to the previously outlined instrumentation, 40 thermocouples were installed onto the surface of both the bottom and the top modules in the recuperator assembly. Presented in Figure 5.2 are the thermocouple placement locations for both the bottom and top recuperator modules. Unsheathed Type K thermocouples

FIGURE 5.2. Thermocouple Placement - Thermal Gradient Tests

of 22 gauge were attached to the surface of the modules in the positions shown using a peening technique. Output from these thermocouples and from the additional thermocouples in the assembly were monitored by a data logger throughout testing.

5.3.3 Testing Procedure

The testing program was formulated to encompass the thermal events to which a high temperature recuperator would be subjected during an operating lifetime. Normal events considered were startup, steady state operation, and shutdown. Upset events included in the test matrix were loss of air flow, loss of gas flow, and loss of both air and gas flow. Because the recuperator thermal response is highly dependent upon convective heat transfer (and associated air and combustion gas mass flow rates), each event was conducted at a lower and upper bound of the flow rate. The test program is outlined in Table 5.1.

5.3.4 Results

Data from all tests were analyzed to select the three most severe operating events (from a temperature and thermal gradient standpoint). The criteria used were as follows:

- 1) maximum module temperature
- 2) maximum module total temperature deviation:

$$D_T = \sum_{j=0}^{39} |T_j - T_r|$$

- 3) maximum module interface temperature deviation:

$$D_{INT} = \sum_{j=0}^9 |T_j - T_r|$$

- 4) maximum module interface temperature drop

$$\Delta T = |T_4 - T_r|$$

Table 5.1
Thermal Gradient Test Matrix

Event No.	Air Side		Combustion Gas Side				Comments
	Reynolds Number		Reynolds Number		Inlet Temperature		
	Beginning	End	Beginning	End	Beginning	End	
1	30,000	30,000	30,000	30,000	Ambient	800°F	Normal startup transient. Increase combustion gas inlet temperature at a rate of 100°F per hour
2	30,000	30,000	30,000	30,000	800°F	800°F	Normal steady state operation.
3	30,000	30,000	30,000	30,000	800°F	Ambient	Normal shutdown transient. Decrease combustion gas inlet temperature at a rate of 100°F per hour.
4	5,000	5,000	5,000	5,000	Ambient	800°F	Normal startup transient. Increase combustion gas inlet temperature at a rate of 100°F per hour.
5	5,000	5,000	5,000	5,000	800°F	800°F	Normal steady state operation.
6	5,000	5,000	5,000	5,000	800°F	Ambient	Normal shutdown transient. Decrease combustion gas inlet temperature at a rate of 100°F per hour.
7	30,000	0	30,000	30,000	800°F	800°F	Sudden loss of air flow.
8	30,000	30,000	30,000	0	800°F	---	Sudden loss of combustion gas flow.
9	30,000	0	30,000	0	800°F	---	Sudden loss of air and combustion gas flow.
10	5,000	0	5,000	5,000	800°F	800°F	Sudden loss of air flow.
11	5,000	5,000	5,000	0	800°F	---	Sudden loss of combustion gas flow.
12	5,000	0	5,000	0	800°F	---	Sudden loss of air and combustion gas flow.

where:

D_T = total module temperature deviation

D_{INT} = module interface temperature deviation

ΔT = module interface temperature drop

J = thermocouple position

T_j = temperature at position j

T_r = temperature at reference point defined as position 4

Using these criteria, normal startup, steady state operation, and loss of air flow were deemed to be the most severe operating events. Presented in Figure 5.3 and 5.4 are representative module temperature histories for these events. In each case, the most severe temperature gradients occurred in the bottom recuperator module with high air and combustion gas flows ($Re \approx 30,000$). Temperature gradients obtained for the three most severe operating conditions were extrapolated to predict the gradients which would be present during high temperature recuperator operation. Gradients were extrapolated to correspond to a recuperator operating at 70 percent efficiency with a $2400^\circ F$ combustion gas inlet temperature.

The scheme used for extrapolation of the temperature gradients was as follows:

- 1) The thermocouple located at the center of the module on the air side (T_4) was selected as a reference position.
- 2) The following relationship was used to determine the reference temperature at high temperature, steady state operating conditions:

$$T_4 = \left(\frac{1}{h_a} + \frac{\Delta X}{K} + \frac{1}{h_g} \right)^{-1} \left(\frac{1}{h_a} \right) (T_g - T_a) + T_a$$

FIGURE 5.3. Temperature History - Normal Startup and Steady State Operating Events

FIGURE 5.4. Temperature History - Loss of Air Flow Event

where:

h_a = convective heat transfer coefficient on air side (Btu/hrft²°F)

h_g = convective heat transfer coefficient on combustion gas side (Btu/hrft²°F)

K = recuperator material thermal conductivity (BTU/hrft°F)

T_a = bulk air side temperature (°F)

T_g = bulk combustion gas side temperature (°F)

ΔX = distance across interface (ft)

- 3) For extrapolation of reference temperatures corresponding to transient events, the following dimensionless temperature was defined:

$$\phi = \frac{T_{4(t)} - T_{4(0)}}{T_{4(\infty)} - T_{4(0)}}$$

where:

ϕ = dimensionless temperature

$T_{4(t)}$ = temperature at reference point at time t into event (°F)

$T_{4(0)}$ = steady state temperature at reference point prior to event (°F)

$T_{4(\infty)}$ = steady state temperature at reference point following event (°F)

- 4) To extrapolate temperatures for the remaining module positions, the following dimensionless temperature was defined:

$$\psi_i = (T_i - T_4) / (T_g - T_a)$$

where:

T_i = temperature at location i (°F)

T_4 = temperature at reference location (°F)

T_g = temperature of exhaust gas (°F)

T_a = temperature of air (°F)

Table 5.2

Extrapolated Module Temperature Distribution -
Normal Start-Up

Thermocouple Placement	Temperature (°F)	Thermocouple Placement	Temperature (°F)
0	901	20	894
1	833	21	839
2	949	22	940
3	932	23	915
4	931	24	934
5	918	25	920
6	897	26	926
7	879	27	870
8	832	28	886
9	811	29	875
10	862	30	842
11	925	31	886
12	924	32	941
13	850	33	900
14	798	34	935
15	757	35	948
16	754	36	913
17	892	37	779
18	945	38	753
19	941	39	785

Table 5.3
Extrapolated Module Temperature Distribution -
Steady-State

Thermocouple Placement	Temperature (°F)	Thermocouple Placement	Temperature (°F)
0	2090	20	2053
1	2065	21	2016
2	2102	22	2087
3	2085	23	2058
4	2090	24	2092
5	2077	25	2079
6	2058	26	2084
7	2036	27	2017
8	2024	28	2077
9	2004	29	2066
10	2054	30	2030
11	2087	31	2043
12	2063	32	2082
13	1993	33	2066
14	1971	34	2119
15	1931	35	2148
16	1941	36	2115
17	2099	37	1967
18	2160	38	1935
19	2150	39	1959

Table 5.4
Extrapolated Module Temperature Distribution -
Loss of Air Flow

Thermocouple Placement	Temperature (°F)	Thermocouple Placement	Temperature (°F)
0	2189	20	2159
1	2181	21	2137
2	2218	22	2191
3	2210	23	2172
4	2206	24	2197
5	2200	25	2192
6	2184	26	2188
7	2176	27	2159
8	2152	28	2166
9	2138	29	2160
10	2167	30	2153
11	2198	31	2176
12	2190	32	2206
13	2150	33	2182
14	2117	34	2201
15	2087	35	2214
16	2089	36	2185
17	2188	37	2098
18	2224	38	2079
19	2219	39	2093

Presented as Tables 5.2, 5.3, and 5.4 are the extrapolated temperature profiles for the startup, steady state, and loss of air flow events, respectively.

5.4 Results of Stress Analysis

Acurex performed a stress analysis for a subscale silicon carbide module under three different thermal loadings. The three conditions correspond to furnace startup, steady state operation, and loss of preheat air flow. The effect of various axial loadings on the stress distributions was not investigated by Acurex. Consideration of axial loads created a nonlinear problem which was beyond Acurex's scope of work.

The maximum and minimum stresses for each of the three conditions are presented in Table 5.5. The maximum tensile stress occurs during loss of air flow. The largest compressive stress occurs during start-up. Figures 5.6 to 5.10 show the stress distributions for the steady state case.

Table 5.5
Maximum and Minimum Stress

<u>Condition</u>	<u>Maximum Stress (psi)</u>	<u>Minimum Stress (psi)</u>
Start up	6181	-6407
Steady state	6310	-4730
Loss of air	6399	-4457

FIGURE 5.5. Steady State, Top View, Maximum Stress (psi)

FIGURE 5.6. Steady State, Bottom View, Maximum Stress (psi)

FIGURE 5.7. Steady State, Maximum Stress (psi)

FIGURE 5.8. Steady State, Top View, Minimum Stress (psi)

FIGURE 5.9. Steady State, Bottom View, Minimum Stress (psi)

Section A-A

Section B-B'

FIGURE 5.10. Steady State, Minimum Stress (psi)

Silicon carbide is a brittle ceramic material. The compressive strength of silicon carbide is much greater than the tensile strength. The tensile strength is approximately half that in a three point bending test (Reference 18).

The structural performance of the module is predicted as the probability of survival, using the statistical theory developed by Weibull (Reference 29):

$$s = \exp - \left\{ \int_v \left[\frac{\sigma(x,y,z) - \sigma_u}{\sigma_o} \right]^m d_x d_y d_z \right\}$$

where: S = probability of survival

$\sigma(x,y,z)$ = applied stress

σ_u = stress below which failure will never occur

σ_o = normalizing scale parameter locating the strength distribution

m = measure of scatter (Weibull material parameter)

v = material volume

It is assumed that $\sigma_u = 0$. This means that there is no lower limit below which there will not be failure. The volume effect was ignored since there was insufficient data to properly define it. The equation is therefore reduced to:

$$S = \exp - \left(\frac{\sigma}{\sigma_o} \right)^m$$

To determine the normalizing scale parameter, σ_o , it was assumed that the probability of survival is 5 percent when the stress reaches the maximum tensile strength.

$$\sigma_o = \frac{\sigma_{3\text{-point}}}{[-\ln(.05)]^{1/8}}$$

The available 4-point bend test data can be converted to 3-point bend test data by the equation:

$$\sigma_{3\text{-point}} = \left(\frac{m+2}{2} \right)^{1/m} \sigma_{4\text{-point}}$$

where $m = 8$ Weibull material parameter for silicon carbide, (Reference 18).

$$\sigma_{3\text{-point}} = (1.22) \sigma_{4\text{-point}}$$

The maximum tensile stress occurs under the "loss of air" conditions. The material temperature during this event is about 2000°F at which the tensile strength is approximately 14,000 psi. The corresponding probability of survival, S , is 99 percent. However, the most severe stress condition occurs during start-up, where the material temperature is about 1000°F and the tensile strength is 11,000 psi. The probability of survival, S , for this condition is 97 percent.

This analysis considers brittle fracture only; it is possible the module may fail in other modes such as creep, static fatigue or cyclic fatigue. These effects were not considered.

6. COMPARISON WITH EXISTING SYSTEMS

6.1 Introduction

Tennery and Wei (Reference 30) completed an assessment of the status of existing technology for recovering waste heat from hot flue gases. Following is a summary and analysis of their assessment.

Two types of devices are currently used for recovering waste heat from flue gases - regenerators and recuperators. Regenerators are periodic flow heat exchangers in which combustion gases and preheat air are alternately cycled through a thermal storage matrix. During the hot cycle, heat is extracted from the combustion gases and then stored in the matrix. During the cold cycle, heat stored in the matrix is transferred to the preheat air. Recuperators are devices operating on a continuous basis which transfer heat from the combustion gases to the preheat air through a material heat transfer interface.

Regenerators are typically constructed using high temperature refractory bricks and thus combustion gas inlet temperatures are not limited by materials considerations. Gunter (Reference 2) reports performance data for operational regenerators which indicate average air preheat temperatures in excess of 2000°F are possible for combustion gas temperatures of approximately 2600°F.

Three basic flow configurations are used in recuperators - counterflow, crossflow, and parallel flow. Thermodynamically, counterflow and crossflow configurations enable the highest heat recovery effectiveness (100% effectiveness is theoretically achievable for units having infinite heat exchange area). However, use of these flow configurations results in higher material temperatures. A parallel flow configuration (with equal flow rate times specific heat ratios on the air and combustion gas sides) has a theoretical

recovery effectiveness of 50%. Lower material temperatures result, however, due to the flow configuration. Figures 6.1, 6.2, and 6.3 schematically depict counterflow, crossflow, and parallel flow configurations and resulting fluid and material temperature profiles.

Metallic recuperator materials currently in use are limited to maximum operating temperatures of 1800°F (Reference 30). Presented in Table 6.1 are the operational limits for metallic recuperators dictated by material capabilities.

Ceramic recuperators have not experienced widespread application in U.S. industry. Carborundum (Reference 30) and Hague International (Reference 31) have produced units operating in a crossflow mode. Amsler-Morton (Reference 30) has produced units operating in a counterflow mode for use in steel reheat furnace applications.

The Carborundum unit consists of horizontal ceramic tubes spanning a flue gas duct. Preheat air passes through the tubes while the combustion gas passes across the exterior of the tubes. The tubes are sealed to ceramic headers using fibrous alumina-silicate packings. Application of the device has been limited due to air distribution and manifolding problems and due to leakage problems.

The Amsler-Morton unit consists of octagonal ceramic tubes oriented vertically with their ends located in fiber sealed recesses in ceramic tile. A typical unit consists of six or more vertical tube sets. Combustion gases flow down through the tubes while preheat air flows upward on the exterior of the tubes. Leakage in the unit has been found to be a problem. Air side leakages of 10 to 15 percent or more of the total flow result with air to gas side pressure differentials of 2 to 3 inches of water.

FIGURE 6.1. Counterflow Heat Exchanger Configuration

FIGURE 6.2. Crossflow Heat Exchanger Configuration

FIGURE 6.3. Parallel Flow Heat Exchanger Configuration

Table 6.1
Metallic Recuperator Operational Limits

Recuperator Flow Configuration	Maximum Combustion Gas Inlet Temperature (°F)	Theoretically Attainable Preheat Temperature ¹ (°F)	Probable Attainable Preheat Temperature ³ (°F)
Parallel Flow	>2600	1350 ²	975
Counterflow and Crossflow	1800	1800	1290

¹Based upon 100°F air inlet temperature, theoretical effectiveness of 50% for parallel flow and 100% for crossflow and counterflow.

²Based upon 2600°F combustion gas inlet temperature.

³Seventy percent of theoretical maximum effectiveness assumed.

The Hague International "CERHX" heat exchanger is a recently introduced crossflow unit in which finned silicon carbide tubes are oriented horizontally across a combustion gas flow passage. Preheat air is circulated through the tubes. Air inlet and exhaust plenums are positioned at the ends of the ceramic tubes. Due to a unique flexible seal between the tubes and the plenums, the manufacturer claims that air side leakages are as low as 2 percent of the total flow for air side pressures of 28 inches of water.

From the preceding, it is apparent that current commercially available alternatives for high temperature air preheat are checker regenerator systems and ceramic recuperators. The recuperator concept investigated in this study was compared to checker regenerators on a performance basis. Comparison with other ceramic recuperators was not possible due to a lack of specific design information. Comparison of metallic and ceramic recuperators introduces the economic considerations of different preheat temperatures and thus was beyond the scope of this investigation.

6.2 Analysis

The performance of the high temperature range recuperator was compared to that of a standard regenerator at two operating conditions. The conditions used in the evaluation represent a typical 150 ton/day glass furnace operating with either 2000°F or 1600°F preheated air. The systems were evaluated on the basis of required heat transfer area, physical size, pressure drop, material weight, and operating characteristics and limitations. Table 6.2 presents the design conditions used in the evaluation.

In sizing regenerators for the outlined conditions, the design procedure outlined in Trinks (Reference 1) was used. The following parameters were used in the design:

Table 6.2

Design Conditions - Preheat Systems Comparison

	<u>Case 1</u>	<u>Case 2</u>
Temperatures (°F)		
Air Inlet	100	100
Air Outlet (Preheat)	2,000	1,600
Combustion Gas Inlet	2,600	2,600
Combustion Gas Outlet	906	1,310
Mass Flow Rates (lbm/hr)*		
Air	32,000	36,500
Combustion Gas	33,600	38,325
Effectiveness	.76	.60

*Based upon 150 ton/day glass furnace

Brick size - $4\frac{1}{2}$ " X $2\frac{1}{2}$ " X 9"

Velocity - 1 ft/sec @ 60°F

Standard Checker Stacking

Cycling Time - 20 minutes

Velocity was limited to 1 ft/sec by leakage considerations (per Trinks, Reference 1). Presented in Figure 6.4 is the regenerator configuration. Using the above parameters, an overall coefficient of 1.5 BTU/hr·ft²·°F was calculated for the regenerator. The unit was sized based upon this overall coefficient. Presented in Table 6.3 are the results of this evaluation.

A high temperature range recuperator was also sized for the same service conditions. The following parameters were used in the analysis:

Module size - 30 inch diameter

Flow rate per stack - 100 lb/min

The module size and flow parameters were based upon economic optimization studies presented in Section 7.0. Recuperator sizing was per the information presented in Section 3.4 of the report. Results of the evaluation are presented in Table 6.3

6.3 Discussion

As shown in Table 6.3, the high temperature recuperator requires less heat exchange area than the regenerator for the same heat recovery duty. Corresponding unit volumes and material weights for the high temperature recuperator are less than those for the regenerator as a result. The high temperature recuperator provides this heat exchange area advantage because:

- 1) The recuperator can be operated at much higher velocities (with corresponding heat transfer enhancement) due to better leakage characteristics.

STANDARD CHECKER BRICK

REGENERATOR CONFIGURATION

FIGURE 6.4. Regenerator Configuration

Table 6.3
Preheat Systems Comparison

Case No.	Type	Heat Transfer Area (ft ²)	Overall Heat Coefficient (Btu/hrft ² °F)	Enclosed Volume of Unit ft ³	Material Weight (lbs)	Total Pressure Drop (in H ₂ O)	Flow Reynolds No.
1 (2000°F Preheat)	High Temperature Recuperator	3,300 ¹	6.9	3,034	60,000	25	≈60,000
	Regenerator	34,900	1.5	8,630	387,000	<0.1	≈ 900
2 (1600°F Preheat)	High Temperature Recuperator	1,408 ²	9.0	1,160	22,900	9	≈60,000
	Regenerator	19,030	1.5	4,070	182,000	<0.1	≈ 800

¹Requires 6 stacks of 165 modules each

²Requires 6 stacks of 63 modules each

Note: Regenerator numbers include required redundant unit

- 2) The geometry of the recuperator provides enhanced convective heat transfer.
- 3) A redundant regenerator unit is required due to the cyclic nature of its operation.

The higher pressure drops indicated in Table 6.3 for the high temperature recuperator are a result of the selected optimum operating point. As was previously discussed, the optimum operating point was based upon economic considerations.

The high temperature recuperator also provides the advantage of continuous, steady state operation. The transient nature of regenerator systems introduces additional design and operational concerns in furnace applications.

The high temperature recuperator cannot be classified as a compact heat exchange unit. Several recuperator columns would be required to service a furnace of the size considered in the study. As a result, manifolding, distribution, and valving problems could be expected in using the high temperature recuperator in large scale applications. Considerable plant area would be required for an installation due to access requirements for each stack. Crossflow-type heat exchangers discussed previously but not included in the performance evaluation would provide an advantage in this respect, and thus warrant further investigation.

7.0 ECONOMICS

The economics of the helical recuperator were investigated by sizing recuperators of different diameters (prototype, 15 and 30 inch diameter) for a range of combustion air preheat temperatures for a specific furnace application. The furnaces used in the analysis include 25, 150, 300 and 600 ton/day glass furnaces. Once the recuperator was sized for a specific application, cost was estimated. A detailed description of the sizing and cost estimating appear in Appendix 7. The result of the analysis is the determination of the payback period using a discounted cash flow approach for a specific situation. The payback period was determined by finding the time when the net present value equals zero. At this point, the stream of cash flows discounted to the present at the required rate of return equals the capital investment in the system. The net present value is calculated from the following expression.

$$NPV = \sum_{t=1}^n \frac{D_t(i_t)}{(1+r)^t} + I(c) + \sum_{t=1}^n \left[\frac{[S(f)^t - (B(e)^t + E)]}{(1+r)^t} (1-i_t) \right] - I$$

where:

B = initial blower energy cost (\$)

C = investment tax credit rate

= .033 n=3,4

.067 n=5,6

.100 n≥7

D_t = depreciation (sum of years digits method) in period t (\$)

$$= (I) \frac{2(n-t+1)}{n(n+1)}$$

E = annual operating and maintenance cost (\$), equal to 20 percent of the equipment cost (Reference 32)

e = price escalation of electricity per year = 1.10

f = price escalation of natural gas per year = 1.05

i_t = income tax rate in period t = .48

I = capital investment (\$)

NPV = net present value (\$)

n = life cycle (years)

r = required rate of return per year = .15

S = initial annual energy savings (\$)

t = period from 1 to n (years)

The payback period was interpolated from the above equation at interval years to find when the NPV was zero.

The major assumptions in the analysis are the following.

- 1) The installed cost of the recuperator is 3.3 times the cost of the modules, transition pieces, insulation, and blowers (Reference 33). In other words, the process equipment cost typically represents 30 percent of the total installed cost of battery-limit (plant subsystem) installations.
- 2) The process temperature of the furnace is 2600°F, and the exhaust gas recuperator inlet temperature is 2400°F.
- 3) The furnace is operated at close to atmospheric pressure. This requires blowers on both the air and exhaust gas side of the recuperator (Section 3.4).
- 4) The thermal performance of the recuperator is evaluated by use of the thermal model, described in Section 3.2.

- 5) The electricity cost for the blower is based on 3¢ per kwh (Reference 34).
- 6) The cost of natural gas is \$2.00 per million Btu (Reference 34).¹
- 7) Recuperator module costs were based upon the use of KT-Silicon Carbide material having a cost of \$24/lb and a density of 200 lbm/ft³ (Reference 3). The \$24/lb material cost provides a 14 percent contingency (corrected for density variation) over 1978 Coors estimates for large scale production costs of AD998 Alumina modules (Reference 36) and a 17 percent contingency over 1980 Carborundum estimates for large scale production costs of KT-Silicon Carbide modules (Reference 37).
- 8) The volume of the larger diameter modules was scaled-up assuming that the wall and interface thickness remained the same as the prototype diameter module.

For a given module diameter, furnace size, and preheat temperature, the mass flow rate through an individual stack was varied. The flow rate was varied by assuming different numbers of stacks per installation. Varying the flow rate varies the equipment cost. The trend of cost versus flow rate is shown in Figure 7.1. So, for a given module diameter, furnace size, and preheat temperature, the key variable to the installed cost is the amount of air and combustion gas that is pushed through an individual stack. At high mass flow rates the blower size dominates the total cost. At low mass flow rates, the number of stacks dominates the cost. The optimum operating point for the prototype module is approximately 10 lbm/min mass flow through an

¹ \$2.00 per million BTU is the average price paid by pipeline industrial natural gas users in the U.S. as of February, 1980. Payback periods presented herein do not apply to localized areas of the U.S. in which natural gas prices vary significantly from the average.

FIGURE 7.1. Equipment Costs Versus Mass Flow Rate Per Stack - 15 Inch Diameter Modules

FIGURE 7.2. Payback Period Versus Flow Rate for Various Preheat Temperatures - 15 Inch Diameter Modules

Table 7.1
Economic Results

Furnace Size (ton/day)	Payback Period (Years)					
	Prototype		15" Module		30" Module	
	Preheat (F)		Preheat (F)		Preheat (F)	
	1000	1700	1000	1700	1000	1700
25	4.5	11.4	4.2	12.9	3.8	—
150	4.8	15.9	3.9	11.8	3.2	11.4
300	4.6	13.8	4.0	10.7	3.1	12.0
600	4.8	15.1	4.0	12.4	3.0	11.8

THE FOLLOWING COMMENTS INSERTED BY DOE:

The payback periods listed in Table 7.1 are sensitive to the assumptions made in performing the analysis. Each specific application would result in a different set of starting assumptions. Examples are the variation in fuel prices from one region of the U.S. to another, variations in the installed cost multiplier from one industry to another (and from plant to plant within the same industry), and in the assumed fuel price escalation rate. In Terra Tek's analysis the natural gas price of \$2.00 per million Btu is below the national average industrial price of \$2.51 per million Btu (Federal Register, Vol. 45, No. 16 dated 1/23/80). The fuel price escalation rate of 5% is also considered low with 15% being more commonly used.

Using an installed cost multiplier of 2.5, a natural gas price of \$2.51 per million Btu, and a fuel price escalation rate of 15% the payback periods would be reduced by 54% or would be approximately one-half those in Table 7.1.

individual stack. The optimum operating point for the 15 and 30 inch diameter modules is approximately 50 and 150 lbm/min, respectively.

For a specific furnace size and module diameter, the payback period is a function of the preheat temperature. The recuperator designed efficiency is an important economic variable for all furnace sizes and module diameters. The shortest payback period always occurred at the lowest preheat temperature. The lowest preheat temperature used in the analysis was 1000°F; the corresponding recuperator efficiency was approximately 40 percent. The highest preheat temperature (1700°F) produced the longest payback period. Figure 7.2 shows the effect of preheat temperature on the economics.

The results of the economic analysis are shown in Table 7.1. The payback period is not a function of furnace size for the assumptions used in the analysis. The analysis does not, however, take into account the problem of process control for a large number of stacks. A larger diameter (30 inch) recuperator is more economic than the prototype diameter (8 inches). For a 600 ton/day furnace and 1000°F preheat the payback periods are 4.8, 4.0 and 3.0 years for the prototype (8 inch), 15 inch and 30 inch diameter recuperator. The shortest payback period is approximately 3.0 years corresponding to the 30 inch module and 1000°F preheat. For the prototype diameter (8 inch), the shortest payback period is approximately 4.6 years for the 1000°F preheat. For the 30 inch diameter recuperator, the predictions are less certain than for the prototype due to the assumptions in scale-up. The major scale-up assumption is that the wall and interface thickness for the larger diameter module is the same as the prototype. Fabricability and structural considerations could dictate greater thicknesses for larger diameter modules; however, this has not been taken into consideration.

Other cost factors cannot be assessed without operational experience. Among these are:

- 1) Cost of separating device.
- 2) Downtime and cost associated with cleaning the recuperator to keep the system operational.
- 3) Replacement costs during the recuperator's lifetime.
- 4) Cost of manifolding and furnace modifications.
- 5) Cost and performance of the gasketing scheme.
- 6) Cost due to the complexity of the flow rate control for a large number of stacks.
- 7) The space required for installation.

Even without these added cost considerations, it is obvious that the helical recuperator is not economical for high temperature preheat systems. At a preheat temperature of 1700°F, the payback period is greater than 10 years. A payback period of approximately 3 years is realized for systems designed for 1000°F preheat using the 30 inch diameter recuperator. For the low preheat temperature service, the prudence of using a ceramic recuperator capable of withstanding high temperatures is questionable. Metallic recuperator or checker systems of proven designs which can provide the low preheat temperatures will be more desirable. For example, a metallic recuperator for a 3.5 ton/day glass furnace has been reported to produce a 1.5 year payback period (Reference 35). This is compared to a payback period of 3 and 4.6 years for the helical recuperator of prototype (8 inch diameter) and 30 inch diameter, respectively.

8.0 APPLICATIONS STUDIES

Investigation of possible industrial applications for the high temperature range recuperator was conducted by Millcreek Glass Corporation. Millcreek Glass also assessed of the problems associated with joining, sealing, cleaning, and separating recuperator modules used in a glass furnace environment. Reports summarizing work conducted by Millcreek Glass are included as Appendices 8 and 9.

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APPENDICES

HIGH TEMPERATURE RANGE RECUPERATOR

Phase II Final Report

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APPENDIX 1
DETAILED THERMAL MODEL

HIGH TEMPERATURE RANGE RECUPERATOR THERMAL MODEL

INTRODUCTION

The thermal model simulates the steady-state performance of a helix-shaped recuperator. The major application of the recuperator is to preheat combustion air utilizing the heat of the exhaust gas. Given the input variables of system geometry, material properties, air and exhaust gas flow rates, and air and gas inlet temperatures, the model predicts fluid and material temperature distributions and thus the recuperator efficiency.

The model consists of a system of heat balance equations derived between finite control volumes within the recuperator. Since radiation heat exchange is incorporated into the analysis, the initial form of the system of equations is non-linear. The radiation equations are linearized which involves knowledge of the temperature distribution. So, the temperature distribution is initially guessed and the system of equations solved giving a new temperature distribution. The process is repeated until it converges. Convergence is attained when on two consecutive equation solutions the temperature distribution changes less than the inputted convergence criteria which is on the order of .1°F.

CONTROL VOLUMES

The control volumes within the recuperator are taken to be a half turn of the helix. (See Figures A1.1 and A1.2). Within a control volume four zones are defined; the temperature within a zone is assumed constant or lumped. The four zones are the gas, wall, upper interface, and lower interface. (Figure A1.3) The wall and interfaces are the boundaries of the gas within a control volume.

Figure A1.1. Recuperator Schematic.

$(h)_i$ = hot exhaust gas control volume

$(c)_i$ = cold air control volume.

Figure A1.2. Control Volume Schematic

FRONT VIEW

SIDE VIEW

BOTTOM VIEW

Figure A1.3. Control Volume Zones

GOVERNING EQUATIONS

Within a control volume, heat balance equations relate the four zones and the environment. Conduction equations relate the hot exhaust gas control volumes to the cold air control volumes. The heat given up by the hot exhaust gas in a control volume is related to the temperature drop into the next hot gas control volume. The heat absorbed by the cold air is related to the temperature increase into the next cold air control volume.

The conduction equation between a hot exhaust gas and a cold air control volume is,

$$q_k = K (T_{i,h} - T_{i,c})$$

$$K = \frac{k_m A_i}{\Delta x_m}$$

Where:

q_k = conduction heat transfer (Btu/hr)

k_m = recuperator material thermal conductivity (Btu/hr·ft·F)

A_i = interface area (ft²)

Δx_m = thickness of recuperator between control volumes (ft)

$T_{i,h}$ = hot gas interface temperature (F)

$T_{i,c}$ = cold gas interface temperature (F)

The convection heat transfer equations are based on the determination of a convection heat transfer coefficient. The relationship used for this coefficient was experimentally determined for the prototype recuperator

during warm flow testing. The convection coefficient is a function of the fluid velocity and the fluid physical properties. The fluid properties are evaluated at the temperature of the air or gas in a control volume. It is assumed that the fluid properties of the exhaust gas are those of air. The convection equations are,

$$q_w = h_w A_w (T - T_w)$$

$$q_i = h_i A_i (T - T_i)$$

Where:

q_w = convection heat transfer between gas and wall (Btu/hr)

q_i = convection heat transfer between gas and interface
(Btu/hr)

h = convection heat transfer coefficient (Btu/hr·ft²·F)

A = area (ft²)

T = gas or air temperature (F)

T_w = wall temperature (F)

T_i = interface temperature (F)

Subscripts

w: wall

i: interface

The radiation heat transfer equations between the air or gas and the control volume boundaries are derived with the assumption that the boundaries are black. This will not introduce serious error if the emissivity of the surface is greater than .9. (Reference 1). This assumption greatly

simplifies the model allowing the radiation transfer to be easily linearized. The model also assumes that there is no radiation transfer between control volumes. This is based on the fact that the shape factors between control volumes are small compared to the shape factors between zones in a control volume.

For the cold air section the air is transparent to radiation. The net heat transfer equations are:

$$q_{r,i-i} = A_i F_{i-i} \sigma (\bar{T}_{i,1}^4 - \bar{T}_{i,2}^4)$$

$$q_{r,i-w} = A_i F_{i-w} \sigma (\bar{T}_i^4 - \bar{T}_w^4)$$

In linear form,

$$q_{r,i-i} = R (\bar{T}_{i,1} - \bar{T}_{i,2})$$

$$q_{r,i-w} = S (\bar{T}_i - \bar{T}_w)$$

$$R = A_i F_{i-i} \sigma (\bar{T}_{i,1} + \bar{T}_{i,2}) (\bar{T}_{i,1}^2 + \bar{T}_{i,2}^2)$$

$$S = A_i F_{i-w} \sigma (\bar{T}_i + \bar{T}_w) (\bar{T}_i^2 + \bar{T}_w^2)$$

Where:

$q_{r,i-i}$ = net radiation transfer between the two interfaces
(Btu/hr)

$q_{r,i-w}$ = net radiation transfer between interface and the wall
(Btu/hr)

A = area (ft²)

F = shape factor

σ = Stefan-Boltzman constant (Btu/hr·ft²·R⁴)

$\bar{T}$ = temperature (R)

R = radiation heat transfer coefficient between interfaces (Btu/hr·R)

S = radiation heat transfer coefficient between an interface and wall (Btu/hr·R)

Subscripts i: interface

w: wall

The radiation heat transfer for the hot exhaust gas control volume takes into account the fact that the gas is not transparent. The gas is assumed to be gray and the emissivity a function of temperature, pressure, and effective beam length. The beam length is assumed the same for each zone and equal to 3.4 times the volume divided by the surface area. The governing equations for the net heat transfer between the boundaries are:

$$q_{r,i-i} = A_i F_{i-i} \sigma ((1-\epsilon_{i,1}) \bar{T}_{i,1}^4 - (1-\epsilon_{i,2}) \bar{T}_{i,2}^4)$$

$$q_{r,i-w} = A_i F_{i-w} \sigma ((1-\epsilon_i) \bar{T}_i^4 - (1-\epsilon_w) \bar{T}_w^4)$$

In linear form:

$$q_{r,i-i} = R ((1-\epsilon_{i,1})^{\frac{1}{4}} \bar{T}_{i,1} - (1-\epsilon_{i,2})^{\frac{1}{4}} \bar{T}_{i,2})$$

$$q_{r,i-w} = S ((1-\epsilon_i)^{\frac{1}{4}} \bar{T}_i - (1-\epsilon_w)^{\frac{1}{4}} \bar{T}_w)$$

$$R = A_i F_{i-i} \sigma ((1-\epsilon_{i,1})^{\frac{1}{4}} \bar{T}_{i,1} + (1-\epsilon_{i,2})^{\frac{1}{4}} \bar{T}_{i,2}) \\ ((1-\epsilon_{i,1})^{\frac{1}{2}} \bar{T}_{i,1}^2 + (1-\epsilon_{i,2})^{\frac{1}{2}} \bar{T}_{i,2}^2)$$

$$S = A_i F_{i-w} ((1-\epsilon_i)^{\frac{1}{4}} \bar{T}_i + (1-\epsilon_w)^{\frac{1}{4}} \bar{T}_w) \\ ((1-\epsilon_i)^{\frac{1}{2}} \bar{T}_i^2 + (1-\epsilon_w)^{\frac{1}{2}} \bar{T}_w^2)$$

Where:

$q_{r,i-i}$ = net radiation transfer between the two interfaces (Btu/hr)

$q_{r,i-w}$ = net radiation transfer between an interface and the wall (Btu/hr)

A = area

F = shape factor

σ = Stefan-Boltzman constant (Btu/hr·ft²·R⁴)

$\bar{T}$ = temperature (R)

ϵ_x = gas emissivity evaluated at temperature of zone x

R = radiation heat transfer coefficient between the two interfaces (Btu/hr·R)

S = radiation heat transfer coefficient between an interface and the wall (Btu/hr·R)

Subscripts

i : interface

w : wall

The net radiation heat transfer between the hot gas and the control volume boundaries are:

$$q_{h-i} = A_i \sigma (\epsilon_h \bar{T}_h^4 - \epsilon_i \bar{T}_i^4)$$

$$q_{h-w} = A_w \sigma (\epsilon_h \bar{T}_h^4 - \epsilon_w \bar{T}_w^4)$$

In linear form:

$$q_{h-i} = G_i (\epsilon_h^{1/4} \bar{T}_h - \epsilon_i^{1/4} \bar{T}_i)$$

$$q_{h-w} = G_w (\epsilon_h^{1/4} \bar{T}_h - \epsilon_w^{1/4} \bar{T}_w)$$

$$G_i = A_i \sigma (\epsilon_h^{1/4} \bar{T}_h + \epsilon_i^{1/4} \bar{T}_i) (\epsilon_h^{1/2} \bar{T}_h^2 + \epsilon_i^{1/2} \bar{T}_i^2)$$

$$G_w = A_w \sigma (\epsilon_h^{1/4} \bar{T}_h + \epsilon_i^{1/4} \bar{T}_i) (\epsilon_h^{1/2} \bar{T}_h^2 + \epsilon_i^{1/2} \bar{T}_i^2)$$

Where:

q_{h-i} = net radiation transfer between exhaust gas and an interface (Btu/hr)

q_{h-w} = net radiation transfer between exhaust gas and wall (Btu/hr)

A = area (ft²)

σ = Stefan-Boltzman constant (btu/hr·ft²·R⁴)

ϵ_x = gas emissivity evaluated at the temperature of zone x

$\bar{T}$ = temperature (R)

G_i = radiation heat transfer coefficient between gas and an interface (Btu/hr·R)

G_w = radiation heat transfer coefficient between gas and the wall (Btu/hr·R)

Subscripts

i: interface

w: wall

h: exhaust gas

Each control volume also loses heat to the environment through the recuperator walls and the outside insulation. The heat loss equation in linear form is:

$$q_l = [1/(H_{in} + I_{in}) + 1/K_{in}]^{-1}(\bar{T}_w - \bar{T}_\infty)$$

$$K_{in} = \pi k_{in} \Delta x / \ln(r_o/r_i)$$

$$H_{in} = \pi r_o h_{in} \Delta x_{in}$$

$$I_{in} = A_{in} \epsilon_{in} \sigma (\bar{T}_{in,0}^2 + \bar{T}_\infty^2) (\bar{T}_{in,0} + \bar{T}_\infty)$$

Where:

k_{in} = insulation conductivity (Btu/hr·ft·F)

Δx = height of control volume, $\frac{1}{2}$ the height per turn of helix (ft)

r_o = outside insulation radius (ft)

r_i = inside insulation radius (ft)

h_{in} = convection heat transfer coefficient from outside insulation (Btu/hr·ft²·F)

A_{in} = outside area of insulation per control volume (ft²)

ϵ_{in} = emissivity of outside of insulation

T = temperature (F)

$\bar{T}$ = temperature (R)

Subscripts

in: insulation

in,o: outside insulation

w: wall

∞ : environment

The equations for the heat absorbed in a cold air control volume and the heat lost in a hot exhaust gas control volume are:

$$q_{c,n} = (\dot{m} c_p)_{c,n} (T_{c,n-1} - T_{c,n})$$

$$q_{h,n} = (\dot{m} c_p)_{h,n} (T_{h,n+1} - T_{h,n})$$

Where:

$q_{c,n}$ = heat absorbed by cold air (Btu/hr)

$q_{h,n}$ = heat lost by hot gas (Btu/hr)

$\dot{m}$ = mass flow rate (Lb/hr)

c_p = heat capacity (Btu/Lb·F)

T = temperature (F)

Subscripts c: cold air

h: hot gas

NUMERICAL TECHNIQUE

For each control volume there are five variables, the gas or air temperature, wall temperature, lower interface temperature, upper interface temperature, and outside insulation temperature. Since the equations relating these zones or nodes are linear in form, the number of variables can be reduced. In the final form there are two variables per control volume, the variables being the air or gas temperature and the lower interface temperature. For every variable an equation can be written about its node. So, a system of linear equations is generated. The constants in the nodal equations are a function of the temperature distribution. These constants are a function of the convection and radiation heat transfer coefficients.

