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MARTIN MARIETTA

Technology Assessment of External Heat Systems for Stirling Heat Pumps

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TECHNOLOGY ASSESSMENT OF EXTERNAL HEAT SYSTEMS FOR STIRLING HEAT PUMPS

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ACRONYMS AND ABBREVIATIONS

AGA	American Gas Association
Al	aluminum
AMTI	Advanced Mechanical Technology, Inc.
ANSI	American National Standards Institute
ASTM	American Society for Testing and Materials
CGR	combustion gas recirculation
CO	carbon monoxide
COP	coefficient of performance
DOE	U.S. Department of Energy
EGR	exhaust gas recirculation
EHS	external heat system
GE	General Electric
J	joule
MTI	Mechanical Technology, Inc.
ng	nanogram
NTIS	National Technical Information Service
NTU	number of transfer units
NO_x	oxides of nitrogen
ORNL	Oak Ridge National Laboratory
PC	pulsed combustion
PTB	premixed transpiration burner
SCAQMD	South Coast Air Quality Management District of California
SE	Stirling engine
SEHP	Stirling engine heat pump
STL	steel
SST	stainless steel
TDB	turbulent diffusion burner
UL	Underwriters Laboratories, Inc.

ABSTRACT

A technology assessment and design improvement effort was undertaken for the Stirling engine heat pump external heat system (EHS) in order to reduce costs. It was found that only two applicable EHS design approaches have been developed to any extent: a relatively undeveloped design featuring a premixed fuel and air transpiration burner, and a turbulent diffusion type burner system developed by Mechanical Technology, Inc.

To evaluate and optimize the design concepts, an analytical model was developed that examined design and performance variables. The model calculated key temperatures, allowing the specification of materials requirements. Adherence to American National Standards Institute appliance furnace code material specifications was assumed. Concepts for EHS control systems were evaluated, and a cost-effective control system design was developed for the turbulent diffusion burner EHS.

The study reveals that optimizing the diffusion burner EHS design can result in significant cost savings. No clear choice between the diffusion burner and transpiration burner systems could be determined from this study, but the designs of both were further developed and improved. Estimates show the EHS based on a transpiration burner to have a manufactured cost that is roughly 70% of the turbulent diffusion burner EHS cost, but fuel efficiency is lower by about 18%.

1. INTRODUCTION AND SUMMARY

The major objective of this program was to assess the technology of the Stirling engine heat pump (SEHP) external heat system (EHS) and to improve its cost effectiveness. Current Stirling engine technology has many proven features that could make it a superior driver for natural-gas-fueled heat pumps; these include high efficiency, low air pollutant emissions, and low noise. A major barrier to the economic competitiveness of this application is high capital cost. As a major contributor to the high cost, the EHS of the SEHP is a prime target for cost reduction. Achieving significant EHS cost reduction would be a major step toward commercializing the Stirling heat pump, and it could have additional applications. The EHS has been considered a costly part of the SEHP largely because the requirement for high temperature at the engine heater head necessitates the use of high-temperature materials.

Specifically, the scope of this program was to

- examine the work performed in the past through an assessment of the literature,
- examine the work being performed by current developers,
- develop new concepts for evaluation, and
- use the background of Advanced Mechanical Technology, Inc. (AMTI), in appliance development to develop cost-effective designs and control strategies.

The intent of the first phase was to survey the body of Stirling engine literature, focusing on the EHS. The EHS as defined for this project includes the combustor, air preheater, air and fuel delivery system, associated ductwork, and controls. It was felt that EHS development in the past had played a secondary role to the mechanical and cycle efficiency issues. Because cost and reliability concerns are now becoming as important as performance, it was deemed proper to review the past work to evaluate various investigators' solutions to EHS system design problems, determine the performance and cost effectiveness of these solutions, and develop new and more cost-effective designs.

A literature survey focused on the EHS system was conducted. Other related systems were examined only when they could contribute to the design of the EHS. The literature was found to contain little EHS-related engineering design information. It also became obvious that only two major EHS design approaches had been developed to any extent. They are being pursued by two companies that have been developing SEHPs with the assistance of DOE contracts. Visits were made to Sunpower, Inc., in Athens, Ohio, and Mechanical Technology, Inc., (MTI), in Latham, N.Y., to update the reviewer on the status of their SEHP programs.

A critical review was made of the recent reports and papers to identify areas that might yield cost savings in the EHS design. Unfortunately, the findings were not dramatic. For the conditions selected, the MTI design, based on a turbulent diffusion flame combustor, uses the right approach if preheating the combustion air to a high temperature is a priority. It was felt, however, that additional optimization could be performed, especially regarding the preheater. The use of an EHS based on a transpiration-cooled combustion system appears to be a good choice, although optimization and some experimentation are needed to develop this design further.

To evaluate the designs and design concepts, an analytical model was developed that tied the performance variables together. It allowed specification of the design conditions, as well as the component details. The model then calculated temperatures in various parts of the EHS together with the system efficiency. It also allowed the specification of excess air, exhaust gas recirculation (EGR), and combustion gas recirculation (CGR). It computed not only the effect of the preceding variables on efficiency, but also an NO_x emission factor. Although the model did not predict the absolute value of emission levels, it did allow a comparison between designs in terms of relative NO_x levels.

Because material costs play an important role in the EHS cost, American National Standards Institute (ANSI) appliance furnace codes were reviewed and material specifications were developed to comply with them. In addition, costs for materials of interest were obtained for large-quantity lots. The code also proved to be a source for emission limits, as well as for input on control strategies.

The EHS system was examined critically using the analytical model described earlier. It was determined that, in the case of MTI's EHS design, a small change in performance could result in a large cost savings because it dropped the material requirement (and cost). A control system using speed control was evaluated and found to be expensive. Two alternative concepts were developed which were less expensive.

The Sunpower, Inc., system was also evaluated, but its design was not as well developed as that of MTI. Sunpower was considering the pros and cons of the turbulent diffusion flame burner and a transpiration-cooled burner developed by General Electric (GE). The evaluation in this report assumed the use of the transpiration-cooled burner and its associated EHS system. The analysis assumed the use of an "MTI"-style preheater based on the finding that it is the most cost-effective design. Estimates showed the EHS based on a transpiration burner to cost roughly 70% of the EHS based on turbulent diffusion.

Finally, conclusions and recommendations are stated for this project. In the case of the MTI design, a smaller preheater with less expensive material is recommended. For the transpiration-cooled combustion system, some testing is recommended to define allowable levels of preheat and to measure emission levels. Pulsed combustion showed some technical promise but was not recommended because of the developmental risks and costs. The need for a more complete cost tradeoff comparison of EHS designs for diffusion burners and transpiration burners is apparent, but it is beyond the scope of this work. Such a comparison would need to include design and cost changes for the heater head and the Stirling engine to be accurate and meaningful.

2. STATE-OF-THE-ART REVIEW

2.1 LITERATURE SEARCH

The literature search started with an evaluation of the reports on hand in AMTI's library. This search resulted in the collection of over 150 volumes that included papers, reports, and conference proceedings containing information on Stirling systems. An open literature data search also was conducted using a computerized data base search service.

Literature searches have the shortcoming of depending on published reports that have progressed into services such as the National Technical Information Services (NTIS). Thus access to reports of recent and current research is limited. Because of this limitation, certain recent research publications were obtained from Oak Ridge National Laboratory (ORNL), including task reports and recent references that had not reached the available data bases.

In addition to publications that concentrated on heat pumps and natural-gas-fired engines, some of the more notable references from the U.S. Department of Energy (DOE) automotive program were included. They include reports concerning the P-40 engine from United Stirling, the MOD I and MOD II from MTI, and the 4-215 from Ford Motor Company.

The references were screened to eliminate those that contained no information concerning EHSs. The remaining reports were scrutinized to find those with relevant information.

The main body of literature was found to lack any significant engineering design information. While there appear to be only a few basic designs or variations of the EHS system, even these are not well documented. A few references included items such as firing rate, heater head temperature, and efficiency; and a few others included gas flow rates and temperatures. Construction details such as heat transfer areas or component dimensions were given in a few rare instances.

It was concluded that two basic designs for combustion systems and two or three relevant designs of preheaters were applicable. These systems were designed primarily under DOE programs, and the design information was obtained by site visits to the developers.

2.2 REVIEW OF RELEVANT STIRLING ENGINE WORK

2.2.1 Basic Stirling Background

The basic Stirling engine on which much subsequent development is based is the P-40 system designed by United Stirling (Fig. 2.1).[1] This unit is rated for 40 kW and uses an EHS similar in many ways to the system currently being used by MTI.[2,3] The P-40 EHS fits over a tubular heater head so that the inside diameter of the heater head forms the combustion chamber (Fig. 2.1). As the P-40 uses EGR for emission control, air and exhaust gas enters a plate fin preheater. There it recuperates heat from the exhaust gas

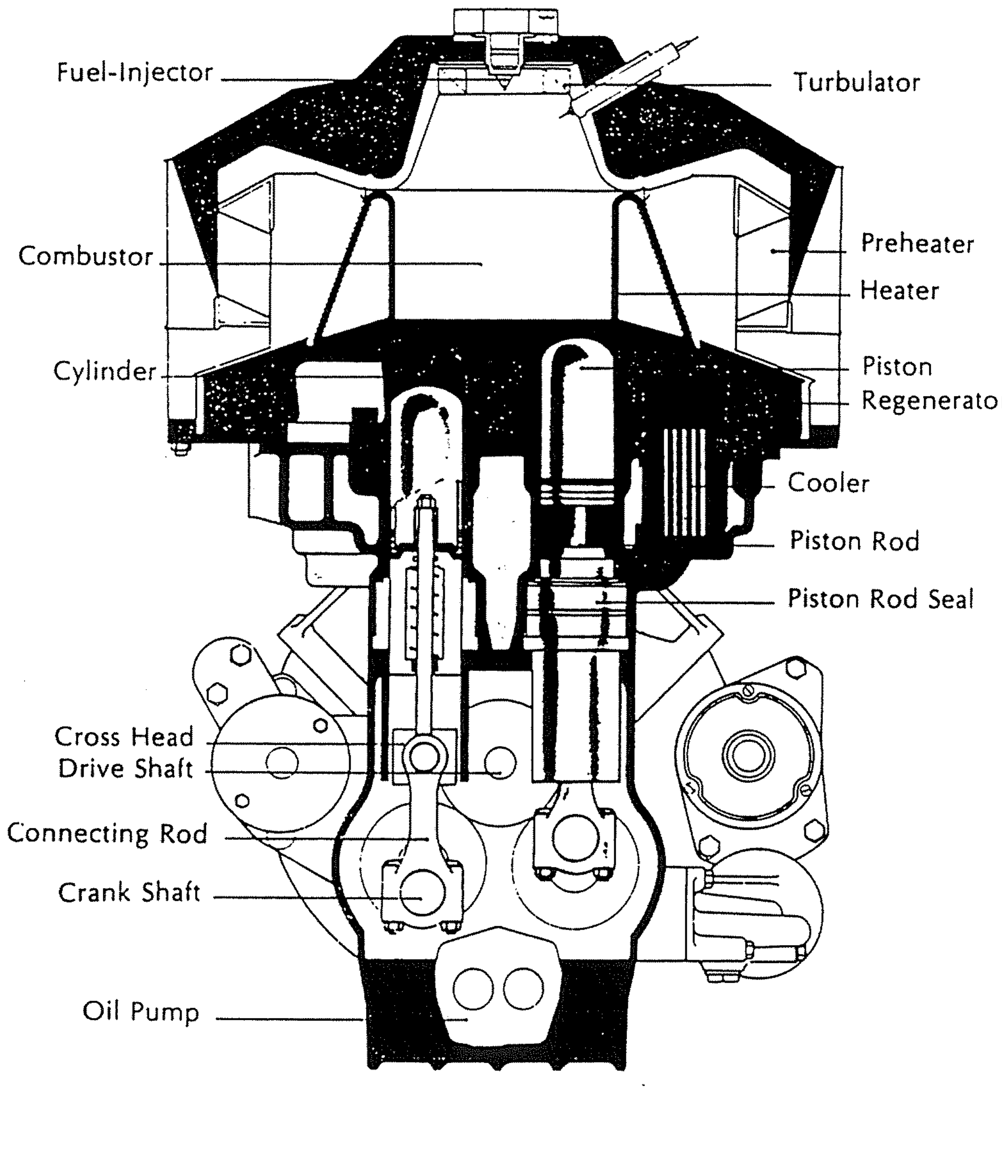


Fig. 2.1. Schematic of the P-40 Stirling engine.

that is passing counterflow to the air/exhaust gas mixture. The air/exhaust mixture passes into a radial inflow turbulator. A fuel nozzle is located at the center of the turbulator where the air/exhaust mixture and the fuel mix before ignition.

The combustor uses a turbulent diffusion flame, a type chosen for compact liquid-fired burners. The plate-fin preheater consists of a great number of thin corrugated stainless steel plates welded together to form a plate-fin style heat exchanger that provides a compact, efficient heat transfer surface. The channels are continued above and below the core heat transfer surface to allow the formation of inlet and outlet plenums for ducting air and exhaust gas into and out of the preheater.

The first major evolution of the MTI Stirling EHS is shown in Fig. 2.2, taken from ref. 4. The major differences between the P-40 and the MTI MOD I are in the ducting of the air delivery system and the use of CGR instead of EGR for NO_x control. In this configuration, air is ducted through a shroud around the EHS assembly, keeping thermal losses to a minimum. The air then passes through the preheater and then through nozzles that induce exhaust gas from the outlet of the heater head tubing into the air stream.

The choice between EGR and CGR involves a trade-off between implementation and effectiveness. CGR and EGR both lower the maximum flame temperature and consequently the formation of NO_x. CGR, requiring a higher temperature than EGR, is less effective. On the other hand, CGR is easier to implement.

The next step in MTI's development was the MOD II EHS design shown in Fig. 2.3. A good description of this configuration can be found in ref. 5. The figure shows the redesign of the CGR induction system using a number of nozzles at the throat of "venturi" tubes rather than the vanes used in the MOD I design. An attempt was made to use both a metal and a ceramic preheater in the design. The ceramic preheater apparently never developed into a full system design, but rather was subjected to component testing. The drive to develop the ceramic preheater was due to the expense of constructing the multi-plate preheater with its myriad welds and difficult fabrication.

The last chapter in the MTI four-cylinder engine development is shown in Fig. 2.4. This is basically the MOD II engine with natural gas, rather than liquid fuels, as the fuel. The application for this configuration was a 100-ton engine-driven chiller.[6]

This short synopsis summarizes EHS development in the DOE automotive program from 1979 to 1990 (refs. 7 and 8).

2.2.2 Heat Pump External Heat Systems

Eight companies were found to be performing research on or developing Stirling-driven heat pump systems (Table 2.1). Collectively, they have been performing research from 1975 (ref. 12) to the present (refs 7, 16).

In the United States, it appears that MTI and Sunpower, Inc., are the two principal companies working on residential-size SEHPs. Recent references indicate activity at Stirling Thermal Motors and Stirling Power Systems, but the level is not known. Current activity (nonpublished) in Japan was not determined in this study, but that is not an indication that research is not being pursued vigorously.

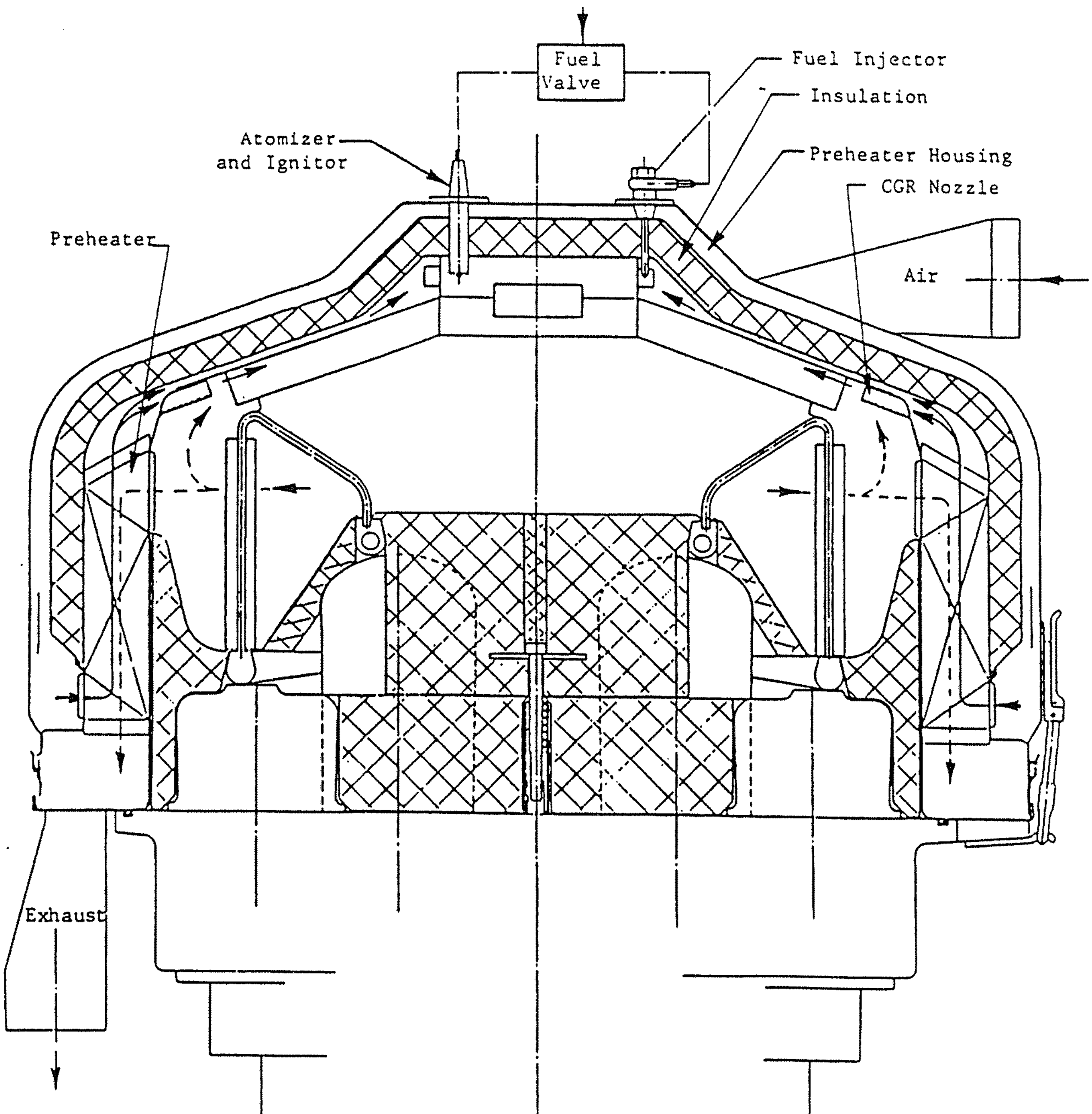


Fig. 2.2. MOD I—external heat system.

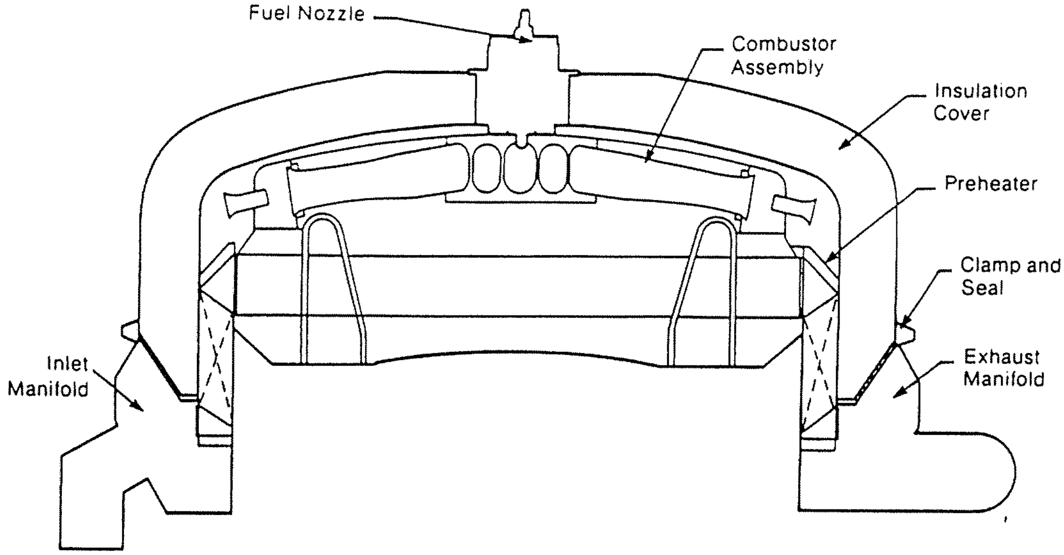


Fig. 2.3. MOD II-external heat system.

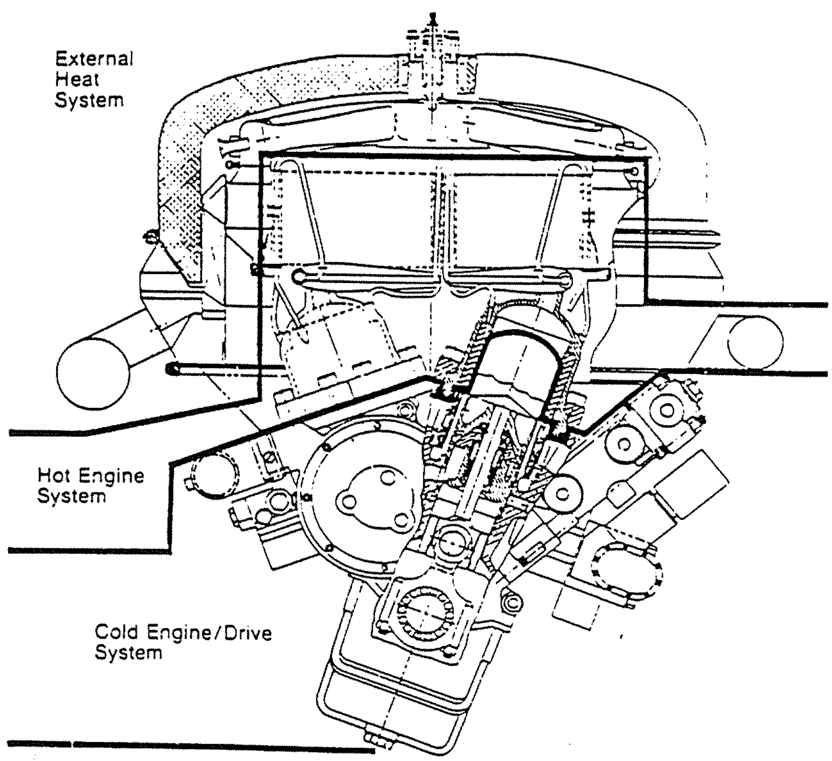


Fig. 2.4. Cross-section of MOD II engine.

Table 2.1 Heat pump system developers

Organization	References
General Electric	9–12
Mitsubishi	13–15
Mechanical Technology, Inc.	2, 3, 7, 16–19
Phillips Research Laboratory	20
Stirling Power Systems	21, 22
Stirling Thermal Motors	23
Sunpower, Inc.	24–26
Toshiba	13, 14, 27

Mechanical Technology Incorporated. As with the Stirling engine automotive program, MTI has probably been the designer most heavily supported by DOE in the development of Stirling engine heat-actuated heat pumps. The most recent configuration of the basic MTI EHS system is shown in Fig. 2.5 (ref. 2). Some of the earlier versions are described in refs. 3, 7, and 17–19.

The EHS system shown in Fig. 2.5 is centered around a monolithic heater head constructed of a high-temperature alloy. The combustion system uses EGR for emission control. Air and exhaust gas pass through the plate-fin preheater and then around the combustion chamber shroud. The burner is a basic turbulent diffusion flame type, firing onto the heater head.

An EHS designed to accommodate a finned tubular heater head rather than a monolithic head is shown in Fig. 2.6. Such a unit has not been built or tested at this time, although liquid fuel burners and tubed heater heads were tested for automotive Stirling engine applications. It should be noted that the combustion volume with the tubular head appears to be smaller than the design using the monolithic head.

The preheater design for the EHS units shown in Figs. 2.5 and 2.6 is depicted in Fig. 2.7. This preheater design differs from the plate-fin arrangements of the past. The construction of this unit eliminates the welded construction of earlier units. The matrix consists of folded sheets of metal that form the walls of the flow channels. The inner and outer shells complete the channels. The ends of the matrix are folded over, and an end piece is placed over the folded ends. Thus flow plenums are conveniently formed for the air and exhaust gas by the placement of the shells (Figs. 2.6 and 2.7). While this design can be efficient in terms of execution, it can require a great deal of expensive material because of its thermal design point.

The type of combustor design selected by MTI is the logical choice for its design conditions. The high degree of air preheating requires the use of a turbulent diffusion flame to avoid flashback and other associated problems (for premixed burners).

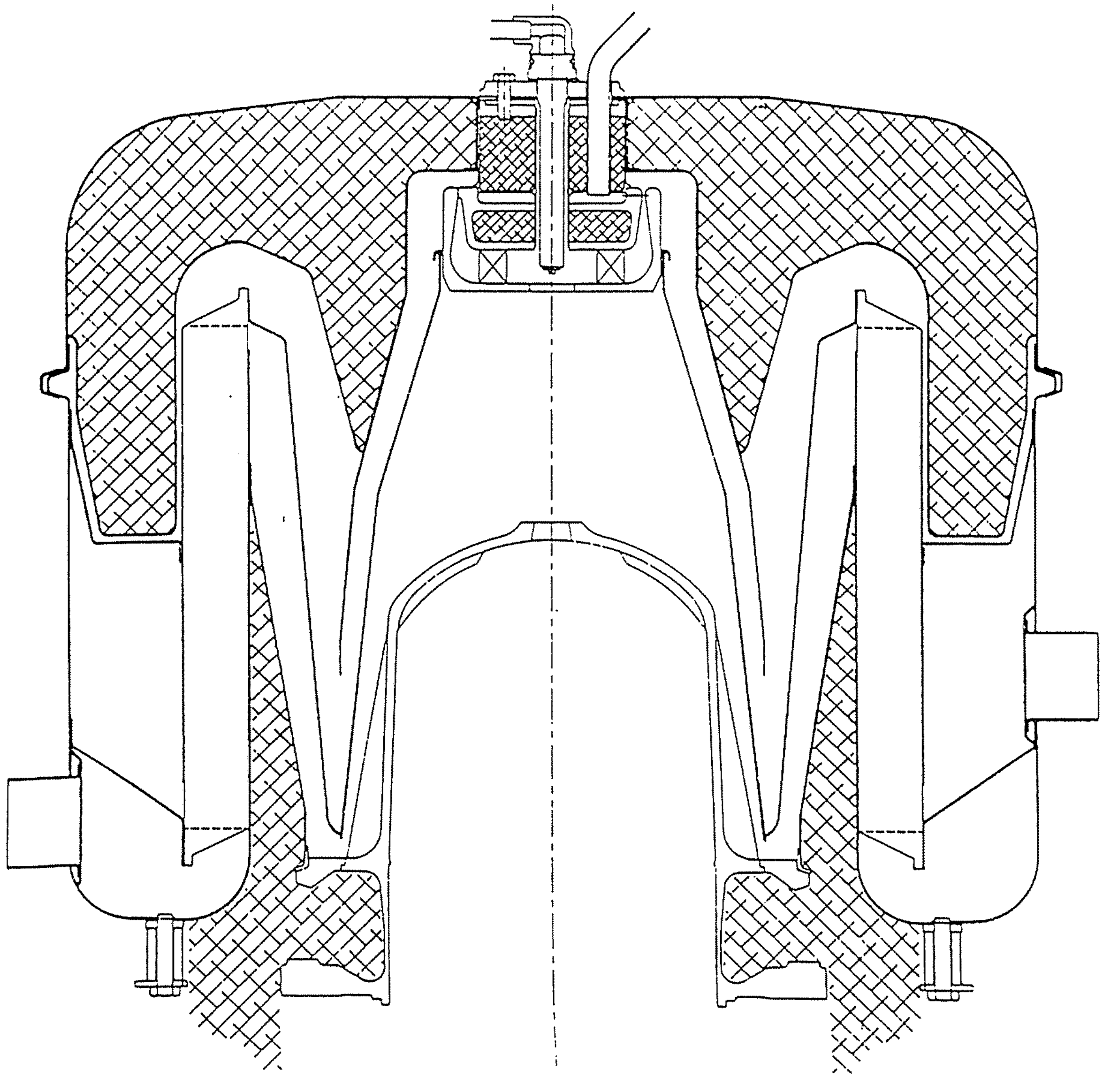


Fig. 2.5. Mechanical Technology, Inc., external heat system for monolithic head.

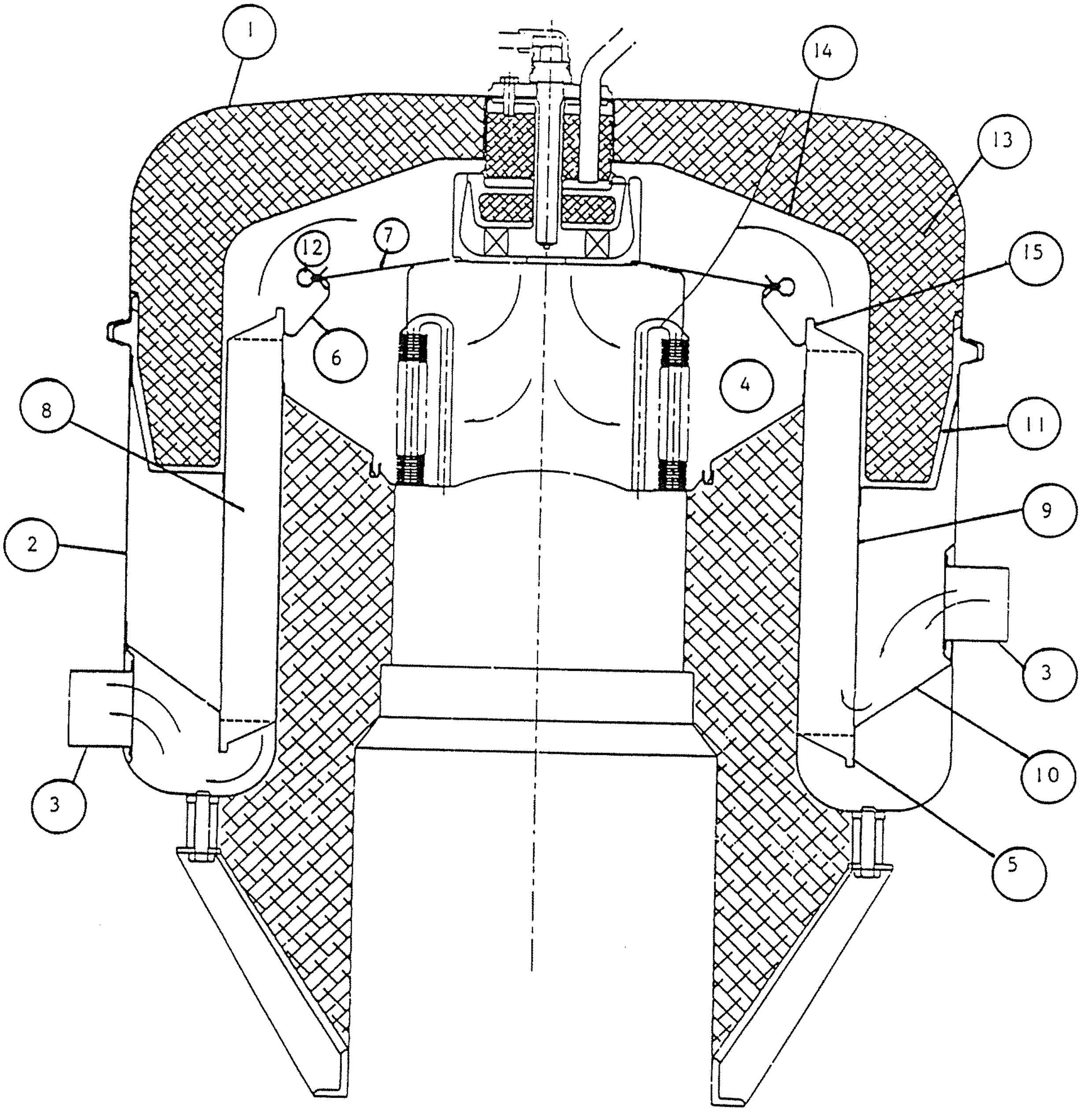


Fig. 2.6. Mechanical Technology, Inc., external heat system with tubular heater head.

