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CERAMIC HEAT RECUPERATORS
FOR
INDUSTRIAL HEAT RECOVERY

Final Report
August 1980

Work Performed Under Contract
No. EX-76-C-01-2162

U. S. DEPARTMENT OF ENERGY
IDAHO OPERATIONS OFFICE

GTE PRODUCTS CORPORATION
TOWANDA, PA. 18848

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CERAMIC HEAT RECUPERATORS
FOR
INDUSTRIAL HEAT RECOVERY

Final Report
August, 1980

Prepared for

The U.S. Department of Energy
Office of Industrial Programs
Washington, DC 20545

Contract No. EX-76-C-01-2162

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FOREWORD

GTE Products Corporation began work on Ceramic Heat Exchangers about nine years ago and has developed a compact cross-flow recuperator core. In October, 1976, DOE and GTE began a jointly funded (75%/25%) program (Contract No. EX-76-C-01-2162) to demonstrate the technical and economic feasibility of using ceramic heat recuperators for industrial heat recovery. In January, 1980, PAR Enterprises, Inc. was contracted by DOE through Oak Ridge National Laboratory (Contract No. 44X-49100V) to develop the data and assist GTE in preparing the Final Report. This report represents the combined efforts of GTE and PAR.

This report describes the GTE Ceramic Recuperator Development Program and presents the results of the jointly funded Demonstration Program.

The results of work performed by GTE prior to the DOE contract have been included in this report to provide the reader with an overall understanding of the ceramic recuperator design. Additionally this report was written to aid engineers in industry to evaluate their specific furnace applications from technical and economic viewpoints to determine the benefits that could be derived from the use of ceramic heat recuperators. The data provided in the body of this report and its appendices will allow an engineer in industry to perform evaluations without the need for an extensive search of other references in the literature. This report is intended to aid in the transfer of recuperator technology to industry and help meet the energy conservation goals of the U.S. Department of Energy.

EXECUTIVE SUMMARY

In most industrial furnaces, the furnace exhaust gases are directly vented to the stack. Furnaces that run at temperatures of about 1800 °F lose about 50% of the input energy in the flue gas exhaust. As fuel costs continue to escalate and industries are faced with curtailments in natural gas supplies, new methods of saving energy are needed. One method of obtaining energy savings is the recovery of waste heat by using recuperators to preheat the combustion air to the furnace burners. An added advantage of preheated air is improved efficiency of the combustion process.

GTE Products Corporation began work on ceramic heat exchangers about nine years ago and has developed a compact cross-flow recuperator core. The objectives of that research program were to develop a ceramic heat exchanger with the following characteristics:

- Capability to handle exhaust gases at temperatures up to 2600 °F.
- Large heat transfer surface area per unit of volume.
- Cost comparable to metallic heat exchangers currently available.

In October, 1976, the Department of Energy and GTE began a jointly funded (75%/25%) program (Contract No. EX-76-C-01-2162) to demonstrate ~~the technical and economic feasibility of using ceramic heat recuperators~~ for industrial heat recovery. The specific objectives of this demonstration program were to:

- Determine the heat transfer performance of the ceramic recuperator.
- Establish the energy savings of recuperation.
- Demonstrate minimum maintenance requirements in typical furnace operations.

It was intended that the demonstration program would also:

- Determine the durability of the ceramic core.
- Determine the operating requirements of the burners and controls when using preheated combustion air.
- Establish the overall system costs and payback period.

Five demonstrations were undertaken on furnaces operating between 1400 °F and 2600 °F with firing rates from 0.4 to 1.2 MM BTU/hour. This report describes the GTE ceramic recuperator development program and presents the results of the jointly funded demonstration program.

The energy savings from exhaust gas recuperation are primarily the result of the increase in energy of the combustion air and, to some extent, improved combustion efficiency. The amount of fuel saved depends on the exhaust gas temperature, amount of excess air, the type of burner and the furnace control system. Fuel savings may be accurately measured by metering the energy consumption per unit of production before and after installation of the recuperator. They may also be calculated by performing a mass and energy balance on the burner, the furnace and the recuperator as a complete system. The effect of recuperation on the mode of heat transfer in the furnace can only be calculated for a specific furnace design. For some of the furnace applications investigated, the estimated fuel savings ranged from 20% to 50% of the fuel previously used depending on the furnace operating conditions.

In designing and evaluating a heat exchanger, one needs to know how efficiently the unit does the job of transferring energy from the hot fluid to the cold fluid. Heat exchanger effectiveness is often defined as the ratio of the actual heat transferred to the maximum possible amount of heat that could be transferred for a given exhaust gas flow and temperature. Effectiveness is a function of the size of the heat exchanger, since sufficient heat transfer surface area must be provided to effectively transfer

energy from the hot gases to the combustion air. The ceramic recuperators used in this program have values of effectiveness in the range of 40% to 60%.

The pressure drop resulting from flow of gases through the ceramic core is an important design parameter. In retrofit recuperator applications, the recuperator must be designed so that the core pressure drop is within the allowable limits for that application. This report presents the procedure to analyze a system within the given design criteria and optimize the recuperator from the standpoint of performance.

The GTE ceramic recuperator was designed for relatively small furnaces with firing rates of 0.3 to 0.6 MM BTU/hour and with exhaust gas temperatures of 1500 to 2600 °F. The ceramic selected as the material of construction is cordierite, a magnesium aluminum silicate (MAS). The selection was based on cordierite's ease of fabrication, relatively low thermal expansion, good thermal shock resistance and good corrosion resistance. The basic fabricated ceramic core is a cube measuring one foot on each side. The flow passages are rectangular in cross section with the individual gas side passages larger than the air passages to minimize the gas side pressure drop. The resulting core is an extremely compact unit, i.e. one with a large heat transfer surface area per unit of volume. The basic core is modular in design and allows several units to be joined together for any application; however, the core can be manufactured in other sizes as well.

The ceramic core is housed in a fabricated metal housing. A refractory seal is provided between the ceramic core and the metal housing. This seal prevents the leakage of air into the gas stream. Excessive leakage reduces the heat exchanger effectiveness and is to be avoided. During the demonstration program, a spring-loaded housing developed by Hamilton Thermal Inc., Hamilton, Ontario, was used and resulted in a reduction of the overall leakage rate to within 5% of the air flow. These leakage rates were determined in tests using a specially designed GTE test furnace under normal temperature operating conditions. The test furnace

was also used to determine the heat transfer and pressure drop characteristics of the recuperators and the durability of the heat exchanger under severe conditions, such as temperature cycling. The performance data, presented in this report, can be used to size a recuperator for a given application.

The housed recuperator is low in capital cost compared with metallic recuperators. For example, a one cubic foot housed unit presently costs \$2,800. The cost of a total system, which includes new burners, controls, duct work, insulation, engineering and installation, is generally 5 to 6 times the cost of the recuperator itself. The net savings and payback periods for any installation will depend on factors such as type of furnace, type of operation, exhaust gas temperature, air preheat, fuel costs, installation cost, etc. In general, payback periods for a ceramic recuperator installation range from nine months to two years.

Five demonstration projects were undertaken on furnaces operating between 1400 and 2600 °F with firing rates of 0.4 to 1.2 MM BTU/hour. Three projects are currently operational and two have been terminated. The following furnaces were selected:

- A reheat furnace used for heating molybdenum slabs to 2600 °F for a Bliss rolling mill.
- A furnace used for reduction of powdered tungsten ore to metal at 1700 °F.
- A York glass tank furnace used for melting borosilicate glass at about 2400 °F.
- A pilot plant U.S. Smelting rotary furnace used for the experimental batch treatment of various ceramic powders at 1700 °F.
- A rotary calciner furnace used for calcining various ores at about 1300 °F.

The first demonstration on the molybdenum reheat furnace performed satisfactorily at temperatures up to 2600 °F. The exhaust gases contained molybdenum oxide dust, which deposited in the core. An air lance system was successfully used to prevent the core from plugging. The recuperator logged 2200 hours of trouble-free operation until an operator's error caused the furnace temperature to rise to 2800 °F. The cordierite softened at 2700 °F and the recuperator failed.

The tungsten reduction furnace application was probably the most successful in demonstrating the durability of the ceramic recuperator at temperatures of 1700 °F. No problems were encountered and the three recuperators on the furnace are currently in operation.

The glass tank furnace demonstration was discontinued after four separate sets of recuperators were tested. The borosilicate in the glass rapidly attacked the cordierite causing repeated failures. Coating the cordierite with alumina seemed to improve the resistance to corrosion by the borosilicates, but this observation is inconclusive because of insufficient time of operation. The monitoring of temperatures with thermocouples was difficult because of glass deposits on the thermocouples. After the fourth failure, no further attempts were made to continue this demonstration.

The U.S. Smelting furnace demonstration presented no problems. The furnace is used only for experimental purposes and while the recuperator is still operable, not enough operating time has been logged on this demonstration to warrant firm conclusions.

The rotary calciner furnace demonstration is currently operating with no major problems. The original recuperator was mechanically damaged during moving of the furnace and was replaced. The combined time of operation was about 3700 hours. The demonstration further attested to the durability of the matrix during continuous variations in the operating variables.

The following conclusions are made from the demonstration program:

- The energy saved from waste heat recovery could amount to 20% to 50% of the fuel that was used before installation of a recuperator.
- The more efficient method of achieving energy savings is to pre-heat the combustion air to the furnace burners rather than heating air for process needs.
- The concept of using a ceramic material in a compact heat exchanger is technically feasible in many heat recovery applications.
- Cordierite is durable in certain applications at furnace temperatures up to 2600 °F.
- Cordierite is rapidly corroded by exhaust gas from a borosilicate glass furnace.
- Particulate matter in furnace exhausts, which deposit in the core, can be removed with an air lance.
- The recuperator can be retrofitted on existing furnaces. Because of its small size and weight, no structural modifications to the furnace are needed.
- The basic ceramic core design allows for optimization of the heat transfer and pressure drop performance of the recuperator.
- Leakage of air to the gas side is less than 5% of the air flow with a spring-loaded housing design.
- Commercially available burner and control systems proved adequate for use with preheated combustion air.
- In most applications, little or no maintenance is required.

- The recuperator is a low capital cost item and complete system installations can result in payback periods of nine months to two years.

As a result of the demonstration program, the following recommendations are made:

- The demonstration program be continued at an accelerated level to aid in the transfer of this technology to industry.
 - An application analysis be conducted to select the industries for the accelerated program.
 - Before each demonstration is undertaken, a testing and monitoring plan should be prepared to maximize the data obtained and the benefit derived from each demonstration.
-

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Mr. Joseph J. Cleveland of GTE was responsible for the initial development of the ceramic core and served as Program Manager on the DOE Contract. Mr. Jeffrey M. Gonzalez participated in the core development, data development, preparation of the computer programs and writing of this report. Mr. Ken H. Kohnken performed the system economics and is responsible for marketing of the recuperator. Dr. W. J. Rebello of PAR was responsible for the data development and preparation of this report and served as Program Manager on the ORNL contract. Special thanks are due Leonard Donahue, Raymond Dorazio, William Mingos, Ray Newman and Beverly Beers for their participation in the design and monitoring efforts. The support staff of Kathy Hammerly, Sally Powers, Anita Moore and Gloria Rebello patiently typed and retyped the report. Thanks are due Hamilton Thermal Inc., Hamilton, Canada, for permission to use a photograph of a recuperator-burner combination constructed by them.

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LIST OF SYMBOLS

A	Heat Transfer Surface Area
A_c	Heat Exchanger Free Flow Area
A_{fr}	Heat Exchanger Frontal Area
C	Stream Capacity Rate (mC_p)
C_a	Air Side Capacity Rate
C_g	Gas Side Capacity Rate
C_{min}	Minimum of C_a or C_g
C_{max}	Maximum of C_a or C_g
C_p	Specific Heat at Constant Pressure
D	Inside Diameter of a Circular Tube
D_h	Hydraulic Diameter of Passage ($D_h = 4r_h = 4A_c L/A$)
E	Heat Exchanger Effectiveness
E_{std}	Friction Horsepower, hp/ft^2
f	Fanning Friction Factor
F	Correction Factor for LMTD
g_c	Gravitational Constant
G	Stream Mass Velocity (m/A_c)
h	Stream Specific Enthalpy or Convective Film Coefficient
h_{std}	Heat Transfer Power, $BTU/hr ft^2 ^\circ F$
H	Total Enthalpy of Fluid
ΔH_R	Heat of Combustion of Fuel
j	Colburn Heat Transfer Factor
k	Unit Thermal Conductivity
K_c	Contraction Loss Coefficient at Heat Exchanger Entrance
K_e	Expansion Loss Coefficient at Heat Exchanger Exit
L	Length of Flow Path
LMTD	Log-Mean Temperature Difference
m	Mass Flow Rate of the Fluid
N_R	Reynolds Number (GD/μ)
N_{Pr}	Prandtl Number ($\mu C_p/k$)
N_{st}	Stanton Number (h/GC_p)
NTU	Number of Transfer Units
P	Pressure of Fluid

LIST OF SYMBOLS (Continued)

ΔP	Pressure Drop
q	Heat Transfer Rate
Q_L	Furnace Load and Losses
r_h	Hydraulic Radius
SCFH	Hourly Volumetric Flow at Standard Conditions
t	Temperature to Any Arbitrary Scale
T	Absolute Temperature
ΔT_{lm}	Log-Mean Temperature Difference
ΔT_m	Mean Temperature Difference
U	Unit Overall Thermal Conductance or Overall Heat Transfer Coefficient
v	Specific Volume of Fluid
V	Fluid Velocity
x	Thickness of Material
α	Channel Aspect Ratio
σ	Free Flow Area/Frontal Area (A_c/A_{fr})
μ	Fluid Dynamic Viscosity
ρ	Fluid Density

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1. INTRODUCTION

GTE Products Corporation began work on a ceramic rotary regenerator for automotive gas turbine applications about nine years ago. Ceramic materials were chosen because of their ease of fabrication and good corrosion resistance. The ceramic honeycomb design had the advantage of providing a large heat transfer surface area per unit of volume. Several designs were fabricated and tested. Because of the inherent sealing problems with a rotary regenerator and other technical problems with the matrix, these heat wheel designs were subsequently discarded. Research in ceramic materials, particularly bonding and fabrication techniques, however, continued.

As fuel costs began to escalate and industries were faced with curtailments in their natural gas supplies, new methods of saving energy were needed. The recovery of waste energy from industrial furnace exhaust gas began gaining in importance. GTE directed its research program in ceramic heat exchangers to the development of a compact, cross-flow recuperator core. Several ceramic recuperators were built and tested, with encouraging results. In October, 1976, GTE Products Corporation was contracted by the Department of Energy (Contract No. EX-76-C-01-2162) to demonstrate the technical and economic feasibility of using ceramic heat recuperators for industrial heat recovery. This project was jointly funded by both DOE and GTE (75%/25%). The specific objectives of the demonstration program were to establish the energy savings by recuperation, determine the heat transfer performance of the ceramic recuperators and demonstrate the durability of the ceramic cores under actual furnace operating conditions. GTE facilities were selected for the five demonstration projects because of the variety of furnaces available.

This final report presents the results of the program. The report is written so that the information presented can be used by engineers in industry to evaluate their specific furnace applications and to estimate the technical requirements and economic benefits from the use of ceramic heat recuperators. The specific GTE core design and the recuperator fabrication, housing and installation are discussed in Chapter 2. The demonstration project results are presented in Chapter 3. Each demonstration is described and the results, economics and problem areas discussed. Appendix A presents methods that can be used to estimate the amount of energy savings by recuperation. Appendix B gives the reader a brief background in heat exchanger design theory. The Appendices provide details that will allow the engineer in industry to select a core, size a heat exchanger, calculate the performance, determine energy saved and estimate the economics of using a ceramic recuperator for a specific industrial application.

2. GTE PRODUCTS CORPORATION RECUPERATOR DESIGN

This chapter describes the GTE ceramic recuperator design. The results of work performed by GTE prior to this DOE contract are included in this discussion to provide the reader with an overall understanding of the ceramic recuperator design.

2.1 Objectives

The near term goals of the ceramic recuperator project were to develop a ceramic recuperator with the following characteristics:

- capability to handle exhaust gas temperature up to 2600 °F,
- large surface area per unit volume,
- cost comparable to metallic heat exchangers currently available.

The target applications for the ceramic recuperator are small furnaces with firing rates from 0.3-6 MM BTU/hour and with exhaust gas temperatures from 1500-2600 °F.

2.2 General Considerations

Convection recuperators used for gas-to-gas heat exchange are usually made of metals fabricated into plates or finned tubes for cross-flow applications. Common construction materials are type 309 or 446 stainless steels and certain nickel-base alloys; however, high temperature creep and oxidation limit the maximum metal temperatures to about 1600 °F. Another factor that affects metallic recuperators is the corrosive materials in furnace exhausts, such as, sulfur compounds, fluorides, chlorides and vanadium. These corrosive materials can cause corrosion of the recuperator and necessitate periodic cleaning. In applications where furnace exhaust temperatures are very high, the exhaust gas stream is diluted by cold air, water or steam injection upstream of the recuperator to reduce the exhaust gas temperature to a level that is safe for the metal. This dilution, of course, reduces the amount of energy that can be recovered from the exhaust gases.

Certain refractory ceramic materials show promise for use in recuperators for high temperature applications. Candidate materials are lithium aluminum silicate (LAS), magnesium aluminum silicate (MAS), mullite, silicon carbide, silicon nitride and various zirconia compositions. In general, thermal shock resistance is an important factor in determining the potential of a ceramic material for use in a recuperator. Temperature gradients and thermal expansion of the ceramic materials can cause cracks that lead to unacceptable high rates of leakage. Ceramic materials must have suitable mechanical properties, resistance to crack growth, low gas permeability, and good chemical stability. The candidate materials must be fabricable at reasonable cost into required shapes with strong leak-proof joints. Ceramic recuperators are available on the market that employ silicon carbide tubes in a cross-flow mode. Many of these ceramic materials cost about 10 times

as much as stainless steel on a per pound basis, and, therefore, economics influence the selection of a recuperator for a given application. For a more detailed state of the art assessment of recuperator types and materials of construction, reference may be made to Tennery and Wei(1).

GTE Products Corporation's interest in ceramic heat exchangers began with the development of a MAS body for use in a rotary regenerator for automotive gas turbine applications. MAS was selected because of ease of fabrication; relatively low thermal expansion, good thermal shock resistance and good corrosion resistance. In order to obtain these properties, MAS must be fired in an extremely narrow temperature range and this factor initially presented some difficulty. Over a period of three years, processing techniques were developed and circular heat wheels were investigated for use in industrial heat recovery; however, due to problems of breakage, high pressure drop, leakage through seals, and high cost and maintenance, work on rotating regenerator wheel designs was terminated.

Continued research resulted in the development of a process for forming a compact cross-flow recuperator core. Development of a bonding material and refinement of the fabrication process resulted in the present cross-flow recuperator core. The remainder of this chapter deals with the specific GTE design, testing, and development of the heat transfer characteristics of the ceramic recuperator.

2.3 Design Criteria

The following design criteria were selected by GTE for its ceramic recuperator development:

- Temperature capability in the range 1600-2600 °F.
- Compact size with a large heat transfer surface area per unit volume.
- Capacity for furnace firing rates of 0.3 to 6.0 MM BTU/hour.
- Good thermal shock resistance.
- High resistance to corrosion and good chemical stability.
- Leakage of combustion air into the gas stream of less than 15%.
- Capacity to withstand pressures of 2 psig.
- Exhaust side pressure drop not exceeding 0.3 inch of water.
- Modular design to allow add-on capacity.
- Cost comparable with present metallic recuperators.

2.4 Ceramic Core and Housing

A gas to gas heat exchanger of small size and light weight requires a large surface area, commonly referred to as a compact surface. The heat transfer rate per unit of surface area may be increased by increasing the fluid velocity since the rate increases approximately linearly with velocity. On the other hand friction increases as a function of velocity to the second or third power. The core design, therefore, should maximize the heat transfer rate and at the same time meet the given pressure drop specification. Friction may be reduced by increasing the number of individual flow passages thereby reducing gas velocities. While reduced gas velocities decrease heat transfer per unit of surface area, additional heat transfer area may be provided by increasing the length of the heat exchanger, thus compensating for the lower transfer rate. While increasing the heat exchanger length will cause an increase in the power needed, it does so in direct proportion to the heat transfer surface area increase.

The size of the individual flow passages for the air and gas streams is an important factor in the design of a compact heat exchanger. Flow passages of small cross-sectional area provide a more compact surface but produce a higher pressure drop and increase the possibility of plugging if the gas stream contains particulate matter. Since the gas side pressure drop is critical, particularly in retrofit applications, a much larger free flow area is normally provided on the gas side than on the air side. Thus, the cross-sectional area of the individual passages on the gas side are usually larger than on the air side of the cross-flow heat exchanger.

The GTE cross-flow design is based on the preceding considerations. Figure 2.1 shows a cubical ceramic recuperator core with multiple flow passages. The flow passages are formed by two flat plates separated by webs or ribs. The flow passages on the gas side, as previously discussed, are much larger than those on the air side. The cube edges have additional side ribs to allow for sealing of the cube in either a ceramic or metal housing. A photograph of the sealed cube providing a surface for the gasket material is shown in Figure 2.2. The ceramic cube installed in a housing is depicted in Figure 2.3.

The total assembly consists of the ceramic core, the resilient refractory seal, and the housing. This housing design is as important to the successful system operation as the ceramic core design. The housing must keep the ceramic in compression over a wide temperature range to prevent leakage. The housing must also withstand high temperatures and provide ease of assembly with other parts of the system, (i.e. pipes, flues and stacks).

Two basic types of housings are being employed in the demonstration contract. The first housing investigated is shown in Figure 2.4. The inlet air duct transition is designed to achieve an even flow through the core. The outlet stainless steel transition is insulated on the inside to reduce metal temperatures and heat loss. The furnace exhaust flange is insulated by a 2-inch thick layer of refractory to avoid overheating and

resultant warping of the metal. This housing is designed for furnace exhaust temperatures of approximately 1600 °F. More insulation would be used at higher exhaust temperatures to insure that the housing does not shift due to thermal expansion. During a test with insufficient insulation an exhaust gas temperature of 2000 °F caused a 1/8-inch wide separation between the core and the air outlet bell.

The latest housing design, illustrated in Figure 2.5, is a unit designed for high temperature operation by Hamilton Thermal Inc., Hamilton, Canada. It consists of a steel casing enclosing a high-temperature ceramic refractory lining. The ceramic core is sealed into the refractory lining by the 1-inch wide refractory edge seal. This unit also utilizes high-temperature springs that exert a compressive force parallel to the direction of air flow, thus allowing for a slight thermal expansion and assuring a constant and uniform seal. The spring loading also allows easy replacement or removal of the ceramic core from the unit for cleaning and maintenance.