The scheme is to guess the nodal equation constants, solve the system of linear equations, find the value of the eliminated temperatures, and calculate the nodal equation constants. The process is repeated until it converges. Convergence is attained when the temperatures calculated from the system of equations in two consecutive iterations varies less than the inputted convergence criteria.

The matrix generated is sparse and can be arranged into a diagonal form. A Gauss-Siedel iteration scheme is used to solve the system of linear equations. Gauss-Siedel converges since the matrix is diagonally dominant.

USER'S MANUAL AND PROGRAM LISTING

The computer program consists of three sections, the main program and two subroutines. The main program receives the input, packs the matrix by calling the subroutines, solves the matrix, iterates to the solution and prints the desired output. The program uses Gauss-Siedel iteration to solve the matrix. The output can be only the exhaust gas and air outlet temperatures, the entire temperature distribution, or the matrix. The subroutine 'hot' calculates the nodal equation constants in the exhaust gas control volumes. The subroutine 'cold' calculates the nodal equation constants in the cold air control volumes.

The input variables are,

NUM = number of gas or air control volumes, must be odd number, number of full turns of helix = $(\text{num}-1)/2$, is an integer

TINH = inlet temperature of exhaust gas (F)

TINC = inlet temperature of cold air (F)

TINF = ambient temperature (F)

FLOH = mass flow rate of exhaust gas (lbm/sec)

FLOC = mass flow rate of cold air (lbm/sec)

DIAMH = hydraulic diameter of recuperator (ft)

AINH = interface area in control volume (ft^2)

AWALL = wall area in control volume (ft^2)

AFLOW = flow area cross-section (ft^2)

FINT = interface area times radiation shape factor between interfaces (ft^2)

FWALL = interface area times radiation shape factor between an interface and wall (ft^2)

HPT = height per turn of helix (ft)
 THICK = recuperator material thickness between control volumes (ft)
 COND = recuperator material thermal conductivity (Btu/hr.ft.F)
 CIN = insulation thermal conductivity (Btu/hr.ft.F)
 RI = inside insulation radius (ft)
 RO = outside insulation radius (ft)
 EMIN = emissivity of outside surface of insulation
 APL = mean path length for radiation within control volume (ft)
 H2OP = partial pressure of water vapor in exhaust gas (ATM)
 CO2P = partial pressure of carbon dioxide in exhaust gas (ATM)
 CONVG = convergence criteria for total solution (F)
 CONIT = convergence criteria for Gauss-Siedel iteration (F)
 KEND = maximum number of iterations for total solution, integer
 MAXIT = maximum number of iterations for Gauss-Siedel, integer
 NOT = print logic, integer
 = 2, prints matrix, and each Gauss-Siedel iteration [temperatures (R) and heat transfer in exit gas and air control volume (Btu/hr)]
 = 1, prints each Gauss-Siedel iteration [temperatures (R) and heat transfer in exit and air control volumes (Btu/hr)]
 = 0, prints only last iteration [temperatures (R) and heat transfer in exit and air control volume (Btu/hr)]
 NPR = print logic
 = 0, prints only exit temperatures (F)
 $\neq$ 0, prints exit temperatures (F), temperature distribution (R), and heat transfer in exit gas and air control volumes (Btu/hr)

The curve fits for the fluid physical properties and gas emissivity bound some of the input variables. The input temperatures must be greater than 32°F and less than 3000°F. The product of the mean path length and the partial pressure must be between .005 and .20 ft-atm for water vapor and between .006 and .10 ft-atm for carbon dioxide.

In most cases the output wanted is just the final exit temperatures. For this case 'NOT' should equal 0. If the temperature distribution is also needed 'NPR' should not equal 0. The cases of 'NOT' equal to 1 or 2 are used for debugging the matrix packing and Gauss-Siedel iteration.

If the temperature distribution is printed, the program returns $T(J)$, $J=1$, to $LENG$, where $LENG$ equals $4 \times NUM - 2$. $T(1)$ equals the heat absorbed by the cold air in its exit control volume (Btu/hr). $T(LENG)$ equals the heat lost by the exhaust gas in its exit control volume (Btu/hr). The rest of the T 's printed are temperatures (R). $T(2)$ is the temperature in the bottom cold air control volume defined as $C1$. (See Figure A1.4). $T(3)$ is the lower interface temperature and $T(4)$ the exhaust gas temperature in gas control volume 2. The scheme is repeated up to the lower interface temperature in the cold inlet control volume, this being $T(LENG/2)$. $T(LENG/2+1)$ is the lower interface temperature of the cold control volume 2. Again, the scheme is repeated to $T(LENG-1)$, which is the gas temperature in control volume 'NUM'.

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If the temperature distribution is printed, the program returns $T(J)$, $J=1$, to $LENG$, where $LENG$ equals $4 \times NUM - 2$. $T(1)$ equals the heat absorbed by the cold air in its exit control volume (Btu/nr). $T(LENG)$ equals the heat lost by the exhaust gas in its exit control volume (Btu/hr). The rest of the T 's printed are temperatures (R). $T(2)$ is the temperature in the bottom cold air control volume defined as C1. (See Figure A1.4). $T(3)$ is the lower interface temperature and $T(4)$ the exhaust gas temperature in gas control volume 2. The scheme is repeated up to the lower interface temperature in the cold inlet control volume, this being $T(LENG/2)$. $T(LENG/2+1)$ is the lower interface temperature of the cold control volume 2. Again, the scheme is repeated to $T(LENG-1)$, which is the gas temperature in control volume 'NUM'.

CONTROL VOLUMES

DESCRIPTION OF OUTPUT, "NPR" $\neq$ 0

LENG = $4 \times \text{NUM}-2$

$\dot{T}(1)$ = Heat absorbed in cold outlet C1 (Btu/hr)

T(2) = Air temperature in C1 (R)

T(3) = Lower interface temperature in H2 (R)

T(4) = Gas temperature in H2 (R)

T(LENG/2) = Lower interface temperature in C NUM (R)

T(LENG/2+1) = Lower interface temperature in H1 (R)

T(LENG-2) = Lower interface temperature in C NUM-1 (R)

T(LENG-1) = Gas temperature in H NUM (R)

T(LENG) = Heat lost in H NUM (Btu/hr)

FIGURE A1.4: Temperature Description

DATA
INPUT

ORDER

FORMAT

1	NUM, TINH, TINC, TINF	1I5, 310.0
2	FLOH, FLOC, DIAMH, AINT, AWALL	510.0
3	AFLOW, FINT, FWALL, HPT, THICK	510.0
4	COND, CIN, RI, RO, EMIN	510.0
5	APL, HZOP, COZP, CONVG, CONIT	510.0
6	KEND, MAXIT, NOT, NPR	4I5

DATA, IL TPFs.

DATA USR1 S73MOD 05/07/80 15:15:32 (0)

```
1. DIMENSION A(300),B(300),C(300),D(300),E(300)
2. DIMENSION Z(300),DEN(300),T(300),TOLD(300),TCK(300)
3. DIMENSION REM(300)
4. C INPUT VARIABLES
5. READ(5,10)NUM,TINH,TINC,TINF
6. 10 FORMAT(15,3F10.0)
7. READ(5,20)FLOH,FLOC,DIAMH,AINT,AWALL
8. 20 FORMAT(5F10.0)
9. READ(5,20)AFLOW,FINT,FWALL,HPT,THICK
10. READ(5,20)COND,CIN,RI,RO,EMIN
11. READ(5,20)APL,H2OP,CO2P,CONVG,CONIT
12. READ(5,10) KEND,MAXIT,NOT,NPR
13. 16 FORMAT(4I5)
14. LENG=4*NUM-2
15. LHALF=2*NUM-1
16. KCOUNT=1
17. ANUM=NUM
18. RLEN=HPT*(ANUM-1.0)/2.0
19. AINS=3.1416*RO*HPT/2.0
20. JGUESS=1
21. nINS=.5*AINS
22. NSECT=(NUM-1)/2
23. WRITE(6,21)NSECT
24. 21 FORMAT(1X,' NO. OF FULL SECTIONS=',I5)
25. WRITE(6,22)TINH,TINC,TINF
26. 22 FORMAT(1X,' TEMPERATURES (OF)'/20X,' HOT INLET= ',
27. 1F8.3/20X,' COLD INLET= ',F8.3/20X,' AMB TEMP= ',
28. 2F8.3//)
29. WRITE(6,23) FLOH,FLOC,DIAMH,AINT,AWALL
30. 23 FORMAT(1X,' SYSTEM PARAMETERS ARE '/20X,' HOT GAS FLOW RATE (LB
31. 1/SEC)= ',F8.5/20X,' COLD AIR FLOW RATE (LBM/SEC)= ',F8.5/20X,
32. 2' HYDRAULIC DIAMETER (FT)= ',F8.5/20X,' INTERFACE AREA (FT2)= ',
33. 3F8.5/20X,' WALL AREA (FT2)= ',F8.5//)
34. WRITE(6,24)AFLOW,FINT,FWALL,HPT,THICK
35. 24 FORMAT(/20X,' FLOW AREA (FT2)= ',F8.5/20X,
36. 1' INTERFACE SHAPE FACTOR*AREA (FT2)= ',F8.5/20X,
37. 3' WALL SHAPE FACTOR*AREA (FT2)= ',F8.5/20X,
38. 4' HEIGHT PER TURN OF HELIX (FT)= ',F8.5/20X,
39. 5' RECUPERATOR THICKNESS (FT)= ',F8.5//)
40. WRITE(6,25)COND,CIN,RI,RO,EMIN
41. 25 FORMAT(/20X,' RECUP CONDUCTIVITY (BTU/HR*FT*F)= ',F8.5/20X,
42. 1' INSULATION CONDUCTIVITY (BTU/HR*FT*F)= ',F8.5/20X,
43. 2' INSIDE INSULATION RADIUS (FT)= ',F8.5/20X,
44. 3' OUTSIDE INSULATION RADIUS (FT)= ',F8.5/20X,
45. 4' INSULATION EMISSIVITY= ',F8.5//)
46. WRITE(6,26)APL,H2OP,CO2P,CONVG,CONIT
47. 26 FORMAT(/20X,' MEAN PATH LENGTH (FT)= ',F8.5/20X,
48. 1' WATER VAPOR PART PRESSURE (ATM)= ',F8.5/20X,
49. 2' CARBON DIOXIDE PART PRESSURE (ATM)= ',F8.5/20X,
50. 3' FINAL SOLUTION CONVERGENCE CRITERIA= ',F8.5/20X,
51. 4' G-S CONVERGENCE CRITERIA= ',F8.5//)
52. WRITE(6,27) KEND,MAXIT
53. 27 FORMAT(/20X,' MAX SOLUTION ITERATION= ',I5/20X,
54. 2' MAX G-S ITERATION= ',I5//)
55. APLH=APL*H2OP
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56.      AP=APL*CU2P
57.      AK=COND*AIN/THICK
58.      CINS=3.1416*CIN*HPT/(ALOG(R0/R1)*2)
59.      AEIN=AIN*EMIN
60.      TINTC=200.0
61.      TINBC=200.0
62.      TINBH=200.0
63.      TINTH=200.0
64.      C ALL TEMPS RANKINE EXCEPT FOR OUTSIDE INSULATION CONVECTION COEFF
65.      TINH=TINH+460.0
66.      TINC=TINC+460.0
67.      TINFDF=TINF
68.      TINF=TINF+460.0
69.      TB=(TINH+TINC)/2.0
70.      L=LENG+1
71.      C INITIALLY SET TEMPERATURES
72.      DO 40 J=1,LENG
73.      T(J)=TB
74.      TOLD(J)=TB
75.      TCK(J)=TB
76.      40 CONTINUE
77.      IF(JGUESS.EQ.1) GO TO 35
78.      30 CONTINUE
79.      C FIND CONVECTION COEFF OFF INSULATION
80.      TINAVE=(TINTH+TINTC+TINBH+TINBC)/4.0
81.      IF(TINAVE.LT.2600.0) GO TO 71
82.      HINSE=.5*AINS
83.      GO TO 35
84.      71 CONTINUE
85.      TFILM=(TINAVE+TINFDF)/2.0
86.      GR=14.6951-5.466E-3*TFILM+.9962E-6*TFILM**2
87.      GRR=EXP(GR)*ABS(TINAVE-TINFDF)*RLEN**3
88.      HINS=.61293+2.2068E-5*TFILM-2.746E-9*TFILM**2
89.      IF(GRR.GT.1.0E9) GO TO 32
90.      GR=EXP(.25*GR)*(ABS(TINAVE-TINFDF)*RLEN**3)**.25
91.      HINS=.555*HINS*.921*GR*AINS/RLEN
92.      GO TO 35
93.      32 CONTINUE
94.      GR=EXP(.333*GR)*(ABS(TINAVE-TINFDF))**.333
95.      HINS=.13*HINS*.896*GR*AINS
96.      35 CONTINUE
97.      JINT=0
98.      C SECTION TO PACK MATRIX
99.      DUM=1.0
100.     L=LENG+1
101.     C SECTION TO PACK UPPER MATRIX
102.     CALL COLD(FLOW,FINT,FWALL,AK,AWALL,AIN,HINS,CINS,
103.     1,AEIN,TINF,DIAM,CPM,CC1,CC2,CC3,ALC1,ALC2,ALC3,D1,D2,D3,D4,E1,
104.     2,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,A11,ENC1,ENC2,
105.     3,ENC3,EIN1,EIN2,EIN3,TINTC,TINBC,JGUESS,
106.     4 T(2),DUM,T(3),2,1)
107.     CPC=CPM
108.     A(1)=1.0
109.     B(1)=ALC1-CC1
110.     C(1)=ALC2-CC2
111.     D(1)=(CC3-ALC3)*TINF
112.     A(2)=C(1)

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113.      CALL HOT(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,AP,
114.      1AEIN,TINF,DIAMH,APLH,CPM,CH1,CH2,CH3,ALH1,ALH2,ALH3,D1,D2,D3,
115.      2D4,E1,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,AI1,AI2,
116.      3ENH1,ENH2,ENH3,EOUT1,EOUT2,EOUT3,TINBH,TINTH,JGUESS,
117.      8 T(4),T(3),T(5),4,2)
118.      B(2)=C(2)+D1-AI1-H1
119.      C(2)=D2-H2-AI2
120.      D(2)=D3-H3
121.      Z(2)=- (CC3+D4-H4)*TINF
122.      LE=LHALF-1
123.      DO 100 J=4,LE,4
124.      IF(J.EQ.4) GO TO 105
125.      CALL HOT(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,AP,
126.      1AEIN,TINF,DIAMH,APLH,CPM,CH1,CH2,CH3,ALH1,ALH2,ALH3,D1,D2,D3,
127.      2D4,E1,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,AI1,AI2,
128.      3ENH1,ENH2,ENH3,EOUT1,EOUT2,EOUT3,TINBH,TINTH,JGUESS,
129.      8T(J),T(J-1),T(J+1),J,2)
130.      C(J-2)=D1-AI1-H1
131.      D(J-2)=D2-H2-AI2
132.      E(J-2)=D3-H3
133.      Z(J-2)=- (D4-H4)*TINF
134.      105 CONTINUE
135.      A(J-1)=G1+AI1+F1
136.      B(J-1)=-CPM+G2+AI2+F2
137.      C(J-1)=G3+F3
138.      Z(J-1)=- (G4+F4)*TINF
139.      DEN(J-1)=CPM
140.      A(J)=D1+F1+E1
141.      B(J)=D2+F2+E2
142.      C(J)=D3+E3+F3
143.      Z(J)=- (D4+E4+F4)*TINF
144.      100 CONTINUE
145.      LE=LHALF-3
146.      DO 200 J=6,LE,4
147.      CALL COLD(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,
148.      1AEIN,TINF,DIAMH,CPM,CC1,CC2,CC3,ALC1,ALC2,ALC3,D1,D2,D3,D4,E1,
149.      2E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,HC1,HC2,HC3,HC4,AI1,ENC1,ENC2,
150.      3ENC3,EIN1,EIN2,EIN3,TINTC,TINBC,JGUESS,
151.      8T(J),T(J-1),T(J+1),J,2)
152.      C(J-2)=C(J-2)-D1-HC1-AI1
153.      D(J-2)=-D2-HC2+AI1
154.      E(J-2)=-D3-HC3
155.      Z(J-2)=Z(J-2)- (-D4-HC4)*TINF
156.      A(J-1)=AI1+F1+G1
157.      B(J-1)=CPM-AI1+F2+G2
158.      C(J-1)=F3+G3
159.      DEN(J-1)=-CPM
160.      Z(J-1)=- (F4+G4)*TINF
161.      A(J)=-L1+F1+E1
162.      B(J)=-L2+F2+E2
163.      C(J)=C(J)-D3+E3+F3
164.      Z(J)=Z(J)- (-L4+E4+F4)*TINF
165.      200 CONTINUE
166.      J=LENG+1
167.      CALL COLD(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,
168.      1AEIN,TINF,DIAMH,CPM,CC1,CC2,CC3,ALC1,ALC2,ALC3,D1,D2,D3,D4,E1,
169.      2E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,HC1,HC2,HC3,HC4,AI1,ENC1,ENC2,

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170.      3ENC3,EIN1,EIN2,EIN3,TINTC,TINBC,JGUESS,
171.      8TINC,T(LHALF),DUM,J,3)
172.      J=LHALF+1
173.      C(J-2)=C(J-2)-ENC2
174.      Z(J-2)=Z(J-2)+ENC3*TINF+ENC1*TINC
175.      A(J-1)=EIN2
176.      DEN(J-1)=-CPM
177.      Z(J-1)=- (EIN1+CPM)*TINC-EIN3*TINF
178. C SECTION TO PACK LOWER MATRIX
179.      J=LHALF+1
180.      CALL HOT(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,AP,
181.      1AEIN,TINF,DIAMH,APLH,CPM,CH1,CH2,CH3,ALH1,ALH2,ALH3,D1,D2,D3,
182.      2D4,E1,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,AI1,AI2,
183.      3ENH1,ENH2,ENH3,EOUT1,EOUT2,EOUT3,TINBH,TINTH,JGUESS,
184.      8TINH,DUM,T(J),0,1)
185.      A(J)=CH2+ALH2
186.      DEN(J)=CPM
187.      Z(J)=- (ALH1+CH1-CPM)*TINH- (ALH3+CH3)*TINF
188.      LB=LHALF+2
189.      LE=LENG-3
190.      DO 300 J=LB,LE,4
191.      CALL CULU(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,
192.      1AEIN,TINF,DIAMH,CPM,CC1,CC2,CC3,ALC1,ALC2,ALC3,D1,D2,D3,D4,E1,
193.      2E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,HC1,HC2,HC3,HC4,AI1,ENC1,ENC2,
194.      3ENC3,EIN1,EIN2,EIN3,TINTC,TINBC,JGUESS,
195.      8T(J),T(J-1),T(J+1),J,2)
196.      C(J)=D1+AI1+HC1
197.      U(J)=D2-AI1+HC2
198.      E(J)=D3+HC3
199.      Z(J)=- (D4+HC4)*TINF
200.      A(J+1)=AI1+F1+G1
201.      B(J+1)=-AI1+F2+G2+CPM
202.      C(J+1)=F3+G3
203.      DEN(J+1)=-CPM
204.      Z(J+1)=- (F4+G4)*TINF
205.      A(J+2)=D1-F1-E1
206.      B(J+2)=D2-F2-E2
207.      C(J+2)=D3-F3-E3
208.      Z(J+2)=- (D4-F4-E4)*TINF
209. 300 CONTINUE
210.      J=LHALF+2
211.      C(J)=C(J)-CH2
212.      Z(J)=Z(J)+CH3*TINF+CH1*TINH
213.      LB=LHALF+4
214.      LE=LENG-5
215.      DO 400 J=LB,LE,4
216.      CALL HOT(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,AP,
217.      1AEIN,TINF,DIAMH,APLH,CPM,CH1,CH2,CH3,ALH1,ALH2,ALH3,D1,D2,D3,
218.      2D4,E1,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,AI1,AI2,
219.      3ENH1,ENH2,ENH3,EOUT1,EOUT2,EOUT3,TINBH,TINTH,JGUESS,
220.      8I(J),T(J-1),T(J+1),J,2)
221.      C(J)=C(J)-D1+AI1+H1
222.      U(J)=-D2+AI2+H2
223.      E(J)=-L3+H3
224.      Z(J)=Z(J)- (-D4+H4)*TINF
225.      A(J+1)=AI1+F1+G1
226.      B(J+1)=-CPM+AI2+F2+G2

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227.      C(J+1)=F3+G3
228.      DEN(J+1)=CPM
229.      Z(J+1)=- (F4+G4)*TINF
230.      A(J+2)=-U1-E1-F1
231.      B(J+2)=-U2-E2-F2
232.      C(J+2)=C(J+2)-D3-E3-F3
233.      Z(J+2)=Z(J+2)+(D4+E4+F4)*TINF
234. 400 CONTINUE
235.      J=LENG
236.      J1=LENG-1
237.      CALL HUT(FLOW,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,AP,
238. 1AEIN,TINF,DIAMH,APLH,CPM,CH1,CH2,CH3,ALH1,ALH2,ALH3,D1,D2,D3,
239. 2D4,E1,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,AI1,AI2,
240. 3ENH1,ENH2,ENH3,EOUT1,EOUT2,EOUT3,TINBH,TINTH,JGUESS,
241. 8T(J-1),T(J-2),DUM,J1,3)
242.      CPM=CPM
243.      C(J-1)=C(J-1)+ENH2
244.      D(J-1)=ENH1
245.      Z(J-1)=Z(J-1)-ENH3*TINF
246.      A(J)=EOUT2
247.      B(J)=EOUT1
248.      C(J)=-1.0
249.      Z(J)=-EOUT3*TINF
250.      IF (NOT.NE.2) GO TO 500
251.      DO 49 I=1,LENG
252.      WRITE(6,51) A(I),B(I),C(I),D(I),E(I),DEN(I)
253. 51 FORMAT(1X,6F10.5)
254.      WRITE(6,58) Z(I)
255. 58 FORMAT(1X,F14.5)
256. 49 CONTINUE
257. 500 CONTINUE
258. C SOLVING MATRIX BY GAUSS-SIEDEL
259. 75 CONTINUE
260.      T(1)=(-B(1)*T(2)-C(1)*T(3)+Z(1))/A(1)
261.      T(3)=(-A(2)*T(2)-C(2)*T(4)-D(2)*T(5)+Z(2))/B(2)
262.      LE=LHALF-1
263.      DO 52 J=4,LE,4
264.      T(J+LHALF)=(-A(J-1)*T(J-1)-B(J-1)*T(J)-C(J-1)*T(J+1)+Z(J-1))
265. 1/DEN(J-1)
266. 52 CONTINUE
267.      LE=LHALF-2
268.      DO 53 J=5,LE,2
269.      T(J)=(-A(J-1)*T(J-2)-B(J-1)*T(J-1)-D(J-1)*T(J+1)-E(J-1)*T(J+2)
270. 1+Z(J-1))/C(J-1)
271. 53 CONTINUE
272.      LE=LHALF-3
273.      DO 54 J=6,LE,4
274.      T(J+LHALF-4)=(-A(J-1)*T(J-1)-B(J-1)*T(J)-C(J-1)*T(J+1)+Z(J-1))
275. 1/DEN(J-1)
276. 54 CONTINUE
277.      J=LHALF
278.      T(J)=(-A(J-1)*T(J-2)-B(J-1)*T(J-1)+Z(J-1))/C(J-1)
279.      T(LENG-3)=(-A(J)*T(J)+Z(J))/DEN(J)
280. C LOWER MATRIX
281.      J=LHALF+1
282.      T(4)=(-A(J)*T(J)+Z(J))/DEN(J)
283.      J=LHALF+2

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284.      T(J-1)=(-D(J)*T(J)-E(J)*T(J+1)+Z(J))/C(J)
285.      LB=LHALF+2
286.      LE=LENG-3
287.      DO 55 J=LB,LE,4
288.      T(J-LHALF)=(-A(J+1)*T(J-1)-B(J+1)*T(J)-C(J+1)*T(J+1)+Z(J+1))
289.      1/DEN(J+1)
290. 55 CONTINUE
291.      LB=LHALF+3
292.      LE=LENG-4
293.      DO 56 J=LB,LE,2
294.      T(J)=(-A(J+1)*T(J-2)-B(J+1)*T(J-1)-D(J+1)*T(J+1)-E(J+1)*T(J+2)
295.      1+Z(J+1))/C(J+1)
296. 56 CONTINUE
297.      LB=LHALF+4
298.      LE=LENG-5
299.      DO 57 J=LB,LE,4
300.      T(J-LHALF+4)=(-A(J+1)*T(J-1)-B(J+1)*T(J)-C(J+1)*T(J+1)+Z(J+1))
301.      1/DEN(J+1)
302. 57 CONTINUE
303.      J=LENG-1
304.      T(J-1)=(-A(J)*T(J-3)-B(J)*T(J-2)-D(J)*T(J)+Z(J))/C(J)
305.      J=LENG
306.      T(J)=(-A(J)*T(J-2)-B(J)*T(J-1)+Z(J))/C(J)
307.      CHECK=0.0
308.      JINT=JINT+1
309. C CHECK GAUSS-SIEDEL CONVERGANCE
310.      DO 80 J=1,LENG
311.      CNEW=ABS(TOLD(J)-T(J))
312.      TOLD(J)=T(J)
313.      IF(CNEW.GT.CHECK) CHECK=CNEW
314.      IF(NOT.EQ.0) GO TO 80
315.      WRITE(6,81) J,T(J)
316. 81 FORMAT(1X,'T',13,'=' ,F14.5)
317.      IF(J.NE.LENG) GO TO 80
318.      WRITE(6,82)
319. 82 FORMAT(///)
320. 80 CONTINUE
321.      IF(CHECK.LT.CONIT) GO TO 83
322.      IF(JINT.GT.MAXIT) GO TO 84
323.      GO TO 500
324. 83 CONTINUE
325. C CHECK SOLUTION,ANOTHER CHECK OF GAUSS-SIEDEL
326.      REM(1)=A(1)*T(1)+B(1)*T(2)+C(1)*T(3)-Z(1)
327.      REM(2)=A(2)*T(2)+B(2)*T(3)+C(2)*T(4)+D(2)*T(5)-Z(2)
328.      LE=LHALF-1
329.      DO 552 J=4,LE,4
330.      REM(J-1)=A(J-1)*T(J-1)+B(J-1)*T(J)+C(J-1)*T(J+1)
331.      1+DEN(J-1)*T(J+LHALF)-Z(J-1)
332. 552 CONTINUE
333.      LE=LHALF-2
334.      DO 553 J=5,LE,2
335.      REM(J-1)=A(J-1)*T(J-2)+B(J-1)*T(J-1)+D(J-1)*T(J+1)+E(J-1)*T(J+2)
336.      1+C(J-1)*T(J)-Z(J-1)
337. 553 CONTINUE
338.      LE=LHALF-3
339.      DO 554 J=6,LE,4
340.      REM(J-1)=A(J-1)*T(J-1)+B(J-1)*T(J)+C(J-1)*T(J+1)

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341.      1+DEN(J-1)*T(J+LHALF-4)-Z(J-1)
342. 554 CONTINUE
343.      J=LHALF
344.      REM(J-1)=A(J-1)*T(J-2)+B(J-1)*T(J-1)+C(J-1)*T(J)-Z(J-1)
345.      REM(J)=A(J)*T(J)+DEN(J)*T(LENG-3)-Z(J)
346. C LOWER MATRIX
347.      J=LHALF+1
348.      REM(J)=A(J)*T(J)+DEN(J)*T(4)-Z(J)
349.      J=LHALF+2
350.      REM(J)=C(J)*T(J-1)+D(J)*T(J)+E(J)*T(J+1)-Z(J)
351.      LB=LHALF+2
352.      LE=LENG-3
353.      DO 555 J=LB,LE,4
354.          REM(J+1)=A(J+1)*T(J-1)+B(J+1)*T(J)+C(J+1)*T(J+1)
355.          1+DEN(J+1)*T(J-LHALF)-Z(J+1)
356. 555 CONTINUE
357.      LB=LHALF+3
358.      LE=LENG-4
359.      DO 556 J=LB,LE,2
360.          REM(J+1)=A(J+1)*T(J-2)+B(J+1)*T(J-1)+D(J+1)*T(J+1)
361.          1+E(J+1)*T(J+2)+C(J+1)*T(J)-Z(J+1)
362. 556 CONTINUE
363.      LB=LHALF+4
364.      LE=LENG-5
365.      DO 557 J=LB,LE,4
366.          REM(J+1)=A(J+1)*T(J-1)+B(J+1)*T(J)+C(J+1)*T(J+1)
367.          1+DEN(J+1)*T(J-LHALF+4)-Z(J+1)
368. 557 CONTINUE
369.      J=LENG-1
370.      REM(J)=A(J)*T(J-3)+B(J)*T(J-2)+C(J)*T(J-1)+D(J)*T(J)-Z(J)
371.      J=LENG
372.      REM(J)=A(J)*T(J-2)+B(J)*T(J-1)+C(J)*T(J)-Z(J)
373.      CHECK=0.0
374.      DO 558 J=1,LENG
375.          IF (REM(J).GT.CHECK) CHECK=REM(J)
376. 558 CONTINUE
377.      IF (CHECK.GT.CONIT) GO TO 500
378.      CHECK=0.0
379. C CHECK TOTAL SOLUTION CONVERGANCE
380.      DO 86 J=1,LENG
381.          CNEW=ABS(TCK(J)-T(J))
382.          TCK(J)=T(J)
383.          IF (CNEW.GT.CHECK) CHECK=CNEW
384. 86 CONTINUE
385.      IF (CHECK.LT.CONVG) GO TO 98
386.      KCOUNT=KCOUNT+1
387.      IF (KCOUNT.GT.KEND) GO TO 99
388.      JGUESS=5
389.      GO TO 80
390. 84 CONTINUE
391.      WRITE(6,93) KCOUNT
392. 93 FORMAT(1X,'GAUSS SEIDEL DOES NOT CONVERGE ',I3)
393.      GO TO 99
394. 98 CONTINUE
395.      WRITE(6,91) KCOUNT
396. 91 FORMAT(1X,'CONVERGED',I5/)
397. 99 CONTINUE

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398.      IF(NOT.NE.0) GO TO 102
399.      TCOLD=T(1)/CPC+T(2)-460.0
400.      THOT=T(LENG)/CPH+T(LENG-1)-460.0
401.      WRITE(6,97) TCOLD,THOT
402.      97 FORMAT(1X,' COLD OUTLET (OF)= ',F14.5/1X,' HOT OUTLET (OF)= ',
403.      1F14.5///)
404.      IF(NPR.EQ.0) GO TO 102
405.      DO 92 J=1,LENG
406.      WRITE(6,81)J,T(J)
407.      92 CONTINUE
408.      102 CONTINUE
409.      END
410.      SUBROUTINE HOT(FLOWS,AFLOW,FINT,FWALL,AK,AWALL,AINT,HINS,CINS,AF
411.      1AEIN,TINF,DIAMH,APLH,CPM,CH1,CH2,CH3,ALH1,ALH2,ALH3,D1,D2,D3,
412.      2D4,E1,E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,H1,H2,H3,H4,AI1,AI2,
413.      3ENH1,ENH2,ENH3,EOUT1,EOUT2,EOUT3,TINBH,TINTH,JGUESS,
414.      6TEMP,T1,TN2,J,ISECT)
415.      DIMENSION W1(300),W2(300),W3(300),W4(300)
416.      DIMENSION Z1(300),Z2(300),Z3(300),Z4(300)
417.      DIMENSION X1(300),X2(300)
418.      APLC=AP
419.      IF(JGUESS.EQ.1) GO TO 500
420.      TEMP=TEMP-460.0
421.      C HEAT CAPACITY TIMES FLOW RATE
422.      CPM=(.2384+1.458E-5*TEMP+1.1073E-8*TEMP**2-3.184E-12*TEMP**3)
423.      1*FLOWS*3600.0
424.      C CURVE FITS AND CONVECTION COEFF,H
425.      VIS=(1.13+1.595E-3*TEMP-2.275E-7*TEMP**2)*1.E-5
426.      REYN=(FLOWS*DIAMH)/(AFLOW*VIS)
427.      IF (TEMP.LT.32.0) GO TO 610
428.      IF (TEMP.LT.200.0) GO TO 620
429.      IF (TEMP.LT.500.0) GO TO 630
430.      IF (TEMP.LT.1000.0) GO TO 640
431.      IF (TEMP.LT.3000.0) GO TO 650
432.      610 CONTINUE
433.      WRITE(6,611)TEMP,J
434.      611 FORMAT(1X,'TEMP OUT OF RANGE H= ',F10.2,' J= ',I6)
435.      620 CONTINUE
436.      PR=.72
437.      GO TO 655
438.      630 CONTINUE
439.      PR=.7467-1.32E-4*TEMP
440.      GO TO 655
441.      640 CONTINUE
442.      PR=.649+6.2E-5*TEMP
443.      GO TO 655
444.      650 CONTINUE
445.      PR=.6464+8.56E-5*TEMP-1.486E-8*TEMP**2
446.      655 CONTINUE
447.      H=.01293+2.2068E-5*TEMP-2.746E-9*TEMP**2
448.      H=H*.05422*REYN**.786*PR**.33/DIAMH
449.      SIGMA=1.714E-9
450.      TEMP=TEMP+460.0
451.      GO TO 510
452.      500 CONTINUE
453.      H=2.0
454.      CPM=.24*FLOWS*3600.0

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455.      510 CONTINUE
456.      C ISECT=1,INLET SECTION      2,MIDDLE      3,EXIT
457.      IF(ISECT.EQ.2) GO TO 100
458.      IF(ISECT.EQ.3) GO TO 200
459.      C INLET SECTION
460.      IF(JGUESS.EQ.1) GO TO 656
461.      C FIND WALL,UPPER INTERFACE AND OUTSIDE INSULATION TEMPERATURES
462.      TW=W1(J)*TEMP+W2(J)*TN2+W3(J)*TINF
463.      T2=Z1(J)*TEMP+Z2(J)*TW+Z3(J)*TN2
464.      TIN=X1(J)*TW+X2(J)*TINF
465.      TINRH=TIN-460.0
466.      C GAS EMISSIVITY AT GIVEN TEMP
467.      C APLH SHOULD BE BETWEEN .005 AND .20 FR-ATM
468.      C APLC SHOULD BE BETWEEN .006 AND .10 FT-ATM
469.      I=1
470.      T=TW-460.0
471.      660 CONTINUE
472.      IF(APLH.GT..06.AND.APLH.LE..10) GO TO 400
473.      IF(APLH.GT..10.AND.APLH.LE..20) GO TO 410
474.      EMH=.007526+2.43552*APLH-14.5792*APLH**2
475.      EMH=EMH*EXP((- .94495E-3+.73195E-2*APLH-.047773*APLH**2)*T)
476.      GO TO 450
477.      400 CONTINUE
478.      EMH1=.10125*EXP(-.0006758*T)
479.      EMH2=.13084*EXP(-.0005869*T)
480.      EMH=(EMH2-EMH1)*(APLH-.06)/.04+EMH1
481.      GO TO 450
482.      410 CONTINUE
483.      EMH1=.13084*EXP(-.0005869*T)
484.      EMH2=.18934*EXP(-.0005132*T)
485.      EMH=(EMH2-EMH1)*(APLH-.10)/.10+EMH1
486.      450 CONTINUE
487.      IF(APLC.GT..02.AND.APLC.LE..04) GO TO 460
488.      IF(APLC.GT..04.AND.APLC.LE..10) GO TO 470
489.      IF(T.LE.1000.0) GO TO 665
490.      CFF=-4.07561+82.7321*APLC-2216.0714*APLC**2
491.      CFF=(-1.00712+209.311*APLC+1029.7653*APLC**2)*1.E-4
492.      CFFF=(-.611146-106.8041*APLC+1888.0714*APLC**2)*1.E-7
493.      EMC=EXP(CF+CFF*T+CFFF*T**2)
494.      GO TO 670
495.      665 CONTINUE
496.      EMC=.04257+1.5714*APLC
497.      GO TO 670
498.      460 CONTINUE
499.      IF(T.LI.1000.0) GO TO 465
500.      EMC1=EXP(-3.5074+.3591E-3*T-.1992E-6*T**2)
501.      EMC2=EXP(-2.9737+2.8345E-4*T-1.7389E-7*T**2)
502.      EMC=(EMC2-EMC1)*(APLC-.02)/.02+EMC1
503.      GO TO 670
504.      465 CONTINUE
505.      EMC=(.057-.044)*(APLC-.02)/.02+.044
506.      GO TO 670
507.      470 CONTINUE
508.      IF(T.LI.1000.0) GO TO 475
509.      EMC1=EXP(-2.9737+2.8345E-4*T-1.7389E-7*T**2)
510.      EMC2=EXP(-2.2012+4.384E-6*T-1.2137E-7*T**2)
511.      EMC=(EMC2-EMC1)*(APLC-.04)/.06+EMC1

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512.      GO TO 670
513.      475 CONTINUE
514.      EMC=(.08-.057)*(APLC-.04)/.06+.057
515.      670 CONTINUE
516.      IF(I.EQ.1) EMW=EMC+EMH
517.      IF(I.EQ.2) EMI=EMC+EMH
518.      IF(I.EQ.3) EMG=EMC+EMH
519.      IF (I.EQ.3) GO TO 675
520.      I=I+1
521.      IF(I.EQ.2) T=T2-460.0
522.      IF(I.EQ.3) T=TEMP-460.0
523.      GO TO 660
524.      675 CONTINUE
525.      TAW=(1.-EMW)**.25
526.      TA2=(1.-EMI)**.25
527.      BW=EMW**.25
528.      BG=EMG**.25
529.      B2=EMI**.25
530.      TINFR=1INF
531.      S=FWALL*SIGMA*(TA2*T2+TAW*TW)*((TA2*T2)**2+(TAW*TW)**2)
532.      GW=AWALL*SIGMA*(BG*TEMP+BW*TW)*((BG*TEMP)**2+(BW*TW)**2)
533.      GINT=AINT*SIGMA*(BG*TEMP+B2*T2)*((BG*TEMP)**2+(B2*T2)**2)
534.      RADI=AIN*SIGMA*(TIN+TINFR)*(TIN**2+TINFR**2)
535.      GO TO 657
536.      656 CONTINUE
537.      C FOR INITIAL GUESS
538.      TAW=1.0
539.      TA2=1.0
540.      BG=1.0
541.      BW=1.0
542.      B2=1.0
543.      S=FWALL*H
544.      GW=H*AWALL
545.      GINT=H*AIN
546.      RADI=H*INS
547.      657 CONTINUE
548.      C CALCULATION OF NODAL EQUATION CONSTANTS AND CONSTANTS FOR
549.      C ELIMINATED TEMPERATURE EQUATIONS
550.      DENOM=AK+H*AIN+GINT*B2
551.      U=1.0/(1.0/(S*TA2)+1.0/DENOM)
552.      A1=TAW/TA2
553.      A2=AK/DENOM
554.      A3=(H*AIN+GINT*BG)/DENOM
555.      AL=1.0/(1.0/(H*INS+RADI)+1.0/CINS)
556.      DENOM=H*AWALL+GW*BW+U*A1
557.      ALOSS=1.0/(1.0/AL+1.0/DENOM)
558.      AL1=(H*AWALL+G*BG+U*A3)/DENOM
559.      AL2=(U*A2)/DENOM
560.      A1(U)=ALOSS*AL1/AL
561.      A2(U)=ALOSS*AL2/AL
562.      A3(U)=1.0-ALOSS/AL
563.      ZBT=H*AIN+GINT*B2+S*TA2+AK
564.      Z1(U)=(H*AIN+GINT*BG)/ZBT
565.      Z2(U)=S*TAW/ZBT
566.      Z3(U)=Z2(U)*AK/(S*TAW)
567.      DENOM=H*AIN+GINT*B2+S*TA2
568.      UP=1.0/(1.0/AK+1.0/DENOM)

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569.      C      Q CONDUCTI
570.      CH1=((H*INT+GINT*BG+S*TAW*ALOSS*AL1/AL)/DENOM)*UP
571.      CH2=((S*TAW*ALOSS*AL2/AL)/DENOM-1.0)*UP
572.      CH3=((S*TAW*(1.0-ALOSS/AL))/DENOM)*UP
573.      C      Q LOSS
574.      ALH1=ALOSS*AL1
575.      ALH2=ALOSS*AL2
576.      ALH3=-ALOSS
577.      X=HINS+RADI+CINS
578.      X1(J)=CINS/X
579.      X2(J)=(HINS+RADI)/X
580.      GO TO 25
581.      C MIDDLE SECTIONS
582.      100 CONTINUE
583.      IF (JGUESS.EQ.1) GO TO 756
584.      C FIND WALL, UPPER INTERFACE AND OUTSIDE INSULATION TEMPS
585.      TW=W1(J)*TEMP+W2(J)*T1+W3(J)*TN2+W4(J)*TINF
586.      T2=Z1(J)*TEMP+Z2(J)*TW+Z3(J)*TN2+Z4(J)*T1
587.      TIN=X1(J)*TW+X2(J)*TINF
588.      C GAS EMISSIVITY AT GIVEN TEMP
589.      C APLH SHOULD BE BETWEEN .005 AND .20 FR-ATM
590.      C APLC SHOULD BE BETWEEN .006 AND .10 FT-ATM
591.      I=1
592.      T=TW-400.0
593.      760 CONTINUE
594.      IF (APLH.GT..06.AND.APLH.LE..10) GO TO 401
595.      IF (APLH.GT..10.AND.APLH.LE..20) GO TO 411
596.      EMH=.00752812.43352*APLH+14.5792*APLH**2
597.      EMH=EMH*EXP((- .94495E-3+.73195E-2*APLH-.047773*APLH**2)*T)
598.      GO TO 451
599.      401 CONTINUE
600.      EMH1=.10125*EXP(-.0006758*T)
601.      EMH2=.13084*EXP(-.0005869*T)
602.      EMH=(EMH2-EMH1)*(APLH-.06)/.04+EMH1
603.      GO TO 451
604.      411 CONTINUE
605.      EMH1=.13084*EXP(-.0005869*T)
606.      EMH2=.18934*EXP(-.0005132*T)
607.      EMH=(EMH2-EMH1)*(APLH-.10)/.10+EMH1
608.      451 CONTINUE
609.      IF (APLC.GT..02.AND.APLC.LE..04) GO TO 461
610.      IF (APLC.GT..04.AND.APLC.LE..10) GO TO 471
611.      IF (T.LE.1000.0) GO TO 765
612.      CF=-4.67561+82.7321*APLC-2216.0714*APLC**2
613.      CFF=(-1.00712+.09.311*APLC+1029.7653*APLC**2)*1.E-4
614.      CFFF=(-.611146-136.8041*APLC+1888.0714*APLC**2)*1.E-7
615.      LMC=EXP(CF+CFF*T+CFFF*T**2)
616.      GO TO 770
617.      765 CONTINUE
618.      LMC=.01257+1.5714*APLC
619.      GO TO 770
620.      461 CONTINUE
621.      IF (T.LI.1000.0) GO TO 466
622.      LMC1=EXP(-3.3074+.3591E-3*T-.1992E-6*T**2)
623.      LMC2=EXP(-2.9737+2.8345E-4*T-1.7389E-7*T**2)
624.      LMC=(LMC2-EMC1)*(APLC-.02)/.02+EMC1
625.      GO TO 770

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626.      466 CONTINUE
627.      EMC=(.057-.044)*(APLC-.02)/.02+.044
628.      GO TO 770
629.      471 CONTINUE
630.      IF(T.LT.1000.0) GO TO 476
631.      EMC1=EXP(-2.9737+2.8345E-4*T-1.7389E-7*T**2)
632.      EMC2=EXP(-2.2012+4.384E-6*T-1.2137E-7*T**2)
633.      EMC=(EMC2-EMC1)*(APLC-.04)/.06+EMC1
634.      GO TO 770
635.      476 CONTINUE
636.      EMC=(.08-.057)*(APLC-.04)/.06+.057
637.      770 CONTINUE
638.      IF(I.EQ.1) EMW=EMC+EMH
639.      IF(I.EQ.2) EM1=EMC+EMH
640.      IF(I.EQ.3) EM2=EMC+EMH
641.      IF(I.EQ.4) EM6=EMC+EMH
642.      IF(I.EQ.4) GO TO 775
643.      I=I+1
644.      IF(I.EQ.2) T=T1-460.0
645.      IF(I.EQ.3) T=T2-460.0
646.      IF(I.EQ.4) T=TEMP-460.0
647.      GO TO 760
648.      775 CONTINUE
649.      TA1=(1.0-EM1)**.25
650.      TA2=(1.0-EM2)**.25
651.      TAW=(1.0-EMW)**.25
652.      BG=EM6**.25
653.      B1=EM1**.25
654.      B2=EM2**.25
655.      BW=EMW**.25
656.      TINFR=TINF
657.      K=FINT*SIGMA*(TA1*T1+TA2*T2)*((TA1*T1)**2+(TA2*T2)**2)
658.      S1=FWALL*SIGMA*(TA1*T1+TAW*TW)*((TA1*T1)**2+(TAW*TW)**2)
659.      S2=FWALL*SIGMA*(TA2*T2+TAW*TW)*((TA2*T2)**2+(TAW*TW)**2)
660.      GINT2=AINTE*SIGMA*(BG*TEMP+B2*T2)*((BG*TEMP)**2+(B2*T2)**2)
661.      GINT1=AINTE*SIGMA*(BG*TEMP+B1*T1)*((BG*TEMP)**2+(B1*T1)**2)
662.      GW=AWALL*SIGMA*(BG*TEMP+BW*TW)*((BG*TEMP)**2+(BW*TW)**2)
663.      RAD1=ALIN*SIGMA*(TIN+TINFR)*((TIN**2+TINFR**2)
664.      GO TO 757
665.      756 CONTINUE
666.      C INITIAL GUESS
667.      TAW=1.0
668.      TA2=1.0
669.      TA1=1.0
670.      B1=1.0
671.      B2=1.0
672.      BW=1.0
673.      BG=1.0
674.      K=H*FI,T
675.      S1=H*FWALL
676.      S2=S1
677.      GINT1=H*AINTE
678.      GINT2=GINT1
679.      GW=H*AWALL
680.      RAD1=H*INS
681.      757 CONTINUE
682.      C CALCULATION OF NODAL EQUATION CONSTANTS AND CONSTANTS FOR

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683.      C ELIMINATED TEMPERATURE EQUATIONS
684.      AL=1.0/(1.0/(H1NS+RADI)+1.0/CINS)
685.      EH2=(AK+R*TA2+S2*TA2)/(H*AIN+GINT2*B2)
686.      EH2=1.0/(EH2+1.0)
687.      ER=(AK+S2*TA2+H*AIN+GINT2*B2)/(R*TA2)
688.      ER=1.0/(ER+1.0)
689.      ER2=(AK+H*AIN+GINT2*B2+R*TA2)/(S2*TA2)
690.      ER2=1.0/(ER2+1.0)
691.      BONE=H*AWALL+GW*BW+ER2*(AK+H*AIN+GINT2*B2+R*TA2)*TAW/TA2
692.      BONE=1.0/(BONE+AL+S1*TAW)
693.      W1(J)=BONE*(H*AWALL+GW*BG+ER2*(H*AIN+GINT2*BG))
694.      W2(J)=BONE*(ER2*R*TA1+S1*TA1)
695.      W3(J)=BONE*ER2*AK
696.      W4(J)=BONE*AL
697.      D1=ER*(AK+S2*TA2+H*AIN+GINT2*B2)*TA1/TA2-S2*TAW*W2(J)
698.      D2=-ER*(H*AIN+GINT2*BG)+S2*TAW*W1(J)
699.      D3=-ER*(AK+S2*TA2+H*AIN+GINT2*B2)*W3(J)
700.      D4=-ER*S2*TAW*W4(J)
701.      TOP=(AK+H*AIN+GINT2*B2+R*TA2)*TAW/TA2
702.      E1=ER2*(-R*TA1+TOP*W2(J))
703.      E2=ER2*(-(H*AIN+GINT2*BG)+TOP*W1(J))
704.      E3=ER2*(-AK+TOP*W3(J))
705.      E4=ER2*TOP*W4(J)
706.      F1=EH2*(-R*TA1-S2*TAW*W2(J))
707.      F2=EH2*((H*AIN+GINT2*BG)*(AK+R*TA2+S2*TA2)/(H*AIN+GINT2*B2)
708.      1-S2*TAW*W1(J))
709.      F3=EH2*(-AK-S2*TAW*W3(J))
710.      F4=EH2*(-S2*TAW*W4(J))
711.      TOP=H*AWALL+GW*BW
712.      G1=-TOP*W2(J)
713.      G2=H*AWALL+GW*BG-TOP*W1(J)
714.      G3=-TOP*W3(J)
715.      G4=-TOP*W4(J)
716.      H1=-S1*TA1+S1*TAW*W2(J)
717.      H2=S1*TA1+H*AWALL+GW*BW-TOP*W1(J)
718.      H3=S1*TA1+H*AWALL+GW*BW-TOP*W3(J)
719.      H4=S1*TA1+H*AWALL+GW*BW-TOP*W4(J)
720.      AI1=-(H*AIN+GINT1*B1)
721.      AI2=H*AIN+GINT1*BG
722.      Z=R*TA2+H*AIN+GINT2*B2+S2*TA2+AK
723.      Z1(J)=(H*AIN+GINT2*BG)/Z
724.      Z2(J)=S2*TAW/Z
725.      Z3(J)=AK/Z
726.      Z4(J)=R*TA1/Z
727.      X=HINS+RADI+CINS
728.      X1(J)=LINS/X
729.      X2(J)=(HINS+RADI)/X
730.      GO TO 25
731.      C      NOT TOP SECTION
732.      200 CONTINUE
733.      IF(JGUESS.EQ.1) GO TO 856
734.      C FIND WALL AND OUTSIDE INSULATION TEMPS
735.      TW=W1(J)*TEMP+W2(J)*T1+W3(J)*TINF
736.      TIN=X1(J)*TW+X2(J)*TINF
737.      TINTH=TIN-460.0
738.      C GAS EMISSIVITY AT GIVEN TEMP
739.      C APLH SHOULD BE BETWEEN .005 AND .20 FR-ATM

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740. C APLC SHOULD BE BETWEEN .006 AND .10 FT-ATM
741. I=1
742. T=TW-460.0
743. 860 CONTINUE
744. IF (APLH.GT..06.AND.APLH.LE..10) GO TO 402
745. IF (APLH.GT..10.AND.APLH.LE..20) GO TO 412
746. EMH=.007528+2.43352*APLH-14.5792*APLH**2
747. EMH=EMH*EXP((- .94495E-3+.73195E-2*APLH-.047773*APLH**2)*T)
748. GO TO 452
749. 402 CONTINUE
750. EMH1=.10125*EXP(-.0006758*T)
751. EMH2=.13084*EXP(-.0005869*T)
752. EMH=(EMH2-EMH1)*(APLH-.06)/.04+EMH1
753. GO TO 452
754. 412 CONTINUE
755. EMH1=.13084*EXP(-.0005869*T)
756. EMH2=.18934*EXP(-.0005132*T)
757. EMH=(EMH2-EMH1)*(APLH-.10)/.10+EMH1
758. 452 CONTINUE
759. IF (APLC.GT..02.AND.APLC.LE..04) GO TO 462
760. IF (APLC.GT..04.AND.APLC.LE..10) GO TO 472
761. IF (T.LE.1000.0) GO TO 862
762. CF=-4.07561+82.7321*APLC-2216.0714*APLC**2
763. CFF=(-1.00712+209.311*APLC+1029.7653*APLC**2)*1.E-4
764. CFFF=(-.611146-106.8041*APLC+1888.0714*APLC**2)*1.E-7
765. EMC=EXP(CF+CFF*T+CFFF*T**2)
766. GO TO 870
767. 862 CONTINUE
768. EMC=.01257+1.5714*APLC
769. GO TO 870
770. 462 CONTINUE
771. IF (T.LI.1000.0) GO TO 467
772. EMC1=EXP(-3.3074+.3591E-3*T-.1992E-6*T**2)
773. EMC2=EXP(-2.9737+2.8345E-4*T-1.7389E-7*T**2)
774. EMC=(EMC2-EMC1)*(APLC-.02)/.02+EMC1
775. GO TO 870
776. 467 CONTINUE
777. EMC=(.057-.044)*(APLC-.02)/.02+.044
778. GO TO 870
779. 472 CONTINUE
780. IF (T.LI.1000.0) GO TO 477
781. EMC1=EXP(-2.9737+2.8345E-4*T-1.7389E-7*T**2)
782. EMC2=EXP(-2.2012+.4384E-6*T-1.2137E-7*T**2)
783. EMC=(EMC2-EMC1)*(APLC-.04)/.06+EMC1
784. GO TO 870
785. 477 CONTINUE
786. EMC=(.08-.057)*(APLC-.04)/.06+.057
787. 870 CONTINUE
788. IF (I.EQ.1) ENH=EMC+EMH
789. IF (I.EQ.2) ENI=EMC+EMH
790. IF (I.EQ.3) ENG=EMC+EMH
791. IF (I.EQ.3) GO TO 875
792. I=I+1
793. IF (I.EQ.2) T=T1-460.0
794. IF (I.EQ.3) T=TEMP-460.0
795. GO TO 860
875 CONTINUE

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797.     TAW=(1.0-EMW)**.25
798.     TA1=(1.0-EMI)**.25
799.     BW=EMW**.25
800.     BG=EMG**.25
801.     B1=EMI**.25
802.     TINFR=TINF
803.     S=FWALL*SIGMA*(TAW*TW+TA1*T1)*((TAW*TW)**2+(TA1*T1)**2)
804.     GINT=AIN*SIGMA*(BG*TEMP+B1*T1)*((BG*TEMP)**2+(B1*T1)**2)
805.     GW=AWALL*SIGMA*(BG*TEMP+BW*TW)*((BG*TEMP)**2+(BW*TW)**2)
806.     RAD1=AEIN*SIGMA*(TIN+TINFR)*(TIN**2+TINFR**2)
807.     GO TO 857
808.
809. C INITIAL GUESS
810.     TA1=1.0
811.     TAW=1.0
812.     B1=1.0
813.     BW=1.0
814.     BG=1.0
815.     S=H*FWALL
816.     GINT=H*AIN*
817.     GW=H*AWALL
818.     RAD1=H*INS
819.
820. C CALCULATION OF NODAL EQUATION CONSTANTS AND CONSTANTS FOR
821. C ELIMINATED TEMPERATURE EQUATIONS
822.     AL=1.0/(1.0/(HINS+RAD1)+1.0/CINS)
823.     UPH=1.0/((GW*BW+H*AWALL+AL)/(S*TAW)+1.0)
824.     A1=GW*BG+H*AWALL
825.     A2=(GW*BW+H*AWALL+AL)*TA1/TAW
826.     ENH1=UPH*A1+GINT*BG+H*AIN*
827.     ENH2=-(UPH*A2+GINT*B1+H*AIN*)
828.     ENH3=UPH*AL
829.     W1=1.0/(GW*BW+H*AWALL+AL+S*TAW)
830.     EOUT1=W1*(AL+S*TAW)*(GW*BG+H*AWALL)+GINT*BG+H*AIN*
831.     EOUT2=-W1*(GW*BW+H*AWALL)*S*TA1-(GINT*B1+H*AIN*)
832.     EOUT3=-W1*AL*(GW*BW+H*AWALL)
833.     DENOM=GW*BW+H*AWALL+AL+S*TAW
834.     W1(J)=(GW*BG+H*AWALL)/DENOM
835.     W2(J)=S*TA1/DENOM
836.     W3(J)=AL/DENOM
837.     X=HINS+RAD1+CINS
838.     X1(J)=CINS/X
839.     X2(J)=(HINS+RAD1)/X
840.
841. C 25 CONTINUE
842.     RETURN
843.
844. SUBROUTINE COLD(FLOWS,AFLOW,FINT,FWALL,AK,AWALL,AIN,HINS,CINS,
845. 1AEIN,TINF,DIAMH,CPM,CC1,CC2,CC3,ALC1,ALC2,ALC3,D1,D2,D3,D4,E1,
846. 2E2,E3,E4,F1,F2,F3,F4,G1,G2,G3,G4,HC1,HC2,HC3,HC4,AI1,ENC1,ENC2,
847. 3ENC3,EIN1,EIN2,EIN3,TINTC,TINBC,JGUESS,
848. 4TEMP,T1,T2,T3,T4,ISECT)
849.     DIMENSION W1(300),W2(300),W3(300),W4(300)
850.     DIMENSION Z1(300),Z2(300),Z3(300),Z4(300)
851.     DIMENSION X1(300),X2(300)
852.     IF(JGUESS.EQ.1) GO TO 500
853.     TEMP=TEMP+460.0

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C HEAT CAPACITY TIMES FLOW RATE (BTU/HR*F)


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854.         CPM=(.2384+1.458E-5*TEMP+1.1073E-8*TEMP**2-3.1846E-12*TEMP**3)
855.         1*FLOWS*3600.0
856. C CURVE FITS AND CONVECTION COEFF,H
857.         VIS=(1.13+1.595E-3*TEMP-2.275E-7*TEMP**2)*1.E-5
858.         REYN=(FLOWS*DIAMH)/(AFLOW*VIS)
859.         IF (TEMP.LT.32.0) GO TO 610
860.         IF (TEMP.LT.200.0) GO TO 620
861.         IF (TEMP.LT.500.0) GO TO 630
862.         IF (TEMP.LT.1000.0) GO TO 640
863.         IF (TEMP.LT.3000.0) GO TO 650
864. 610 CONTINUE
865.         WRITE(6,611) TEMP
866. 611 FORMAT(1X,'TEMP OUT OF RANGE=',F10.2)
867. 620 CONTINUE
868.         PR=.72
869.         GO TO 655
870. 630 CONTINUE
871.         PR=.7467-1.32E-4*TEMP
872.         GO TO 655
873. 640 CONTINUE
874.         PR=.649+6.2E-5*TEMP
875.         GO TO 655
876. 650 CONTINUE
877.         PR=.6434+8.56E-5*TEMP-1.486E-8*TEMP**2
878. 655 CONTINUE
879.         H=.01293+2.2068E-5*TEMP-2.746E-9*TEMP**2
880.         H=H*.05422*REYN**.786*PR**.33/DIAMH
881.         SIGMA=1.714E-9
882.         TEMP=TEMP+460.0
883.         GO TO 510
884. 500 CONTINUE
885.         H=2.0
886.         CPM=.24*FLOWS*3600.0
887. 510 CONTINUE
888. C 1SECT=1,BOTTOM      2,MIDDLE      3,TOP SECTIONS
889.         IF(ISECT.EQ.2) GO TO 100
890.         IF(ISECT.EQ.3) GO TO 200
891. C BOTTOM OR EXIT SECTION
892.         IF(JGUESS.EQ.1) GO TO 31
893. C FIND WALL AND UPPER INTERFACE AND OUTSIDE INSULATION TEMPS
894.         TW=W1(J)*TEMP+W2(J)*TN2+W3(J)*TINF
895.         T2=Z1(J)*TEMP+Z2(J)*TW+Z3(J)*TN2
896.         TIN=X1(J)*TW+X2(J)*TINF
897.         TINBC=TIN-460.0
898.         TINFR=TINF
899.         S=FWALL*SIGMA*(T2+TW)*(T2**2+TW**2)
900.         RAD1=A2IN*SIGMA*(TIN+TINFR)*(TIN**2+TINFR**2)
901.         GO TO 32
902. 31 CONTINUE
903. C INITIAL GUESS
904.         S=H*FWALL
905.         RAD1=HINS
906. 32 CONTINUE
907. C CALCULATION OF NODAL EQUATION CONSTANTS AND CONSTANTS FOR
908. C ELIMINATED TEMPERATURE EQUATIONS
909.         AL=1.0/(1.0/(HINS+RAD1)+1.0/CINS)
910.         DENOM=AK+H+ALNT

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911.      U=1.0/(1.0/S+1.0/DENOM)
912.      A1=H*AINT/DENOM
913.      A2=AK/DENOM
914.      DENOM=U+H*AINT
915.      ALOSS=1.0/(1.0/AL+1.0/(U+H*AINT))
916.      AL1=(U*A1+H*AINT)/DENOM
917.      AL2=U*A2/DENOM
918.      ALC1=ALOSS*AL1
919.      ALC2=ALOSS*AL2
920.      ALC3=-ALOSS
921.      W1(J)=ALOSS*AL1/AL
922.      W2(J)=ALOSS*AL2/AL
923.      W3(J)=1.0-ALOSS/AL
924.      DENOM=H*AINT+S
925.      UP=1.0/(1.0/AK+1.0/DENOM)
926.      CC1=-(H*AINT+S*AL1*ALOSS/AL)/DENOM
927.      CC2=-1.0*(-1.0+(S*ALOSS*AL2/AL)/DENOM)
928.      CC3=S*(1.0-ALOSS/AL)/DENOM
929.      CC1=UP*CC1
930.      CC2=UP*CC2
931.      CC3=UP*CC3
932.      X=HINS+RADI+CINS
933.      X1(J)=CINS/X
934.      X2(J)=(HINS+RADI)/X
935.      DENOM=H*AINT+S+AK
936.      Z1(J)=H*AINT/DENOM
937.      Z2(J)=S/DENOM
938.      Z3(J)=AK/DENOM
939.      GO TO 25
940.      100 CONTINUE
941.      C MIDDLE SECTIONS
942.      IF(GUESS.EQ.1) GO TO 41
943.      C FIND WALL AND UPPER INTERFACE AND OUTSIDE INSULATION TEMPS
944.      TW=W1(J)*TEMP+W2(J)*T1+W3(J)*TN2+W4(J)*TINF
945.      T2=Z1(J)*TEMP+Z2(J)*TW+Z3(J)*TN2+Z4(J)*T1
946.      TIN=X1(J)*TW+X2(J)*TINF
947.      TINF=1INF
948.      R=FINT*SIGMA*(T1+T2)*(T1**2+T2**2)
949.      S2=FWALL*SIGMA*(T2+TW)*(T2**2+TW**2)
950.      S1=FWALL*SIGMA*(T1+TW)*(T1**2+TW**2)
951.      RADI=AEIN*SIGMA*(TIN+TINF)*(TIN**2+TINF**2)
952.      GO TO 42
953.      41 CONTINUE
954.      C INITIAL GUESS
955.      R=H*FINT
956.      S2=H*FWALL
957.      S1=S2
958.      RADI=HINS
959.      42 CONTINUE
960.      C CALCULATION OF NODAL EQUATION CONSTANTS AND CONSTANTS FOR
961.      C ELIMINATED TEMPERATURE EQUATIONS
962.      AL=1.0/(1.0/(HINS+RADI)+1.0/CINS)
963.      ER=1.0/((S2+H*AINT+AK)/R+1.0)
964.      ER2=1.0/((R+AK+H*AINT)/S2+1.0)
965.      EH2=1.0/((R+AK+S2)/(H*AINT)+1.0)
966.      BONE=1.0/(S1+ER2*(R+AK+H*AINT)+H*AWALL+AL)
967.      W1(J)=(H*AWALL+ER2*AINT)*BONE

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968.      W2(J)=BONE*(S1+ER2*R)
969.      W3(J)=BONE*AK*ER2
970.      W4(J)=BONE*AL
971.      U1=ER*(S2+H*AJNT+AK)-ER*S2*W2(J)
972.      U2=-ER*(H*AJNT+S2*W1(J))
973.      U3=-ER*(AK+S2*W3(J))
974.      U4=-ER*S2*W4(J)
975.      TOP=R+AK+H*AJNT
976.      E1=ER2*(R-TOP*W2(J))
977.      E2=ER2*(H*AJNT-TOP*W1(J))
978.      E3=ER2*(AK-(R+AK+H*AJNT)*W3(J))
979.      E4=-ER2*TOP*W4(J)
980.      G1=H*AWALL*W2(J)
981.      G2=(-H*AWALL+H*AWALL*W1(J))
982.      G3=H*AWALL*W3(J)
983.      G4=H*AWALL*W4(J)
984.      F1=EH2*(R+S2*W2(J))
985.      F2=EH2*(-(R+AK+S2)+S2*W1(J))
986.      F3=EH2*(AK+S2*W3(J))
987.      F4=EH2*S2*W4(J)
988.      HC1=S1*(1.0-W2(J))
989.      HC2=-S1*W1(J)
990.      HC3=-S1*W3(J)
991.      HC4=-S1*W4(J)
992.      AI1=H*AJNT
993.      Z=R+S2+AK+H*AJNT
994.      Z1(J)=H*AJNT/Z
995.      Z2(J)=S2/Z
996.      Z3(J)=AK/Z
997.      Z4(J)=R/Z
998.      X=HINS+RADI+CINS
999.      X1(J)=CINS/X
1000.     X2(J)=(HINS+RADI)/X
1001.     GO TO 25
1002.     200 CONTINUE
1003.     C TOP OR INLET SECTION
1004.     IF(JGUESS.EQ.1) GO TO 51
1005.     C FIND WALL AND OUTSIDE INSULATION TEMPS
1006.     TW=W1(J)*TEMP+W2(J)*T1+W3(J)*TINF
1007.     TIN=X1(J)*TW+X2(J)*TINF
1008.     TINTC=TIN-460.0
1009.     TINFR=TINF
1010.     S=FWALL*SIGMA*(TW+T1)*(T1**2+TW**2)
1011.     RADI=AEIN*SIGMA*(TIN+TINFR)*(TIN**2+TINFR**2)
1012.     GO TO 52
1013.     51 CONTINUE
1014.     C INITIAL GUESS
1015.     S=H*FWALL
1016.     RADI=HINS
1017.     52 CONTINUE
1018.     C CALCULATION OF NODAL EQUATION CONSTANTS AND CONSTANTS FOR
1019.     C ELIMINATED TEMPERATURE EQUATIONS
1020.     AL=1.0/(1.0/(HINS+RADI)+1.0/CINS)
1021.     DENOM=(H*AWALL+AL)/S+1.0
1022.     ENC1=-H*AJNT-H*AWALL/DENOM
1023.     ENC2=H*AJNT+(H*AWALL+AL)/DENOM
1024.     ENC3=-AL/DENOM