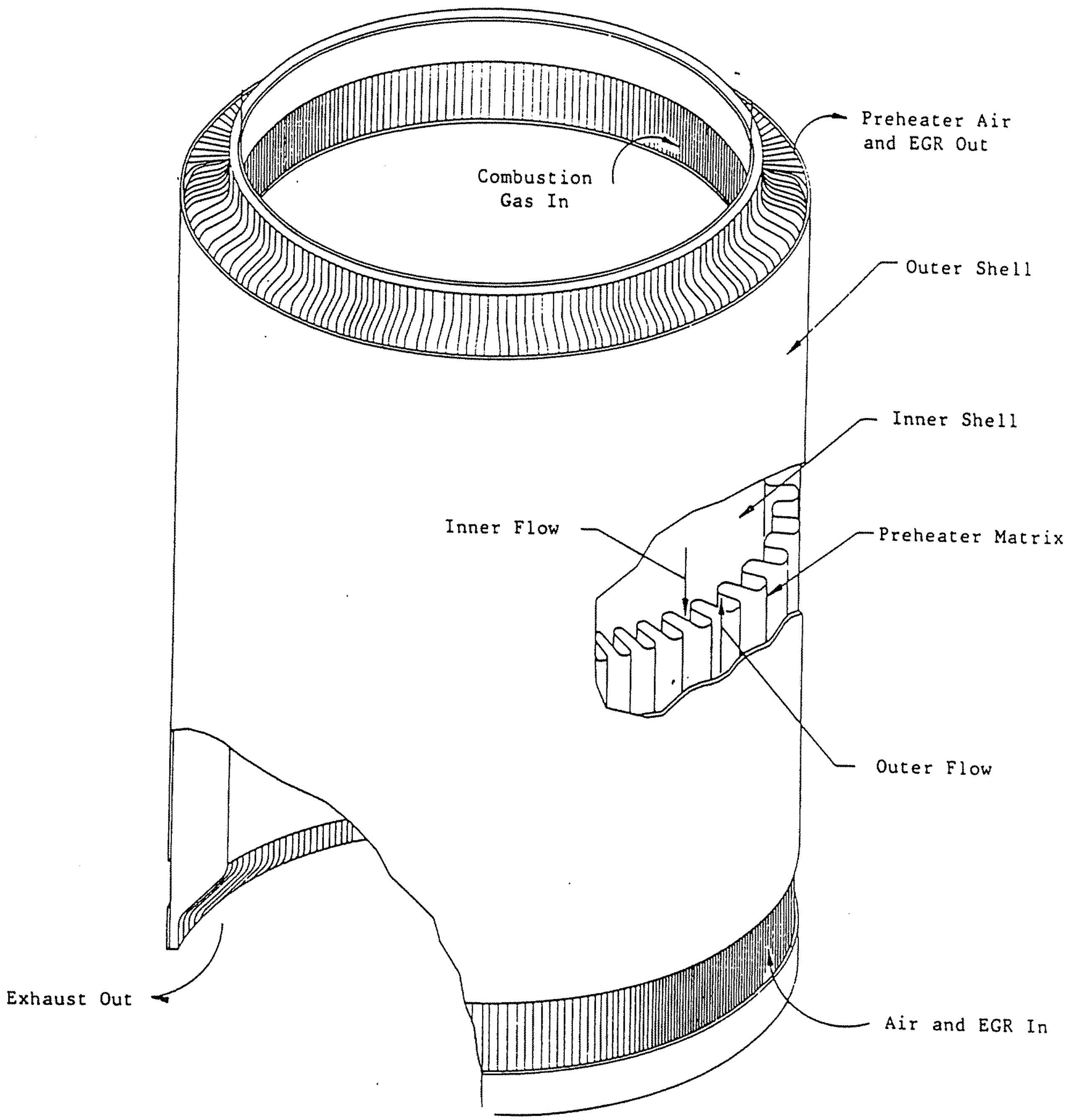


Fig. 2.7. Mechanical Technology, Inc., preheater design.

The control system shown for this system is reproduced in Fig. 2.8 (ref. 2). This is a development system; therefore, it cannot be concluded that it will be the production system. It is apparent, however, that for the combustion system selected, active and separate air and fuel controls are required and are processed by a main engine control. If they are varied uniformly, the air/fuel ratio control will have to be very accurate, with one slaving off the other. While the electronics may be inexpensive, the flow measurement and control elements will be expensive. Again, this is not the production system, so its execution to a production design remains to be seen. Control strategies will be evaluated in Sect. 4.

General Electric. Although GE is no longer pursuing gas-fired SEHP, its approach is of interest because it was different from that used by United Stirling and MTI. The GE heat pump system EHS featured a transpiration burner for combustion, rather than a turbulent diffusion flame burner. A transpiration burner is shown schematically in Fig. 2.9. Sunpower is contemplating the use of a similar system for its design.

The transpiration burner has many advantages over the diffusion flame burner for the combustion of natural gas. First, the transpiration burner uses premixed air and fuel, a technique that helps ensure combustion of a well-mixed air/gas stream with subsequent low emissions. If the burner is operated at elevated temperatures, some radiative heat transfer can take place, lowering the requirement for heater head convective heat transfer. Another positive feature of the transpiration burner is that gas valves and controls have been developed for other appliance applications and the design of the control system is relatively inexpensive. The drawback is that this type of burner cannot tolerate high-temperature preheating because of pre-ignition of the fuel before it reaches the burner surface. In the GE design, the air preheat temperature was limited to 800°F.

Sunpower, Inc. Though Sunpower is a major developer of Stirling heat pump systems, its EHS design is not as advanced as the work at MTI. A heat pump assembly designed by Sunpower is shown in Fig. 2.10 (ref. 25). It includes an engine/magnetic coupling/compressor assembly. The heater head is a tubular type, heated by a transpiration burner (Fig 2.9). Sunpower has also considered using a heat pipe system, but because of its prohibitive cost, it will not be described in this report. An example of this system can be found in ref. 28.

In a recent visit to Sunpower, it was learned that an EHS similar in design to the one developed by GE is being contemplated for the heat pump system EHS.[24] The preheater Sunpower intends to use is in many ways similar to the MTI unit in basic configuration (Fig. 2.11). It was not developed to the point that the detailed design could be evaluated.

Phillips Research Laboratory. A 3-kW Stirling heat pump system design has been reported by Phillips Research Laboratory.[20, 29] This design features a diffusion flame burner similar to those in previously discussed designs, but it has a different heater head design. The heater head consists of U-formed multimet tubes presenting a double row to the combustion products. The first tube is unfinned, and the second row has a longitudinal strip of stainless steel welded to the tube. The gap of adjacent fins forms the flow path for heat transfer. It appears that this design might result in a high pressure drop or require a low heater head temperature. A dense fin arrangement might be a better approach.

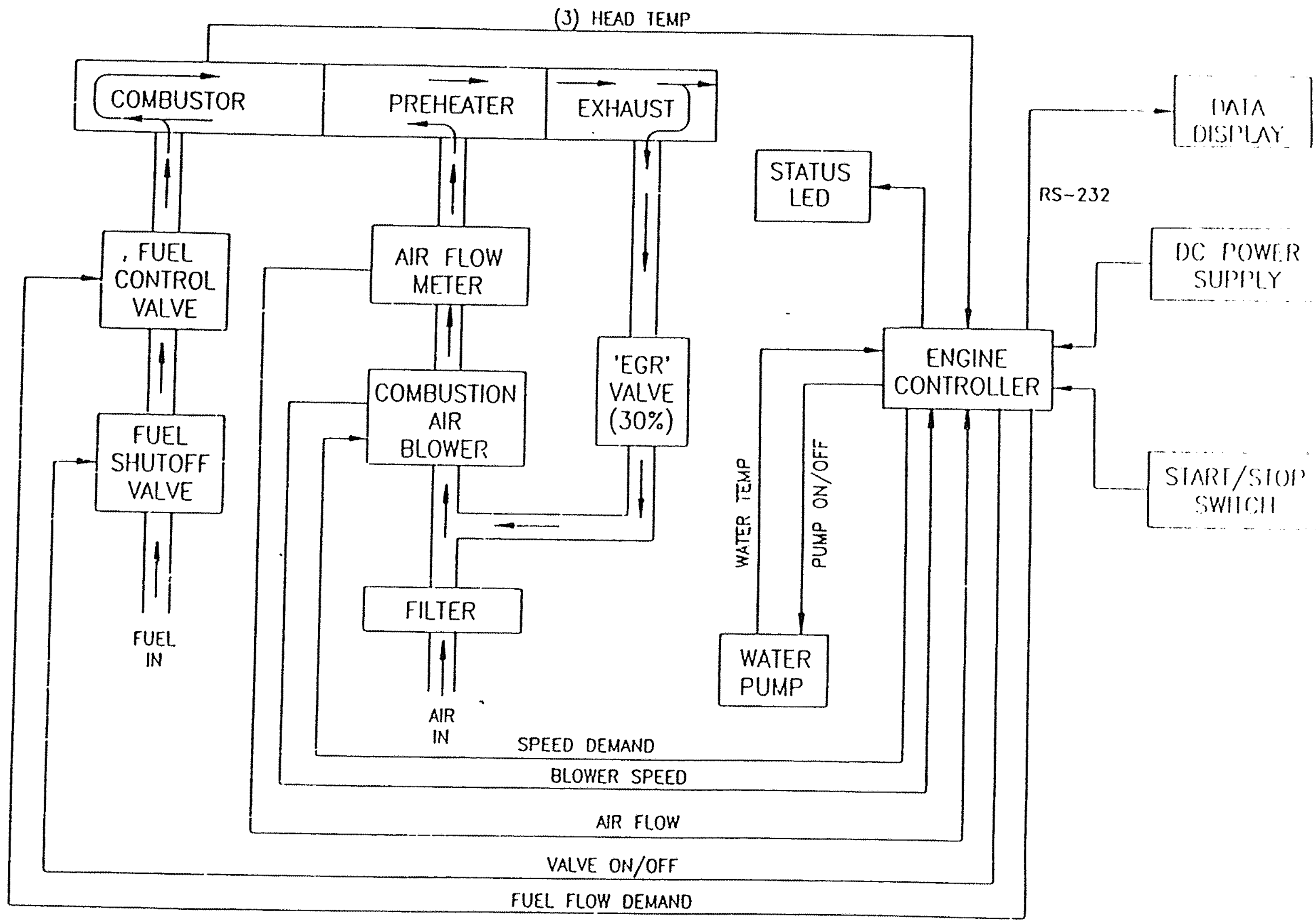


Fig. 2.8. Mechanical Technology, Inc., Stirling engine heat pump control system.

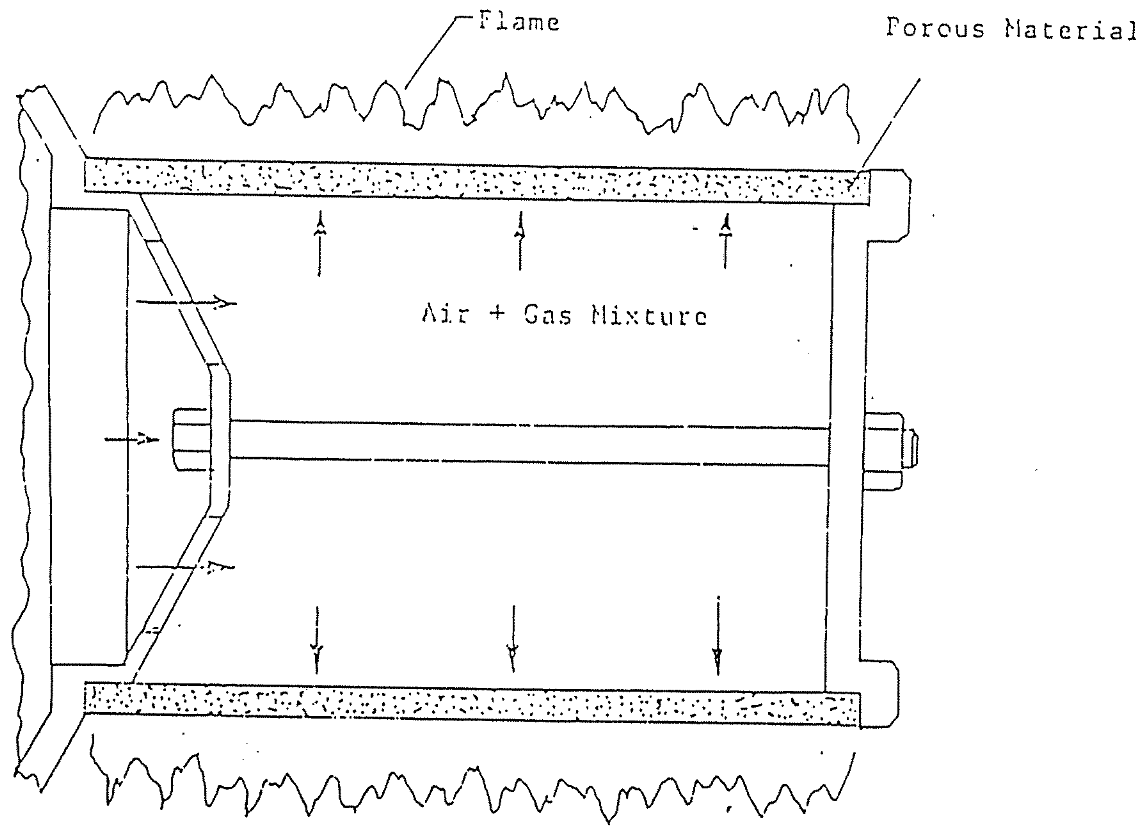


Fig. 2.9. Transpiration burner schematic.

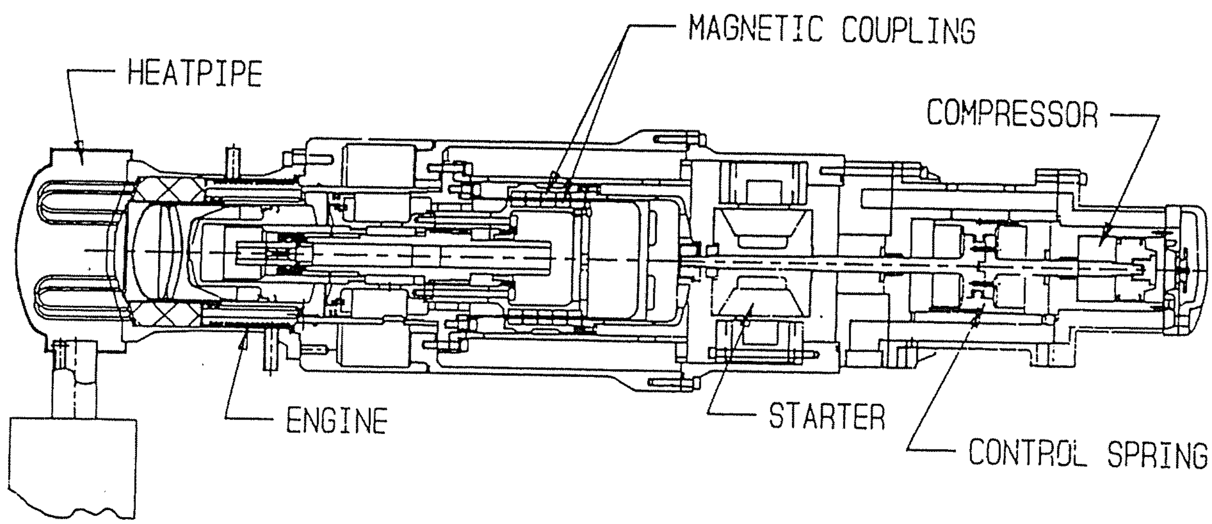


Fig. 2.10. Sunpower heat pump with heatpipe external heat system.

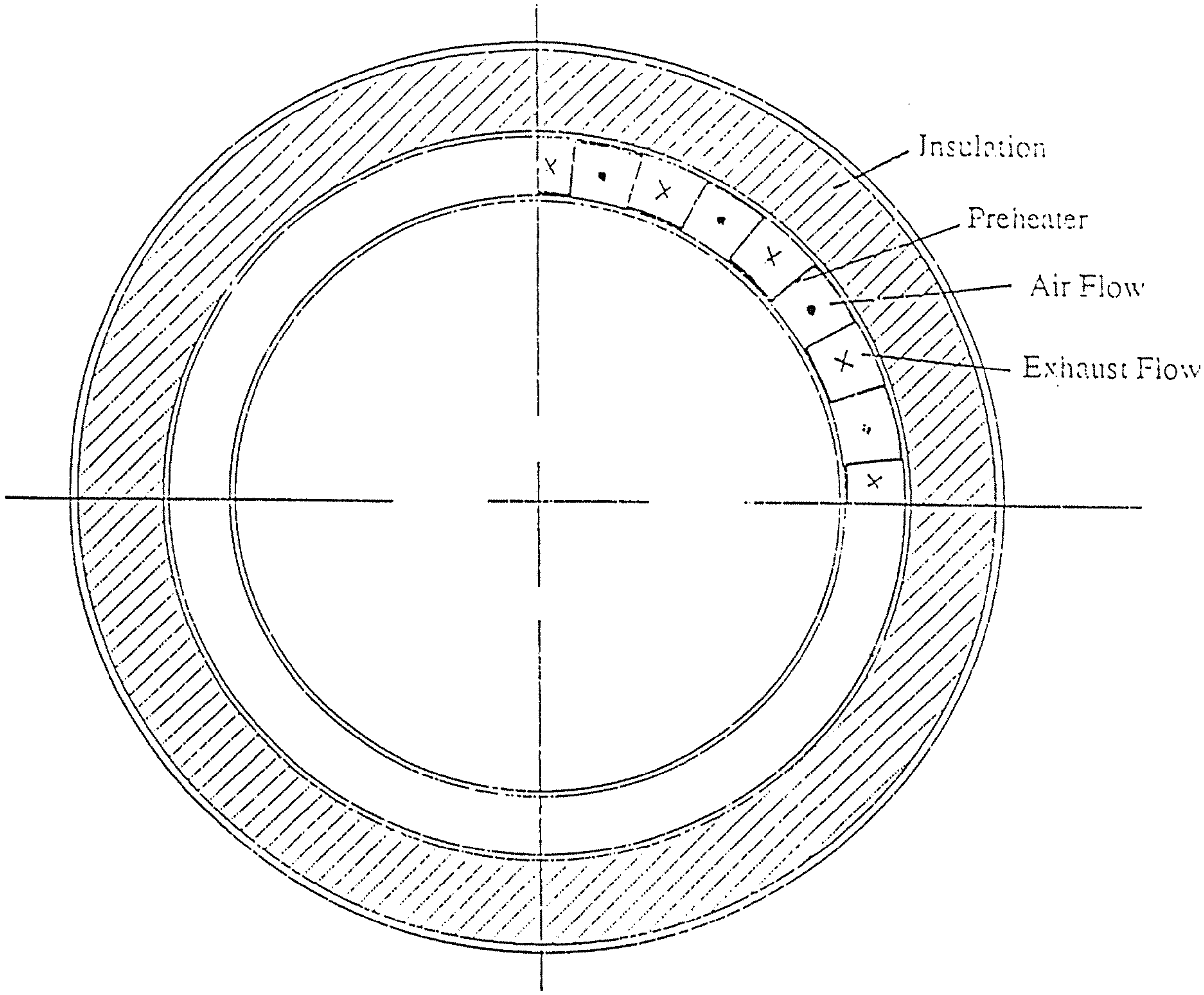


Fig. 2.11. Annular air preheater.

The preheater design appears similar to the MTI design (see Fig. 2.7). It consists of a 0.15-mm stainless steel sheet folded in accordion form. Sealing was done by pinching adjacent flat parts and clinching them with a staple. The reference did not show the technique, but based on the description, it apparently depends on a mechanical means of sealing as does the MTI design, rather than extensive welding. Little else of interest regarding EHS design was reported.

Japanese Development: Toshiba and Mitsubishi. Toshiba and Mitsubishi have also done development work on 3-kW natural-gas–fired heat pumps).[13-15, 27] The designs are designated as NS03M (Mitsubishi) and NS03T (Toshiba), and both use diffusion flame burners and tubular heater heads. The Toshiba unit uses a star-shaped finned tube arrangement, which appears to be a coaxial flow design that uses annuli to direct the working fluid out and back along the finned section of the tube. Its air preheater is listed as a plate-fin type, but no details are provided. No significant information was obtained about the Mitsubishi heater head or air preheater.

Stirling Thermal Motors. Some natural-gas-burning Stirling engine design work has been done by Stirling Thermal Motors, as evidenced by a patent (ref. 23). that describes a multi-piston engine design with each cylinder having its own burner. The method of heat transfer to the heater head is not clear, although it appears that the exhaust from all the heaters may be collected and passed through a common preheater. The type of preheater was not shown. Stirling Thermal Motors has also done development work concerning a liquid-fuel-fired 40-kW Stirling engine that features a heat pipe system for the heater head (ref. 28).

Stirling Power Systems. Stirling Power Systems has done development work on a 7.5-kW kinematic type Stirling engine system.[21, 22, 30] Included is a heater head design that differs from the standard one most developers have adopted. The design presented employed a "flat" tube heater head design and appears to use a turbulent diffusion flame burner. No details of the preheater were found. The arrangement is unusual in that it is strung out in a linear fashion with little attempt at packaging. The MTI and Sunpower systems are much more compact, efficient designs.

2.3 FINDINGS AND CONCLUSIONS

The significant findings of the literature review, communications, and site visits are explored in this section for relevance to SEHP EHS development and optimization work.

2.3.1 Combustion System

Perhaps the dominant EHS design issue is the requirement for high-temperature preheated air. This is a direct consequence of the need for a preheater to keep the system efficiency at reasonable levels. Design specifications for air preheat temperature are seen to vary from 427 to 760°C (800 to 1400°F). Furthermore, the heater head temperature requirements can affect preheat requirements. The heater head design temperature variation found in the literature ranged from 600 to 800°C (1112 to 1472°F).

The selection of a high heater head and preheated air temperature dictates the selection of a turbulent diffusion burner (the temperature is too high for premixed systems). In this type of system, the air and fuel must be mixed quickly, just at the point of ignition. High velocities are required, which can result in high pressure drops. In addition to the effect on burner selection, the effect on other parameters such as emissions is great. For a diffusion flame burner, every 200°F increase in preheat temperature will result in a roughly 75% increase in NO_x level. The use of 800 to 1200°F temperatures without EGR or CGR will result in NO_x emissions in the 200- to 1200-ppm range.

The common solution cited to the NO_x emission problem is the use of EGR or CGR. The effect of EGR on NO_x is shown in Fig. 2.12, based on data produced by MTI (refs. 2 and 16). It can be seen that EGR rates above 20% cause NO_x emissions to drop below 100 ppm. Although the drop shows that emissions due to high preheat can be controlled, keep in mind that this control requires excess flow and pressure drop in the combustor and more elaborate hardware requirements.

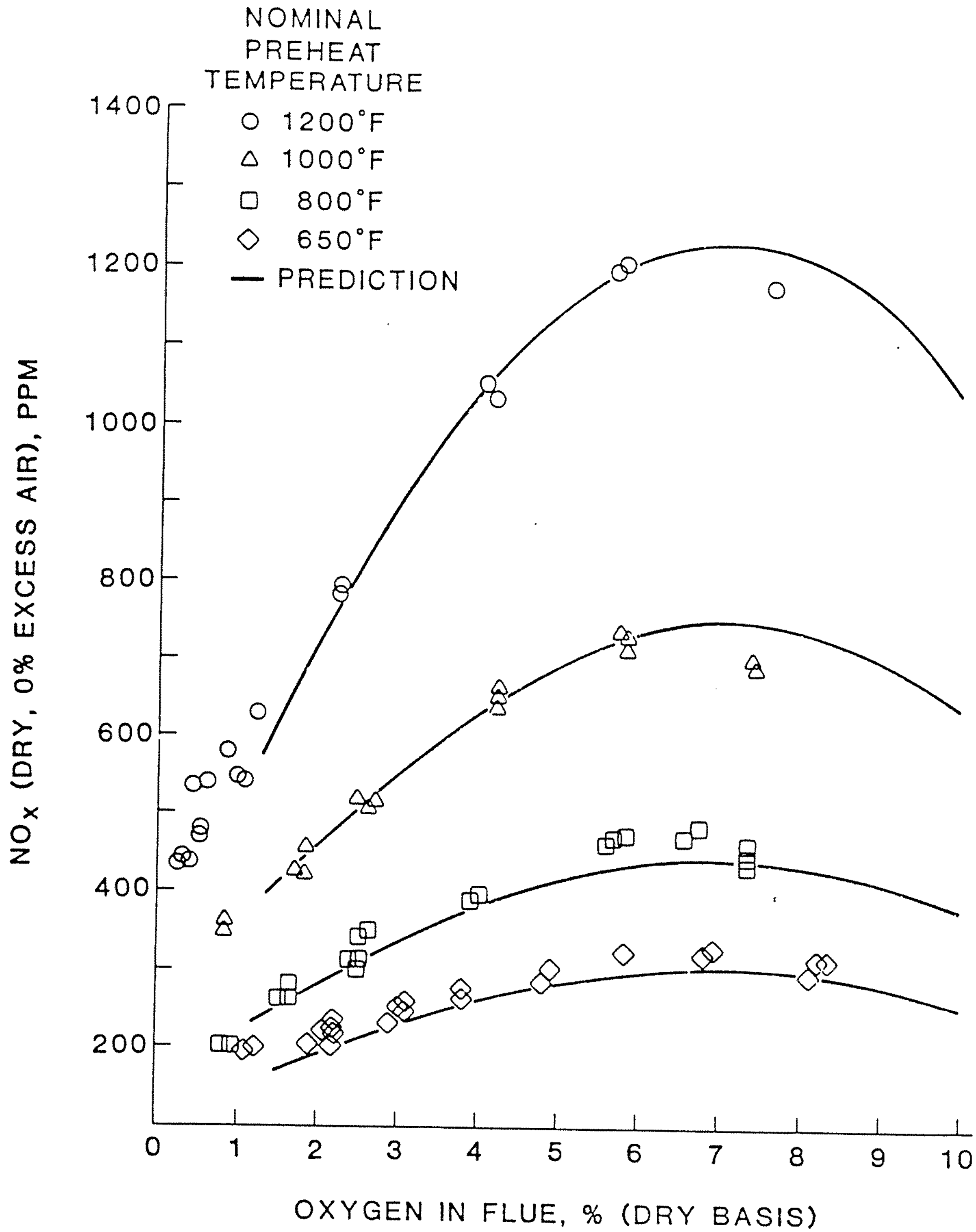


Fig. 2.12. The effect of exhaust gas recirculation on NO_x.

At the other end of the spectrum, low preheat and heater head temperatures, using a pre-mixed combustion system like a transpiration burner, could also be a wise choice. This system can be as compact as a turbulent diffusion flame burner while offering additional benefits. Emission levels typically are low, and very low NO_x emissions have been achieved with premixed radiant type burners. In addition, off-the-shelf control components exist for some transpiration burners.

It is not clear what type of combustion system would be best. There appear to be two main alternatives: the pre-mixed transpiration burner and the turbulent diffusion burner. The transpiration burner is seen as the low capital investment choice, but it results in significantly lower Stirling engine efficiency. At best, it is felt that design conditions can be reexamined to ensure that optimum values have been selected. Further evaluation of these issues is presented in Sect. 4.

2.3.2 Preheater Design

At this time, the plate-fin preheater appears to be the best choice among the candidates existing in the literature. The requirement for a compact design with a high ratio of surface area to volume alone tends to make it appear the most logical choice. The execution of the plate-fin approach found in the literature varied from an expensive fabrication with many plates welded together to form a heat exchanger, to the approach used by MTI [2] or Phillips [20] that avoids extensive welding and uses a mechanical means of sealing. The latter appears to be the best approach. Although a plate-fin unit may be the best choice, it is important to optimize the design so that material is conserved.

The concept of a ceramic preheater has been familiar for some time. It has been investigated seriously by both MTI and Ford Motor Company in the Stirling automotive program. MTI reported a potential savings of a factor of four if a ceramic preheater were used instead of the extensively welded plate-fin design MTI had used in the MOD I design.[31] It assumed quantities of 300,000 units per year for the analysis. In spite of this incentive and the resources available to that program, problems were still being discussed 4 years later. [32] The major problem appeared to be the inability of the ceramic to withstand the thermal cycling. Ford Motor Company also used a rotary ceramic preheater in its 4-215 automotive Stirling engine. After this preheater core had run for 20 hours, a crack started on the inside diameter and propagated outward. The application of ceramic preheaters to the SEHP is judged to be unworkable at this time.

2.3.3 Other Components

Because of a general lack of specific engineering design information in the available literature, little was found concerning other EHS components such as control systems, fans or blowers, ignitors, valves, and other auxiliaries.

3. CONCEPTS FOR IMPROVED PERFORMANCE

The intent of this section is to review new concepts, designs, components, and systems that could improve the performance and/or cost effectiveness of EHS designs. The new approaches came from suggestions from those involved in SEHP work at ORNL, the review of the state of the art (Sect. 2) and concepts that evolved during the course of this study. The following concepts are from the original statement of work:

- ceramic rotary preheater,
- combustion gas recirculation,)
- improved combustor designs,
- simplified and/or off-the-shelf controls,
- combustion chamber cooling, and
- improved preheater approaches.

As evident in Sect. 2, MTI and Sunpower, Inc. are the only two SEHP developers active in this country at the writing of this report. They selected different approaches to their EHS systems that are reflected in their divergent designs. Early analysis revealed that these two approaches were both technically sound. In fact, an analysis of the two approaches revealed no middle design path or completely different approach that appeared viable. The major goal of this work, finding or developing more cost-effective EHSs, remained unchanged.

Note that one criterion of the evaluation was to avoid concepts that need extensive development or require significant breakthroughs before becoming available. Where it was felt that the potential gains justified some development risk, the concept was included but the risks or unknowns were evaluated.

3.1 PREHEATER DESIGNS

The design of the preheater is critical to the cost effectiveness of the design. Three types of preheaters were identified as having have been used in the past:

- plate-fin,
- rotary ceramic, and
- stationary ceramic.

3.1.1 Ceramic preheaters

One of the approaches that appeared to have merit at the beginning of the program was ceramic preheaters. The lure of ceramics is that the raw material used in their manufacture is inherently inexpensive. Thus products that use ceramic materials may ultimately be inexpensive. The problem is that unless large (automotive market) volumes are involved, the processing costs to produce acceptable products at a low cost are not recovered. MTI, in a development program with Coors, tried to develop a fixed-bed type of ceramic preheater.[31] The project was unsuccessful. At a recent Automotive Technology Development Contractors' coordination meeting, many papers were presented on the state of the art of ceramic processing and fabrication. There was nothing to indicate new developments in the area of ceramics or their processing that would make them readily applicable to the SEHP EHS.