Minimizing leakage from the air to the gas side is an important consideration in recuperator design. Leakage occurs by permeation through the porous material, through cracks in the ceramic core caused by temperature induced stresses, and through seals between the ceramic core and metal housing. Leakage of air into the gas stream results in a reduction in mass flow of air through the heat exchanger as well as a reduction in the temperature of the gas flowing through the heat exchanger. This results in a reduction in the effectiveness of the heat exchanger in extracting the waste heat from the flue gases. The GTE recuperator is made of a relatively dense ceramic material and permeation of air through the material to the gas side is considered negligible.

Considerable effort has been expended on prevention of cracking of the core because of temperature induced stresses. Material specifications with regard to impurities, adequate blending of materials, research on new bonding materials, improved fabrication techniques and proper control of the firing process have all contributed to improvement in the mechanical properties of the finished core. Recent cores tested under extreme conditions have shown improved resistance to slow crack growth. Considerable experimentation with different refractory sealing materials and the spring-loaded housing shown in Figure 2.5 have resulted in the prevention of excessive leakage through the seals. While one of the initial design criteria was to maintain leakage at under 15% of the maximum air flow, present units tested under representative conditions indicate leakages of less than 5% of the maximum air flow.

Another aspect of leakage is the effect on the combustion process. Burner systems in use today usually require a static air pressure at the burner of about 16 ounces/inch². In the design of a new installation, a suitable blower could be selected that would allow for leakage and pressure drop across the recuperator and still provide the necessary air pressure at the burner. However, in a retrofit application where the original air blower is to be used, excessive air leakage through the recuperator can affect the operation of the burner. For this reason, leakage should be kept to a minimum.

Figure 2.1

GTE Cross-Flow Recuperator Cube

Figure 2.2

GTE Cross-Flow Recuperator Sealed Cube

Figure 2.3

GTE Sylvania Ceramic Cube Assembly

Figure 2.4
Welded Stainless Steel Housing
For Ceramic Core

Figure 2.5
Cast Steel Housing
For Ceramic Core

The ceramic cross-flow core can be manufactured in a variety of standard sizes. Table 2.1 lists some of the standard flow passage sizes available for preheat air and exhaust passages. Table 2.2 lists the sizes of the standard cubes that have been developed and are being utilized in this contract. The surface characteristics such as hydraulic diameter, free flow area, etc., have been calculated for some of the heat transfer surface geometries and are presented in Appendix D.

The largest size ceramic core utilized in this contract is 6 ft³, 17 in. x 18 in. x 34 in. Units of this size caused difficulties in maintaining uniform flow passages in the fabricating process, and, in the future, large units can be built up of smaller modules. The 18 in. x 18 in. x 18 in. units or the anticipated 24 in. x 24 in. x 12 in. units can be grouped together to provide heat exchangers for a range of gas flows from 40,000 to 1 million SCFH. Two possible arrangements are shown in Figure 2.6. The housing illustrated in Figure 2.5 was specifically designed so that several units could be joined together in modular construction.

The concept of modular design is being utilized in this demonstration contract. For example, a 12 million BTU/hour furnace can utilize twelve 1 ft³ units rated at 1 million BTU/hour each to achieve optimum preheats to twelve burners or two units of 6 ft³ if fewer burners can be employed.

2.5 The Ceramic Core

Candidate ceramic materials for high-temperature heat exchanger applications must have good thermal shock resistance. Resistance to thermal shock is a function of the thermal expansion coefficient, tensile and compressive strength, elastic modulus, modulus of rupture, fracture toughness, Poisson's ratio, thermal conductivity, heat capacity and density. A material with good thermal shock resistance should have the following properties:

- Relatively low thermal expansion
- High strength
- Resistance to fracture
- Low elastic modulus
- Small Poisson's ratio.

Properties of some of the candidate materials are presented in Table 2.3. The other desirable characteristics for a ceramic material are a high melting point, resistance to corrosion, and ease of fabrication.

Most of the candidate materials have certain limitations but it is possible to compare them qualitatively. The usual operating temperature for vitreous silica is limited to 2,100 °F because devitrification (crystallization) occurs

TABLE 2.1AVAILABLE CERAMIC CORE HOLE SIZES

Dimensions (Inches) Before Firing the Cube

0.750 in. x 0.125 in.
 0.750 in. x 0.200 in.
 0.750 in. x 0.300 in.
 0.750 in. x 0.350 in.
 0.750 in. x 0.500 in.
 0.750 in. x 0.600 in.

TABLE 2.2STANDARD SIZE CERAMIC CUBES

Dimensions After Firing the Cubes

<u>Overall Size (in)</u>	<u>Usable Heat Transfer Volume (in³)</u>
10 x 10 x 10	512
12 x 12 x 12	1,000
12 x 12 x 18	1,600
12 x 12 x 24	2,200
18 x 18 x 18	4,096
17 x 18 x 34	7,680

Figure 2.6

Two Arrangements of Grouped Ceramic Cores

with long exposures above 2,100 °F. LAS ceramics have a low coefficient of thermal expansion but have a lower melting point and react with sulfur which may be present in the exhausts of furnaces fired by fuel oil. Silicon nitride, both reaction bonded and sintered material, is high in cost. Silicon carbide oxidizes at high temperatures and is difficult to fabricate into thin cross-sections. Alumina and mullite, although very high strength and refractory in nature, are difficult to sinter into thin walled complex shapes. Fluxed zirconium silicate (Z-1000) was developed by GTE during this demonstration program to resist the chemical attack of borosilicate glass. However, its use is limited because of its high firing shrinkage which leads to poor dimensional stability and its high fabrication costs due to its abrasive nature.

Cordierite (MAS) or $2\text{MgO} \cdot 2\text{Al}_2\text{O}_3 \cdot 5\text{SiO}_2$ was selected as the ceramic material principally because of ease of fabrication, good thermal shock resistance, and good corrosion resistance. Three basic raw materials were developed by GTE prior to this demonstration program. Their properties are listed in Table 2.4. Cordierite's average softening point is about 2,550 °F and average melting point is about 2,615 °F. These properties would limit their application to furnace temperatures of about 2,600 °F. MAS 8400 proved to be the best in terms of structural integrity, fabricability and sinterability. It was, therefore, selected as the material for the demonstration cores.

TABLE 2.3
PROPERTIES OF VARIOUS CERAMIC MATERIALS⁽²⁾

<u>Material</u>	<u>Mean Coefficient Of Linear Thermal Expansion in/in °F x 10⁶</u>	<u>Youngs Modulus x10⁶ psi</u>	<u>Modulus of Rupture psi</u>	<u>Melting Points °F</u>
Vitreous Silica, SiO_2	0.28	10.5	3,120	
LAS	0.30	30	8,000	2,370
Cordierite or MAS, $2\text{MgO} \cdot 2\text{Al}_2\text{O}_3 \cdot 5\text{SiO}_2$	0.93	34	7,000	2,650
Silicon Nitride, Si_3N_4	1.35	55	10,000	3,450
Silicon Carbide, SiC	2.56	55	24,000	3,990
Fluxed Zirconium Silicate, Z1000	1.95			2,985
Alumina, Al_2O_3	4.34	25	20,000	3,720
Mullite $3\text{Al}_2\text{O}_3 \cdot 2\text{SiO}_2$	2.95	22	10,000	3,290

TABLE 2.4
PROPERTIES OF CERAMIC MATERIALS
USED FOR FABRICATING CERAMIC CORES

<u>Property</u>	<u>Material^(A)</u>		
	<u>MAS 8200</u>	<u>MAS 8400</u>	<u>MAS 8252</u>
Density, % of theoretical	91	92	97
Melting Point, °F	2570	2615	2640
Softening Temperature, °F ^(B)	2550	2550	2580
Modules of Rupture, psi ^(C)	4500	3500	6500
Modules of Elasticity, psi x10 ⁶	7.6	4.5	13.8

(A) All materials are processed from standard grade raw materials prior to final processing.

(B) The temperature at which a specimen, heated at a rate of 18 °F/hour, deflects 1/4 inch in a 4 inch span.

(C) 75 percent theoretical density at 77 °F.

The ceramic core (cube) is sealed into the housing by a proprietary refractory sealing material applied to the end ribs of the ceramic cube along with a refractory gasket that expands two or three times its original volume on heating. The sealant must be a refractory capable of withstanding temperatures above 1,400 °F and be relatively impervious to air flow at 32 oz./in.² differential pressure.

2.6 Fabrication, Assembly and Installation

The fabrication procedure for ceramic cores is illustrated in Figure 2.7. The core is then sealed in the housing as described in Section 2.4 and the unit is ready for installation in a furnace application. Because of small size and weight, the installation of the ceramic recuperator in an existing furnace system is a relatively simple matter. The recuperator can usually be mounted on top of the furnace exhaust duct with a minimum of structural support. A stack is attached to the top of the recuperator housing. Insulated pipe is required to duct preheated combustion air to the burner. If the original air blower is used, it must have sufficient capacity to provide the combustion air at the required pressure to the burner. Burner control and safety system modifications can be made in conjunction with the burner

or furnace manufacturer. Standard equipment for monitoring flows, temperatures, and pressures can be installed. A typical ceramic recuperator installation on a furnace is shown in Figure 2.8. An important component of a recuperator installation is an air by-pass system to supply air to the burner in the event of a recuperator failure. Figure 2.8 shows such a by-pass system.

2.7 Tests and Measurements

Two types of laboratory tests were performed in this program. Each cube is tested for leakage after firing. The assembled recuperator was then tested on a specially built furnace to determine the heat transfer and pressure drop characteristics.

2.7.1 Leak Tests

Ceramic cubes were tested for delaminations and internal leakage in the leak test facility shown in Figure 2.9. Tests were conducted at room temperature on both the air and gas flow paths. The test fluid was air, which was applied alternately to the air and gas flow paths. The cube was placed in the test stand and the rubber plates which simulate the refractory sealant, secured with tensioning screws. The inlet to the cube was then pressurized to 10 psig to ensure that the cube was structurally sound. The top of the cube was open to the atmosphere. Thus, air that leaked from the pressurized flow path through pores and internal cracks, flows out the top.

At a pressure of 10 psig any weak laminations blew open and the cube was rejected. Cubes that passed the delamination test were then leak tested. The upstream pressure was maintained at 2 psig, and the air flow, as measured by the orifice plates, was recorded.

Rejection of a cube for excessive leakage was initially based on a "rule of the thumb" rather than on a measurement of the actual percent leakage. The initially recommended flow rates for each cube were based on early tests. The recommended flow rate for a one foot cube was 10,000 SCFH and the maximum leak rate was 5% of this value. If the measured leakage exceeded 5% of the flow rate (500 SCFH), the cube was rejected.

2.7.2 Furnace Tests

The need for testing an assembled recuperator in an actual furnace atmosphere resulted in the construction of a special test furnace depicted schematically in Figure 2.10 and illustrated conceptually in Figure 2.11. The test furnace was designed to be versatile and allows testing to determine the following:

- Heat transfer characteristics
- Pressure drop measurements

- Leakage measurements
- Core geometry variations
- Housing and gasket materials integrity
- Operation under unbalanced flow conditions
- Furnace temperature cycling effects
- Temperature limitations
- Furnace load variations
- Energy savings
- Recuperator durability
- Quality assurance

A key feature of the test furnace is the load simulator, which consists of Inconel tubes through which air can be blown. This arrangement allows simulation of various furnace load conditions. The furnace load is measured by allowing the furnace to come to a steady state condition without the load simulator fan in operation. The maximum furnace temperature is limited by the Inconel tubes, which tend to sag when heated above 2,100 °F. Air is then blown through the Inconel tubes and the exhaust gas temperature from the furnace is noted for the same burner flow conditions. The new furnace load is calculated from the reduction in the furnace exhaust gas temperature. Various furnace loads can be calculated directly from a compilation of such data.

The measured performance of a 10 x 10 x 10 inch and a 12 x 12 x 12 inch core is presented in Figures 2.12 and 2.13 respectively. These figures show the preheated air temperature, and the gas and air side pressure drops for various furnace exhaust flows and temperatures. As a result of these tests, the recommended flow rate of 10,000 SCFH per cubic foot of core was revised to include the range shown in Table 2.5. Testing of various matrices is continuing and the flow recommendations may change depending on the additional data collected.

One of the important objectives of these furnace tests was to determine the heat transfer and flow friction characteristics of each core geometry so as to be able to size a heat exchanger to operate within the given specifications of a particular furnace application. The usefulness of the Fanning friction factor "f" and the Colburn heat transfer factor "j" in sizing and predicting the performance of a heat exchanger are discussed in Appendix B. The usual design problem specifies both pressure drop and heat transfer performance. In the equations presented in Appendix B, no direct relationship exists between pressure drop and heat transfer performance. An approximate expression(3) relating these factors is:

$$\frac{V_1^2/2g_c}{P_1/\rho_1} \sim \frac{\Delta P/P}{NTU} \times \frac{\ell_m}{\ell_1} \times \frac{Nst}{f} \times E \quad 2-1$$

one side

FIGURE 2.7

FLOW CHART ILLUSTRATING STEPS IN MANUFACTURE OF THE CERAMIC CORE

Figure 2.8

Typical Ceramic Recuperator Installation

Figure 2.9

Conceptual Illustration of Leak Testing System

FIGURE 2.10

SCHEMATIC OF TEST FURNACE

Figure 2.11

Conceptual Illustration of Test Furnace

NOTES:

ΔH input = ΔH wall loss + ΔH load + ΔH exhaust

ΔH exhaust = ΔH out of recuperator exhaust +
 ΔH out of recuperator preheat + ΔH wall
 loss of recuperator

ΔH input is controlled by Control Motor #2

Vary load by Control Motor #1

By varying load we can maintain input and ex-
 haust flows constant and vary temperature.

Figure 2.12

Recuperator Performance

Matrix Size - 10" x 10" x 10"
(Model RC600)

Specifications

- Product Code: 8111014
- Material: Cordierite
- Recommended Flow:
2500-12000 SCFH
- Recommended BTU:
250,000 - 1,000,000
- Exhaust Hole Size: .300" x .750"
- Air Hole Size: .125" x .750"

Temperature Rise of Preheated Air
Under Balanced Flows Using 10%
Excess Air With Model R0600

Example: 1MM BTU Burner using 10% excess air.
Air Flow = 5500 SCFH
Gas Flow = 500 SCFH
Exhaust Flow = 6000 SCFH

Ceramic Recuperator Pressure Drop
at Various Temperatures And Flows -
Model R0600

Preheated Air Temp. at 2000°F Exhaust = 770+RT(70)=840°F.
Pressure Drop Exhaust .13" H₂O Column
Pressure Drop Air .8" H₂O Column

Pressure Drop (Inches Water Column)

Figure 2.13

Recuperator Performance

Matrix Size - 12" x 12" x 12"
(Model R1000)

Specifications

- Product Code: 8112014
- Material: Cordierite
- Recommended Flow:
6000-18000 SCFH
- Recommended BTU:
500 000 - 1,500,000
- Exhaust Hole Size: .300" x .750"
- Air Hole Size: .125" x .750"

Temperature Rise of Preheated Air
Under Balanced Flows Using 10%
Excess Air With Model R1000

Example: 1111 BTU Burner using 10% excess air.
Air Flow = 11000 SCFH
Gas Flow = 1000 SCFH
Exhaust Flow = 12000 SCFH

Ceramic Recuperator Pressure Drop
at Various Temperatures And Flows -
Model R1000

Preheated Air Temp. at 2000°F Exhaust = $770 + RT(70) = 840^\circ\text{F}$
Pressure Drop Exhaust .35" H₂O Column
Pressure Drop Air 1.3" H₂O Column

TABLE 2.5CERAMIC RECUPERATOR PERFORMANCE

Effectiveness, E	- Min.	40 - 50%
	- Max.	60 - 70%
Gas Flow (Recommended)	- Min.	3,000 SCFH
	- Max.	15,000 SCFH
Heat Rating (Recommended)	- Min.	300,000 BTU/h
	- Max.	1,500,000 BTU/h
Cost	- Min.	\$2,000
	- Max.	\$2,800

Another useful expression is:

$$NTU_{\text{one side}} = (E \frac{L}{r_h} N_{st})_{\text{one side}} \quad 2-2$$

Where:

- V_1 = inlet velocity
- P_1 = inlet pressure
- ρ_1 = inlet density
- ΔP = pressure drop
- NTU = number of transfer units
- N_{st} = Stanton number
- f = friction factor
- E = effectiveness
- L = length of flow path
- r_h = hydraulic radius

The experimental data from the test furnace and the above two equations are then used to calculate the terms " f " and N_{st} from which " j " is obtained. These calculations are illustrated in Appendix E. A computer program was written to calculate the various parameters, and the computer output for a single test run on a one cubic foot matrix is given in Table 2.6. The values of " f " and " j " determined from the test runs to date are plotted versus the Reynolds Number in Figure 2.14. Since the maximum Reynolds Number was about 2,000, flows were primarily in the laminar flow region. Tests are presently being conducted to determine " f " and " j " at higher Reynolds Numbers. While the " f " and " j " data presented in Figure 2.14 are based on actual furnace tests of a housed core, the values are approximate because of the use of Equation 2.1. Therefore core transient tests in a wind tunnel were initiated. These tests will permit more accurate sizing of the heat exchangers and will give " f " and " j " data over a wide Reynolds Number range.

2.7.3 Wind Tunnel Tests

The heat exchanger designer is usually required to meet some specification that requires a design on the basis of a minimum for one or more of the following parameters: frontal area, length, volume, weight, pressure drop and cost. He may have various compact heat exchanger surfaces available from which to choose an optimum design that would satisfy the design criteria. The prediction of the performance of a heat exchanger can be accomplished by using theoretical or other idealized test results reported in the literature. However, actual performance could diverge considerably from the theoretically predicted performance because of the complicated shape of the individual flow passages, nonuniformities in manufacture, surface roughness, etc. To increase confidence in a specific design, it is desirable to base the design on actual test data.

Geometry of the Crossflow Recuperator Matrix

Distance of Air Travel in Matrix = 12.0 (In.)
Distance of Gas Travel in Matrix = 12.0 (In.)

	FRE FLO AREA (SQ IN)	HYDRAL DMTR (IN)	SRFC AREA (SQ IN)	PRESSURE PSI	PRES DRO INWC	L/HD
Preheat	22.801	0.192	5695.434	15.740	0.740	62.45
Exhaust	57.206	0.397	6916.918	14.766	0.140	30.23

	FLCW	SP.GR.	MASS FLOW .	DENSITY	CH4	C2H6	O2	N2
	SCFH	AIR=1.	LB/HR	LB/CUFT	WT%	WT%	WT%	WT%
Gas Birkahn	500.0	0.6000	22.950	0.0459	83.25	8.67	0.0	8.08
Air	5090.0	1.0000	389.385	0.0765	0.0	0.0	23.14	75.53
Comb Propts	5531.5	0.9578	405.308	0.0733	0.0	0.0	1.68	87.38

AVERAGE PREHEAT TEMP = 511.5 (DEG. F) (539.4 DEG K)
AVERAGE EXHAUST TEMP = 1427.0 (DEG. F) (1048.0 DEG K)

C MIN/C MAX	0.79338
LOG MEAN TEMP DIFFERENCE	477.66162

	AVG TEMP F	THER OND BTU/HR FT F	EFFECTNESS	HT TRANS BTU/HR	THER CONDUCTANC	RENIDS #
Preheat	511.50	0.0244	0.4837	85549.9	4.5283	570.72
Exhaust	1427.00	0.0398	0.6175	-109203.8	4.7596	329.44

	NTU	PRANDTL #	STANTON #	HT TRANS COEF	FRICTION FACTOR
Preheat	1.807	0.719	0.014952	0.011999	0.087235
Exhaust	2.306	0.779	0.030889	0.026152	0.097679

Figure 2.14

"f" and "j" Versus Reynolds Number for a 1 Ft³ Matrix

Gas Flow Inside a Rectangular Tube:
 $\alpha = 6.02$; air side
 $\alpha = 2.40$; gas side

Compact heat exchanger surfaces are usually tested by either the condensing steam method or the single blow method used for regenerative type surfaces. The condensing steam method was rejected for the ceramic heat recuperator because the porosity of the ceramic material could cause the transpiration or absorption of some of the steam, thereby ruining the calorimetry. Air Preheater Company(4), Wellsville, New York, with over 20 years of experience in the single blow testing method, was contracted by GTE to conduct tests on two cores of different flow passage dimensions. This subsection discusses the test procedure, the results and the use of the information in optimizing the design of the heat exchanger.

The Single Blow Technique

The single blow method is summarized on Figure 2.15. At the top of the figure is a diagram of the wind tunnel set-up. Flow is from right to left through an almost square duct. An orifice plate is used for flow measurement. Next, the flow passes through a nichrome screen which can be electrically heated. Next, comes a thermocouple grid, then the heat transfer surface (core), and another thermocouple grid. Static pressure rings are located upstream and downstream of the test core. The flow exits the tunnel through a variable size restriction which is used to set the flow rate of the tunnel.

The single blow procedure is conducted as follows:

1. The test core is carefully mounted in the special flanged box that becomes part of the tunnel during the test. Insulating plywood is used to separate the core from the metal walls to reduce conduction effects. Care is exercised to ensure smooth flow conditions around the edges of the core at both the entrance and the exit. The core itself is carefully positioned so that the geometry of the passages is identical to that actually employed in the heat exchanger.
2. The fan is turned on and the exit plug adjusted until the desired flow is indicated by the orifice plate measurement.
3. The power to the screen is turned on to its fixed setting. This setting is adjusted for different flows so that the air leaving the screen is heated to about 20 degrees Fahrenheit above ambient.
4. The temperature of the air flow downstream of the test core is measured and the screen left on until the test core is at a uniform temperature as indicated by steady air out temperature.
5. Power is switched off the screen but the air flow is maintained at its steady rate and a recording of the downstream air temperature is made. This temperature trace is the vital measurement: from it, the heat transfer characteristic of the heat transfer surface will be determined by the use of mathematical data reduction procedures.

FIGURE 2.15
SINGLE BLOW WIND TUNNEL TEST PROCEDURE

1. TEST SET-UP

2. ANALYSIS

OBTAIN: F FROM Δp , TEMPERATURE, GEOMETRY, AND FLOW MEASUREMENTS

OBTAIN: J FROM TEMPERATURE COOLING CURVE SHAPE, FLOW, GEOMETRY, AND TEMPERATURES

COMMENT: A TRANSIENT PROCEDURE

6. The data reduction method is based upon an analysis of the test core as a heat exchanger. The rate at which the core cools, and thus the shape of the outlet air temperature trace, depends upon the following variables:
 - a. the air flow rate and properties (specific heat, thermal conductivity, and viscosity),
 - b. the mass and specific heat of the test core,
 - c. the heat transfer coefficient of the test core at the test conditions, and
 - d. the heat transfer surface area of the test core.

Basically, the analysis is simple: the warm core experiences a step function change in the air temperature. Energy in the core flows to the cooler air flowing through it. As time goes on, the core will eventually be cooled to the inlet air temperature. The key result--the shape of the air outlet temperature versus time plot is, for all other variables being the same, a unique function of the heat transfer coefficient of the core surface.