```

```

1025.      DENOM=AL+H*AWALL+S
1026.      W1(J)=H*AWALL/DENOM
1027.      W2(J)=S/DENOM
1028.      W3(J)=AL/DENOM
1029.      DENOM=(S+AL)/(H*AWALL)+1.0
1030.      EIN1=-H*AJNT-(S+AL)/DENOM
1031.      EIN2=H*AJNT+S/DENOM
1032.      EIN3=AL/DENOM
1033.      X=HINS+RADI+CINS
1034.      X1(J)=CINS/X
1035.      X2(J)=(HINS+RADI)/X
1036.      25 CONTINUE
1037.      RETURN
1038.      END
END DATA.

```

QFIN

VALIDATION OF THERMAL MODEL

To determine the validity of the thermal model, the model's results are compared to the experimental results from hot flow testing. The input to the model represented the test conditions realized in testing. The input consisted of the air and exhaust gas mass flow rates, the air and exhaust gas inlet temperatures to a test section (48 interior recuperator modules), the partial pressures of water vapor and carbon dioxide in the exhaust gas (calculated by stoichiometry), and the system geometry. The results of this comparison are shown graphically in Figures A1.5 to A1.22. The graphs are plots of exhaust gas and air temperatures as a function of recuperator position. The graphs show that the thermal model works well over the test range, validating the model's assumptions and structure.

FIGURE A11.5. Comparison of Model and Experimental Results, Test No. 21

FIGURE A1.6. Comparison of Model and Experimental Results, Test No. 22-A

1-41

FIGURE A1.7. Comparison of Model and Experimental Results, Test No. 22

FIGURE A1.8. Comparison of Model and Experimental Results, Test No. 23
1-43

FIGURE A1.9. Comparison of Model and Experimental Results, Test No. 22/32
1-44.

FIGURE A1.10. Comparison of Model and Experimental Results, Test No. 25
1-45

FIGURE A1.11. Comparison of Model and Experimental Results, Test No. 26
1-46

FIGURE A1.12. Comparison of Model and Experimental Results, Test No. 27
1-47

FIGURE A1.13. Comparison of Model and Experimental Results, Test No. 29/30

FIGURE A1.14. Comparison of Model and Experimental Results, Test No. 30.
1-49

FIGURE A1.15. Comparison of Model and Experimental Results, Test No. 33-4

TEST NO. 33-B

- △ Combustion gas, model
- ⊙ Combustion gas, experimental
- Air, model
- Air, experimental

FIGURE A1.16. Comparison of Model and Experimental Results, Test No. 33-B

FIGURE A1.17. Comparison of Model and Experimental Results, Test No. 34

FIGURE A1.18. Comparison of Model and Experimental Results, Test No. 37
1-53

FIGURE A1.19: Comparison of Model and Experimental Results, Test No. 37/38

FIGURE A1.20. Comparison of Model and Experimental Results, Test No. 39

1-55

FIGURE A1.21: Comparison of Model and Experimental Results, Test No. 47.
1-56.

FIGURE A1.22. Comparison of Model and Experimental Results, Test No. 47/48
1-57

SYSTEM GEOMETRY

Recuperator Flow Parameter Calculations

All dimensions in the following discussion are illustrated in Figure A1.23.

The prototype recuperator has a rise of 8-inches in a 360° helix turn.

The internal diameter perpendicular to the axis is 7.8 inches. At the outer wall the interface plate makes an angle with this diameter as follows:

$$\phi = \tan^{-1} \left(\frac{L/2}{D} \right) = 27.15^\circ$$

The maximum height of the flow chamber perpendicular to flow is given by:

$$\text{Height} = (L/D) \cos \phi = 3.56''$$

The height represents one side of the flow duct. This side is considered to be a straight line. The other sides are best represented by a parabola of the form:

$$y = Ax^2 + B = 1.23x^2 + 0$$

$$y(0) = 0, y(1.78) = 3.9$$

This accounts for the plate thickness at the center of the recuperator. The flow cross section area is given by:

$$A = 2[(1.78)(3.9) - \int_0^{1.78} (1.23)x^2 dx]$$

$$A = 9.26 \text{ in}^2$$

The perimeter is then given by:

$$P = \sqrt{4y^2 + x^2} + \frac{x^2}{2y} \ln \left(\frac{2y + \sqrt{4y^2 + x^2}}{x} \right) + 2x$$

$$P = 12.45''$$

Section A-A

FIGURE A1.23. Recuperator Geometry

by definition the hydraulic diameter is defined as,

$$D_H = \frac{4 \times \text{cross sectional area}}{\text{wetted perimeter}}$$

$$D_H = (4 \times 9.26)/12.45 = 2.98"$$

To define the flow path length, the assumption is made that the mean radius is on the line of cross section symmetry at a point where a line perpendicular to the radius in the cross section plane divides the cross section into two equal areas. Then we can say that,

$$dA = 2 \times dy$$

and,

$$A = \int_0^{y^1} 2 \left(\frac{y}{a}\right)^{1/2} dy = \frac{4}{3} a \left(\frac{y^1}{a}\right)^{3/2}$$

since half the area is known we can calculate $y^1 = R$ (where R is the mean radius) as follows,

$$\frac{A}{2} = \frac{9.26}{2} = \frac{4}{3} (1.23) \left(\frac{R}{1.23}\right)^{3/2}$$

$$R = 2.46"$$

Then the path length is given by

$$L_H = [(2\pi R)^2 + 8^2]^{1/2}$$

$$L_H = 17.39"$$

The interface plate perimeter is the total perimeter minus the height or

$$P' = 12.45 - 2(1.78) = 8.89"$$

The heat exchange area is calculated as follows:

$$A = P' \times \text{path length}$$

$$A = (8.89) (17.73) \text{ in}^2/\text{rev}$$

$$A = 157.6 \text{ in}^2/\text{rev}$$

The radiation shape factors between the two interfaces and an interface and the wall approximated by simplifying the geometry. The method used calculates the shape factors involved in a cross-sectional view of the flow path. (See Figure A1.2A). The shape factors are calculated as follows:

$$\begin{aligned} \bar{A}_i F_{i-i} &= \bar{A}_{i,p} F_{i,p-i,p} = \bar{A}_{i,p} (1 - \sin \phi) \quad (\text{Reference 2}) \\ &= \sqrt{x^2 + y^2} (1 - \sin \phi) \end{aligned}$$

$$\sin \phi = x / \sqrt{x^2 + y^2}$$

$$\bar{A}_i = P' / 2$$

$$\bar{A}_{i,p} = \sqrt{x^2 + y^2}$$

$$F_{i-i} = \frac{\bar{A}_{i,p} (1 - \sin \phi)}{\bar{A}_i}$$

$$\bar{A}_i F_{i-w} = \bar{A}_{i,p} F_{i,p-w,p} = \bar{A}_{i,p} (1 - (1 - \sin \phi))$$

$$F_{i-w} = \frac{\bar{A}_{i,p} (1 - (1 - \sin \phi))}{\bar{A}_i}$$

where,

$\bar{A}_i$ = Area of section $i,1$ and $i,2$ in Figure A.6

$\bar{A}_{i,p}$ = Area of section P in Figure A.6

F = Shape factor

P' = Heat exchange perimeter

for the prototype recuperator

$F_{i-i} = .5640$

$F_{i-w} = .4004$

The mean path length is approximated as follows:

$$L_m = 3.4 V/A_T$$

where,

L = Mean path length (ft)

V = Flow volume per revolution (ft³/ rev)

A_T = Total area per revolution (ft²/rev)

For the prototype recuperator

$$L_m = .210 \text{ ft.}$$

FIGURE A1.24. Geometry for Shape Factor Calculation

CONVECTIVE HEAT TRANSFER COEFFICIENT AND PHYSICAL PROPERTIES CURVE FITS

The convective coefficient between the air or exhaust gas and the control volume boundaries was determined experimentally for the prototype recuperator. (See Section 3.3). The coefficient takes the following form,

$$h = .05422 \text{ Re}_{D_H}^{.786} \text{ Pr}^{1/3} K/D_H$$

where:

D_H = hydraulic diameter (ft)

h = convective heat transfer coefficient (Btu/hr·ft²·F)

K = fluid thermal conductivity (Btu/hr·ft·F)

Pr = Prandtl number

Re = Reynolds number

This relationship was determined experimentally with the bounds on the Reynolds number of 900 to 50,000.