A rotary preheater using ceramics might be usable because it could use existing, available ceramic matrix materials. In such a design, a ceramic wheel would alternately be heated by the exhaust products and then cooled by incoming combustion air. The wheel sector area exposed to gas and air would be balanced against the speed of the wheel to obtain optimum heat transfer. This approach has several drawbacks. The first problem would be sealing the rotating wheel and the drive in a high-temperature environment. Doing so reliably over a 20,000-hour life would be difficult, if not very expensive. The second problem would be the durability of the ceramic: the thermal cycling as it heated and cooled would be a demanding application. Ford Motor Company found the thermal cycling to be a problem in its Stirling engine development program. [33] Ford researchers found that cracks developed almost immediately in their rotary preheaters (see Sect. 2). Another minor consideration would be the additional power requirement of a motor to turn the preheater.

These drawbacks led to the conclusion that ceramics are not a good choice for the preheater. Although this approach offers potential low costs, it was felt the development path was beyond the scope of the current DOE heat pump program.

3.1.2 Plate-fin Preheaters

Although a number of other surface types exist, the packing density of the plate-fin heat exchanger makes it one of the most compact designs possible. Its costs can run the gamut from expensive and exotic to inexpensive. The materials and construction techniques used account for the cost differences. Another low-cost alternative is a finned tube similar to those used in air conditioning systems. These, while inexpensive, are more suited to gas-liquid heat exchange, and the preheater of concern here is a gas-to-gas application.

Once it is decided that a plate-fin heat exchanger is the best approach for this program, then the goal is optimizing the design for cost effectiveness. The major design elements to consider are construction, size, and material selection. The construction technique selected by MTI (see Fig. 2.7) was judged to be well developed and probably the lowest cost approach. [2, 34] It uses an accordion-shaped core trapped between an inner and outer annular shroud. Sealing is accomplished by bending over the ends and capping them with a ring that is welded or crimped closed.

Although this design can be efficient, the size must be optimized to avoid using a great deal of expensive material. For the design shown in Fig. 2.7, the preheater takes exhaust gas at 1462°F and uses it to heat the incoming air (which may be mixed with recirculated exhaust gas) to 1365°F. The preheater effectiveness for this set of conditions is about 0.930, good from a performance viewpoint but probably too high for optimum cost effectiveness. For high thermal effectiveness values, each increment in efficiency gain comes at an accelerating increase in area. This effect can be seen in Fig. 3.1, which shows the EHS thermal efficiency as a function of the number of transfer units (NTU), the quantity related to heat transfer area. This quantity is defined as

$$NTU = AU/C_m \ , \tag{1}$$

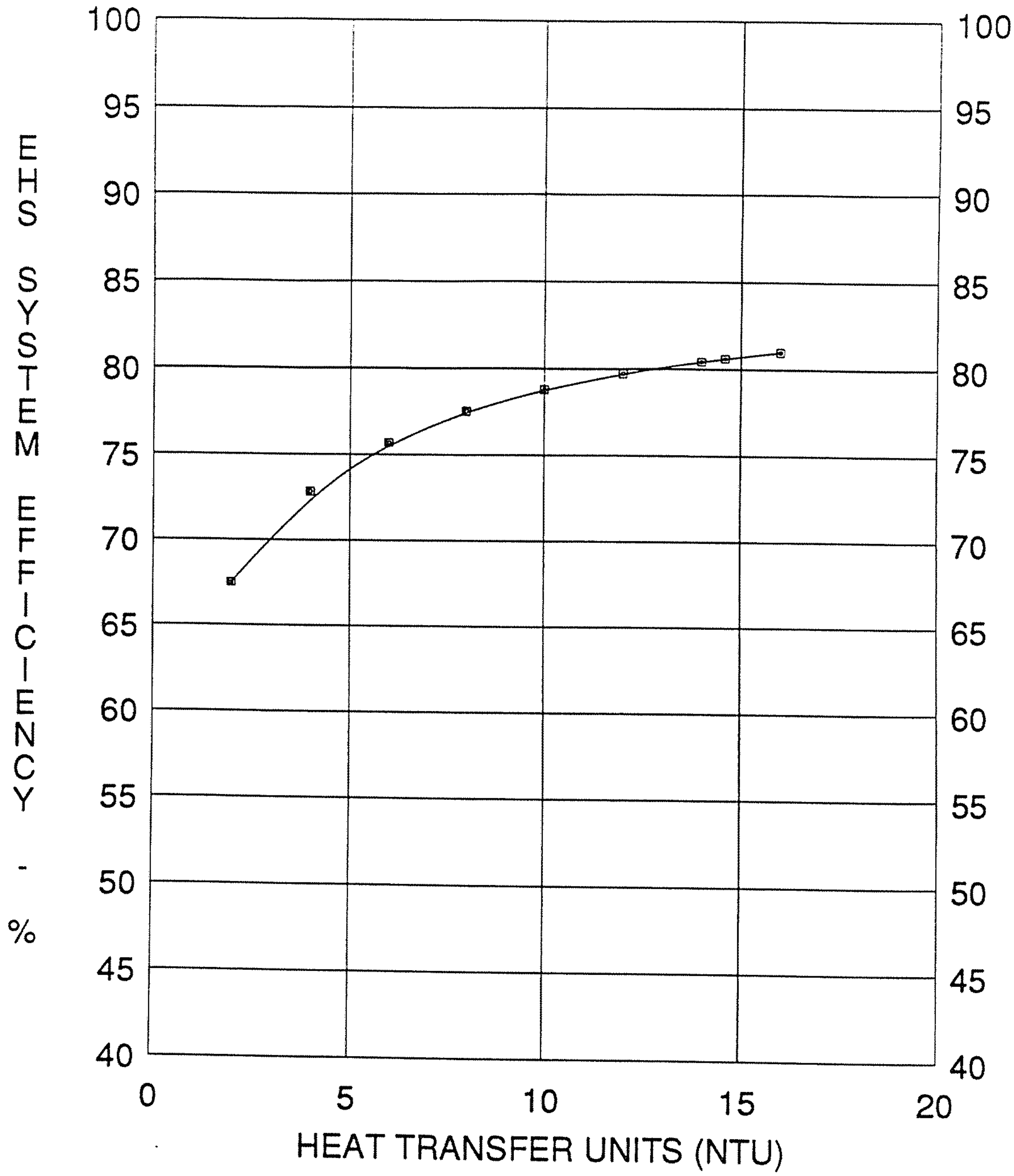


Fig. 3.1. The effect of preheater size on external heat system efficiency.

where

A is heat transfer area,
U is heat transfer coefficient,
C_m is the product of flow and specific heat.

Thus if the flow rate and heat exchanger cross-sectional geometry are kept constant, then the NTU term is directly proportional to area. This can be related to the heat exchanger effectiveness as follows:

$$NTU = \frac{1}{1 - EFF}, \tag{2}$$

where

$$EFF = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}},$$

T_{co} is air temperature exiting the preheater,
T_{ci} is air temperature entering the preheater,
T_{hi} is exhaust gas temperature entering the preheater.

With these equations, preheater size (NTU) can be related to effectiveness (EFF). This relationship will be used together with material costs in Sect. 4. to perform a cost effectiveness evaluation of the EHS system.

In addition to the cost in terms of heat transfer area, some additional secondary savings may occur from a reduced preheater size. The pressure drop of the system will decrease, allowing the use of a smaller blower and lowering the blower power cost. Also, lower preheat temperatures will lower the flame temperature, reduce emissions (or reduce EGR levels), and make material requirements less exotic. Further analysis to optimize the preheater is presented in Sect. 4.

3.2 COMBUSTION SYSTEMS

3.2.1 Combustor Type

The predominant type of combustor used for Stirling engine systems has been the turbulent diffusion flame type. This choice stems from necessity rather than from design or cost reasons. The Stirling cycle requires the use of high temperatures (600 to 700°C) in the heater head to achieve high engine thermal efficiency. These high temperatures in turn require high exhaust temperatures from the heater head and potentially high stack losses. To avoid the high stack losses, the combustion air is preheated with the exhaust products before they are discharged. This heat recovery process is good from the standpoint of efficiency; however, it has two negative impacts on the design of the combustor. (1) The temperature of the combustion air entering the burner can vary from 800 to 1400°F. At the high end of this temperature range, the designer must use a turbulent diffusion flame. The preheated air can meet the fuel only in the combustion chamber, as is the case with a diffusion flame, because the fuel/air mixture will spontaneously ignite. (2) Preheating increases the flame temperature, which in turn increases NO_x emissions. Since any advanced appliance should be designed to meet current and proposed emissions standards, it is important that any combustion system for the Stirling EHS meet at least the current California South Coast Air Quality Management District (SCAQMD) requirement (ref. 35) of 40 ng of NO_x/J of useful energy output (the most stringent to date). It would be prudent to set a target below this standard.

The SCAQMD standard is shown converted to more conventional terms in Fig. 3.2. The equivalent of the standard (40 ng/J of useful energy output) in ppmv is plotted versus excess air. The individual curves represent the effect of efficiency. The standard is based on efficiency as well as emission levels. For example, at 20% excess air and 80% efficiency, emission levels at or below 60 ppmv are in compliance; while at 180% efficiency (COP=1.8), a burner could have 135 ppmv and still meet the 40 ng/J standard. A standard based on overall efficiency as well as burner output offers an advantage to a SEHP with a COP above 1.0 .

Other things being equal, a premixed combustion system with or without a radiant transpiration cooled flameholder would be the combustion system of choice. Not only would it have the lowest potential NO_x emissions of any combustion system, but also its compactness and the availability of existing approved controls, valves, and accessories would be a distinct advantage. The disadvantage of this choice is that it would not be the most tolerant of high preheat temperatures. It has been used successfully by one developer (ref. 9) with air-preheat temperatures of only up to 800°F. This limitation to relatively low air-preheat temperatures means high stack losses are an inherent feature of premixed combustion systems applied to the SEHP.

Another system that was evaluated was a pulsed combustion system. Pulse combustion offers several potential benefits that make it an attractive candidate for the Stirling EHS. Foremost, it does not require any continuous source of power to provide system draft, thereby eliminating a significant element of operating cost (no blower or blower power is needed). Also, pulsed combustion generates a fairly high pressure rise, thus permitting higher gas velocities and pressure drops with resultant high heat transfer rates. The oscillating flows produced by the pulse combustion may also give higher heat transfer coefficients than steady flows at the same velocity. Finally, the key elements of the pulse combustor may be integrated synergistically with those of the Stirling EHS. A pulse combustion system is evaluated in Sect. 4. It shows some promise, but this technology has not been developed for use with preheated air. A significant development program would be needed to evaluate whether it is truly feasible.

3.2.2 Combustion Gas Recirculation and Exhaust Gas Recirculation

One of the main concerns regarding the design of the combustor was selecting a control strategy for emissions, especially oxides of nitrogen. Currently, EGR is the primary means of control for NO_x. Another technique used is CGR. In the case of EGR, a portion of exhaust gas that has passed through the heater head zone and the preheater is recycled back to the air intake, usually by introducing the exhaust gas at the blower intake. Since the blower intake is at a lower pressure than the exhaust, exhaust gas is sucked into the

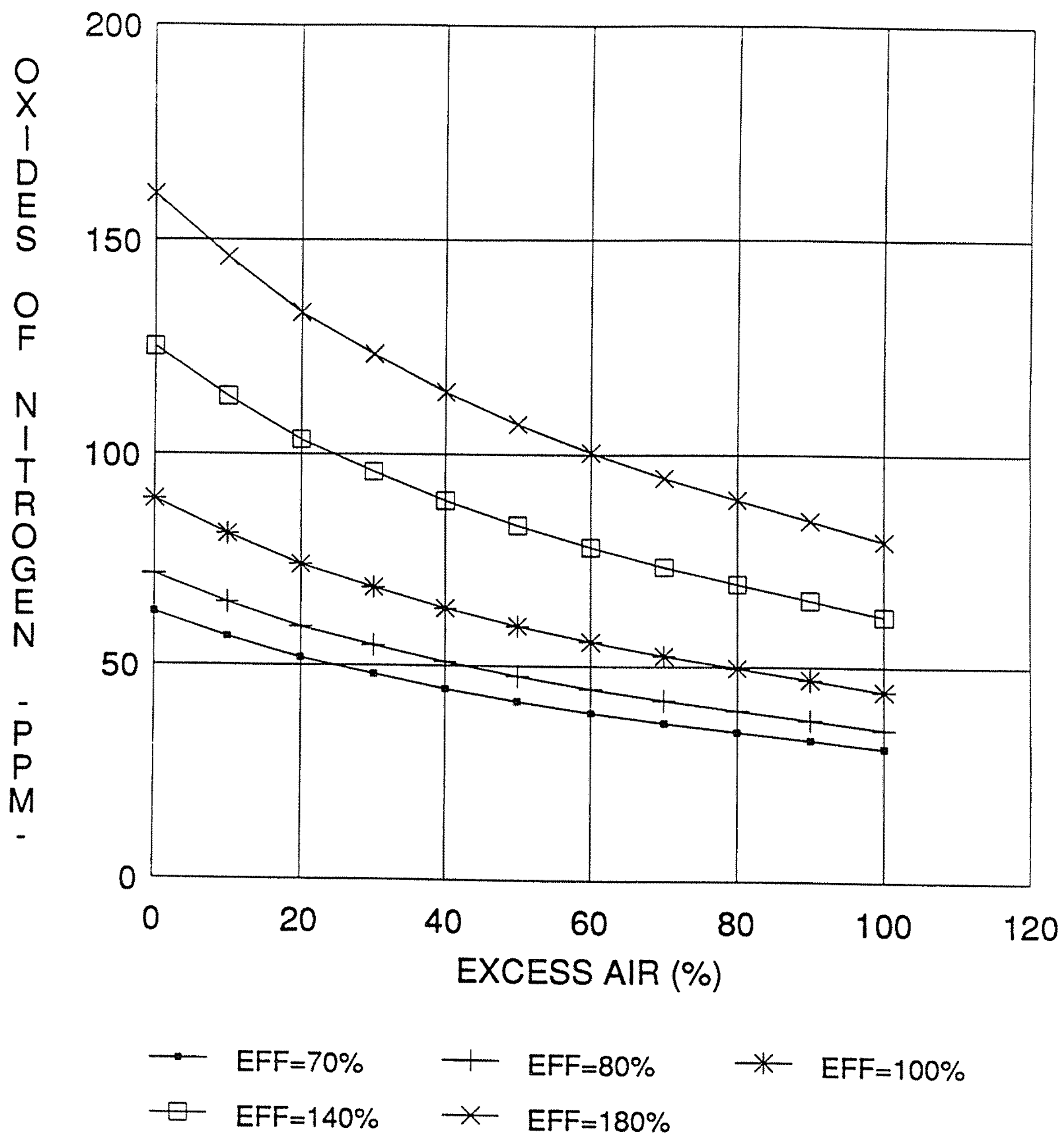


Fig. 3.2. The 40-ng emission level expressed in ppmv as a function of excess air and system efficiency.

blower. The drawback is that this method raises the gas temperature in the blower, increasing the blower power needed to move the less dense gas stream.

CGR is similar to EGR except that the exhaust is recirculated after it has passed the heater head but before it passes through the preheater. If nozzles, such as those pictured in Fig. 2.3, are used to create a low static pressure region, the exhaust gas can be drawn back to the combustion chamber, lowering the combustion temperature. The blower will need additional power, just as with EGR; and the nozzle arrangement will require that hardware be included in the hotter regions of the combustor, sometimes an expensive proposition. The advantage is that the blower will not be pumping high-temperature air and gas. It was decided to include this feature for further evaluation.

3.2.3 Advanced Cooling

Another concept targeted for evaluation was advanced cooling methods for selected combustion parts, most likely channeling of relatively cool air to the specific part. This is a difficult concept to implement because performance goals are not easily defined; it is actually a technique that can be used to make other concepts workable. For example, if preheat temperatures are so high that a combustion system has the tendency to overheat, selective air cooling can be used to make the system work. This is true for any combustion system, whether it be a turbulent diffusion or a premixed system. In addition, the technique can be used to cool a particular part, allowing the use of a less expensive material. This concept was retained to be utilized as the opportunity arises.

4. EVALUATION OF SELECTED CONCEPTS

In this section, the concepts and ideas developed in Sect. 3 are evaluated from a technical and cost perspective. An analytical model was developed that could interrelate the various performance parameters so that a "what if" analysis could be performed on the various EHS components. For example, the preheater size could be varied and the effect on combustion efficiency and cycle temperatures could be evaluated. The model also was capable of doing an "order of magnitude" analysis of the effect on NO_x emissions for various strategies. This model is described in Sect. 4.1. While the developed model also is capable of performing a partial cost analysis, a more detailed cost analysis was done separately using a parts list/spreadsheet approach.

In order to choose materials and accurately reflect their costs, the appliance codes dealing with natural gas-fired appliances were analyzed to determine the factors that should be used in selecting materials. Costs were obtained by having vendors quote to supply material for building 10,000 units per year. A model was then used to develop designs that could be priced. Finally, all of the control elements that would satisfy both the code requirements and the technical requirements were selected and priced at least on a preliminary basis. Because neither hardware designs for the systems nor specifications for control systems existed in detail, some assumptions had to be made.

4.1 ANALYTICAL MODEL

The EHS model has been used to analyze the effects of various design parameters on the performance and cost of the Stirling heat pump. The model is not intended to be a precise predictor of all performance parameters; instead, it is intended to gauge the influence of changes in the basic sizing parameters, such as heat exchanger surface area and flow area, on performance variables such as temperature, heat transfer, and efficiency.

Lumped-parameter models of the main elements of the EHS are used to analyze the influence of design changes in each element on the performance of the other elements. The lumped model, shown in Fig. 4.1, consists of the following elements:

- forced-draft combustion air blower,
- air preheater/exhaust recuperator,
- combustor, and
- heater head.

Provisions are made in the model for (1) EGR, which is assumed to flow from the outlet of the recuperator to the inlet of the combustion air blower; (2) CGR, which is assumed to flow from the outlet of the heater head into the air inlet of the combustor; and (3) preheater/recuperator internal leakage, which is assumed to flow from the preheater to the recuperator because of the higher pressure of the preheater.

The analytical model is programmed as a Quattro® Pro/Lotus 1-2-3® spreadsheet. To simplify programming and to facilitate comprehension of the model, all of the elements use similar analytical formulations of the following processes:

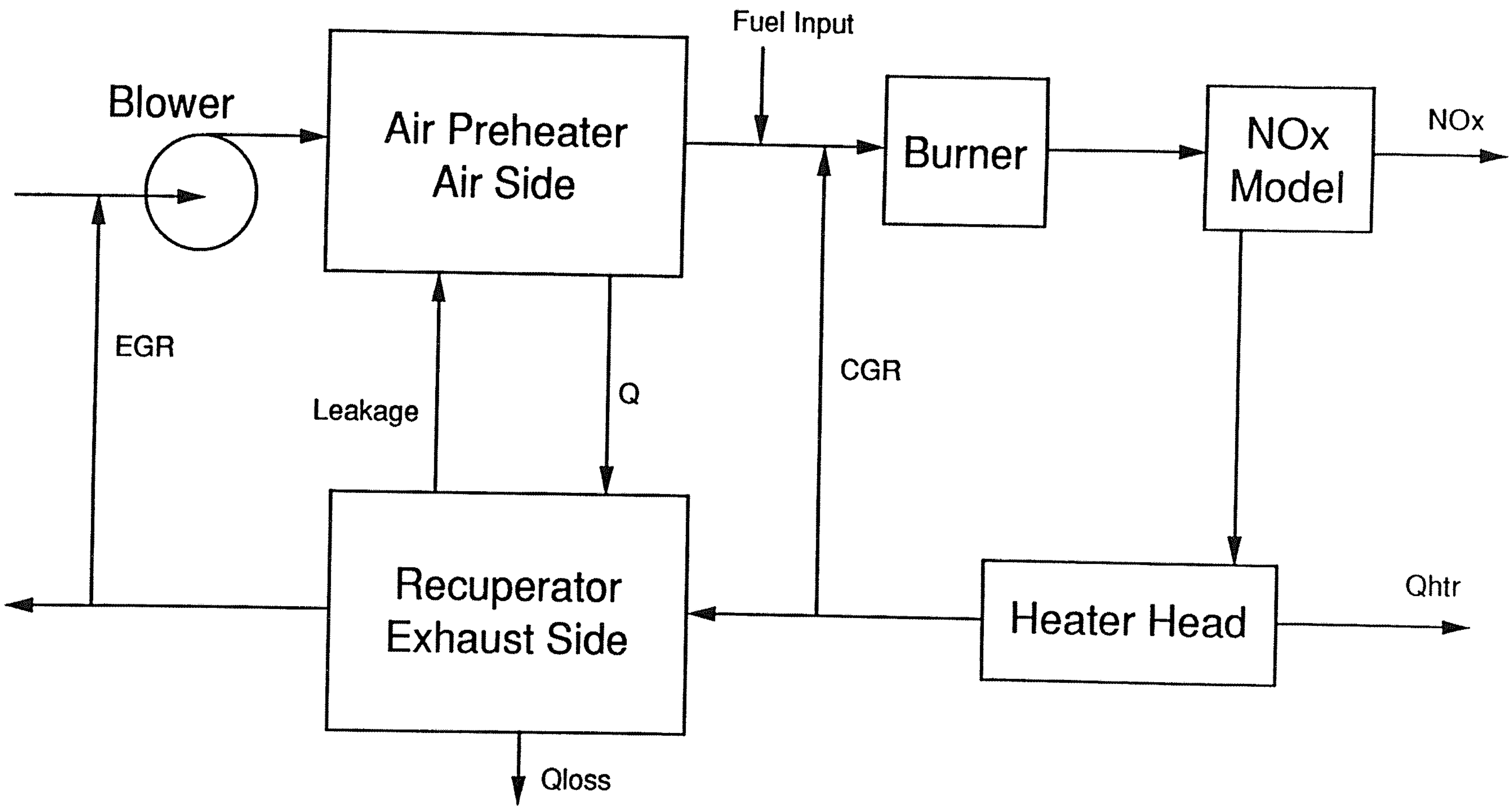


Fig. 4.1. Schematic of an analytical model.

- mass flow (conservation of mass, conservation of species),
- heat transfer (conservation of energy, rate correlations), and
- pressure drop (friction factor correlations).

The basic lumped model of an element of the EHS is shown in Fig. 4.2. The input variables, temperature (T_{in}), mass flow rate (w_{in}), specific enthalpy (h_{in}), and concentration of species (C_{in}), are identical to the corresponding outlet variable from the preceding element. Provision is made for additional flow of EGR; CGR, or leakage to or from the element; and heat transfer between elements (no provision is made for heat losses to the environment).

Simplified relationships for physical properties are used to avoid the computational complexities resulting from variable properties (discussed later). A complete output of the EHS model together with formula listings is given in Appendix A.

Referring to the sample tables in Appendix A, the first 33 rows contain the input parameters and the output variables. The input parameters are mostly self-explanatory. The cycle heat input is calculated as the base case heat input to the heater head times the ratio of Carnot efficiency of the base case to Carnot efficiency of the sample case. Thus all cases are evaluated at the same engine output.

The basic performance parameter of the heat exchanger elements is the NTU, defined here as:

$$NTU = \eta_f * htc * A/(w * c_p) , \qquad (3)$$

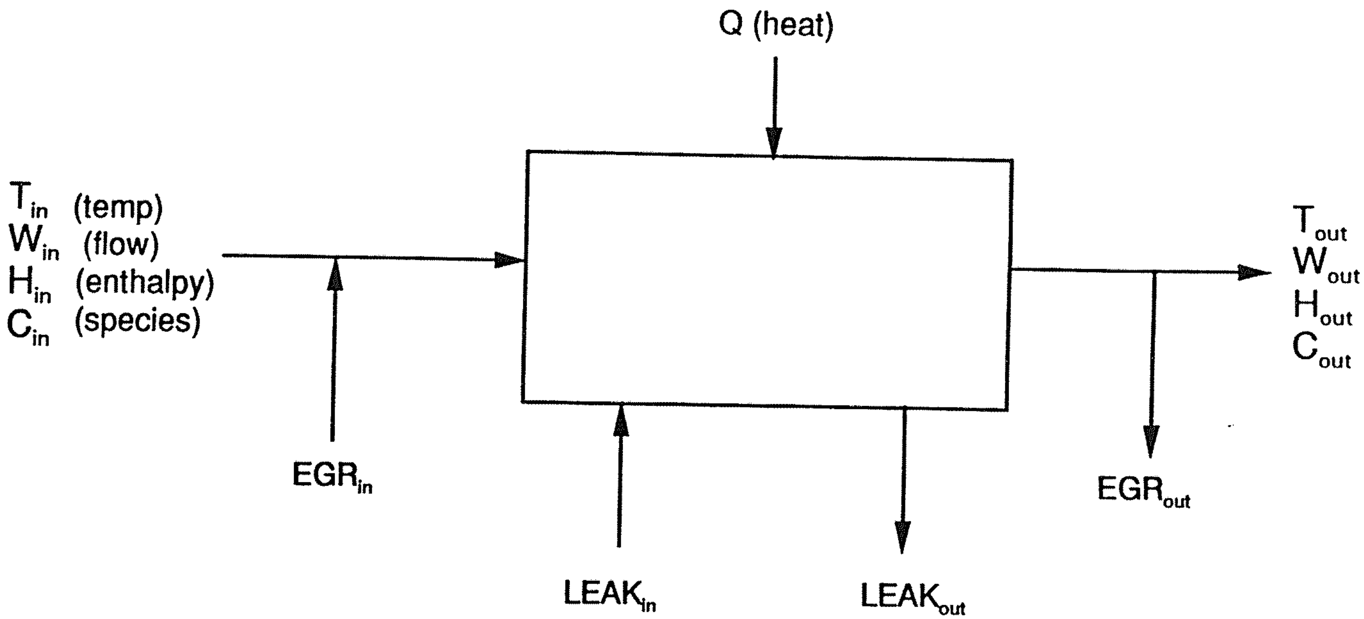
where:

η_f = fin efficiency,
htc = heat transfer coefficient,
A = surface area,
W = mass flow rate, and
C_p = specific heat of the gas of that element.

The NTU is a measure of the amount of heat transfer conductance relative to the amount of flow and is directly related to the heat exchanger effectiveness (size). The NTU of each element is selected as an independent variable, and the program sizes the heat exchanger accordingly and calculates the effect on temperatures, pressure drops, etc.

The mass velocity within the heat exchangers is also specified. It governs the required flow cross-sectional area, and hence the pressure drop in the heat exchanger. Although the mass velocity in the preheater may be specified independently, the mass velocity in the recuperator depends upon the mass velocity in the preheater and cannot be set independently, because the flow areas are assumed to be equal (a corrugated plate fin design is used).

The output of the model is summarized in lines 35-55. The firing rate is calculated to provide the same engine output as the base case, and it takes into account changes in heater head temperature (and hence Carnot efficiency) and stack loss. Next, the key



GOVERNING EQUATIONS

$$Q_{in} + (W_{in} + EGR_{in} + LEAK_{in})H_{in} = (LEAK_{out} + EGR_{out} + W_{out})H_{out} .$$

$$W_{in} + EGR_{in} + LEAK_{in} = W_{out}\ EGR_{out} + LEAK_{out} .$$

$$W_{out} = W_{in} + LEAK_{in} + EGR_{in} - LEAK_{out} - EGR_{out} .$$

$$H_{out} = [Q_{in} + (W_{in} + EGR_{in} + LEAK_{in})H_{in}] / (LEAK_{out} + EGR_{out} + W_{out})$$

$$C_{out} = \frac{\sum W_{in}\ C_{in}}{\sum W_{out}}$$

Fig. 4.2. Lumped-element model of external heat system.

temperatures throughout the EHS are displayed, as calculated by the models of the individual elements.

Overall operating cost is calculated from fuel cost plus the cost of running the combustion air blower. The theoretical blower power takes explicit account of only the pressure drops in the preheater, heater head, and recuperator. If CGR is employed, it assumes that the pressure drop in the ejector is equal to the pressure drop in the combustor and heater head (which equals the required pressure rise of the CGR stream). It does not explicitly model the other pressure drops in the EHS. Actual blower power is calculated as five times theoretical power, to account for blower and motor losses and other pressure drops.

Since all of the elements use a similar formulation, we may refer to any one of the elements to describe the general formulation. Inlet flow rate is the same as the outlet flow from the previous element. Added to this is any EGR/CGR plus any in-leakage, resulting in the "total flow in." From the total flow out, we subtract any outflowing leakage or EGR/CGR to obtain the outlet flow. The outlet enthalpy is determined from the incoming enthalpy fluxes, less the leakage flux, plus the heat input. Any leakage enthalpy is calculated using the average of the inlet and outlet enthalpy.

The outlet temperature is then calculated by dividing the outlet enthalpy by the specific heat. The specific heat is calculated as the mass weighted average of the specific heat of air and flue gas. Although the specific heat varies with composition, the specific heats of the air and flue gas are assumed to be independent of temperature. The heat transfer rate is calculated from standard NTU/effectiveness relationships.[36]

Specifying the mass velocity and hydraulic diameters of the heat exchangers permits the calculation of the heat transfer coefficient through the use of heat transfer correlations appropriate to the particular surface. With this information, the surface area required to produce the specified NTU can be determined.

The combustor is treated similarly to the heat exchangers, except for the heat of combustion. Here, most of the combustion properties are extracted from the MTI data for purposes of comparison. The average specific heat of the products of combustion (air-free) is calculated as the heat of combustion divided by the mass of reactants divided by the temperature rise. The specific heat of the air-free flue gas is assumed to be 13% greater than that of the air (which is necessary to account for the temperatures in and out of the preheater/recuperator).

The assumption of constant specific heat greatly simplifies the calculations. However, it overestimates heat capacity at the low temperatures and underestimates it at high temperatures. The most important result is that the stack loss tends to be overestimated.

Because of the high preheat of the EHS, either EGR or CGR will be required in order to limit NO_x. To evaluate various thermal design alternatives on a realistic basis, the NO_x emissions should be kept constant. To calculate the NO_x emission index, the NO_x formation rate $d(NO_x)/dt$ is calculated using the following equation:

$$\frac{d(NO_x)}{dt} = 3 \times 10^{14} e^{-129{,}000/RT} \times N_2 \times O_2^{.5} , \qquad (4)$$

where:

NO_x	=	oxides of nitrogen (mol/cm^3,)
R	=	gas constant (cal/mol/°K),
T	=	temperature (°K),
N_2	=	nitrogen (mol/cm^3),
O_2	=	oxygen (mol/cm^3).

This rate equation is generally recognized as describing NO_x generation even though some of the constants may vary.[37, 38] Although it recognizes that, in fact, NO_x does not form at a constant rate, this factor is thought to be a meaningful index that quantifies the influences of temperature and species concentration. It was assumed that residence time was constant for all cases. This assumption would be valid for combustors of similar design.

4.2 APPLIANCE CODE-RELATED ISSUES

It is important to be concerned with appliance code issues at this stage of technical development. The appliance market is closely regulated. Agencies such as the American National Standards Institute (ANSI) write test procedures for testing laboratories such as Underwriters Laboratories, Inc. (UL) and American Gas Association (AGA) that perform operating and safety tests to ensure that appliances meet these standards. It was recognized that much could be gained by examining these standards and incorporating the appropriate information into the design to help avoid an early development path that could lead to serious problems in certifying the appliance.

The specific code examined was the ANSI gas-fired furnace code.[39] The following code issues were found relevant:

- allowable emissions,
- material temperature limits,
- material thickness requirements, and
- controls and safety.

