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WASTE HEAT EXCHANGER FOR DIESEL EXHAUST POWERED AIR CONDITIONING

PHASE I SBIR PROGRAM FINAL REPORT

October 6, 1989

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he waste heat exchanger is often the largest single component in diesel-exhaust powered air onditioning systems, and on account of its high material content, is often the heaviest and most costly as well. The need exists for a heat exchanger design and fabrication technique which provides a configuration that is compact and lightweight, resists fouling from exhaust ases, is easily cleaned, uses inexpensive materials, and is amenable to high volume manufactur-ing. An experimental program aimed at the initial development of a heat exchanger meeting these requirements was conducted. The program was based on a design approach which integrates a low-ost, fouling resistant, easily cleanable heat exchanger with the engine muffler. A prototype proof-of-principle waste heat exchanger was designed, fabricated, and tested to demonstrate feasibility. The overall results obtained were sufficiently encouraging to warrant further onsideration of the concept.

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1.0 INTRODUCTION

1.1 Problem Definition

A key element in waste heat powered air conditioning or cogeneration systems is the waste heat exchanger. This is often the largest single component, and on account of its high material content is often the heaviest and most costly. Developers of waste heat powered air conditioning systems are forced to rely on available components and manufacturing techniques to fabricate their heat exchangers. Commonly, the approach is to utilize small diameter heat exchanger tubing to contain the high pressure working fluid, usually with external fins to enhance heat transfer from the exhaust gases. While this approach may yield a heat exchanger that may be fabricated without undue difficulty, it is not well suited to high volume production techniques, and yields a product which is expensive to manufacture. Furthermore, external flow of the exhaust gases over the heat exchanger tubing results in a high rate of deposition of soot, which tends to be difficult to remove due to the inaccessibility of the close-packed tube bundles.

Due to the low exhaust temperature of diesel exhaust - 600 to 800°F - a waste heat exchanger generally requires larger surface area than a direct fired heat exchanger of similar capacity. Also, since soot deposition increases the thermal resistance, provision is often made for additional surface area to provide some margin for performance under fouling conditions. Heat exchangers designed for fouling service often also have larger passage sizes for ease of cleaning and to prevent too-rapid a build-up of pressure drop when fouling occurs. The requirements for large surface area plus the large passage size combine to produce designs which are heavy, bulky, and as a result tend to be costly as well.

The magnitude of the fouling problem varies widely and is highly dependent on such factors as soot content of the exhaust, flow velocity, passage geometry and gas and surface temperature. While catalysts may be helpful in reducing soot content of the exhaust stream or in changing its fouling tendency, they do not eliminate the fouling problem. Keeping the exhaust stream and the heat exchanger surfaces above the vapor dew point, typically about 300°F, keeps the heat exchange surfaces dry and tends to prevent hard-to-remove deposits from forming. This is also important in reducing the corrosiveness of the exhaust gases. Abrupt changes in the velocity or the direction of the exhaust gases tends to cause greater fallout of soot particles and should be avoided. Cavities and narrow, difficult-to-reach spaces such as the gaps between fins or closely spaced tubes create a servicing problem, as they are difficult to reach by manual, soot blowing or washing methods.

The need exists for a heat exchanger design and fabrication technique which provides a configuration that is compact and lightweight, resists fouling from exhaust gases, is easy to clean, uses inexpensive materials, and is amenable to high-volume manufacturing. A desirable feature of such a design is that it would be suitable for use with most working fluids, whether liquid or gaseous, high-pressure or low-pressure, single-phase or multi-phase.

1.2 Potential Solution Approach

A heat exchanger designed for diesel exhaust heat recovery should feature straight, easy-to-reach gas flow passages which are readily accessible for servicing. Such heat exchanger geometries are typically found in gas-to-gas exchangers, such as air preheaters and recuperators. These are generally of the plate-fin type, in which corrugated fin sheets are placed between plates which separate the two gas streams. This type of design, however, is not amenable to high-pressure fluid-to-gas heat exchangers, as generally tubular elements are needed to contain the pressure of the fluid. The common approach of using multiple rows of small diameter finned tubing satisfies the heat transfer and pressure containment requirements, but yields a heat exchanger that is difficult to clean, and uses large amounts of fairly expensive finned tubing. A better approach would utilize much shorter lengths of large diameter tubing for pressure containment, while retaining the features of straight-through flow plate-fin heat exchangers.