The natural convective heat transfer coefficient off of the outside of the insulation is assumed to take the following form, (Reference 1)

$$h_{in} = .555 \frac{K}{L} (\text{Gr} \cdot \text{Pr}) \quad 10 < \text{Gr} < 10^9$$

$$h_{in} = .13 \frac{K}{L} (\text{Gr} \cdot \text{Pr}) \quad 10^9 < \text{Gr}$$

$$\text{Gr} = \rho^2 g \beta (T - T_\infty) L^3 / \mu^2$$

where:

Gr = Grashof number

g = gravitational acceleration (ft/sec^2)

h_{in} = natural convective heat transfer coefficient
($\text{Btu}/\text{hr}\cdot\text{ft}^2\cdot\text{F}$)

K = Thermal conductivity of air ($\text{Btu}/\text{hr}\cdot\text{ft}\cdot\text{F}$)

L = vertical height of recuperator (ft)

Pr = Prandtl number of air

T = outside insulation temperature (F)

T_{∞} = ambient air temperature (F)

β = coefficient of thermal expansion ($1/\text{F}$)

ρ = air density (lbm/ft^3)

μ = air viscosity ($\text{lbm}/\text{ft}\cdot\text{sec}$)

The physical properties of air in the above equations are evaluated at the film temperature. The film temperature is the average of the outside insulation temperature and the ambient temperature. The natural convective coefficient is assumed to be constant along the recuperator height using an average outside insulation temperature.

The curve fits of the physical properties of air used in the model take the following form,

$$Pr = \begin{cases} .72 & 32^{\circ}\text{F} \leq T \leq 200^{\circ}\text{F} \\ .7467 - 1.32 \times 10^{-4}T & 200^{\circ}\text{F} < T \leq 500^{\circ}\text{F} \\ .649 + 6.2 \times 10^{-5}T & 500 < T \leq 1000^{\circ}\text{F} \\ .6434 + 8.5 \times 10^{-5}T - 1.486 \times 10^{-8}T^2 & 1000 < T \leq 3000^{\circ}\text{F} \end{cases}$$

$$\mu = (1.13 + 1.595 \times 10^{-3}T - .2275 \times 10^{-6}T^2) \times 10^{-5}$$

$$K = .01293 + .22068 \times 10^{-4}T - .02746 \times 10^{-7}T^2$$

$$C_p = .2384 + 1.458 \times 10^{-5}T + 1.107 \times 10^{-8}T^2 \\ - 3.1846 \times 10^{-12}T^3$$

$$g\beta\rho^2/\mu^2 = \text{EXP}(14.6951 - 5.466 \times 10^{-3}T + .9662 \times 10^{-6}T^2)$$

where:

C_p = air heat capacity (Btu/hr·ft²·F)

g = gravitational acceleration (ft/sec²)

K = air thermal conductivity (Btu/hr·ft·F)

Pr = air Prantdl number

T = air temperature (F)

β = coefficient of thermal expansion (1/F)

ρ = air density (lbm/ft³)

μ = air viscosity (lbm/ft·sec)

HOT INTERIOR CONTROL VOLUME

$$1) \quad q_k = K(T_{hn,2} - T_{cn+1,1})$$

$$2) \quad q_{r,i-i} = R(t_1 T_{hn,1} - t_2 T_{hn,2})$$

$$t_1 = (1 - \epsilon_1)^{1/4}$$

$$t_2 = (1 - \epsilon_2)^{1/4}$$

$$3) \quad q_{v,q-i_2} = h_i A_i (\phi_n - T_{hn,2})$$

$$4) \quad q_{r,g-i_2} = G_{i,2} (b_g \phi_n - b_2 T_{hn,2})$$

$$b_g = \epsilon_g^{1/4}$$

$$b_2 = \epsilon_2^{1/4}$$

$$5) \quad q_{v,g-i_1} = h_i A_i (\phi_n - T_{hn,1})$$

$$6) \quad q_{r,g-i_1} = G_{i,1} (b_g \phi_n - b_1 T_{hn,1})$$

$$b_g = \epsilon_g^{1/4}$$

$$b_1 = \epsilon_1^{1/4}$$

$$7) \quad q_{v,g-w} = h_w A_w (\phi_n - T_w)$$

$$8) \quad q_{r,g-w} = G_w (b_g \phi_n - b_w T_w)$$

$$b_w = \epsilon_w^{1/4}$$

$$9) \quad q_{r,w-i_2} = S_2 (t_w T_w - t_2 T_{hn,2})$$

$$t_w = (1 - \epsilon_w)^{1/4}$$

$$t_w = (1 - \epsilon_2)^{1/4}$$

$$10) \quad q_{r,w-i_1} = S_1 (t_w T_w - t_1 T_{hn,1})$$

$$11) \quad q_l = L (T_w - T_\infty)$$

$$12) \quad q_g = \dot{m} C_p (\phi_n - \phi_{n+1})$$

Reduce equations to following form:

$$q_2 = (h_i A_i + G_{i,2} b_g) \phi_n - (h_i A_i + G_{i,2} b_2) T_{hn,2}$$

$$q_1 = (h_i A_i + G_{i,1} b_g) \phi_n - (h_i A_i + G_{i,1} b_1) T_{hn,1}$$

$$q_w = (h_w A_w + G_w b_g) \phi_n - (h_w A_w + G_w b_w) T_w$$

Eliminate $T_w, T_{hn,2}$

$$q_r = q_k + q_{r,2} - q_2$$

$$q_r = E_R [(K + S_2 t_2 + h_i A_i + G_{i,2} b_2) \frac{t_1}{t_2} T_{hn,1}$$

$$- K T_{cn+1,1} - S_2 t_w T_w - (h_i A_i + G_{i,2} b_g) \phi_n]$$

$$E_R = [(K + S_2 t_2 + h_i A_i + G_{i,2} b_2)/(R t_2 + 1)]^{-1}$$

$$q_{r,2} = q_k - q_2 - q_r$$

$$q_{r,2} = E_{r,2} \left[(K + h_i A_i + G_{i,2} b_2 + R t_2) \frac{t_w}{t_2} T_w - (h_i A_i + G_{i,2} b_g) \phi_n - K T_{cn+1,1} - R t_1 T_{hn,1} \right]$$

$$E_{r,2} = [(K + h_i A_i + G_{i,2} b_2 + R t_2) / S_2 t_2 + 1]^{-1}$$

$$q_2 = q_k - q_r - q_{r,2}$$

$$q_2 = E_{h,2} \left[(h_i A_i + G_{i,2} b_g) (K + R t_2 + S_2 t_2) \phi_n / (h_i A_i + G_{i,2} b_2) - K T_{cn+1,1} - R t_1 T_{hn,1} - S_2 t_w T_w \right]$$

$$E_{h,2} = [(K + R t_2 + S_2 t_2) / (h_i A_i + G_{i,2} b_2) + 1]^{-1}$$

Find an expression for T_w

$$q_w = q_{r,2} + q_{r,1} + q_1$$

$$T_w = \text{BONE} (B_1 \phi_n + B_2 T_{hn,1} + B_3 T_{cn+1,1} + B_4 T_\infty)$$

$$\text{BONE} = [h_w A_w + G_w b_w + E_{r,2} (K + h_i A_i + G_{i,2} b_2 + R t_2) \frac{t_w}{t_2} + L + S_1 t_w]^{-1}$$

$$B_2 = E_{r,2} R t_1 + S_1 t_1$$

$$B_3 = E_{r,2} K$$

$$B_4 = L$$

Eliminate T_w in equations

$$q_r = D_1 T_{hn,1} + D_2 \phi_n + D_3 T_{cn+1,1} + D_4 T_\infty$$

$$D_1 = E_R [(K + S_2 t_2 + h_i A_i + G_{i,2} b_2) t_1 / t_2 \\ - S_2 t_w \cdot BONE \cdot B_2]$$

$$D_2 = -E_R [(h_i A_i + G_{i,2} b_g) + S_2 t_w \cdot BONE \cdot B_1]$$

$$D_3 = -E_R (K + S_2 t_w \cdot BONE \cdot B_3)$$

$$D_4 = -E_R \cdot S_2 t_w \cdot BONE \cdot B_4$$

$$q_{r,2} = E_1 T_{hn,1} + E_2 \phi_n + E_3 T_{cn+1,1} + E_4 T_\infty$$

$$E_1 = E_{r,2} [-Rt_1 + (K + h_i A_i + G_{i,2} b_g + Rt_2) t_w / t_2 \\ \cdot BONE \cdot B_2]$$

$$E_2 = E_{r,2} [-(h_i A_i + G_{i,2} b_g) + (K + h_i A_i + G_{i,2} b_2 \\ + Rt_2) \cdot BONE \cdot B_1 t_w / t_2]$$

$$E_3 = E_{r,2} [-K + (K + h_i A_i + G_{i,2} b_2 + Rt_2) t_w / t_2 \\ \cdot BONE \cdot B_3]$$

$$E_4 = E_{r,2} (K + h_i A_i + G_{i,2} b_2 + Rt_2) \cdot BONE \cdot B_4 \\ \cdot t_w / t_2$$

The equations for the air properties were curve fit using least squares technique from property tables. (Reference 1).

The emittance of water vapor and carbon dioxide were curve fit from graphs of emittance versus temperature for various products of partial pressure and mean path length. (Reference 1). The curve fits take the following form,

$$\epsilon_{H_2O} = [.007528 + 2.43352 P_{H_2O} \cdot L_m - 14.5792 (P_{H_2O} \cdot L_m)^2] \\ \times \text{EXP} [(-.94195 \times 10^{-3} + .73195 \times 10^{-2} P_{H_2O} L_m \\ - .047773 (P_{H_2O} \cdot L_m)^2) T]$$

$$\epsilon_{CO_2} = \begin{cases} .01257 + 1.5714 \cdot P_{CO_2} \cdot L_m & T \leq 1000^\circ F \\ \text{EXP} (a + bT + cT^2) & 1000^\circ F < T \leq 3200^\circ F \end{cases}$$

$$a = -4.07561 + 82.7321 P_{H_2O} \cdot L_m - 22.16.0714 (P_{H_2O} \cdot L_m)^2$$

$$b = [-1.00712 + 209.311 \cdot P_{H_2O} \cdot L_m + 1029.7653 \\ (P_{H_2O} \cdot L_m)^2] 1 \times 10^{-4}$$

$$c = [-.611146 - 106.8041 \cdot P_{H_2O} L_m + 1888.0714 (P_{H_2O} \cdot L_m)^2] \\ 1 \times 10^{-7}$$

where:

L_m = mean path length (ft)

P = partial pressure (atm)

ϵ = gas emittance

subscripts H_2O = water vapor

CO_2 = carbon dioxide

DERIVATION OF NODAL EQUATIONS

Notation:

- A = Area (ft^2)
- C_p = Heat capacity ($\text{Btu/lbm}\cdot\text{F}$)
- G_i = Radiation heat transfer coefficient between gas and interface ($\text{Btu/hr}\cdot\text{R}$)
- G_w = Radiation heat transfer coefficient between gas and wall ($\text{Btu/hr}\cdot\text{R}$)
- h = Convective heat transfer coefficient ($\text{Btu/hr}\cdot\text{ft}^2\cdot\text{F}$)
- K = Conduction heat transfer coefficient ($\text{Btu/hr}\cdot\text{F}$)
- L = Heat loss coefficient to environment ($\text{Btu/hr}\cdot\text{F}$)
- $\dot{m}$ = Mass flow rate (lbm/hr)
- q = Heat transfer rate (Btu/hr)
- R = Radiation heat transfer coefficient between interfaces ($\text{Btu/hr}\cdot\text{R}$)
- S = Radiation heat transfer coefficient between interface and wall ($\text{Btu/hr}\cdot\text{R}$)
- T = Temperature (R)
- ϵ_x = Gas emissivity evaluated at temperature at zone x
- ϕ = Gas temperature (R)
- θ = Air temperature (R)

Subscripts:

- $cn, 1$ = Lower interface in air control volume n
- $cn, 2$ = Upper interface in air control volume n
- g = Gas
- $hn, 1$ = Lower interface in gas control volume n
- $hn, 2$ = Upper interface in gas control volume n

g-i = Relation between gas and interface

i-i = Relation between interfaces

i-w = Relation between interface and wall

r = Radiation

v = Convection

HOT INLET CONTROL

$$1) \quad q_k = K(T_{h1,2} - T_{c2,1})$$

$$2) \quad q_{r,g-i} = G_i (b_g \phi_1 - b_2 T_{h1,2})$$

$$b_g = \epsilon_g^{1/4}$$

$$b_2 = \epsilon_i^{1/4}$$

$$3) \quad q_{v,i} = h_i A_i (\phi_1 - T_{h1,2})$$

$$4) \quad q_{r,i-w} = S(t_w T_w - t_2 T_{h1,2})$$

$$t_w = (1 - \epsilon_w)^{1/4}$$

$$t_2 = (1 - \epsilon_L)^{1/4}$$

$$5) \quad q_{v,w} = h_w A_w (\phi_1 - T_w)$$

$$6) \quad q_{r,g-w} = G_w (b_g \phi_1 - b_w T_w)$$

$$b_w = \epsilon_w^{1/4}$$

$$7) \quad q_1 = L(T_w - T_\infty)$$

$$8) \quad q_g = \dot{m} C_p (\phi_1 - \phi_2)$$

REDUCE EQUATIONS

$$q = q_{r,g-i} + q_i$$

$$= (h_i A_i + G_i b_g) \phi_1 - (h_i A_i + G_i b_2) T_{h1,2}$$

$$q_w = q_{v,w} + q_{r,g-w}$$

$$= (h_w A_w + G_w b_g) \phi_1 - (h_w A_w + G_w b_w) T_w$$

$$q_r = q_{r,i-w}$$

$$q_g = \dot{m} C_p (\phi_1 - \phi_2)$$

Find expressions for q_r and q_l in terms

of $\phi_1, T_{C2,1}, T_\infty$

$$q_k = q_r + q$$

$$q_r = U(a_1 T_w - a_2 T_{C2,1} - a_3 \phi_1)$$

$$U = (1/(St_2) + 1/(K + h_i A_i + G_L b_2))^{-1}$$

$$a_1 = t_w/t_2$$

$$a_2 = K/(K + h_i A_i + G_i b_2)$$

$$a_3 = (h_i A_i + G_i b_g) / (K + h_i A_i + G_i b_2)$$

$$q_1 = q_w - q_r$$

$$q_1 = \text{LOSS} (l_1 \phi_1 + l_2 T_{C_{2,1}} - T_\infty)$$

$$\text{LOSS} = (1/L + 1/(h_w A_w + G_w b_w + U_1 a_1))^{-1}$$

$$l_1 = (h_w A_w + G_w b_g + U_1 a_3) / (h_w A_w + G_w b_w + U_1 a_1)$$

$$l_2 = U_1 a_2 / (h_w A_w + G_w b_w + U_1 a_1)$$

Find an expression for T_w in terms of ϕ , $T_{C_{1,2}}$, T_∞

$$q_1 = L(T_w - T_\infty)$$

$$T_w = \frac{\text{LOSS}}{L} (l_1 \phi_1 + l_2 T_{C_{2,1}}) + (1 - \text{LOSS}/L) T_\infty$$

Substitute in to eliminate T_w

$$q_k = \text{UP} (u_1 \phi_1 + u_2 T_{C_{2,1}} + u_3 T_\infty)$$

$$\text{UP} = (1/K + 1/(h_i A_i + G_i b_2 + St_2))^{-1}$$

$$u_1 = \frac{(h_i A_L + G_i b_g + St_w \cdot \text{LOSS} \cdot 1/L)}{(h_i A_L + G_i b_2 + St_2)}$$

$$u_2 = \frac{S \cdot t_w \cdot \text{LOSS} \cdot 1/L}{(h_i A_i + G_L b_2 + St_2)} - 1.0$$

$$u_3 = St_w (1 - \text{LOSS}/L) / (h_i A_i + G_i b_2 + St_2)$$

Find an expression for $T_{h1,2}$

$$q + q_r - q_k = 0$$

$$T_{h1,2} = Z_1 \phi_1 + Z_2 T_{C2,1} + Z_3 T_w$$

$$Z_1 = (h_i A_i + G_i b_g) / (h_i A_i + G_i b_2 + St_2 + K)$$

$$Z_2 = K / (h_i A_i + G_i b_2 + St_2 + K)$$

$$Z_3 = St_w / (h_i A_i + G_i b_2 + St_2 + K)$$

COLD OUTLET CONTROL VOLUME

$$q_{V,i} = h_i A_i (T_{C1,2} - \theta_1)$$

$$q_k = K(T_{h1,2} - T_{C1,2})$$

$$q_r = S(T_{C1,2} - T_w)$$

$$q_{V,w} = h_w A_w (T_w - \theta_1)$$

$$q_l = L(T_w - T_\infty)$$

$$q_a = \dot{m} c_p (\theta_{exit} - \theta_1)$$

Eliminate $T_{c1,2}$

$$q_r = q_k - q_{v,i}$$

$$q_r = U(a_1 \theta_1 + a_2 T_{h2,1} - T_w)$$

$$U = (1/s + 1/(K + h_i A_i))^{-1}$$

$$a_1 = h_i A_i / (K + h_i A_i)$$

$$a_2 = K / (K + h_i A_i)$$

$$q_l = q_r - q_{v,i}$$

$$q_l = \text{LOSS} (l_1 \theta_1 + l_2 T_{h2,1} - T_\infty)$$

$$\text{LOSS} = (1/L + 1/(U + h_i A_i))^{-1}$$

$$l_1 = (U a_1 + h_i A_i) / (U + h_i A_i)$$

$$l_2 = U a_2 / (U + h_i A_i)$$

Find an expression for T_w

$$q_l = L (T_w - T_\infty)$$

$$T_w = \frac{\text{LOSS}}{L} (l_1 \theta_1 + l_2 T_{h2,1}) + (1 - \text{LOSS}/L) T_\infty$$

Find q_k in terms of θ_i , $T_{h_2,1}$, T_∞

$$q_k = q_r + q$$

$$q_k = UP (-u_1 \theta_1 - u_2 T_{h_2,1} + u_3 T_\infty)$$

$$UP = (1/K + 1/(h_i A_i + S))^{-1}$$

$$U_1 = (h_i A_i + S) \text{ LOSS}/L / (h_i A_i + S)$$

$$U_2 = (S \cdot \text{LOSS} \cdot 1_2/L) / (h_i A_i + S) - 1$$

$$U_3 = (1 - \text{LOSS}/L) / (h_i A_i + S)$$

Find expression for $T_{c1,2}$

$$q_k = q_r + q$$

$$T_{c1,2} = (h_i A_i \theta_1 + K T_{h_2,1} + S \cdot T_w) / (h_i A_i + S + K)$$

$$q_2 = F_1 T_{hn,1} + F_2 \phi_n + F_3 T_{cn+1,1} + F_4 T_\infty$$

$$F_1 = E_{h,2} (-R t_1 - S_2 t_w \cdot BONE \cdot B_2)$$

$$F_2 = E_{h,2} [(h_i A_i + G_{i,2} b_g)(K + R t_2 + S_2 t_w) / (h_i A_i + G_{i,2} b_g) - S_2 t_w \cdot BONE \cdot B_1]$$

$$F_3 = E_{h,2} (-K - S_2 t_w \cdot BONE \cdot B_3)$$

$$F_4 = E_{h,2} (-S_2 t_w \cdot BONE \cdot B_4)$$

$$q_w = G_1 T_{hn,1} + G_2 \phi_n + G_3 T_{cn+1,1} + G_4 T_\infty$$

$$G_1 = - (h_w A_w + G_w b_w) \cdot BONE \cdot B_2$$

$$G_2 = (h_w A_w + G_w b_g) - (h_w A_w + G_w b_w) \cdot BONE \cdot B_1$$

$$G_3 = - (h_w A_w + G_w b_w) \cdot BONE \cdot B_3$$

$$G_4 = - (h_w A_w + G_w b_w) \cdot BONE \cdot B_4$$

$$q_{r,1} = H_1 T_{hn,1} + H_2 \phi_n + H_3 T_{cn+1,1} + H_4 T_\infty$$

$$H_1 = -S_1 t_1 + S_1 t_w \cdot BONE \cdot B_2$$

$$H_2 = S_1 t_w \cdot BONE \cdot B_1$$

$$H_3 = S_1 t_w \cdot BONE \cdot B_3$$

$$H_4 = S_1 t_w \cdot BONE \cdot B_4$$

$$q_1 = I_1 T_{hn,1} + I_2 \phi_n$$

$$I_1 = - (h_i A_i + G_{i,1} b_i)$$

$$I_2 = h_i A_i + G_{i,1} b_g$$

Find an expression for $T_{hn,2}$

$$q_r + q_2 + q_{r,2} = q_k$$

$$T_{hn,2} = [R t_1 T_{hn,1} + (h_i A_i + G_{i,2} b_g) \phi_n + S_2 t_w T_w + K T_{cn+1,1}] / (R t_2 + h_i A_i + G_{i,2} b_2 + S_2 t_2 + K)$$

COLD INTERIOR SECTION

$$q_R = K (T_{hn+1,1} + T_{cn,2})$$

$$q_r = R (T_{cn,1} - T_{cn,2})$$

$$q_{v,1} = h_i A_i (T_{cn,1} - \theta_n)$$

$$q_{v,2} = h_i A_i (T_{cn,2} - \theta_n)$$

$$q_{v,w} = h_w A_w (T_w - \theta_n)$$

$$q_{r,2} = S_2 (T_{cn,2} - T_w)$$

$$q_{r,1} = S_1 (T_{cn,1} - T_w)$$

$$q_l = L (T_w - T_\infty)$$

$$q_r = E_R [(S_2 + h_i A_i + K) T_{cn,1} - S_2 T_w - h_i A_i \theta_n - K T_{hn+1,1}]$$

$$E_R = [(S_2 + h_i A_i + K) / R + 1]^{-1}$$

$$q_{r,2} = E_{r,2} [R T_{cn,1} + K T_{hn+1,1} + h_i A_i \theta_n - (R + K + h_i A_i) T_w]$$

$$E_{r,2} = [(R + K + h_i A_i) / S_2 + 1]^{-1}$$

Find an expression for T_w

$$q_{r,1} + q_{r,2} - q_{v,w} - q_l = 0$$

$$I_w = BONE (B_1 T_{cn,1} + B_2 \theta_n + B_3 T_{hn+1,1} + B_4 T_\infty)$$

$$BONE = [S_1 + E_{r,2} (R + K + h_i A_i) + h_w A_w + L]^{-1}$$

$$B_1 = S_1 + E_{r,2} R$$

$$B_2 = h_w A_w + E_{r,2} h_i A_i$$

$$B_3 = E_{r,2} K$$

$$B_4 = L$$

Plug into to eliminate T_w

$$q_r = d_1 T_{cn,1} + d_2 \theta_n + d_3 T_{hn+1,1} + d_4 T_\infty$$

$$d_1 = E_R (S_2 + h_i A_i + K) - E_R S_2 \cdot BONE \cdot B_1$$

$$d_2 = -E_R (h_i A_i + S_2 \cdot BONE \cdot B_2)$$

$$d_3 = -E_R (K + S_2 \cdot BONE \cdot B_3)$$

$$d_4 = -E_R S_2 \cdot BONE \cdot B_4$$

$$q_{r,2} = e_1 T_{cn,1} + e_2 \theta_n + e_3 T_{hn+1,1} + e_4 T_\infty$$

$$e_1 = E_{r,2} [R - (R + K + h_i A_i) \cdot BONE \cdot B_1]$$

$$e_2 = E_{r,2} [h_i A_L - (R + K + h_i A_i) \cdot BONE \cdot B_2]$$

$$e_3 = E_{r,2} [K - (R + K + h_i A_i) \cdot BONE \cdot B_3]$$

$$e_4 = - E_{r,2} (R + K + h_i A_i) \cdot BONE \cdot B_4$$

$$q_{v,2} = f_1 T_{cn,1} + f_2 \theta_n + f_3 T_{hn+1,1} + f_4 T_\infty$$

$$f_1 = E_{h,2} (R + S_2 \cdot BONE \cdot B_1)$$

$$f_2 = E_{h,2} [-(R + K + S_2) + S_2 \cdot BONE \cdot B_2]$$

$$f_3 = E_{h,2} (K + S_2 \cdot BONE \cdot B_3)$$

$$f_4 = E_{h,2} \cdot BONE \cdot B_4 S_2$$

$$q_{v,w} = g_1 T_{cn,1} + g_2 \theta_n + g_3 T_{hn+1,1} + g_4 T_\infty$$

$$g_1 = h_w A_w \cdot BONE \cdot B_1$$

$$g_2 = - h_w A_w + h_w A_w \cdot BONE \cdot B_2$$

$$g_3 = h_w A_w \cdot BONE \cdot B_3$$

$$g_4 = h_w A_w \cdot BONE \cdot B_4$$

$$q_{r,1} = h_{c1} T_{cn,1} + h_{c2} \theta_n + h_{c3} T_{hn+1,1} + h_{c4} T_{\infty}$$

$$h_{c1} = S_1 (1 - \text{BONE} \cdot B_1)$$

$$h_{c2} = -S_1 \cdot \text{BONE} \cdot B_2$$

$$h_{c3} = -S_1 \cdot \text{BONE} \cdot B_3$$

$$h_{c4} = -S_1 \cdot \text{BONE} \cdot B_4$$

$$q_{r,1} = i_1 T_{cn,1} - i_1 \theta_n$$

$$i_1 = h_i A_i$$

Find expression for $T_{cn,2}$

$$q_r + q_k = q_{r,2} + q_{v,2}$$

$$T_{cn,2} = (R T_{cn,1} + K T_{hn+1,1} + S_2 T_w + h_i A_i \theta_n) / (R + S_2 + K + h_i A_i)$$

HOT EXIT SECTION

$$q_{r,g-i} = G_i (b_g \phi_m - b_l T_{hm,1})$$

$$b_g = \epsilon_g^{\frac{1}{4}}$$

$$b_l = \epsilon_i^{\frac{1}{4}}$$

$$q_{v,i} = h_i A_i (\phi_m - T_{hm,1})$$

$$q_{r,g-w} = G_w (b_g \phi_m - b_w T_w)$$

$$b_w = \epsilon_w^{\frac{1}{4}}$$

$$q_{v,w} = h_w A_w (\phi_m - T_w)$$

$$q_{r,w-i} = S_l (t_w T_w - t_l T_{hm,1})$$

$$t_w = (1 - \epsilon_w)^{\frac{1}{4}}$$

$$t_l = (1 - \epsilon_i)^{\frac{1}{4}}$$

$$q_\ell = L (T_w - T_\infty)$$

$$\dot{m}_g = \dot{m} C_p (\phi_{\text{exit}} - \phi_m)$$

q_n = heat transfer to lower control volume

Reduce to the following form

$$q = (G_i b_g + h_i A_i) \phi_m - (G_i b_l + h_i A_i) T_{hm,1}$$

$$q_w = (G_w b_g + h_w A_w) \phi_m - (G_w b_w + h_w A_w) T_w$$

$$q_r = S_1 (t_w T_w - t_1 T_{hm,1})$$

Eliminate I_w

$$q_r = q_w - q_l$$

$$q_r = UPH (z_1 \phi_m - z_2 T_{hm,1} + z_3 T_\infty)$$

$$UPH = [(G_w b_w + h_w A_w + L)/(S_1 t_w) + 1]^{-1}$$

$$z_1 = G_w b_g + h_w A_w$$

$$z_2 = (G_w b_w + h_w A_w + L) t_1 / t_w$$

$$z_3 = L$$

$$q_n = q_r + q$$

$$q_n = ENH_1 \phi_m + ENH_2 T_{hm,1} + ENH_3 T_\infty$$

$$ENH_1 = UPH \cdot z_1 + (G_i b_g + h_i A_i)$$

$$ENH_2 = - (UPH \cdot z_2 + G_i b_1 + h_i A_i)$$

$$ENH_3 = UPH \cdot z_3$$

$$q_w = q_\lambda + q_r$$

$$q_w = [(G_w b_g + h_w A_w) \phi_m - (G_w b_w + h_w A_w) S_1 t_1 T_{hm,1} \\ / (L + S_1 t_w) - L (G_w b_w + h_w A_w) T_\infty / (L + S_1 t_w)] \\ / [(G_w b_w + h_w A_w) / (L + S_1 t_w) + 1]$$

$$q_g = q_w + q$$

$$q_g = EOUT_1 + EOUT_2 T_{hm,1} + EOUT_3 T_\infty$$

$$EOUT_1 = W (L + S_1 t_w) (G_w b_g + h_w A_w) + G_i b_g + h_i A_i$$

$$EOUT_2 = - W (G_w b_w + h_w A_w) S_1 t_1 - (G_i b_1 + h_i A_i)$$

$$EOUT_3 = - W (G_w b_w + h_w A_w)$$

$$W = (G_w b_w + h_w A_w + L + S_1 t_w)^{-1}$$

Find an expression for t_w

$$q_w = q_\lambda + q_r$$

$$T_w = [(G_w b_g + h_w A_w) \phi_m + S t_1 T_{hm,1} + L \cdot T_\infty] \\ / (G_w b_w + h_w A_w + L + S \cdot t_w)$$

COLD INLET SECTION

$$q_{v,i} = h_i A_i (T_{cm,1} - \theta_m)$$

$$q_{v,w} = h_w A_w (T_w - \theta_m)$$

$$q_r = S (T_{cm,1} - T_w)$$

$$q_l = L (T_w - T_\infty)$$

$$q_n = \text{HEAT TRANSFER FROM NEXT SECTION}$$

Eliminate T_w

$$q_r = q_{v,w} + q_l$$

$$q_r = [(h_w A_w + L) T_{cm,1} - h_w A_w \theta_m - L T_\infty] / [(h_w A_w + L) / S + 1]$$

$$q_n = q_{v,i} + q_r$$

$$q_n = ENC_1 \theta_m + ENC_2 T_{cm,1} + ENC_3 T_\infty$$

$$ENC_1 = -h_i A_i - (h_w A_w) / [(h_w A_w + L) / S + 1]$$

$$ENC_2 = h_i A_i + (h_w A_w + L) / [(h_w A_w + L) / S + 1]$$

$$ENC_3 = -L / [(h_w A_w + L) / S + 1]$$

$$q_{v,w} = q_r - q_\ell$$

$$q_{v,w} = [-(S + L) \theta_m + S \cdot T_{cm,1} + L \cdot T_\infty] / [(S + L) / (h_w A_w) + 1]$$

$$q_a = q_{v,w} + q_{v,i}$$

$$q_a = E1N_1 \theta_m + E1N_2 T_{cm,1} + E1N_3 T_\infty$$

$$E1N_1 = -h_i A_i - (S + L) / [(S + L) / (h_w A_w) + 1]$$

$$E1N_2 = h_i A_i + S / [(S + L) / (h_w A_w) + 1]$$

$$E1N_3 = L / [(S + L) / (h_w A_w) + 1]$$

Find expression for T_w

$$T_w = (h_w A_w \theta_m + L \cdot T_\infty + S \cdot T_{cm,1}) / (h_w A_w + L + S)$$

EQUATION SUMMARY

BOTTOM SECTIONS

HOT

$$q_{\ell} = LH_1 \phi_1 + LH_2 T_{c2,1} + LH_3 T_{\infty}$$

$$q_K = KH_1 \phi_1 + KH_2 T_{c2,1} + KH_3 T_{\infty}$$

$$LH_1 = \text{LOSS} \cdot \ell_1$$

$$LH_2 = \text{LOSS} \cdot \ell_2$$

$$LH_3 = -\text{LOSS}$$

$$KH_1 = UP \cdot u_1$$

$$KH_2 = UP \cdot u_2$$

$$KH_3 = UP \cdot u_3$$

COLD

$$q_{\ell} = LC_1 \theta_1 + LC_2 T_{h2,1} + LC_3 T_{\infty}$$

$$q_K = KC_1 \theta_1 + KC_2 T_{h2,1} + KC_3 T_{\infty}$$

$$LC_1 = \text{LOSS} \cdot \ell_1$$

$$LC_2 = \text{LOSS} \cdot \ell_2$$

$$LC_3 = -\text{LOSS}$$

$$KC_1 = -UP \cdot u_1$$

$$KC_2 = -UP \cdot u_2$$

$$KC_3 = UP \cdot u_3$$

INTERIOR SECTIONS

HOT

$$q_r = D_1 T_{hn,1} + D_2 \phi_n + D_3 T_{cn+1,1} + D_4 T_\infty$$

$$q_{r,2} = E_1 T_{hn,1} + E_2 \phi_n + E_3 T_{cn+1,1} + E_4 T_\infty$$

$$q_2 = F_1 T_{hn,1} + F_2 \phi_n + F_3 T_{cn+1,1} + F_4 T_\infty$$

$$q_w = G_1 T_{hn,1} + G_2 \phi_n + G_3 T_{cn+1,1} + G_4 T_\infty$$

$$q_{r,1} = H_1 T_{hn,1} + H_2 \phi_n + H_3 T_{cn+1,1} + H_4 T_\infty$$

$$q_1 = I_1 T_{hn,1} + I_2 \phi_n$$

COLD

$$q_r = d_1 T_{cn,1} + d_2 \theta_n + d_3 T_{hn+1,1} + d_4 T_\infty$$

$$q_{r,2} = e_1 T_{cn,1} + e_2 \theta_n + e_3 T_{hn+1,1} + e_4 T_\infty$$

$$q_2 = f_1 T_{cn,1} + f_2 \theta_n + f_3 T_{hn+1,1} + f_4 T_\infty$$

$$q_w = g_1 T_{cn,1} + g_2 \theta_n + g_3 T_{hn+1,1} + g_4 T_\infty$$

$$q_{r,1} = h_{c1} T_{cn,1} + h_{c2} \theta_n + h_{c3} T_{hn+1,1} + h_{c4} T_\infty$$

$$q_1 = i T_{cn,1} - w_i \theta_n$$

TOP SECTIONS

HOT

$$q_n = ENH_1 \phi_m + ENH_2 T_{hm,1} + ENH_3 T_\infty$$

$$q_g = EOUT_1 \phi_m + EOUT_2 T_{hm,1} + EOUT_3 T_\infty$$

COLD

$$q_n = ENC_1 \theta_m + ENC_2 T_{cm,1} + ENC_3 T_\infty$$

$$q_a = EIN_1 \theta_m + EIN_2 T_{cm,1} + EIN_3 T_\infty$$

Example of Nodal Equations

Case of two full turns of helix

J = Mass flow rate times heat capacity

Node 1, see Figure A1.25.

$$Q_c = q_k - q_\ell$$

$$Q_c = (KC_1 - LC_1) \theta_1 + (KC_2 - LC_2) T_{h2,1} + (KC_3 - LC_3) T_\infty$$

Node 2

$$q_k + q_r - q_1 - q_{r,1} = 0$$

$$KC_1 \theta_1 + (KC_2 + D_1 - I_1 - H_1) T_{h2,1} + (D_2 - H_2 - I_2) \phi_2$$

$$+ (D_3 - H_3) T_{c3} + (KC_3 + D_4 - H_4) T_\infty = 0$$

Node 3

$$-J (\phi_2 - \phi_3) + q_w + q_1 + q_2 = 0$$

$$T_{h2} (G_1 + I_1 + F_1) + \phi_2 (-J + G_2 + I_2 + F_2) + T_{c3} (G_3 + F_3) \\ + T_{\infty} (G_4 + F_4) + J \phi_3 = 0$$

Node 4

$$T_{h2} (D_1 + F_1 + E_1) + \phi_2 (D_2 + F_2 + E_2) + T_{c3} (D_3 + E_3 \\ + F_3 - d_1 - h_{c1} - i) + \theta_3 (-d_2 - h_{c2} + i) + T_{h4} \\ (-d_3 - h_{c3}) + (D_4 + E_4 + F_4 - d_4 - h_{c4}) T_{\infty} = 0$$

Node 5

$$T_{c3} (i + f_1 + g_1) + \theta_3 (J - i + f_2 + g_2) + T_{h4} (f_3 + g_3) \\ - J \theta_2 + (f_4 + g_4) T_{\infty} = 0$$

Node 6

$$T_{c3} (-d_1 + f_1 + e_1) + \theta_3 (-d_2 + f_2 + e_2) + T_{h4} (-d_3 + c_3 \\ + f_3 + D_1 - I_1 - H_1) + \phi_4 (D_2 - H_2 - I_2) + T_{c5} \\ (D_3 - H_3) = 0$$

Node 7

$$T_{h4} (G_1 + I_1 + F_1) + \phi_4 (-J + G_2 + I_2 + F_2) + T_{c5} (G_3 + F_3) + T_{\infty} (G_4 + F_4) + J \phi_5 = 0$$

Node 8

$$T_{h4} (D_1 + F_1 + E_1) + \phi_4 (D_2 + F_2 + E_2) + T_{c5} (D_3 + F_3 - E_3 - ENC_2) - FNC_1 \theta_5 + (D_4 + E_4 + F_4 - ENC_3) T_{\infty} = 0$$

Node 9

$$T_{c5} (EIN_2) + \phi_5 (EIN_1 + J) + EIN_3 T_{\infty} - J \theta_4 = 0$$

Similar equations for nodes 10 through 18

NODAL EQUATION SCHEMATIC
CASE OF TWO FULL TURNS OF HELIX

ϕ_1 = Hot Inlet

θ_5 = Cold Inlet

FIGURE A1.25. Schematic of Nodal Equations

REFERENCES

1. F. Kreith, Principles of Heat Transfer, 3rd Edition, International Text Book Company, Scranton, Pa., 1973.
2. E. M. Sparrow and R. D. Cess, Radiation Heat Transfer, Wadsworth Publishing Co., New York, 1966.