Currently, there is no NO_x emission standard in the ANSI furnace code, but the current SCAQMD standard was chosen as a target (see Sect. 3.2.1) (ref. 35). A prudent goal would be 25% to 50% of this standard. The ANSI code requirement for CO emissions calls for no more than 400 ppm (air-free) in the stack under normal operation. Normal operation means normal line gas pressure, tested as designed, etc. The difficult part of the code is a test requiring 400 ppmv (air-free) when a 12.5% overfire condition exists without allowing an adjustment of the air flow. In our experience, this test requires that the CO emissions be much less at normal operation than the allowed value. This simple test of overfire should be part of the testing procedure for the combustor. Although the Stirling engine enjoys an efficiency advantage that helps regarding NO_x

emissions, CO emissions are a health (rather than environmental) problem, so the code agencies set limits without regard to efficiency.

Another issue that could make the SEHP differ from most appliances is that some developers contemplate a continuously varying firing rate. How the ANSI code will treat that approach is not known. If the air/fuel control must keep the CO in balance at all times, meeting that requirement might prove difficult. If, on the other hand, a step control were used, emissions could be proved at discrete firing rates and continuously variable firing would probably be acceptable.

One of the concerns that the ANSI furnace standard addresses is the durability of the materials used in constructing the flue passageways. A general ANSI guideline for acceptable temperature limits for metals is approximately 90% of the scaling temperature. The scaling temperatures of various stainless steels, as well as some other useful information, are shown in Table 4.1. The first column lists materials that either are of interest to the project or were used to develop useful data (as will be explained). The ANSI-standard-approved maximum use temperatures for aluminum-coated steel, 410 SST, and 321 SST are listed, along with minimum thicknesses for use in flue passages. These values were set to ensure corrosion resistance over the life of the appliance under normal circumstances. Since natural gas-fired furnaces have lives of around 15 to 20 years, the same ANSI temperature limits should apply to the Stirling heat pump EHS system.

Materials in Table 4.1 that were not specifically listed in the ANSI standard were 304, 316, and 310 stainless steel. Comparing those materials with the listed materials allows an estimate of the temperatures likely to be approved. A scaling temperature of 1650°F is given for 304, 316, and 321 SST. The ANSI standard gives a use temperature limit of 1390°F for the 321 SST and a thickness requirement of 0.0195 in., so it was inferred that these same values would apply to 304 and 316 SST. Because MTI was considering using 310 SST for the preheater, this material was included in the table. The assumption was made that the 260°F difference between the scaling temperature and the ANSI-approved temperature of 321 SST could also be used for 310 SST; therefore, a limit of 1540°F was selected for 310 SST (based on 1800°F as the scaling temperature). An allowable thickness of 0.0195 in. was chosen for this material.

The chrome and nickel content of the materials is listed in Table 4.1 to determine whether there is a relationship between these and material cost. The cost per pound of the material, based on pricing from sheet metal vendors, is given for sufficient material to build around 10,000 units per year. Note that the cost follows nickel content, and the cost penalty for higher temperature metals is significant. At 1090°F, Al-coated steel costs are \$.49/lb, while at 1540°F and above, the cost climbs to \$3.28/lb.

The appliance code issues regarding control and safety of the EHS system require a control strategy that results in safe operation using AGA- and UL-approved controls and safety devices. If special controls are developed, they must undergo testing to the ANSI standards. The only inherent control problem found in MTI's design was the omission of a flame safety in the combustion system. Unless a system uses a pilot gas flame for ignition, it must have a flame safety system. A temperature monitor is not reliable or quick enough. Normally, a Kanthal rod is used to measure flame current to a ground to prove the presence of a flame. Some hot surface ignition systems use the ignitor for flame monitoring.

Table 4.1. Material selection criteria

Material	Scaling temperature (°F)	ANSI appproved temperature (°F)[a]	ANSI approved thickness (in.)[b]	Chrome (%)	Nickel (%)	Cost ($/lb)
Al-coated steel		1090	0.0254			0.49
410 SST	1250	1160	0.0225	11.5–13.5		1.46
304 SST	1650	1390[c]	0.195[d]	18–20	8–10	1.51
310 SST	1800	1540[c]	0.0195[d]	24–26	19–22	3.28
316 SST	1650	1390[c]	0.0195[d]	16–18	10–14	2.11
321 SST	1650	1390	0.0195	17–19	9–12	

[a]Furnace ANSI Code approved temperature for load bearing flue gas passages assuming an ambient temperature of 60°F (ref. 39).
[b]ANSI-listed minimum allowed thickness for furnace parts (ref. 39).
[c]Estimated for material listed, 90% or less than scaling temperature.
[d]AMTI estimates for material, not listed.

4.3 BASELINE CONDITIONS FOR SELECTED SYSTEMS

EHS designs based on the two distinct combustion systems were selected for evaluation. The primary approach was an EHS based on a diffusion flame combustion system identical to the 3-kW system under development at MTI. [34] This concept is the most typical of EHS systems used in Stirling applications. The turbulent diffusion flame approach allows high thermal efficiency (76–83%) because it can accommodate highly preheated air. Although these high efficiencies are desirable, they are achieved at higher cost because of the high-temperature materials requirements. A second approach was also selected, featuring a transpiration-cooled burner,[9] which was being considered as a candidate for the Sunpower SEHP system. [24]

The design conditions for these two systems are shown in Table 4.2. The top portion of the table generally corresponds to the input to the model, including the heat input to the cycle, the ambient temperature conditions, the preheater and heater head heat transfer area (represented by NTU), the heater head temperature, and flow strategies (excess air, EGR, and CGR).

The model output corresponds to the lower half of Table 4.2. The first item is the firing rate or input to the burner. Based on the size of the heater head and preheater, heater head temperature, etc., the efficiency of the combustor is calculated. The temperature profile is also calculated and aids in the selection of materials. For example, the average recuperator hot-end temperature is approximately the highest metal temperature in the preheater and is used to select the preheater/recuperator material. The combustor inlet temperature (preheater outlet) is used to select the ducting leading up to the combustion chamber, and the recuperator inlet temperature defines the material selection criteria for the ducting from the heater head to the exhaust side of the preheater. The Carnot efficiency is based on the heater head and ambient temperature. Although a complete and rigorous analysis would have to go through the entire Stirling engine cycle and engine design, the approach here was simply to tie the cycle efficiency to the heater head temperature and make adjustments to fuel consumption based on the cycle efficiency.

The first column of Table 4.2 contains design conditions reported by MTI for its diffusion flame design with a tubular heater head. The principal features are a firing rate of 29,054 Btu/h, a combustion efficiency of 80.6%, and a shaft output of 2.34 kW. The latter assumes a power output approximated as 50% of the Carnot efficiency. The second column shows the design conditions for a transpiration-cooled burner, while keeping the same shaft power output (2.34 kW). A few other differences or assumptions should be mentioned. First, a maximum preheat temperature of 850°F was assumed for the transpiration burner case. This limited the heater head temperature to 600°C; the preheater size is estimated as 17% of that for the diffusion flame burner. The combustion efficiency dropped from 80.6% for the diffusion burner to 73.3% for the transpiration burner. On the other hand, it is estimated that the transpiration burner only needed 6.8% EGR, compared with 30% for the diffusion burner, to control NO_x.

Table 4.2. Model comparison of diffusion flame and transportation-cooled external heat system

Case	Diffusion flame[a]	Transpiration cooled
Cycle heat input (Btu/h)	23,420	24,730
Inlet temperature (°F)	95	95
Excess air at blower inlet (%)	25	25
Preheater size (NTU)[b]	14.60	2.50
Recuperator size (NTU)[b]	15.12	3.05
Leakage (preheater recuperator) (% air flow)	0.0	0.0
EGR[c] (exh blr to inlet) (% air flow)	30.0	6.8
CGR[d] (heater head to comb) (% air flow)	0	0
Average heater head temperature (°C)	700	600
Heater head gas-side size (NTU)	2.55	2.55
Heater head fin efficiency (%)	90	90
HHV[e] firing rate (Btu/h)	29,054	34,340
Combustor inlet temperature (°F)	1,365	850
Combustor outlet temperature (°F)	3,496	3,491
Recuperator inlet temperature (°F)	1,462	1,295
Exhaust temperature (°F)	395	673
Average recuperator hot end temperature (°F)	1,413	1,073
d(NO)/dt–ppm/s	9.64×10^4	9.27×10^4
Stack loss (% HHV)	19.39	27.97
Combustor efficiency (%)	80.61	73.30
Carnot × combustion efficiency (%)	55.07	46.60
Shaft power at 50% Carnot (kW)	2.34	2.34

[a]MTI design conditions (ref. 63).
[b]NTU = number of transfer units.
[c]EGR = exhaust gas recirculation.
[d]CGR = combustion gas recirculation
[e]HHV = higher heating value.

4.4 MTI DIFFUSION FLAME EHS SYSTEM

4.4.1 MTI EHS System Description

The MTI EHS system is designed to accommodate two approaches in heater head design, a tubular style (Fig. 4.3) and a monolithic version (Fig. 4.4) (ref. 2). Actual data were not provided; only the design conditions shown in Table 4.2 (ref. 34) were used. MTI currently is evaluating several designs. For the purpose of this report, a tubular heater head is assumed. It is designed for operation at a nominal metal temperature of 700°C.

The first step in the evaluation was to establish the cost of the basic design of the EHS. This was broken down into two sections:

- preheater, combustor, and ducting and
- controls and accessories.

The data in Table 4.3 show the output of the cost model developed for the EHS (excluding the controls and accessories). The number labels of Fig. 4.3 correspond to the item numbers in Table 4.3. The metal type and the wall thickness for each item was selected based on the recommendations of the ANSI standard criteria in Table 4.1. The only exception was the preheater material that forms the preheater passageways (8) by accordion folding of 0.008-in. sheet metal. Although this part is in the flue passageway, it was assumed that this configuration would be acceptable because the thin sheet metal is enclosed between metal sheets (4 and 9) selected to meet the code. The material cost quotes were then used to evaluate the costs of each individual part. It was felt that calculating the actual labor costs was beyond the scope of this program and that those costs could be estimated by using some guidelines. A manufacturing company that produces heating parts and accessories using sheet metal stampings was contacted; staff there estimated that the labor component of their product cost was about 10% of material costs. AMTI had been in the hydronic heating business, which involved a boiler assembly along with associated controls and sheet metal costs. This business had labor costs that were 25% of the material costs. For this evaluation, a compromise was struck and labor was assumed to be 15% of the material costs.

The costs also must be burdened to reflect overhead costs such as rent, utilities, and general and administrative (G&A) costs such as selling and accounting. While these numbers can vary a great deal, the ratios usually agree fairly well when they are all applied. An overhead rate of 300% was applied to labor costs, and a G&A rate of 25% was applied to material, labor, and labor overhead costs. These are all reflected in the costing detailed in Table 4.3.

Using the basic MTI design with the material selections MTI provided, a cost of $147.14 was estimated. This estimate excluded the cost of the fan, control system, and auxiliaries; but it did include the cost of the burner nozzle assembly. A profit for the company and a distributor, as well as an installation cost, must be provided before the user gets any benefit. An installed cost to manufactured cost ratio of 2.4 has been suggested [40] and used by MTI.[3] Although a detailed basis was not provided, it appears to be a reasonable number and was used in this study.

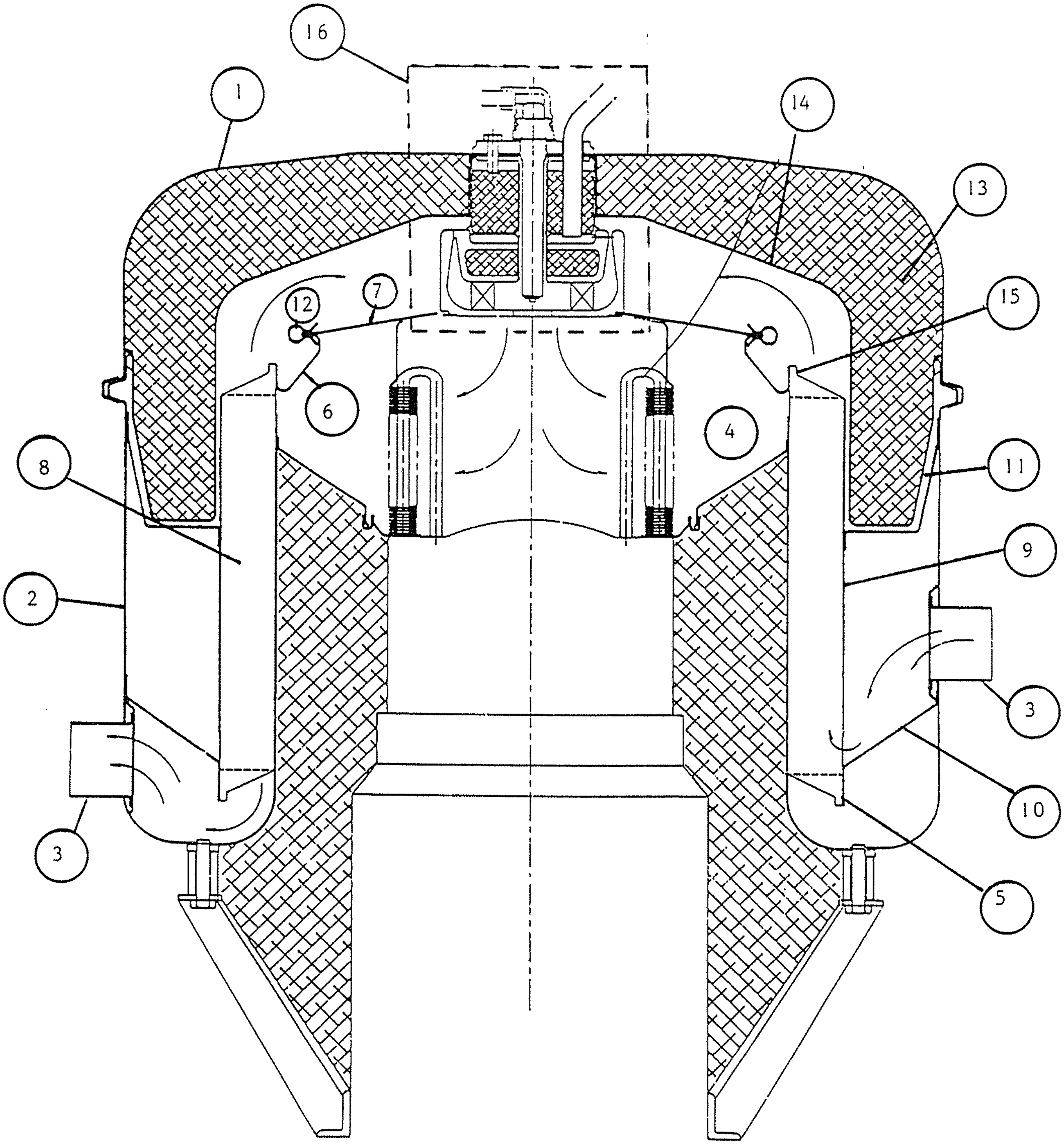


Fig. 4.3. Mechanical Technology, Inc., external heat system with tubular heater head.

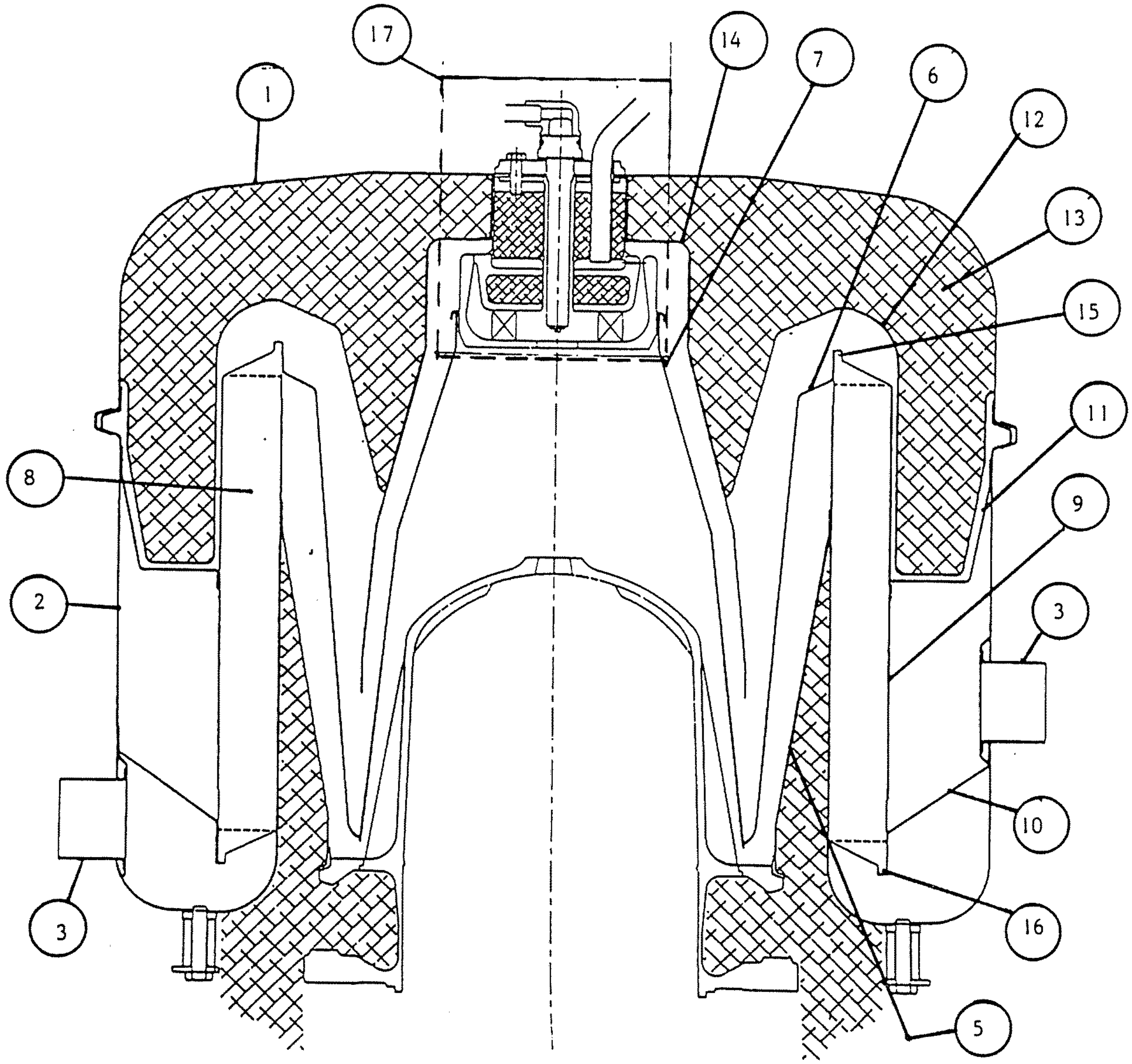


Fig. 4.4. Mechanical Technology, Inc., external heat system for monolithic head.

Table 4.3. Tubular heater head external heat system; dominant material 310 stainless steel

Manufacturing cost =	$147.14			Installed cost	=	$353.14		
Cost factors:	Overhead	=	300.00%	Material cost	=	$71.91	Labor =	$11.45
	General and adm.	=	25.00%	General and adm.	=	$29.43	Overhead =	$34.35
				Manuf. cost	=	$147.14		
Geometry factors:	Preheater ID (in.)	=	10	Preheater thickness (in.)	=	0.4		
	Preheater OD (in.)	=	10.8	Preheater length (in.)	=	8.75		
	Folds/in.	=	14					
	INS density (lbm/ft^3)	=	4					

Item no.	Quantity	Material	Thickness (in.)	Area (ft^2)	Volume (ft^3)	Weight (lb)	Cost/lb ($)	Cost ($)	Labor $ (15%)
1	1	CS	0.0195	2.2429662	0.003644	2.296236	0.35	0.80	0.12
2	1	ALS	0.0195	2.9354256	0.004770	3.005141	0.49	1.47	0.22
3	2	ALS	0.0254	0.0327249	0.000069	0.043638	0.49	0.04	0.01
4	1	310	0.0195	2.5634523	0.004165	2.624334	3.21	8.42	1.26
5	1	ALS	0.0195	0.1767145	0.000287	0.180911	0.49	0.27	0.04
6	1	310	0.0195	0.3926990	0.000638	0.402025	3.21	1.29	0.19
7	1	HANES	0.0195	0.3490568	0.000567	0.357356	16.40	5.86	0.88
8	1	310	0.008	15.0511740	0.010034	6.321493	3.21	20.29	3.04
9	1	310	0.0195	2.0616701	0.003350	2.110634	3.21	6.78	1.02
10	1	ALS	0.0254	0.5148721	0.001089	0.686581	0.49	0.34	0.05
11	1	304	0.0195	0.7723081	0.001255	0.790560	1.51	1.19	0.18
12	1	310	0.0195	0.1145372	0.000186	0.117257	3.21	0.38	0.06
13	1	Insulation	1.5	2.2429662	0.280370	1.121483	3.00	3.36	0.50
14	1	310	0.0195	1.1074114	0.001799	1.133712	3.21	3.64	0.55
15	1	310	0.0195	0.1636246	0.000265	0.167510	3.21	0.54	0.08
16 (nozzle assembly)	1	Various						10.69	2.21
						Subtotal		65.37	10.41
						Miscellaneous Hardware 10%		6.54	1.04
						Total		71.91	11.45

Before proceeding, it should be mentioned that the analysis is somewhat optimistic. It assumes that a plant with production tooling exists. This assumption of automated production results in low labor costs. Stamping and automatic welding equipment was also assumed.

The same process was used to evaluate the EHS for the monolithic heater head shown in Fig. 4.4. The cost of this design is detailed in Table 4.4; table entries are keyed to the part numbers in Fig. 4.4. The cost of this design is estimated to be $226.48. The drastic difference between the tubular and the monolithic head is due to the design of the combustion chamber and the associated ductwork. A Haynes 214 liner (7) is used to duct the combustion gases to the heater head. According to one reference,[41] Haynes 214 is five times more expensive than 310 SST, making its cost about $16.40 per pound. The design also requires additional 310 SST parts (6 and 14) to duct air to the liner for cooling (7). As configured, this approach is very costly. A previous approach developed by MTI using high-temperature insulation for the combustion liner might be a less costly approach than the use of high-temperature materials such as Haynes 214.

The cost of the burner nozzle subassembly (item 16 of Fig. 4.3 and item 17 of Fig. 4.4) is relatively insensitive to the design of the EHS system. This subassembly consists mainly of the gas pipe inlet, a turbulator, and an ignitor. The main cost item in the subassembly is the ignitor, estimated to be 50% of the subassembly cost. A performance issue that should be addressed is the durability of a spark ignition system in this application. AMTI's experience has been that the reliability and life of a spark ignitor in a totally enclosed hot combustion chamber is poor. When the ignitor is used only to light a pilot or a natural draft flame, the electrodes are cooled to some degree by the air entering the burner. In a hot combustion chamber, they tend to oxidize and consequently reduce reliability or increase maintenance intervals.

4.4.2 Material Options for the MTI Design

The approach of this section was to look at the effect of design conditions on material requirements and cost. This approach entails exploring the cost trade-off between thermal efficiency and capital investment.

The first step in determining the suitability of materials is to use the model to develop temperature profiles of the system for various design conditions. The first two variables to be examined were preheater size (NTU) and the heater head temperature. This analysis is summarized in Fig. 4.5, where preheater hot-end temperature is plotted versus preheater NTU. The hot-end temperature is the arithmetic average of the recuperator inlet and preheater outlet temperature. This temperature was the criterion for choosing the metal type for the preheater components. Heater head temperatures of 700° and 600°C were examined.

The MTI design point of a tubular heater head temperature of 700°C and 14.6 preheater NTU is shown on the figure. This condition had a hot-end temperature of 1413°F. When the preheater NTU is dropped to 10, the temperature drops to 1386°F, allowing a less expensive material to be used. Using Table 4.1 as a guide, it was determined that 310 SST could handle any case examined, while 304 SST could be used if the preheater NTU value is 10 or less.

Table 4.4 Monolithic heater head external heat system costs (number of transfer units = 14.6); dominant materials are Haynes 214 and 310 stainless steel

Manufacturing cost =	$226.48			Installed cost	=	$353.14			
Cost factors:	Overhead	=	300.00%	Material cost	=	$111.72	Labor	=	$17.36
	General and adm.	=	25.00%	General and adm.	=	$45.30	Overhead	=	$52.09
				Manufacturing cost	=	$226.48			
Geometry factors:	Preheater ID (in.)	=	10	Preheater thickness (in.)	=	0.4			
	Preheater OD (in.)	=	10.8	Preheater length (in.)	=	8.75			
	Folds/in.	=	14						
	INS density (lbm/ft³)	=	4						

Item No.	Quantity	Material	Thickness (in.)	Area (ft^2)	Volume (ft^3)	Weight (lbs)	Cost/lb ($)	Cost ($)	Labor $-15%
1	1	CS	0.0195	2.242966	0.003644	2.296236	0.35	0.80	0.12
2	1	ALS	0.0195	2.935425	0.004770	3.005141	0.49	1.47	0.22
3	2	ALS	0.0254	0.032724	0.000069	0.043638	0.49	0.04	0.01
4	1	310	0.0195	2.563452	0.004165	2.624334	3.21	8.42	1.26
5	1	310	0.0195	1.166000	0.001894	1.193692	3.21	3.83	0.57
6	1	310	0.0195	1.769283	0.002875	1.811304	3.21	5.81	0.87
7	1	Haynes 214	0.0195	1.308178	0.002125	1.339248	16.40	21.96	3.29
8	1	310	0.008	15.051170	0.010034	6.321493	3.21	20.29	3.04
9	1	310	0.0195	2.061670	0.003350	2.110634	3.21	6.78	1.02
10	1	ALS	0.0254	0.514872	0.001089	0.686581	0.49	0.34	0.05
11	1	310	0.0195	0.772308	0.001255	0.790650	3.21	2.54	0.38
12	1	310	0.0195	3.000000	0.004875	3.071250	3.21	9.86	1.48
13	1	Insulation	1.5	2.242966	0.280370	1.121483	3.00	3.36	0.50
14	1	310	0.0195	1.290000	0.002096	1.320637	3.21	4.24	0.64
15	1	310	0.0195	0.163624	0.000265	0.167510	3.21	0.54	0.08
16	1	310	0.0195	0.176714	0.000287	0.180911	3.21	0.58	0.09
17 (nozzle assembly)	1	Various						10.69	2.21
						Subtotal		101.57	15.84
						Miscellaneous Hardware	10%	10.16	1.52
						Total		111.73	17.36

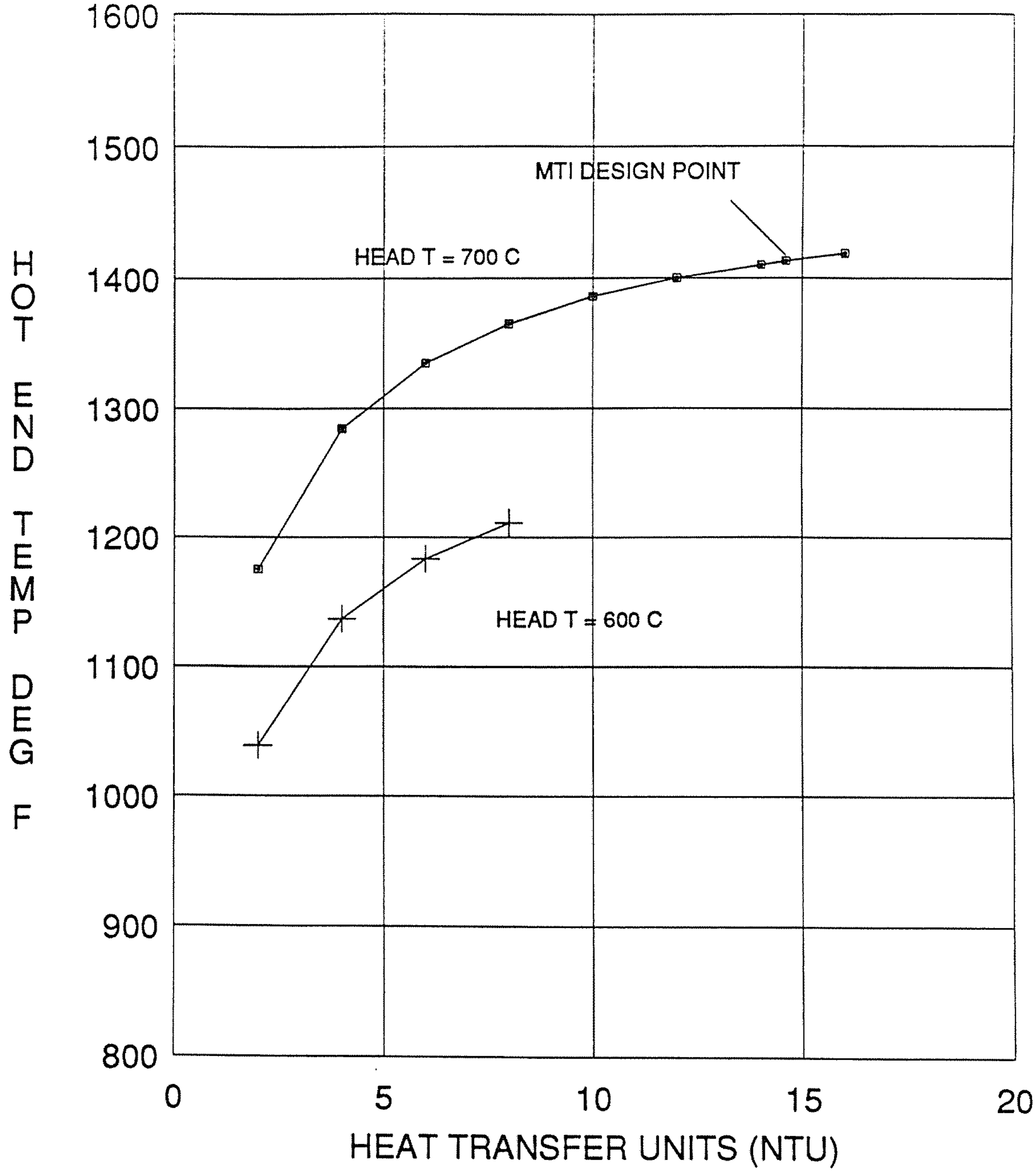


Fig. 4.5. Maximum metal temperature as a function of preheater size and head temperature.

To allow the preheater to be made of aluminum-coated steel required a heater head temperature of 600°C and a preheater NTU of 3. This design allows the maximum preheater metal temperature to be below the ANSI standard limit of 1090°F.

This benefit of using lower cost materials is, of course, counterbalanced by a performance penalty. Figure 4.6 shows the effect on performance of using lower preheater NTU values and lower heater head temperatures. This figure plots the ratio of burner input (Q_{in}) to the MTI base case (Q_{base}) versus preheater NTU for the two different heater head temperatures. By definition, the ratio is 1.0 for the base case of a preheater NTU of 14.6 and a heater head temperature of 700°C. For the case using 304 SST for the preheater with an NTU of 10, a fuel increase of about 2.5% was encountered. The manufactured cost of the EHS, however, drops from $147 to $75, which translates to an installed cost decrease from $353 down to $180, if the scaling factor of 2.4 is used. The case of using aluminum-coated steel with an NTU value of 3 required a fuel increase in excess of 17%.

Normally, a payback analysis would be performed to determine the best investment for various options. To perform this type of analysis, a relationship between operating costs and efficiency is required. The effect of combustion efficiency on the operating costs is shown in Fig. 4.7, which plots added operating fuel costs versus the drop in efficiency for units installed in Chicago and Atlanta. These data were taken from a more comprehensive analysis performed by MTI. [3] Because the two curves representing the two cities fell close together, an average was used.