Friction Factor Determination

The pressure drop across a heat exchanger surface is generally broken down into several components:

1. the actual frictional pressure drop of the surface,
2. the pressure drops associated with the entrance contraction and exit expansion, and
3. the flow acceleration effect due to the heating of the gas as it flows through the heat exchanger.

For the wind tunnel test, the procedure is simply to measure the total pressure drop across the core. The flow acceleration term will be zero because the flow is isothermal. The measured loss is thus the combination of entrance and exit effects plus passage flow friction and can be treated as desired. The entrance and exit effects are subtracted from this pressure drop leaving only the heat exchanger passage flow friction.

Heat Transfer Coefficient

The single blow method is conducted at about 20 °F ambient conditions. Thus, the effect of variations in properties with temperature is not significant in the data reduction.

In the mathematical method employed by Air Preheater, it is assumed that the thermal conductivity of the heat exchanger material is not a factor. Specifically, it is assumed that the coefficient is infinity in the direction normal to the flow (material temperature is uniform throughout its thickness) and zero in the flow direction (heat cannot transfer by conduction through the material from hot places to cold places). Calculations indicate that for the ceramic core under test conditions this condition is adequately met.

The GTE heat exchanger is a cross-flow configuration with passages of different geometry for the air and gas. These passages are straight rectangular shaped channel on each side. By constructing a test core from just the passages of just one side (air or gas), a core is formed that resemble the matrix of a rotary regenerator. The configuration meets the requirements of a valid measurement by the single blow method. The procedure is valid as long as the special test core has passages that are identical to those of the crossflow standard product. The specially built test core in the regenerator configuration is shown in Figure 2.16.

The range of test flow rates through the test cores must be consistent with the range of flow conditions experienced in the practical application of the heat exchanger. The parameter that best expresses this is the Reynolds Number since the test data will be correlated using it and the performance prediction method also uses this correlation. Reynolds Number is principally a measure of flow rate but is modified by variations in gas viscosity and passage geometry. About 25 test runs were performed for each friction factor and heat transfer correlations for each core.

Test Results and Discussion

The results of the wind tunnel tests are summarized for the air side core in Figure 2.17 and for the gas side (higher porosity) core in Figure 2.18. The dimensions shown on these figures were the measured average values for fifty passages for the test cores and were used for the subsequent calculations. The passage dimensions of the unfired or raw cores were 0.125 x 0.750 in. (air side) and 0.300 x 0.750 in. (gas side).

The friction factor, f , is due strictly to wall shear effects and causes a pressure drop:

$$\Delta p = \frac{4fL}{D} \frac{\rho V^2}{2} \quad 2.3$$

In a heat exchanger design, the pressure drop due to entrance, exit, and flow acceleration must be added to the passage pressure drop to compute the total pressure losses.

A dimensionless heat transfer number j , the Colburn number, is defined in Equation 2.4 with $G = m/A_c$, the mass velocity through the core; h , the heat transfer coefficient; C_p , the gas specific heat; and Pr , the gas Prandtl Number:

FIGURE 2.16
CROSS FLOW RECUPERATOR
AND REGENERATOR CONFIGURATION

FIGURE 2.17

DIMENSIONLESS CHARACTERISTICS

GTE SYLVANIA TEST CORE

$L/D_H = 139.1$

CERAMIC MATERIAL

$C_p = 0.1801 \text{ BTU/lb-}^\circ\text{F}$

(AIR SIDE)

$f = 8.7357 \text{ Re}^{-0.9119}$
 $\text{Re} < 1750$

$J = 7.2466 \text{ Re}^{-0.9972}$
 $\text{Re} < 1900$

HYDRAULIC DIAM. - $D_H = 0.01393 \text{ FT}$

FREE AREA - $FA = 0.5271 \text{ FT}^2/\text{FT}^2$

HEATING SURFACE - $HS = 151.33 \text{ FT}^2/\text{FT}^3$

$f = 0.02467 \text{ Re}^{-0.1281}$
 $\text{Re} > 3300$

$J = 0.02177 \text{ Re}^{-0.1316}$
 $\text{Re} > 3800$

FIGURE 2.18

DIMENSIONLESS CHARACTERISTICS
GTE SYLVANIA TEST CORE

$$L/D_H = 64.3$$

CERAMIC MATERIAL

$$C_p = 0.1801 \text{ BTU/lb-}^\circ\text{F}$$

(GAS SIDE)

$$j = (h/G C_p) Pr^{2/3} \quad 2.4$$

These dimensionless characteristics are plotted versus the Reynolds Number. The Reynolds Number is based on hydraulic diameter, D_h ; the mass flow rate through the core, m ; the passage flow area, A_c ; and the dynamic viscosity, μ ;

$$N_R = \frac{m D_h}{A_c \mu} \quad 2.5$$

Figures 2.17 and 2.18 indicate that the transition zone from laminar to turbulent flow occurs at a Reynolds Number of 2000 for the air side passage and about 2700 for the gas side passage. The significant effect shown on the j curve for both size passages is the enhanced heat transfer as the flow enters the turbulent range. (Note the bump in the j curves.) The enhanced heat transfer is probably due to the roughness of the ceramic surfaces. The practical value of this phenomena can be realized by designing the heat exchanger to obtain Reynolds Numbers that correspond to the peak in the j curves.

Figure 2.19 summarizes comparisons with Kays and London results shown as solid lines and dashed lines representing the present study. Corresponding results can be compared: $\alpha = 2.6$ and $L/D_h = 64$ with $\alpha = 3.0$ and $L/D_h = 40$, and $\alpha = 6.9$ and $L/D_h = 139$ with $\alpha = 8.0$ and $L/D_h = 100$. These are the closest available aspect ratios and length ratios.

The heat transfer results, j , on Figure 2.19 show excellent agreement considering the slightly different α and L/D_h in the laminar range (below $Re = 2000$) and considering an error analysis of j suggests an uncertainty of 15%. The turbulent range gives expected divergence since the ceramic channel of this test is much rougher than the metal channels of the Kays and London experiment: surface roughness enhances heat transfer and flow friction in the turbulent region.

The friction factor data on Figure 2.19 shows less agreement, although the same trends. The friction factor data is generally within 5%, but in the gas side case where $\alpha = 2.6$ and $L/D_h = 64$ a relatively large hydraulic diameter and porosity, p , caused larger uncertainties as the Reynolds Number decreased. This is reflected in larger data scatter. Additionally, the raw data is modified by subtracting the entrance and exit losses to give the passage friction factor. This experiment used an Air Preheater in-house entrance and exit loss correction which gives excellent results for the parallel plate regenerator and for other typical regenerator surfaces. Kays and London used a different entrance and exit loss factor as reported in (5) and summarized in (3). This discrepancy could cause some of the disagreement.

Experimental results gave good agreement with the Kays and London data. This implies that the geometry of the ceramic passages is sufficiently uniform to use their data for aspect ratios and passage length to hydraulic diameter ratios which were not tested here.

FIGURE 2-19

COMPARISON OF EXPERIMENTAL RESULTS
WITH KAYS AND LONDON RESULTS, REFERENCE 2

NOTE: $L/d = L/D_h$

The heat transfer data and friction factor characteristics should be incorporated into a crossflow recuperator design procedure. Since crossflow recuperators have a hot corner which may govern the failure of the entire unit, and since large temperature differences require some allowance for variable properties, a multi-zone numerical model is almost mandatory. Additionally, the heat transfer coefficient should always be used with a fin effectiveness.

The passage friction factor can be incorporated into a recuperator design by adding on flow acceleration effects, entrance and exit losses. Since Air Preheater's entrance and exit loss relationships have never been validated for rectangular passages, those of Kays and London are recommended.

Optimization and Design Considerations

Heat exchangers can be optimized on the basis of either surface area, frontal area, length, volume, weight, pressure drop or cost. It is difficult to assess, from j and f data alone, the relative merits of two or more surfaces. Further, since the heat exchanger is but one component of a whole system, it is difficult to select a universal method of comparing surfaces short of a complete design study. However, for one mode of fabrication which establishes the cost in terms of geometry, there are comparison techniques that will provide a guide to the designer. The following methods illustrate how the j and f curves can be used to compare the surfaces tested.

Figure 2.20 compares the surfaces by the commonly used parameter j/f . A high value of the ratio j/f indicates a high heat transfer rate for the same pressure drop. Selecting a surface with the highest value of j/f will result in a heat exchanger with the lowest surface area requirement. For this reason the parameter j/f is referred to as the flow area goodness factor. A review of the flow area goodness factor in Figure 2.38 in Kays and London(3) indicate that the j/f values for the two ceramic passages are generally higher than for the plate fin and other metallic surfaces. This indicates that the ceramic heat exchanger would require less surface area for the same pressure drop as metallic surfaces resulting in a more compact heat exchanger.

Specifically with regard to the two surfaces tested, the air side surface (0.125 x 0.750 in.) is the preferred surface up to a Reynolds Number of 4,500. In the highly turbulent range the gas side (0.300 x 0.750 in.) surface is preferable. There is another interesting aspect to the j/f curve in Figure 2.20. Note that the air side passage peaks at a Reynolds Number of about 4,000. The recuperator should therefore be designed for the air flow in this Reynolds Number range. On the other hand the gas side passage peaks at a Reynolds Number of about 6000. While the design of the gas side surface for this flow condition would result in a high heat transfer rate, the absolute values of the pressure drop would be higher as the gas velocity is directly proportional to the Reynolds Number for a constant temperature. If the gas side pressure drop is a limiting design criteria,

FIGURE 2.20
FLOW AREA GOODNESS FACTOR

one would prefer to design the gas side surface for a flow condition that results in a Reynolds Number of from 300 to 500.

Another method(6) of comparing heat exchanger surfaces is a plot of the heat transfer power, h_{std} , versus the friction power expenditure, E_{std} .

$$h_{std} = \frac{C_p \mu}{N_{pr}^{2/3}} \frac{1}{D_h} j N_R \quad 2.6$$

$$E_{std} = \frac{1}{2g_c} \frac{\mu^3}{\ell^2} \left(\frac{1}{D_h}\right)^3 f N_R \quad 2.7$$

The above equations were evaluated for the two surfaces at standard conditions of dry air at an average temperature of 500 °F and a pressure of one atmosphere and reduce to:

$$h_{std} = 1.521 j N_R \quad (\text{air side passage})$$

$$E_{std} = 0.0393 f (N_R / 1000)^3$$

and

$$h_{std} = 0.683 j N_R \quad (\text{gas side passage})$$

$$E_{std} = 0.00356 f (N_R / 1000)^3$$

The plot of h_{std} versus E_{std} in Figure 2.21 provides a comparison of the volume (and hence weight) of an exchanger with the friction power expenditure. High values of h_{std} indicate a small volume and this type of plot is usually referred to as the volume goodness factor. Thus for any specific friction power expenditure E_{std} , that is allowable, the surface with the highest h_{std} is desired. Of the two ceramic surfaces tested, the air side surface (0.125 x 0.75 in.) is the preferred surface.

Another method of comparing surfaces is that of Grober et al(7). The length of the flow path in a heat exchanger is inversely proportional to j and consequently the face area is proportional to $(1/\sigma)(f/j)^{1/2}$ and the volume is proportional to $(1/\sigma)(f/j^3)^{1/2}$. The length, face area and volume also depend on other parameters, such as, the hydraulic diameter, the film coefficient, the passage width, the pressure drop and the temperature of operation. The use of a modified Reynolds Number, $(f/j)^{1/2} N_R$, provides a parameter that also depends on these factors. The length, face area, and

FIGURE 2-21
VOLUME GOODNESS FACTOR

volume parameters are plotted versus the modified Reynolds Number in Figures 2.22, 2.23 and 2.24. Surfaces with lower values of these parameters would result in a more compact heat exchanger and would be the preferred surface.

In summarizing the j and f versus N_R curves and the subsequent analysis, it may be concluded that the 0.125 x 0.750 in passage is, in general, the preferable surface and would result in a more compact heat exchanger.

Analysis of the f and j curves indicates that the performance of the recuperator could be improved by operation at the optimum Reynolds Number and that an increase in the air path length would increase the temperature effectiveness of the recuperator. This led to the decision to consider a multi-pass cross-flow system on the air side. A 12 x 12 x 12 in. and a 12 x 12 x 18 in. were designed with a triple pass arrangement on the air side and are currently being tested on the test furnace. These triple pass units have increased the effectiveness, and hence heat recovery, of the recuperator while also increasing the air side pressure drop. These encouraging results would prompt the use of the triple pass ceramic recuperator on future installations.

FIGURE 2.22
LENGTH PARAMETERS FOR MATRICES

FIGURE 2-23
FACE AREA PARAMETERS FOR MATRICES

FIGURE 2.24
VOLUME PARAMETERS FOR MATRICES

2.8 System Economics

The most important consideration in deciding whether to install a recuperator on an existing furnace is the system economics. The average fuel savings and pay-back period depend on many factors such as the type of furnace, type of operation, exhaust gas temperature, air preheat, fuel costs, controls, installation costs, etc. In general, pay-back periods are favorably influenced by:

- High furnace temperatures
- High fuel costs
- Continuous furnace operation
- High combustion air temperatures
- High excess air

In order to determine the economics of a recuperator installation, the following information is required:

- Type of fuel, heating value, excess air at the burner
- Fuel costs \$/MM BTU or \$/ft³ or \$/gallon
- Cost of electric power \$/KWH, \$/year
- Operating time/day or year
- Fuel input MM BTU/hour with cold combustion air
- Exhaust gas temperature

The cost of a total retrofit installation is generally 5 to 6 times the cost of the recuperator itself. This is because new burners, controls, air duct work, and insulation are required. Table 2.7 presents the cost of a ceramic recuperator system. The major operating cost for a recuperator system is the cost of pumping power required to overcome the air and gas side pressure drops. In some cases, the recuperator can be sized so that the existing blower can be used. The ceramic recuperator requires little or no maintenance and therefore O&M costs were not considered in this study.

A common approach to analyzing the capital investment for a ceramic recuperator is to determine the annual net savings after considering depreciation, interest and taxes. The annual net savings can then be compared with the capital costs to determine the investment payback period. Pay-back periods can vary for each installation but are generally from nine months to two years.

TABLE 2.7CERAMIC RECUPERATOR SYSTEM COSTS

<u>Cost Item</u>	<u>\$ Minimum</u>	<u>\$ Maximum</u>
Ceramic Recuperator	2,000	2,800
New Burners	1,000	1,500
New Air Duct Work and Insulation	1,000	1,500
Modifications to Exhaust Gas System	500	1,000
Controls	500	1,000
Engineering and Installation	<u>3,000</u>	<u>5,000</u>
TOTAL COSTS	8,000	12,800

2.9 Cost Performance Analysis

The cost of a recuperator also depends upon operating temperatures, the allowable air and gas pressure drops, and the thermal effectiveness required. The plant engineer needs to determine whether or not the installation of a recuperator is justifiable. An extremely good analysis of this problem is presented by Anderson and Thurner(8) and forms the basis for this discussion. Data used in this analysis was collected in tests performed in the test furnace on three recuperators, a 10 x 10 x 10 in. single pass, a 12 x 12 x 12 in. single pass and a 12 x 12 x 12 in. triple pass unit.

Figure 2.25 shows the capital cost of a recuperator in dollars per SCFM of air as a function of the air side effectiveness. For a given recuperator and air flow, the cost increases rapidly with increased air effectiveness. Furthermore, for a given recuperator, incremental increases in effectiveness result in larger increases in the capital costs per SCFM. Note that for a given recuperator increasing the air flow reduces the capital cost but also decreases the thermal effectiveness of the unit. Of significant interest is the fact that for a given capital cost per SCFM, the triple pass arrangement is considerably superior to the single pass in terms of air side effectiveness. Figure 2.26 indicates that if the air flow is increased for a given size of recuperator, the heat recovery rate increases even though the air side effectiveness decreases. This would result in lower payback periods.

The major operating cost of a recuperator is the blower power consumption to overcome the friction losses on the air and gas sides of the crossflow unit. Figure 2.27 shows the variation in air side pressure drop with effectiveness. It is obvious that increasing the air flow through a given size recuperator would increase the pressure drop. It is important to note that the triple pass 12 in. cube shows a significant increase in pressure drop as compared to the 12 in. single pass cube. Retrofitting a furnace with a triple pass 12 x 12 x 12 in. recuperator must therefore take into consideration the pressure rating of the blower.

In deciding to invest capital in heat recovery equipment, a plant engineer must perform an investment payback analysis. This analysis should properly consider the annual net savings after consideration of depreciation, interest, maintenance and taxes. The payback period could then be computed by dividing the total installed costs by the net annual savings after taxes. Note that the proper procedure should subtract the annual depreciation, interest, etc. together with the annual operating cost from the annual fuel savings to obtain the net annual savings before taxes. Therefore, for a fixed installed cost, the payback period would increase with increasing operating costs. The results of Anderson and Thurner(8) indicate that as the thermal effectiveness decreases (by increasing air flow for a given recuperator), the payback period decreases, goes through a minimum and then again increases. This occurs because, as the air flow is increased, the pressure drop increases and this in turn results in increased operating costs. Their results show that the operating effectiveness of the recuperator should be selected for the minimum payback period.

FIGURE 2-25
CAPITAL COST VS AIR SIDE EFFECTIVENESS

FIGURE 2-26

CAPITAL COST VS HEAT RECOVERY

FIGURE 2-27
AIR SIDE PRESSURE DROP

Figure 2.28 is a plot of the payback period as a function of the effectiveness for the three ceramic recuperators tested. The simplified method of calculating payback period, detailed in section 2.8, was used. It was assumed that the installed cost was 2.5 times the capital cost of the recuperator, the natural gas cost was \$3.00 per MMBTU and that operating time was 8000 hours per year. The results of Figure 2.28 indicate that the 12 in. triple pass cube is distinctly superior to the 12 in. single pass cube.

The curves show that it is possible that the payback period could go through a minimum. However, a minimum payback period was not obtained because the test furnace blower and air measurement devices prevented further increases in the air flow.

FIGURE 2-28
PAYBACK PERIOD VS EFFECTIVENESS

3. DEMONSTRATION PROGRAM AND RESULTS

The overall objective of the demonstration program was to establish the technical and economic feasibility of using ceramic heat recuperators for industrial heat recovery. The specific objectives of the program were to:

- o Determine the heat transfer performance of the ceramic recuperator.
- o Establish the energy savings by recuperation.
- o Demonstrate minimum maintenance requirements in typical furnace operation.

It was intended that the demonstration program would also:

- o Determine the durability of the ceramic core.
- o Determine the operating requirements of the burners and controls with recuperation.
- o Establish the overall system costs and pay-back period.

Five demonstrations were undertaken on furnaces operating between 1,400 and 2,600 °F with ratings of 0.4 to 2.2 MM BTU/hour. Three projects are currently operational and two have been terminated. This section describes each of the demonstrations and discusses the results.

3.1 Bliss Mill Furnace Demonstration

Description

The furnace selected for the first demonstration was a molybdenum reheat furnace for a Bliss rolling mill. A photograph of the furnace and rolling mill are shown in Figure 3.1. Billets or slabs of molybdenum are manually fed through the furnace door in a batch-type operation, are heated in the furnace, and then passed through the rolling mill. The slabs pass through the mill repeatedly till the desired size reduction is achieved. The furnace operates at 2000 °F at idle and at 2600 °F at high fire. The firing rate of the furnace is 6.4 MM BTU/hour at 2600 °F. Gas and air are premixed and pass through a flexible manifold to 16 premix gas burners (eight on each side of the furnace) that fire directly into a base plate or hearth. The burner nozzles are cooled by the flowing premixed fluid. The furnace is operated in a reducing atmosphere by providing about 10% excess gas. In spite of this fact, some surface oxidation of the molybdenum occurs and a fine yellowish oxide powder can be seen exhausting through the hood.

The ceramic recuperator is placed directly in the furnace exhaust stream and is located on the top of the furnace. A schematic of the reheat furnace and recuperator is depicted in Figure 3.2. The air supply to the

Figure 3.1
Furnace and Bliss Rolling Mill

FIGURE 3.2

SCHEMATIC OF REHEAT FURNACE

DEMONSTRATION #1

recuperator is from a separate blower and the preheated air from the recuperator is vented to the stack. This was necessary because burners that can handle preheated air at 1600 °F were not available commercially.

Results and Discussion

The ceramic core selected for this demonstration was 12 x 12 x 22 inch. The geometry and core properties are presented in Table 3.1. Note that the size of each rectangular flow passage is 0.237 inch x 0.600 inch for both the gas and the air sides. These flow passages are relatively large and have been reduced in the current core models. The air side passages on current models are smaller than the gas side because the air side can accommodate a higher pressure drop. The operating variables for the Bliss Mill furnace demonstration are listed in Table 3.2 while the results of the heat transfer calculations for a typical run are listed in Table 3.3. The calculated Reynolds Numbers on both the gas and air side are very low indicating that the flow regime is laminar. Higher flow rates could have been used for the same size core or the core could have been reduced in size. The low pressure drop on the air side (1.7 in. water column) also indicates that higher air flows could have been used. It is emphasized here that the original core design for this demonstration was not optimized from the heat transfer or pressure drop viewpoints because the initial objective was merely to test the durability of the matrix in an actual furnace atmosphere.

The temperature of the exhaust gases leaving the recuperator was not monitored during the demonstration but was calculated to be about 1450 °F. While the air flow to the recuperator was measured, the air flow out of the recuperator was not monitored. Thus, actual leakage rates of air into the gas stream and the effect of this leakage in reducing the effectiveness of the unit are unknown. However, the air side effectiveness of 55% was calculated from actual temperatures and thus represents an accurate number.

This demonstration operated for a total time of 2,200 hours. At this time, the furnace exhaust temperature accidentally rose to 2800 °F, which caused softening and slumping of the ceramic material and destruction of the demonstration unit. The demonstration was then terminated.

Economics

Since the preheated air was vented and not fed to the burners as combustion air, actual process fuel savings were not realized. For the purpose of determining the payback period, the savings in energy were assumed to be equal to the heat transferred to the preheated air. The actual savings in energy for a typical system where the burners are supplied with preheated air can be expected to be higher because of the increase in the furnace efficiency. Nevertheless, based on this assumption, the pay-back period for a typical installation of this type was calculated to be 2.63 years as presented in Table 3.4.