APPENDIX 2
COLD/WARM FLOW TEST DATA

TABLE A2.1

COLD FLOW TEST RESULTS

INLET AIR FLOW (lbm/min)	RECUPERATOR, AIR PRESSURE (inH ₂ O)		GAS SIDE PRESSURE (in. H ₂ O)	LEAKAGE		
	Pave	ΔP		Total (lbm/min)	%	Interpath (lbm/min)
9.4	5.36	4.53	.43	.25	3	2.9×10^{-4}
8.3	4.07	3.54	.32	.13	2	2.3×10^{-4}
7.35	3.24	2.83	.23	.05	1	1.5×10^{-4}
6.21	3.02	2.56	.23	.01	.2	1.3×10^{-4}
4.50	1.30	1.09	.08	0	0	0
8.58	6.43	3.80	.58	.23	3	3.6×10^{-4}
7.30	7.7	2.68	.70	.35	5	4.9×10^{-4}
2.31	2.52	.24	.12	.17	7	8.8×10^{-4}
1.84	1.65	.15	.08	.13	7	1.7×10^{-5}
1.36	.90	.06	.02	.07	5	0
1.04	.49	.04	.02	.03	5	0
.71	.11	.02	.01	.02	3	0
.495	1.17	.01	.03	.15	30	0
.425	.92	.02	.02	.13	29	0
.35	.64	.01	0	.09	24	0
.185	.15	.01	0	.025	13	0
.345	.60	.01	0	.08	23	0
.335	.62	.01	0	.07	21	0
.485	1.27	.01	0	.125	26	0
.325	.72	.01	0	.055	17	0
.19	.21	.01	0	.015	8	0
.185	.20	.01	0	.02	11	0
.330	.71	.01	0	.05	15	0

APPENDIX 3
HIGH TEMPERATURE TEST DATA

TABLE A3.1. AMBIENT TEMPERATURE LEAKAGE TEST RESULTS
HIGH TEMPERATURE TEST ASSEMBLY

Air Inlet Flow Rate (lbm/min)	Air Outlet Flow Rate (lbm/min)	Leakage (lbm/min)	Average Pressure (in H ₂ O)	Average Temp. (°F)	Ave. Density (lbm/ft ³)
9.45	8.65	0.80	6.11	70°	0.064
9.20	8.15	1.05	5.77	70°	0.064
8.10	7.35	0.75	4.68	70°	0.064
6.40	6.15	0.25	3.02	70°	0.064
4.55	4.15	0.40	1.72	70°	0.064
1.81	1.53	0.28	1.32	70°	0.064
2.04	1.70	0.34	1.60	70°	0.064
1.19	1.14	0.05	0.66	70°	0.064
0.59	0.35	0.24	1.18	70°	0.064
0.49	0.31	0.18	0.82	70°	0.064
0.28	0.22	0.06	0.41	70°	0.064
0.175	0.11	0.07	0.16	70°	0.064

TABLE A3.2. LEAKAGE DURING HIGH TEMPERATURE HEAT TRANSFER TESTS

Run No.	Air Inlet Flow Rate (lbm/min)	Air Outlet Flow Rate (lbm/min)	Leakage (lbm/min)	Average Pressure (in H ₂ O)	Average Temp. (°F)	Ave. Density (lbm/ft ³)
1	0.26	0.219	0.041	0.46	366	0.041
2	1.17	1.06	0.110	0.84	302	0.045
3	2.17	1.86	0.31	2.46	224	0.050
4	3.22	3.05	0.17	1.06	158	0.055
5	6.33	5.99	0.34	3.46	129	0.058
6	0.25	0.19	0.06	0.35	341	0.042
7	1.20	1.12	0.08	0.87	305	0.044
8	2.04	1.83	0.21	2.39	266	0.047
9	3.53	3.27	0.26	1.35	231	0.049
10	6.52	5.93	0.59	3.97	180	0.053
11	0.28	0.23	0.05	0.47	608	0.032
12	1.34	1.22	0.12	1.25	504	0.035
13	2.60	2.51	0.09	0.96	346	0.042
14	3.34	3.39	0.05	1.60	291	0.046
15	6.94	6.52	0.42	5.01	207	0.051
16	0.33	0.24	0.09	0.59	595	0.032
17	1.47	1.29	0.18	1.53	484	0.036
18	3.14	3.02	0.12	1.56	365	0.041
19	3.75	3.56	0.19	2.13	323	0.043
20	7.49	6.95	0.54	6.62	239	0.049
21	0.50	0.36	0.14	1.45	902	0.025
22	1.91	1.73	0.18	2.94	538	0.034

Table A3.2 Cont.

Run No.	Air Inlet Flow Rate (lbm/min)	Air Outlet Flow Rate (lbm/min)	Leakage (lbm/min)	Average Pressure (in H ₂ O)	Average Temp. (°F)	Ave. Density (lbm/ft ³)
22A	1.70	1.42	0.28	2.11	707	0.029
22-32	1.75	1.46	0.29	2.31	783	0.027
23	2.87	2.68	0.19	1.39	521	0.035
24	3.95	3.64	0.31	2.40	477	0.036
25	4.96	4.47	0.49	3.41	393	0.040
26	0.91	0.84	0.07	0.71	890	0.025
27	1.75	1.55	0.20	2.57	783	0.027
28	3.65	3.51	0.14	2.38	580	0.033
29	4.61	4.38	0.23	3.50	533	0.034
29-30	5.41	4.99	0.42	4.64	499	0.035
30	6.95	5.91	1.04	6.16	500	0.035
32	2.28	2.14	0.14	1.10	746	0.028
32A	1.73	1.45	0.28	2.41	887	0.025
33	2.87	2.75	0.12	1.59	704	0.029
33A	2.60	2.77	-0.17	1.62	733	0.029
33B	3.43	3.30	0.13	2.50	715	0.029
34	2.94	2.53	0.41	1.43	673	0.030
37	1.90	1.89	0.01	0.81	916	0.025
37-38	2.90	2.75	0.15	1.78	812	0.027
38	2.94	2.79	0.15	1.81	831	0.026
39	4.24	4.13	0.11	3.61	659	0.030

Table A3.2 Cont.

Run No.	Air Inlet Flow Rate (lbm/min)	Air Outlet Flow Rate (lbm/min)	Leakage (lbm/min)	Average Pressure (in H ₂ O)	Average Temp. (°F)	Ave. Density (lbm/ft ³)
47	2.11	1.99	0.12	1.03	1025	0.023
47-48	3.07	2.91	0.16	2.00	876	0.025
48	3.49	3.29	0.20	2.59	781	0.027

TABLE A3.3. PRESSURE DROPS DURING HIGH TEMPERATURE HEAT TRANSFER TESTS

TEST NO.	1	2	3	4	5	6	7	8	9
Re_{air}	896	4,413	8,675	14,684	29,829	863	4,617	8,028	14,669
Re_{gas}	3,809	3,680	4,568	5,117	5,157	13,758	14,933	15,524	15,916
$\dot{M}_{air}$ (lb/min)	.24	1.12	2.02	3.14	6.16	.224	1.16	1.94	3.4
$\dot{M}_{gas}$ (lb/min)	1.13	1.04	1.24	1.28	1.29	4.04	4.2	4.27	4.35
$T_{air_{in}}$	127	105	95	91	84	138	132	111	105
$T_{air_{out}}$	605	498	352	224	173	543	478	420	356
$T_{air_{avg}}$	366	302	224	158	129	341	305	266	235
$T_{g_{in}}$	625	637	576	470	493	557	517	512	529
$T_{g_{out}}$	373	235	195	142	113	443	369	317	284
$T_{g_{avg}}$	499	436	386	306	303	500	443	415	407
$P_{air_{in}}$ (in H ₂ O)	.50	.95	2.70	1.34	4.34	.38	.93	2.52	1.66
$P_{air_{out}}$ (in H ₂ O)	.42	.73	2.22	.78	2.58	.31	.80	2.25	1.03
ΔP_{air}	.08	.22	.48	.56	1.76	.07	.13	.27	.63
$P_{g_{in}}$ (in H ₂ O)	0	0	0	0	-.01	-.24	-.24	-.26	-.21
$P_{g_{out}}$ (in H ₂ O)	-.02	-.02	-.02	-.02	-.03	-1.30	-1.28	-1.30	-1.33
ΔP_g	.02	.02	.02	.02	.02	1.06	1.04	1.04	1.12
P_{air}	.0409	.044	.044	.055	.057	.042	.044	.046	.049
P_g	.035	.038	.040	.044	.044	.035	.037	.039	.039
V_{air}	1.52	6.56	6.09	14.90	27.87	1.38	6.82	10.82	18.15
V_{gas}	8.32	7.16	8.06	7.53	7.56	29.79	29.13	28.70	28.97
f_{air}	3.12	.427	.496	.17	.146	3.23	.234	.183	.146
f_{gas}	.030	.038	.029	.030	.029	.126	.121	.121	.127

Table A3.3 Cont.

TEST NO.	10	11	12	13	14	15	16	17	18
Re_{air}	28,436	816	4,349	9,825	13,581	30,080	913	4,756	11,652
Re_{gas}	16,985	4,921	5,646	5,996	5,769	5,899	9,567	10,163	9,787
$\dot{M}_{air}$ (lb/min)	6.23	.255	1.28	2.56	3.37	6.73	.283	1.38	3.08
$\dot{M}_{gas}$ (lb/min)	4.5	1.77	1.91	1.94	1.85	1.86	3.48	3.57	3.33
$T_{air_{in}}$	100	178	131	586	101	87	217	95	64
$T_{air_{out}}$	260	1,037	877	105	481	326	973	873	665
$T_{air_{avg}}$	180	608	504	346	291	207	595	484	365
$T_{g_{in}}$	506	1,094	1,030	1,011	1,035	1,062	993	972	1,063
$T_{g_{out}}$	230	624	438	307	258	180	772	648	427
$T_{g_{avg}}$	368	859	734	659	647	621	883	810	745
$P_{air_{in}}$ (in H ₂ O)	4.85	.50	1.35	1.20	1.96	6.06	.64	1.66	1.89
$P_{air_{out}}$ (in H ₂ O)	3.09	.44	1.15	.71	1.23	3.96	.54	1.39	1.22
ΔP_{air}	1.76	.06	.20	.49	.73	2.1	.10	.27	.67
$P_{g_{in}}$ (in H ₂ O)	-.24	-.02	-.05	-.05	-.08	-.14	-.32	-.32	-.16
$P_{g_{out}}$ (in H ₂ O)	-1.33	-.22	-.35	-.26	-.28	-.28	-1.86	-1.75	-1.12
ΔP_g	1.09	.20	.30	.21	.20	.14	1.54	1.43	.96
P_{air}	.053	.032	.035	.042	.045	.051	.032	.036	.041
P_g	.041	.026	.028	.030	.030	.031	.025	.027	.028
V_{air}	30.63	2.09	9.48	15.85	19.44	34.48	2.29	10.01	19.52
V_{gas}	28.62	17.93	17.52	16.67	15.73	15.44	35.90	34.82	30.82
f_{air}	.121	1.604	.235	.172	.159	.129	2.197	.279	.159
f_{gas}	.132	.090	.128	.093	.098	.070	.176	.164	.133

Table A3.3 Cont.

TEST NO.	19	20	21	22	22A	22-32	23	24	25
Re_{air}	14,342	30,844	1,166	6,063	4,692	4,631	9,336	13,165	15,442
Re_{gas}	9,506	10,401	4,352	4,608	4,942	4,827	5,223	5,498	5,484
$\dot{M}_{air}$ (lb/min)	3.66	7.22	.43	1.82	1.56	1.61	2.78	3.80	4.72
$\dot{M}_{gas}$ (lb/min)	3.12	3.33	1.86	1.73	1.99	2.03	2.02	2.09	2.06
$T_{air_{in}}$	74	82	268	68	88	84	94	98	94
$T_{air_{out}}$	573	390	1,535	1,009	1,326	1,181	947	856	692
$T_{air_{avg}}$	324	239	902	539	707	783	521	477	393
$T_{g_{in}}$	992	995	1,590	1,461	1,566	1,732	1,580	1,551	1,507
$T_{g_{out}}$	369	290	940	434	664	715	456	407	398
$T_{g_{avg}}$	681	643	1,265	948	1,115	1,224	1,018	979	953
$P_{air_{in}}$ (in H ₂ O)	2.66	8.08	1.5	3.14	2.30	2.52	1.78	3.03	4.26
$P_{air_{out}}$ (in H ₂ O)	1.60	5.16	1.40	2.74	1.92	2.10	.99	1.76	2.56
ΔP_{air}	1.06	2.92	.10	.4	.38	.42	.79	1.27	1.70
$P_{g_{in}}$ (in H ₂ O)	-.18	-.18	.06	-.04	.04	.05	.01	.02	.02
$P_{g_{out}}$ (in H ₂ O)	-1.02	-1.00	-.42	-.34	-.40	-.46	-.40	-.39	-.38
ΔP_g	.84	.82	.48	.30	.44	.51	.41	.41	.40
P_{air}	.043	.048	.025	.034	.029	.027	.034	.036	.040
P_g	.030	.031	.020	.024	.021	.020	.023	.023	.024
V_{air}	22.04	38.76	4.50	13.97	13.98	15.37	20.95	27.35	30.92
V_{gas}	27.34	28.21	24.64	18.71	24.07	26.26	22.93	23.10	22.36
f_{air}	.187	.149	.737	.224	.248	.242	.193	.174	.166
f_{gas}	.140	.124	.149	.132	.131	.136	.126	.121	.124

Table A3.3 Cont.

TEST NO.	26	27	28	29	29-30	30	32	33B	33A
Re_{air}	2,399	4,761	11,649	15,019	17,708	21,879	6,479	10,028	7,954
Re_{gas}	7,304	7,870	9,135	9,382	10,279	10,529	5,088	7,779	6,301
$\dot{M}_{air}$ (lb/min)	.88	1.65	3.58	4.50	5.2	6.43	2.21	3.37	2.69
$\dot{M}_{gas}$ (lb/min)	3.21	3.40	3.69	3.81	4.03	4.17	2.19	3.42	2.76
$T_{air_{in}}$	226	121	91	82	83	97	91	94	82
$T_{air_{out}}$	1,554	1,444	1,069	984	914	902	1,401	1,336	1,385
$T_{air_{avg}}$	890	783	580	533	499	500	746	715	734
$T_{g_{in}}$	1,637	1,609	1,566	1,610	1,540	1,589	1,957	1,937	1,996
$T_{g_{out}}$	1,040	977	679	661	561	560	609	741	662
$T_{g_{avg}}$	1,339	1,293	1,123	1,136	1,051	1,075	1,283	1,339	1,329
$P_{air_{in}}$ (in H ₂ O)	.84	2.80	3.08	4.45	5.84	7.70	1.47	3.24	2.14
$P_{air_{out}}$ (in H ₂ O)	.58	2.34	1.68	2.54	3.44	4.62	.73	1.76	1.10
ΔP_{air}	.26	.46	1.40	1.91	2.40	3.08	.74	1.48	1.04
$P_{g_{in}}$ (in H ₂ O)	-.38	-.36	-.40	-.46	-.56	-.54	-.05	-.38	-.12
$P_{g_{out}}$ (in H ₂ O)	-2.32	-2.36	-2.20	-2.28	-2.38	-2.42	-.72	-1.92	-1.17
ΔP_g	1.94	2.0	1.80	1.82	1.82	1.88	.67	1.54	1.05
P_{air}	.025	.027	.032	.034	.035	.035	.028	.029	.028
P_g	.019	.019	.021	.021	.022	.022	.019	.019	.019
V_{air}	9.12	15.75	28.60	34.32	38.30	47.41	20.47	30.41	24.67
V_{gas}	44.36	45.78	44.87	46.71	46.77	49.17	29.32	47.26	37.93
f_{air}	.462	.252	.195	.176	.172	.144	.233	.206	.233
f_{gas}	.194	.183	.155	.146	.138	.131	.149	.136	.143

Table A3.3 Cont.

TEST NO.	32A	33	34	37	37-38	38	39	47	47-48
Re_{air}	4,363	8,450	8,374	5,131	8,034	8,467	12,934	5,286	8,245
Re_{gas}	4,650	6,345	4,590	6,660	6,435	7,109	7,841	6,338	6,685
M_{air} (lb/min)	1.59	2.81	2.74	1.90	2.83	2.86	4.19	2.05	2.99
M_{gas} (lb/min)	2.06	2.73	1.96	3.10	2.88	3.22	3.45	3.09	3.14
$T_{air_{in}}$	86	81	87	112	96	95	85	129	130
$T_{air_{out}}$	1,688	1,327	1,259	1,719	1,527	1,567	1,233	1,920	1,622
$T_{air_{avg}}$	887	704	673	916	812	831	659	1,025	876
$T_{g_{in}}$	1,944	1,921	1,933	1,962	1,955	1,949	1,913	2,217	2,135
$T_{g_{out}}$	776	643	592	1,030	820	891	770	1,089	912
$T_{g_{avg}}$	1,360	1,282	1,263	1,496	1,388	1,420	1,342	1,653	1,524
$P_{air_{in}}$ (in H ₂ O)	2.64	2.09	1.86	1.14	2.32	2.38	4.60	1.42	2.64
$P_{air_{out}}$ (in H ₂ O)	2.18	1.08	1.00	.48	1.24	1.24	2.62	.64	1.36
ΔP_{air}	.46	1.01	.86	.66	1.08	1.14	1.98	.78	1.28
$P_{g_{in}}$ (in H ₂ O)	.06	-.18	.06	-.32	-.18	-.36	-.44	-.34	-.34
$P_{g_{out}}$ (in H ₂ O)	-.44	-1.18	-.38	-2.08	-1.60	-2.08	-2.12	-2.10	-2.15
ΔP_g	.50	1.00	.44	1.76	1.42	1.72	1.68	1.76	1.81
P_{air}	.025	.029	.030	.025	.027	.026	.030	.023	.025
P_g	.019	.019	.020	.017	.018	.018	.019	.016	.017
V_{air}	16.45	25.12	23.84	20.08	27.65	28.36	36.01	23.38	30.68
V_{gas}	28.80	36.53	25.94	46.57	40.88	46.50	47.75	50.15	47.85
f_{air}	.251	.204	.188	.247	.197	.200	.188	.232	.199
f_{gas}	.120	.143	.123	.174	.172	.164	.145	.162	.172

Table A3.3 Cont.

TEST NO. 48

Re _{air}	9,786								
Re _{gas}	7,053								
M _{air} (lb/min)	3.39								
M _{gas} (lb/min)	3.23								
T _{air in}	91								
T _{air out}	1,470								
T _{air avg}	781								
T _{g in}	2,126								
T _{g out}	771								
T _{g avg}	1,449								
P _{air in} (in H ₂ O)	3.38								
P _{air out} (in H ₂ O)	1.80								
ΔP _{air}	1.58								
P _{g in} (in H ₂ O)	-.32								
P _{g out} (in H ₂ O)	-1.94								
ΔP _g	1.62								
P _{air}	.027								
P _g	.018								
V _{air}	32.31								
V _{gas}	47.36								
f _{air}	.206								
f _{gas}	.151								

TABLE A3.3 Cont.

TEST NUMBER	1	2	3	4	5	6	7	8
AIR SIDE STACK DRAFT (in. H ₂ O)	.046	.040	.040	.018	.014	.044	.040	.036
COMBUSTION SIDE STACK DRAFT (in. H ₂ O)	.058	.052	.048	.040	.048	.058	.054	.050
f CORRECTED (air side)	1.33	.352	.997	.164	.146	1.203	.163	.161
f CORRECTED (gas side)	.119	.137	.097	.089	.100	.133	.129	.126

TEST NUMBER	9	10	11	12	13	14	15	16
AIR SIDE STACK DRAFT (in H ₂ O)	.030	.022	.064	.058	.044	.038	.026	.064
COMBUSTION SIDE STACK DRAFT (in H ₂ O)	.050	.046	.075	.072	.068	.068	.066	.077
f CORRECTED (air side)	.138	.129	-.106	.167	.156	.151	.127	.794
f CORRECTED (gas side)	.132	.125	.122	.160	.123	.134	.103	.186

TEST NUMBER	17	18	19	20	21	22	22A	22-23
AIR SIDE STACK DRAFT (in. H ₂ O)	.056	.046	.042	.032	.078	.060	.070	.073
COMBUSTION SIDE STACK DRAFT (in. H ₂ O)	.074	.072	.068	.066	.087	.079	.085	.087
f CORRECTED (air side)	.219	.148	.180	.140	.161	.190	.202	.201
f CORRECTED (gas side)	.170	.144	.150	.133	.173	.167	.160	.160

TABLE A3.3 Cont.

TEST NUMBER	23	24	25	26	27	28	29	29-30
AIR SIDE STACK DRAFT (in. H ₂ O)	.060	.056	.048	.077	.074	.064	.060	.058
COMBUSTION SIDE STACK DRAFT (in. H ₂ O)	.081	.081	.079	.089	.089	.085	.085	.083
f CORRECTED (air side)	.181	.167	.160	.183	.213	.189	.171	.169
f CORRECTED (gas side)	.150	.148	.148	.201	.194	.165	.154	.146

TEST NUMBER	30	32	33B	33A	32A	33	34	37
AIR SIDE STACK DRAFT (in H ₂ O)	.058	.072	.070	.072	.077	.070	.068	.077
COMBUSTION SIDE STACK DRAFT (in H ₂ O)	.083	.089	.089	.089	.089	.089	.087	.093
f CORRECTED (air side)	.142	.211	.195	.210	.209	.190	.172	.214
f CORRECTED (gas side)	.137	.172	.142	.154	.138	.159	.145	.167

TEST NUMBER	37-38	38	39	47	47-48	48		
AIR SIDE STACK DRAFT (in. H ₂ O)	.074	.075	.068	.081	.077	.074		
COMBUSTION SIDE STACK DRAFT (in. H ₂ O)	.091	.091	.089	.095	.093	.091		
f CORRECTED (air side)	.180	.188	.182	.206	.189	.198		
f CORRECTED (gas side)	.186	.172	.151	.171	.181	.157		

TABLE A3.4. HIGH TEMPERATURE HEAT TRANSFER TEST RESULTS

Test No.	COMBUSTION GAS				AIR			
	Flow Rate (lb/min)	Reynolds Number	Inlet Temp. (°F)	Outlet Temp. (°F)	Flow Rate (lb/min)	Reynolds Number	Inlet Temp. (°F)	Outlet Temp. (°F)
1	1.13	3,809	625	373	0.24	896	127	605
2	1.04	3,680	637	235	1.12	4,413	105	498
3	1.24	8,675	576	195	2.02	8,675	95	352
4	1.28	5,117	470	142	3.14	14,684	91	224
5	1.29	5,157	493	113	6.16	29,829	84	173
6	4.04	13,758	557	443	0.22	863	138	543
7	4.20	14,933	517	369	1.16	4,617	132	478
8	4.27	15,524	512	317	1.94	8,028	111	420
9	4.35	15,916	529	284	3.40	14,669	105	356
10	4.50	16,985	506	230	6.23	28,436	100	260
11	1.77	4,921	1,094	624	0.26	816	178	1,037
12	1.91	5,646	1,030	438	1.28	4,349	131	877
13	1.94	5,996	1,011	307	2.56	9,825	105	586
14	1.85	5,769	1,035	258	3.37	13,581	101	481
15	1.86	5,899	1,062	180	6.73	30,080	87	326
16	3.48	9,567	993	772	0.28	913	217	973
17	3.57	10,163	972	648	1.38	4,756	95	873
18	3.33	9,787	1,063	427	3.08	11,652	64	665
19	3.12	9,506	992	369	3.66	14,342	74	573
20	3.33	10,401	995	290	7.22	30,844	82	396

Table A3.4 Cont.

Test No.	COMBUSTION GAS				AIR			
	Flow Rate (lb/min)	Reynolds Number	Inlet Temp. (°F)	Outlet Temp. (°F)	Flow Rate (lb/min)	Reynolds Number	Inlet Temp. (°F)	Outlet Temp. (°F)
21	1.86	4,352	1,590	940	0.43	1,166	268	1,535
22	1.73	4,608	1,461	434	1.82	6,063	68	1,009
22A	1.99	4,942	1,566	664	1.56	4,692	88	1,326
22-32	2.03	4,827	1,732	715	1.61	4,631	84	1,481
23	2.02	5,223	1,580	456	2.78	9,336	94	947
24	2.09	5,498	1,551	407	3.80	13,165	98	856
25	2.06	5,484	1,507	398	4.72	15,442	94	692
26	3.21	7,304	1,637	1,040	0.88	2,399	226	1,554
27	3.40	7,870	1,609	977	1.65	4,761	121	1,444
28	3.69	9,135	1,566	679	3.58	11,649	91	1,069
29	3.81	9,382	1,610	661	4.50	15,019	82	984
29-30	4.03	10,279	1,540	561	5.20	17,708	83	914
30	4.17	10,529	1,589	560	6.43	21,897	97	902
32	2.19	5,088	1,957	609	2.21	6,479	91	1,401
33	2.73	6,345	1,921	643	2.81	8,450	81	1,327
33A	2.76	6,301	1,996	662	2.69	7,954	82	1,385
33B	3.42	7,779	1,937	741	3.37	10,028	94	1,336
34	1.96	4,590	1,933	592	2.74	8,374	87	1,259
37	3.10	6,660	1,962	1,030	1.90	5,131	112	1,719
32A	2.06	4,650	1,944	776	1.59	4,363	86	1,688

Table A3.4 Cont.

[illegible]

APPENDIX 4
SIMPLIFIED HEAT TRANSFER MODEL

SIMPLIFIED HEAT TRANSFER MODEL

The heat transfer model uses nineteen equations to define the energy transfer modes and energy balance conditions for the recuperator geometry. The geometry and transfer modes are shown in simplified form in Figure A1. Table A4.1 shows the basic form of the governing equations for the transfer modes. Table A4.2 shows the specific 15 modal equations and the 4 energy balance equations used in the model. Notations are defined at the end of the discussion.

The model is a one dimensional computerized model and uses an iterative technique to obtain a convergent solution based upon internal consistency of the governing equations. The model calculates all temperatures and heat transfer quantities based upon defined temperatures of exhaust and cooling gases. To modify the model for the base case and various parametric calculations, it is only necessary to insert the appropriate coefficients. Separate calculations at different temperature zones and levels are required.

The model as shown in Table A4.2 includes the modes necessary to calculate air doping, transparent interface and variations in emissivities. In the model it was elected to not calculate radiative transfer from hot confining wall and/or exhaust gas to cooling air directly but rather to assume the transfer was by radiation from the hot gas and hot wall to the cold confining wall and then by both radiation and conduction/convection to the cooling air. This greatly simplifies the model and results in negligible errors.

Figure A4.1. Schematic of heat transfer indicating the 15 modes of heat transfer

Table A4.1
Mode Types for Heat Transfer

<u>Mode</u>	<u>Transfer Media</u>	<u>Equation</u>
Convection/Conduction	gas ↔ solid	$\dot{Q} = Ah (T_1 - T_2)$
Radiation	gas ↔ solid	$\dot{Q} = A\sigma (\epsilon_{g1}T_1^4 - \alpha_{g2}T_2^4)$
	solid ↔ solid	$\dot{Q} = AF_a F_b \sigma (T_1^4 - T_2^4)$
Conduction	contact solids	$\dot{Q} = A (\bar{u}) (T_1 - T_2)$

TABLE A4.2

Modal Equations

Mode or Condition	Media	Equation
Convection/Conduction	exhaust gas to hot interface	$\dot{Q}_1 = A_i h_e (T - T_{ih})$
Radiation	exhaust gas to hot interface	$\dot{Q}_2 = A_i (1-\tau) \sigma F_{ei} (\epsilon_e T_e^4 - \alpha_{ih} T_{ih}^4)$
Radiation	exhaust gas to cold confining wall	$\dot{Q}_3 = A_i \sigma F_{ecw} (\epsilon_e T_e^4 - \alpha_{cw} T_{cw}^4)$
Convection/Conduction	exhaust gas to hot confining wall	$\dot{Q}_4 = A_w h_e (T_e - T_{hw})$
Radiation	exhaust gas to hot confining wall	$\dot{Q}_5 = A_w \sigma F_{ehw} (\epsilon_e T_e^4 - \alpha_{hw} T_{hw}^4)$
Radiation	hot confining wall to hot interface	$\dot{Q}_6 = A_i \sigma F_{hwi} (1-\tau) \epsilon'_{hw} \epsilon'_{ih} (T_{hw}^4 - T_{ih}^4)$
Conduction	across interface	$\dot{Q}_7 = A_i (K/L) (T_{ih} - T_{ic})$
Conduction/Convection	cold interface to cooling air	$\dot{Q}_8 = A_i h_a (T_{ic} - T_a)$
Radiation	cold interface to cooling air	$\dot{Q}_9 = A_i F_{ica} (\epsilon_{ih} T_{ic}^4 - \alpha_a T_a^4)$
Radiation	cold interface to cold confining wall	$\dot{Q}_{10} = A_w F_{iccw} \sigma (1-\tau) (\epsilon'_{ic} \epsilon'_{cw}) (T_{ic}^4 - T_{cw}^4)$
Radiation	hot confining wall to cold confining wall	$\dot{Q}_{11} = A_w F_{hwcw} \sigma \epsilon_{hw} \epsilon_{cw} (T_{hw}^4 - T_{cw}^4)$
Convection/Conduction	cold confining wall to cooling air	$\dot{Q}_{12} = A_w h_a (T_{cw} - T_a)$
Radiation	cold confining wall to cooling air	$\dot{Q}_{13} = A_w F_{cwa} \sigma (\epsilon_{cw} T_{cw}^4 - \alpha_a T_a^4)$
Conduction	hot confining wall to environment	$\dot{Q}_{14} = A_w (\bar{u}) (T_{hw} - T_r)$
Conduction	cold confining wall to environment	$\dot{Q}_{15} = A_w (\bar{u}) (T_{cw} - T_r)$
Hot interface balance	interface	$\dot{Q}_1 + \dot{Q}_2 + \dot{Q}_6 = \dot{Q}_7$
Hot confining wall balance	hot confining wall	$\dot{Q}_4 + \dot{Q}_5 = \dot{Q}_6 + \dot{Q}_{11} + \dot{Q}_{14}$
Cold interface balance	interface	$\dot{Q}_7 = \dot{Q}_8 + \dot{Q}_9 + \dot{Q}_{10}$
Cold confining wall balance	cold confining wall	$\dot{Q}_3 + \dot{Q}_{10} + \dot{Q}_{11} = \dot{Q}_{12} + \dot{Q}_{13} + \dot{Q}_{15}$

Note: Emissivity ϵ' is for solid to solid radiation transfer which is different than ϵ which here is used for gas to solid radiation transfer.

NOTATION

Parameters

$\dot{Q}$ = heat flow

T = temperature

$\bar{u}$ = Complex Thermal Conductivity

K = Material Thermal Conductivity

L = Thickness

h = Convection/conduction coefficient

A = Area

F = Form factor for radiation transfer

σ = Stefan - Boltzman constant

ϵ = Emissivity

α = Absorptivity

$\hat{h}$ = Enthalpy of component

$\hat{H}$ = Sum of enthalpies

Subscripts

a = Cooling air

e = Exhaust gas

i = Interface

c = Cold side

h = Hot side

w = Wall of recuperator

r = Environment

g = Gas

s = Solid

P = Products

R = Reactants

APPENDIX 5

OAK RIDGE NATIONAL LABORATORY'S MATERIALS
EXAMINATION REPORT

MATERIALS ANALYSIS FOR SiC HEAT

EXCHANGER PROGRAM

SUMMARY

Test specimens of KT-SiC were examined and compared to a heat exchanger module of the same material. It was found that the test specimens contained 1 to 2% porosity and had a density of about 2.95 to 2.97 g/cm³. The porosity was not distributed uniformly throughout the test bars; most had distinct regions of "good" and "bad" material. The module, on the other hand, had no "bad" material at all. It had a density of 3.02 g/cm³ and no detectable porosity.

These observations partially explain the specimen-to-specimen variations in rupture modulus. They indicate that the test specimens are not representative of the heat exchanger module. Any inferences regarding the strength of the module based on these test bars will be substantially in error on the low side.

1. Introduction:

The purpose of this project was to evaluate a number of SiC test specimens and compare them to a full-size heat exchanger module made of the same material. The structure of the module and its macroscopic defects were examined destructively and nondestructively. Test specimens (fracture toughness rods and modulus-of-rupture test bars) were examined metallographically and their densities and pore size distributions were measured by mercury porosimetry. The heat exchanger module was examined for cracks using a fluorescent dye penetrant. Selected areas of the module were then sectioned for metallography and porosimetry. In addition, bend test specimens were cut from the module wall to compare its strength to that of the MOR test bars being used by TerraTek.

2. Fracture Toughness Test Rod

The FT test rod as received was 1/2 in. diameter by approximately 3/4 in. long. The rod was cut into several pieces using a low-speed diamond saw.

The specimen submitted for mercury porosimetry had an apparent density of $\sim 2.97 \text{ g/cm}^3$ at atmospheric pressure. At 2000 psi the apparent density was 3.01 g/cm^3 . The volume of open porosity represented $\sim 1.45\%$ of the specimen. The majority of open porosity was in the size range 1.05 to $4.86 \mu\text{m}$.

Transverse and longitudinal sections of the FT test rod were prepared metallographically. It was found that the rod contained a layer of "good" material ($\sim 1/16$ in. thick) around the circumference, which contained no porosity. The interior of the specimen, however, contained a fairly large amount of porosity. Figs. 1 and 2 show representative photomicrographs of the FT test rod.

3. Modulus-of-Rupture Test Bar:

One untested test bar was received from TerraTek for examination. Later, five broken MOR test bars were sent for metallography in an effort to explain the wide variation in measured rupture modulus.

The unbroken MOR test bar was checked by fluorescent dye penetrant for cracks and porosity. No cracks were found, but a large amount of open porosity was present. The bar was then sectioned as shown in Fig. 3. The apparent density of the bar was $\sim 2.97 \text{ g/cm}^3$ and the open porosity amounted to $\sim 2.3\%$ by volume. The pore size distribution was nearly identical to that present in the FT test rod. Optical metallography

revealed that, as in the FT test rod, the porosity was not uniformly distributed in the MOR test bar. Instead, there were definite "good" and "bad" areas, shown schematically in Fig. 4.

The five broken MOR test bars were sectioned in the transverse direction approximately 1/4 in. from the fracture surface. The porosity distributions are shown schematically in Fig. 5. From this, it would be expected that the specimens could be ranked in order of decreasing strength, as follows:

No. 11	Best
10	
7	
2	
corroded sample	Worst

It is, of course, possible that the porosity distribution could vary along the length of the bars, although the distribution in the unbroken specimen (Figs. 3 and 4) was fairly uniform along its length.

Therefore, this ranking must be interpreted with caution. It mainly serves to emphasize the specimen-to-specimen variation, which could be further enhanced by the specimen loading direction. (That is, if the "good" area is concentrated on the "tensile" side of beam the measured strength will be much greater than if the "good" area is on the "compression" side.)

The microstructure of the specimen that had been exposed to glass volatiles was unaltered in the interior. There was a thin glassy layer on the outside and evidence of material removal, particularly at the corners. However, there was no evidence of corrosive penetration or any changes deeper than the outermost layer of grains.

4. Heat Exchanger Module:

X-radiography of the module from several angles showed no detectable flaws. Fluorescent dye penetrant testing detected several cracks in the vicinity of a large chipped area on the flange. There was also a small crack, ~1 in. long in the very center of the module (in the groove at the center of the mating surface). The module was then sectioned as shown in Fig. 6.

Sections of the cylinder wall and the center web were found to have no measurable porosity by mercury intrusion. The density was ~3.02 g/cm³. Optical metallography showed the microstructure to be very dense and free from defects and visible porosity. The cylinder wall, web, and web-to-wall joint were all examined metallographically. A typical micrograph is shown in Fig. 7.

As indicated in Fig. 6, MOR test specimens were machined from sections of the cylinder wall. Bending strength was measured in two directions as shown in Fig. 8. The results were as follows:

<u>Sample</u>	<u>X-Bend (ksi)</u>	<u>Sample</u>	<u>Y-Bend (ksi)</u>
1	31.4	1	33.0
2	30.0	2	28.3
3	32.3	3	29.4
4	33.0	4	30.5
5	31.5	5	30.7
Ave	31.6 ± 1.1	Ave	30.4 ± 1.7

5. Comparison of Test Specimens and Module

It is clear from the micrographs and porosity measurements that the material in the heat exchanger module is of appreciably higher quality than that in the MOR test bars and the FT test rod. The test specimens,

therefore, are not representative of the module. If strength measurements made on these specimens were taken at face value, it would result in a serious underestimation of the module's strength and overall performance.

a

b

Fig. 1. Longitudinal section of FT test rod, showing (a) good and (b) bad areas (all micrographs as-polished 100X).

d

b

Fig. 2. Transverse section of FT test rod, showing (a) good and (b) bad areas.

Fig. 3. Modulus of Rupture test bar, showing locations of specimens for metallography and porosimetry.

Fig. 4. Diagram of MOR test bar metallographic specimens showing good and bad areas. (For appearance of good and bad regions, refer to Figs. 1 and 2.) Orientation corresponds to section in Fig. 3.

ITEM 3 NO.2

ITEM 3 NO.7

GLASS

CORRODED SPECIMEN

ITEM 3 NO.10

ITEM 3 NO.11

GOOD

HOLES

BAD

VERY BAD

Fig. 5. Porosity distributions in five MOR test specimens. (Note that item 3, no. 2 was not as easily divided into good and bad areas. The entire cross section had some porosity but the amount was highly variable.)

Fig. 6. Photo of heat exchanger module, showing locations of specimens.
(a) Bend Test Bars
(b) Metallography
(c) Mercury Porosimetry

Fig. 7. Typical microstructure of KT-SiC heat exchanger module.

Fig. 8. Diagram of bend specimens taken from wall of heat exchanger module, showing definition of bend directions.

APPENDIX 6
ACUREX AEROTHERM'S THERMOSTRUCTURAL
ANALYSIS REPORT

THERMOSTRUCTURAL ANALYSIS
OF A
SUBSCALE RECUPERATOR MODULE

V. S. KULKARNI

MARCH 1980

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Section 7 RESULTS

Section 8 FAILURE CRITERIA AND DESIGN MARGINS

Section 9 CONCLUSIONS AND RECOMMENDATIONS

REFERENCES

SECTION 1

INTRODUCTION

TerraTek Inc. is developing a unique heat exchanger concept incorporating a helical ceramic (silicon carbide) recuperator. A recuperator is a gas to gas to heat exchanger in which there is a flow of hot gas on one side of a heat exchanging surface and colder air on the other side.

Silicon carbide, being a brittle material, requires special considerations in the design. Brittle materials fail at the peak stress no matter how localized as compared to ductile materials where redistribution of stress takes place. This, in turn, leads to a requirement for much more refined stress analysis. An added complication in the present case is the complexity of geometry and thermal loading conditions. This calls for a full three dimensional analysis. These requirements can be met by finite element analysis which is being used extensively over the last few years for complex stress analysis.