The data of Fig. 4.7 were combined with those of Fig. 4.6 to produce the economic analysis of Fig. 4.8. In Fig. 4.8 the incremental payback is plotted versus preheater NTU for two materials, 310 SST and 304 SST. This analysis was performed by using the performance models described in Sect. 4.1 with the cost model shown in Table 4.3. The incremental payback was calculated by dividing the increase in installed cost by the fuel savings as the preheater size increased. Based on the curves of Fig. 4.7, the fuel savings were estimated to be $5 per one percentage point increase in EHS system efficiency. Based on the assumption that a 3-year payback is acceptable, the data in Fig. 4.8 show that a 310 SST preheater size slightly under 10 NTU would be optimum, rather than the current 14.6 NTU design. This change would decrease the installed cost from $383 to $302, while the system efficiency would decrease from 80.6% to 78.8% (1.8 percentage points).

A decrease in preheater size also would decrease metal temperatures in the EHS as shown in Fig. 4.5, and might allow the use of less expensive materials. The lower curve of Fig. 4.8 shows the incremental payback when 304 SST is used in place of 310 SST. This change decreases the installed cost from $353 to $180 with a loss of only 1.8 percentage points in efficiency for a preheater size of about 10 NTU. A preheater made with 304 SST is restricted to 10 NTU because of metal temperature limitations. The conclusion is clear for the case of the diffusion flame burner. A preheater of 10 NTU using 304 SST as the predominant material is the clear choice. Using aluminum-coated steel with a 3-NTU preheater was found to be less economical than the original base case. The next section describes a transpiration-cooled burner that did use aluminum-coated steel.

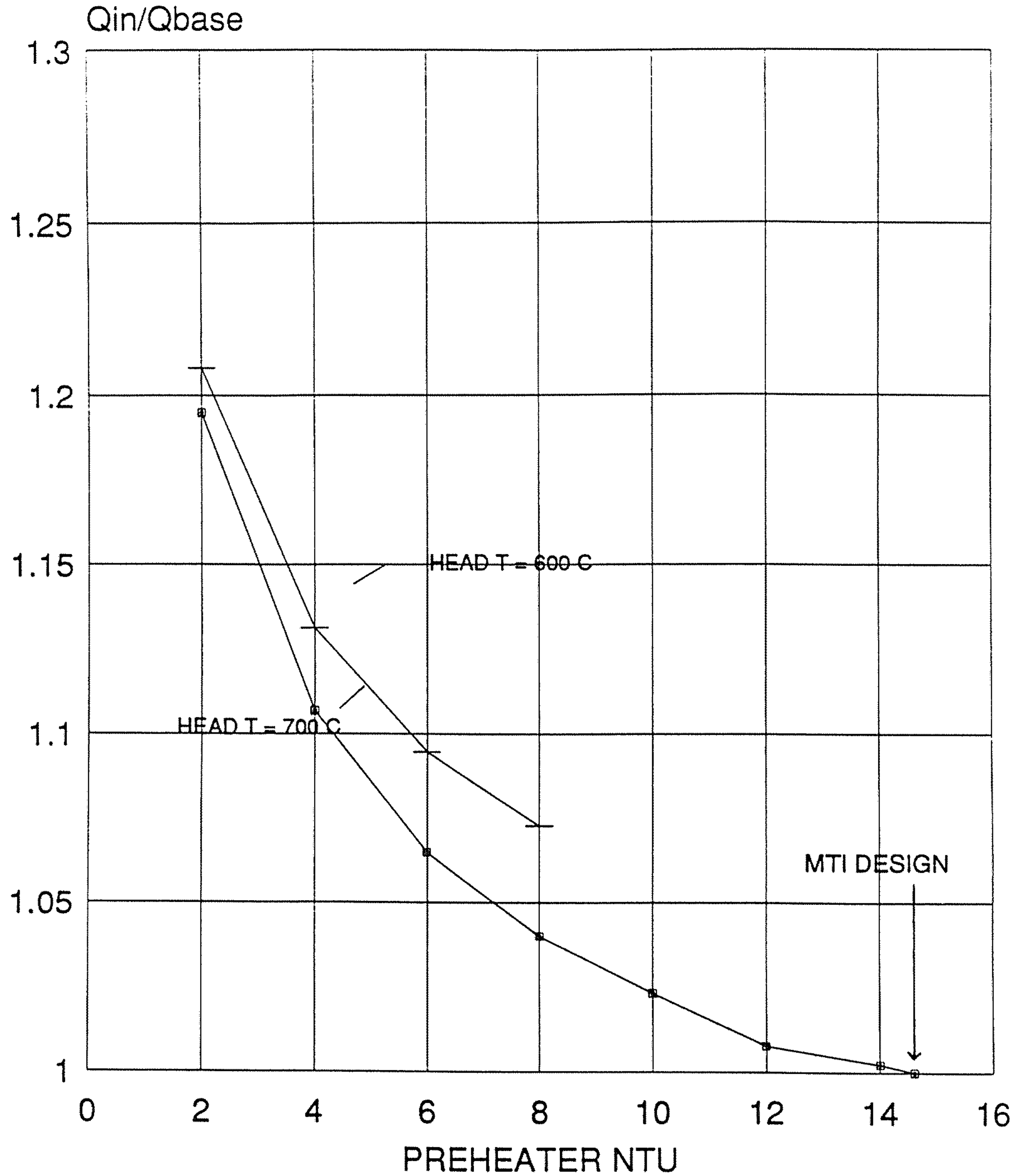


Fig. 4.6. Effect of number of transfer units on efficiency for constant NO_x figure of merit.

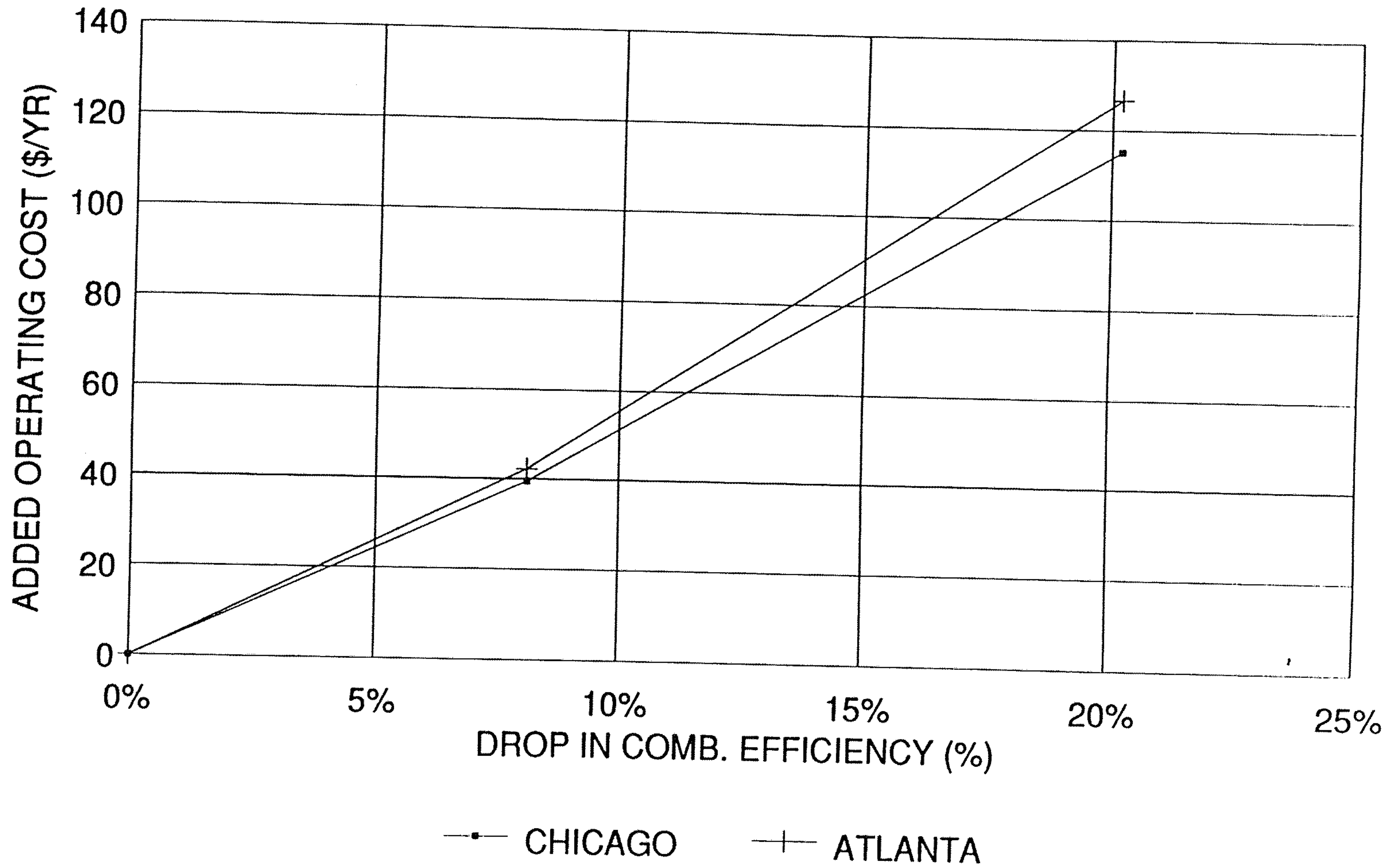


Fig. 4.7. The effect on cost of a decrease in combustion efficiency. *Source:* Holliday, J. C., S. G. Howell, and M. Richter, "Free-Piston Linear Alternator Solar Stirling Engine Concept," in *Proceedings of the 21st IECEC,* San Diego, August 25–29, 1986.

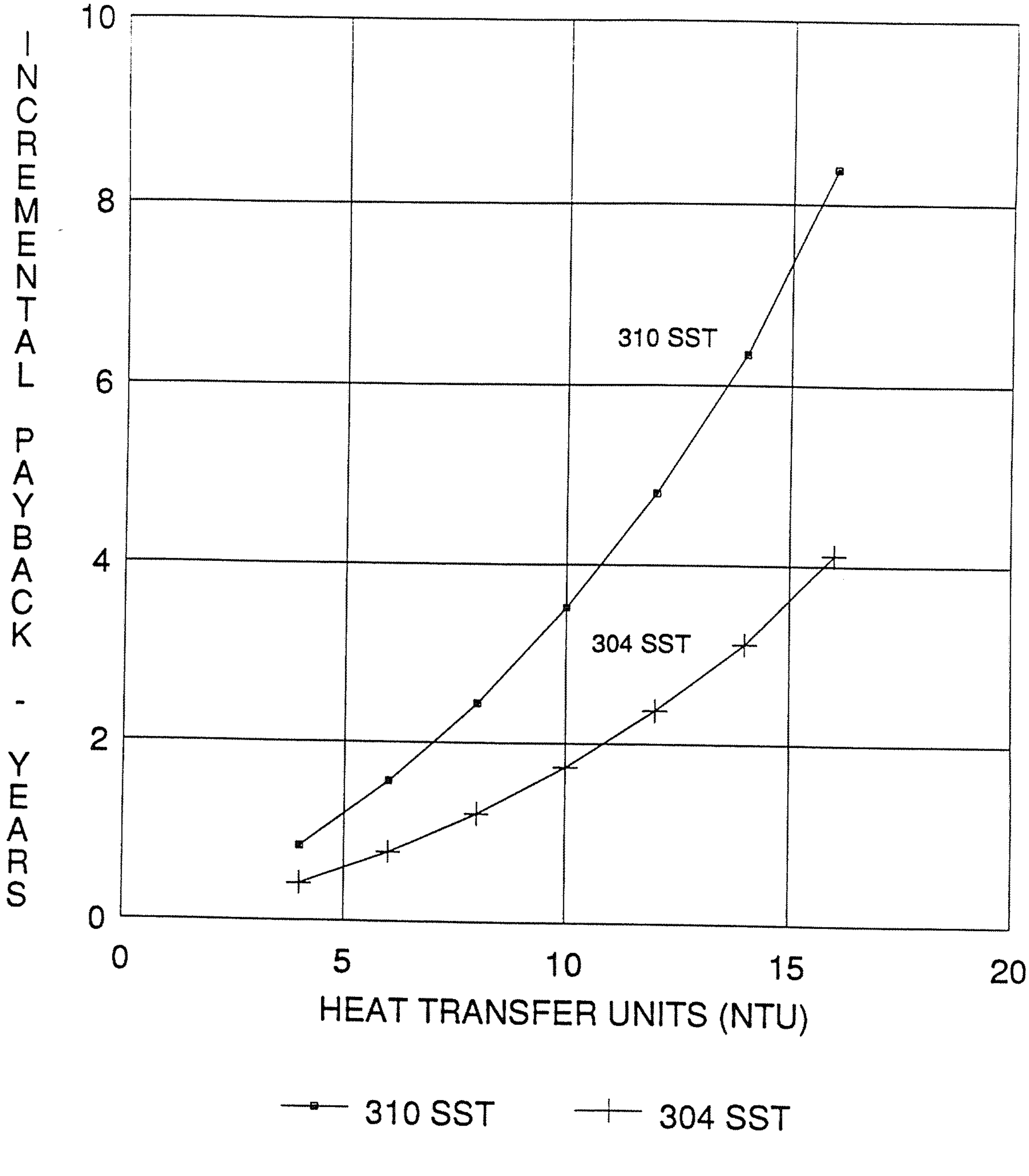


Fig. 4.8. Incremental payback for the preheater as a function of size and material choice.

It was felt that before this analysis was completed, the heater head NTU should also be examined for effectiveness because the MTI design had a high preheat/low heater head NTU design. The heater head NTU varied from 2 to 6 (base case was 2.55), and the preheater NTU varied from 4 to 16 (base case was 14.6). Two heater head temperature cases were run for this analysis, 600°C and 700°C. A summary of the analysis is provided in Table 4.5. The shortcoming of this analysis is that heater head costs were not evaluated because appropriate design details were lacking (heater head costing was beyond the scope of this study).

The object of this analysis was to determine whether there was an optimum proportioning of heat transfer area between the heater head and the preheater. The MTI base case (Case 1 of Table 4.5) is shown as having a maximum metal temperature of 1400°F and no fuel consumption penalty because it is the base case. At 700°C, it appears that a heater head NTU of 4 and a preheater NTU of 10 would result in performance close to the original design condition with a 0.6% fuel penalty increase but at a lower metal temperature (1250°F), which would easily allow the use of 304 SST. Alternatively, increasing the heater head to 4 NTU and leaving the preheater at 14.6 NTU (base case) would allow the use of a 304 SST preheater design with no fuel penalty. The savings in the preheater would have to be balanced against the cost of increasing the heater head NTU from 2.55 to 4.

Another goal of this analysis was to explore the use of an aluminum-coated steel preheater. It appears that a heater head NTU of 4 combined with a heater head input temperature of 600°C and a preheater NTU of 14.6 could achieve the desired objective. It could allow the use of the lower cost aluminum steel preheater and result in a fuel consumption increase of only about 1%.

Note that it is possible to use some exhaust gas heat that is not recovered by the recuperator for heating the conditioned air space (during the heating season). This could have the effect of making designs using relatively small air preheaters/recuperators somewhat more favorable economically.

4.5 THE TRANSPIRATION-COOLED EHS BURNER

A transpiration-cooled burner was included for several reasons. An EHS design featuring a transpiration burner was considered recently by Sunpower, Inc.,[24] and earlier by General Electric for SEHP applications. Transpiration burners using pre-mixed fuel and air are known to achieve low NO_x emissions. An added benefit is that many off-the-shelf components are available for this combustion system. Preliminary cost analysis reveals that the transpiration burner design holds some promise for significantly reduced capital cost, although thermal efficiency would be limited.

For this evaluation, the design conditions shown in Table 4.6 are slightly different from those for the diffusion flame burner. Although the power output at 2.34 kW is the same, the fuel input rate at 34,340 Btu/h is 18% higher. The reasons for higher fuel use are a lowered heater head temperature (600°C) and a small preheater (2.5 NTU, 17% of the base case value of 14.6 NTU), which stems from the inability to use highly preheated air. The specific conditions selected allow the use of aluminum-coated steel in the preheater and other components, which lowers the EHS cost.

Table 4.5. Effect of heater head number of transfer units

Parameter	Case (base) 1	2	3	4
Heater head NTU[a]	2.55	4	4	4
Preheater NTU	14.6	10	14.6	14.6
Heater head temperature (°C)	700	700	700	600
Fuel consumption increase (%)	0	0.6	0	1
Maximum metal temperature (°F)	1400	1250	1270	1100

[a]NTU = number of transfer units.

An EHS system based on a transpiration-cooled burner developed by General Electric is shown in Fig. 4.9, and the burner is shown in Fig. 4.10 (ref. 9). This burner uses premixed air and gas. The burner surface operates at temperatures in the 1500°F range, providing a radiant component in addition to the convective heat transfer in the combustion chamber. One side benefit to this design is that the products of combustion are quenched below NO_x producing temperatures, freezing the production of NO_x rapidly and minimizing generation. This is in contrast to a turbulent diffusion flame, which maintains high combustion temperatures until the gases hit the heat transfer surface (heater head).

A major disadvantage of this system is the inability to use a high air preheat temperature. The system has been used with about 800°F of preheat, but there are few data to indicate what preheat temperature can safely be used. The barrier is the auto-ignition temperature of the premixed fuel and air, which can vary with air/fuel ratio, flow conditions (including residence time), and gas constituencies. Auto-ignition temperatures for methane and other hydrocarbons commonly found in natural gas are given in Table 4.7 (ref. 42) for time-lagged and instantaneous ignition. How auto-ignition temperature relates to the allowable preheat temperature is not known, except to establish some absolute limits. Realistically, the maximum allowable preheat temperature may be lower for other reasons. Establishing the preheat temperature limit is a good research issue.

Comparing the EHS design featuring the transpiration burner with the design featuring the turbulent diffusion burner required putting them on an equal footing from an emissions standpoint. A calculated figure of merit for NO_x emissions (using the model explained in Sect. 4.1) was used as an index, so only systems with similar NO_x emissions are compared. The diffusion flame burner at 25% excess air and 30% EGR had a NO_x figure of merit of 9.64×10^4 ppm/s. With less preheat, the transpiration-cooled design reached a similar NO_x figure of merit (9.27×10^4 ppm/s) at 25% excess air and 6.80% EGR.

In addition to the effect of lower preheat, the transpiration-cooled burner probably would have lower emissions than the model indicates. This means that the model predicts

Table 4.6. Design conditions for transpiration-cooled burner

Input parameters	Value
Cycle heat input (Btu/h)	24,730
Inlet temperature (°F)	95
Excess air at blower inlet (%)	25
Preheater size (NTU)[a]	2.50
Recuperator size (NTU)	3.05
Leakage out of preheater to recuperator (%)	0.0
EGR[b] (exhaust stream to air intake) (% air flow)	6.8
CGR[c] (heater head to combustor) (% air flow)	0.0
Average heater head temperature (°C)	600
Heater head gas side size (NTU)	2.55
Heater head fin efficiency (%)	90
Calculated parameters	**Value**
Firing rate (Btu/h, HHV)[d]	34,340
Combustor inlet temperature (°F)	850
Combustor outlet temperature (°F)	3,491
Recuperator inlet temperature (°F)	1,295
Exhaust temperature (°F)	673
Average recuperator hot end temperature (°F)	1,073
d(NO)/dt (ppm/s)	9.27×10^4
Stack loss (%, HHV)	27.97
Combustor efficiency (%)	73.30
Carnot efficiency × combustion efficiency (%)	46.60
Shaft power at 50% Carnot efficiency (kW)	2.34

[a]NTU = number of transfer units.
[b]EGR = exhaust gas recirculation.
[c]CGR = combustion gas recirculation.
[d]HHV = higher heating value.

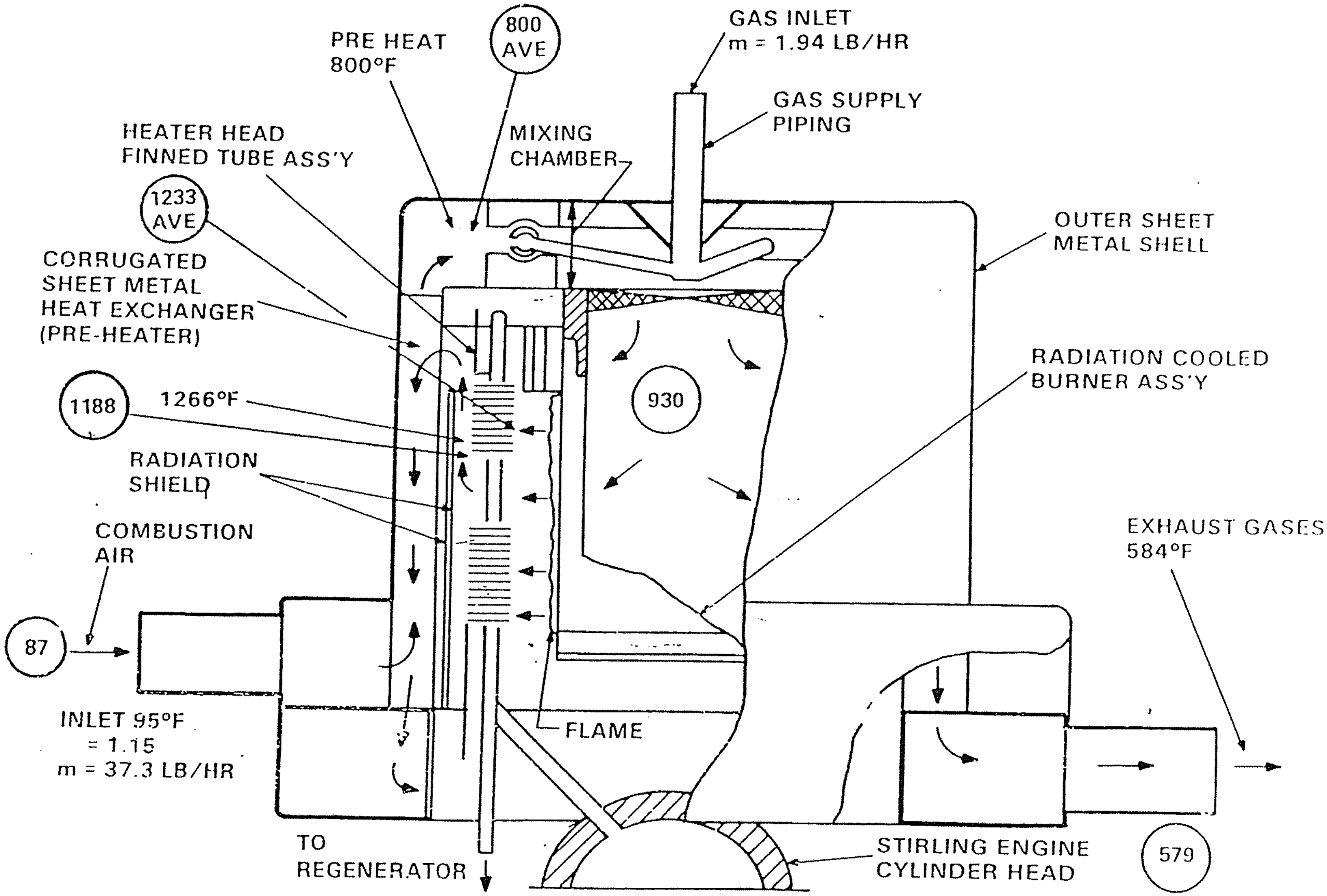


Fig. 4.9. The external heat system developed by General Electric.

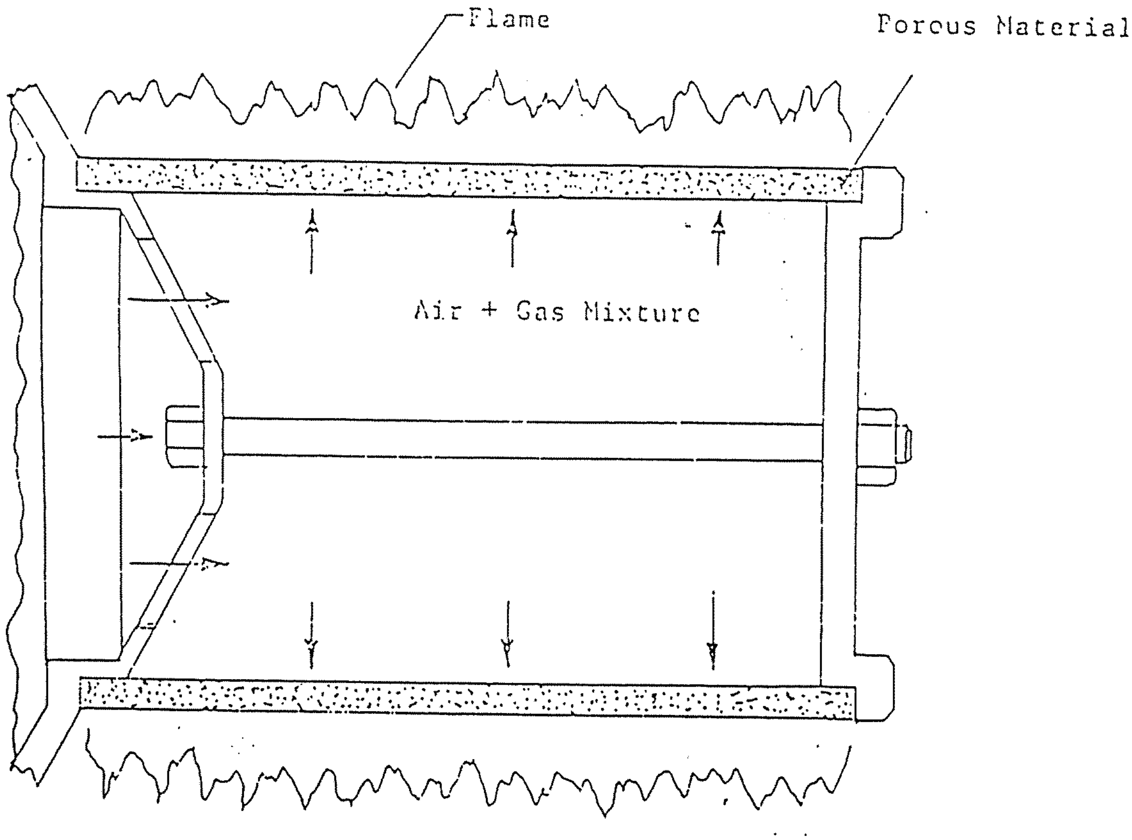


Fig. 4.10. Transpiration burner.

Table 4.7. Ignition temperatures of air–gas mixtures

Gas	With lag: Percentage gas (%)	With lag: Average temperature (°F)	Instantaneous: Percentage gas (%)	Instantaneous: Average temperature (°F)
Hydrogen	8–24	1130	10	1377
Carbon monoxide	13–47	1204	50	1708
Methane	5–38	1202	25	>1832
Ethane	N/A	968		
Propane	N/A	914		
Acetylene	4–22	804		
Ethylene	6–19	1004	10	1832

higher emissions from the transpiration burner than would actually be encountered because the "quenching" characteristic of the transpiration burner is not reflected in the model. Some development testing is required to determine NO_x emissions more accurately, but that is beyond the scope of this project. The best that can be done at this time is to note this information and use it to qualify decisions.

Figure 4.11 shows a composite of the unit that incorporates what appears to be the lowest cost design for the EHS system. From the MTI design, the plate-fin preheater with the clinched ends is probably the least costly unit. Because of the lower temperatures, aluminum-coated steel was used primarily in this design; some 304 SST was used in critical areas. For ignition, a hot surface ignition system would be used with controls that operate the ignitor both for start-up and as a flame monitoring system.

The manufacturing cost of this design was priced at $57.89. The installed cost of this EHS system is $138, not including the control system. That figure compares with $353 and $180 for the diffusion flame EHS designs with the dominant materials 310 SST and 304 SST, respectively.

The burner design will be based on the burner loading and balanced against the combustion intensity. For radiant type burners, a combustion loading of 1000 to 2000 Btu/h $\times$ in^{-2} is used to keep the combustion process in the surface of the flameholder for a burner with no air preheat.[43] At loadings above 2000 Btu/h $\times$ in^{-2}, the combustion process takes place above the flameholder surface and loadings up to 6000 Btu/h $\times$ in^{-2} have been achieved at AMTI. Normally, without preheat, the loading for a radiant burner would have to be below 2000 Btu/h $\times$ in^{-2}; but it is predicted that when preheat is added, the burner loading can be increased above 2000 Btu/h $\times$ in^{-2} and still retain the radiant features. Without some experimental work, however, this prediction cannot be proved. The other combustion parameter that must be examined is the combustion intensity. A good value of combustion intensity would be .5 million to 1 million Btu/h $\times$ ft^{-3}. This value would provide sufficient volume for low CO emission levels and low residence time for low NO_x.

4.6 PULSE COMBUSTION SYSTEM

Pulse combustion (PC) offers several potential benefits over conventional combustion systems that make it an attractive candidate for a Stirling EHS. Foremost, it does not require any continuous external source of power to provide system draft, thereby eliminating a significant element of operating cost. Second, it is capable of generating a fairly high pressure rise, permitting higher gas velocities and pressure drops, with resultant high heat transfer rates. Additionally, the oscillating flows produced by the PC develop higher heat transfer coefficients than do steady flows at the same velocity, further improving heat transfer. Finally, as will be described, the key elements of the pulse combustor may be integrated synergistically with those of the Stirling EHS.

To assess the feasibility of a PC-based EHS, several conceptual designs were generated, and the most promising of these were analyzed to estimate the sizes and performance parameters of the key elements.

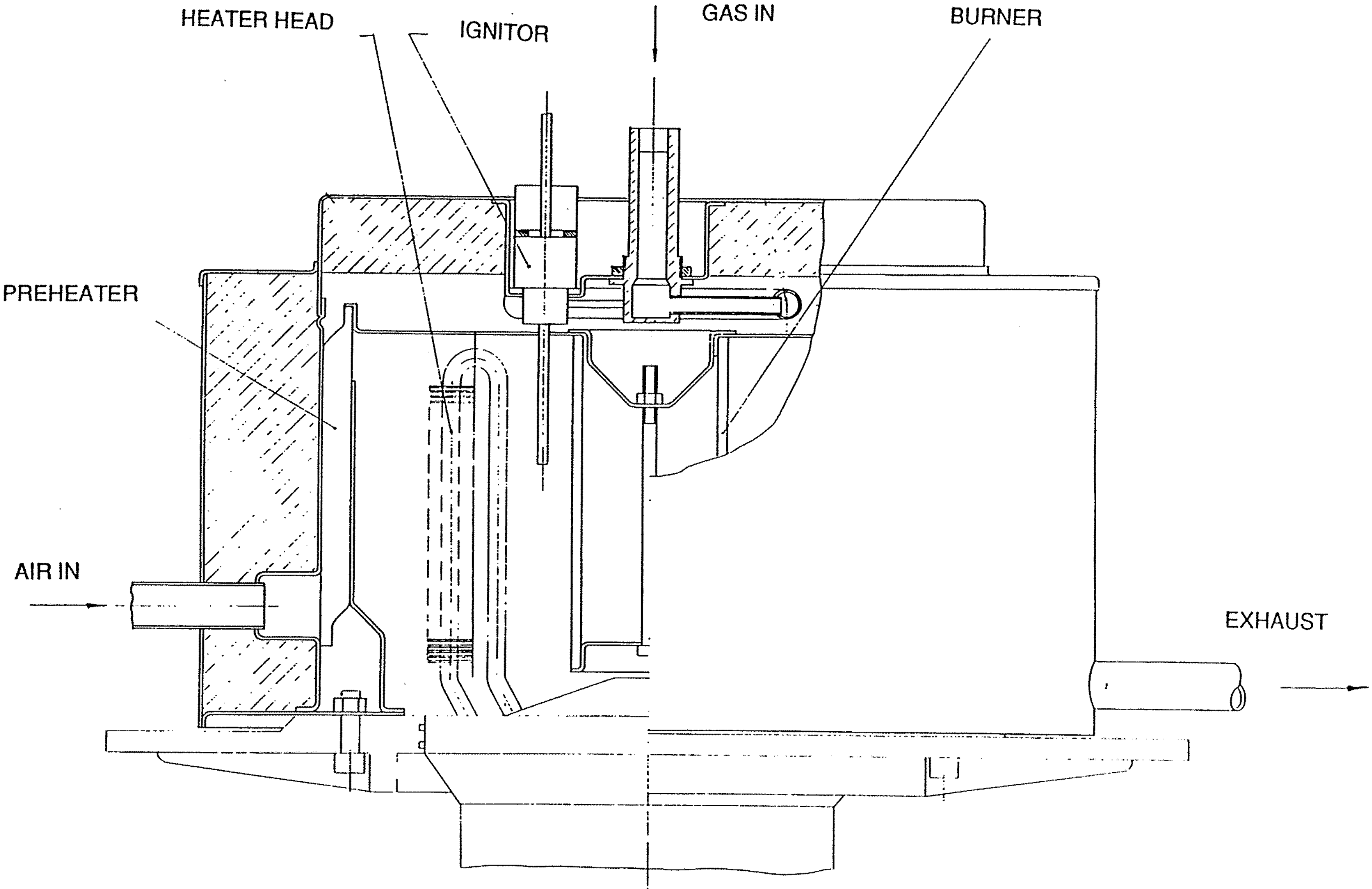


Fig. 4.11. An external heat system with transpiration-cooled burner.