TABLE 3.1

CORE GEOMETRY FOR BLISS MILL DEMONSTRATION

Crossflow Recuperator - 12 x 12 x 22 inch Core

Preheat Window = $10.1 \times 20.2 = 204.02$ (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = $10.1 \times 20.2 = 204.02$ (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.050	0.237	0.600	556	0.045	2.53
***	0.050	0.237	0.600	556	0.045	2.53
	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	79.137	0.340	11179.355	15.091	1.700	35.32
***	79.137	0.340	11179.355	14.748	0.220	35.32

* = Aspect Ratio = Distance between ribs/rib height

** = Preheat

*** = Exhaust

TABLE 3.2BLISS MILL FURNACE DEMONSTRATION OPERATING VARIABLES

(12 x 12 x 22 inch Core)

Natural Gas Consumption	650 SCFH
Exhaust Gas Flow	7,049 SCFH
Preheat Air Flow	6,400 SCFH
Preheat Air Inlet Temperature	100 °F
Preheat Air Outlet Temperature	1465 °F
Furnace Temperature	2600 °F
Exhaust Gas Inlet Temperature	2600 °F
Air Side Effectiveness	55%
Total Time in Operation	2,200 hours

TABLE 3.3

BLISS MILL FURNACE DEMONSTRATION
RECUPERATOR PERFORMANCE CALCULATIONS

	<u>Temp In</u> <u>Deg F.</u>	<u>Temp Out</u> <u>Deg F.</u>	<u>Flow</u> <u>SCFH</u>	<u>Flow</u> <u>ACFH</u>	<u>Capacity</u> <u>Rate</u> <u>BTU/hr °F</u>
**	100.0	1465.0	6400.0	6892.3	130.7891
***	2600.0	1465.0	7049.3	41482.6	162.3458

	<u>Flow</u> <u>SCFH</u>	<u>Velocity</u> <u>ft/sec</u>	<u>Mass</u> <u>Flow</u> <u>lb/hr</u>	<u>Density</u> <u>lb/ft³</u>	<u>Spec</u> <u>Heat</u> <u>BTU/</u> <u>lb °F</u>	<u>Viscosity</u> <u>lb/hr ft</u>	<u>Ther</u> <u>Cnd</u> <u>BTU/</u> <u>hr ft °F</u>	<u>Prandtl</u> <u>Number</u>
**	6400.00	3.484	489.60	0.0319	0.267	0.0818	0.0297	0.736
***	7049.33	20.967	512.47	0.0152	0.317	0.1223	0.0488	0.782

	<u>Effec-</u> <u>tiveness</u>	<u>Ht</u> <u>Trans</u> <u>BTU/hr</u>	<u>U</u>	<u>NTU</u>	<u>Renlds</u> <u>Number</u>	<u>Stanton</u> <u>Number</u>	<u>j</u>	<u>f</u>
**	0.5480	178527.1	4.6778	2.777	308.33	0.035999	0.029356	2.111140
***	0.5710	-185597.7	4.8919	2.904	215.82	0.035999	0.030555	0.118485

U = Unit overall thermal conductance - BTU/hr ft² F

j = Colburn heat transfer coefficient

f = Fanning friction factor

Cmin/Cmax

0.80562

Log Mean Temp Difference

491.59521

** = Preheat

*** = Exhaust

TABLE 3.4BLISS MILL FURNACE DEMONSTRATION SYSTEM ECONOMICS

Average Heat Transferred to Preheat Air = 172,143 BTU/hour

Nominal Operating Time = 120 hours/week

Recovered Energy = 20,657,160 BTU/week

Annual Savings¹ = \$3,140/year

Ceramic Recuperator	4,000
Piping	400
Controls	350
Burners	1,400
Insulation	100
Installation	1,000
Engineering	<u>1,000</u>
TOTAL SYSTEM COSTS	= \$8,250

Payback Period = 2.63 years

¹Based on 50 weeks/year operation with gas of 1,000 BTU/ft³ and costs of \$3.04/MM BTU

Problem Areas

- The operating control systems on this furnace did not lend themselves to good control of the furnace temperatures and flows. A high temperature protection system is recommended to obviate the operator error that led to the destruction of the core. Sophisticated control systems are available on the market and are recommended for use on all future applications.
- The air flow from the recuperator was not measured. Actual leakage rates were therefore not available.
- In spite of the fact that the furnace heats the molybdenum and tungsten slabs in a reducing atmosphere, surface oxidation of the metal occurs and a fine oxide powder was seen exiting through the hood. Oxidation was especially noticeable when the furnace doors were opened for feeding or rolling of the slabs. The yellow molybdenum oxide dust was deposited on the top of the core, particularly on the cooler surfaces, after the demonstration was started. Because of the possibility that the gas side passages could become plugged with the oxide deposit, an air lance system was designed and installed. This jet of air was used to blow away the deposited oxide. While the operation of the air lance system proved effective, not enough operating time was logged because soon after the installation, the core was destroyed by the high temperature excursion.
- The destroyed core was sent to Oak Ridge National Laboratory for analysis. A summary of the ORNL report is presented in Appendix F. The analysis indicated that the oxide in the waste gas stream significantly penetrated the cordierite material. The leading edges of the exhaust side inlet section showed significant microstructural modification. The oxide dust was also found to be present on the air side passages. Since leakage within the core would occur from the air to the gas side because of the pressure differential, the oxide must have entered the air side through the air blower. This was confirmed by checking the air blower filter, which was found to be clogged with the oxide dust. The long-term effect of the oxide dust on the ceramic material cannot be determined because of insufficient time of operation.

Conclusions

- While the recuperator failed at a temperature of 2800 °F due to operator error, the cordierite material proved to be stable at temperatures of 2600 °F.
- The air lance system was effective in removing oxide dust.
- The demonstration indicated that this type of installation would have a payback period of less than 2.63 years.

- The long-term effect of molybdenum oxide dust on the core material cannot be determined because of insufficient operating time of the demonstration project.

3.2 Tungsten Reduction Furnace Demonstration

Description

The second demonstration project was performed on a tungsten reduction furnace. The reduction of oxide powders to metal powders is standard practice in refractory metal production where the extremely high melting points of the metals prevent the use of smelting operators as is common in the production of steel. A schematic of the furnace and recuperator is depicted in Figure 3.3. The furnace is fired by 10 natural gas burners with a total maximum firing rate of about 2.7 MM BTU/hour. The combustion gases flow on the outside of a series of metal tubes containing the tungsten ore powder, heating the tubes to a temperature of 1700 °F. The tungsten ore is fed into the tubes from one side of the furnace and is withdrawn from the other end of the furnace after the powdered ore has been reduced to metal. Note that the atmosphere inside of the metal tubes is reducing. Since the metal tubes in the furnace are heated to 1700 °F primarily by convection, these furnaces usually use large amounts of excess air (about 125%). The furnace is divided into three zones by baffles so that the heat release rates can be varied in order to maintain the tubes at a uniform temperature of 1700 °F. Zones 1 and 2 are heated by three burners each while Zone 3 has four burners.

The furnace exhaust from each zone passes through three recuperators located on top of the furnace. Zones 1 and 2 each have a 1.83 ft³ recuperator while Zone 3 has a 5.35 ft³ unit. The preheated air is piped to each of the North American high velocity burners. Each recuperator is provided with an air by-pass so that the furnace can continue to operate in the event of a recuperator failure or shut down for maintenance.

Results and Discussion

Two matrices of 12 x 12 x 22 inch and one of 17 x 17 x 32 inch were selected for this demonstration. The core geometry and properties are listed in Tables 3.5 and 3.6. Note that the size of the flow passages are the same as those used in the Bliss Mill furnace demonstration. The size of the flow passages were selected subjectively with the thought of keeping the pressure drop low on the gas side. At the time the cores were selected for this demonstration, no attempt was made to optimize the core geometry on the basis of performance. The operating variables for the tungsten reduction furnace demonstration are listed in Table 3.7. The results of the heat transfer calculations for a typical run are detailed in Tables 3.8 and 3.9. One observes from these tables that all the recuperators were considerably oversized (the air side pressure drop was of the order of 0.6 in. water column). While the intent of this demonstration was to prove the durability of the core, it is recommended that the size of recuperators for similar applications be optimized.

Figure 3.3

SCHEMATIC OF TUNGSTEN REDUCTION

FURNACE DEMONSTRATION No. 2

**** Note:** This unit actually has three separate zones, each with its own control system. Fuel is throttled. The preheated air to the burners is shown as a single path. The furnace actually utilizes three separate preheated air lines. The schematic above has been simplified due to space limitations.

TABLE 3.5
CORE GEOMETRY FOR THE
TUNGSTEN REDUCTION FURNACE DEMONSTRATION (1.83 cu ft)

Crossflow Recuperator - 12 x 12 x 22 inch Core

Preheat Window = $10.1 \times 20.2 = 204.02$ (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = $10.1 \times 20.2 = 204.02$ (sq in)

Length of Gas Passage = 12.0 inch

<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
** 0.050	0.237	0.600	556.519	0.045	2.53
*** 0.050	0.237	0.600	556.519	0.045	2.53

<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
** 79.137	0.340	11179.355	15.524	1.600	35.32
*** 79.137	0.340	11179.355	14.766	0.080	35.32

* = Aspect Ratio = Distance between ribs/rib height

** = Preheat

*** = Exhaust

TABLE 3.6

CORE GEOMETRY FOR THE
TUNGSTEN REDUCTION FURNACE DEMONSTRATION (5.35 cu ft)

Crossflow Recuperator - 17 x 17 x 32 inch Core

Preheat Window = $15.1 \times 30.1 = 454.51$ (sq in)

Length of Air Passage = 17.0 inch

Exhaust Window = $15.1 \times 30.1 = 454.51$ (sq in)

Length of Gas Passage = 17.0 inch

<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
** 0.050	0.237	0.600	1235.69	0.045	2.53
*** 0.050	0.237	0.600	1239.80	0.045	2.53

<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
** 175.715	0.340	35165.328	15.524	0.500	50.03
*** 176.299	0.340	35282.148	14.766	0.060	50.03

* = Aspect Ratio = Distance between ribs/rib height
 ** = Preheat
 *** = Exhaust

TABLE 3.7

TUNGSTEN REDUCTION FURNACE DEMONSTRATION
OPERATING VARIABLES

	<u>Zone 1 or 2</u>	<u>Zone 3</u>
Matrix Size	12 x 12 x 22	17 x 17 x 32
Gas Consumption (SCFH)	300	700
Exhaust Gas Flow (SCFH)	6,345	15,605
Preheat Air Flow (SCFH)	6,040	14,895
Air Inlet Temperature (°F)	104	104
Air Outlet Temperature (°F)	1285	1250
Furnace Temperature (°F)	1700	1700
Gas Inlet Temperature (°F)	1700	1700
Air Side Effectiveness (%)	74	71.8
Total Time in Operation (hours)	4,200	4,200

TABLE 3.8

TUNGSTEN REDUCTION FURNACE DEMONSTRATION
ZONE 1 OR 2 RECUPERATOR PERFORMANCE CALCULATIONS

	<u>Temp In</u> <u>Deg F.</u>	<u>Temp Out</u> <u>Deg F.</u>	<u>Flow</u> <u>SCFH</u>	<u>Flow</u> <u>ACFH</u>	<u>Capacity</u> <u>Rate</u> <u>BTU/hr °F</u>
**	104.0	1285.0	6040.0	6551.1	121.3884
***	1700.0	970.0	6344.5	26353.9	133.3990

	<u>Flow</u> <u>SCFH</u>	<u>Velocity</u> <u>ft/sec</u>	<u>Mass</u> <u>Flow</u> <u>lb/hr</u>	<u>Density</u> <u>lb/ft³</u>	<u>Spec</u> <u>Heat</u> <u>BTU/</u> <u>lb °F</u>	<u>Viscosity</u> <u>lb/hr ft</u>	<u>Ther</u> <u>Cnd</u> <u>BTU/</u> <u>hr ft °F</u>	<u>Prandtl</u> <u>Number</u>
**	6040.00	3.311	462.06	0.0344	0.263	0.0778	0.0280	0.729
***	6344.47	13.321	469.69	0.0214	0.284	0.1002	0.0373	0.754

	<u>Effec-</u> <u>tiveness</u>	<u>Ht</u> <u>Trans</u> <u>BTU/hr</u>	<u>U</u>	<u>NTU</u>	<u>Renlds</u> <u>Number</u>	<u>Stanton</u> <u>Number</u>	<u>j</u>	<u>f</u>
**	0.7400	143359.7	1.5680	1.003	305.93	0.009593	0.007774	0.900343
***	0.5026	-97381.2	1.0651	0.681	241.50	0.009593	0.007945	0.072311

U = Unit overall thermal conductance - BTU/hr ft² F

j = Colburn heat transfer coefficient

f = Fanning friction factor

Cmin/Cmax

0.90997

Log Mean Temp Difference

1177.67944

** = Preheat

*** = Exhaust

TABLE 3.9

TUNGSTEN REDUCTION FURNACE DEMONSTRATION
ZONE 3 RECUPERATOR PERFORMANCE CALCULATIONS

	<u>Temp In</u> <u>Deg F.</u>	<u>Temp Out</u> <u>Deg F.</u>	<u>Flow</u> <u>SCFH</u>	<u>Flow</u> <u>ACFH</u>	<u>Capacity</u> <u>Rate</u> <u>BTU/hr °F</u>			
**	104.0	1250.0	14895.0	16155.3	298.3950			
***	1700.0	1010.0	15605.5	64822.7	327.8364			

	<u>Flow</u> <u>SCFH</u>	<u>Velocity</u> <u>ft/sec</u>	<u>Mass</u> <u>Flow</u> <u>lb/hr</u>	<u>Density</u> <u>lb/ft³</u>	<u>Spec</u> <u>Heat</u> <u>BTU/</u> <u>lb °F</u>	<u>Viscosity</u> <u>lb/hr ft</u>	<u>Ther</u> <u>Cnd</u> <u>BTU/</u> <u>hr ft °F</u>	<u>Prandtl</u> <u>Number</u>
**	14895.00	3.678	1139.47	0.0349	0.262	0.0770	0.0277	0.728
***	15605.47	14.707	1156.46	0.0212	0.283	0.1011	0.0376	0.753

	<u>Effec-</u> <u>tiveness</u>	<u>Ht</u> <u>Trans</u> <u>BTU/hr</u>	<u>U</u>	<u>NTU</u>	<u>Renlds</u> <u>Number</u>	<u>Stanton</u> <u>Number</u>	<u>j</u>	<u>f</u>
**	0.7180	341960.7	1.4450	1.183	343.37	0.008230	0.006661	0.435959
***	0.4750	-226107.1	0.9527	0.782	264.60	0.008230	0.006814	0.031026

U = Unit overall thermal conductance - BTU/hr ft² F

j = Colburn heat transfer coefficient

f = Fanning friction factor

Cmin/Cmax

0.91019

Log Mean Temp Difference

969.06152

** = Preheat

*** = Exhaust

The performance calculations detailed in Tables 3.8 and 3.9 point out a minor discrepancy in the results. The heat transferred to the air in the recuperator is considerably larger than the heat transferred from the gas. Normally the latter is slightly greater than the former because of losses. The observed difference cannot be due to leakage of the air into the gas stream because the indicated gas side temperature difference would increase, and the calculated heat transferred from the gas would be larger than with no leakage. During operation air left the recuperator at about 1285 °F but was only 900 °F at the burner. This large temperature drop between the recuperator air outlet and the burner air inlet cannot all be attributed to poor insulation. A more logical explanation is that the temperature of the preheated air leaving the recuperator was incorrectly measured, perhaps because the thermocouples received radiation from the core. It is suspected that the air outlet temperature was closer to 900 °F than 1285 °F.

If the air outlet temperatures were incorrectly measured, then the calculated air side recuperator effectiveness of 74%, in Table 3.8, is incorrect. The calculated air side effectiveness is usually slightly lower than the gas side effectiveness because the gas side capacity rate, (mass flow x heat capacity), is always higher than the air side capacity rate in these recuperator applications. The air side effectiveness was probably about 45%.

As was the case in the first demonstration, the temperature of the exhaust gas leaving the recuperator was not measured but was calculated to be 970 °F and 1010 °F for Zones 1 and 2 and Zone 3 respectively. Since the air flow out of the recuperator was not measured, actual leakage rates of air into the gas stream are not available.

The Tungsten Reduction Furnace Demonstration may be considered extremely successful because it has demonstrated that considerable energy is saved, that the associated burners and control systems operated adequately, and that the core is very durable at these operating temperatures. The three recuperators on this furnace were still in operation as of August, 1980.

Economics

Since the gas flow rates, before and after installation of the recuperator were not metered, actual percent fuel saved is not available. However, the GTE recuperator installation on a similar tungsten reduction furnace was metered and indicated that the calculated fuel savings are accurate enough, if not on the conservative side, for payback period calculations. For the purpose of computing the energy saved, the air leaving the recuperator was assumed to be the same value (900 °F) as that entering the burners. The payback period for an installation of this type was calculated to be 2.85 years and is presented in Table 3.10.

TABLE 3.10TUNGSTEN REDUCTION FURNACE DEMONSTRATION ECONOMICS

(3 Recuperators)

Average Heat Transferred to Preheat Air = 395,000 BTU/hour

Nominal Operating Time = 120 hours/week

Recovered Energy = 47.4 MM BTU/week

Annual Savings¹ = \$7,205/year

Recuperator	10,000
Piping	1,000
Controls	1,000
Burners	2,000
Insulation ²	1,500
Installation	3,000
Engineering	<u>2,000</u>
TOTAL SYSTEM COSTS	= \$20,500

Payback Period = 2.85 years

¹Based on 50 weeks/year and fuel cost of \$3.04/MM BTU²Insulation costs include labor

Problem Areas

Except for the probable error in the air outlet thermocouple, which was not corrected, there were no problems encountered during this demonstration. All three recuperators were still in operation as of August, 1980.

Conclusions

- This demonstration project is considered extremely successful because it demonstrated the following:
 - The ceramic core was proven durable at operating temperatures of 1700 °F.
 - The savings in energy were of the order of 395,000 BTU/hour.
 - The available burners and control system were suitable for using preheated air at 900 °F.
 - Minimal maintenance was required.
- The demonstration indicated that this type of installation would have a payback period of about 2.85 years.

3.3 Glass Tank Demonstration

Description

The third demonstration was attempted on a borosilicate glass furnace. The furnace is located at the York, Pennsylvania, plant of GTE Products Corporation. The operation is semi-continuous, a blend of raw materials and cullet charge are continuously fed into one end of the furnace, and melted glass is channeled into the stamping machines at the discharge end of the tank. The furnace has two exhaust flues in its center. A dual fuel, flat flame (radiation) burner at each end of the furnace is responsible for maintaining the melt temperature. Figure 3.4 shows a schematic of the recuperators positioned over the exhaust flues. A photograph of the recuperator on the furnace is shown in Figure 3.5

Results and Discussions

This demonstration was attempted on four separate occasions without success. The original design called for two ceramic cubes to be mounted in a single housing, similar to that shown in Figure 2.4 and Figure 3.5. The cores were fabricated of a fluxed zirconia silicate, Z-1000, developed by GTE because of its high resistance to borosilicate glass corrosion. Difficulties were encountered in maintaining dimensional tolerances. During heatup of this recuperator, the metal housing separated from the ceramic. Two separate recuperators were fabricated for the second attempt and the

Figure 3.4

SCHEMATIC OF YORK GLASS TANK
DEMONSTRATION #3

Figure 3.5
Glass Tank with Recuperator

furnace was modified to accommodate both. During heatup, however, the matrix and housing incompatibility again caused large leakage rates and the second attempt was terminated.

A series of material tests was conducted concurrently with the above two efforts. These tests consisted of placing bars of different materials in the exhaust stacks for varying periods of time up to six weeks. The materials were Z-1000, MAS 8200, MAS 8400 and MAS 8400 coated with alumina. The results indicated that the Al_2O_3 coating protected the cordierite from chemical attack. A technique was then developed to apply a coating of Al_2O_3 to the entire cross-flow matrix. This core was housed in the spring-loaded housing shown in Figure 2.5 and was successfully tested on the test furnace. Two of these units were installed on the glass tank. The third attempt failed when a delamination of the ceramic caused leakage in amounts greater than 3000 SCFH (50% of the air flow). It was possible that the delamination was caused by mechanically weak ceramic bonds. Therefore, a technique was developed to test the cores for mechanical strength by suspending a load of 400 pounds from the core. The 400 pound load was equivalent to three times the load on a matrix operating at a static pressure of 1 psi. The ceramic bonds successfully supported the load without delaminating.

Two Al_2O_3 ceramic cores that passed the above test were used in the fourth demonstration attempt. To ensure adequate exhaust gas flow through the recuperators, an aspiration system was installed on the exhaust outlet from the recuperators. This type of an ejector system would overcome the increased pressure drop resulting from glass deposits in the core. The core geometries of the two 12 x 12 x 12 inch matrices are shown in Table 3.11. Note that the gas side flow passage is 0.648 inch x 0.281 inch and that of the air side is 0.648 inch x 0.112 inch. The fourth demonstration effort ran for approximately 200 hours and operated at the glass melt temperatures for about 105 hours. At about 200 hours, the leakage rates reached 2000 SCFH and the demonstration was terminated. Figure 3.6 is a plot of the glass melt temperature, air preheat temperature and leakage rates versus time. Note that the leakage increased dramatically once the feeder was turned on (production started) and reached 2000 SCFH in about 50 hours. While the original melt seemed to have a high vapor pressure, the glass became more volatile after the feed material was introduced into the tank. The material carryover and deposition in the core attacked the ceramic causing excessive leakage and failure.

The operating variables for the fourth demonstration attempt are listed in Table 3.12. The natural gas consumption was measured at 790 SCFH compared with 1667 SCFH without recuperation. This reduction in fuel consumption was primarily due to the air preheated to 1175 °F and better heat transfer from the radiation type burners due to the higher adiabatic flame temperature. Some of the savings may be attributed to better stoichiometric controls on the burner. The results of the heat transfer calculations are presented in Table 3.13. The air side effectiveness was calculated to be about 47%.

TABLE 3.11CORE GEOMETRY FOR GLASS TANK RECUPERATORS

Crossflow Recuperator - 12 x 12 x 22 inch

Preheat Window = $10.3 \times 10.1 = 104.03$ (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = $10.3 \times 10.1 = 104.03$ (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	320.10	0.039	5.78
***	0.042	0.281	0.648	320.10	0.039	2.30

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	23.232	0.191	5838.652	15.740	1.500	62.83
***	58.287	0.392	7136.984	14.766	0.370	30.61

* = Aspect Ratio = Distance between ribs/rib height

** = Preheat

*** = Exhaust

Figure 3.6

RECUPERATOR LEAKAGE (SCFH) AND GLASS TEMPERATURES DURING EFFORT #4
HOURS OF OPERATION

TABLE 3.12GLASS TANK FURNACE DEMONSTRATION OPERATING VARIABLES

	<u>Front Burner</u>	<u>Rear Burner</u>	<u>Average</u>
Natural Gas Consumption (SCFH)	380	410	395
Exhaust Gas Flow (SCFH)	4,250	4,550	4,400
Preheat Air Flow (SCFH)	3,880	4,145	4,012
Preheat Air Inlet Temperature (F)	100	100	100
Preheat Air Outlet Temperature (F)	1115	1241	1178
Furnace Temperature (F)	2370	2280	2325
Exhaust Gas Inlet Temperature (F)	2460*	2370*	2415
Exhaust Gas Outlet Temperature (F)	1325	1345	1335
Total Time in Operation (hours)	200	200	200

*Assumed to be 122 °F greater than actual melt temperature - 2370 °F by optical methods.