Heat exchangers, because of their function, invariably have temperature gradients. The temperature gradients, restraint of free thermal expansion if any, and temperature dependent material properties

all cause thermal stresses to which brittle ceramic materials are very susceptible.

This report describes a three dimensional linear elastic thermostructural analysis of a sub-scale recuperator module. The analysis technique is described in the next section. The material properties used for the analysis are given in Section 3. The finite element model, loading and boundary conditions are described in Sections 4 and 5. The method used to interpolate the temperatures is explained in Section 6. Results of the analysis are presented in Section 7 followed by design margins and failure criteria in Section 8. Finally conclusions and recommendations are given in Section 9.

SECTION 2

ANALYSIS TECHNIQUE

The complexity of the recuperator geometry and the loading conditions required a full three dimensional finite element analysis. This was performed using the well known general purpose computer program ANSYS (Engineering Analysis System), described in Reference 1.

In a finite element analysis, the body is first subdivided into a large number of small discrete pieces called finite elements. The state of the displacements within these elements depends on the element's geometry, material properties, applied loadings, temperatures etc. The elements are mathematically connected such that displacement continuity exists between the elements. The resulting set of simultaneous equations is solved for the displacement of the corners (nodes) of the elements from which the strains and stresses within each element are calculated.

SECTION 3

MATERIAL PROPERTIES

For linear elastic thermostructural analysis, assuming isotropic material behavior, the required mechanical properties of the material are: elastic modulus, Poisson's ratio and coefficient of thermal expansion; all of which may be temperature dependent. Also strength data are required to evaluate design margins. All these data were supplied by TerraTek (Reference 2) and are shown in Figures 1 through 3. The Poisson's ratio was assumed to be 0.15 based on the range of 0.14 to 0.16 estimated by TerraTek. The properties are listed in Table 1 (with appropriate conversions). In this table, the coefficient of thermal expansion was based on a reference temperature of 70 °F.

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Figure 1 Elastic Modulus of KT-SiC

Figure 2 Thermal Expansion of KT-SiC

Figure 3 Flexural Strength of KT-SiC

Table 1 Mechanical Properties of KT-SiC

Temperature (°F)	Elastic Modulus (10^6 psi)	Coefficient of thermal expansion (10^{-6} in/in/°F)	Poisson's ratio
70	53	2.60	0.15
392		2.67	0.15
752		2.73	0.15
1112	50	2.78	0.15
1472		2.57	0.15
1832	49	2.72	0.15
2192	48	2.97	0.15
2408		3.12	0.15

SECTION 4

FINITE ELEMENT MODEL

The finite element model of one module of the subscale recuperator was constructed from the engineering drawings supplied by Terratek (Figure 4). The three dimensional model, shown in Figures 5 through 9 comprises of 204 elements and 438 nodes. Each node has three degrees of freedom: translations in the x , y and z directions.

One notable departure from the original program plan was the substitution of the solid element STIF45 in ANSYS for the STIF94 element. STIF94 is a higher order element which is most useful to model curved boundaries with very few elements. However, if many elements are used, the boundaries of individual elements approach straight lines. Therefore elements like STIF45 which is a hexahedron (8 node brick) can be substituted with no loss in accuracy. Consequently, since many elements are used, a better resolution of the stress distribution is obtained. Furthermore the computational cost per element for STIF45 is less than the more complex STIF94 element.

Figure 5 Finite Element Model of Subscale Recuperator Module (Isometric View)

Figure 6 Finite Element Model of Subscale Recuperator Module (Transparent View)

Figure 7

Finite Element Model of Subscale Recuperator Module (Top View)

Figure 8 Cross-sectional View of Finite Element Model, Section A-A.

Figure 9 Cross-sectional View of Finite Element Model, Section B-B

Also some of the post processors in ANSYS are not yet applicable to STIF94.

STIF45 is an isoparametric element with extra displacement shapes. This gives better accuracy compared to the conventional constant strain elements.

SECTION 5

LOADING AND BOUNDARY CONDITIONS

The only loading considered was that due to the temperature distribution throughout the module. The loading due to pressure was small hence neglected. One important loading not considered was the axial load on the module. The axial load is estimated to be 1200 pounds and is due to the weight of the stacked modules and the clamping force. The axial load has to be applied and reacted in a "Compression only" manner because the modules are merely stacked and are not bonded or joined in any way. They can transmit compressive forces only but not tensile forces. Any attempt to transmit tensile forces will produce fictitious stresses. To treat the axial load correctly, it is necessary to use gap elements. This however makes the problem non linear and beyond the present work-scope. It is possible that the stresses due to the axial load add or subtract to the thermal stresses in different areas of the module. Also the loads may be transmitted as point loads rather than distributed loads giving rise to higher stresses. Thus it may be unconservative to ignore the axial load.

Since the only loading considered was the thermal load, which is self equilibrating in nature, there was no need to specify any boundary conditions. However external restraints such as the effects of other modules were not considered. These may cause additional stresses.

The temperature distribution data were generated by TerraTek in a test program. Details of the test program are given in Reference 3. From the instrumented subscale tests, TerraTek selected three most severe conditions based on several criteria. One criterion was the highest temperature reached. The others involved the determination of the deviation of the module temperature distribution relative to a module reference temperature. The test results are given in Reference 4.

The three conditions selected for the thermostructural analysis were:

- 1) Transient start up condition which produced high global and interface temperature deviation

- 2) Steady State Condition which produced the greatest temperature drop across the interface of the module from the combustion products to the air side.

3) Loss of air flow to the recuperator : an upset condition which produced the highest temperatures.

The temperatures were measured by forty thermocouples placed on the module. The thermocouple locations are shown in Figure 10. For each condition, the temperatures were interpolated for thermostructural analysis. The interpolation scheme is explained in the next section.

Figure 10 Thermocouple placement and coordinate system

SECTION 6

TEMPERATURE INTERPOLATION

The thermal field was available as temperatures measured by thermocouples at forty discrete locations on the module (Figure 10). To do thermostructural analysis, it is necessary to specify ^{the} temperatures at each and every nodal point (438 in all) of the finite element model. Thus it was necessary to interpolate the measured temperatures. The approach used was as follows:

First, selected nodes were placed at the thermocouple locations. For these nodes, there was no need to interpolate the temperatures. However, no nodes could be located at the thermocouples on the rim because there was only one element through the thickness. In this case, the thermocouple temperature was assigned to both the inside and outside nodes (Figure 11).

The interpolation was done by solving the Laplace equation:

$$\nabla^2 T = 0 \quad \text{for temperatures}$$

T. Basically this is the steady state heat conduction equation. Two of the cases (startup and loss of Air) were transient. For these, the application of the steady state equation could be thought of as an "instantaneous Steady State" solution. This amounts to neglecting any past history

Figure 11: Specification of rim temperature

effects but accounting for the transient boundary conditions. This may not be a bad approximation considering the slow rates of changes and the high conductivity of the material.

The Laplace equation was solved over curved surfaces in three dimensions. In Figure 12 a schematic diagram of a portion of the module is shown. In this case the surfaces are ABDC, CDFE, GHJI and IJLK. Further details are explained for the surface ABDC but they apply to all the surfaces.

In order to solve the Laplace equation, it is necessary to specify the boundary conditions "properly." If the equation is to be solved on the surface ABDC, it is not enough to specify the temperatures at the corner points only (A, B, C and D), which are the only temperatures available from thermocouple measurements. One of the proper boundary conditions for this case is the specification of all the temperatures on all four edges (Figure 13). This is the Dirichlet problem in the theory of partial differential equations (Reference 5). The solution of the Laplace equation (as isotherms on a surface) are shown for a test case with and without proper boundary conditions in Figure 14. For proper boundary conditions, the temperatures were specified along all the boundaries. For the improper boundary conditions, the temperatures were specified

Figure 12: Schematic diagram of a portion of the module

- ◇ Thermocouple
- Boundary point
- △ Interior point

Figure 13 : Specification of boundary conditions for surface interpolation

With proper boundary conditions

With improper boundary conditions

Figure 14: Solution of Laplace equation

Only at the corner points. The need for specifying proper boundary conditions is evident from this Figure.

The interpolation was done in two steps:

Step 1: Find boundary temperatures from the corner (thermocouple) temperatures.

Step 2: Find interior temperatures from the boundary temperatures.

For the present case, these two steps were sufficient, since all the nodes in the finite element model were on the surface. Otherwise it would have been necessary to go one more step, i.e. solve for the interior temperatures when all the surface temperatures are known from step 2. The Laplace equation would be solved over a volume in this case.

Step 1: The boundary temperatures were obtained by linearly interpolating along the arc length between the thermocouple data (Figure 15). Note that this is just a special case of the Laplace equation in one dimension (even though the points themselves are in three dimensional space). For this case specification of temperatures at the end points (A and B) is a proper boundary condition. This step was performed using the ANSYS 1-D heat Conduction bar element STIF33. The 1-D bar model

- ◇ : Thermocouple
- : Interpolated boundary point.
- S : Arc length

Figure 15: Determination of boundary temperature

for the entire module is shown in Figure 16.

Step 2: The interior temperatures were obtained by solving the Laplace equation for each surface with boundary temperatures obtained from step 1.

This was done using the ANSYS 3-D thermal shell element STIF57 having only in-plane thermal conduction capability. It was necessary to use triangular rather than quadrilateral shell elements (Figure 17) because of the twisted geometry of the module since any four points do not always lie in a plane but three points always define a plane.

Isotherm plots were made to check the adequacy of the temperature interpolation. The plots are presented in Figures 18 through 26.

6-34

Figure 16

- ◇ : Thermocouple
- : Boundary point
- △ : Interior point

Figure 17: Schematic of surface interpolation

FIGURE 18 Start up, top view, isotherms (deg F)

Figure 19

Start up, bottom view, isotherms (deg F)

Section A-A

Section B-B

Figure 20

Start up, isotherms (deg F)

Figure 21

Steady state, top view, isotherms (deg. F)

Figure 22

Steady state, bottom view, isotherms (deg F)

Steady state, isotherms (deg F)

Figure 23

Figure 24 Loss of air, top view, isotherms (deg F)

Figure 25 Loss of air, bottom view, isotherms (deg F)

Figure 26

Loss of air, isotherms (deg F)

SECTION 7

RESULTS

Results of the thermostructural analysis are presented as stress contour plots in the following pages. For each of the three conditions considered contours of maximum and minimum stresses are plotted. The plots are made for top view, bottom view, and two section views (see Figure 7 for section definitions). The maximum and minimum stresses are summarized below:

Condition	Maximum stress (psi)	Minimum stress (psi)
Start up	6181	- 6407
Steady state	6310	- 4730
Loss of air	6399	- 4457

Figure 33 Steady state, top view, maximum stress (psi)

Figure 34

Steady state, bottom view, maximum stress (psi)

Figure 35

Steady state, maximum stress (psi)

Figure 36 Steady state, top view, minimum stress (psi)

Figure 37 Steady state, bottom view, minimum stress (psi)

Section A-A

Section B-B

Figure 38

Steady state, minimum stress (psi),

SECTION 8

FAILURE CRITERIA AND DESIGN MARGINS

In general, the mechanical properties of a ceramic material vary considerably. Causes of variability include such macroscopic and microscopic factors ~~as~~ as porosity, density variations, impurities, microcracks and compositional variations. While the degree of variability can be influenced to some extent by quality control procedures, it cannot be reduced to the low level normally associated with ductile materials. Any attempt to design with brittle materials must recognize this variability of properties.

The strength behavior of brittle materials is statistical in nature. Therefore the allowable stresses must be related to a probability of failure. Hence the assessment of structural integrity becomes a problem of determining the probability of failure under given loading conditions rather than whether or not the part will fail.

The best known statistical theory of strength was developed by W. Weibull (Reference 6). With this theory, the strength data from tests is fitted with a statistical distribution to predict the probability of survival. The distribution function is:

$$S = \exp - \left\{ \int_V \left[\frac{\sigma(x,y,z) - \sigma_u}{\sigma_0} \right]^m dx dy dz \right\} \quad \text{--- (1)}$$

Where S = probability of survival

$\sigma(x,y,z)$ = applied stress

σ_u = stress below which failure will never occur

σ_0 = normalizing scale parameter locating the strength distribution

m = measure of scatter (Weibull material parameter)

V = material volume

It is conventional to set σ_u to zero.

This is the safest approach because there are generally insufficient experimental data to determine it accurately.

In effect $\sigma_u = 0$ states that there is no upper limit to the size of the largest flaw in the material.

In the present case, for lack of sufficient data the volume effect will be ignored. Therefore the equation can be simplified to:

$$S = \exp - \left\{ \frac{\sigma}{\sigma_0} \right\}^m \quad \text{--- (2)}$$

Reference 7 gives a value of 8 for the Weibull parameter m . To evaluate σ_0 it is necessary to make some assumptions.

TerraTek supplied flexural strength data generated by four point bending test (see Figure 3). Values of tensile strength are usually lower than those obtained in bending because a larger volume of material is under maximum stress in a tensile specimen compared to bending test where ^{the} stresses decrease from a maximum at the surface to zero at neutral axis (Reference 8). It has been observed that for silicon carbide, the tensile strength is approximately half that in a three point bending test (Reference 7). The available four point bend test data was converted to three point bend data by the equation:

$$\frac{\sigma_{3\text{-point}}}{\sigma_{4\text{-point}}} = \left(\frac{m+2}{2} \right)^{\frac{1}{m}} \quad \text{Reference 7}$$

For $m=8$,

$$\sigma_{3\text{-point}} = 1.22 \sigma_{4\text{-point}}$$

$\sigma_{4\text{-point}}$ is approximately 18000 psi upto 1400°F (see Figure 3), after which it increases gradually to 25000 psi at about 2200°F .

$$\begin{aligned} \text{Thus } \sigma_{3\text{-point}} &= 1.22 \times 18000 \\ &\approx 22000 \text{ psi} \end{aligned}$$

$$\text{Therefore tensile strength} = \frac{22000}{2} = 11000 \text{ psi}$$

Let us arbitrarily assume that the probability of survival when the stress reaches tensile strength is 5 %. Therefore from equation (2),

$$0.05 = \exp - \left\{ \frac{11000}{\sigma_0} \right\}^8$$

$$\begin{aligned} \text{Thus } \sigma_0 &= \frac{11000}{(-\ln 0.05)^{1/8}} \\ &= 9590 \text{ psi} \end{aligned}$$

For multi-axial stress states, several failure criteria are postulated such as:

Maximum stress theory

Maximum shear stress theory

Distortion energy theory

Coulomb-Mohr theory etc.

For brittle ceramic materials, the maximum stress theory appears most appropriate (Reference 9).

The maximum tensile stress (6399 psi) occurs for loss of air condition. However, for this condition, the temperatures are in 2000°F range where the tensile strength is higher compared to lower temperatures. The more severe stress condition (6181 psi) occurs during start up when the tensile

strength is lower because of lower temperatures (below 1000°F).

From equation ②, the probability of survival can be calculated as

$$S = \exp - \left\{ \frac{\sigma}{\sigma_0} \right\}^m$$

For $\sigma = 6181 \text{ psi}$

$\sigma_0 = 9590 \text{ psi}$

$m = 8,$

$$S = 0.97$$

Thus there is a 97% chance of survival.

In addition to the brittle fracture considered above, there are other failure modes related to time/temperature effects such as delayed fracture (static fatigue), cyclic mechanical and thermal fatigue, Creep etc. For lack of sufficient data these effects were not considered.

SECTION 9

CONCLUSIONS AND RECOMMENDATIONS

The analysis indicates that the worst case stress condition occurs during the start up transient. For the loading considered (thermal stress only in a single module), the maximum stress during startup is 6181 psi. Weibull statistical analysis shows a 97% chance of survival under these conditions.

It should be emphasized that there are major uncertainties and approximations with regard to some aspects of the analysis. These are described next.

- Only loading considered was the temperature distribution in a single module. Axial load and external constraints such as effect of the other modules on a given module were not considered.

- The temperature distribution applied to the module needed several approximations:

- a) The temperatures were measured with thermocouples on an intermediate temperature metallic recuperator at a gas temperature of 800°F. These temperatures were scaled to the ceramic recuperator operating at a gas temperature of

2400°F. At these temperatures radiation is the predominant mode of heat transfer (Reference 10) which is highly nonlinear.

b) The temperatures were measured at discrete locations and interpolated. Due to this local thermal gradients may not be adequately represented especially on the rim where thermocouples were placed only in the middle (See Figure 11)

- The preliminary nature of the analysis and the cost and schedule constraints precluded refined modelling near stress concentration regions. In view of the uncertainties in the applied loading such a refined model was not justified any way.

- Time/temperature effects such as creep and fatigue were not considered.

Recommendations

- The major uncertainties in the present analysis are related to the applied loading i.e. temperature distributions, axial loads and external constraints.

A heat transfer analysis would obviate the need to scale up and interpolate temperatures. However good judgements about the convective and

radiative heat transfer should be made.

Axial load could be applied using gap elements. External constraints could be considered by analyzing several modules together. To do this cost effectively use of super elements in the finite element analysis is recommended.

- More sophisticated failure criteria such as fracture mechanics may be used.

- It is difficult to make any judgement on scaling up (geometrically) the results. If the temperature gradients were predominantly through the thickness, and the thickness was ^{not} changed, the scale up would not have affected the thermal stresses, however this is not the case.

APPENDIX 7
ECONOMIC CALCULATIONS

APPENDIX 7 - ECONOMIC CALCULATIONS

Fuel Savings

The fuel saving by air preheating is based on the following assumptions:

- 1) The fuel is natural gas which is assumed to be methane,
- 2) Complete combustion,
- 3) 10% excess air,
- 4) Process temperature is 2600°F.

The combustion equation is:

The equations governing the fuel savings are the following:

$$Q_T = \bar{h}_{\text{CH}_4} + n\bar{h}_{\text{O}_2} + n\bar{h}_{\text{N}_2}$$

$$Q_F = n\bar{h}_{\text{CO}_2} + n\bar{h}_{\text{O}_2} + n\bar{h}_{\text{N}_2} + n\bar{h}_{\text{H}_2\text{O}}$$

$$Q_L = Q_T - Q_F$$

where,

Q_T = Heat input to furnace, evaluated at preheat temperature
(Btu/lbmole CH_4)

Q_F = Heat output to flue, evaluated at 2600°F temperature
(Btu/lbmole CH_4)

Q_L = Heat load to furnace, includes heat to melt feed and for heat loss (Btu/lbmole CH_4)

h = Enthalpy (Btu/lbmole)

n = Moles of molecule per mole of CH_4

The fuel savings is defined as:

$$S = \frac{1/Q_{L,w/o} - 1/Q_{L,T}}{1/Q_{L,w/o}} \times 100$$

where,

S = Percent fuel saving for preheat versus no preheat

$Q_{L,w/o}$ = Heat load without preheat (Btu/lbmole CH_4)

$Q_{L,T}$ = Heat load at preheat temperature T. (Btu/lbmole CH_4).

The results are shown in Table A7.1.

Furnace Variables

The furnace size and preheat temperature specify the total air and exhaust gas flow rate, and the fuel requirements. The analysis is made with the following assumptions:

- 1) Heat required to process glass is 2,200,000 Btu/ton. (Reference 1)
- 2) The heat loss from the furnace equals the heat required to melt glass
- 3) Process temperature is 2600°F.

The governing equations for mass flows are:

$$\dot{M}_{CH_4} = \frac{q_L}{Q_L / \hat{M}_{CH_4}}$$

$$\dot{M}_{air} = \dot{M}_{CH_4} \times \frac{\text{no. moles air}}{\text{no. moles } CH_4} \times \frac{\hat{M}_{air}}{\hat{M}_{CH_4}}$$

$$\dot{M}_g = \dot{M}_{CH_4} \times \frac{\text{no. moles exhaust gas}}{\text{no. moles } CH_4} \times \frac{\hat{M}_{air}}{\hat{M}_{CH_4}}$$

TABLE A7.1: FUEL SAVINGS

Air Preheat (°F)	Q_L (Btu/lbmole CH_4)	S (%)
77	106,290	-
200	115,318	7.8
400	130,090	18.3
600	145,135	26.8
800	160,644	33.8
1,000	176,385	39.7
1,200	192,649	44.8
1,400	209,354	49.2
1,700	235,368	54.8
2,000	260,941	59.3
2,500	304,536	65.1

where,

$\dot{M}$ = Mass flow rate (lbm/hr)

$\hat{M}$ = Molecular weight (lbm/lbmole)

Q_L = Heat load (Btu/lbmole CH_4)

q_L = Heat load (Btu/hr).

With no preheat the ratio of the total heat input needed to the furnace to the heat required in the furnace to match the load is 3.3 for a process temperature of 2600°F. The equation for the heat that must be bought for no preheat is,

$$q_T = F \cdot 2 \cdot 2.2 \times 10^6 \cdot 3.3/24$$

F = Furnace size (ton/day)

q_T = Heat that must be purchased (Btu/hr).

The results for the different furnaces used in the analysis are shown in Table A7.2.

TABLE A7.2. Energy Requirements & Mass Flow Rate,
No Preheat

Furnace Size (Ton/Day)	Energy Requirements (Btu/hr)	Air Mass Flow Rate (lbm/hr)
25	1.51×10^7	13,030
150	9.07×10^7	78,170
300	1.81×10^8	156,330
600	3.63×10^8	312,660

Recuperator Sizing

The recuperator size is determined by using the results of the thermal model. These results are shown graphically in Figures A7.1, A7.2, and A7.3 for the prototype module (diameter = 7.8"), the 15" module, and the 30" module respectively. The graphs portray the number of modules in a column as a function of the mass flow rate for various preheat temperatures.

The sizing method is the following:

- 1) A furnace size is selected. The analysis includes glass furnaces of 25, 150, 300 and 600 ton/day.
- 2) A range of preheat temperatures is selected for a given furnace size. The analysis includes 1,000, 1,200, 1,400, and 1,700°F preheat temperatures. The total mass flow rate through the recuperator is specified once the furnace and preheat is selected.
- 3) The mass flow rate through an individual recuperator column is bounded between Reynolds numbers of approximately 1,000 to 100,000 for a given recuperator diameter. This is done by varying the number of columns in the system.
- 4) The recuperator size (number of modules per column) is determined from Figures A7.1, A7.2, and A7.3.

Module Cost

The cost of the modules for a particular installation is given by:

$$\text{Cost}(\$) = N \cdot p \cdot V \cdot C$$

N = Number of modules

p = Module material density (lbm/ft³)

V = Module volume (ft³)

C = Cost of module material per pound (\$/lbm)

FIGURE A7.1. Number of modules versus mass flow rate - prototype module.

FIGURE A7.2. Number of modules versus mass flow rate - 15 inch diameter modules,

FIGURE A7.3. Number of modules versus mass flow rate - 30 inch diameter modules.

The analysis assumes a module material density of 200 lbm/ft³ and a cost per pound of \$24. (Reference 2).

The volume term, V, is scaled from the prototype diameter to the larger diameters by assuming the wall and interface thickness remains the same as the prototype. The volumes are:

$$V \begin{cases} = .02105 \text{ ft}^3/\text{module} & 7.8 \text{ in. diameter} \\ = .07414 \text{ ft}^3/\text{module} & 15 \text{ in. diameter} \\ = .3031 \text{ ft}^3/\text{module} & 30 \text{ in. diameter} \end{cases}$$

Blower Sizing

The blowers are sized by determining the pressure drop on the air and exhaust gas side of the recuperator for a particular installation. It is assumed that blowers on the air inlet and combustion gas side is required. This constraint allows the furnace to be run at near atmospheric pressures. The pressure drop is calculated from curve fits of pressure drop versus mass flow rate and the number of modules. The curve fits are based on the following equation:

$$\Delta P = f \frac{L}{D_H} \frac{\rho V^2}{2g_c} \pm H (\rho_{AIM} - \rho) g/g_c$$

where,

D_H = Module hydraulic diameter (ft)

f = Friction factor

g = Gravitational acceleration (ft/sec²)

g_c = Gravitational constant ($\frac{\text{lbm} \cdot \text{ft}}{\text{lbf} \cdot \text{sec}^2}$)

H = Recuperator height (ft)

L = Path length (ft)

V = Fluid velocity at average temperature and pressure (ft/sec)

ρ = Fluid density at average temperature and pressure (lbm/ft³)

ρ_{ATM} = Atmospheric air density (lbm/ft³)

+ = Downward recuperator flow

- = Upward recuperator flow

The values for friction factor used in the analysis are:

$$f = \begin{cases} 720/\text{Re} & \text{Re} \leq 1000 \\ .250 & 1000 < \text{Re} \leq 2000 \\ 1.066 \text{Re}^{-.196} & 2000 < \text{Re} \end{cases}$$

The resulting curve fits take the following form:

$$\Delta P = (a + b\dot{m} + c\dot{m}^2) N/100$$

where,

ΔP = Pressure drop (in H₂O)

$\dot{m}$ = Mass flow rate (lbm/min)

N = Number of modules in a column.

The values of the parameter a , b , and c are shown in Table A7.3.

TABLE A7.3. Curve Fit Parameters for Pressure Drop

Module Size	Recuperator Side	Bounds on Mass Flow Rate (lbm/min)	a	b	c
Proto-type	Air	$1.0 \leq m \leq 10.0$	-.057	.3452	.1061
		$10 < m \leq 50$	-15	2.80	.01
	Exhaust Gas	$1.0 \leq m \leq 10.0$	-.29	.2322	.2197
		$10 < m \leq 50$	20.8	-3.13	3.45
15" Module	Air	$1.0 \leq m \leq 20.0$	.25	.0486	.00895
		$20 < m \leq 100$	-2.6	.281	.00424
	Exhaust Gas	$1 \leq m \leq 10$	.074	.10	.0263
		$10 < m \leq 40$	-.53	.065	.0158
		$40 < m \leq 100$	24.5	-1.01	.027
30" Module	Air	$10 \leq m \leq 100$	.50	.0134	.00043
		$100 < m \leq 200$	-.70	.04	.00028
	Exhaust Gas	$10 \leq m \leq 25$	-.03	.00266	0
		$25 < m \leq 100$	-.76	.0108	.000848
		$100 \leq m \leq 200$	1.3	-.0110	.00086

Blower Cost

The blower costs are determined from a series of equations relating cost as a function of volumetric flow rate and pressure drop. (Reference 3) The costs are adjusted to January 1980 prices by using the Marshall - Steven equipment cost index. The pressure drop on the air and gas side are assumed to be double that calculated for the recuperator. This takes into account the pressure drop across piping, valving, and transition pieces. The final form of the blower cost equation is derived through curve fitting and is

$$C = 150 (\Delta P)^{.79} \left(\frac{Q}{100} \right)^{.48}$$

where

C = Blower cost (\$)

ΔP = Pressure drop (in H₂O)

Q = Volumetric flow rate (ft³/min)

The blower operating cost is determined by assuming an efficiency of 70%.

The equation for the blower work is

$$\dot{W} = \dot{Q} (\Delta P) / \eta$$

where,

ΔP = Pressure drop (lbf/ft²)

$\dot{Q}$ = Volumetric flow rate (ft³/min)

$\dot{W}$ = Work (ft-lbf/min)

η = Blower efficiency

Insulation Cost

The cost of the insulation is based on the amount of insulation required

to keep the heat loss from the recuperator less than five percent of the heat transferred from the exhaust gas to the air side of the recuperator. The governing equation is

$$t = r \left[\text{EXP} \left(\frac{2\pi k l (T_L - T_0)}{q_1} \right) - 1 \right]$$

where:

- R = Insulation conductivity (Btu/hr·ft·F)
- l = Recuperator height (ft)
- q_1 = Allowable heat loss (Btu/hr)
- r = Module outside diameter (in)
- t = Insulation thickness (in)
- T_L = Average inside insulation temperature (°F)
- T_0 = Ambient temperature (°F)

The cost of the insulation is assumed to be \$1.00/ft² for a 1" thickness. This is based on the hot flow testing costs. The total cost for a column takes the form

$$c = (a + bt + ct^2) l \times 1.2$$

where,

- a,b,c = Constant for different module diameters, see Table A7.4
- t = Insulation thickness (in)
- l = Recuperator column height (ft)

TABLE A7.4. INSULATION PARAMETERS

Recuperator Size	a	b	c
7.8	.01	2.22	.263
15.0	-.03	3.66	.2625
30.0	.01	7.85	.2625

Transition Pieces Cost

The transition pieces are the manifolding pieces which help support the module stack and provide a transition from tubing to the flow passage in the recuperator. The cost is estimated with the following assumptions:

1. Cost of the prototype transitions are taken to be twice the cost of the castable used in the hot flow testing.
2. The cost is scaled to larger dimension by the second power relationship.

The costs assumed in the analysis are:

$$\text{Cost} = \begin{cases} \$1,300/\text{Stack} & \text{Prototype} \\ \$4,800/\text{Stack} & 15" \text{ Diameter} \\ \$19,000/\text{Stack} & 30" \text{ Diameter} \end{cases}$$

Results

The results of the cost estimating are shown in Tables A7.5 through A7.8.

TABLE A7.5. 25 TON/DAY FURNACE

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
PROTOTYPE MODULE								
1,000	18.7	7	238	24,100	115	9,100	65,600	14,650
1,000	13.1	10	290	29,300	130	13,000	37,300	6,000
1,000	6.5	20	466	47,000	530	26,000	12,470	1,310
1,000	2.6	50	800	80,800	2,240	65,000	3,300	170
1,000	1.3	100	1,300	131,350	10,174	130,000	1,000	33
1,200	20	6	306	30,920	174	7,800	80,000	2.08×10^4
1,200	12	10	410	41,430	340	13,000	40,000	6.0×10^3
1,200	6.0	20	660	66,690	1,000	26,000	12,100	1.39×10^3
1,200	2.4	50	1,200	121,250	5,900	65,000	4,000	1.8×10^2
1,200	1.2	100	1,900	192,000	30,340	130,000	1,000	50
1,400	18.4	6	432	43,650	340	1,800	105,500	2.08×10^4
1,400	10	10	580	58,600	800	13,000	37,500	5.3×10^3
1,400	5.5	20	940	95,000	2,450	26,000	19,000	1.4×10^3
1,400	2.2	50	1,650	166,700	14,000	65,000	3,800	1.7×10^2

25 TON/DAY FURNACE CONT.
Prototype Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,400	1.1	100	2,800	283,000	131,500	130,000	1,140	40
1,700	19.6	5	690	69,720	930	6,500	140,000	3.55×10^4
1,700	9.8	10	1,140	115,186	3,600	1,300	55,500	7.7×10^3
1,700	6.5	15	1,515	153,100	8,600	19,500	32,600	3.08×10^3
1,700	4.9	20	1,820	184,000	16,000	26,000	24,000	1.7×10^3
1,700	1.96	50	3,250	328,400	193,000	65,000	5,000	2.16×10^2

25 TON/DAY FURNACE CONT.

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
15" MODULE								
1,000	65.4	2	80	28,500	141	9,600	33,000	1.46×10^4
1,000	43.6	3	102	36,300	240	14,400	22,000	5.78×10^3
1,000	26.2	5	135	48,000	450	24,000	10,800	1.8×10^3
1,000	13.1	10	200	71,200	1,100	48,000	4,000	3.4×10^2
1,000	6.5	20	300	106,800	3,060	96,000	1,700	70
1,000	2.6	50	500	178,000	13,000	240,000	300	10
1,200	60	2	112	39,900	260	9,600	34,100	1.46×10^4
1,200	40	3	144	51,250	450	14,400	23,800	5.8×10^3
1,200	24	5	195	69,400	890	24,000	8,200	1.9×10^3
1,200	12	10	290	103,200	2,320	48,000	4,400	3.5×10^2
1,200	6	20	430	153,000	6,560	96,000	1,750	70

25 TON/DAY FURNACE CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	2.4	50	800	284,700	46,700	240,000	400	10
1,400	55	2	160	57,000	500	9,600	36,600	6.6×10^3
1,400	36.7	3	201	71,500	860	14,400	25,300	2.7×10^3
1,400	22	5	270	96,100	1,700	24,000	12,300	8.5×10^2
1,400	11	10	410	146,000	5,000	48,000	4,600	1.8×10^2
1,400	5.5	20	620	220,700	16,460	96,000	1,800	50
1,400	2.2	50	1,150	409,300	155,000	240,000	600	15
1,700	49	2	306	109,000	2,060	9,600	45,400	1.7×10^4
1,700	32.7	3	384	136,700	3,630	14,400	31,500	6.9×10^3
1,700	24.5	4	456	162,300	5,700	19,200	19,000	3.55×10^3
1,700	19.6	5	525	186,800	8,400	24,000	15,500	2.1×10^3
1,700	9.8	10	790	281,100	29,200	48,000	5,900	4.3×10^2
1,700	6.5	15	990	352,300	65,400	72,000	3,400	1.5×10^2
1,700	4.9	20	1,180	420,000	128,400	96,000	2,300	7.7×10^1
1,700	2	50	2,200	783,000	2.9×10^5	240,000	800	20

25 TON/DAY FURNACE CONT.

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min.)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
30" MODULE								
1,000	131	1	31	45,100	352	19,000	6,000	2.4×10^3
1,000	65.4	2	44	64,000	740	38,000	3,100	4.6×10^2
1,000	43.6	3	56	81,500	1,230	57,000	1,800	170
1,000	32.7	4	66	96,000	1,800	76,000	1,270	90
1,000	26.2	5	75	109,100	2,400	95,000	700	30
1,000	21.8	6	84	122,200	3,120	114,000	650	25
1,000	18.7	7	95	138,200	4,100	133,000	570	20
1,000	16.4	8	100	145,500	4,670	152,000	550	17
1,000	14.5	9	108	157,100	5,660	171,000	530	15
1,200	13.1	10	118	171,700	6,970	190,000	530	14
1,200	120	1	44	64,000	652	19,000	8,600	2.5×10^3
1,200	60	2	54	93,100	1,470	38,000	3,430	500
1,200	40	3	80	116,400	2,400	57,000	1,850	150
1,200	30	4	94	137,000	3,484	76,000	1,190	70

25 TON/DAY FURNACE CONT.
30" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	24	5	108	157,100	4,760	95,000	850	40
1,200	20	6	120	174,600	6,200	114,000	750	30
1,200	15	8	148	215,300	10,300	152,000	690	22
1,200	10	12	198	288,000	22,000	228,000	580	14
1,400	110	1	62	90,200	1,250	19,000	9,300	2.54×10^{-3}
1,400	55	2	91	132,400	2,920	38,000	3,700	500
1,400	36.7	3	114	165,900	4,900	57,000	2,080	170
1,400	27.6	4	136	198,900	7,430	76,000	1,200	70
1,400	22	5	155	225,500	10,300	95,000	950	40
1,400	11	10	250	363,700	36,400	190,000	800	23
1,700	98	1	120	174,600	4,900	19,000	11,800	3.1×10^{-3}
1,700	49	2	168	244,400	10,960	38,000	4,500	5.5×10^{-2}
1,700	32.7	3	210	305,500	19,300	57,000	2,400	190
1,700	24.5	4	244	355,000	28,800	76,000	1,500	90
1,700	19.6	5	280	407,400	42,350	95,000	1,300	60
1,700	9.8	10	440	640,150	174,000	190,000	1,200	35

TABLE A7.6. 150 TON/DAY FURNACE

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
PROTOTYPE MODULE								
1,000	49	16	816	82,500	207	20,800	1.43×10^6	1.0×10^6
1,000	39.3	20	920	93,000	266	26,000	1.04×10^6	5.7×10^5
1,000	26.2	30	1,170	118,200	440	39,000	590,000	2.0×10^5
1,000	19.6	40	1,400	141,500	660	52,000	400,000	1.0×10^5
1,000	9.8	80	2,160	218,300	1,760	104,000	170,000	2.08×10^4
1,000	4.9	160	3,360	335,500	5,130	208,000	70,000	4.2×10^3
1,000	2.6	300	4,800	485,000	13,410	390,000	26,000	1.0×10^3
1,200	48	15	1,110	112,200	350	19,500	1.72×10^6	1.16×10^6
1,200	39.3	20	1,350	137,400	490	26,000	1.31×10^6	7.02×10^5
1,200	23.9	30	1,550	166,700	830	39,000	610,000	2.0×10^5
1,200	18	40	1,380	200,100	1,240	52,000	470,000	1.0×10^5
1,200	9	80	3,040	307,200	3,370	104,000	170,000	2.0×10^4
1,200	4.5	160	4,640	463,800	9,900	208,000	73,000	4.16×10^3
1,200	2.4	300	6,900	697,200	31,000	390,000	24,000	1.0×10^3

150/TON/DAY FURNACE CONT.
Prototype Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,400	44	15	1,650	166,700	750	19,500	1.9×10^6	1.23×10^5
1,400	33	20	1,820	184,000	930	26,000	1.15×10^6	5.47×10^5
1,400	22	30	2,310	233,400	1,600	39,000	650,000	1.93×10^5
1,400	16.5	40	2,760	278,900	2,400	52,000	440,000	1.0×10^5
1,400	8.3	80	4,280	432,500	6,950	104,000	186,000	2.0×10^4
1,400	4.1	160	6,720	679,000	24,600	208,000	79,800	4.32×10^3
1,400	2.2	300	10,500	1.06×10^6	105,400	390,000	30,400	1.2×10^3
1,700	39.2	15	2,445	247,000	1,607	19,500	1.9×10^6	1.08×10^6
1,700	29.4	20	3,120	315,250	2,820	26,000	1.3×10^6	5.55×10^5
1,700	19.6	30	4,140	418,300	5,600	39,000	760,000	2.08×10^5
1,700	14.7	40	5,080	513,300	9,500	52,000	530,000	1.08×10^5
1,700	7.35	80	8,240	832,600	37,400	104,000	230,000	2.39×10^4
1,700	3.68	160	13,600	1,374,000	219,122	208,000	106,000	5.55×10^3