The basic elements of a PC system are illustrated in Fig. 4.12. Aside from the control hardware, such as gas and air valves, the key dynamic elements that must be integrated with the EHS are the combustion chamber, the tailpipe, the inlet and exhaust decouplers, and the inlet and vent pipes. For PC to be technically and economically practical, the components required for the proper dynamic operation of the PC must be applied as well to the thermal requirements of the EHS. This requires sizing the dynamic elements for their appropriate PC dynamics and then using these same components as the main elements of the Stirling EHS without compromising their dynamic characteristics.

An example of how this process may be accomplished is illustrated schematically in Fig. 4.13. Here, the tailpipe is used as a device to heat the heater head. As will be demonstrated, this arrangement has the potential to provide significantly higher heat transfer to the heater head than traditional approaches. The exhaust decoupler also serves as a recuperator, exchanging heat with the inlet decoupler. The inlet and vent pipes are used similarly to recuperate exhaust heat.

To determine quantitatively whether the dynamic and thermal requirements could be met simultaneously, information from American Gas Association laboratories was used to size the key components of the PC.[44] Once their dimensions had been established, their thermal performance was estimated conservatively using conventional (i.e., non-oscillatory) heat transfer correlations.

The preferred approach is illustrated in Fig. 4.14. Here, the tailpipe is used to transfer heat to the cylindrical monolithic heater head of the engine. Portions of the inlet and exhaust decouplers serve as the preheater/recuperator. However, the volume contained within the preheater/recuperator is insufficient to buffer the pulsations adequately, and additional decoupler volume must be provided by the concentric chambers surrounding the preheater/recuperator. The inlet/vent pipes, which must be of sufficient length to enable the decouplers to absorb the pulsations, are arranged concentrically, thereby providing additional recuperation of exhaust energy.

The dynamic and thermal analytical results are shown in Table 4.8. Although the information from the American Gas Association research recommends specific mixer and combustion chamber dimensions, these may not be amenable to the Stirling EHS; however, the recommended volumes and most passage dimensions have been followed. The PC input was sized nominally at 30,000 Btu/h rated load and 10,000 Btu/h part load.

The case analyzed in Table 4.8 indicates that a PC-based EHS may be capable of superior thermal performance. Even without allowing for the enhanced heat transfer produced by the oscillatory motion, the heater head and preheater/recuperator effectiveness are equal or superior to those obtained with conventional forced draft systems, but without the expenditure of external power. The dynamic and thermal requirements appear to be easily met. Although the configuration of the heater head, with its concentric heat exchangers, is not conducive to the long length-to-diameter ratios generally recommended for the PC mixer and combustion chamber, that is not considered a serious impediment. Although the calculated sound levels may appear high, allowance has not been made for attenuation in the inlet/vent pipes; and they are functions of the sizes of the decouplers, which have been selected arbitrarily in this example.

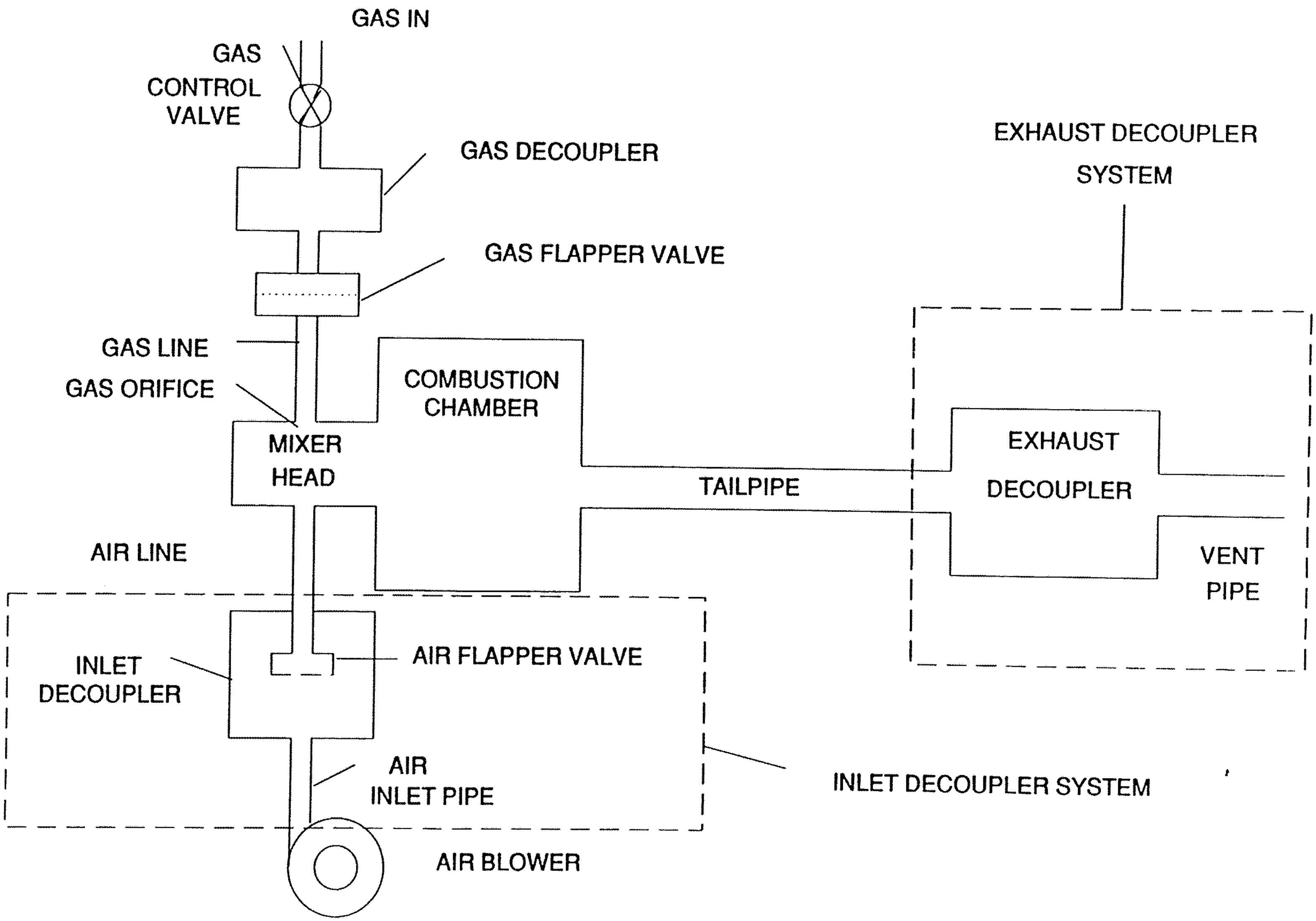


Fig. 4.12. Pulse combustion burner system.

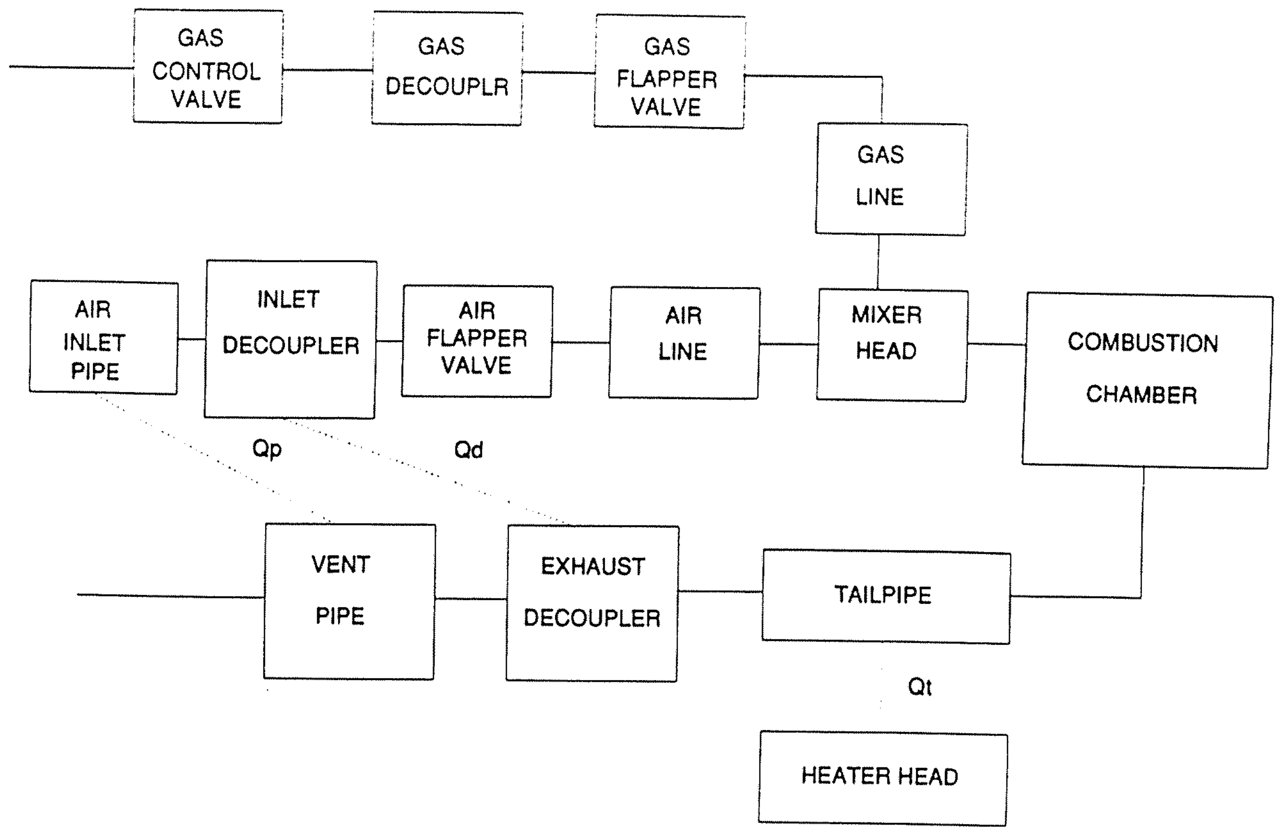


Fig. 4.13. Pulsed combustion system schematic.

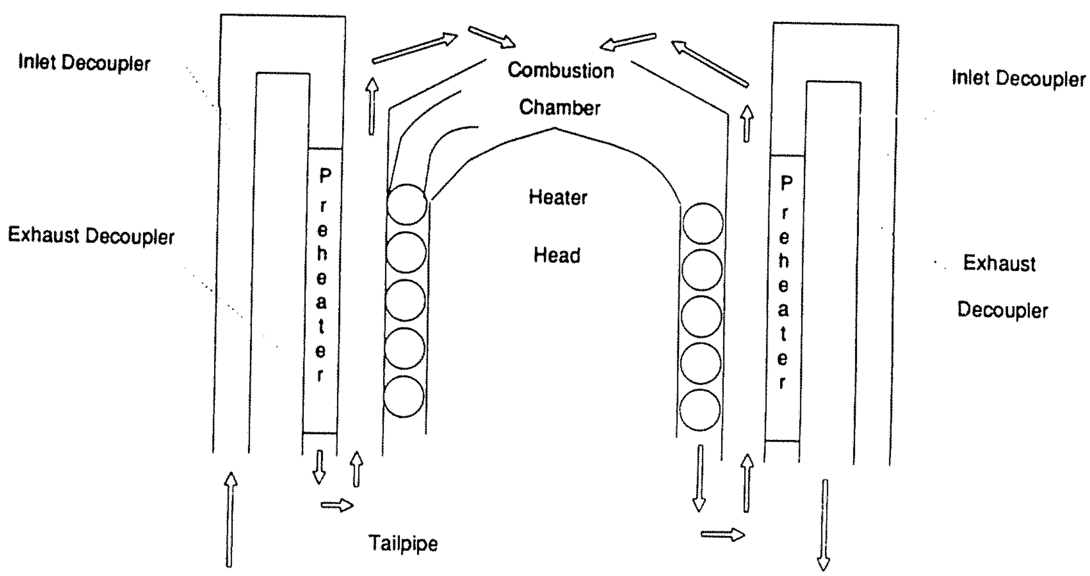


Fig. 4.14. Conceptual design of pulsed external heat system.

Table 4.8. Pulse combustor specifications

	Burner input (Btu/h)	
	30,000	10,000
Excess air (%)	40	40
Air flow (SCFH)	400	133
Gas flow (SCFH)	30	10
Mixer diameter (in.)	4	4
Mixer length (in.)	3	3
Combustion chamber height (in.)	1.7	1.7
Combustion chamber diameter (in.)	7.8	7.8
Combustion chamber volume (in.^{3})	83	83
Peak pressure in WC	27	13
Frequency (Hz)	52	29
Mean pressure limit in WC	2.4	0.8
Heater head ID (in.)	6.0	6.0
Heater head height (in.)	6.0	6.0
Tailpipe diameter (in.)	0.9	0.9
Number of tailpipes	2	2
Tailpipe length (in.)	72	72
Average recuperator temperature (°F)	917	825
Exhaust length limit (in.)	79	138
Average preheater temperature (°F)	787	761
Inlet length limit (in.)	75	134
Annulus I.D. (in.)	8.8	8.8
Annulus height (in.)	0.4	0.4
Exhaust decoupler ID (in.)	9.6	9.6
Exhaust decoupler OD (in.)	15.6	15.6
Inlet decoupler ID (in.)	15.6	15.6
Inlet decoupler OD (in.)	17.0	17.0
Decoupler height	10.0	10.0
EXH decoupler volume (in.^{3})	1238	1238
Inlet decoupler volume (in.^{3})	407	407
EXH sound pressure level (dB)	93	93
Inlet sound pressure level (dB)	85	85
Vent diameter (in.)	1.0	1.0
EXH decoupler frequency (Hz)	31	17

Table 4.8. (continued)

	Burner input (Btu/h)	
	30,000	10,000
Critical vent length (in.)	8	24
Vent length (in.)	90	90
Vent flow area (in.^{2})	0.79	0.79
Average vent temperature (°F)	368	262
Inlet diameter (in.)	1.5	1.5
Inlet equivalent diameter (in.)	1.1	1.1
Inlet hydraulic diameter (in.)	0.5	0.5
Inlet decoupler frequency (Hz)	31	17
Critical inlet length (in.)	27	87
Inlet length (in.)	90	90
Inlet flow area (in.^{2})	0.98	0.98
Average inlet temperature (°F)	211	171
Effectiveness (%)	65	70
Air inlet temperature (°F)	70	70
Preheater inlet temperature (°F)	351	272
Preheater outlet temperature (°F)	1222	1251
Combustor temperature (°F)	4024	4051
Heater head temperature (°C)	700	700
Recuperator inlet temperature (°F)	1333	1292
Recuperator outlet temperature (°F)	502	359
Exhaust temperature (°F)	234	166
Inlet/vent heat transfer (Btu/h)	2563	613
Decoupler heat transfer (Btu/h)	7942	2975
Heater head heat transfer (Btu/h)	25,732	8793
Stack loss (Btu/h)	4268	1207
Stack loss (%)	14	12

In the conceptual design analyzed in Table 4.8, the heater would consist of one or two tailpipes coiled about a cylindrical heater head (although other configurations might be equally or more practical). Although the structural aspects of this configuration have not been considered, the thermal and dynamic aspects appear feasible. The preheater/recuperator would be similar to the annular corrugated foil heat exchangers being developed for conventionally fired Stirling heat pumps. The decoupler chambers operate at relatively low temperatures and pressures and should represent no great technical risk or cost. None of these components represents substantially higher material costs than those of a conventionally fired EHS.

Although the technical analysis is somewhat encouraging, there are many unanswered questions. Virtually all pulse combustion systems developed have used ambient air, and this application would require substantial air preheating. Communication with individuals in several agencies provided insufficient information to determine the developmental difficulties in the design of a pulsed combustion system with preheated air. This approach would have to be rated as risky in that considerable development will be required to determine the feasibility. Another issue is that NO_x emissions from pulsed combustion systems have been on the order of 30 to 40 ppm without combustion modification such as EGR. One researcher has achieved lower levels using EGR but is still unpublished. Considering that these systems are not using preheated air, the levels are twice those expected for a transpiration pre-mixed burner. The pulsed combustion system, therefore, is not recommended for development based on the expected difficulty and cost of needed research and development coupled with the fact that it has not been tested with preheat.

4.7 COMBUSTION GAS RECIRCULATION

Until now, only the effects of EGR as a means of reducing NO_x have been considered. An alternative to EGR is CGR, which means recirculation of products of combustion from the exit of the heater head to the inlet of the combustor. The attraction of CGR is that unlike EGR, in which flow is added to the total mass flow in the preheater/recuperator and the blower, CGR involves no additional mass flow in any of these components. Thus when CGR is used, the mass loading of the preheater/recuperator is lower, resulting in either lower pressure drop, higher effectiveness, or a reduction of the surface area and/or flow area requirement. Furthermore, when EGR is used, because the preheater inlet temperature is increased, the final exhaust temperature is higher even if the effectiveness of the recuperator is held constant, resulting in greater stack loss.

While CGR has none of those disadvantages of EGR, its main drawback is that it requires a means of raising the pressure of the products of combustion leaving the heater head to the higher pressure of the air entering the combustor. Although a small blower could be used to raise the pressure, the elevated temperature (900-1400°F) makes that solution costly and impractical. The preferred approach is to accelerate the combustion-air stream leaving the preheater in a nozzle by aspirating the CGR stream in an ejector. The pressure drop in the ejector (due to both reversible momentum transfer to the CGR stream and to irreversible mixing and recovery losses) must be provided through increased head rise in the blower.

The MTI EHS design was analyzed to determine the influence of CGR. Although there are many bases of comparison, the approach used was to keep the design parameters of component mass velocity and NTUs similar and to evaluate the influence of switching from EGR to CGR while maintaining the same NO_x formation rate. The comparative results are in the form of changes in heat exchanger area, stack loss, and blower power.

The analytical results are presented in Table 4.9. In the MTI case, the CGR flow must be 31.8% of total air flow to produce the same NO_x as 30% EGR flow. As a result of the lower preheater inlet temperature, the final exhaust temperature is 36°F lower, resulting in about a 1% improvement in combustion efficiency.

Table 4.9. Effect of combustion gas recirculation

	A: MTI[a] base With EGR[b] (No CGR[c])	B: MTI[a] With CGR[c] (No EGR[b])
Cycle heat input (BTU/h)	23,420	23,420
Inlet temperature (°F)	95	95
Excess air at blower inlet (%)	25	25
Preheater size (NTU[d])	14.60	14.60
Preheater mass velocity (#/h/ft^2)	837	837
Recuperator size (NTU[d])	15.12	15.09
Recuperator mass velocity (#/h/ft^2)	868	878
Leakage (preheater to recuperator) (% air)	0.0	0.0
EGR[b] (exh blr to inlet) (% air F)	30.0	0.0
CGR[c] (heater head to combustor) (% air F)	0.0	31.8
Average heater head temperature (°C)	700	700
Heater head gas-side size (NTU[d])	2.55	2.55
Heater head mass velocity (#/h/ft^2)	3,500	3,500
Heater head fin efficiency (%)	90	90
Heater head firing rate (BTU/h)	29,054	28,665
Combustor inlet temperature (°F)	1,365	1,373
Combustor outlet temperature (°F)	3,496	3,497
Recuperator inlet temperature (°F)	1,462	1,462
Exhaust temperature (°F)	395	359
Average recuperator hot end temperature (°F)	1,413	1,418
d (NO)/dt (PPM/s)	9.64E+04	9.68E+04
Stack loss (HHV[e]) (%)	19.39	18.30
Combustor oxygen (%)	4.00	4.00
Combustor excess air (%)	32.15	32.58
Stack oxygen (%)	4.00	4.00
Preheater pressure drop (in WC)	0.12	0.11
Recuperator pressure drop (in WC)	0.15	0.15
Heater head pressure drop (in WC)	1.54	3.08
Blower theoretical power and inlet (W)	1.84	2.28
Blower power at 5 × theo. (W)	9.20	11.41
Carnot × comb efficiency (%)	55.07	55.82
Shaft power at 50% Carnot (W)	2,344	2,344

[a]MTI = Mechanical Technology, Inc.
[b]EGR = exhaust gas recirculation
[c]CGR = combination gas recirculation
[d]NTU = number of transfer units
[e]HHV = higher heating value

The analysis of the MTI case (which takes into account the heater head pressure drop but not the combustor pressure drop) shows that the blower power increases by about 25%; but because the operating cost of the blower is small compared with the cost of the fossil input, the overall operating cost should be about 1% lower because of the improved combustion efficiency.

Because of the high effectiveness of the baseline MTI preheater, the inlet temperature to the combustor is insensitive to the choice of EGR versus CGR. As a result, the amount of CGR is only about 6% higher than the EGR required to produce the same NO_x.

The decision whether to use CGR rather than EGR is thought to be unclear. The application of EGR is straightforward and an easy part of the development program. The design of a CGR system, however, is much more difficult and will result in hardware inflexibility if conditions change. CGR should be reconsidered after the other technical issues are resolved and the product enhancements are focusing on small improvements.

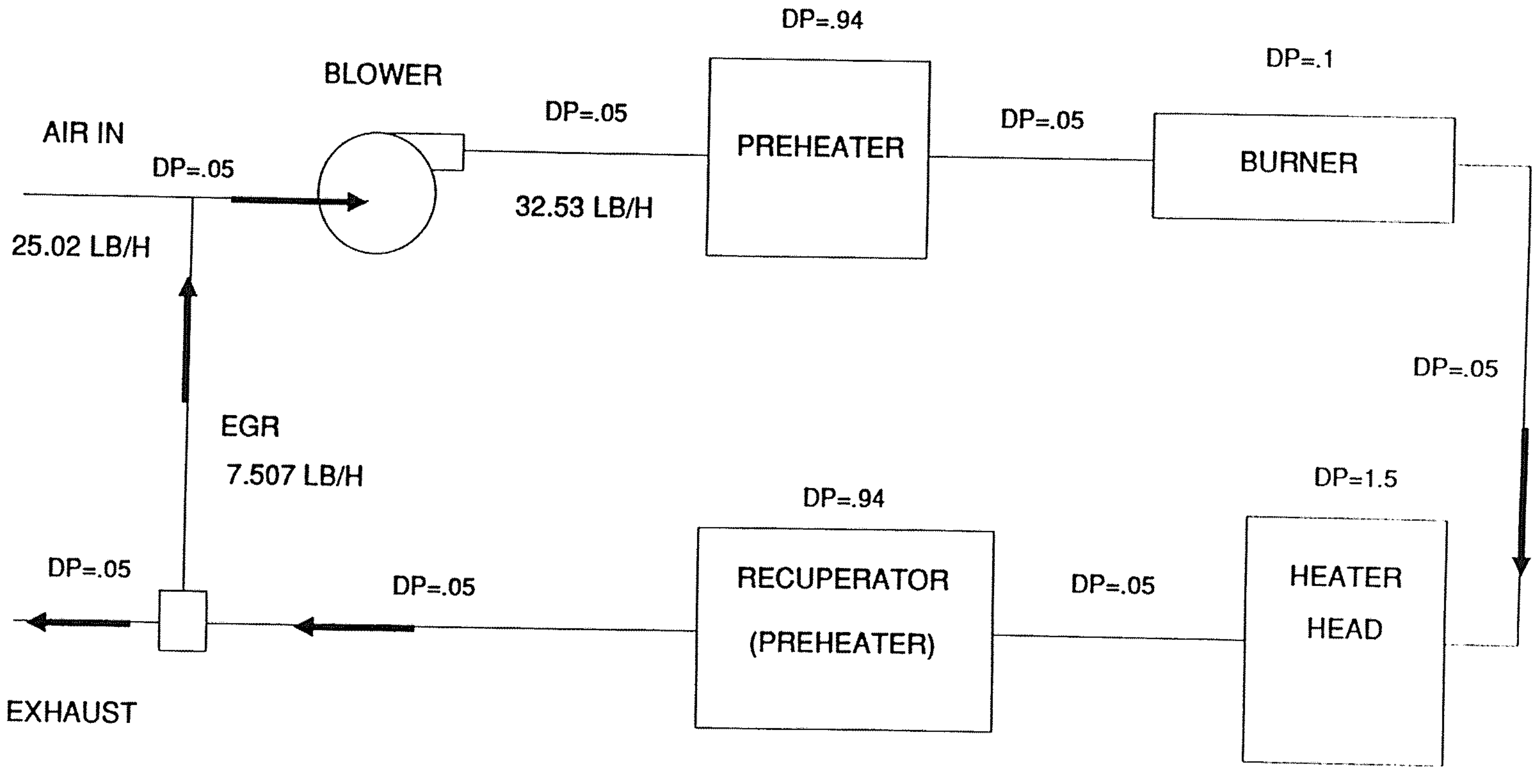
4.8 CONTROL SYSTEM

4.8.1 DIFFUSION FLAME-BASED EHS CONTROL

The basic requirements of the EHS system are control of the air/fuel ratio consistent with efficiency and emission requirements, along with variation in the fuel input to meet the thermal demands of the system. According to MTI, the turndown required for the burner is 3:1 for performance and 5:1 for light-off, with a continuously variable firing rate. The 5:1 turndown requirement at start-up is for a spark ignition system used by AMTI, which probably will be replaced by a hot-surface ignition system.

The design of the control system for the diffusion flame EHS begins with the examination of the air flow through the system. Air flow is shown schematically in Fig. 4.15, which shows the pressure drop component by component for the system design. The design pressure drop at steady state full load conditions is 3.83 in. of water at a flow of 25.02 lb/h.

In general, it is desirable to have a larger pressure drop in the colder parts of a flow system rather than the heated parts; this minimizes flow variation due to temperature changes, especially at start-up. In the case of the EHS system shown in Fig. 4.15, the pressure drops are in the preheater/recuperator and in the heater head; as a result, the difference in flow from "cold" start-up to "hot" steady conditions could be substantial. This is illustrated in Fig. 4.16, where pressure drop curves are plotted on a typical blower curve. At the system design conditions, the pressure drop is 3.8 in. of water at about 9.5 ft^3/min, while at the cold condition, it is 3.7 in. of water at about 11 ft^3/min. Since the density of room temperature air is 0.075 lb_m/ft^3, compared with 0.061 lb_m/ft^3 at the 195°F design condition, the ratio of cold to hot flow is based not only on volume flow but also on the ratio of volume flow times the density. Therefore, the change in flow is about 40% in going from hot to cold. At the same time, the natural gas flow will not change because its major pressure drop is across a regulator and an orifice.



TOTAL PRESSURE DROP = 3.83 INCHES OF WATER

Fig. 4.15. Mechanical Technology, Inc., external heat system flow-pressure drop.

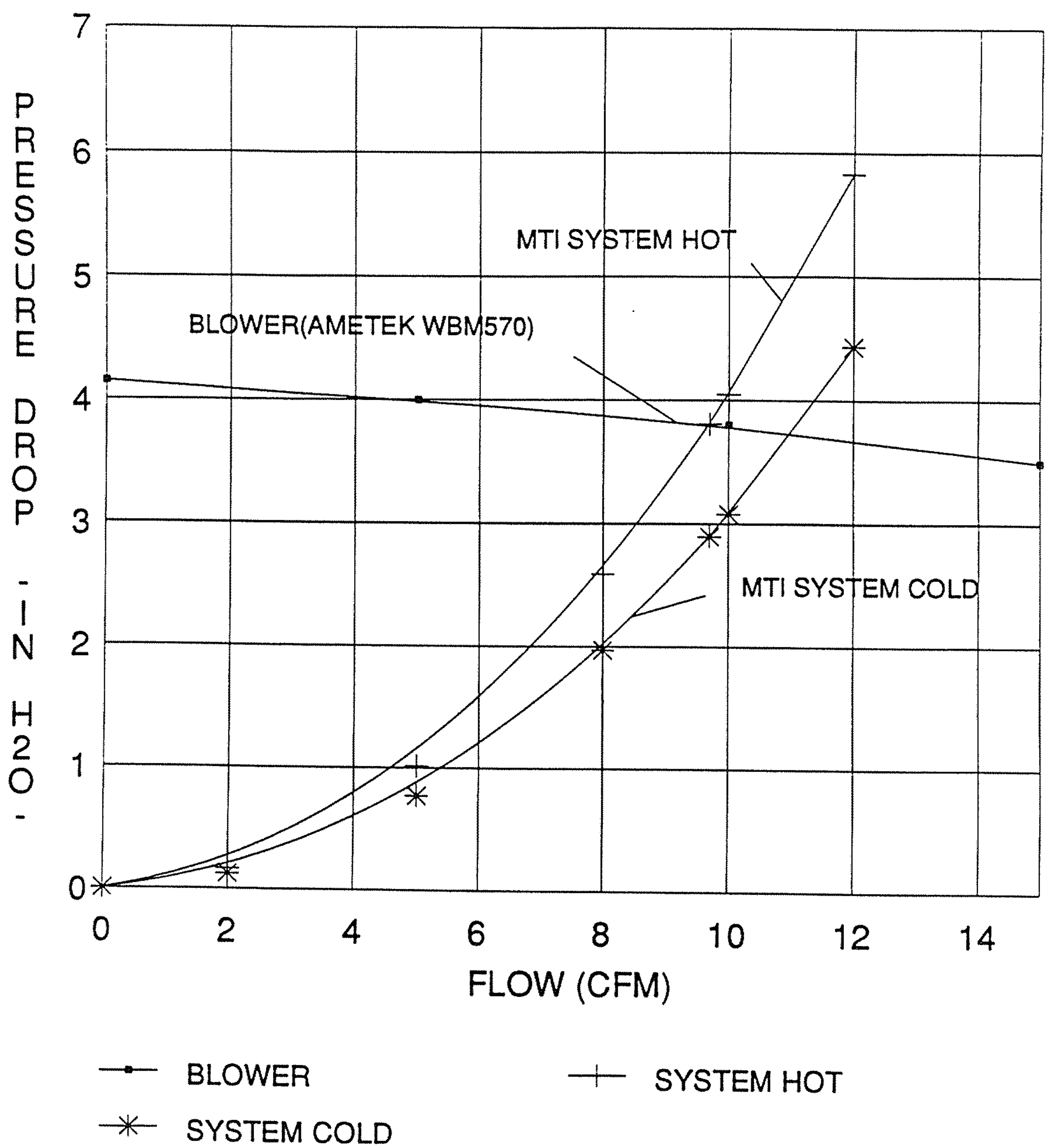


Fig. 4.16. Effect of temperature on system pressure drop.