TABLE 3.13

GLASS TANK DEMONSTRATION
RECUPERATOR PERFORMANCE CALCULATIONS

	<u>Temp In</u> <u>Deg F.</u>	<u>Temp Out</u> <u>Deg F.</u>	<u>Flow</u> <u>SCFH</u>	<u>Flow</u> <u>ACFH</u>	<u>Capacity</u> <u>Rate</u> <u>BTU/hr °F</u>			
**	100.0	1178.0	4400.0	4738.5	87.5484			
***	2415.0	1335.0	4800.8	26542.7	108.7688			

	<u>Flow</u> <u>SCFH</u>	<u>Velocity</u> <u>ft/sec</u>	<u>Mass</u> <u>Flow</u> <u>lb/hr</u>	<u>Density</u> <u>lb/ft³</u>	<u>Spec</u> <u>Heat</u> <u>BTU/</u> <u>lb °F</u>	<u>Viscosity</u> <u>lb/hr ft</u>	<u>Ther</u> <u>Cnd</u> <u>BTU/</u> <u>hr ft °F</u>	<u>Prandtl</u> <u>Number</u>
**	4400.00	8.159	336.60	0.0361	0.260	0.0752	0.0270	0.726
***	4800.77	18.215	350.26	0.0162	0.311	0.1176	0.0463	0.777

	<u>Effec-</u> <u>tiveness</u>	<u>Ht</u> <u>Trans</u> <u>BTU/hr</u>	<u>U</u>	<u>NTU</u>	<u>Renlds</u> <u>Number</u>	<u>Stanton</u> <u>Number</u>	<u>j</u>	<u>f</u>
**	0.4657	94377.2	2.9025	1.344	441.47	0.011486	0.009275	0.215834
***	0.5796	-117470.2	2.9555	1.673	240.45	0.023575	0.019921	0.285148

U = Unit overall thermal conductance - BTU/hr ft² F

j = Colburn heat transfer coefficient

f = Fanning friction factor

Cmin/Cmax

0.80490

Log Mean Temp Difference

801.95630

** = Preheat

*** = Exhaust

Economics

Table 3.14 shows the cost breakdown for the system. The economic calculations only utilize the data as presented by an enthalpy balance, i.e., the calculated fuel savings is directly related to the amount of energy recovered, not the observed fuel savings. It should be pointed out that the payback period of 2.41 years is somewhat less if the increased system efficiency is incorporated into the calculations.

Problem Areas

- Plugging of the matrix by condensation of the vaporized glass proved to be a serious problem. One method of avoiding this problem would require a two-stage heat exchanger with a removable collector or transition zone between them. The exhaust would enter the first recuperator, be cooled and leave at a temperature above 1830 °F (condensation temperature). The gases would then enter the transition zone where they would deposit on the removable collectors. After this zone, the gases enter the second stage recuperator to continue the heat transfer process.
- Other methods to reduce plugging are 1) reduce the velocity of the impinging flame, thereby reducing the batch carryover, 2) increase the hydraulic diameter of the individual exhaust passages.
- The corrosive nature of the borosilicate glass makes difficult the prediction of actual modes of failure. Plugging was evident; however, previous chemical attack may have caused cracking. The leakage that resulted could have caused localized cooling followed by plugging in the cooled exhaust stream.
- The inability of the furnace to maintain a positive pressure calls for the utilization of exhaust fans to promote flow of exhaust gases through the cube.

Conclusions

- The recuperator material Z-1000 was difficult to fabricate, which reduced its potential as a candidate material. Not enough operating time was logged to conclusively state that a matrix made of Z-1000 would withstand attack from borosilicate glass.
- The borosilicate glass attack on MAS 8400 is reduced with the application of a protective coating of Al_2O_3 .
- Plugging of the matrix, by whatever mechanism, is a problem that must be minimized in the future.
- Fuel savings were in the 40 to 60% range.

TABLE 3.14GLASS TANK DEMONSTRATION ECONOMICS

Average Heat Transferred to Preheat Air for Both Recuperators =

94377 x 2 BTU/hour

Nominal Operating Time = 120 hours/week

Recovered Energy = 22,650,480 MM BTU/week

Annual Savings¹ = \$3,442.87/year

Ceramic Recuperator	4,000
Piping	300
Controls	500
Burners	1,400
Insulation	100
Installation	1,000
Engineering	<u>1,000</u>

TOTAL SYSTEM COSTS = \$8,300

Payback Period = 2.41 years

¹Based on 50 weeks/year operation with gas of 1,000 BTU/ft³ and costs of \$3.04/MM BTU

- Flat flame burners functioned satisfactorily with preheated air and saved considerable fuel during the 180 hours of operation.
- The payback period of this system may be as little as 2.41 years if a suitable system to reduce fouling can be developed.

3.4 Pilot Plant U.S. Smelting Furnace Demonstration

Description

The pilot plant smelting furnace is a small rotary kiln type furnace and is used for the treatment of various ceramic powders. It is fed in a batch-type operation, and because it is used primarily for experimental purposes, it is not operated on a continuous basis. A schematic of the furnace is depicted in Figure 3.7.

The material to be calcined is loaded into a 2 foot diameter, 3 foot long refractory-lined kiln. A 1-4 MM BTU/hour burner using premixed natural gas and air fires directly into the kiln. Secondary air enters through the open front end of the kiln. The furnace exhausts into a plenum chamber to allow for some settling of particulate matter. The exhaust gas from the plenum passes directly through a 6 ft³ ceramic recuperator. The air supply to the recuperator is from a separate blower and the preheated air is vented to the stack as the burner was not modified to use the preheated air.

Results and Discussion

The geometry and core properties of the 17 x 17 x 32 inch matrix used in this demonstration are presented in Table 3.15. Note that the air side flow passage (0.123 inch x 0.67 inch) is smaller than the gas side flow passage (0.237 inch x 0.67 inch). The operating variables for one of many tests performed during the demonstration are listed in Table 3.16 while the results of the heat transfer calculations are presented in Table 3.17.

As in the case of the other demonstrations, the selected matrix was considerably oversized. This is evident from the low values of the Reynolds Number and the low pressure drop on the air side (0.96 inch H₂O). The exhaust gas outlet temperature was measured by a thermocouple in the exhaust pipe leading to the stack. The natural gas flow and the preheat air flow to the recuperator were measured. The exhaust gas flow was based on the excess air burner setting. The flow of preheated air out of the recuperator was not measured; therefore, actual leakage rates were not determined.

The calculated heat transfer from the gas side was considerably less than the heat transferred to the air in the recuperator. This may be due either to air leaking to the gas side before entering the core or to inaccurate temperature measurements. This question cannot be resolved at

Figure 3.7

SCHEMATIC OF THE PILOT PLANT
U.S. SMELTING FURNACE DEMONSTRATION #4

TABLE 3.15
CORE GEOMETRY FOR
THE PILOT PLANT U.S. SMELTING FURNACE DEMONSTRATION

Crossflow Recuperator - 17 x 17 x 32 inch Core

Preheat Window = $15.1 \times 29.8 = 449.98$ (sq in)

Length of Air Passage = 17.0 inch

Exhaust Window = $15.1 \times 29.8 = 449.98$ (sq in)

Length of Gas Passage = 17.1 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.050	0.123	0.600	1533.30	0.045	4.88
***	0.050	0.237	0.600	1538.39	0.045	2.53

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	113.157	0.204	37691.551	15.740	0.960	83.27
***	218.759	0.340	44037.098	14.766	0.110	50.33

* = Aspect Ratio = Distance between ribs/rib height

** = Preheat

*** = Exhaust

TABLE 3.16

PILOT PLANT U.S. SMELTING FURNACE DEMONSTRATION
OPERATING VARIABLES ON A SINGLE TEST

(17 x 17 x 32 inch Core)

Natural Gas Consumption	1,000 SCFH
Exhaust Gas Flow	13,015 SCFH
Preheat Air Flow	12,000 SCFH
Preheat Air Inlet Temperature	100 °F
Preheat Air Outlet Temperature	1240 °F
Furnace Temperature	1700 °F
Exhaust Gas Inlet Temperature	1700 °F
Exhaust Gas Outlet Temperature	1025 °F
Air Side Effectiveness	71%
Time of Operation	10 hours

TABLE 3.17

PILOT PLANT U.S. SMELTING FURNACE DEMONSTRATION
PERFORMANCE CALCULATIONS

	<u>Temp In</u> <u>Deg F.</u>	<u>Temp Out</u> <u>Deg F.</u>	<u>Flow</u> <u>SCFH</u>	<u>Flow</u> <u>ACFH</u>	<u>Capacity</u> <u>Rate</u> <u>BTU/hr °F</u>			
**	100.0	1240.0	12000.0	12923.1	240.0943			
***	1700.0	1025.0	13014.6	54060.8	282.6682			

	<u>Flow</u> <u>SCFH</u>	<u>Velocity</u> <u>ft/sec</u>	<u>Mass</u> <u>Flow</u> <u>lb/hr</u>	<u>Density</u> <u>lb/ft³</u>	<u>Spec</u> <u>Heat</u> <u>BTU/</u> <u>lb °F</u>	<u>Viscosity</u> <u>lb/hr ft</u>	<u>Ther</u> <u>Cnd</u> <u>BTU/</u> <u>hr ft °F</u>	<u>Prandtl</u> <u>Number</u>
**	12000.00	4.568	918.00	0.0351	0.262	0.0767	0.0276	0.728
***	13014.65	9.885	951.70	0.0208	0.297	0.0996	0.0381	0.763

	<u>Effec-</u> <u>tiveness</u>	<u>Ht</u> <u>Trans</u> <u>BTU/hr</u>	<u>U</u>	<u>NTU</u>	<u>Reynlds</u> <u>Number</u>	<u>Stanton</u> <u>Number</u>	<u>j</u>	<u>f</u>
**	0.7125	273707.5	1.1564	1.261	259.19	0.005312	0.004298	0.323322
***	0.4967	-190801.0	0.6900	0.879	178.18	0.008790	0.007342	0.126338

U = Unit overall thermal conductance - BTU/hr ft² F

j = Colburn heat transfer coefficient

f = Fanning friction factor

Cmin/Cmax

0.84939

Log Mean Temp Difference

904.28052

** = Preheat

*** = Exhaust

this time because not enough of the operating variables were actually monitored. The calculated air side effectiveness of 71% appears to be high because of the above two possible reasons.

The data presented in Table 3.16 were obtained on a test of 10 hours duration. Other tests at different conditions were performed with the time of operation on each test varying from 10 hours to 25 hours. The recuperator is still installed on the furnace but is now used occasionally for experimental purposes.

Economics

Because this demonstration does not represent a typical installation, a payback period analysis was not performed.

Problem Areas

Besides the question of accuracy of the temperature measurements and the lack of flow and leakage rate data, there were no problems encountered during this demonstration. The recuperator is still installed on the furnace but the furnace is sparingly used at the present time.

Conclusions

Because of the experimental nature of the pilot plant smelting furnace operation, its varying operating conditions and the limited time of operation, no specific conclusions can be drawn from this demonstration.

3.5 Rotary Calciner Furnace Demonstration

The furnace chosen for the fifth demonstration was a rotary calciner furnace depicted schematically in Figure 3.8. The material to be calcined is continuously fed to the inside of a slowly rotating tube which is surrounded by a furnace. The speed of rotation of the tube and its tilt with the horizontal causes the material to gradually flow down the tube and finally out the other end. The furnace is multi-zoned and temperatures in the zones vary from 1200 to 1400 °F depending on the material being calcined. The furnace is maintained at the desired temperatures by six natural gas burners located at the bottom of one side of the furnace. The burners have an independent air supply and are automatically controlled. The calcining operation requires process air to be blown into the rotating tube at the feed end. This process air flow is continuously varied depending on the feed material and the rates of reaction of the material in the tube.

The ceramic recuperator is located on the top of the furnace where part of the furnace exhaust passes before entering the exhaust hood. Preheat air is supplied to the recuperator by a separate blower and the preheated air

Figure 3.8

SCHEMATIC of the ROTARY CALCINER

FURNACE DEMONSTRATION No. 5

from the recuperator is directed to the rotating tube and is used for the process. Thus, the air flowing through the recuperator is continuously varied depending on process conditions.

Results and Discussion

The ceramic matrix used in this demonstration was 17 x 17 x 32 inch and was identical to the one used in the Pilot Plant U.S. Smelting Furnace Demonstration. The core properties are presented in Table 3.15. Since the operating variables were continuously varied, data for a range of operating conditions is presented in Table 3.18. The exhaust gas flow through the recuperator was not measured but was only estimated. Heat transfer calculations were not performed as a specific set of data was not available. The range of air side effectiveness values are estimated.

On March 15, 1979, after about 2,000 hours of operation, the recuperator was mechanically damaged. An identical replacement on the calciner has logged about 1,800 hours. The initial core was inspected and no signs of corrosion or deterioration were found.

Economics

The energy savings were estimated at 213,000 BTU/hour, (an average value of the heat transferred to the preheat air). The economics of a typical system, based on the estimated savings, results in a payback period of 4.13 years and is presented in Table 3.19.

Problem Areas

No problems with the recuperator were encountered during this demonstration. The preheated air was used as process air and was not directed to the burners. This latter mode of operation would require the air to be at a higher pressure and would therefore increase leakage, which was not monitored during this demonstration.

Conclusion

- The durability of the ceramic material has been proven at furnace temperatures of 1400 °F.
- Higher energy savings could be attained if preheated air were used as combustion air. This would result in a shorter payback period.
- This demonstration subjected the core to rather severe conditions particularly when the process air flow was shut off and then started again. No damage to the core was observed.

TABLE 3.18

ROTARY CALCINER FURNACE DEMONSTRATION
OPERATING VARIABLES

Exhaust Gas Flow (estimated)	7,000 - 12,000	SCFH
Preheat Air Flow	0 - 11,220	SCFH
Preheat Air Inlet Temperature	104 °F	
Preheat Air Outlet Temperature	800 - 1120 °F	
Furnace Operating Temperature	1200 - 1400 °F	
Exhaust Gas Inlet Temperature	1200 - 1400 °F	
Air Side Effectiveness (%)	65 - 78%	
Time of Operation	3,800 hours	

TABLE 3.19

ROTARY CALCINER FURNACE DEMONSTRATION
SYSTEM ECONOMICS

Energy Transferred to Preheated Air = 213,000 BTU/hour

Nominal Operating Time = 80 hours/week

Recovered Energy = 17.04 MM BTU/week

Annual Savings¹ = \$2,590/year

Ceramic Recuperator	4,000
Piping	300
Controls	1,000
Burners	2,400
Insulation	1,000
Installation	1,000
Engineering	<u>1,000</u>
TOTAL SYSTEM COSTS	= \$10,700

Payback Period = 4.13 years

¹Based on 50 weeks/year operation with gas of 1,000 BTU/ft³ and costs of \$3.04/MM BTU

4. INDUSTRIAL APPLICATIONS

The ceramic recuperator has the potential to save energy in a number of different manufacturing processes. Some industries with high temperature processes, such as the steel industry, have historically utilized recuperation or other waste heat recovery techniques to save energy. With the cost and availability of fuels becoming an ever increasing problem, there is more incentive for other industries to improve the efficiency of their heat processing operations.

Identification of the industries where recuperators can be used and the annual potential energy savings by recuperation of furnace exhausts is necessary before the impact of this technology on energy conservation can be assessed. A comprehensive analysis to assess the impact of this technology on energy conservation is considered outside the scope of this study. The purpose of this chapter is to develop the methodology and show how an impact analysis may be performed.

The basic approach used in the determination of the potential energy savings by recuperation was the following:

- Determine the energy profiles of at least the four-digit SIC level.
- Analyze the specific process used in each four-digit industry.
- Establish the fuel usage in those unit processes where recuperation can be applied.
- Estimate potential savings by recuperation for the four-digit industry.

The sources of information used in this analysis were the following:

- 1977 Census of Manufacturers
- Annual Survey of Manufacturers, 1976
- Industrial Applications Study, Drexel University, 1978
- Analysis of the Potential of Solar Thermal Energy to Provide Industrial Process Heat, Intertechnology Corporation, 1977
- Energy Analysis of 108 Industrial Processes, Drexel University, 1979.

The potential energy savings figures for each four-digit industry obtained by the above procedure represent a realistic goal that can be achieved by installing recuperators on those exhaust streams that are now exhausted into the atmosphere. The final decision whether or not to install a heat recovery system will be based primarily on economic factors.

Industrial Energy Profiles

The energy consumption in the manufacturing industries is concentrated in a small number of industries. The 1976 consumption of purchased fuels and electricity by all the manufacturing industries was 12.625 quads. (This figure does not include fuels produced and consumed on site such as blast furnace gas, fuels for feedstocks or fuels converted to other types of fuels.) The top ten two-digit SICs listed in Table 4.1 account for 91% of the above consumption. The consumption figures presented in Table 4.1 include energy use for process steam and hot water, furnaces and mechanical energy needs. The energy requirements for furnaces operating at all temperatures amounts to 3.721 quads. Of this amount, 2.746 quads or 74% is rejected through the furnace stack. While only a part of rejected energy can be recovered, the potential for saving energy from furnace exhausts is indeed significant. Figure 4.1 shows the furnace energy requirements and furnace stack losses for the various two-digit SICs. The petroleum (SIC29), primary metals (SIC33), stone clay and glass (SIC32), paper (SIC26), chemicals (SIC28) and food (SIC20) use significant quantities of energy in furnace operations. The average furnace stack temperature is about 650 °F. The food and paper industries are usually low temperature operations compared with primary metals and stone, clay and glass industries; therefore, the latter offer considerably greater potential for saving energy.

Potential Savings by Recuperation

The first step in determining the potential energy savings is to study the specific four-digit industry's process to determine where recuperation could be applied. Since one of the objectives of this program was to develop a high temperature recuperator, only those processes with stack gases above 1,400 °F were considered in this analysis. The rationale was that the higher the stack gas temperature, the greater is the amount of energy that can be saved by recuperation and the lower the payback period.

However, it is emphasized that there are many unit processes in industry where stack gas temperatures range from 1,000 °F to 1,400 °F. Further, industrial processes where recuperation or other forms of waste heat utilization are employed, were excluded from the potential savings figures.

A few of the four and six-digit industries selected for this analysis and the results analysis are presented in Table 4.2. Listed in the table are the fuel used in the specific unit process, the stack gas temperature and the potential savings by recuperation. The potential savings figures were estimated from Table A-1 and represent the maximum possible savings. The following example, primary copper (SIC3331), illustrates the methodology used.

TABLE 4.1
INDUSTRIAL CONSUMPTION OF FUELS
AND PURCHASED ELECTRICITY

<u>SIC</u>	<u>Description</u>	<u>10¹⁵ BTU</u>
28	Chemicals, allied products	3.017
33	Primary metals	2.380
26	Paper and allied products	1.295
29	Petroleum and coal products	1.292
32	Stone, clay and glass products	1.220
20	Food and kindred industries	0.937
34	Fabricated metal products	0.381
37	Transportation equipment	0.380
35	Machinery, except electric	0.330
22	Textile mill products	0.329

FIGURE 4.1

1976 TOTAL FUEL & PURCHASED ELECTRICITY ————— $12,625 \times 10^{12}$ BTU/YR.
 □ TOTAL FURNACE ENERGY REQUIREMENTS ————— $3,721 \times 10^{12}$ BTU/YR.
 ■ TOTAL ENERGY REJECTED THROUGH FURNACE STACK — $2,746 \times 10^{12}$ BTU/YR.

TABLE 4.2
POTENTIAL SAVINGS BY RECUPERATION
FOR SELECTED FOUR-DIGIT SICs

SIC No.	SIC Description	1977 Production	Fuel and Electricity Purchased for Heat and Power 10 ¹² BTU	Fuel Used in Process Where a Recuperator Can Be Used 10 ¹² BTU	Stack Temp. °F	Potential Savings by Use of a Recuperator 10 ¹² BTU
2813 28127-20	Inorganic Gases H ₂ -Steam Reforming	872 x 10 ⁶ lbs	116.0	Furnace 45.0	1,550	8.0
2816 28161-11	Inorganic Pigments Titanium Pigments (Chloride and Sulfate Processes)	0.743 x 10 ⁶ short tons	38.0	Heater and Kin 8.0	1,650	1.5
28162-21	Zinc Oxide Pigments (French and Electrothermal Process)	0.2 x 10 ⁵ short tons	4.0	Sintering Settling Comb.	2,900) 1,500)	1.0
2873 28731-31	Nitrogenous Fertilizer Anhydrous Ammonia	17.8 x 10 ⁶ short tons	293.0	Primary Reformer 42.0	1,500	7.0
3211 3211- 32112-13) -15) -17)	Flat Glass Sheet Plate and Float Glass (Various Thicknesses)	6.84 x 10 ⁹ lbs	40.0	Melting 31.0 Refining	2,800) 2,000)	10.0
3321	Gray Iron Foundries	29.0 x 10 ⁹ lbs	243.0	Cupole Furnace 13.0	2,900	44.0
3331 33311-11	Primary Smelting and Refining of Copper Blister and Anode Copper	3.37 x 10 ⁶ short tons	116.0	(Converter Smelter Gas 2.3 x 10 ¹²) Anode Furnace (11.8 x 10 ¹²)	2,200) 1,400)	4.2
33312-13	Cathode, wire, bars, ingots and shot			Melting (10.0 x 10 ¹²)	1,400)	
3341 33412-11 33412-31	Secondary Smelting and Refining of Non-ferrous Metals Copper	0.512 x 10 ⁶ short tons	2.0	Reverberatory Furnace 0.424	1,400	0.06
3462	Iron and Steel Forgings	1.76 x 10 ⁶ short tons	74.0	Heating 15.8 Heat Treating 23.2 49.0	2,400 1,600	12.5
3523	Farm Machinery and Equipment	N/A	35.0	Heat Treating 9.0	2,000	3.0
3531	Construction Machinery	N/A	40.0	Heat Treating 9.0	2,000	3.0

EXAMPLE OF SIC3331 (PRIMARY COPPER)

Process Description

The primary product from this SIC is copper. The total production of copper in 1977 was 3.37×10^6 short tons. By-products produced in primary copper production include sulfuric acid, selenium, tellurium, gold and silver.

A process flow diagram for smelting and refining primary copper is shown in Figure 4.2. Low grade sulfide ores require an initial step of concentration. Roasting is done to dry the ore and to regulate the sulfur content prior to smelting. No fuel is required for roasting if the sulfur content is 25% or more. In some cases, the sulfur content in the ore is sufficiently controlled in the concentrator to eliminate roasting prior to smelting in a reverberatory furnace.