A promising design approach which provides the desired characteristics and features is to contain the working fluid in the annular space between two concentric large diameter tubes. Helical baffles, in the form of helical fins wound about the inner tube, force the working fluid to flow in a helical path through the annulus. The exhaust gas also flows through the annular space formed between a pair of similar large diameter tubes. Longitudinal fins, in the form of accordion-pleated sheet metal, join the two tubes, forming more or less rectangular flow passages for the exhaust gas. The helical working fluid passages are placed adjacent to the inner and outer tubes of the working gas, forming a three-pass heat exchanger: two working fluid passes and one exhaust gas pass. A waste heat exchanger designed in this fashion would be compact and lightweight, would be amenable to high-volume low-cost manufacturing techniques, would be resistant to fouling, and would be easily serviceable.

1.3 Phase I SBIR Study

The effort described in this report was sponsored by the Department of the Army under Contract No. DAAK 70-89-C-0085 as a Phase I study in the overall Defense Small Business Innovation Research (SBIR) Program. The principal objective of this investigation was to demonstrate the feasibility of the proposed design approach for waste heat exchangers suitable for use in diesel exhaust powered air conditioning and other waste heat powered systems.

The Phase I effort consisted of a combined analytical and experimental program involving the analysis, design, and initial development of a proof-of-principle waste heat exchangers for a representative set of design conditions. A series of design calculations was conducted to investigate the effects of various design parameters on heat exchanger size, shape, and performance. Based on the results of the analysis, a prototype heat exchanger was designed in detail. The prototype configuration was fabricated and installed in the AMTI laboratory facility. The prototype heat exchanger was tested over a representative range of operating conditions to measure its performance and demonstrate the feasibility of the design approach. The overall results obtained in the Phase I program are presented and discussed in the remainder of this report.

2.0 DESCRIPTION OF CONCEPT

The conceptual design of a waste heat exchanger based on the proposed approach is illustrated in Figure 2.1. It is made up of four concentric tubes and the three annular spaces between the tubes. The middle annulus is the gas flow passage and contains longitudinal fins. The inner and outer annuli are the working fluid passages and contain helical fins. With this arrangement, the heat from the middle gas-flow passage can flow radially inward and outward to the fluid passages on either side. The ends of the fluid-containing tubes are sealed to contain the working fluid pressure. The ends of the gas flow annulus are open to permit exhaust gas flow. Access covers on one or both ends of the heat exchanger may be removed to permit cleaning of the gas flow passages.

In most practical heat exchangers of this design, the diameter of the innermost tube will be large enough that the exhaust gas muffler can be incorporated within the heat exchanger. This is illustrated in Figure 2.1, which shows a simple multi-chamber muffler. The exhaust gases exit at the end of the muffler and turn 180° to enter the gas passage of the heat exchanger, exiting at the same end as the inlet to the muffler. With this arrangement, there need be no fluid or gas fittings at one end of the heat exchanger, thereby facilitating an easily removable end cap for access for cleaning. If necessary, it would also be feasible to remove the exhaust inlet/outlet end; however, this would require disassembly of the exhaust piping.