TABLE 150 TON/DAY CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,000	98.1	8	380	135,200	518	38,400	360,000	2.62×10^5
1,000	65.4	12	480	170,800	850	57,600	195,000	9.25×10^4
1,000	49	16	568	202,100	1,230	76,800	130,000	4.39×10^4
1,000	39.3	20	640	227,800	1,590	96,000	100,000	2.7×10^4
1,000	26.2	30	810	288,300	2,700	144,000	59,000	1.08×10^4
1,000	15.7	50	1,075	382,600	5,100	240,000	31,000	3.31×10^3
1,000	7.9	100	1,600	569,400	13,300	480,000	12,000	5.8×10^2
1,000	3.92	200	2,400	854,000	40,000	960,000	4,400	100
1,000	2.6	300	3,000	1.07×10^6	77,000	1.44×10^6	2,500	50
1,200	89.8	8	528	187,900	1,160	38,400	370,000	2.47×10^6
1,200	60	12	672	239,100	1,540	57,600	205,000	8.48×10^4
1,200	45	16	800	284,700	2,260	76,800	142,000	4.55×10^4
1,200	36	20	920	327,400	3,090	96,000	109,000	2.78×10^4
1,200	24	30	1,170	416,400	5,400	144,000	65,000	1.08×10^4
1,200	14.4	50	1,550	551,600	10,500	240,000	34,000	3.47×10^3

150 TON/DAY FURNACE CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	7.2	100	2,300	818,500	28,800	480,000	12,000	5.4×10^{-2}
1,200	3.6	200	3,600	1.28×10^6	106,000	960,000	5,000	1.1×10^{-2}
1,400	82.7	8	752	267,600	1,800	38,400	390,000	2.47×10^{-5}
1,400	55.1	12	960	341,600	3,070	57,600	220,000	9.25×10^{-4}
1,400	41.4	16	1,136	404,300	4,500	76,800	162,000	5.0×10^{-4}
1,400	22.1	30	1,620	576,500	10,300	144,000	68,000	1.08×10^{-4}
1,400	13.2	50	2,200	782,900	22,400	240,000	35,000	3.3×10^{-3}
1,400	6.6	100	3,300	1.17×10^6	69,000	480,000	14,000	6.1×10^{-2}
1,400	3.3	200	5,200	1.85×10^6	303,500	960,000	5,000	1.2×10^{-2}
1,700	49	12	1,836	653,400	12,360	57,600	272,000	1.0×10^{-5}
1,700	36.8	16	2,160	768,700	18,470	76,800	191,000	5.4×10^{-4}
1,700	29.4	20	2,460	875,500	25,900	96,000	144,000	3.2×10^{-4}
1,700	19.6	30	3,090	1.10×10^6	47,860	144,000	94,000	1.46×10^{-4}
1,700	11.8	50	4,200	1.49×10^6	117,740	240,000	46,000	4.1×10^{-3}
1,700	6.0	100	6,400	2.28×10^6	485,000	480,000	17,000	7.1×10^{-2}
1,700	2.9	200	10,000	3.56×10^6	4.07×10^6	960,000	7,000	1.8×10^{-2}

150 TON/DAY FURNACE CONT.
30" Module

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PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,000	196	4	148	215,300	1,310	76,000	56,000	3.78×10^4
1,000	157	5	170	247,300	1,750	95,000	46,000	2.24×10^4
1,000	131	6	186	270,600	2,100	114,000	33,000	1.46×10^4
1,000	98	8	216	314,300	2,900	152,000	23,000	7.55×10^3
1,000	78.5	10	240	349,200	3,630	190,000	17,000	4.3×10^3
1,000	56.1	14	294	427,700	5,620	266,000	11,000	1.9×10^3
1,000	39.3	20	360	523,800	8,750	380,000	7,000	7.7×10^2
1,000	15.7	50	625	909,300	31,000	950,000	2,200	1.0×10^2
1,200	180	4	208	302,600	2,360	76,000	59,000	3.7×10^4
1,200	144	5	235	341,900	3,050	95,000	50,000	2.3×10^4
1,200	120	6	261	379,700	3,820	114,000	36,000	1.46×10^4
1,200	90	8	304	442,300	5,300	152,000	25,000	7.55×10^3
1,200	72	10	340	494,700	6,750	190,000	18,000	4.3×10^3
1,200	51.3	14	413	600,900	10,400	266,000	12,000	1.92×10^3
1,200	36	20	500	727,400	15,900	380,000	7,000	7.7×10^2

150 TON/DAY FURNACE CONT.
30" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	14.4	50	900	1.31x10 ⁶	64,000	950,000	3,000	1.3 x 10 ²
1,400	165.4	4	300	436,500	4,700	76,000	66,000	3.93 x 10 ⁴
1,400	132.3	5	338	491,800	6,030	95,000	54,000	2.24 x 10 ⁴
1,400	110.3	6	372	541,200	7,500	114,000	39,000	1.54 x 10 ⁴
1,400	82.7	8	436	634,300	10,600	152,000	27,000	7.71 x 10 ³
1,400	66.2	10	490	712,900	13,700	190,000	20,000	4.55 x 10 ³
1,400	47.3	14	595	865,700	21,220	266,000	13,000	1.93 x 10 ³
1,400	33	20	720	1.05x10 ⁶	33,100	380,000	7,500	7.5 x 10 ²
1,400	13.2	50	1,300	1.89x10 ⁶	146,800	950,000	3,500	1.62 x 10 ²
1,700	196	3	534	776,900	14,800	57,000	148,000	1.0 x 10 ⁵
1,700	147	4	600	872,900	19,300	76,000	85,000	4.9 x 10 ⁴
1,700	117.7	5	660	960,200	23,970	95,000	68,000	2.78 x 10 ⁴
1,700	58.8	10	920	1.34x10 ⁶	52,600	190,000	24,500	1.85 x 10 ⁴
1,700	42	14	1,092	1.59x10 ⁶	80,310	266,000	15,000	9.25 x 10 ³
1,700	29.4	20	1,320	1.92x10 ⁶	131,031	380,000	9,000	2.24 x 10 ³
1,700	11.8	50	2,300	3.35x10 ⁶	654,420	950,000	5,000	8.5 x 10 ²

TABLE A7.7. 300 TON/DAY FURNACE

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
PROTOTYPE MODULE								
1,000	49.0	32	1,632	154,900	413	41,600	2.9×10^6	2.0×10^6
1,000	39.3	40	1,840	185,900	531	52,000	2.09×10^6	1.16×10^6
1,000	26.2	60	2,340	236,400	890	78,000	1.16×10^6	4.0×10^5
1,000	19.6	80	2,800	282,900	1,320	104,000	790,000	2.0×10^5
1,000	13.1	120	3,600	363,700	2,300	156,000	450,000	7.32×10^4
1,000	9.8	160	4,320	436,500	3,500	208,000	316,000	3.85×10^4
1,000	5.2	300	6,300	636,600	8,800	390,000	145,000	9.25×10^3
1,000	2.6	600	9,600	970,000	26,600	780,000	52,000	2.0×10^3
1,200	45.0	32	2,304	232,800	750	41,600	3.03×10^6	2.0×10^6
1,200	36.0	40	2,600	262,700	980	52,000	2.2×10^6	1.16×10^6
1,200	24.0	60	3,300	333,400	1,650	78,000	1.24×10^6	4.0×10^5
1,200	18.0	80	3,960	400,100	2,540	104,000	840,000	1.93×10^5
1,200	12.0	120	5,100	515,300	4,440	156,000	493,000	7.55×10^4
1,200	9.0	160	6,080	614,300	6,700	208,000	346,000	4.0×10^4

300 TON/DAY FURNACE CONT.
Prototype Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	4.8	300	9,000	909,400	18,215	390,000	179,000	1.0×10^4
1,200	2.4	600	1,380	1,394,000	62,000	780,000	53,000	2.0×10^3
1,400	44.0	30	3,030	306,200	1,240	39,000	3.4×10^6	2.24×10^6
1,400	33.0	40	3,640	367,800	1,860	52,000	2.3×10^6	1.08×10^6
1,400	22.0	60	4,620	466,800	3,200	78,000	1.3×10^6	3.93×10^5
1,400	15.5	80	5,520	557,700	4,800	104,000	880,000	1.93×10^5
1,400	11.0	120	7,200	727,500	9,100	156,000	528,000	7.71×10^4
1,400	8.3	160	8,640	873,000	14,200	208,000	374,000	4.09×10^4
1,400	4.4	300	13,200	1.33×10^6	46,000	390,000	177,000	1.0×10^4
1,400	2.2	600	21,000	2.12×10^6	211,000	780,000	60,000	2.3×10^3
1,700	39.2	30	5,070	512,300	3,500	39,000	3.73×10^6	2.16×10^6
1,700	29.4	40	6,240	630,500	5,600	52,000	2.58×10^6	1.08×10^6
1,700	19.6	60	8,280	836,600	11,200	78,000	1.52×10^6	4.24×10^5
1,700	14.7	80	10,160	1.027×10^6	19,000	104,000	1.06×10^6	2.16×10^5
1,700	9.8	120	13,560	1.37×10^6	42,000	156,000	659,000	9.25×10^4

300 TON/DAY FURNACE CONT.
Prototype Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,700	7.4	160	16,640	1.68×10^6	76,000	208,000	472,000	4.9×10^4
1,700	3.9	300	25,800	2.607×10^6	361,000	390,000	225,000	1.2×10^4
1,700	1.96	600	42,000	4.24×10^6	3.45×10^6	780,000	73,000	2.9×10^3
15" MODULE								
1,000	98.0	16	752	267,600	1,000	76,800	710,300	5.16×10^5
1,000	78.5	20	860	306,100	1,350	96,000	510,000	2.85×10^5
1,000	52.3	30	1,080	384,300	2,200	144,000	284,000	1.0×10^5
1,000	39.3	40	1,280	455,500	3,200	192,000	200,000	5.4×10^4
1,000	26.2	60	1,620	576,500	5,350	288,000	119,000	2.1×10^4
1,000	15.7	100	2,200	782,900	10,811	480,000	60,500	6.3×10^3
1,000	7.9	200	3,200	1.14×10^6	26,600	960,000	24,000	1.16×10^3
1,000	3.9	400	4,800	1.71×10^6	80,000	1.92×10^6	7,300	1.46×10^2
1,200	90.0	16	1,056	375,800	1,830	76,800	742,000	5.0×10^5
1,200	72.0	20	1,200	427,000	2,400	96,000	533,000	2.85×10^5
1,200	48.0	30	1,530	544,500	4,100	144,000	303,000	1.0×10^5

300 TON/DAY FURNACE CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	36.0	40	1,840	654,800	6,200	192,000	219,000	5.6×10^4
1,200	24.0	360	2,340	832,700	10,700	288,000	128,000	2.2×10^4
1,200	14.4	100	3,100	1.103×10^6	21,000	480,000	65,000	6.5×10^3
1,200	7.2	200	4,600	1.64×10^6	57,500	960,000	25,000	1.1×10^3
1,200	3.6	400	7,200	2.56×10^6	211,000	1.92×10^6	10,000	2.2×10^2
1,400	82.7	16	1,504	535,200	3,600	76,800	793,000	5.01×10^5
1,400	66.2	20	1,720	612,100	4,800	96,000	575,000	2.85×10^5
1,400	44.0	30	2,190	779,400	8,300	144,000	331,000	1.08×10^5
1,400	33.0	40	2,560	911,000	12,000	192,000	229,000	5.4×10^4
1,400	22.0	60	3,240	1.15×10^6	20,800	288,000	136,000	2.2×10^4
1,400	13.2	100	4,400	1.57×10^6	44,800	480,000	72,000	6.6×10^3
1,400	6.6	200	6,600	2.35×10^6	137,000	960,000	26,000	1.08×10^3
1,400	3.3	400	10,400	3.7×10^6	607,000	1.92×10^6	12,500	2.7×10^2
1,700	39.2	30	4,170	1.48×10^6	34,000	144,000	353,000	1.23×10^5
1,700	29.4	40	4,920	1.75×10^6	51,700	192,000	246,000	6.6×10^4

300 TON/DAY FURNACE CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,700	19.6	60	6,180	2.2×10^6	96,000	238,000	146,000	2.5×10^4
1,700	11.8	100	8,400	2.99×10^6	236,000	480,000	78,000	7.7×10^3
1,700	5.9	200	12,800	4.56×10^6	102,000	960,000	33,000	1.5×10^3
1,700	2.94	400	20,000	7.12×10^6	7.7×10^5	1.92×10^6	16,000	3.6×10^2
30" MODULE								
1,000	196.3	8	296	430,600	2,600	152,000	112,000	7.6×10^4
1,000	157.0	10	340	494,700	3,500	190,000	84,000	4.5×10^4
1,000	112.0	14	406	590,700	5,100	266,000	55,000	2.1×10^4
1,000	78.5	20	480	698,300	7,300	380,000	34,000	8.5×10^3
1,000	79.3	40	720	1.05×10^6	17,500	750,000	13,600	1.6×10^3
1,000	26.2	60	900	1.31×10^6	28,900	1.14×10^6	5,600	3.5×10^2
1,000	15.7	100	1,250	1.82×10^6	62,000	1.9×10^6	5,000	1.9×10^2
1,200	180.0	8	416	605,200	4,720	152,000	119,000	7.5×10^4
1,200	144.0	10	470	683,800	6,100	190,000	88,000	4.3×10^4
1,200	102.7	14	574	835,100	9,400	266,000	59,000	2.1×10^4

300 TON/DAY FURNACE CONT.
30" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	72.0	20	680	989,300	13,500	380,000	36,000	8.5×10^3
1,200	36.0	40	1,000	1.45×10^6	32,000	760,000	14,000	1.6×10^3
1,200	24.0	60	1,290	1.88×10^6	57,000	1.14×10^6	7,000	4.7×10^2
1,200	14.4	100	1,800	2.62×10^6	128,000	1.9×10^6	5,800	2.62×10^2
1,400	190.0	7	560	814,700	8,000	133,000	168,000	1.1×10^5
1,400	147.6	9	639	929,700	10,700	171,000	119,000	5.94×10^4
1,400	102.2	13	780	1.13×10^6	16,400	247,000	74,000	2.54×10^4
1,400	66.4	20	980	1.43×10^6	27,200	380,000	39,500	9.3×10^3
1,400	33.2	40	1,440	2.1×10^6	65,900	760,000	14,700	1.5×10^3
1,400	22.1	60	1,860	2.71×10^6	122,600	1.14×10^6	7,800	5.0×10^2
1,400	13.3	100	2,600	3.78×10^6	290,000	1.9×10^6	7,000	3.2×10^2
1,700	196.0	6	1,068	1.55×10^6	29,700	114,000	256,000	2.0×10^5
1,700	147.0	8	1,200	1.75×10^6	38,600	152,000	170,000	1.0×10^5
1,700	117.7	10	1,320	1.92×10^6	48,000	190,000	125,000	5.6×10^4
1,700	84.0	14	1,540	2.24×10^6	69,000	266,000	77,500	2.4×10^4

30" Module

[illegible]

TABLE A7.8. 600 TON/DAY FURNACE

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
PROTOTYPE MODULE								
1,000	39.3	80	3,680	371,800	1,062	104,000	4.16×10^6	2.24×10^6
1,000	31.4	100	4,200	424,400	1,410	130,000	3.05×10^6	1.31×10^6
1,000	26.2	120	4,680	472,900	1,780	156,000	2.36×10^6	8.48×10^5
1,000	17.4	180	5,940	600,200	3,010	234,000	1.33×10^6	2.92×10^5
1,000	10.5	300	8,400	848,700	6,540	390,000	721,000	9.25×10^4
1,000	5.2	600	12,600	1.27×10^6	17,600	780,000	290,000	1.85×10^4
1,200	36.0	80	5,200	525,400	1,960	104,000	4.37×10^6	2.24×10^6
1,200	29.0	100	6,000	606,200	2,700	130,000	3.24×10^6	1.31×10^6
1,200	24.0	120	6,600	666,900	3,300	156,000	2.48×10^6	7.71×10^5
1,200	16.0	180	8,460	854,800	5,760	234,000	1.41×10^6	2.93×10^5
1,200	9.6	300	11,700	1.18×10^6	12,300	390,000	744,000	9.25×10^4
1,200	4.8	600	18,000	1.82×10^6	36,400	780,000	316,000	1.93×10^4
1,400	33.0	80	7,280	735,600	3,700	104,000	4.6×10^6	2.16×10^6
1,400	26.5	100	8,300	838,600	4,960	130,000	3.38×10^6	1.23×10^6

600 TON/DAY FURNACE CONT.
Prototype Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,400	22.0	120	9,240	933,600	6,350	156,000	2.6×10^6	7.71×10^5
1,400	14.7	180	12,060	1.22×10^6	11,800	234,000	1.53×10^6	3.0×10^5
1,400	8.8	300	16,500	1.67×10^6	25,600	390,000	784,000	9.25×10^4
1,400	4.4	600	26,400	2.67×10^6	92,000	780,000	784,000	2.08×10^4
1,700	29.4	80	12,480	1.26×10^6	11,300	104,000	5.16×10^6	2.24×10^6
1,700	23.5	100	14,600	1.48×10^6	16,500	130,000	3.85×10^6	1.31×10^6
1,700	19.6	120	16,560	1.67×10^6	22,400	156,000	3.05×10^6	8.48×10^5
1,700	13.1	180	22,140	2.24×10^6	47,300	234,000	1.85×10^6	3.39×10^5
1,700	7.8	300	31,800	3.21×10^6	135,400	390,000	1.0×10^6	1.08×10^5
1,700	3.9	600	51,600	5.21×10^6	723,000	780,000	450,000	2.47×10^4
15" MODULE								
1,000	89.7	35	1,575	560,500	2,200	168,000	1.25×10^6	8.09×10^5
1,000	62.8	50	2,000	711,700	3,700	240,000	745,000	3.31×10^5
1,000	39.3	80	2,560	911,000	6,300	384,000	398,000	1.08×10^5
1,000	26.2	120	3,240	1.15×10^6	10,700	576,000	236,000	4.16×10^4

600 TON/DAY FURNACE CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,000	17.4	180	4,050	1.44×10^6	17,900	864,000	136,000	1.62×10^4
1,000	10.5	300	5,400	1.92×10^6	35,000	1.44×10^6	70,600	4.78×10^3
1,200	32.1	35	2,205	784,700	4,030	168,000	1.31×10^6	7.71×10^5
1,200	57.5	50	2,750	978,600	6,500	240,000	769,000	3.16×10^5
1,200	36.0	80	3,680	1.31×10^6	12,400	384,000	437,000	1.16×10^5
1,200	24.0	120	4,680	1.67×10^6	21,400	576,000	256,000	4.32×10^4
1,200	16.0	180	5,940	2.11×10^6	37,800	864,000	152,000	1.70×10^4
1,200	9.6	300	7,800	2.78×10^6	74,500	1.44×10^6	76,300	4.86×10^3
1,400	75.6	35	3,185	1.13×10^6	8,100	168,000	1.4×10^6	7.71×10^5
1,400	52.9	50	3,900	1.39×10^6	12,700	240,000	830,000	3.24×10^5
1,400	33.1	80	5,120	1.82×10^6	23,700	384,000	460,000	1.08×10^5
1,400	22.0	120	6,480	2.31×10^6	41,600	576,000	270,000	4.24×10^4
1,400	14.7	180	8,280	2.95×10^6	76,300	864,000	162,000	1.70×10^4
1,400	8.8	300	11,100	3.95×10^6	167,000	1.44×10^6	81,000	4.78×10^3
1,700	47.0	50	7,500	2.67×10^6	52,100	240,000	1.03×10^6	3.78×10^5

600 TON/DAY FURNACE CONT.
15" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,700	29.4	80	9,840	3.5×10^6	103,400	384,000	575,000	1.31×10^5
1,700	19.6	120	12,360	4.4×10^6	191,400	576,000	338,000	5.09×10^4
1,700	13.1	180	15,480	5.51×10^6	367,000	864,000	200,000	2.0×10^4
1,700	7.8	300	21,300	7.58×10^6	1.05×10^6	1.44×10^6	103,000	5.78×10^3
30" MODULE								
1,000	209.3	15	555	807,500	4,600	285,000	248,000	1.7×10^5
1,000	157.0	20	680	989,300	7,000	380,000	169,000	9.3×10^4
1,000	105.0	30	840	1.22×10^6	11,000	570,000	101,000	3.55×10^4
1,000	62.8	50	1,100	1.6×10^6	19,500	950,000	49,000	1.0×10^4
1,000	39.3	80	1,440	2.09×10^6	35,000	1.52×10^6	25,800	2.93×10^3
1,000	22.4	140	1,960	2.85×10^6	70,300	2.66×10^6	10,700	6.1×10^2
1,000	15.7	200	2,500	3.64×10^6	124,000	3.8×10^6	9,000	3.9×10^2
1,200	192.0	15	795	1.16×10^6	8,600	285,000	268,000	1.77×10^5
1,200	144.0	20	940	1.37×10^6	12,200	380,000	181,000	9.25×10^4
1,200	96.0	30	1,140	1.66×10^6	18,400	570,000	104,000	3.39×10^4

600 TON/DAY FURNACE
30" Module

PREHEAT TEMPERATURE (°F)	MASS FLOW RATE (lbm/min)	NUMBER OF STACKS	NUMBER OF MODULES	MODULE COST (\$)	INSULATION COST (\$)	TRANSITION COST (\$)	BLOWER COST (\$)	BLOWER OPERATING COST (\$)
1,200	57.5	50	1,600	2.33×10^6	38,600	950,000	54,500	1.0×10^4
1,200	35.9	80	2,000	2.91×10^6	64,000	1.52×10^6	28,500	3.08×10^3
1,200	20.5	140	2,800	4.07×10^6	139,000	2.66×10^6	12,500	7.17×10^2
1,200	14.4	200	3,600	5.24×10^6	256,000	3.8×10^6	11,600	5.24×10^2
1,400	189.0	14	1,120	1.63×10^6	16,000	266,000	313,000	2.16×10^5
1,400	177.0	15	1,155	1.68×10^6	17,300	285,000	292,000	1.77×10^5
1,400	132.0	20	1,350	1.96×10^6	24,200	380,000	195,000	9.25×10^4
1,400	88.0	30	1,680	2.44×10^6	39,000	570,000	116,000	3.6×10^4
1,400	53.0	50	2,250	3.27×10^6	74,600	950,000	60,000	1.08×10^4
1,400	33.0	80	2,880	4.19×10^6	133,000	1.52×10^6	29,500	3.01×10^3
1,400	19.0	140	4,060	5.91×10^6	304,000	2.66×10^6	14,000	8.5×10^2
1,400	13.2	200	5,200	7.57×10^6	587,000	3.8×10^6	13,900	6.4×10^2
1,700	196.0	12	2,136	3.11×10^6	59,000	228,000	511,000	3.93×10^5
1,700	157.0	15	2,340	3.4×10^6	72,900	285,000	385,000	2.24×10^5
1,700	117.7	20	2,680	3.9×10^6	99,000	380,000	253,000	1.16×10^5

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APPENDIX 8

MILLCREEK GLASS CORPORATION'S APPLICATIONS STUDIES - SUMMARY AND CONCLUSIONS

SUMMARY AND CONCLUSIONS (TERRA TEK INTERPRETATION)

A study was conducted by Millcreek Glass Corporation to ascertain industrial needs for high temperature heat recovery equipment. The aluminum, cement, copper, forging, glass, and steel industries were selected for detailed investigation due to the high temperature, energy intensive processes inherent to these industries. Following is a summary of the major conclusions of this study itemized by industry.

Aluminum Industry

1. The majority of the fuel-fired heating processes in the aluminum industry were found to yield relatively clean, intermediate temperature (less than 1800°F) flue gases amenable to heat recovery via currently existing metallic recuperators.
2. Exhaust flows from reverberatory melting furnaces can be at temperatures as high as 2100-2200°F. Fluxing and gas treating operations in these furnaces produce exhaust gases which are not compatible with metallic recuperators. In view of these factors, the potential for using the high temperature ceramic recuperator in this application exists.

Cement Industry

1. In the drying process of cement production, exhaust gases from the drying kiln are typically at a temperature of approximately 1100°F. Because existing metallic recuperators can be used in this application, no need for the high temperature ceramic recuperator exists.

Copper Industry

1. Heat recovery from slag will be the highest priority conservation measure to be pursued by the copper industry in the near future.
2. Concentrate drying and roasting processes produce exhaust gases amenable to heat recovery via conventional metallic heat exchangers.
3. Extensive retrofits associated with smelter combustion air preheating probably preclude heat recovery via this option.
4. Energy requirements for anode casting and cathode melting operations in reverberatory furnaces could be reduced by high temperature recuperation.

Forging Industry

1. The forging industry offers an opportunity for further development of recuperation systems. The nature of typical exhaust gas streams is such that minimal recuperator corrosion or fouling would be encountered in a temperature range suited to ceramic recuperators.

Glass Industry

1. Heat recovery from the exhaust from glass furnaces is currently being accomplished via the use of regenerative brick checkerwork systems. The high temperature ceramic recuperator could have application in the glass industry if it demonstrates an economic advantage over the presently used regenerator systems.
2. Retrofitting, leakage, plugging, cleaning, corrosion, manifolding, and furnace balancing are technical problems which need to be resolved by pilot plant testing and development prior to introduction of the high temperature ceramic recuperator to the glass industry.

Steel Industry

1. Steel processing involves a series of operations, many of which utilize high temperature furnaces using large amounts of energy.
2. Use of ceramic recuperators for heat recovery has been previously attempted in the steel industry. Operational problems related to recuperator leakage limited the success of these previous heat recovery attempts.
3. The high temperature ceramic recuperator would have application in the steel industry if essentially leak-free operation could be assured.
4. The current economic condition of the steel industry in the United States could limit investment in developmental heat recovery equipment.

APPENDIX 9

MILLCREEK GLASS CORPORATION'S MODULE JOINING, SEALING, CLEANING, AND SEPARATING STUDIES

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SUMMARY AND CONCLUSIONS (TERRA TEK INTERPRETATION)

Primary efforts of the development program were directed towards application of the high temperature ceramic recuperator in a glass furnace exhaust environment. Problems inherent to such recuperator application, resulting due to the nature of glass furnace exhaust flows (which contain significant amounts of condensable vaporized glass constituents), include recuperator cleaning and module separation. Another problem intrinsic to any application of the recuperator concept is that of joining and sealing of the individual modules. These three areas of concern, critical to recuperator development, were investigated by Millcreek Glass Corporation. The details of these investigations are included herein. Following are the major conclusions of the study.

Cleaning

1. The nature of deposits within the recuperator exhaust flow passage is dependent upon localized flow and material temperatures. The following three deposition horizons (and corresponding temperature bounds) have been identified for typical glass furnace exhaust compositions:

Temperature Range (°F)	Condensate Characteristics
>1700	free flowing liquid, heat exchange surfaces remain relatively clean
≈1650	highly viscous liquid/solid, bonds to recuperator surfaces
<1500	powder-like crystalline deposit, easily dislodged from surfaces

2. A review of candidate cleaning processes has indicated that the following two concepts offer the potential of effective cleaning of recuperators used in glass furnace applications:

Temperature Range (°F)	Cleaning Method
>1500	elevated temperature condensate burnout
<1500	swabbing of recuperator internals with a chain/brush apparatus

Module Separation

1. The modular recuperator concept enables replacement of cracked, corroded, or plugged portions of a recuperator assembly without the need for replacing the entire unit.
2. To enable selective replacement, a means of separating used modules (at times bonded together by glazed deposits) is required.
3. A mechanical separation device in conjunction with module lifting surfaces will enable separation of used recuperator modules.

Module Joining and Sealing

1. In previously used industrial ceramic recuperators, effective sealing of the recuperator flow passages has been a major problem.
2. Ceramic refractory fiber gasket material performed acceptably during the short term high temperature tests conducted during the current program. Further testing to ascertain the long term stability and the resistance to corrosive exhaust environments of these gasket materials is warranted.
3. Should refractory fiber gasket material prove to be unacceptable for long term usage in a glass furnace environment, module sealing via viscous glass materials should be investigated.

Joining and Sealing, Cleaning, and Separating Subsystems for the Helical Recuperator Applied to the Glass Industry

INTRODUCTION

Terra Tek and its subcontractor, Millcreek Glass Corporation, have been working since April of 1977 on a Department of Energy (formerly Energy Research and Development Administration) contract to develop a modular, helical high temperature recuperator concept. To date this concept has been aimed primarily at the glass industry, and although this may not be the only industry to which this concept is applicable, this remains a target industry for the concept.

To be economically viable, one of the primary obstacles a recuperator must overcome, especially in the glass industry, is that of fouling. In a glass container melting tank, for example, there are approximately 2-1/2 lbs. of condensable volatiles leaving the surface of the glass for every ton of glass melted.¹⁶ Thus in a 250 ton per day tank, approximately 625 lbs. of these volatiles will exit in the exhaust every twenty-four hours. These volatiles consist primarily

of sodium sulfate and some sodium carbonate. In addition to these volatiles, there is some amount of dust carry-over from the batching operation. The quantity of this carry-over varies greatly from almost none near the firing end of a well run furnace to several pounds per ton of glass melted near the charging end of a dirty furnace. As these materials move through the recuperator with the furnace exhaust, they are cooled and they settle and condense on the surfaces in the recuperator, leaving a condensate layer which is characteristic of the temperature of the collecting surface. At the very high temperatures (above approximately 1700°F), condensation and dust is a minor problem from the point of view of fouling. Condensation and carry-over settling in this region is in a liquid phase and tends to flow downward whenever possible, leaving relatively clean heat transfer surfaces in the recuperator. At approximately 1650°F is a region known in the glass industry as the sodium sulfate horizon¹. This is the region in which sodium sulfate solidifies. In a glass furnace flue or heat recovery system, this is the region of the greatest fouling problems. The volatiles condensing on the surfaces do so as a solid mass. If the flue or heat

recovery system is oriented so that the hotter regions are above the cooler ones (as in the glass furnace recuperators) the liquid condensate and melted dust from the hotter regions above flow down into the sulfate horizon zone and solidify also. At lower temperatures, starting at approximately 1500°F, the condensation forms a fuzz of crystals (see Figure 1) a little like hoar frost on a cold, foggy morning. These crystals build upon one another in layers and in the Millcreek Glass Corporation's test furnace flues, thicknesses on the order of half an inch have accumulated in approximately four months.

In a recuperator two problems arise from these fouling condensates. The more serious is plugging of the passages of the recuperator. The other is the insulation effect of the additional layers of material on the heat exchange surfaces. Clearly, if the recuperator passages become plugged, it not only ceases to act as a recuperator, it also ceases to act as a flue and the exhaust gases from the furnace must find someplace else to go. The results can be catastrophic to the operation of a furnace. The insulation effect causes the recuperator to operate at lower and lower

Figure 1. Low temperature condensate of materials exposure furnace flue at Millcreek Glass Corp.

effectiveness as the heat transfer becomes more and more impeded by successive layers of material on the heat exchange surfaces. Assigning thermal conductivity values to this material would be difficult at best because of its inhomogeneity. However, it is worth examining the quantities of material that are being discussed. Approximately 2-1/2 lbs. of sodium sulfate are produced for every ton of glass melted.^{16,23} It is estimated that approximately 10-20% of this condenses in a standard regenerator.²³ This fraction will be greater for a system that cools the exhaust more than the 1100-1200°F output of most regenerators. In addition to the condensate, over 50% of the carry-over dust settles out in the regenerator.^{4,23} An average size container glass furnace today melts 250 to 300 tons of glass per day, so this implies a minimum of 50 to 100 pounds of deposit per day into the recuperator system. Because of this, periodic burning out of glass furnace regenerators is required to keep them from becoming plugged. And ultimately even this fails and rebuilding is required. (This will be discussed in greater detail below.)

The helical recuperator concept under development in this program has two design features which make it relatively promising for dirty environments

such as a glass furnace. (The basic design concept is shown in Figure 2.) The first of these features is the relatively large flow passages. In the prototype 8 inch diameter model with which the experimental work has been done to date, the flow passages have roughly a half ellipse shaped cross section with the two axial dimensions approximately 3.5 inches each (see Figure 3). It is anticipated that full-scale industrial models will be at least 24 inches in diameter. In that case the major axis of the half ellipse will be approximately 11 inches (allowing for wall thicknesses) and the minor axis can be any size up to approximately 11 inches depending upon the pitch chosen for the helical interface. Of course there is a payoff, for the larger the flow passages, the less heat transfer area there is per unit volume of passage and the less the heat transfer per unit length of recuperator. But relatively large flow passages are required for very dirty environments such as the glass industry. And even these, in time, become plugged.

The second design feature of this recuperator concept, which makes it suitable for the glass environment, is its modularity. This feature has sev-

Figure 2. Schematic of the modular counter-flow recuperator with a helical interface.

Figure 3. Cutaway diagram of helical recuperator and approximately half-ellipse flow cross section.

eral potential or hoped-for advantages. But the one potential advantage of interest in the present discussion is that of partial replaceability. It is hoped that if one section of the recuperator becomes excessively and irreparably fouled (or cracked or broken), that section can be removed and replaced. This modularity, however, creates the need to seal the joints between the modules. With ceramic recuperators in the past, this has been a serious problem.

Thus, three problems that need to be solved in the development of the helical recuperator are (1) cleaning, in place and on-stream as a primary preference or in place but not during operation as a secondary preference, (2) sealing, and (3) separation of used modules and replacement of sections of the column. The purpose of this report is to examine these problems, review historical solutions to similar problems as well as other potential solutions, to choose the most promising potential solutions for the present problem, and to present basic plan outlines for testing these solutions.

CLEANING

As has been stated above, in a heat recovery system on a glass furnace there are really three zones of condensation and fouling. The first zone, from the ducts leading from the furnace down to the sodium sulfate horizon at approximately 1650°F, is characterized by liquid condensation and the settling and melting of any carry-over dusts from the batching operation. Because these fouling agents are in a liquid state, they tend to drain away from the surface on which they were deposited whenever the geometry of the surface allows this. Hence a buildup of fouling agents in this region is not a problem. Some corrosion and dissolutions of refractories in this region may take place, and then these materials contribute to the volume of the fluid that is draining away from the region. Below approximately 1650°F, the sodium sulfate in the flue gases begins to condense out as a sticky solid coating on the heat-absorbing refractories of the system¹. If this region is geometrically below the hotter region, the liquid condensates and any dissolved refractories from the upper region drain to this level and solidify. This is the region that creates the major problem for keeping regenerators open in the glass industry.

As the temperature decreases, the condensates gradually change in character. Below approximately 1500°F, observations at Millcreek Glass Corporation show that the condensates form a light, downy crystalline growth. This growth is easily removed from the host surface by light brushing, rubbing or, if the growth is considerable, blowing or vibrating. An observation of this material leaves little doubt that it should be a good insulator, much like a wool blanket, but no actual measurements of its heat transfer have been done. Indeed, it is doubtful that any such measurements would be very meaningful because of the variability of the density of the material. But this assumed insulating quality means that the material would be very undesirable in any quantity in a recuperator. The zone of this light crystalline growth extends to the cool end of the recuperator with quantities of condensation apparently tending to taper off somewhat as temperatures decrease.

Thus, while there are three condensation zones in a glass furnace recuperator, only the cooler two zones can be expected to require cleaning. In studying potential cleaning methods, several approaches were used:

1) Methods used today for cleaning glass regeneration systems were investigated.

2) Methods of restoring metal recuperators used in segments of the glass industry were reviewed.

3) Methods used for cleaning recuperation systems in other industries were studied.

4) Methods used for cleaning exhaust streams for EPA requirements were examined.

5) New methods for solving this problem were brainstormed.

Some of these approaches reviewed systems and problems rather far afield of the present problem but occasionally a suggestion was found. Here we shall review a few of the systems of interest and point out some aspects of those systems which, with suitable modifications, might be applicable. We will then list a series of potential approaches to the problem.

Glass Furnace Regeneration:

Except for the very small unit melters, glass furnaces in the container and plate glass industries are of the regenerative type (see Figure 4). That is, on each side of the furnace and connected to it by

Figure 4. Illustration of side port container glass furnace layout, including checkerworks; plan view of side port flat glass furnace.

firing port ducts, there is a large chamber filled with a loosely laid stack of bricks. These stacks of bricks are called checkers or checker works. The furnace is fired with air for the burners being blown up through the bricks in the chamber on one side of the furnace. The air then goes into the furnace, being mixed with fuel as it does so. Combustion takes place as these gases pass across the furnace. The combustion products then exit through the opposite chamber, heating the checker stack. Every fifteen to thirty minutes the direction is reversed so that the bricks heated by the furnace exhaust in one half-cycle, preheat the combustion air during the other half-cycle.

In newer furnaces, one trend has been toward what are called multiple pass checker systems. Here each checker chamber is enlarged and divided into two or more chambers with the air flowing down through one chamber and up through the next, and then into the furnace. This allows a more extensive checker system without excessive excavation to make the checker chambers deeper. It has another advantage as well. As has been explained above, dust and volatiles settling

and condensing in the hottest zone of the recuperator form a liquid slag which tends to flow downward into the checker work. If the bottom of the first (hottest) pass of the system is designed to be just about at the sodium sulfate horizon level, then all of the condensation and dust deposit from above drains out the bottom of this checker stack. It collects in the passages under the rider arches (which support the checker system) where it can be relatively easily removed. Furthermore, if the entire furnace exhaust is mixed in these passages and redistributed before moving up through the second pass, much of the material that is going to condense will have time to do so and most of the problems are eliminated for the second and any subsequent pass checkers. Indeed, all passes except the first should last indefinitely¹.

In both single pass and multipass furnaces, when a checker does become too plugged up to be effective, the technique used is called "burning out". In a multipass system, burners are usually built in below the rider arches under the first pass. These can be turned on to heat the entire first pass well enough that the congestive material melts on through⁵. With single pass furnaces, torches are brought in from the sides to melt

out the congested areas and move the material down into the cooler part of the stack⁹. Eventually this becomes a losing battle because the entire midrange of the stack becomes too congested to effectively clean out. It is for this reason that in the past regenerators have often been the limiting factor in the length of a furnace campaign.

These methods of dealing with the problems of glass furnace regenerators suggest several approaches to consider for the ceramic recuperator. A recuperator can be quickly heated by stopping the flow of cool air and continuing the flow of exhaust. If the recuperator is made to withstand the temperatures and transients associated with this operation, this might be a way to consider cleaning it on a periodic basis. The multipass regenerator suggests a similar arrangement in the recuperator in which much of the sodium sulfate might be collected and into which glaze from the hottest parts of the recuperator might drain. This adds some considerable complications to the recuperator concept, for the recuperator must be divided into at least two separate parts and the number of special manifold end parts is at least doubled, but

it has an elegance of operation that periodic cleaning of the recuperator itself does not have.