The roasted or unroasted ore is radiantly heated in the reverberatory furnace, at 2,200 °F by burning natural gas, oil or coal. Approximately 23.5 MM BTU/ton of product are required. The molten copper sulfide (Cu_2S) known as matte collects at the bottom while the slag floats on the top. The flue gases leave the furnace at about 2,200 °F with an energy content of about 20.5 MM BTU per ton of product. The energy in this waste stream is usually recovered in a waste heat boiler and therefore a recuperator would not be applicable to this waste stream.

The matte is transferred to a refractory lined converter at 2,200 °F and air is blown through the molten material to oxidize the sulfide to sulfur dioxide. The smelter gas from the converter contains about 0.68 MM BTU per ton of product and represents a stream from which some energy may be recovered by recuperation. No fuel is used in the converter. The product from the converter, known as blister copper, is further refined in an anode furnace operating at 2,200 °F. The refining process employs oxidation, fluxing and reduction and the impurities form a slag. The molten metal is agitated by compressed air and fuel in the amount of 3.5 MM BTU ton of product is consumed. The stack gas leaves the furnace at 1,400 °F with an energy content of about 3.3 MM BTU per ton of product. The high grade copper is now cast into anodes for further electrolytic refining.

In the electrolytic step, copper is dissolved from the anode into a solution of copper sulfate and sulfuric acid and deposited on a cathode in a highly purified form. The impurities such as gold, silver, etc. fall to the bottom of the electrolytic tank to form anode slime or mud. About 2.1 MM BTU of electricity per ton of product is used in this process which operates at about 140 °F. The highly purified copper cathode is then melted in a furnace and cast into commercial shapes. The fuel consumption in the melting process is about 3.0 MM BTU per ton of product and the stack gases containing about 2.26 MM BTU per ton of product leave the furnace at 1,400. These stack gases may be recuperated to further recover energy.

FIGURE 4.2
3331-PRIMARY SMELTING AND REFINING OF COPPER
PROCESS FLOW DIAGRAM *

* ENERGY CONTENT OF EACH STREAM IS PRESENTED IN BTU/TON OF PRODUCT

Potential Energy Savings

Energy savings by recuperation may be obtained at three points in the overall process. The potential savings are presented in Table 4.3 and are computed on the assumption the fuel used is natural gas burned with 10% excess air. The percent savings by recuperation are approximated from Table A-1.

Recuperation of the smelter gas (2,200 °F) by the GTE ceramic recuperator could result in preheat air temperature of 1,200 °F and a savings of about 38% of the energy in the waste stream. Recuperation of the stack gas (1,400 °F) from the anode and melting furnaces could result in preheat air at 600 °F and a savings of about 15% of the fuel usage. The maximum potential savings of energy by recuperation in the primary copper industry (SIC3331) amounts to about 4.2 trillion BTU per year.

TABLE 4.3POTENTIAL ANNUAL ENERGY SAVINGS IN SIC3331

<u>Unit Process</u>	<u>Fuel Used BTU/Year</u>	<u>Energy Constant of Waste Gases BTU/Year</u>	<u>Gas Temperature °F</u>	<u>Potential Savings Recuperation BTU/Year</u>
Converter		2.3×10^{12}	2,200	0.9×10^{12}
Anode Furnace	11.8×10^{12}	11.1×10^{12}	1,400	1.8×10^{12}
Melting Furnace	10.0×10^{12}	7.6×10^{12}	1,400	<u>1.5×10^{12}</u>
				4.2×10^{12}

5. CONCLUSIONS AND RECOMMENDATIONS

The following conclusions can be drawn from the ceramic recuperator demonstration program:

- The energy saved from waste gas heat recovery was equivalent to 20 to 50% of the fuel that was previously used.
- The more efficient method of achieving energy savings is to preheat the combustion air to the burners of the furnace rather than to preheat air for process needs. The latter use does not maximize savings from recuperation.
- The concept of using a ceramic material for a compact (large heat transfer surface area per unit volume) heat exchanger is technically feasible in many exhaust gas heat recovery applications.
- The cordierite ceramic material used in these demonstrations was durable in most applications at furnace temperatures up to 2600 °F.
- Certain corrosive furnace atmospheres can affect the cordierite material. For example, borosilicate glass rapidly corroded the cordierite. Coating the cordierite with alumina proved to be helpful but durability of this recuperator was not proven because the tests were of short duration.
- Developments in bonding and fabrication techniques have resulted in a ceramic core that can withstand cycling temperatures and extreme conditions caused by emergency furnace shut down. These tests were performed on a specially designed test furnace.
- Excessive particulate matter in the furnace exhaust streams could lead to plugging of the matrix if air lance systems are not incorporated. An air lance system used in one of the demonstrations proved effective on short duration tests. On another demonstration, a plenum chamber to allow settling of the particulate matter proved to be successful.
- ~~While the demonstrations were performed on furnaces burning clean fuels, i.e. natural gas, independent tests performed by Oak Ridge National Laboratory using fuel oils containing vanadium and sulfur did not show any deleterious effects on the cordierite ceramic matrix.~~
- The ceramic recuperator can be designed for service on new furnace installations and is flexible enough to be installed in many retrofit applications while meeting the exhaust gas side and air blower pressure drop constraints.
- The basic ceramic core design is flexible enough to allow for optimization of the heat transfer and pressure drop performance of a recuperator.
- The heat exchanger effectiveness was found to range from 40 to 60%.

- Excessive leakage of air into the gas stream through cracks in the core and through seals between the matrix and housing can reduce the heat exchanger effectiveness. Leakage rates of 10% of the air flow were acceptable. Cores presently being fabricated have had leakage rates of about 5% during tests performed in a specially designed test furnace.
- A spring-loaded housing design was especially effective in reducing leakage through the seals. While leakage does reduce effectiveness, a proper system design should compensate for leakage with appropriately selected burner and blower systems.
- The commercial burner and control systems available proved adequate for use with preheated combustion air up to a preheat temperature of 1,200 °F.
- The small size and weight of the ceramic recuperator allowed ease of installation without requiring structural modifications to the furnace.
- With the exception of the problem with borosilicate glass and high exhaust gas particulate loadings, the demonstration program indicated that little or no maintenance was required with the ceramic recuperator installations.
- The demonstration program indicated that the GTE ceramic recuperator is economically feasible. The ceramic recuperator is low in capital cost as compared to metallic recuperators for the same heat transfer performance. Costs of a complete system, including modified burners, controls and installation, amount to about five times the cost of the ceramic recuperator itself (Table 2.7). Conservative estimates of payback periods are about two years.

As a result of the development and demonstration program, the following recommendations are made:

- The demonstration program be continued at an accelerated level to aid the transfer of this technology to industry, thus ensuring increased industrial energy conservation.
- An application analysis be conducted to determine those energy intensive industries where the ceramic recuperator could be successfully used to maximize the impact on energy conservation. These selected industries should be targeted for the accelerated demonstration program.
- The ceramic recuperator design for each application should be optimized for performance and economics.
- The selected furnaces should include both those where the primary mode of heat transfer is by convection and those where the primary mode of heat transfer is by radiation.

- Before each demonstration is undertaken, a detailed testing and monitoring plan should be prepared, approved and implemented. Accurate measurements of all pertinent parameters are essential to obtain a complete analysis of each demonstration.

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7. APPENDICES

APPENDIX A

A. ESTIMATION OF FUEL SAVINGS BY RECUPERATION OF FURNACE EXHAUST TO PREHEAT COMBUSTION AIR

A.1 General Considerations

The recovery of waste energy in furnace exhaust gases is gaining in importance as fuel costs continue to escalate. Installation of a recuperator in the furnace exhaust stream to preheat the combustion air can result in considerable savings of fuel. These savings are primarily the result of the sensible heat increase of the combustion air and, to some extent, improved combustion efficiency. The amount of fuel saved will depend on the exhaust gas temperature, amount of excess air used, the type of burner and the furnace control system. These fuel savings can be accurately measured by metering the energy consumption per unit of production before and after installation of the recuperator. In designing a waste heat recuperation system, the fuel saved by the system must be estimated. Standard industrial practice refers to the method described in the North American Combustion Handbook with curves and tables that directly predict the percentage fuel savings. This chapter analyzes the standard estimation technique and suggests a more conservative approach for calculation of percent fuel savings. Mass and enthalpy balances are provided for both methods and a typical furnace recuperation example is detailed to illustrate the differences in the two methods of calculating the percent energy saved.

Most industrial processes for heat treating and reheating of materials employ furnaces that are fueled with natural gas or oil. In such applications, the furnaces are required to heat the materials uniformly and furnaces are designed to ensure a uniform temperature distribution. One method of achieving temperature uniformity is to use large amounts of excess air. The stoichiometric air fuel ratio for natural gas is approximately 10:1. In some steel industrial furnaces operating from 700 °F to 2,200 °F, excess air values are 600% at the low temperature and approximately 50% at the higher temperature. Thus, temperature uniformity achieved with large excess air values requires increased fuel. Therefore, one way to obtain energy savings in these applications is to substantially reduce the quantities of excess air while still maintaining temperature uniformity through appropriate furnace design.

Another method of obtaining energy savings is to use recuperators to preheat the combustion air to the furnace. The fuel saved primarily results from the increase in sensible heat of the combustion air. An added advantage is the improved efficiency of the combustion process. Recovery of waste heat normally exhausted into the atmosphere is now considered to be one of the most viable methods of energy conservation. The price and supply of all forms of energy demands that processes be run as efficiently as possible, and preheating combustion air by waste gas recuperation can save up to 50% of the fuel previously used. The amount of fuel saved depends on control systems, amount of excess air, type of burner and temperature of the furnace exhaust. The higher the temperature of the

furnace, the greater the amount of energy that is lost in exhaust to the atmosphere. Furnaces that run at temperatures of 1,800 °F or above lose about 50% of their energy in flue gas exhaust. The energy input, output and losses are illustrated by the Sankey diagram in Figure A.1.

Experience has shown that furnace operators do not follow set furnace operating specifications, and fuel and excess air vary considerably depending on each individual application and the particular furnace load. Furnace operators are more concerned with meeting production targets and will do so by adjusting fuel and air inputs to the burner so as to obtain the best operating conditions in the furnace irrespective of fuel economy. The ideal system for determining the energy savings resulting from recuperation of furnace exhausts is to meter the fuel input to the furnace before and after the use of the recuperator. In many industrial plants, however, fuel usage in individual furnaces may not have been monitored on a continuous basis. Therefore, base line energy consumption per unit of time or per unit of production, may not have been established prior to installation of the recuperator. If no prior fuel usage data is available, the fuel consumption per unit of time must be calculated for given furnace operating conditions. When a recuperator is designed and sized for the given application the fuel consumption may likewise be calculated so as to obtain the energy savings when a recuperator is used. The remainder of this section deals with the standard technique for estimating fuel savings by recuperation and suggests a more conservative approach in calculating energy savings.

A.2 Mass and Energy Balances

Before detailing the methods used to calculate energy savings, it is desirable to review the basic mass and enthalpy balances around the furnace and recuperator. For the basic calculations dealing with the combustion of a fuel with air, reference may be made to any standard text on thermodynamics. The example discussed in this report uses methane as the fuel for simplicity of calculations. The stoichiometric air/fuel ratio for methane is 9.52 ft³ air/ft³ methane or 17.27 pounds air/pound methane. The lower heating value of methane is 910 BTU/ft³ or 21,500 BTU/pound and the adiabatic flame temperature is about 4,025 °F. For the case of 10% excess air used in the subsequent example, the air fuel ratio is 10.47 ft³ air/ft³ methane or 18.97 pounds air/pound methane and the resulting flame temperature, without losses, is 3,350 °F.

Before Installation of a Recuperator

In a typical furnace operating without a recuperator, the combustion products enter the furnace, transfer energy to the material in the furnace with some losses through the furnace walls and openings, exit the furnace at approximately the furnace temperature, and are exhausted up the stack. A system schematic with and without a recuperator is depicted in Figure A.2. A mass balance on the unrecuperated system is as follows:

Figure A.1
Sankey Diagram

REF. #1 PG. 57

a. No Recuperation

b. With Recuperation

Figure A.2**FURNACE SYSTEM SCHEMATICS** **A). SYSTEM SCHEMATIC WITHOUT RECUPERATION** **B). SYSTEM SCHEMATIC WITH RECUPERATION**

$$m_1 + m_2 = m_3 = m_4 \quad A-1$$

Where

- m_1 = mass of methane entering burner (point 1)
- m_2 = mass of air entering burner (point 2)
- m_3 = mass of products entering furnace (point 3)
- m_4 = mass of exhaust gas exiting the furnace (point 4)

The equivalent energy balance equation for the system is:

$$m_1 h_1 + m_1 \Delta H_R + m_2 h_2 = m_3 h_3 = m_4 h_4 - Q_L \quad A-2$$

Where

- m = mass rate of flow (pound/hour)
- h = enthalpy (BTU per pound)
- ΔH_R = heat of combustion of fuel
- Q_L = heat transferred to the system. (Note that in this instance, heat is transferred from the furnace to the load and surroundings. This represents heat transferred from the system. By convention the numerical value of Q_L inserted into Equation A-2 is negative. Refer to Appendix E for details of the calculations.)

After Installation of a Recuperator

Now consider a recuperator installed in the exhaust gas stream (Figure A.2) that preheats the combustion air. A mass balance for the overall system give the following equations:

$$m_2 = 18.97 m_1 \text{ (with 10\% excess air)} \quad A-3$$

$$m_1 + m_2 = m_3 = m_4 = m_5 \quad A-4$$

$$m_2 = m_6 \quad A-5$$

An energy balance give the following three equations:

$$m_1 h_1 + m_1 \Delta H_R + m_2 h_2 = m_3 h_3 = 19.97 m_1 h_3 \quad A-6$$

$$Q_L + m_3 h_3 = m_4 h_4 \text{ or } Q_L = 19.97 m_1 (h_4 - h_3) \quad A-7$$

$$m_2 h_2 - m_6 h_6 = m_4 (h_4 - h_5) = 19.97 m_1 (h_4 - h_5) \quad A-8$$

Where m = mass rate of flow (pounds/hour)

h = stream specific enthalpy (BTU/pound)

ΔH_R = lower heating value₃ of the fuel (for methane
 $\Delta H_R = 910 \text{ BTU/ft}^3 = 21,500 \text{ BTU/pound}$)

Q_L = Heat transferred to the system. Because in this case, the furnace load and losses represents heat transferred from the system, values of Q_L substituted into Equation A-7 should be negative by convention (see Appendix E).

A.3 Standard Method For Calculating Fuel Savings

Method 1 - (Detailed calculations are presented in Appendix E)

If the methane input to the burner (m_1) and the stack gas temperature (t_4) are given for a specific application, the total enthalpy input to the furnace ($m_3 h_3$), the furnace load and losses (Q_L), and enthalpy lost through the stack ($m_4 h_4$) can be computed by using Equations A-1 and A-2. The temperature of the combustion products (t_3) can be calculated from Equation A-2 by assuming a value of t_3 and balancing Equation (A-2) in an iterative process. The iterative procedure may be avoided by referring to a standard combustion text for a plot of the theoretical flame temperature versus excess air for the specific fuel. For the purposes of this report Figure A.3 can be used if the combustion product enthalpy (h_3) in BTU/pound is calculated from Equation A-2.

When a recuperator is installed in the furnace exhaust stream to preheat the combustion air (see Figure A.2), and the exhaust gas inlet (t_4) and outlet (t_5) and air inlet (t_6) and outlet (t_2) are known, the following questions need to be answered:

- What are the mass and energy balance relationships?
- How much energy is now lost in the stack gases?
- How much fuel is saved?

These questions may be answered providing the following assumptions are made:

Assumption 1: The furnace load and losses are the same as before, i.e., without the recuperator (or Q_L is kept constant).

Assumption 2: The exhaust gas temperature leaving the furnace is the same as before, i.e., without the recuperator (or t_4 is kept constant).

Assumption 3: Burner excess air is the same.

The first assumption is logical because in any industrial application the furnace load must be maintained when a recuperator is used. The second assumption is logical because the furnace operating temperature must be the same even if a recuperator is in use. The third assumption provides for the same stoichiometry. Calculations to determine the methane usage with a recuperator and hence the fuel savings, may be performed using Equations A-3 through A-8. First the enthalpy of the combustion products (h_3) is determined by using Equation A6 modified as below (for 10% excess air):

$$m_1 h_1 + m_1 \Delta H_R + 18.97 m_1 h_2 = 19.97 m_1 h_3$$

With h_3 known, the combustion product temperature (t_3) which is higher with preheated air, can be determined from Figure A.3. Equation A-7 is now used to determine the methane flow rate (m_1) because of the assumptions that Q_L and t_4 were the same as before the recuperator was used. The percent fuel saved is then calculated from the difference in methane usage with and without recuperation.

The percent fuel saved with recuperation can be determined by following the graphs and tables presented in the "North American Manufacturing Company Handbook"⁽⁹⁾. The percent fuel saved is defined in terms of "available heat" which is the gross heat released within the combustion chamber less the stack losses. The fuel savings calculations are presented in tabular form similar to Table A.1 presented here. This table presents the results of energy balances based on the same three assumption listed above and may be used to directly obtain the energy savings with recuperation. Performing the detailed calculations using the above equations, however, gives better insight into a recuperated system.

A.4 Recommended Method For Calculating Fuel Savings

Method 2 - (Detailed calculations are presented in Appendix E)

Recuperator manufacturers have found that the percent fuel saved, as presented in Table A.1, is difficult to achieve in some applications. This is particularly so in cases where the furnace operator has tuned the furnace to operate efficiently and where the furnace design is such that convective heat transfer plays an important role in transferring heat to the furnace load. An efficiently operated furnace is indeed a rare occurrence in practice mainly because the furnace variables are not carefully monitored. Only when a recuperator is installed are the operating parameters adjusted and the furnace fine tuned. This sometimes results in actual fuel savings greater than predicted by Table A.1 partly because the use of preheated air requires less fuel and partly because the furnace operates more efficiently due to the adjustments.

It may be appropriate at this stage to briefly analyze what happens to the system when a recuperator is used. Efficient furnace operation depends on the energy input, the quality of that energy and how the furnace transfers that energy to the load. The use of preheated air to the burner

Figure A.3

Enthalpy for a Given Exhaust Gas Temperature for Methane Combustion
with 10% Excess Air

TABLE A.1
% FUEL SAVINGS AT VARIOUS FURNACE
EXHAUST TEMPERATURES AND AIR PREHEAT

<u>Furnace Exhaust Temp °F</u>	<u>Air Preheat Temperature</u>						
	<u>600</u>	<u>800</u>	<u>1000</u>	<u>1200</u>	<u>1400</u>	<u>1600</u>	<u>1800</u>
2800	37	45	51	56	60	64	67
2600	30	37	43	48	53	56	60
2400	46	32	38	43	47	51	54
2200	23	29	34	39	43	47	50
2000	20	26	31	35	39	43	46
1800	18	24	28	33	37	40	--
1600	17	22	26	30	34	--	--
1400	15	29	24	28	--	--	--
1200	14	19	23	--	--	--	--
1000	13	18	--	--	--	--	--

Fuel - Natural gas at 10% excess air

results in a small improvement in combustion efficiency (especially if oil is used as fuel) a change in the characteristics of the flame front, and a higher temperature of the products of combustion. Combustion products enter the furnace at a higher temperature but at a reduced mass flow rate (as less fuel is combusted with the same air/fuel ratio). The higher temperature increases the heat transferred by radiation while the reduced mass flow rate decreases the heat transferred by convection. The net effect of these processes is difficult to determine unless the specific furnace design is known. Thus, it is difficult to quantitatively predict the change in the furnace efficiency resulting from the use of preheated air.

Since the furnace responds to the total energy input and the furnace design remains the same when a recuperator is installed, it is assumed that the production load will be maintained the same provided the total energy input to the furnace is held constant. This assumption forms the basis of the recommended method of calculating the fuel savings.

Assumption 1: The total furnace energy input is the same as before the recuperator was installed (i.e., $m_3 h_3$ is kept constant).

Assumption 2: The exhaust gas temperature leaving the furnace is the same as before the recuperator was installed (t_4 is kept constant).

Assumption 3: The burner excess air is held constant.

The calculations use the same basic Equations A-1 through A-8 as presented before. The first step is to use Equation A-6 to calculate the specific enthalpy (h_3) at point 3 (refer to Figure A.2). The temperature of the combustion products (t_3) is to be determined from Figure A.3. The total energy input to the furnace ($m_3 h_3$ or H_3), assumed the same as before the recuperator was installed, is now used to determine the mass of methane used (m_1), thus $19.97 m_1 h_3 = H_3$. The savings in methane is then determined and the stack losses and other parameters determined as before using Equations A-1 through A-8. Note that the furnace load and losses are now calculated by using Equation A-7.

A.5 Illustrative Example

The differences in the two methods for estimating energy savings are best illustrated by considering a practical application.

Problem:

A furnace on a galvanizing line is fired by natural gas with 10% excess air. The fuel input to the burner is 3.8 MM BTU/hour. The gases leave the furnace at 1,500 °F and are exhausted out of the stack. Because of restrictions on fuel usage and rising fuel costs, a decision is made to install a recuperator to preheat the combustion air. A quotation from a recuperator manufacturer sizes a unit and estimates the operating characteristics of the unit as follows:

- Exhaust gas temperature entering the recuperator = 1,500 °F.
- Exhaust gas temperature leaving the recuperator = 885 °F.
- Ambient temperature = 77 °F.
- Preheated combustion air temperature = 650 °F.

Based on these estimates, how much fuel is saved by the recuperator?

Solution

For simplicity, methane will be assumed as the fuel. The excess air, 10%, will also be assumed to be constant. The amount of methane currently used is directly obtained from the given burner input. Equations A-1 and A-2 are then used to calculate the furnace input, the energy exhausted out of the stack and the furnace load including losses (Q_L). When the recuperator is installed, Equations A-1 through A-8 are used to calculate the operating parameters by methods 1 and 2. The results of the calculations are summarized in Table A.2. Refer to Appendix E for the detailed calculations using the property values of Appendix C.

In method 1, the furnace load and losses of 2.223 MM BTU/hour have been assumed to be the same and the fuel input to the burner and the enthalpy input to the furnace calculated. Note that the combustion product temperature has increased to 3610 °F and the furnace efficiency increased. The fuel savings amounts to 17.25%. While the calculations are consistent with the first law of thermodynamics, whether or not the furnace production can be maintained with the decreased flow and decreased energy input to the furnace, will depend on the specific furnace design.

In method 2, the energy input to the furnace is kept the same as before recuperation and the increased combustion product temperature ensures that the furnace production will be met. Note the increase in furnace load and losses to 2.424 MM BTU/hour. This small increase may be attributed to the increase in radiation losses due to the increased combustion product temperature. The estimated fuel savings is now 9.6%.

A.6 Recommendation

Method 2 is recommended for estimating fuel savings from recuperation of furnace exhausts. It is a more conservative approach that ensures that the furnace production will be maintained after the recuperator has been installed. The estimation of fuel savings is more realistic, particularly in cases of furnace designs that transfer heat, in large part, by convection.