It is unlikely that the burner can handle such a wide swing in air/fuel ratio. Design problems and increased costs would result if a sufficiently large pressure drop were introduced in the colder regions to decrease the variation. The possibility of decreasing the existing pressure drops to the point where they do not predominate is unlikely because this would require making the preheater/recuperator much larger in cross-section, which would make the cost of the preheater unacceptable.

The most likely possibility will be the use of some software in the microprocessor control in the engine system to provide temperature compensation with a mechanical "choke."

The control system used by MTI for the EHS is shown in Fig. 2.8. It is not discussed further because it is a laboratory control rather than production control system. The objective of either a laboratory control or a production control system is the same, however, as both control fuel/air ratio and firing rate in response to thermal management of the system.

Table 4.10 summarizes the control systems and issues that were considered in this work. The systems are discussed further in the following paragraphs.

On-off control system. The simplest approach is an on-off control system. It is the type of system most appliance manufacturers use, and is generally the least expensive. However, certain requirements for this application may make such a system unacceptable. The turbulent diffusion flame combustor may need an automated start-up procedure for proper light-off, and it seems that some modulation also is necessary. During part load operation, the added response time of a start-up procedure as the unit cycles is probably undesirable. Some further control system development work probably is needed to address the ignition and response time concerns regarding the on-off system.

Blower speed control system. Another type of system considered was based on controlling the speed of the blower motor to control air flow. This type of control has many advantages, including excellent response time, low pressure drop, and low parasitic

Table 4.10. Control strategy options

System type	Advantages	Disadvantages
1.0 On/off	Inexpensive Simple	Slow response Start-up problems
2.0 Speed control	Fast response Low pressure drop Low parasitic power	High cost
3.0 Throttling	Fast response Moderate cost	Medium pressure drop Start-up problems
4.0 Modified mixing	Fast response Moderate cost No start-up problems	Medium pressure drop

power. The disadvantage is that it offers either low cost and short life, or high cost and long life. For simple speed control, a dc motor is required, so that varying voltage results in a variable speed. A low-cost universal ac/dc motor, such as those found in vacuum cleaners, has brushes that can wear out and is limited to a 1000-h life before it has to be repaired; this is unacceptable. An applicable brushless dc motor exists but is quite expensive. A blower manufacturer quoted a price of $150 each for a blower/motor/control for 10,000 units per year.

A capacitor start motor with frequency control also can provide speed control. However, although the cost of such a motor was reasonable, the frequency control even for fractional horsepower motors (1/15–1/20) was very expensive. It had a list price of $300, which would probably result in a manufacturer's price of over $100. A multi-speed blower motor also was examined for speed control. While these motors are available and inexpensive enough to be commonly used in applications for stepping air flow, it appears they are limited to turndown ratios of 2:1 to 3:1.

The conclusion was that the use of speed control for the combustor control system appears to be too expensive or limited in turndown.

Throttle control system. An alternative to speed control is to use throttling to modulate the air and fuel control. A schematic of this system is shown in Fig. 4.17. The interface between the Stirling engine controls and the combustion system will be a burner control. It will have a temperature input (probably heater head temperature) and heat pump feedback from the microprocessor for the start-up algorithm, firing rate, etc. The burner control will supply voltage inputs to the air control and fuel control to vary firing rate and modulate air/fuel ratio. The burner control will also provide ignition and flame safety. This type of burner control exists for on-off and step controls but will have to be developed for the specific EHS controls.

It was not possible to find "off-the-shelf" controls for this system that were suitable for production evaluation. However, items sufficiently similar to the required items to do a cost-effectiveness evaluation were found. The air control would be a butterfly valve operated by a rotary solenoid. The solenoid type selected is manufactured by Ledex Inc. A completely suitable gas control valve was not found, but Maxitrol, Inc., manufactures a modulating gas valve (MR410) that has a voltage as an input. Maxitrol offers an amplifier for the control of this valve which is very expensive. It should not be difficult to integrate this valve into the system microprocessor. This valve also has a built-in regulator but does not include a stop or shutoff valve. The EGR control is expected to be only an orifice working with the air flow to control emissions. It is not expected that an active EGR control will be required. This control system is a good candidate for the diffusion flame based EHS. Its estimated cost is $174 (see Table 4.11), and its installed cost is $417.

Modified mixing control system. Finally, a control system is presented in Fig. 4.18 that includes the cost-effectiveness of the step control with a method of overcoming the "cold/hot" start-up problem. This system uses a special regulator with a double diaphragm that not only regulates the gas pressure, but also compensates for the variation in flow when the pressure drop across the preheater changes as the flow goes from hot to cold. The equation for pressure regulation at the outlet of the regulator was

$$P_8 = P_2 - (P_3 - P_5) \,. \qquad (5)$$

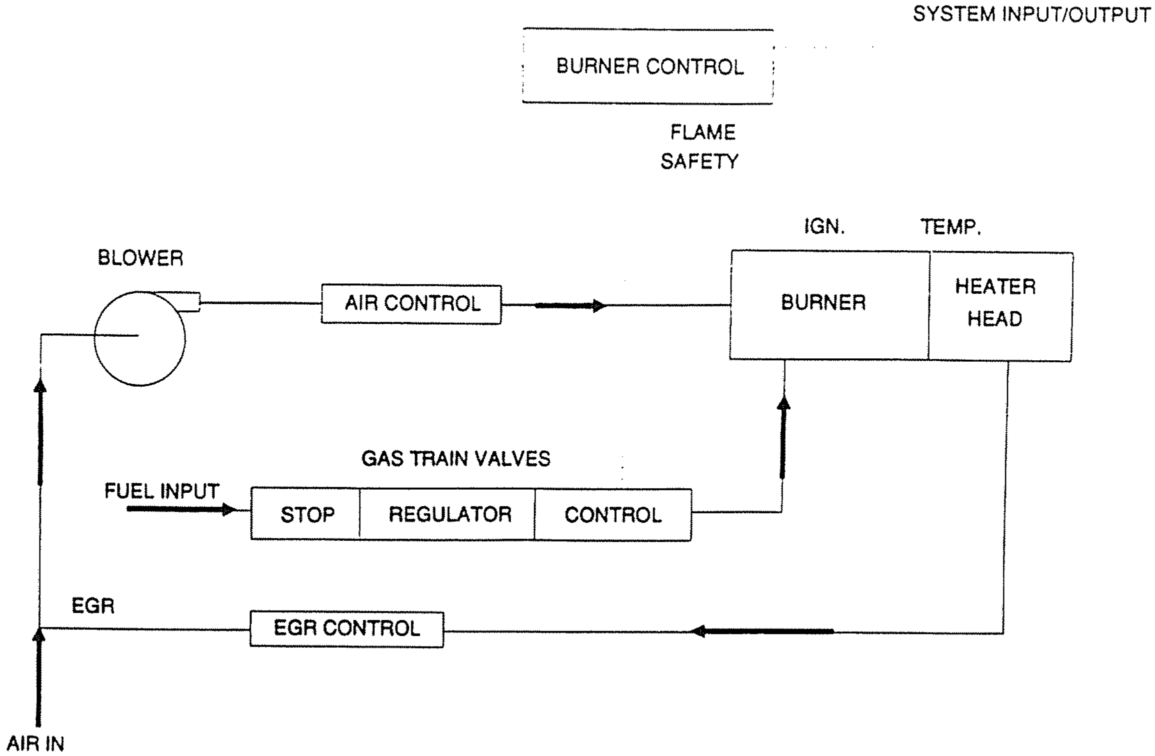


Fig. 4.17. Diffusion burner throttle control system.

Table 4.11. External heat source control costing

Item	Diffusion burner control system	Transpiration-cooled burner control system
Blower	$ 71.25	$ 71.25
Control board	31.25	31.25
Air valve	22.50	22.50
Gas valve	42.50	34.00
Stop valve	6.25	
Total cost	173.75	159.00
Mark-up	× 2.4	× 2.4
Installed cost	417.00	381.60

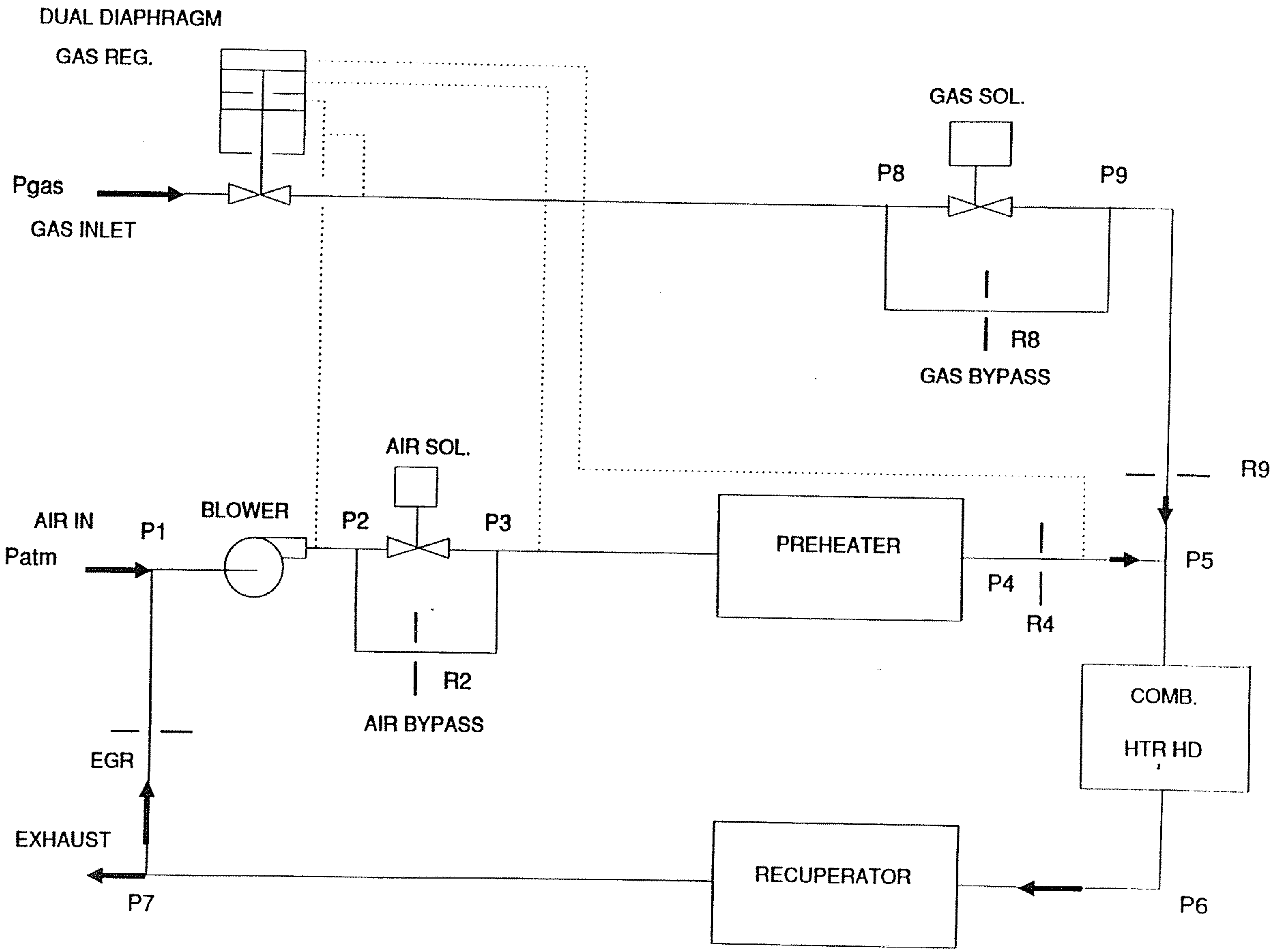


Fig. 4.18. Air/fuel control with dual function regulator.

When the relationships for the various resistances were analyzed, the resultant relationship evolved:

$$W_a^2/W_g^2 = (R_8 + R_9)/R_2 , \tag{6}$$

where:

W_a = air flow,
W_g = gas flow,
R_8 = gas orifice resistance,
R_9 = gas injector/nozzle resistance,
R_2 = air orifice resistance.

Equation 6 says the ratio of gas to air flow is constant when the included resistances are constant. These are all in cold regions and are not temperature dependent, and thus solve the heat up problem at start-up.

The scheme shown uses a step type control system. At high fire, the gas solenoid and gas bypass would be open, as would the air solenoid and orifice. When low fire is desired, the solenoids are closed and the orifices handle the low fire condition. The low fire would be set just below the minimum input of the system. The solenoid valves need not be elaborate since internal leakage is not critical. The entire assembly can be incorporated into a single unit to handle the gas and air. The throttling control of the previous system might also be used for this system; however, the step firing rate would be the first choice and would be abandoned only if the system required a better response time.

The cost of this control system is expected to be about the same as for the throttle control ($417, see Table 4.11). However, it is expected to offer much simpler operation and require less development than the system shown in Fig. 4.17.

4.8.2 Control For Transpiration-Cooled Burner

A control system design for premixed combustion is shown in Fig. 4.19. The system uses an air and gas orifice to regulate the flow into the burner. A special regulator is used that is referenced to the outlet of the blower so that the proportion of air and gas is constant. The system has some turndown capability, on the order of 3:1. The wider 5:1 turndown is not required because a hot surface ignitor is used rather than a spark ignitor. If further turndown is necessary, the system can be modified with second orifices and valves similar to those in Fig. 4.18 to allow step control of the system.

The costs for the transpiration burner EHS control system are shown in the second column of Table 4.11. The cost is $159; if a factor of 2.4 is applied to calculate the installed cost, then the total cost is $381.60.

4.9 TOTAL EHS COSTS

In previous sections, the costs have been examined only for each subsystem. In this section, the costs of the various systems will be combined to provide complete costs for the EHS. Table 4.12 compares the costs of three separate systems: (1) a diffusion flame

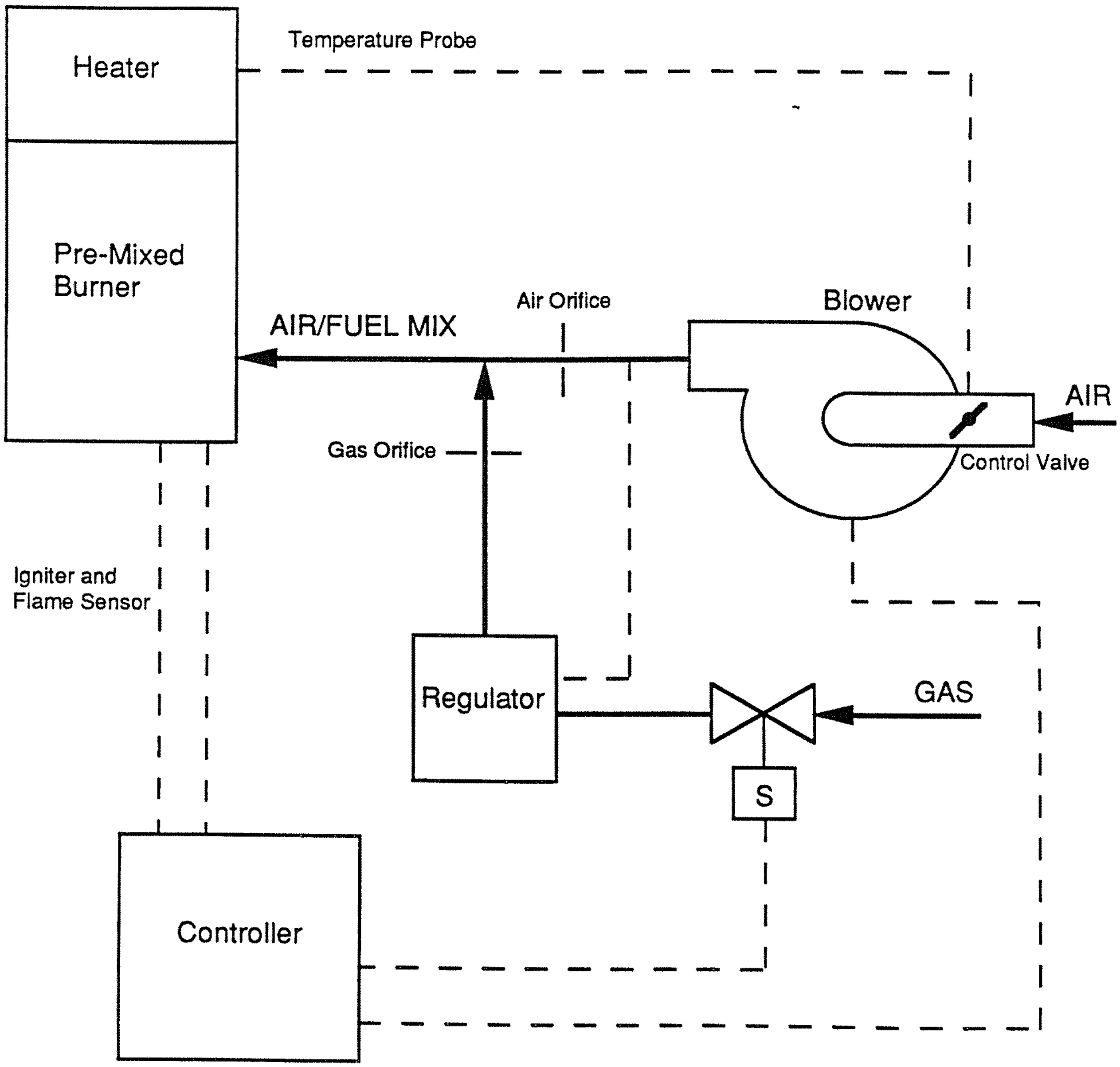


Fig. 4.19. Typical pre-mixed burner control system.

Table 4.12. External heat source assembly and control costs

Item	Diffusion flame: 310 SST	Diffusion flame: 304 SST	Transpiration-cooled: Aluminum-coated steel
Preheater NTU[a]	14.6	10.0	3.0
EHS[b] assembly cost	$147.14	$75.00	$57.50
Control cost	173.75	173.75	159.00
Total cost	$320.89	$248.75	$216.50
Mark-up	× 2.4	× 2.4	× 2.4
Installed cost	$770.14	$597.00	$519.60

[a]NTU = number of transfer units.
[b]EHS = external heat source.

EHS with a 14.6 NTU, 310 SST preheater; (2) a diffusion flame EHS with a 10-NTU 304 SST preheater; and (3) a transpiration-cooled burner with a 3-NTU aluminum-coated steel preheater.

The payback analysis was reexamined for the EHS, including control system costs. There was no change from the previous analysis that considered only the EHS without the controls.

A direct comparison of the diffusion burner and transpiration burner EHS systems involves many issues beyond the scope of this study, such as the cost of the heater head and other SEHP components and the limitations on preheated air for the transpiration burner case. Both concepts should be given further consideration.

5. CONCLUSIONS AND RECOMMENDATIONS

5.1 GENERAL FINDINGS

1. Two fairly different but viable EHS design approaches were found in the state-of-the art review. The major differences between the designs stem from the very different combustion systems chosen: one based on a turbulent diffusion burner and the other based on a transpiration burner using premixed fuel and air.

2. New and/or novel design concepts were explored and reviewed, and the more conventional design concepts were examined parametrically. The result of these efforts was significant improvements and partial optimization of the two major EHS designs.

3. The diffusion burner EHS design is suited for higher air preheat temperatures and therefore superior energy efficiency. This design is the more capital-intensive of the two designs, partly because of more stringent materials requirements associated with higher temperatures.

   The transpiration EHS design can be viewed as a somewhat lower technology approach that uses more currently available components and can make more use of less expensive materials. Although this system appears less costly to manufacture (compared with the diffusion burner EHS design), the system fuel consumption will be significantly higher.

4. Both EHS design paths were found to be worthy of further consideration. It was not possible to eliminate one of the design choices based on the available information. A number of issues could be explored to aid in making the choice between designs, but these are beyond the scope of this work and/or require more development work.

5. This study did not point to any other significantly different design alternatives that seem appropriate to pursue at this time. Other approaches would require much development work, and the associated technical and cost risks are perceived as rather high considering that the goal is to produce an SEHP appliance unit.

5.2 TURBULENT DIFFUSION BURNER EHS DESIGN

The turbulent diffusion burner EHS design features a highly preheated air system that can be costly. The findings of this work indicate that preheater temperatures should be limited to a range where use of steel with a lower nickel content is possible. Specifically, the use of 304 rather than 310 stainless steel is recommended as the preheater construction material, which requires that maximum metal temperatures be limited to about 1390°F. Making the preheater from aluminum-coated steel, a lower cost and lower temperature material, appears not to be cost effective (in conjunction with the turbulent diffusion burner). The reduced preheater temperatures for this design result in a relatively large increase in fuel use.

Another conclusion regarding the preheater is that the type of preheater design and construction specified by MTI appeared to be the best design available and could not be improved upon significantly. Construction techniques and cost issues appear already to have been thoroughly explored for this stage of development.

It is also felt that a combustion chamber design that requires nickel-based alloys (see Fig. 4.4) such as Haynes 214 for a combustion zone liner should be avoided if possible. Such a combustion zone liner only appears to be needed if a monolithic heater head design is to be employed. The use of vacuum-formed high-temperature insulation may be a possible alternative.

One unresolved issue that was not included as part of this study, but that was difficult to ignore, was the issue of whether a tubular or monolithic heater head is the better choice technically and economically. It was beyond the scope of this study for AMTI to do such an analysis. However, it should be done at some time and should include the effect both designs will have on the EHS system and the balance of the SEHP. The monolithic heater head will require a combustion chamber that will need to duct combustion products up to and then away from the heater head. The cost of these added materials may be high. Another design difference is that the monolithic head design would use the preheated air stream to cool the high-temperature combustor liner. This process would accomplish some of the air preheating so that a smaller preheater would be used.

In examining the controls, it was determined that blower speed control is too expensive. Two control systems were explored that appear to be applicable and more cost effective.

5.3 PRE-MIXED TRANSPIRATION BURNER EHS DESIGN

The EHS design based on a pre-mixed transpiration burner appears to be at least marginally competitive with the turbulent diffusion burner EHS design. Although this type of system has inherently lower efficiency, the manufacturing cost is significantly lower. Furthermore, transpiration burners and their controls have been used extensively in appliance application and have good emissions performance, although applications with relatively high air preheat have not been commercialized.

5.4 REGULATORY CODE ISSUES

Several areas were identified in the ANSI furnace code that need to be applied to the EHS design, especially maximum temperature/material and minimum/thickness material combinations. Additionally, there are emission limits of CO that should be used to define combustor performance, and control issues and recommendations were also made for NO_x emissions in light of the California (SCAQMD) standards.

5.5 RESEARCH ISSUES

Although several research issues could be identified for investigation, the most important research issue is to understand the limitations of using preheated air for the premixed transpiration burner system. It is unclear what temperature of preheated air can be used without flashback or auto-ignition problems. Furthermore, the emissions characteristics of a transpiration burner using preheated air are unknown. Experimental data on these issues are needed for further comparison of the premixed transpiration burner and the turbulent diffusion burner EHS designs.

5.6 MARKET PENETRATION

Market penetration can be difficult for a new appliance that costs considerably more than the existing technology. It is hard to sell an appliance on efficiency alone if it exceeds some reasonable cost limit (on the order of 40% more than the technology it would replace, according to AMTI experience). Payback becomes secondary beyond some point if additional consumer features are not also included. It would seem that some limit on total cost should be an objective for designing the SEHP.

At one time, AMTI tried to develop a residential water heater that cost almost twice as much as the "builder's special" glass-lined unit. The attempt was made with a manufacturer that felt the energy savings and durability of the new heater would overcome the increased price. The product was never commercialized, primarily because of its price. Although the payback and life cycle costs were acceptable, the total installed price was not.

This situation also shows that certain "encouraging" regulations may need to be put into place for more capital-intensive but fuel-efficient appliances to compete in the marketplace.

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Appendix A

A.1 TYPICAL SPREADSHEET MODEL OUTPUT

CASE	A	B	C	D
	MTI			
COSTING:	IECEC			
REGEN COST/SQ-FT - $/SQ-FT	$1.00	$1.00	$1.00	$1.00
HTR HEAD COST/SQ-FT - $/SQ-FT	$10.00	$10.00	$10.00	$10.00
PREHTR HYDR DIAM - FT	0.011	0.011	0.011	0.011
RECUP HYDR DIAM - FT	0.011	0.011	0.011	0.011
HTR HEAD HYDR DIAM - FT	0.003	0.003	0.003	0.003
PREHTR SURFACE AREA - FT^2	11.64	11.64	11.28	10.91
RECUP SURFACE AREA - FT^2	11.66	11.66	11.29	10.92
HTR HEAD SURFACE AREA - FT^2	0.78	0.78	0.78	0.79
PREHTR LENGTH - FT	0.79	0.79	0.76	0.73
RECUP LENGTH - FT	0.79	0.79	0.76	0.73
HTR HEAD LENGTH - FT	0.06	0.06	0.06	0.06
REGENERATOR COST	$11.64	$11.64	$11.28	$10.91
HTR HEAD COST	$7.82	$7.82	$7.83	$7.85
TOTAL EHS H-X COST	$19.46	$19.46	$19.11	$18.76
REGEN PAYBACK VS BASE - HRS	BASE	ERR	1097	1061
CYCLE HEAT INPUT - BTU/HR	23420	23420	23420	23420
INLET TEMPERATURE - DEG F	95	95	95	95
EXCESS AIR @ BLOWER INLET- %	25%	25%	25%	25%
PREHEATER NTU	14.60	14.60	14.10	13.60
PREHTR MASS VELOCITY - #/HR-FT^	837	837	837	837
RECUPERATOR NTU	15.14	15.14	14.65	14.16
RECUP MASS VELOCITY - #/HR-FT^2	868	868	868	868
LEAKAGE (PRHTR TO RECUP) - % AI	0.0%	0.0%	0.0%	0.0%
EGR (EXH BLR TO INLET) - % AIR	30.0%	30.0%	30.0%	30.0%
CGR (HTR HEAD TO COMB) - % AIR	0.0%	0.0%	0.0%	0.0%
AVG HTR HEAD TEMP - DEG C	700	700	700	700
HTR HEAD GAS-SIDE NTU	2.55	2.55	2.55	2.55
HTR HEAD MASS VEL - #/HR-FT^2	3500	3500	3500	3500
HTR HEAD FIN EFFIC - %	90%	90%	90%	90%
HHV FIRING RATE - BTU/HR	29028	29028	29084	29144
COMBUSTOR INLET TEMP - DEG F	1367	1367	1362	1357
COMBUSTOR OUTLET TEMP - DEG F	3498	3498	3494	3490
RECUP INLET TEMP - DEG F	1462	1462	1462	1462
EXHAUST TEMP - DEG F	392	392	397	402
AVG RECUP HOT END TEMP - DEG F	1415	1415	1412	1409
d(NO)/dt - PPM/SEC	9.79E+04	9.79E+04	9.49E+04	9.18E+04
STACK LOSS (HHV) - %	19.32%	19.32%	19.47%	19.64%
COMB. PCT OXYGEN - %	4.00%	4.00%	4.00%	4.00%
COMB EXCESS AIR - %	32.16%	32.16%	32.16%	32.16%
STACK PCT OXYGEN - %	4.00%	4.00%	4.00%	4.00%
PREHTR PRESS DROP - IN WC	0.12	0.12	0.11	0.11
RECUP PRESSURE DROP - IN WC	0.15	0.15	0.14	0.14
HTR HEAD PRESSURE DROP - IN WC	1.54	1.54	1.54	1.54
BLOWER THEO. POWER @ INLET - W	1.84	1.84	1.84	1.83
BLOWER POWER @ 5X THEO - W	9.19	9.19	9.18	9.17
CARNOT X COMB EFFIC. - %	55.12%	55.12%	55.02%	54.90%
SHAFT POWER @ 50% CARNOT - W	2344	2344	2344	2344
GAS COST @ $6.00/MMBTU - $/KWsh	$0.0743	$0.0743	$0.0744	$0.0746
BLOWER COST @ $18/MMBTU -$/KWsh	$0.0002	$0.0002	$0.0002	$0.0002

TOTAL POWER COST - $/KWsh	$0.0745	$0.0745	$0.0747	$0.0748
::				
INLET BLOWER:				
HEAT INPUT - BTU/HR				
INLET FLOW RATE - #/HR	25.21	25.21	25.26	25.31
EGRIN - #/HR	7.6	7.6	7.6	7.6
LEAKIN - #/HR				
TOTAL FLOW IN - #/HR	33	33	33	33
TOTAL FLOW OUT - #/HR	33	33	33	33
LEAKOUT - #/HR				
EGROUT - #/HR				
OUTLET FLOW RATE - #/HR	33	33	33	33
INLET ENTHALPY - BTU/#	29	29	29	29
EGRINHIN - BTU/#	133	133	134	136
LEAKINHIN - BTU/#				
LEAKOUTHOUT - BTU/#	41	41	41	42
OUTLET ENTHALPY - BTU/#	53	53	53	54
INLET TEMPERATURE - DEG F	95	95	95	95
OUTLET TEMPERATURE - DEG F	169	169	170	172
INLET INERT - PCT	0.00%	0.00%	0.00%	0.00%
OUTLET INERT - PCT	18.68%	18.68%	18.68%	18.68%
::				
AIR PREHEATER:				
HEAT INPUT - BTU/HR	12313	12313	12275	12235
INLET FLOW RATE - #/HR	33	33	33	33
EGRIN - #/HR				
LEAKIN - #/HR				
TOTAL FLOW IN - #/HR	32.78	32.78	32.84	32.91
TOTAL FLOW OUT - #/HR	32.78	32.78	32.84	32.91
LEAKOUT - #/HR	0.00	0.00	0.00	0.00
EGROUT - #/HR				
OUTLET FLOW RATE - #/HR	33	33	33	33
INLET ENTHALPY - BTU/#	53	53	53	54
EGRINHIN - BTU/#				
LEAKINHIN - BTU/#				
LEAKOUTHOUT - BTU/#	241	241	240	240
OUTLET ENTHALPY - BTU/#	429	429	427	426
INLET TEMPERATURE - DEG F	169	169	170	172
OUTLET TEMPERATURE - DEG F	1367	1367	1362	1357
INLET INERT - PCT	18.68%	18.68%	18.68%	18.68%
OUTLET INERT - PCT	18.68%	18.68%	18.68%	18.68%
NTU(COLD)	14.60	14.60	14.10	13.60
NTU(HOT)	15.14	15.14	14.65	14.16
NTU(TOTAL)	7.84	7.84	7.58	7.32
WC(COLD)	10.28	10.28	10.30	10.32
WC(HOT)	11.51	11.51	11.53	11.55
WC(MIN)	10.28	10.28	10.30	10.32
WC(MAX)	11.51	11.51	11.53	11.55
EFFECTIVENESS	0.925	0.925	0.921	0.917
TCIN	169	169	170	172
TCOUT	1365	1365	1360	1355
THIN	1462	1462	1462	1462
THOUT	394	394	399	404
AVG VISC - #/HR-FT	0.080	0.080	0.080	0.080
PREHTR REYNOLDS NUMBER	111	111	111	111