Metallic Glass Furnace Recuperators:

There is apparently no attempt being made to clean metal recuperators for glass furnaces on stream. Glass furnace condensates that have not been heated enough to form a glass, are water soluble. Thus the technique often used is to remove the recuperator from the furnace and scrub it or flush it with water. At Lennox Glass Company we observed this being done manually with long handled brushes. These recuperators are severely temperature limited and normally operate close to the limit of the metal from which they are made¹² so that burning out of the condensates is not feasible. (Two of the metals used in such recuperators were tested for this environment in the materials testing done in the helical recuperator program at Millcreek Glass Corporation. Inconel 600 was oxidized severely in this testing at 1740°F. This appears to be close to or above the material temperature limit for metal recuperators in a soda lime glass application.)

For the recuperator under development in this

project, the cleaning alternatives suggested here are certainly less desirable ones. But if need be, a recuperator could be taken off stream, cooled and cleaned, either as a whole or as individual modules using water either alone or with some kind of scrubbing system. And, of course, one of the hoped-for advantages of modularity in this system is that parts of the system which fail first can be replaced without the delay time and expense of installing an entire new unit.

At this time, estimating the time and costs involved in such a maneuver is difficult. It might be estimated that a stack could be cooled and dismantled in a day and a replacement installed and reheated in another day. If no intact cleaning is done, we can estimate a condensate deposit of 20% of the 2-1/2 lbs. in a stack per day and perhaps an average of 1/2 lb. of batch carry-over for an 8" diameter stack per day. Thus each stack might be expected to be effective (less than 25% filled with deposits) approximately 40 days before replacement was required. Thus, if each stack is down two days every 40 days, a minimum redundancy of 5% must be designed into the system.

Alpha Glass Corporation has been doing research for the past several years on systems to extract heat from the exhaust of container glass furnaces by means of Brayton cycle turbines. They have been working on several systems and they are complex enough that we shall not take the space to go into them here; but they require the extraction of heat from the exhaust of the glass furnace by means of a recuperator. Because of their turbine materials as well as their heat exchanger materials, they have limited themselves to under 1500°F. In this range they have had to devise techniques for cleaning their high effectiveness heat exchanger. This heat exchanger is designed and made by Garrett Airesearch. An example of the type of heat exchanger is shown in Figure 5. Alpha Glass found that the settled particles and condensates could be cleaned out of the heat exchanger by means of an air cannon, a device that generates a self-propogating pulse of air through the heat exchanger, used in conjunction with a horn which set up sonic vibrations in the metal exchanger.¹⁰

It is known that the air cannon pulse does not travel well around corners²⁷ and the helical recuperator

Figure 5. Metal fin type recuperator
by Garrett Airesearch.

passages are essentially a continuous corner. Thus, it is believed that this concept could not be applied directly to the helical system. However, a blast of air could be blown through the system and perhaps achieve the same result in the low temperature region. Care must be taken not to damage the sealing system in the process. Sonic vibrations could be used in conjunction with that, if necessary, but application of these vibrations to each module might well require a separate probe through the insulation to each module because each module may be essentially isolated from its neighbors by the joint seals. Some empirical work would be required to answer these questions.

Recuperators in Other Industries:

In reviewing recuperator cleaning practices in other industries we see some of the same practices as in the glass industry. In the steel and power industries metal recuperators are removed and cleaned using very high pressure (up to 1,000 psi) streams of water to lance out the deposits around and in the tubes.^{26,25} More novel systems are to be found also. Some places in industry chains are dropped down into

gas to gas heat exchangers to knock out deposits. In liquid systems "pigs", or floating swabs, are forced by the heat exchange medium through the heat exchanger tubes to wipe out deposits.¹⁵ Especially in liquid systems chemical cleaning agents are periodically flushed through the system, or the chemistry of the exchange medium is modified to eliminate or limit the problem.^{25,11} Of course, the design of the heat exchanger itself can influence the rate of fouling. In general smooth surfaces, high fluid velocities and probably high turbulence lessen the rate of deposition.²²

Some of these ideas also suggest approaches to the present problem. Perhaps modified chains could be worked down through a recuperator to clean it. Figure 6 shows the conception of a chain with attached brushes of a type which might be used to clean the fluffy condensate seen below approximately 1550°F. Balls or briquettes of any one of several materials could be rolled down through a recuperator, cleaning not so much by scrubbing as by knocking and rolling through. Modifying the chemistry of the exhaust flow so that no condensation takes place or so that the condensation drains away and is not a problem is also

Figure 6A.

Figure 6A. Cutaway front view,
 Figure 6B. Top view artists conception
 of cleaning chain. Chain shown here in
 a chain of small carts, each carrying a
 brush hoop.

Figure 6B.

suggested. The chemical system necessary to do this is not known at the present time. As for off-stream flushing out of the condensates, throughout much of the recuperator this could be theoretically accomplished with hot water, though there may be agents that could speed the process. This, however, might create problems for the joint sealing system. Options on this system are discussed later in this report.

In terms of the general design criteria which effect the accumulation of fouling, this project has some freedom but not complete freedom. Some of the materials under consideration appear to have less tooth for the development of fouling than others. The silicon carbides and high purity aluminas have considerably less than the coarser grained ceramics such as the ECP materials and the Harbison-Walker materials tested by Millcreek Glass Corporation.

The fluid velocities are limited by the allowable pressure drops in the system and these are probably limited primarily by leakage in the system. If turbulence is not fully developed, one way to increase it is to introduce finning, but fins are liable to increase the fouling problems rather than decrease

them because they tend to create dead spaces in which fouling can develop rapidly.²² Other than high velocities, the other way to increase turbulence is by the overall design. In this respect the circuitous path in the design under discussion is good, inducing full turbulence at a Reynold's number near $Re=3000$. In this particular recuperator the surface smoothness, velocities, and turbulence can be expected to have a greater effect in the range of solid condensation than in that of liquid condensation because of the solid to solid bond in the former versus the liquid to wettable solid bond in the latter.

Air Pollution Control Systems:

Pollution control devices reviewed fell into four categories: static precipitators, cyclone entrapment devices, bag houses and scrubbers.¹⁴ Static precipitators collect particles by giving them a charge and then attracting them to oppositely charged collectors (see Figure 7). Cyclone entrapment systems remove particles from a gas stream by swirling the stream and separating out the particles centrifugally (see Figure 8).

Figure 7. Schematic of a wire and pipe static precipitator.

Figure 8. Schematic of cyclone precipitator.

Figure 9. Schematic of section of ceramic tile type recuperator. Air flows vertically, exhaust horizontally.

Bag houses are basically large filtration systems in which the gas stream is pushed through cloth bag filters to collect entrained particles. And scrubbers remove particles from the gas stream by putting the stream through a dense shower of water particles which wet the entrained particles and settle out with them.

In general these concepts have two hurdles to overcome in going from the field of pollution control to high temperature recuperator cleaning. The first is temperature and the second is the fact that the majority of the fouling agents enter the recuperator as a gas, condensing into a liquid or solid phase inside the recuperator. This implies that if you are going out to catch the fouling agents by particle separation techniques, you will have to do it as the particles are formed. Furthermore, since the condensation is a result of the cooling that is taking place, and since that cooling is being forced by cooler walls and interface throughout the recuperator, it is to be expected that to a large degree the fouling agents never are stream-born as particles. Thus a filtration device can only be expected to work if it remains cooler than the stream and only to the extent

that it cools the stream, inducing condensation of the foulants. But this cooling defeats the purpose of the recuperator.

An example of this is seen in a recent patent by William P. Mahoney and assigned the Ball Corporation¹⁵ which discusses a scrubbing system for glass furnaces using an aqueous solution of sodium silicate as the scrubbing liquid. As droplets of the sodium silicate solution pass through the exhaust stream from the furnace, the water evaporates, leaving first a sticky sphere which is highly attractive to glass batch carry-over and glass volatiles. Finally all of the water is evaporated leaving particles of a solid prill which can be collected. This prill can be mixed into new glass batch to recycle the captured dust and precipitate back into the glass. This system should operate well in a high temperature environment but one reason this system is effective is the cooling of the exhaust stream first by the introduction of the sodium silicate solution and then by evaporation of the water from the particles. Clearly this would defeat the object of a high quality heat recycling system such as the one being discussed in this paper.

For the reasons explained above, i.e., temperature

limitations and condensation behavior, the pollution control concepts reviewed have limited suggested applicability. A simple trap might be used to largely eliminate batch carry-over in applications where this does prove to be a problem. One method for doing this is discussed in a patent by H. R. Swift and Thomas B. O'Connell assigned to the Libby-Owens-Ford Glass Company.²¹ This patent shows generic formulas for glass coatings that are sticky at their respective temperature ranges. Formulas are given for ranges down to 400° to 600°F and up to 3000°F. These coatings can be applied to any suitable refractory substrates (metal plates, bricks, screens, or what-have-you) to collect particles. As a trap for batch carry-over before it enters the recuperator, the collected dust will melt to replenish the coating, and if provisions are made for the successive layers of coating to drain harmlessly away, this could largely stop this type of contaminant from entering the recuperator.

Electrostatic Precipitation is an intriguing concept because of several reasons: 1) It is a continuous, on-stream process. 2) It has the ability to separate very small particles from an exhaust stream (there is no theoretical lower limit to the size of particles

that can be collected). 3) High efficiency (greater than 99% in some cases) can be obtained. 4) Pressure and temperature drops are small (the pressure drop seldom exceeds 0.5 inch water column. 5) Maintenance costs are usually relatively low, in part because there are few, if any, moving parts. 6) Corrosive materials can be collected with special construction. 7) Very large flow rates can be handled. 8) The power requirements for flow handled are relatively low because the energy used is applied directly to the particles to be collected. (For example, in one application, the actual power required to clean 500,000 cubic feet of gas per minute at 95% efficiency, including draft loss, is about 65 kilowatts.) Furthermore, the temperature capability is high for air pollution control equipment. Temperatures up to 700°F are normal; special designs have been used for temperatures as high as 1300°F⁸. But this temperature is well below the sodium sulfate horizon where particles become a serious problem in the glass environment. We have been unable to confirm that 1300°F represents a real barrier to this technology, but it is clear that some development work would be required to reach

the temperatures required for this application. But, again, the real difficulty seen with this application is that it is not capable of removing gaseous materials. Because there is no place in the recuperator where all of the foulants are in particulate form, there is no single phase where a static precipitator could be placed to remove all of the foulants. With the low temperature and pressure drops that the system has, it might seem possible to place several of these devices along the stack, but they are rather expensive. In air pollution applications they are typically the option with the highest initial cost.

Brainstorming:

The final source of cleaning concepts used for this study has been brainstorming on the part of the participants. Suggestions not mentioned earlier in this report have included revolving screen filters, ultrasonics, sonic and mechanical vibrations, air jets, and air blasts.

It has been proposed that a series of circular screens which revolve with one side spanning the exhaust stream could be used to capture the condensates. These condensates could be vibrated or scraped off on

the side of the filter outside of the exhaust stream. Objectively evaluating a relatively novel and untried approach such as this is difficult without empirical data. It is not clear how many such filters would be required nor can one reliably estimate how much heat they would have to remove from the exhaust stream to operate effectively. But some problems are apparent. In the environment under discussion, all of the possible materials for such a screen filter have some severe temperature limitations. Inconel 600 was oxidized severely in Millcreek Glass' materials testing at 1740°F. It appears to hold up well at 1450°F. Just what its threshold is has not been determined more accurately than that at this time. But if a screen material cannot be found to operate up to the 1650°F region, there will be a major problem area that the concept cannot touch. The screens could conceivably be kept cool by relatively rapid rotation, but the cooler they are relative to the stream they are filtering, the more of a drain they are on the recuperator. Another major problem appears to be sealing the recuperator around the filters. Typically, the exhaust stream will be under a negative pressure so

the tendency is to draw air in around the filter. This dilutes the exhaust, lowering the potentially recoverable energy. The filters could be enclosed in air-tight compartments, depressurized only when emptied of accumulated condensate, but the system begins to appear rather elaborate and potentially difficult to maintain.

Whatever its disadvantages, the filter concept has the advantage of being an on-stream approach. To the extent that it would work, it would do so during operation of the recuperator requiring neither shutdown nor removal for a special operation.

Another such on-stream approach is the proposed "air jets" concept. Here the accumulated condensate would be blown off of the inside of the recuperator by a multitude of built-in air jets. Presumably this would be a periodic operation, diluting the exhaust stream only momentarily. The assumption is made that the flow of the exhaust stream would carry this loose condensate out of the recuperator. This concept clearly appears limited to the range below 1450°-1500°F where the deposition is a light powdery material. Above this range it appears unlikely that the air jets could remove the deposit, or even be kept open to op-

erate as air jets. This limitation aside, we see numerous problems with maintaining a multitude of high temperature air lines that penetrate the recuperator housing, insulation and exterior walls of the modules in enough places that a blast of air really would clean the entire inside surface of the exhaust side of the recuperator.

The region of light powdery deposition has invited several more ideas. Indeed, as observed in the Millcreek Glass Corporation exposure furnaces, it seems as if almost anything will knock this deposition loose. And there is need to get rid of it, for, as explained above, it most certainly is a good insulator capable of diminishing the effectiveness of the recuperator. One potentially on-stream approach is to simply vibrate the stack, either mechanically or sonically. The questions relating to this approach are two: 1) Will the powdery deposition vibrate off at a frequency and intensity that are not detrimental to the rest of the system? In a system in which leakage and cracking are two major long-range concerns, any talk of vibrating sounds threatening. 2) Will the deposition that is removed from the chamber walls

really be swept from the recuperator by the movement of the exhaust stream? There is no question that at any reasonable exhaust velocity much of it will. Most of it is very fine down-like crystals which will blow with a gentle breeze. However, in the boundary region (wall temperatures near 1300-1500°F) we have observed that the deposit will start out light and feathery and as deposition continues, it becomes more dense until at some distance out from the wall it becomes glazed. Apparently the feathery deposit insulates later deposition from the cool wall and ultimately new deposition is considerably hotter and of a different character from the old. If this is then knocked off the wall, there is a mixture of down-like crystalline material that will blow away with a whiff with coarser grains and even chunks of glaze.

One cleaning concept that would reach further into this boundary region is an in-place but off-stream technique. Once a stack has been cooled, a blast of air, possibly with entrained sand, other solid particles or water could be used. One caution is that there must be certainty that the pressure and materials of the blast will not disturb the joint seals and cause subsequent leakage in the recuperator.

The word ultrasonics has come up in discussions

of cleaning. Generally ultrasonics are used to clean parts immersed in appropriate cleaning fluids. Such a liquid wets the object to be cleaned as well as the material to be removed. The liquid is then cavitated and the foreign material is forced from the object by the violent action of the fluid. Apparently there is no reason modules could not be cleaned by this method, but it is a relatively high technology, off-stream, out-of-place technique and for this reason is not highly desirable.

Summary of Cleaning Concepts:

Table I gives a summary of the cleaning concepts considered for cleaning of the helical recuperator in a soda lime glass furnace environment. Included is the temperature range for which each concept appears applicable in this environment. It will be observed that few of the concepts are applicable throughout the recuperator. Thus it is possible that more than one cleaning technique will have to be used to cover the entire recuperator. Clearly on-stream concepts are more desirable than off-stream ones; surely in-place ones are preferable to out-of-place ones; in general, continuous concepts are preferred to periodic

TABLE I

POTENTIAL RECUPERATOR CLEANING CONCEPTS

<u>On-Stream, Continuous</u>	<u>Applicable Range</u>	<u>Notes</u>
Condensate collector	Near 1600°F	Requires manifolding recuperator into and out of collector. Requires periodic cleaning out.
Smooth surfaces	Below 1500°F	Not expected to be fully effective.
High fluid velocities	Below 1500°F	Limited by pressure drops and leakage of recuperator.
High turbulence	Below 1500°F	Turbulence should be fully developed. Not expected to be fully effective.
Briquettes or pigs	Below 1500°F	Possibly periodic. Could be batch preheat, otherwise entails some heat loss.
Static percipitators	Below 1300°F	Requires several shapes to be fully effective, even below temperature ceiling.
Particle trap	Top	For batch carry-over only.
Screen filters	Below 1500°F	Sealing problems make system more complex.

(Continued following page)

TABLE I (Cont.)

<u>On-Stream, Periodic</u>	<u>Applicable Range</u>	<u>Notes</u>
Burning out	to 1500°F or throughout	Requires sufficiently refractory materials.
Chemical cleaning	Below 1650°F or throughout	Chemical system not presently known.
Briquettes or pigs	Below 1500°F	Possibly continuous. (See above)
Chains	Below 1500°F	Requires chain of special design.
Air jets	Below 1500°F	Air orifices and lines severe drawback.
Vibration	Below 1500°F	Effects on joints questionable.
<u>Off-Stream, In Place</u>		
Air blast	Below 1650°F	Could carry entrained cleaning agents. Effect on joint seals questionable.
<u>Off-Stream, Out-of-Place</u>		
Washing	Below 1650°F	
Rebuilding	Throughout	
Ultrasonics	Below 1650°F	

ones; and, especially at this stage of development, simpler, lower technology concepts are preferable to complex, high technology ones.

Recommendations for Testing:

A survey of Table I shows two concepts which appear simpler and more certain of success than the others. One of these is burning or melting out. It is probable that this concept could be used alone to clean the entire recuperator if this were desirable. This, however, has some costs because this means that the entire recuperator would have to be made of refractory enough materials to tolerate this operation. It would seem that much of this expense could be saved if another method were used for cleaning the range below 1500°F. The other method is applicable to this lower range. It is the use of special chains with brushes to brush off the fluffy condensate (see Figure 6). While these chains would require special fabrication, the concept is simple and appears fool-proof.

It is expected that the procedure for using these two cleaning techniques would be to use the chains first to clean out the lower temperature range. Then

the 1650-1500°F range would be cleaned using the burning out process. It is anticipated that this could be accomplished by stopping or diminishing the flow on the air side of the recuperator, thus removing that air's cooling effect.

A test plan for these two systems on a laboratory scale furnace might involve the following steps:

1. Instrument a recuperator so that a temperature profile to $\pm 25^{\circ}\text{F}$ can be determined.
2. Run the recuperator using a simulated soda lime glass furnace flue environment until its effectiveness drops 10-15% due to fouling.
3. Dismantle the recuperator step by step to observe the foulant distribution.
4. Reassemble the recuperator and, if necessary, reexpose the recuperator to the environment until the same impairment or other chosen impairment is observed.
5. Clean the recuperator using a brush chain system to the 1600°F temperature level.
6. Dismantle the recuperator to the level at which the brush chain system became ineffective. Note the operating temperature at that

level.

7. Reassemble and commence operation of the recuperator; measure effectiveness.
8. Diminish air side flows through the recuperator until the minimum temperature level to be cleaned reaches a temperature $T=1700^{\circ}\text{F}$. Hold this temperature for one hour, then return to normal operation.
9. Measure "as cleaned" effectiveness.
10. If necessary, return to burn out conditions trying either higher temperatures or longer burn out times. Measure the effectiveness as a means of determining satisfactory cleaning.
11. Disassemble the recuperator and visually inspect cleaned parts.

MODULE JOINING AND SEALING

Introduction:

Most of the industrial experience with ceramic recuperators to date has been in the steel industry with Amsler-Morton type recuperators (see Figure 9). The tiles used in these recuperators have been made out of fire clay materials. Today cross flow tube type recuperators are replacing the old Amsler-Morton type (see Figure 9). In all of these recuperators leakage has been the greatest source of deterioration. In part, this is because the common ceramic materials tend to crack after a few temperature cycles. But it is also because as the ceramic materials expand and contract, the joints between the ceramic parts open up and leaks develop. Together these two mechanisms quickly begin to degrade the performance of a new ceramic recuperator. Because of this problem, this type of recuperator is fitted with oversized fans so that operation can continue at high leakage rates. Then hot air-induced draft fans are used downstream from the recuperator to reduce the pressure in the recuperator and thereby cut down on the leakage rate.²⁴

The problem of cracking is being and will continue to be addressed in materials, material stress and transient studies in the recuperator program. The purpose of this discussion is to address the problem of joint sealing. Five approaches to the problem will be discussed: (1) mechanical methods, molded or machined into the parts to be joined, (2) gaskets of fibrous, non-melting materials, (3) ceramic fillers and adhesives, (4) materials designed to be viscous at operating temperatures, and (5) compliant interlayers. These are not entirely independent categories and, in fact, any ideas of the last four categories would normally be combined with some alignment method as a minimum from the first category.

Mechanical Methods:

Numerous mechanical systems can be devised for joining and sealing a pair of cylinders with webs through them. But, going beyond a mere rough alignment system requires close tolerances on the mating parts. Modules made to date do not have these. As formed, tolerances in the silicon carbide modules are $\pm .06$ inches in flatness of the mating surfaces and in roundness of the cylinders. Tolerances on the ECP

modules are similar. With these tolerances, something will have to be used to fill the spaces. And, while these tolerances can be expected to improve somewhat as the bugs are worked out of fabrication of the modules, to expect as fired tolerances in ceramics comparable with machined tolerances in metal is no more reasonable than expecting similar tolerances in sand cast metal. Thus, if there is a way of mating and sealing modules without some form of gasketing or filling, it will surely require machining. This is an expensive process which has been avoided on ceramic modules to this point. Should it prove necessary for any application to machine parts in order to avoid gasketing or filling, certainly the least expensive type of machining is flat grinding of the mating surfaces. But this is possible only if all mating surfaces are the most protruding surfaces on their respective faces. This creates a problem similar to mating male to male or female to female formed faces. One solution to this problem is shown in Figure 10. Here the joining faces are male formed and machined flat. Alignment is maintained by a ring which fits into the niche created when the two faces are placed together. Rotational alignment is maintained by keys on the ring which fit into corresponding keyways in the modules.

Figure 10. Schematic of male to male joining system.

The required tolerances in roundness are no better than those obtainable without machining. But the system does have one disadvantage: it requires the manufacture of two different parts for each module-- the module and the alignment ring.

Because of the expense of the necessary machining, other types of purely mechanical seals, such as threaded or tapered joints, or male-female fitting butt joints (in form like the ones used in this program to date) do not appear feasible, at least for the ceramic parts. In fact, these types of joints are seldom used in gas tight metal joints without some kind of gasket or filler. For use in conjunction with one of the sealing methods to be discussed below, two of these systems appear feasible. The male-male joint with ring discussed above and the male-female butt joint being used today. The other systems are rejected because they require closer tolerances than the ceramics suppliers can be expected to supply.

Non-melting Ceramic Fibers:

Non-melting ceramic fibers in one of their several forms appear to offer promise as gasket material for the joints of the recuperator. Materials of this type

have been used as seals in conjunction with silicon carbide tubes in a type of horizontal tube heat exchanger used in recent years in the steel industry.^{13,8}

And, indeed, this is the type of seal used on the warm and hot flow prototypes tested in the present phase of the subject program. These materials are made by several manufacturers in a variety of forms. To date, in the helical recuperator project, three materials have been used in testing: 1) a one-eighth inch Kaowool felt made by Babcock and Wilcox and with a constant maximum use temperature of 2300°F, 2) a one-eighth inch Fiberfrax felt made by Carborundum and with a maximum continuous use temperature of 2600°F, and 3) a braided sleeving material made by 3-M and called AB-312. This last material has been used to 3000°F in certain space shuttle testing applications.²⁰

It is believed that the ceramic fiber type of joint sealing material has great promise, at least in relatively clean applications. However, most of these materials have poor resistance to alkali attack and hence their endurance in a glass furnace environment is questionable. Furthermore, they have been used extensively in recuperators in the steel industry, and

even in this relatively benevolent environment, these materials tend to embrittle. Vibrations in the system then knock them out of place.¹³ As far as ease of separation and replacement of modules, this type of material seems to hold the greatest promise.

Ceramic Adhesives and Fillers:

Another possibility in joint materials are ceramic adhesives. They appear to offer some possible advantages along with some anticipated disadvantages. The advantages derive primarily from their convenience in application and adaptability to widely varying gap sizes between modules. There are potential materials for temperatures from room temperature to 3400°F.

However, most of these materials go through some shrinkage upon setting. This is expected to lead to cracking. Even if this cracking does not take place, cracking is liable to occur as the recuperator is heated and cooled unless the expansion curve of the adhesive closely matches that of the modules it is joining. And finally, if this cracking is avoided, the material cannot help but be a problem when it comes to separation and replacement of modules. These problems are not neces-

sarily insurmountable, but they suggest that considerable work would be involved in perfecting this type of seal.

Viscous Joint Seals:

One other type of joint seal is a glassy material which is in a viscous state at operating temperature. The study of this type of seal has merit, not only because this type of seal might be useful itself, but also because, at least in glass applications at above approximately 1650°F, the joints may end up with this type of seal no matter what type they start with. This is because of glassy condensates forming in the recuperator and flowing down through it, soaking into any openings along the way.

One use of this type of seal is discussed by Smeltzer and Metcalfe.¹⁹ They have studied glasses to be used in relaxing joints in the 1250° to 2000°F temperature range. These joints consist of silicon carbide telescoping tubes with .010 to .004 inch clearance and approximately 1.25 inch overlap. Joints were tested for up to 100 hours at up to 100 psig with negligible leakage.

We have theoretically examined the likelihood of penetration of a viscous glass joint seal for the helical

recuperator. Assuming a low side value of 1.34×10^3 lb./in. (235 dynes/cm)¹⁸ for the surface tension of the viscous glass, a joint gap of greater than approximately .015 inches will ultimately be penetrated at a pressure difference across the joint of 10 inches water column. The relationship is linear with pressure and joint gap distance, so for twice the pressure, the allowable gap is half as large. The remaining question involves the rate of penetration for joint gaps larger than the maximum given above.

For a chosen viscosity η of 10^8 poise (gm/cm-sec or 5.60×10^5 lb./in-sec), a joint thickness S of .25 inches and a required seal lifetime L 2×10^8 seconds (approximately 6 years), and assuming a classical liquid behavior, we can estimate the allowed pressure difference across the joint Δp and joint gap H from the equation¹⁹

$$L = \frac{3s^2\eta}{\Delta p H^2}.$$

For $\Delta p = 10$ inches water column, the allowed gap is approximately .085 inch. The assumptions made are that the seal fluid is bubbleless and that it does not get replaced by a lower viscosity glass formed by some of the condensates. Smeltzer and Metcalfe have wit-

nessed the problems that bubbles in the sealant fluid can cause. Apparently the bubble walls are collapsed one at a time, each providing much less resistance than a pure Newtonian fluid.¹⁹ Conceptually it seems as if a fiber material enmeshed in the viscous medium might be an answer to the problems we have been discussing here, but then the problem becomes finding a fiber which would not dissolve in the viscous glass. If the viscosity of the medium is held to 10^8 poise (5.60×10^5 lb/in-sec) or greater, high alumina type fibers can be expected to serve in this way. For lower viscosities, some development work may be required.

It is believed that there are glass formulas which are applicable as viscous seals to operating temperatures from 400°F to 3000°F.^{19,21} It is not known however what reactions might take place between the module materials and the viscous seal compositions throughout the temperature range and, indeed, Smeltzer and Metcalfe suggest that there may be some problems above 2000°F. This question as well as the best method of application should be addressed if it appears that a viscous type seal can be made to work at all on the modules out of which this recuperator will be assembled.

Compliant Interlayer:

This concept was suggested by Smeltzer and Metcalfe¹⁹ and involves putting a refractory but malleable metal between ceramic joints loaded primarily in compression. In this reference, the authors show that the concept is useful in distributing the loading from imperfectly machined or misaligned ceramic parts. Indeed this concept looks as if it might be helpful should it become necessary to machine the mating surfaces of the modules. However, this does not appear to be a method to seal modules nearly as rough as the present generation of KT. Compliant interlayers have been tested successfully at room temperature, 1100°F and 1800°F. The full potential range is not known at this time.

Selection of Joint Sealing Concepts:

Because of the machining required, the compliant interlayer concept is given last resort status. Similarly, a strictly mechanical seal, if practical in any case, would require machining and is given a low preference status. Because of anticipated difficulty in separation and replacement, the ceramic fillers and adhesives concept is considered secondary in preference. Thus the most promising sealing system at

this time is a combination of a mechanical system for alignment with fibrous gaskets and/or viscous seals.

In the just completed short-term helical recuperator testing it was demonstrated that a combination of a mechanical alignment system and dry fibrous gaskets can be used to hold the leakage to approximately 5% for the short term. It is possible that with experience this leakage value could be decreased to perhaps 2%. The remaining questions are what happens to this material over the long term and how is its behavior modified in a glass environment. As pointed out above, it has been the experience of the steel industry that this material embrittles and falls apart in a matter of three to four years.¹³ Furthermore, it is recognized by the manufacturers that the high alumina types of fibers are susceptible to degradation by alkalis.^{6,7} Just how serious these problems are in this configuration and environment can only be determined by experience.

As described above, the questions relative to the viscous seal concept involve blowing through the seal, interactions between the seal medium and the module materials, the creation of bubbles in the seal

medium and modification of the viscosity of the seal by condensate materials. These factors change throughout the recuperator for different temperature, condensation and module material ranges. Also some development of viscous media for the full range of temperatures would be required. At viscosities required to maintain the seal, initial sealing is expected to require overheating of the stack, and separation may require either overheating or cooling until the seal material cracks. This concept may be incompatible with the concept of burning out to clean the recuperator because during this process the viscosity of the seals can be expected to drop, possibly below the point where a seal would be maintained. It quickly becomes apparent that making this type of system workable will require a significant task.

The calculations reviewed on page 50 of this report suggest that a combination of a viscous seal with enmeshed insoluble fibers might solve some of the anticipated problems of a single viscous seal medium. It would greatly diminish the potential diameter of a penetrating bubble, replacing the joint gap size with the fiber separation distance in the above calculations. The only material with effectively no solubility in glass is platinum and the cost of this as a mesh material can be shown to be out of the range of economics.

However, with a viscosity in the range of 10^8 poise (2.56×10^5 lb/in-sec), the diffusion of a solute through the medium should be so slow that a soluble fiber should retain its integrity for the life of the recuperator. A small number of short excursions into a lower viscosity temperature range could be tolerated, but how many depends upon the peak temperature, the viscosity curve of the glass seal and the fiber diameter and solubility.

Recommendations for Testing:

Because of the complexity of the task of making workable seals using viscous media, it is worth exhausting the possibilities in dry fiber seals. There are many ceramic fibers that have not been examined to date, and indeed, on work done to date the materials used have been quite satisfactory. It is long-term operation in the glass environment that causes concern. A suggested test plan outline for the entire task follows:

- I. Dry fibers in mating alignment system
 - A. Determine and procure prospective materials.
 - B. Cold testing:
 1. Construct compression rig for testing cold leakage.
 2. Test prospective materials to determine:
 - a. thickness required
 - b. compression required
 - c. minimum potential leakage
 - C. Glass flue environment testing:
 1. Construct recuperator using gasket materials interspersed throughout their respective ranges.
 2. Test recuperator for leakage; modify as necessary to minimize leakage.
 3. Conduct flue environment testing of the recuperator.
 4. Upon completion of testing or upon significant increase in joint leakage, cool and dismantle recuperator.
 5. Examine joint seals; reject any materials that are clearly inferior in any temperature range.
 6. If necessary, reassemble recuperator and run until adequate durability of materials is demonstrated.

II. Viscous seals with mating alignment system (to be examined only for those ranges for which no successful dry fiber seals can be found)

A. Determine candidate glass compositions for the temperature ranges to be covered:

1. From literature, suppliers and consultants determine possible glass compositions.
2. Empirically determine viscosity curves for potential compositions.
3. Determine seal compatability with burning out method of recuperator cleaning.

B. Determine compatability of glass seal compositions with the potential substrate (module) materials:

1. Empirically determine the wetability of the substrate by candidate joint sealing materials and temperature needed to achieve bonding.
2. Determine bond strength at temperature.
3. Investigate application methods of sealing media that eliminate bubbling.
4. Determine separation or sticking characteristics as bond is cooled to room temperature.
5. Using ceramic sandwiches, study long-term glass stability at temperature.

C. Determine ability to seal modules:

1. Build hot pressure testing system.
2. Perform module sealing and short-term hot test for leakage.
3. Modify application and sealing techniques as necessary.
4. Perform long-term hot test for leakage.

D. Determine seal performance in environment testing:

1. Assemble recuperator on glass environment simulation furnace.
2. Perform viscous medium sealing operation.
3. Determine leakage of recuperator.
4. Conduct long-term environment exposure, monitoring seals by means of leakage measurements.
5. Upon completion of testing or upon significant increase in joint leakage, cool and dismantle recuperator.
6. Examine joint seals. Reject any materials that are inferior in any temperature range.
7. If necessary, reassemble recuperator and run until adequate durability of materials is demonstrated.

III. Combined viscous and fiber seals with mating alignment system (to be examined only for those ranges for which no successful dry fiber seals nor pure viscous seals can be found).

- A. Determine candidate fiber-glass combinations:
 - 1. From literature and solubility and diffusion studies, determine possible fiber compositions and diameters for candidate glass determined in I. above.
 - 2. Determine long-term stability of candidate combinations.
- B. Determine ability to seal modules:
 - 1. Perform module sealing and short-term hot test for leakage.
 - 2. Modify application and sealing techniques as required.
 - 3. Perform long-term hot test for leakage.
- C. Determine seal performances in environmental testing:
 - 1. Assemble recuperator on glass environment simulation furnace.
 - 2. Perform viscous medium sealing operation.
 - 3. Determine leakage of recuperator.
 - 4. Conduct long-term environment exposure, monitoring seals by means of leakage measurements.
 - 5. Upon completion of testing or upon significant increase in joint leakage, cool and dismantle recuperator.

6. Examine joint seals. Reject any materials that are inferior in any temperature range.
7. If necessary, reassemble recuperator and run until adequate durability of materials is demonstrated.

MODULE SEPARATION AND REPLACEMENT

The problem:

One of the considered advantages of the modular helical recuperator is that its modularity is expected to allow selective replacement of cracked, corroded, plugged or otherwise malfunctioning parts. This means that when a malfunction occurs, its location must be identified and then the modules must be separated, the malfunctioning module or modules replaced and the stack resealed. The purpose here is to propose a system and test plan for the separation function during long-term glass flue environment testing of the recuperator.

Figure 11 shows a diagrammatic sketch of the recuperator with insulation removed near the module or modules in need of replacement. It should be remembered that in a glass furnace environment the modules operating at temperatures above approximately 1650°F will be glazed, and this glaze can be expected to seep into the joints between the modules bonding them together. If viscous seals are used, these will create a glass bond between the modules. The strength of these glass bonds after cooling to room temperature is dependent upon the wettability and porosity of the module materials and on the relative thermal coeffi-

Figure 11. Schematic of recuperator with insulation removed around module to be replaced.

coefficients of expansion of the glass and the module material. If the expansion coefficients are markedly different, the glass can be expected to crack, greatly weakening the bond. However, if this cracking does not occur, the tensile strength of the glass bond, $10,000 \pm 5,000 \text{ lb/sq. in.}^{18}$, may be considerably more than that of the ceramic body of the bonded modules. In this case, cold separation of the modules will be difficult without destroying the modules themselves and hot separation may be necessary. For this reason as well as simple convenience, it would be best if replacement operations could be performed without removing insulation from the entire stack.

The device:

In this phase of the work, what is required is a mechanism which performs the required separation and also measures the force necessary to perform that separation. What is necessary is a pair of devices which will each wrap around a joint and exert a controllable and measureable separation force. Once separation has been achieved above the module(s) to be replaced, the stack above that level must be raised enough to allow separation below and to allow modules

to be maneuvered in and out of place. For the configuration used in phase II of the recuperator project, a total raising of one inch would be adequate.

Figure 12 schematically shows one of the pair of separation devices. It actually consists of two pairs of mating parts which slide together around the joint to be separated. The modules are notched so that the separation device can grip the two modules to pull them apart. Four stress cylinder separators are located around the stack and these are adjusted together to exert an even force on the two modules.

This operation is first performed above the modules to be replaced. Then chains or cables from a winch above are attached to the upper part of the device. With this that part of the device and the entire stack above the break is raised enough to allow separation below and removal and replacement of modules. The lower part of the separation device is removed. Separation is performed below the defective section in the same way, but once separation is achieved, it is anticipated that the module(s) to be replaced can be removed by hand or with the use of tongs.

Recommendations for Testing:

Because of the different condensate ranges inside

Figure 12. Schematics of recuperator separation device. A. Side view of system in place; B. Top view of system open; C. Top view of system closed around module.

Device shown will separate the module to be replaced from the one above. Another such device is used to separate the module to be replaced from the one below it.

the recuperator, separation should be attempted at locations throughout the stack. It is anticipated that no sticking of the modules will be observed in the temperature range below about 1500°F unless viscous seals are used. However, this may not be true if the condensates are burned out. A suggested test plan outline follows:

- I. Perform detailed design and specifications of separator parts and modules.
- II. Fabricate separator parts and stack lifting winch.
- III. Perform pretesting on cold stack.
- IV. Perform separation and testing in conjunction with glass flue environment testing of recuperator:
 - A. Perform separation of shut down and cold system in the temperature ranges where this is possible. Include attempts in all temperature ranges and all candidate seal types.
 - B. Attempt hot separation in all above ranges.
 - C. Measure forces required to obtain separation in every instance.
 - D. Perform complete replacement of modules and resealing of stack, hot if possible. Monitor leakage to determine successful resealing.

E. Modify replacement and resealing system as necessary until system competence can be demonstrated.

Integration with other subsystems:

No attempt has been made here to integrate the subsystems described with other subsystems of the total recuperator, e.g., insulation, stack supports, manifolding and ducting above and below the recuperator, etc. These systems must be resolved in at least a preliminary sense before that integration can begin. And because of the interrelationships between all of the various subsystems, it is anticipated that the subsystems described here and the rough test outlines for those subsystems may well be modified as that integration process develops. The attempt here has been to provide a basic starting point for the selected subsystems.

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