TABLE A.2
SUMMARY OF ILLUSTRATIVE EXAMPLE RESULTS

Furnace/Recuperator Parameters	Without Recuperator	With Recuperator Installed	
		Standard Technique Method 1	Recommended Technique Method 2
Mass Flow Rate - lb/hr			
Methane	177	146.5	160
Air	3358	2779.0	3030
Exhaust Gas	3535	2925.5	3190
Temperatures °F			
Methane	77	77	77
Air	77	650	650
Combustion Products	3280	3610	3610
Furnace Exhaust	1500	1500	1500
Stack Gas	1500	885	885
Fuel Input to Burner MM BTU/hr	3.800	3.150	3.440
Enthalpy Input to Furnace - MM BTU/hr	4.291	3.935	3.440
Furnace Load & Losses			
MM BTU/hr	2.223	2.223	2.424
% of Furnace Input	51.81	56.60	56.50
Exhaust Gas Enthalpy Entering Recuperator			
MM BTU/hr		1.711	1.866
% of Furnace Input		43.50	43.50
Stack Gas Enthalpy			
MM BTU/hr	2.068	1.126	1.228
% of Furnace Input	48.19	28.61	28.61
Heat Transferred from Exhaust Gas Recuperator - MM BTU/hr		0.585	0.638
Heat Transferred to Combustion Air in Recuperator - MM BTU/hr		0.389	0.424
% Fuel Savings		17.25	9.6

APPENDIX B

RECUPERATOR DESIGN THEORY

The selection of a heat exchanger for a particular application will involve a thermal analysis to estimate the size and performance of the unit as well as an economic analysis. The purpose of this appendix is to provide the reader with a ready reference of the basic equations used in heat exchanger design. The material presented is brief and pertains to convective heat transfer only and specifically to the cross-flow ceramic heat recuperator. For a detailed analysis, reference may be made to Kays and London(3) or to the various textbooks on heat transfer.

B.1 The Overall Heat Transfer Coefficient

In the ceramic heat recuperator, the high temperature exhaust gases are separated from the low temperature combustion air by the ceramic material through which heat is transferred. The rate of heat transfer through an element of heat transfer area, dA , and temperature difference, dT , is given by:

$$dq = m_h C_{Ph} dT_h = m_c C_{Pc} dT_c \quad \text{B-1}$$

Where "m" and "C" are mass flow rates (pounds/hour) and specific heats (BTU/pound F) of the fluid, and subscripts "h" and "c" refer to the hot and cold fluids. The rate of heat transfer may also be expressed as:

$$dq = U \Delta T_m dA \quad \text{B-2}$$

Where:

U = the overall heat transfer coefficient (BTU/hour ft^2 F) or overall thermal conductance.

ΔT_m = a mean temperature difference across the heat exchanger (F).

dA = element transfer area (ft^2).

The reciprocal of U represents a resistance to flow of heat through the hot fluid boundary layer, the ceramic material, and the cold fluid boundary layer.

$$\frac{1}{U} = \frac{1}{h_h} + \frac{\Delta X}{k} + \frac{1}{h_c} \quad \text{B-3}$$

Where:

h_h = hot side convective film coefficient (BTU/hour ft² F).

ΔX = thickness of the ceramic material (ft).

k = thermal conductivity of the ceramic material.

h_c = cold side convective film coefficient (BTU/hour ft² F).

B.2 The Logarithmic-Mean Temperature Difference

In a cross-flow heat exchanger, the two fluids flow at right angles to each other on either side of the material that separates them. Since the fluids are unmixed, the temperatures vary both in the direction of flow and normal to the flow as shown in Figure B.1. To determine the actual rate of heat transfer, Equation B-2 is integrated over the heat transfer area "A" along the length of the heat exchanger by assuming U as a constant. In the case of a counterflow heat exchanger, integration of Equation B-2 yields:

$$q = UA \frac{(T_{h2} - T_{c2}) - (T_{h2} - T_{c1})}{\ln [(T_{h2} - T_{c2}) / (T_{h1} - T_{c1})]} \quad B-4$$

$$\text{or } q = UA \Delta T_{lm} \quad B-5$$

Where:

T_{h1} = temperature of the hot inlet gas

T_{h2} = temperature of the hot outlet gas

T_{c1} = temperature of the cold inlet air

T_{c2} = temperature of the preheated outlet air

ΔT_{lm} is called the logarithmic-mean temperature difference (LMTD) and is defined as:

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln \Delta T_2 / \Delta T_1} \quad B-6$$

Figure B.1

Temperature Distribution in a Cross-Flow Heat Exchanger
With Both Fluids Unmixed

In the case of a cross-flow heat exchanger, the integration process becomes complex and the above (LMTD) is modified by a correction factor, F , published in the form of a graph by Bowman et. al.(10) and depicted in Figure B.2. Therefore, Equation B-5 becomes:

$$q = UAF \Delta T_{lm} \quad B-7$$

The curves for F presented in Figure B.2 have a large slope for lower values of F and thus small changes in temperatures could result in large changes of LMTD. If a correction factor of less than 0.8 results, design practice usually dictates multiple passes. The LMTD is still an approximation because U is not actually constant as was assumed; however, it is common practice to estimate U for a mean section of the heat exchanger.

The LMTD method of performing heat exchanger calculations is relatively simple provided the terminal temperatures are known. In cases where the terminal temperatures are to be calculated, an iterative procedure must be employed by first assuming a value of terminal temperatures, using the equations to calculate the terminal temperature, and repeating the process until the assumed and calculated values converge.

B.3 Heat Exchanger Effectiveness

The user of a heat exchanger needs to know how efficiently energy is transferred from the hot fluid to the cold fluid with external losses neglected. Heat exchanger effectiveness, (E), is defined as the ratio of the actual heat transferred to the maximum possible amount of heat that could be transferred.

Thus,

$$E = \frac{q}{q_{\max}} = \frac{C_h (T_{h1} - T_{h2})}{C_{\min} (T_{h1} - T_{c1})} = \frac{C_c (T_{c2} - T_{c1})}{C_{\min} (T_{h1} - T_{c1})} \quad B-8$$

Where:

C = capacity rate in BTU/hour F .

= $m C_p$

and $C_{\min}$ = lesser of the hot and cold fluid capacity rates.

Effectiveness is therefore some function of the mass flow rate of the fluid, the fluid specific heat and the flow arrangement of the particular heat exchanger which includes the overall heat transfer coefficient and the heat transfer surface area. Obviously, the maximum possible heat transfer could be achieved by having a unit of infinite surface area, but this is impractical. Limiting the surface area of a heat exchanger for economic and space considerations would therefore result in effectiveness values of

Figure B.2

CORRECTION FACTOR (F) FOR LOG-MEAN TEMPERATURE
DIFFERENCE (LMTD) FOR A SINGLE-PASS CROSS-FLOW HEAT
EXCHANGER WITH BOTH FLUIDS UNMIXED (3)

less than unity. If the effectiveness of a heat exchanger is known, one can calculate the heat transferred without knowledge of the outlet terminal temperatures.

$$q = EC_{\min} (T_{h_1} - T_{c_1}) = UA \Delta T_{lm} \quad B-9$$

The term $UA/C_{\min}$ is dimensionless and is called the number of transfer units (NTU).

$$NTU = \frac{UA}{C_{\min}} \quad B-10$$

From the definition of NTU, one sees that it physically relates to the heat transfer size of the heat exchanger. The larger the value of NTU the more closely will the exchanger approach the theoretical limit. The heat exchanger effectiveness is thus a function of the NTU, the capacity ratio of the two fluids and the specific arrangement of the heat exchanger.

$$E = f \left(NTU, \frac{C_{\min}}{C_{\max}}, \text{flow arrangement} \right) \quad B-11$$

The effectiveness can be calculated for various flow arrangements, such as cross-flow as was done in the case of the LMTD in the previous section. The results are presented graphically for a cross-flow heat exchanger in Figure B.3. The physical significance and use of the NTU-E curves is extremely important in heat exchanger design. If the heat transfer effectiveness of a given heat exchanger is low, for a constant capacity rate ratio, the value of the NTU is low. This would indicate that the heat transfer area may be too small to effectively use the available energy. One could improve the performance of the heat exchanger by providing more surface area (higher values of NTU). In the region of asymptotic approach in Figure B.3 increasing the surface area to increase the NTU does not result in much gain in heat transfer effectiveness, but does increase cost, weight, space and pressure drop for that exchanger design. These four practical considerations thus limit the values of NTU that are commonly used in heat exchanger design.

The NTU-E approach has some advantages over the LMTD approach in heat exchanger design. It gives the designer a better feel for what is physically happening in the unit as well as the practical considerations of cost, surface area and pressure drop. It is easy to perform calculations if the outlet terminal temperatures are unknown.

B.4 Heat Exchanger Pressure Drop

When a gas flows in a pipe isothermally, its pressure decreases. This pressure drop is a function of the type of flow and the pipe roughness. The type of flow is usually expressed in terms of a dimensionless number called the Reynolds Number (N_R) defined below:

Figure B.3

Heat-Exchanger Effectiveness for a Cross-Flow
Recuperator With Both Fluids Unmixed

$$N_R = \frac{VD\ell}{\mu} = \frac{GD}{\mu} \quad \text{B-12}$$

Where V = fluid velocity, ft/hour
 D = pipe diameter, ft
 ℓ = gas density, pound/ft³
 μ = dynamic viscosity of the gas, pounds/ft hour
 G = the mass velocity, pounds/hour ft²

$$= V\ell = \frac{\text{mass flow rate}}{\text{free flow area}} = \frac{m}{A_c}$$

The physical significance of N_R is that it is the ratio of the inertia forces to the viscous forces in a flowing fluid. The pressure drop resulting from a fluid flowing in a pipe is a function of the Reynolds number and may be calculated by the equation below:

$$P = f \frac{2 G^2 L}{g_c \ell D} \quad \text{B-13}$$

Where f = friction factor
 L = length of pipe
 g_c = gravitational constant

The friction factor is usually plotted versus the Reynolds Number and may be found in any standard text on heat transfer.

The pressure drop resulting from gases flowing through a heat exchanger core is a very important design parameter because it relates to the energy required to force the gases through the heat exchanger. In retrofit heat exchanger applications the allowable pressure drop is a critical design criteria that impacts the design of the heat exchanger. The pressure losses in a heat exchanger core are made up of entrance, acceleration, surface friction and exit losses:

$$\begin{aligned} \frac{\Delta P}{P_1} = & \frac{G^2}{2g_c} \frac{v_1}{P_1} \left[\underbrace{(K_c + 1 - \sigma^2)}_{\text{Entrance Effect}} + \underbrace{2\left(\frac{v_2}{v_1} - 1\right)}_{\text{Flow Acceleration}} \right] \\ & + \underbrace{\left(f \frac{A}{A_c} \frac{v_m}{v_1}\right)}_{\text{Core Friction}} - \underbrace{(1 - \sigma^2 - K_e) \frac{v_2}{v_1}}_{\text{Exit Effect}} \end{aligned} \quad \text{B-14}$$

Where, g_c = gravitational constant
 v = fluid specific volume
 v_m = mean specific volume
 K_c = entrance loss coefficient
 σ = free flow area/frontal area = A_c/A_f
 f = friction factor
 K_e = exit loss coefficient

It is common practice to test heat exchangers isothermally in a wind tunnel to determine an overall friction factor which includes entrance and exit loss effects and present the results in a plot of "f" versus N_R (see Section 2.7.3). Equation B-14 then reduces to:

$$\frac{\Delta P}{P_1} = \frac{G^2}{2g_c} \frac{v_1}{P_1} \left[f \frac{A}{A_c} \frac{v_m}{v_1} \right] \quad \text{B-15}$$

Note that since $L/D_h = A/4A_c$, Equation B-15 reduces to Equation B-13 presented earlier. The calculated pressure drop will be affected by the variation of fluid properties with temperatures and the friction factor "f" can be corrected to account for these variations.

B.5 Heat Transfer and Flow Friction Design Data

In order to determine the overall heat transfer coefficient U , one must know the values of the convective film coefficients on the air and gas sides of the heat exchanger. Methods of evaluating individual coefficients are presented in the literature. Evaluation of these coefficients from semi-empirical equations and advanced analogies is complicated. For this reason, heat exchanger manufacturers test their cores in wind tunnels under a range of flow conditions to determine the heat transfer and pressure drop characteristics of the cores. The heat transfer and friction data collected can then be expressed by means of a correlation. It is common practice to plot "f" and "j" versus Reynolds Number. "j" is called the Colburn factor and is defined as:

$$j = N_{st} N_{pr}^{2/3} \quad \text{B-16}$$

Where N_{st} = the Stanton Number, a dimensionless heat transfer modulus.
 $= h/GC_p$
 N_{pr} = the Prandtl Number, a dimensionless fluid property modulus.
 $= \mu C_p / k$

For a given heat exchanger and flow conditions, the film coefficient h for each side can be calculated from the j value and thus the overall heat transfer coefficient determined. Figure B.4 is an illustration of an " f " and " j " plot(3) for flow through a rectangular passage. Because fluid properties are temperature dependent, a correction factor may be applied when the difference between the fluid and surface temperatures is large.

B.6 Recuperator Design Procedure

A recuperator manufacturer will usually have tested various surface geometries to be used in the heat exchanger core element. When called upon to supply a recuperator for a specific furnace application, he will have to decide on the size and type of recuperator to accomplish the heat transfer function while still meeting any pressure drop constraints in that application. The latter factor is extremely important in retrofit applications. Usually the supplier will be given certain specifications that may include air and exhaust flow rates, exhaust gas temperatures and the maximum allowable pressure drop on the exhaust gas side. He will have to select a core geometry and determine the recuperator effectiveness and pressure drops on the air and gas sides. The procedure to be followed in this analysis will use the equations and data outlined in the previous subsections and is as follows:

1. Determine the surface characteristics for the selected core that will include the hydraulic diameter, frontal and free flow surface areas on both the gas and air sides.
2. Determine the air and gas properties for the operating temperatures in the system and the excess air used in the burner. Appendix C provides these properties as a function of temperature.
3. Determine the Reynolds and Prandtl Numbers.
4. From the experimental data for the selected core (such as Figure 2.18) obtain the " f " and " j " values.
5. ~~The Stanton Number and the convective film coefficients can be calculated for both the air and gas sides. The overall heat transfer coefficient (U) can then be calculated.~~
6. The NTU and heat exchanger effectiveness are determined and the outlet air and gas temperatures calculated.
7. The air and gas side pressure drops are computed using the " f " values values obtained from the flow friction versus Reynolds Number curves.

The supplier is now in a position to analyze the system, apply any trade-offs and thus optimize his design. Obviously, if the gas side pressure drop exceeds the given specification, another surface geometry needs to be selected and the calculations repeated.

Figure B.4

GAS FLOW INSIDE RECTANGULAR TUBES WITH ABRUPT CONTRACTION

ENTRANCES: A SUMMARY OF EXPERIMENTAL AND ANALYTICAL DATA.(2)

If the calculated pressure drop is well within the specified limits, the supplier may elect to increase the surface area by increasing the length of the core and thus improve the heat exchanger effectiveness. While the design can thus be optimized from the space, weight and technical considerations, it must be emphasized that economics of both the heat exchanger and the associated equipment will play a key role in the final selection of the recuperator. Appendix E provides a typical furnace application example so that the reader can perform the calculation and follow the logic involved in optimizing the recuperator.

APPENDIX C
FLUID PROPERTIES

<u>Figure Number</u>	<u>Title</u>
C.1	Properties of Air at Elevated Temperatures
C.2	Properties of Combustion Products of Alaska Natural Gas at 0% Excess Air
C.3	At 5% Excess Gas
C.4	At 2% Excess Air
C.5	At 5% Excess Air
C.6	At 10% Excess Air
C.7	At 100% Excess Air
C.8	At 200% Excess Air

<u>Table Number</u>	
C.1	Enthalpy of Air at Low Pressure
C.2	Enthalpy of Ideal Gases
C.3	Enthalpy of Gaseous Hydrocarbon Fuels at Low Pressure

Figure C.1

AIR (SP. GR. = 1.0) (0.0765 LB/CU. FT.)

MOLECULAR WT. = 28.964

PROPERTIES OF DRY AIR AT ELEVATED TEMPERATURES

$$C_p, K, U = \left(\sum_{i=1}^4 A_i T^{(i-1)} \right) * 10.0^{(-3 * (i-1))}$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U * C / K$$

	A1	A2	A3	A4
C _p	0.231770	0.015217	0.039501	0.016213
K	0.001796	0.047570	-0.010380	0.000000
U	0.005924	0.144511	-0.063234	0.016094

Figure C.2

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 30.617

PROPERTIES OF COMBUSTION PRODUCTS AT 0% XS-AIR

$$C_p, K, U = \left(\sum_{i=1}^n A_i T^{(i-1)} + \sum_{j=1}^m B_j T^{(-3+(-1)j)} \right) * 10.0$$

NOTE: IN THE ABOVE EQUATIONS, SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U * C_p / K$$

	A1	A2	A3	A4
C _p	0.247362	0.039402	0.040946	-0.022752
K	0.000658	0.045407	-0.009530	0.001507
U	0.003458	0.133860	-0.052690	0.013037

Figure C.3

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 30.681

PROPERTIES OF COMBUSTION PRODUCTS AT 5% XS-GAS

$$C_p, K, U = \left(\sum_{i=1}^4 A_i T^{(i-1)} + (-3 \cdot (i-1)) \right) * 10.0$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U * C / K$$

	A1	A2	A3	A4
C _p	0.247802	0.040624	0.042045	-0.023354
K	0.000636	0.045403	-0.009365	0.001410
U	0.003442	0.133614	-0.052499	0.012981

Figure C.4

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 30.594

PROPERTIES OF COMBUSTION PRODUCTS AT 2% XS-AIR

$$C_p, K, U = \left(\frac{A}{1-A} \right) T^{(1-1)} \times 10.0 \quad (-3 \cdot (1-1))$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
 VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U \cdot C_p / K$$

	A1	A2	A3	A4
C _p	0.246955	0.039304	0.040565	-0.022603
K	0.000659	0.045531	-0.009846	0.001578
U	0.003585	0.133771	-0.052583	0.012992

Figure C.5

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 30.560

PROPERTIES OF COMBUSTION PRODUCTS AT 5% XS-AIR

$$C_p, K, U = \left(\sum_{i=1}^4 A_i T^{(i-1)} + (-3 * (1-1)) \right) * 10.0$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL} \# = U * C / K$$

	A1	A2	A3	A4
C _p	0.246582	0.038367	0.040887	-0.022574
K	0.000706	0.045535	-0.009950	0.001513
U	0.003665	0.134034	-0.052870	0.013077

Figure C.6

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 30.508

PROPERTIES OF COMBUSTION PRODUCTS AT 10% XS-AIR

$$C_p \cdot K \cdot U = \left(\frac{A_1}{A_2} T^{(1-1)} \right) \cdot 10.0^{(-3 \cdot (1-1))}$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U \cdot C / K$$

	A1	A2	A3	A4
C _p	0.245946	0.037130	0.041151	-0.022707
K	0.000775	0.045553	-0.010154	0.001675
U	0.003847	0.134211	-0.053047	0.013115

Figure C.7

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 30.004

PROPERTIES OF COMBUSTION PRODUCTS AT 100% XS-AIR

$$C_p, K, U = \left(\sum_{i=1}^4 A_i T^{(i-1)} + (-3 \cdot (1-1)) \right) * 10.0$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U * C / K$$

	A1	A2	A3	A4
C _p	0.239234	0.027316	0.041242	-0.022190
K	0.001481	0.045604	-0.011875	0.002213
U	0.005262	0.137302	-0.056321	0.014017

Figure C.8

ALASKA NATURAL GAS (SP. GR. = 0.60)

MOLECULAR WT. = 29.791

PROPERTIES OF COMBUSTION PRODUCTS AT 200% XS-AIR

$$C_p, K, U = \left(\frac{A}{1-A} \right) T^{(1-1)} * 10.0^{(-3*(1-1))}$$

NOTE: IN THE ABOVE EQUATIONS SUBSTITUTE THE TEMPERATURE IN DEG. KELVIN.
VARIABLE VALUES OBTAINED ARE IN THE ABOVE STANDARD UNITS

$$\text{PRANDTL \#} = U * C / K$$

	A1	A2	A3	A4
C _p	0.236555	0.022231	0.042351	-0.022334
K	0.001770	0.045656	-0.012536	0.002452
U	0.005796	0.132241	-0.057965	0.014486

TABLE C.1
ENTHALPY OF AIR AT LOW PRESSURE*

<u>Absolute Temperature °R</u>	<u>Temperature °F</u>	<u>Enthalpy BTU/lb m</u>
0	-460	0
500	40.3	119.48
537	77.3	128.34
600	140.3	143.47
700	240.3	167.56
800	340.3	191.81
900	440.3	216.26
1000	540.3	240.98
1100	640.3	256.99
1200	740.3	291.30
1300	840.3	316.94
1400	940.3	342.90
1500	1040.3	369.17
1600	1140.3	395.74
1700	1240.3	422.59
1800	1340.3	449.71
1900	1440.3	477.09
2000	1540.3	504.71
2500	2040.3	645.78

*J. Keenan and J. Kaye, "Thermodynamic Properties of Air," John Wiley and Sons, New York.