PREHTR J FACTOR	0.039	0.039	0.039	0.039
PREHTR F/J	2.0	2.0	2.0	2.0

: :

COMBUSTOR:

HEAT INPUT - BTU/HR	29028	29028	29084	29144
INLET FLOW RATE - #/HR	33	33	33	33
EGRIN - #/HR	0.0	0.0	0.0	0.0
LEAKIN - #/HR				
TOTAL FLOW IN - #/HR	33	33	33	33
TOTAL FLOW OUT - #/HR	34	34	34	34
LEAKOUT - #/HR				
EGROUT - #/HR				
OUTLET FLOW RATE - #/HR	34	34	34	34
INLET ENTHALPY - BTU/#	429	429	427	426
EGRINHIN - BTU/#	495	495	495	495
LEAKINHIN - BTU/#				
LEAKOUTHOUT - BTU/#	806	806	805	803
OUTLET ENTHALPY - BTU/#	1184	1184	1182	1181
INLET TEMPERATURE - DEG F	1367	1367	1362	1357
OUTLET TEMPERATURE - DEG F	3498	3498	3494	3490
INLET INERT - PCT	18.68%	18.68%	18.68%	18.68%
OUTLET INERT - PCT	80.93%	80.93%	80.93%	80.93%
EXCESS AIR	32.16%	32.16%	32.16%	32.16%
HHV - BTU/#	23617	23617	23617	23617
LHV - BTU/#	21255	21255	21255	21255
DENSITY #/FT^3	0.05	0.05	0.05	0.05
THEO. A/F #/#	16.41	16.41	16.41	16.41
THEO. FLAME TEMP - DEG F	3600	3600	3600	3600
COMB PROD SPEC HEAT - BTU/#-F	0.346	0.346	0.346	0.346
AIR SPECIFIC HEAT - BTU/#-F	0.31	0.31	0.31	0.31
FUEL FLOW	1.229	1.229	1.231	1.234

: :

HEATER HEAD:

HEAT INPUT - BTU/HR	-23428	-23428	-23429	-23429
INLET FLOW RATE - #/HR	34.0	34.0	34.1	34.1
EGRIN - #/HR				
LEAKIN - #/HR				
TOTAL FLOW IN - #/HR	34.0	34.0	34.1	34.1
TOTAL FLOW OUT - #/HR	34.0	34.0	34.1	34.1
LEAKOUT - #/HR				
EGROUT - #/HR	0.0	0.0	0.0	0.0
OUTLET FLOW RATE - #/HR	34.0	34.0	34.1	34.1
INLET ENTHALPY - BTU/#	1183.6	1183.6	1182.2	1180.7
EGRINHIN - BTU/#				
LEAKINHIN - BTU/#				
LEAKOUTHOUT - BTU/#	839.2	839.2	838.4	837.6
OUTLET ENTHALPY - BTU/#	494.7	494.7	494.6	494.5
INLET TEMPERATURE - DEG F	3498	3498	3494	3490
OUTLET TEMPERATURE - DEG F	1462	1462	1462	1462
INLET INERT - PCT	80.93%	80.93%	80.93%	80.93%
OUTLET INERT - PCT	80.93%	80.93%	80.93%	80.93%
NTU(COLD)	2.55	2.55	2.55	2.55

NTU(HOT)	2.55	2.55	2.55	2.55
NTU(TOTAL)	2.52	2.52	2.52	2.52
WC(COLD)	1150.61	1150.61	1152.83	1155.21
WC(HOT)	11.51	11.51	11.53	11.55
WC(MIN)	11.51	11.51	11.53	11.55
WC(MAX)	1150.61	1150.61	1152.83	1155.21
EFFECTIVENESS	0.92	0.92	0.92	0.92
TCIN	1282	1282	1282	1282
TCOUT	1302	1302	1302	1302
THIN	3498	3498	3494	3490
THOUT	1462	1462	1462	1462
AVG HTR HEAD TEMP - DEG F	1292	1292	1292	1292
AVG HTR HEAD VISC - #/HR-FT	0.098	0.098	0.098	0.098
HTR HEAD REYNOLDS NUMBER	107	107	107	107
HTR HEAD J FACTOR	0.028	0.028	0.028	0.028
HTR HEAD F/J	5.5	5.5	5.5	5.5

: :

RECUPERATOR:				
HEAT INPUT - BTU/HR	-12313	-12313	-12275	-12235
INLET FLOW RATE - #/HR	34	34	34	34
EGRIN - #/HR				
LEAKIN - #/HR	0.0	0.0	0.0	0.0
TOTAL FLOW IN - #/HR	34	34	34	34
TOTAL FLOW OUT - #/HR	34	34	34	34
LEAKOUT - #/HR				
EGROUT - #/HR				
OUTLET FLOW RATE - #/HR	34	34	34	34
INLET ENTHALPY - BTU/#	495	495	495	494
EGRINHIN - BTU/#				
LEAKINHIN - BTU/#	241	241	240	240
LEAKOUTHOUT - BTU/#	314	314	314	315
OUTLET ENTHALPY - BTU/#	133	133	134	136
INLET TEMPERATURE - DEG F	1462	1462	1462	1462
OUTLET TEMPERATURE - DEG F	392	392	397	402
INLET INERT - PCT	80.93%	80.93%	80.93%	80.93%
OUTLET INERT - PCT	80.93%	80.93%	80.93%	80.93%
NTU(COLD)	14.60	14.60	14.10	13.60
NTU(HOT)	15.14	15.14	14.65	14.16
NTU(TOTAL)	7.84	7.84	7.58	7.32
WC(COLD)	10.28	10.28	10.30	10.32
WC(HOT)	11.51	11.51	11.53	11.55
WC(MIN)	10.28	10.28	10.30	10.32
WC(MAX)	11.51	11.51	11.53	11.55
EFFECTIVENESS	0.92	0.92	0.92	0.92
TCIN	169	169	170	172
TCOUT	1365	1365	1360	1355
THIN	1462	1462	1462	1462
THOUT	394	394	399	404
AVG RECUP VISCOSITY - #/HR-FT	0.086	0.086	0.086	0.086
RECUP REYNOLDS NUMBER	107	107	107	107
RECUP J FACTOR	0.040	0.040	0.040	0.040
RECUP F/J	2.0	2.0	2.0	2.0

: :

OUTLET BLOWER:				
HEAT INPUT - BTU/HR				
INLET FLOW RATE - #/HR	34	34	34	34
EGRIN - #/HR				
LEAKIN - #/HR				
TOTAL FLOW IN - #/HR	34	34	34	34
TOTAL FLOW OUT - #/HR	34	34	34	34
LEAKOUT - #/HR				
EGROUT - #/HR	7.56	7.56	7.58	7.59
OUTLET FLOW RATE - #/HR	26	26	26	27
INLET ENTHALPY - BTU/#	133	133	134	136
EGRINHIN - BTU/#				
LEAKINHIN - BTU/#				
LEAKOUTHOUT - BTU/#	133	133	134	136
OUTLET ENTHALPY - BTU/#	133	133	134	136
INLET TEMPERATURE - DEG F	392	392	397	402
OUTLET TEMPERATURE - DEG F	392	392	397	402
INLET INERT - PCT	80.93%	80.93%	80.93%	80.93%
OUTLET INERT - PCT	80.93%	80.93%	80.93%	80.93%

: :

HEAT BALANCE:				
MASS IN - MASS OUT	-0.01	-0.01	-0.01	-0.01
INLET ENTHALPY	733	733	735	736
FUEL INPUT (LHV)	26195	26195	26246	26300
HEATER OUTPUT	-23428	-23428	-23429	-23429
STACK LOSS (LHV)	-3508	-3508	-3559	-3615
STACK LOSS (HHV)	-6410	-6410	-6468	-6529
HEAT IN - HEAT OUT	-7.52	-7.52	-7.75	-7.98
NET STACK LOSS (HHV)	5608	5608	5664	5724
STACK LOSS (HHV) - %	19.32%	19.32%	19.47%	19.64%
CARNOT EFFICIENCY - %	68.32%	68.32%	68.32%	68.32%
CARNOT X COMB EFFIC. - %	55.12%	55.12%	55.02%	54.90%

NOx MODEL:				
OXYGEN - %	4.00%	4.00%	4.00%	4.00%
OXYGEN^.5	0.200	0.200	0.200	0.200
NITROGEN - %	72.53%	72.53%	72.53%	72.53%
GAS CONST - CAL/GM-MOL-DEG K	1.986	1.986	1.986	1.986
TEMPERATURE - DEG K	2199	2199	2197	2194
d(NO)/dt - GM-MOL/CM^3-SEC	6.47E+00	6.47E+00	6.27E+00	6.06E+00
d(NO)/dt - PPM/SEC	9.79E+04	9.79E+04	9.49E+04	9.18E+04

A.2 MODEL EQUATIONS

```
B3: [W22] 'CASE
D3: "A
D4: "MTI
B5: [W22] 'COSTING:
D5: "IECEC
B6: [W22] 'REGEN COST/SQ-FT - $/SQ-FT
D6: (C2) 1
B7: [W22] 'HTR HEAD COST/SQ-FT - $/SQ-FT
D7: (C2) 10
B8: [W22] 'PREHTR HYDR DIAM - FT
D8: (F3) 0.01057
B9: [W22] 'RECUP HYDR DIAM - FT
D9: (F3) +D8
B10: [W22] 'HTR HEAD HYDR DIAM - FT
D10: (F3) 0.003
B11: [W22] 'PREHTR SURFACE AREA - FT^2
D11: (F2) +D25*0.7^0.667*0.5*(D85+D86)/(D113*D26)
B12: [W22] 'RECUP SURFACE AREA - FT^2
D12: (F2) +D11
B13: [W22] 'HTR HEAD SURFACE AREA - FT^2
D13: (F2) +D33*0.7^0.667*0.5*(D154+D155)/(D35*D193*D34)
B14: [W22] 'PREHTR LENGTH - FT
D14: (F2) +D11*D8*D26/(0.5*(D85+D86)*4)
B15: [W22] 'RECUP LENGTH - FT
D15: (F2) +D14
B16: [W22] 'HTR HEAD LENGTH - FT
D16: (F2) +D10*D13*D34/(4*0.5*(D154+D155))
B17: [W22] 'REGENERATOR COST
D17: (C2) +D6*D11
B18: [W22] 'HTR HEAD COST
D18: (C2) +D7*D13
B19: [W22] 'TOTAL EHS H-X COST
D19: (C2) +D17+D18
B20: [W22] 'REGEN PAYBACK VS BASE - HRS
D20: "BASE
B22: [W22] 'CYCLE HEAT INPUT - BTU/HR
D22: 23420*$D$257/D257
B23: [W22] 'INLET TEMPERATURE - DEG F
D23: 95
B24: [W22] 'EXCESS AIR @ BLOWER INLET- %
D24: (P0) 0.25
B25: [W22] 'PREHEATER NTU
D25: (F2) 14.6
B26: [W22] 'PREHTR MASS VELOCITY - #/HR-FT^2
D26: 837
B27: [W22] 'RECUPERATOR NTU
D27: (F2) 4*D15*D221/(0.7^0.667*D9)
B28: [W22] 'RECUP MASS VELOCITY - #/HR-FT^2
D28: 4*D15*0.5*(D193+D194)/(D9*D12)
B29: [W22] 'LEAKAGE (PRHTR TO RECUP) - % AIR FLOW
D29: (P1) 0
B30: [W22] 'EGR (EXH BLR TO INLET) - % AIR FLOW
D30: (P1) 0.3
B31: [W22] 'CGR (HTR HEAD TO COMB) - % AIR FLOW
D31: (P1) 0
B32: [W22] 'AVG HTR HEAD TEMP - DEG C
D32: 700
B33: [W22] 'HTR HEAD GAS-SIDE NTU
D33: (F2) 2.55
```

```
B34: [W22] 'HTR HEAD MASS VEL - #/HR-FT 2
D34: 3500
B35: [W22] 'HTR HEAD FIN EFFIC - %
D35: (P0) 0.9
B37: [W22] 'HHV FIRING RATE - BTU/HR
D37: +D22+D255
B38: [W22] 'COMBUSTOR INLET TEMP - DEG F
D38: +D134
B39: [W22] 'COMBUSTOR OUTLET TEMP - DEG F
D39: +D135
B40: [W22] 'RECUP INLET TEMP - DEG F
D40: +D203
B41: [W22] 'EXHAUST TEMP - DEG F
D41: +D242
B42: [W22] 'AVG RECUP HOT END TEMP - DEG F
D42: (D38+D40)/2
B43: [W22] 'd(NO)/dt - PPM/SEC
D43: (S2) +D267
B44: [W22] 'STACK LOSS (HHV) - %
D44: (P2) +D256
B45: [W22] 'COMB. PCT OXYGEN - %
D45: (P2) +D261
B46: [W22] 'COMB EXCESS AIR - %
D46: (P2) +D138
B47: [W22] 'STACK PCT OXYGEN - %
D47: (P2) 0.21*(1-D244)
B48: [W22] 'PREHTR PRESS DROP - IN WC
D48: (F2) (D114*D25*D26^2*(460+(D95+D96)*0.5)*0.7^0.667/(0.076*520*2))*12/(62.4*418000000)
B49: [W22] 'RECUP PRESSURE DROP - IN WC
D49: (F2) (D222*D27*D28^2*0.7^0.667*(460+(D203+D204)*0.5)/(0.076*520*2))*12/(62.4*4180000
B50: [W22] 'HTR HEAD PRESSURE DROP - IN WC
D50: (F2) (D184*D33*0.7^0.667*D34^2/(2*0.076*(520/(460+D180))*D35))*(12/(62.4*417000000))
B51: [W22] 'BLOWER THEO. POWER @ INLET - W
D51: (F2) (D48+D49+D50)*D64*(D75+460)*62.4*0.0003766/(12*0.076*520)
B52: [W22] 'BLOWER POWER @ 5X THEO - W
D52: (F2) 5*D51
B53: [W22] 'CARNOT X COMB EFFIC. - %
D53: (P2) +D258
B54: [W22] 'SHAFT POWER @ 50% CARNOT - W
D54: +D53*0.5*D37/3.413
B55: [W22] 'GAS COST @ $6.00/MMBTU - $/KWsh
D55: (C4) 6*D37/(1000*D54)
B56: [W22] 'BLOWER COST @ $18/MMBTU -$/KWsh
D56: (C4) 18*D52*3.413/(D54*1000)
B57: [W22] 'TOTAL POWER COST - $/KWsh
D57: (C4) +D55+D56
B58: [W22] \::
B59: [W22] 'INLET BLOWER:
B60: [W22] 'HEAT INPUT - BTU/HR
B61: [W22] 'INLET FLOW RATE - #/HR
D61: (F2) (1+D24)*D146*D142
B62: [W22] 'EGRIN - #/HR
D62: (F1) +D234
B63: [W22] 'LEAKIN - #/HR
B64: [W22] 'TOTAL FLOW IN - #/HR
D64: +D61+D62+D63
B65: [W22] 'TOTAL FLOW OUT - #/HR
D65: +D64
B66: [W22] 'LEAKOUT - #/HR
B67: [W22] 'EGROUT - #/HR
```

```
B68: [W22] 'OUTLET FLOW RATE - #/HR
D68: +D65-D66-D67
B69: [W22] 'INLET ENTHALPY - BTU/#
D69: +D$145*D74
B70: [W22] 'EGRINHIN - BTU/#
D70: +D201
B71: [W22] 'LEAKINHIN - BTU/#
B72: [W22] 'LEAKOUTHOUT - BTU/#
D72: (D73+D69)/2
B73: [W22] 'OUTLET ENTHALPY - BTU/#
D73: (D60+D61*D69+D62*D70+D63*D71-D66*D72)/(D65-D66)
B74: [W22] 'INLET TEMPERATURE - DEG F
D74: +D23
B75: [W22] 'OUTLET TEMPERATURE - DEG F
D75: +D73/(D77*D$144+(1-D77)*D$145)
B76: [W22] 'INLET INERT - PCT
D76: (P2) 0
B77: [W22] 'OUTLET INERT - PCT
D77: (P2) (D76*D61+D62*D206)/D64
B79: [W22] ':::
B80: [W22] 'AIR PREHEATER:
B81: [W22] 'HEAT INPUT - BTU/HR
D81: +D102*(D108-D107)
B82: [W22] 'INLET FLOW RATE - #/HR
D82: +D68
B83: [W22] 'EGRIN - #/HR
B84: [W22] 'LEAKIN - #/HR
B85: [W22] 'TOTAL FLOW IN - #/HR
D85: (F2) +D82+D83+D84
B86: [W22] 'TOTAL FLOW OUT - #/HR
D86: (F2) +D85
B87: [W22] 'LEAKOUT - #/HR
D87: (F2) +D29*D24
B88: [W22] 'EGROUT - #/HR
B89: [W22] 'OUTLET FLOW RATE - #/HR
D89: +D86-D87-D88
B90: [W22] 'INLET ENTHALPY - BTU/#
D90: +D73
B91: [W22] 'EGRINHIN - BTU/#
B92: [W22] 'LEAKINHIN - BTU/#
B93: [W22] 'LEAKOUTHOUT - BTU/#
D93: (D94+D90)/2
B94: [W22] 'OUTLET ENTHALPY - BTU/#
D94: (D81+D82*D90+D83*D91+D84*D92-D87*D93)/(D86-D87)
B95: [W22] 'INLET TEMPERATURE - DEG F
D95: +D75
B96: [W22] 'OUTLET TEMPERATURE - DEG F
D96: +D94/(D98*D$144+(1-D98)*D$145)
B97: [W22] 'INLET INERT - PCT
D97: (P2) +D77
B98: [W22] 'OUTLET INERT - PCT
D98: (P2) (D82*D97+D84*D206)/D85
B99: [W22] 'NTU(COLD)
D99: (F2) +D25
B100: [W22] 'NTU(HOT)
D100: (F2) +D27
B101: [W22] 'NTU(TOTAL)
D101: (F2) 1/(D104*((1/(D99*D102))+(1/(D100*D103))))
B102: [W22] 'WC(COLD)
D102: (F2) ((D85+D86)/2)*(D97*D$144+(1-D97)*D$145)
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```
B103: [W22] 'WC(HOT)
D103: (F2) ((D193-D194)/2)*((D205+D206)*0.5*D$144+(1-(D205-D206)*0.5)*D$145)
B104: [W22] 'WC(MIN)
D104: (F2) @MIN(D102,D103)
B105: [W22] 'WC(MAX)
D105: (F2) @MAX(D102,D103)
B106: [W22] 'EFFECTIVENESS
D106: (F3) (1-@EXP(-D101*(1-D104/D105)))/(1-(D104/D105)*@EXP(-D101*(1-D104/D105)))
B107: [W22] 'TCIN
D107: (D82*D90+D83*D91)/((D97*D$144+(1-D97)*D$145)*(D82+D83))
B108: [W22] 'TCOUT
D108: +D107+D106*D104*(D109-D107)/D102
B109: [W22] 'THIN
D109: +D203
B110: [W22] 'THOUT
D110: +D109-D106*D104*(D109-D107)/D103
B111: [W22] 'AVG VISC - #/HR-FT
D111: (F3) 0.00126*(460+(D107+D108)/2)^0.583
B112: [W22] 'PREHTR REYNOLDS NUMBER
D112: +D26*D8/D111
B113: [W22] 'PREHTR J FACTOR
D113: (F3) @IF(D112<2000,4.3/D112,0.019/D112^0.2)
B114: [W22] 'PREHTR F/J
D114: (F1) 2
B118: [W22] ':::
B119: [W22] 'COMBUSTOR:
B120: [W22] 'HEAT INPUT - BTU/HR
D120: +D37
B121: [W22] 'INLET FLOW RATE - #/HR
D121: +D89
B122: [W22] 'EGRIN - #/HR
D122: (F1) +D157
B123: [W22] 'LEAKIN - #/HR
B124: [W22] 'TOTAL FLOW IN - #/HR
D124: +D121+D122+D123
B125: [W22] 'TOTAL FLOW OUT - #/HR
D125: +D124+D146
B126: [W22] 'LEAKOUT - #/HR
B127: [W22] 'EGROUT - #/HR
B128: [W22] 'OUTLET FLOW RATE - #/HR
D128: +D125-D126-D127
B129: [W22] 'INLET ENTHALPY - BTU/#
D129: +D94
B130: [W22] 'EGRINHIN - BTU/#
D130: +D163
B131: [W22] 'LEAKINHIN - BTU/#
B132: [W22] 'LEAKOUTHOUT - BTU/#
D132: (D133+D129)/2
B133: [W22] 'OUTLET ENTHALPY - BTU/#
D133: (D120*D140/D139+D121*D129+D122*D130+D123*D131-D126*D132+D146*0.6*D74)/(D125-D126)
B134: [W22] 'INLET TEMPERATURE - DEG F
D134: +D96
B135: [W22] 'OUTLET TEMPERATURE - DEG F
D135: +D133/(D137*D$144+(1-D137)*D$145)
B136: [W22] 'INLET INERT - PCT
D136: (P2) +D98
B137: [W22] 'OUTLET INERT - PCT
D137: (P2) (D121*D136+D122*D167+D146*(1+D142))/(D124+D146)
B138: [W22] 'EXCESS AIR
D138: (P2) (D125*(1-D137))/(D146*D142)
```

```
B139: [W22] 'HHV - BTU/#
D139: 28813/1.22
B140: [W22] 'LHV - BTU/#
D140: 0.9*D139
B141: [W22] 'DENSITY #/FT 3
D141: (F2) 0.65*0.076
B142: [W22] 'THEO. A/F #/#
D142: (F2) 25.02/(1.25*1.22)
B143: [W22] 'THEO. FLAME TEMP - DEG F
D143: 3600
B144: [W22] 'COMB PROD SPEC HEAT - BTU/#-F
D144: (F3) +D140/((D143-70)*(1+D142))
B145: [W22] 'AIR SPECIFIC HEAT - BTU/#-F
D145: (F2) +D144/1.13
B146: [W22] 'FUEL FLOW
D146: (F3) +D120/D139
B148: [W22] \::
B149: [W22] 'HEATER HEAD:
B150: [W22] 'HEAT INPUT - BTU/HR
D150: +D172*(D179-D178)
B151: [W22] 'INLET FLOW RATE - #/HR
D151: (F1) +D128
B152: [W22] 'EGRIN - #/HR
B153: [W22] 'LEAKIN - #/HR
B154: [W22] 'TOTAL FLOW IN - #/HR
D154: (F1) +D151+D152+D153
B155: [W22] 'TOTAL FLOW OUT - #/HR
D155: (F1) +D154
B156: [W22] 'LEAKOUT - #/HR
B157: [W22] 'EGROUT - #/HR
D157: (F1) +D31*D24
B158: [W22] 'OUTLET FLOW RATE - #/HR
D158: (F1) +D155-D156-D157
B159: [W22] 'INLET ENTHALPY - BTU/#
D159: (F1) +D133
B160: [W22] 'EGRINHIN - BTU/#
B161: [W22] 'LEAKINHIN - BTU/#
B162: [W22] 'LEAKOUTHOUT - BTU/#
D162: (F1) (D163+D159)/2
B163: [W22] 'OUTLET ENTHALPY - BTU/#
D163: (F1) (D150+D151*D159+D152*D160+D153*D161-D156*D162)/(D155-D156)
B164: [W22] 'INLET TEMPERATURE - DEG F
D164: +D135
B165: [W22] 'OUTLET TEMPERATURE - DEG F
D165: +D163/(D167*D$144+(1-D167)*D$145)
B166: [W22] 'INLET INERT - PCT
D166: (P2) +D137
B167: [W22] 'OUTLET INERT - PCT
D167: (P2) +D166
B168: [W22] 'NTU(COLD)
D168: (F2) +D169
B169: [W22] 'NTU(HOT)
D169: (F2) +D33
B170: [W22] 'NTU(TOTAL)
D170: (F2) 1/(D173*((1/(D168*D171))+(1/(D169*D172))))
B171: [W22] 'WC(COLD)
D171: (F2) 100*D172
B172: [W22] 'WC(HOT)
D172: (F2) (D166*D$144+(1-D166)*D$145)*D154
B173: [W22] 'WC(MIN)
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D173: (F2) @MIN(D171,D172)
B174: [W22] 'WC(MAX)
D174: (F2) @MAX(D171,D172
B175: [W22] 'EFFECTIVENESS
D175: (F2) (1-@EXP(-D170*(1-D173/D174)))/(1-(D173/D174)*@EXP(-D170*(1-D173/D174)))
B176: [W22] 'TCIN
D176: 32+1.8*D32-(D172/D171)*(D178-D179)/2
B177: [W22] 'TCOUT
D177: +D176+D175*D173*(D178-D176)/D171
B178: [W22] 'THIN
D178: +D164
B179: [W22] 'THOUT
D179: +D178-D175*D173*(D178-D176)/D172
B180: [W22] 'AVG HTR HEAD TEMP - DEG F
D180: 0.5*(D176+D177)
B181: [W22] 'AVG HTR HEAD VISC - #/HR-FT
D181: (F3) 0.00126*(460+D180)^0.583
B182: [W22] 'HTR HEAD REYNOLDS NUMBER
D182: +D34*D10/D181
B183: [W22] 'HTR HEAD J FACTOR
D183: (F3) 0.18/D182^0.4
B184: [W22] 'HTR HEAD F/J
D184: (F1) 5.5
B187: [W22] |::
B188: [W22] 'RECUPERATOR:
B189: [W22] 'HEAT INPUT - BTU/HR
D189: -D81
B190: [W22] 'INLET FLOW RATE - #/HR
D190: +D158
B191: [W22] 'EGRIN - #/HR
B192: [W22] 'LEAKIN - #/HR
D192: (F1) +D87
B193: [W22] 'TOTAL FLOW IN - #/HR
D193: +D190+D191+D192
B194: [W22] 'TOTAL FLOW OUT - #/HR
D194: +D193
B195: [W22] 'LEAKOUT - #/HR
B196: [W22] 'EGROUT - #/HR
B197: [W22] 'OUTLET FLOW RATE - #/HR
D197: +D194-D195-D196
B198: [W22] 'INLET ENTHALPY - BTU/#
D198: +D163
B199: [W22]  EGRINHIN - BTU/#
B200: [W22] 'LEAKINHIN - BTU/#
D200: +D93
B201: [W22] 'LEAKOUTHOUT - BTU/#
D201: (D202+D198)/2
B202: [W22] 'OUTLET ENTHALPY - BTU/#
D202: (D189+D190*D198+D191*D199+D192*D200-D195*D201)/(D194-D195)
B203: [W22] 'INLET TEMPERATURE - DEG F
D203: +D165
B204: [W22] 'OUTLET TEMPERATURE - DEG F
D204: +D202/(D206*D$144+(1-D206)*D$145)
B205: [W22] 'INLET INERT - PCT
D205: (P2) +D167
B206: [W22] 'OUTLET INERT - PCT
D206: (P2) (D190*D205+D192*(D97+D98)/2)/D193
B207: [W22] 'NTU(COLD)
D207: (F2) +D99
B208: [W22] 'NTU(HOT)
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D208: (F2) +D1[illegible]
B209: [W22] 'NTU TOTAL
D209: (F2) 1/(D212*(1/(D207*D210))+(1/(D208*D211)))
B210: [W22] 'WC(COLD)
D210: (F2) +D102
B211: [W22] 'WC(HOT)
D211: (F2) +D103
B212: [W22] 'WC(MIN)
D212: (F2) +D104
B213: [W22] 'WC(MAX)
D213: (F2) +D105
B214: [W22] 'EFFECTIVENESS
D214: (F2) +D106
B215: [W22] 'TCIN
D215: +D107
B216: [W22] 'TCOUT
D216: +D108
B217: [W22] 'THIN
D217: +D109
B218: [W22] 'THOUT
D218: +D110
B219: [W22] 'AVG RECUP VISCOSITY - #/HR-FT
D219: (F3) 0.00126*(460+(D217+D218)/2)^0.583
B220: [W22] 'RECUP REYNOLDS NUMBER
D220: +D28*D9/D219
B221: [W22] 'RECUP J FACTOR
D221: (F3) @IF(D220<2000,4.3/D220,0.019/D220^0.2)
B222: [W22] 'RECUP F/J
D222: (F1) 2
B225: [W22] \::
B226: [W22] 'OUTLET BLOWER:
B227: [W22] 'HEAT INPUT - BTU/HR
B228: [W22] 'INLET FLOW RATE - #/HR
D228: +D197
B229: [W22] 'EGRIN - #/HR
B230: [W22] 'LEAKIN - #/HR
B231: [W22] 'TOTAL FLOW IN - #/HR
D231: +D228+D229+D230
B232: [W22] 'TOTAL FLOW OUT - #/HR
D232: +D231
B233: [W22] 'LEAKOUT - #/HR
B234: [W22] 'EGROUT - #/HR
D234: (F2) +D30*D61
B235: [W22] 'OUTLET FLOW RATE - #/HR
D235: +D232-D233-D234
B236: [W22] 'INLET ENTHALPY - BTU/#
D236: +D202
B237: [W22] 'EGRINHIN - BTU/#
B238: [W22] 'LEAKINHIN - BTU/#
B239: [W22] 'LEAKOUTHOUT - BTU/#
D239: (D240+D236)/2
B240: [W22] 'OUTLET ENTHALPY - BTU/#
D240: (D227+D228*D236+D229*D237+D230*D238-D233*D239)/(D232-D233)
B241: [W22] 'INLET TEMPERATURE - DEG F
D241: +D204
B242: [W22] 'OUTLET TEMPERATURE - DEG F
D242: +D240/(D244*D$144+(1-D244)*D$145)
B243: [W22] 'INLET INERT - PCT
D243: (F2) +D206
B244: [W22] 'OUTLET INERT - PCT
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D244: (P2) -D243
B246: [W22] \::
B247: [W22] 'HEAT BALANCE:
B248: [W22] 'MASS IN - MASS OUT
D248: (F2) +D61+D146-D235
B249: [W22] 'INLET ENTHALPY
D249: +D61*D69
B250: [W22] 'FUEL INPUT (LHV)
D250: +D146*(D140+0.6*D74)
B251: [W22] 'HEATER OUTPUT
D251: +D150
B252: [W22] 'STACK LOSS (LHV)
D252: -D235*D240
B253: [W22] 'STACK LOSS (HHV)
D253: +D252-D146*(D139-D140)
B254: [W22] 'HEAT IN - HEAT OUT
D254: (F2) @SUM(D249..D252)
B255: [W22] 'NET STACK LOSS (HHV)
D255: -D252+D146*(D139-D140)-D249-D146*0.6*D74
B256: [W22] 'STACK LOSS (HHV) - %
D256: (P2) +D255/D120
B257: [W22] 'CARNOT EFFICIENCY - %
D257: (P2) 1-(460+D74)/(460+32+1.8*D32)
B258: [W22] 'CARNOT X COMB EFFIC. - %
D258: (P2) +D257*(1-D44)
B260: [W22] 'NOx MODEL:
B261: [W22] 'OXYGEN - %
D261: (P2) 0.21*(1-D137)
B262: [W22] 'OXYGEN^.5
D262: (F3) +D261^0.5
B263: [W22] 'NITROGEN - %
D263: (P2) 0.79*(1-D137)+0.71*D137
B264: [W22] 'GAS CONST - CAL/GM-MOL-DEG K
D264: (F3) 1.986
B265: [W22] 'TEMPERATURE - DEG K
D265: (D135+460)/1.8
B266: [W22] 'd(NO)/dt - GM-MOL/CM^3-SEC
D266: (S2) 3E+14*@EXP(-129000/(D264*D265))*D263*D262
B267: [W22] 'd(NO)/dt - PPM/SEC
D267: (S2) 1000000*D266/(D264*D265)^0.5
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