TABLE C.2
ENTHALPY OF IDEAL GASES*
 (BTU/mole)

<u>T, °R</u>	<u>t, °F</u>	<u>CO₂</u>	<u>H₂O</u>	<u>N₂</u>	<u>O₂</u>
0	-460	0	0	0	0
500	40	3,706	3,962	3,472	3,466
537	77	4,030	4,258	3,730	3,725
1000	540	8,682	8,079	6,978	7,088
1500	1,040	14,576	12,551	10,649	11,017
2000	1,540	21,019	17,439	14,534	15,164
3000	2,540	34,807	28,386	22,762	23,818
4000	3,540	49,231	40,489	31,329	32,806
5000	4,540	64,000	53,327	40,080	42,086

*Keenan and Kay, "Gas Tables," John Wiley and Sons, Inc., New York, 1948

TABLE C.3

ENTHALPY OF GASEOUS HYDROCARBON FUELS AT LOW PRESSURE*
(BTU/mole)

<u>T, °R</u>	<u>t, °F</u>	Methane <u>CH₄</u>	Ethane <u>C₂H₆</u>	Propane <u>C₃H₈</u>	m-Butane <u>C₄H₁₀</u>
0	-460	0	0	0	0
537	77	4,312	5,137	6,318	8,356
600	140	4,875	5,982	7,501	9,929
800	340	6,791	9,042	11,880	15,734
1,000	540	9,012	12,752	17,259	22,805
2,000	1,540	24,334	38,894	54,944	71,473

*"Selected Values of Properties of Hydrocarbons," National Bureau of Standards Circular C-461, 1947

APPENDIX D
CORE GEOMETRIES

<u>Table No.</u>	<u>Title</u>
D.1	Summary Table
D.2	Geometry of a .125 x .125 Matrix
D.3	Geometry of a .125 x .200 Matrix
D.4	Geometry of a .125 x .250 Matrix
D.5	Geometry of a .125 x .300 Matrix
D.6	Geometry of a .125 x .400 Matrix
D.7	Geometry of a .125 x .500 Matrix
D.8	Geometry of a .125 x .600 Matrix
D.9	Geometry of a .200 x .200 Matrix
D.10	Geometry of a .300 x .300 Matrix

TABLE D.1
SUMMARY TABLE

<u>Table No.</u>	<u>Air Side Core Geometry</u>		<u>Gas Side Core Geometry</u>	
	<u>Green Rib Height</u>	<u>Fired Rib Height</u>	<u>Green Rib Height</u>	<u>Fired Rib Height</u>
D.2	.125	.112	.125	.112
D.3	.125	.112	.200	.187
D.4	.125	.112	.250	.224
D.5	.125	.112	.300	.281
D.6	.125	.112	.400	.375
D.7	.125	.112	.600	.467
D.8	.125	.112	.600	.552
D.9	.200	.187	.200	.187
D.10	.300	.381	.300	.281

TABLE D.2

GEOMETRY OF A .125 IN. x .125 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = $10.3 \times 9.8 = 100.42$ (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = $10.3 \times 9.8 = 100.42$ (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	481.93	0.039	5.88
***	0.042	0.112	0.648	481.93	0.039	5.88

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	34.977	0.191	8790.434	15.740	1.500	62.83
***	34.977	0.191	8790.434	14.766	0.370	62.83

⁺Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.3

GEOMETRY OF A .125 IN. x .200 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = 10.3 x 9.9 = 101.97 (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = 10.3 x 9.9 = 101.97 (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	392.00	0.039	5.88
***	0.042	0.187	0.648	392.00	0.039	3.45

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	28.449	0.191	7150.004	15.740	1.500	62.83
***	47.500	0.290	7855.590	14.766	0.370	41.34

⁺Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.4

GEOMETRY OF A .125 IN. x .250 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = $10.3 \times 10.2 = 105.06$ (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = $10.3 \times 10.2 = 105.06$ (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	367.78	0.039	5.88
***	0.042	0.224	0.648	367.78	0.039	2.86

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	26.692	0.191	6708.297	15.740	1.500	62.83
***	53.384	0.333	7696.891	14.766	0.370	36.04

⁺ Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.5

GEOMETRY OF A .125 IN. x .300 IN. CORE[†]

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = 10.3 x 10.1 = 104.03 (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = 10.3 x 10.1 = 104.03 (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	320.10	0.039	5.88
***	0.042	0.281	0.648	320.10	0.039	2.33

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	23.232	0.191	5838.652	15.740	1.500	62.83
***	58.287	0.392	7136.984	14.766	0.370	30.61

[†] Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.6

GEOMETRY OF A .125 IN. x .400 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = $10.3 \times 10.3 = 106.09$ (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = $10.3 \times 10.3 = 106.09$ (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	272.13	0.039	5.88
***	0.042	0.375	0.648	272.13	0.039	1.72

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	19.750	0.191	4963.648	15.740	1.500	62.83
***	66.128	0.475	6681.332	14.766	0.370	25.26

⁺Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.7GEOMETRY OF A .125 IN. x .500 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = 10.3 x 9.8 = 100.94 (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = 10.3 x 9.8 = 100.94 (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	222.66	0.039	5.88
***	0.042	0.467	0.648	222.66	0.039	1.39

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	16.160	0.191	4061.378	15.740	1.500	62.83
***	67.382	0.543	5958.469	14.766	0.370	22.11

⁺ Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.8

GEOMETRY OF A .125 IN. x .600 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = 10.3 x 9.4 = 96.82 (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = 10.3 x 9.4 = 96.82 (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.112	0.648	189.11	0.039	5.88
***	0.042	0.552	0.648	189.11	0.039	1.18

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	13.725	0.191	3449.345	15.740	1.500	62.83
***	67.643	0.596	5446.332	14.766	0.370	20.13

⁺ Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.9

GEOMETRY OF A .200 IN. x .200 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = 10.3 x 9.9 = 101.97 (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = 10.3 x 9.9 = 101.97 (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.187	0.648	326.95	0.039	3.45
***	0.042	0.187	0.648	326.95	0.039	3.45

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	39.619	0.290	6552.117	15.740	1.500	41.34
***	39.619	0.290	6552.117	14.766	0.370	41.34

⁺ Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

TABLE D.10

GEOMETRY OF A .300 IN. x .300 IN. CORE⁺

Crossflow Recuperator - 12 x 12 x 12 inch Core

Preheat Window = 10.3 x 9.7 = 99.91 (sq in)

Length of Air Passage = 12.0 inch

Exhaust Window = 10.3 x 9.7 = 99.91 (sq in)

Length of Gas Passage = 12.0 inch

	<u>Rib Thickness (in)</u>	<u>Rib Height (in)</u>	<u>Dist Btwn Ribs (in)</u>	<u>Number of Openings</u>	<u>Thickness Plate (in)</u>	<u>Aspect* Ratio</u>
**	0.042	0.281	0.648	226.25	0.039	2.33
***	0.042	0.281	0.648	226.25	0.039	2.33

	<u>Free Flow Area (sq in)</u>	<u>Hydrl Dmtr (in)</u>	<u>Surface Area (sq in)</u>	<u>Pressure psia</u>	<u>Press. Drop in H₂O</u>	<u>L/D_h</u>
**	41.197	0.392	5044.363	15.740	1.500	30.61
***	41.197	0.392	5044.363	14.766	0.370	30.61

⁺Size shown is the rib height in the unfired state. During firing settling and shrinkage reduce the actual height of the passages to that shown below.

* Aspect Ratio = Distance between ribs/rib height

** Preheat

*** Exhaust

APPENDIX EEXAMPLES OF CALCULATIONS PERFORMED FOR A
TYPICAL FURNACE APPLICATIONExample 1

A furnace on a galvanizing line is fired by natural gas with 10% excess air. The fuel input to the burner is 3.8 MM BTU/hour. The gases leave the furnace at 1500 °F and are exhausted out the stack. Because of restrictions on fuel usage and rising fuel costs, a decision is made to install a recuperator to preheat the combustion air. A quotation from a recuperator manufacturer sizes a unit and estimates the operating characteristics of their unit as follows:

Exhaust gas temperature entering the recuperator = 1500 °F

Exhaust gas temperature leaving the recuperator = 885 °F

Ambient temperature = 77 °F

Preheated combustion air temperature = 650 °F

How much fuel is saved by the use of the recuperator?

Solution

The results of this problem were presented in Table A.2. The detailed calculations are presented here.

$$m_1 h_1 + m_1 \Delta H_R + 18.97 m_1 h_2 = 19.97 m_1 h_3 \quad \begin{array}{l} \text{A-2} \\ \text{(modified)} \end{array}$$

m_1 cancels out on both sides of the equation and Equation A-2 provides

$$h_3 = \frac{270 + 21,500 + 18.97 \times 128.4}{19.97} = 1,214 \text{ BTU/lb}$$

The enthalpy of ambient air is 128.4 BTU/lb.

$$\text{Also, } m_1 = \frac{3.8 \times 10^6}{21,500} = 177 \text{ lb/hour}$$

$$\therefore m_2 = 18.97 \times 177 = 3,358 \text{ lb/hour}$$

$$\text{and } m_3 = 19.97 \times 177 = 3,535 \text{ lb/hour}$$

Energy Input to Furnace ($m_3 h_3$)

$$m_3 h_3 = 3535 \times 1214 = 4.291 \times 10^6 \text{ BTU/hour}$$

From Figure A.3, $h_4 = 585$ BTU/lb since $t_4 = 1500$ °F

$$\therefore \text{Energy out the stack } m_4 h_4 = 3535 \times 585 = 2.068 \times 10^6 \text{ BTU/hour}$$

From Equation A-2, the furnace load and losses

$$Q_L = (m_4 h_4 - m_3 h_3)$$

$$Q_L = - (4.291 - 2.068)10^6 = -2.223 \times 10^6 \text{ BTU/hour}$$

The negative sign denotes that heat is transferred from the system to the load and the surroundings.

The maximum recoverable energy from the stack gases

$$= m_4 (h_4 - h_{\text{atm}})$$

$$= 3535(585 - 128)$$

$$= 1.615 \times 10^6 \text{ BTU/hr}$$

With Recuperation - Method 1

$$m_1 h_1 + m_1 W H_R + 18.97 m_1 h_2 = 19.97 m_1 h_3$$

$$\therefore h_3 = \frac{270 + 21,500 + 18.97 \times 268.5}{19.97}$$

Where the enthalpy of the preheated air is 268.5 BTU/lb

$$h_3 = 1345 \text{ BTU/lb}$$

From Figure A.3, $t_3 = 3610$ °F, and from Equation A-7

$$Q_L = 19.97 m_1 (h_4 - h_3), \text{ note } Q_L \text{ is assumed constant in Method 1}$$

$$\text{or } m_1 = \frac{-2.223 \times 10^6}{19.97(585 - 1345)}$$

$$= 146.5 \text{ lb/hr}$$

$$\therefore m_2 = 18.97 \times 146.5 = 2779 \text{ lb/hr}$$

$$\text{and } m_3 = 19.97 \times 146.5 = 2925.5 \text{ lb/hr}$$

$\therefore$ the energy input to the furnace

$$H_3 = 2925.5 \times 1345 = 3.935 \times 10^6 \text{ BTU/hr}$$

The total enthalpy of the gases leaving the furnace H_4 may now be determined.

$$\therefore H_4 = 2925.5 \times 585 = 1.711 \times 10^6 \text{ BTU/hr}$$

$$\text{and } H_5 = 2925.5 \times 385 = 1.126 \times 10^6 \text{ BTU/hr}$$

Heat transferred from the exhaust gas to the air

$$\begin{aligned} &= H_4 - H_5 \\ &= 0.585 \times 10^6 \text{ BTU/hr} \end{aligned}$$

Heat transferred to the combustion air

$$\begin{aligned} &= H_2 - H_6 \\ &= 2779 (268.5 - 128.4) \\ &= 0.389 \times 10^6 \text{ BTU/hr} \end{aligned}$$

The heat transferred from the exhaust gas is higher than that transferred to the combustion air because of losses from the recuperator.

The Percent Fuel Savings

$$= \frac{177 - 146.5}{177} = 17.25\%$$

Instead of performing the above calculations, one could refer to Table A.1 from the North American Combustion Handbook to directly obtain energy savings as 17.25%.

With Recuperator - Method 2

Again, Equation 2-6 will give

$$h_3 = 1345 \text{ BTU/lb}$$

$$\therefore t_3 = 3610 \text{ F}$$

The total enthalpy input to the furnace is assumed constant in this case.

$$\therefore H_3 = m_3 h_3 = 4.291 \times 10^6 \text{ BTU/hour}$$

$$\text{from which } 19.97 m_1 h_3 = 4.291 \times 10^6$$

$$\therefore m_1 = 160 \text{ lb/hour}$$

$$\text{and } m_2 = 18.97 \times 160 = 3030 \text{ lb/hour}$$

$$m_3 = 3190 \text{ lb/hour}$$

The furnace load and losses is now determined as

$$\begin{aligned} Q_L &= H_4 - H_3 \\ &= 3190 (585 - 1345) \\ &= -2.424 \times 10^6 \text{ BTU/hour} \end{aligned}$$

The negative sign is because heat is transferred out of the system.

Note that the furnace load and losses have increased in value because of the small increase in radiation losses from the furnace because of the higher temperature of the exhaust gases entering the furnace. The calculated value of Q_L indicates that the load production of the furnace will be maintained.

The enthalpy lost up the stack (H_5) is:

$$H_5 = 3190 \times 385 = 1.228 \times 10^6 \text{ BTU/hour}$$

$$H_4 = 3190 \times 585 = 1.866 \times 10^6 \text{ BTU/hour}$$

$$\begin{aligned} \text{The heat transferred from the exhaust gases in the recuperator } (H_4 - H_5) \\ = 0.638 \times 10^6 \text{ BTU/hour} \end{aligned}$$

Heat transferred to the combustion air

$$= 3030 \times 140.1$$

$$= 0.424 \times 10^6 \text{ BTU/hour}$$

$$\text{Percent Fuel Saved} = \frac{177 - 160}{177}$$

$$= 9.6\%$$

Example 2

Given a 12 x 12 x 12 inch ceramic cube with the following data on the test furnace:

Airflow	=	11000 SCFH
Exhaust Gas Flow	=	12000 SCFH
Exhaust Gas Temperature Out	=	1115 °F
Air Temperature In	=	116 °F
Air Temperature Out	=	887 °F
Gas Side Pressure Drop	=	0.33 inch H ₂ O
Air Side Pressure Drop	=	2.16 inch H ₂ O

Determine the heat transfer and pressure drop performance of the ceramic recuperator.

Solution

1. Determination of the Core Properties

The core properties may be found in Appendix D

	<u>Air</u>	<u>Gas</u>
Thickness of Plate, inch	0.043	0.043
Thickness of Ribs, inch	0.043	0.043
Distance Between Ribs, inch	0.71	0.71
Height of Flow Passage, inch	0.121	0.294
Length of Flow Passage (L), inch	12.0	12.0
Open Window, inch x inch	10 x 10	
Frontal Area (A_{fr}) in ²	100	100
Free Flow Area (A_c) in ²	22.77	55.33
Surface Area (A) in ²	5286.5	6387.1
Hydraulic Diameter ($\frac{4A_c}{A} = D_h$) in.	0.2067	0.4158
L/ D_h	58.1	28.9

2. Fluid Properties

The average fluid properties may be determined from Appendix C

	<u>Air</u>	<u>Gas</u>
Average Temperature °F	501	1559
Viscosity lbs/hour ft	0.065	0.106
Specific heat, C_p BTU/lb °F	0.245	0.285
Thermal Conductivity k - BTU/hour ft ² /ft	0.025	0.043
Prandtl Number N_{Pr}	0.68	0.71

Performance Calculations

$$m_a = \frac{14.7 \times 144 \times 11000}{53.34 \times 540} = 808.4 \text{ lb/hour}$$

$$m_g = \frac{14.7 \times 144 \times 12000}{53.34 \times 540} = 881.9 \text{ lb/hour}$$

The mass velocity = $\frac{\text{mass flow rate}}{\text{freeflow area}}$

$$G_a = \frac{808.4}{22.77} \times 144 = 5112 \text{ lb/hour ft}^2$$

$$G_g = \frac{881.9}{55.33} \times 144 = 2295 \text{ lb/hour ft}^2$$

The Reynolds Numbers may now be determined from Equation B-12

$$N_{Ra} = 5112 \times \frac{0.2067}{12} \times \frac{1}{0.065} = 1354$$

$$N_{Rg} = 2295 \times \frac{0.4158}{12} \times \frac{1}{0.106} = 750$$

The Capacity Rates (C) are now calculated

$$C_a = m_a C_{Pa} = 808.4 \times 0.245 = 195.05 \text{ BTU/hour °F}$$

$$C_g = 881.9 \times 0.285 = 251.33 \text{ BTU/hour °F}$$

$$\therefore \frac{C_{min}}{C_{max}} = \frac{195.05}{251.33} = 0.788$$

The heat exchanger effectiveness can now be determined from Equation B-8

$$E_a = \frac{198.05}{198.05} \times \frac{771}{1887} = 0.4085$$

$$E_g = \frac{251.33}{198.05} \times \frac{888}{1887} = 0.5972$$

From Equation B-6 the logarithmic mean temperature difference is determined

$$\Delta T_{lm} = \frac{(1115 - 887) - (2003 - 116)}{\ln \frac{228}{1887}}$$

$$= 1659/2.1133$$

$$\therefore \Delta T_{lm} = 785$$

The heat transferred to the air

$$\begin{aligned} Q_a &= 808.4 \times 0.245 \times 771 \\ &= 152,700 \text{ BTU/hour} \end{aligned}$$

The overall heat transfer coefficient is now determined from Equation B-5

$$U = \frac{152,700 \times 144}{5286.5 \times 785} = 5.298$$

From Equation B-10

$$NTU_a = \frac{5286.5}{144} \times \frac{5.298}{198.05} = 0.982$$

From Equation 2-2

$$\begin{aligned} N_{st} &= \frac{NTU_a}{E_a} \times \frac{r_h}{L} \\ &= \frac{0.982 \times 0.2067}{0.4085 \times 4 \times 12} = 0.01035 \end{aligned}$$

From Equation B-16, the Colburn factor

$$j = 0.01035 \times 0.773 = 0.008$$

The superficial air velocity entering the core is now determined

$$V_a = \frac{11000}{3600} \times \frac{144}{22.77} = 19.323 \text{ ft/sec}$$

The Fanning Friction Factor can now be determined from the approximate Equation 2-1

$$\frac{373.4}{64.4} \times \frac{0.0735}{15.7 \times 144} = \frac{0.0779}{15.7 \times 0.982} \times \frac{0.0408}{0.0735} \times \frac{0.01035}{f} \times 0.4085$$

$$\therefore f = 0.063$$

The above calculations are merely an illustration as to how the heat transfer and flow friction performance calculations can be performed.

Example 3

Determine the preheat air temperature and pressure drops if a 12 x 12 x 12 in. matrix is used on the galvanizing furnace in Example 1.

Solution

From the mass flow rates in Example 1, the volumetric flow is calculated as:

Air Flow = 42544 SCFH

Exhaust Gas Flow = 47834 SCFH

The core properties may be determined from Appendix D.

	<u>Air</u>	<u>Gas</u>
Frontal Area, A_{fr} in ²	100	100
Free Flow Area, A_c in ²	22.77	55.33
Surface Area, A in ²	5286.5	6387.1
Hydraulic Diameter, D_h , in.	0.2067	0.5533
L/D_h	58.1	28.9
σ	0.2277	0.5533

The fluid properties may be determined from Appendix C.

	<u>Air</u>	<u>Gas</u>
Average Temperature Assumed	430	1150
Cp, BTU/lb °F	0.25	0.295
Viscosity μ , lb/hour ft	0.07	0.09
k, BTU/hour ft ² °F/ft	0.33	0.34
N _{Pr}	0.73	0.776
v, ft ³ /lb	21.0	40.57
ℓ , lb/ft ³	0.0476	0.0246

The mass velocity and Reynolds Numbers for both air and gas are now calculated:

$$G_a = \frac{3358}{22.77} \times 144 = 21,236 \text{ lb/hour ft}^2$$

$$G_g = \frac{3535}{55.33} \times 144 = 9,200 \text{ lb/hour ft}^2$$

$$N_{Ra} = \frac{21,236 \times 0.2067}{0.07 \times 12} = 5225$$

$$N_{Rg} = \frac{9,200 \times 0.4158}{0.09 \times 12} = 3542$$

The friction factor and Colburn factor are now determined from Figures 2.17 and 2.18.

$$f_a = 0.0082 \quad j_a = 0.0042$$

$$f_g = 0.0065 \quad j_g = 0.003$$

$$\therefore N_{sta} = j_a / N_{Pra}^{2/3} = \frac{0.0042}{0.8107} = 0.0052$$

$$N_{stg} = \frac{0.003}{0.844} = 0.00355$$

$$h_a = N_{sta} G_a C_{Pa} = 0.0052 \times 21236 \times 0.25 = 27.60 \text{ BTU/hour ft } ^\circ\text{F}$$

$$h_g = 0.00355 \times 9200 \times 0.295 = 9.63 \text{ BTU/hour ft } ^\circ\text{F}$$

The overall heat transfer coefficient is now calculated as follows:

$$\frac{1}{U_a} = \frac{1}{h_a} + \frac{1}{A_g/A_a h_g}$$

$$\therefore U_a = \frac{27.6 \times 9.63 \times 6387/5286}{6387/5286 \times 9.63 + 27.60} = 8.18 \text{ BTU/hour ft}^2 \text{ F}$$

$$\therefore NTU_a = \frac{A_a U_a}{C_a} = \frac{5286.5 \times 8.18}{144 \times 3358 \times 0.265} = 0.3375$$

The air side effectiveness is now determined from Figure B.3:

$$E_a = 0.25 = \frac{t_{a_{out}} - 77}{1423}$$

$$\therefore t_{a_{out}} = 433 \text{ } ^\circ\text{F}$$

The air and gas side pressure drops are now calculated from Equation B-14.

$$\Delta P_a = 6.08'' \text{ H}_2\text{O or } 3.52 \text{ oz/in}^2$$

$$\Delta P_g = 0.976'' \text{ H}_2\text{O or } 0.56 \text{ oz/in}^2$$

The calculated pressure drops seem reasonable and could be handled by the same blower in a retrofit application. However, if a higher preheat air temperature is desired (higher effectiveness), it becomes evident that the heat transfer surface area needs to be increased. Thus the next bigger size recuperator, a 12 x 12 x 18 in. may be selected and the calculations repeated.

APPENDIX FANALYSIS OF A CORDIERITE (MAS) RECUPERATOR FROM A
MOLYBDENUM HEAT-TREATING FURNACE

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ABSTRACT

With increasing fuel costs, efficient fuel use is greatly important in U.S. process heat industries. Significant increases in fuel usage efficiency are realizable by transferring exhaust heat to incoming air, which is subsequently used for fuel combustion. This air-heating process, or recuperation, can be accomplished by using fixed-boundary heat exchangers, called recuperators, to transfer this waste heat to the combustion air.

Available alloys cannot currently satisfy the demand for higher thermodynamic efficiencies from higher temperature recuperated combustion air because of their creep, corrosion, and oxidation properties at high temperatures. Therefore, selected ceramic materials are principal candidates for use in waste-heat recovery devices for high temperatures [1100 to 1650 °C (2000 to 3000 °F)] in flue gas. A leading configuration now under investigation is the cross-flow fixed-boundary extended-surface ceramic "honeycomb" heat exchanger, which has evolved through the development of regenerative heat wheels for vehicular engines. The need for a material that has both low thermal shock resistance and is also fabricable and relatively inexpensive has led to the development of cordierite (magnesium aluminosilicate - MAS).

A DOE contractor exposed a cordierite honeycomb recuperator for about 2200 hours to representative process temperatures from 1100 to 1500 °C (2000 to 2800 °F) in a molybdenum and tungsten reheating furnace.

This recuperator performed satisfactorily for test and exhaust inlet temperatures of 1100 to 1400 °C (2000 to 2600 °F). Electron microprobe and ceramographic examination demonstrated that molybdenum and tungsten carryover in the waste gas stream significantly penetrated in the cordierite at temperatures as low as 675 °C (1250 °F). The leading edges of the exhaust inlet section that were exposed to maximum temperatures and extensive metallic carryover showed significant microstructural modification. A subsequent accidental overfiring to 1560 °C (2850 °F) resulted in a slumping of this modified material.

Exhaust-to-air channel leakage caused deposits in air channels and modified the microstructure in bonding lines between layers in the honeycomb. Cracking also occurred in the air outlet section between the air and exhaust channels. Significant microstructural modification, including coarsening within the walls and glass formation on the air-channel wall surfaces, was observed.