This design should be inherently easy to manufacture, and

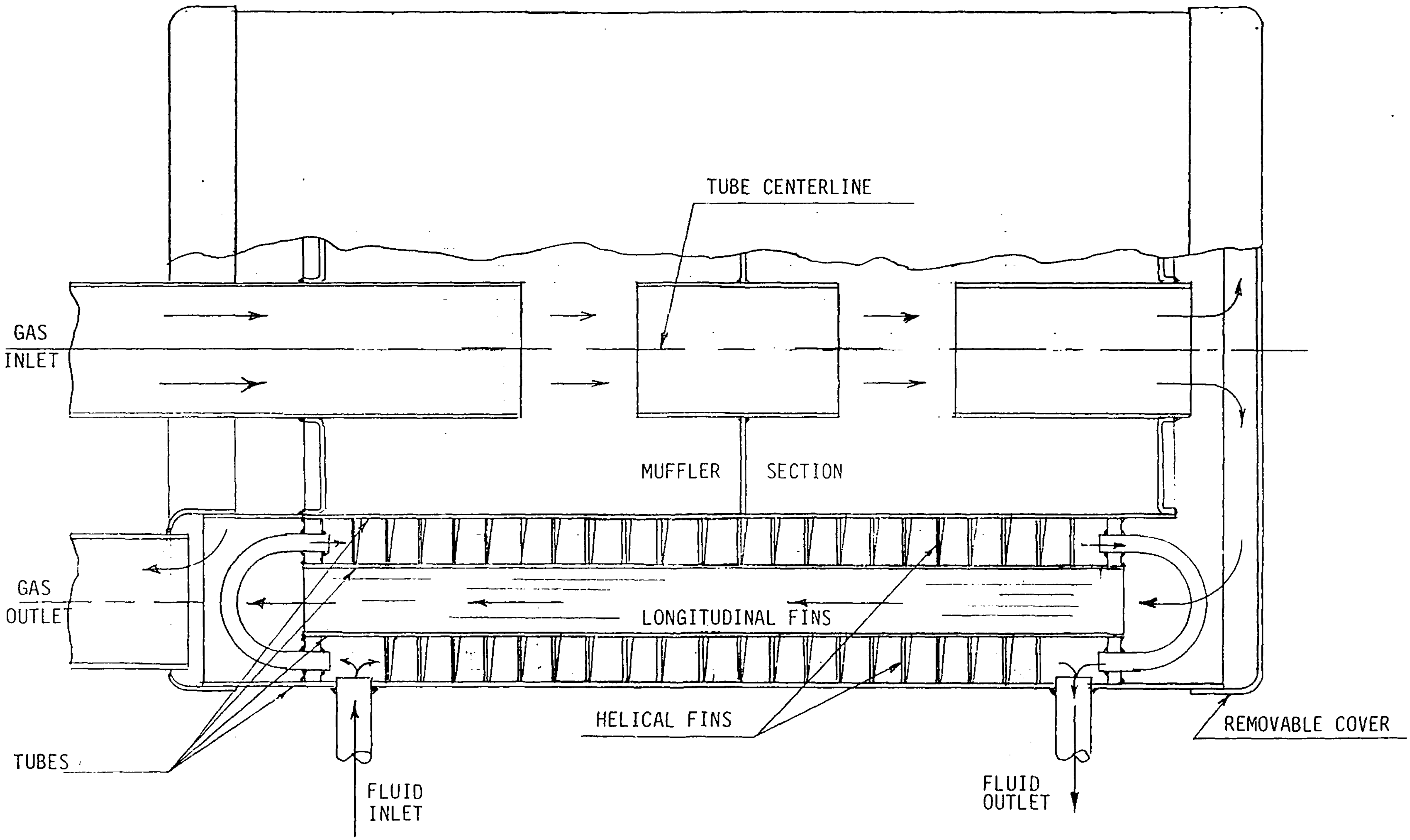


Figure 2 1 CONCEPTUAL HEAT EXCHANGER DESIGN should be highly amenable to automated fabrication procedures, leading to low cost. Helically finned tubes are currently manufactured at relatively low cost by either brazing, welding, or tension winding techniques. Longitudinal finned tubing is also manufactured by welding or brazing "U-channels" onto the outer diameter of the tubes. This technique could be used in the proposed design, or alternatively, continuous accordion-pleated sheet metal assemblies could be inserted into the gas flow annulus and brazed, welded or shrink-fitted in place. Following assembly of the concentric sections, the ends of the fluid passages would be sealed by brazing or welding, and the fluid inlet and outlet fittings welded or brazed in place. Attachment of the end caps would complete the assembly.

Most of the heat transfer surfaces in such a design are extended surfaces, not subject to pressure stresses. Therefore, they can be made of light gauge material, reducing overall weight and material costs. Although the wall thickness of the pressure-containing tubing must theoretically be larger for a large diameter tube versus a small diameter tube, in practice the additional wall thickness required as a margin for corrosion causes the required increase in wall thickness to be less than linear with diameter. Moreover, the wall thickness requirements can be reduced by brazing the entire assembly, such that the helical and longitudinal fins become stiffening ribs. Compensating for any cost increase due to material content of the large diameter tubing is the fact that only a few feet of such tubing, typically less than 10 feet, would be required, versus up to 100 feet of small diameter tubing in a coil-type heat exchanger. As material cost is but one component of the cost of finned tubing, the significantly shorter total length of tubing should result in an overall cost saving.

While the basic design approach appears fairly straightforward, there are a number of potential technical issues which should be considered in the practical implementation of the concept. For example, contact resistance may become an issue if the heat exchanger is assembled by shrink fitting or if the fins are tension-wound without a metallurgical bond between tubing and fins. If the heat exchanger is assembled by shrink fitting concentric sections, it is likely that the fins would be metallurgically joined to the outer diameter of the inner tube, and an outer tube would be shrink fitted over the finned inner tube. In this case, a significant contact resistance may exist between the fin tips and the inner diameter of the outer tube. This is likely to be of importance only on the exhaust-gas side which typically provides most of the thermal resistance. On the working-fluid side, the heat transfer coefficient tends to be high. As a result, surface enhancement on the fluid side is not necessary and contact resistance of the widely spaced fin/baffles is not critical.

The tolerances to which the finned tubing can be manufactured will determine whether low contact resistance can be achieved by shrink fitting a tube over fins. In the event that it is not possible to achieve the required tolerances, then shrink fitting may not be practical and brazing the fins to both the inner and outer tubes would be a more reasonable approach.

In order to minimize fouling, it is important to keep the metal temperatures above the dew point of the vapor in the flue gases. For fuels such as diesel fuel, especially those having a significant sulphur content, dew point temperatures may be as high as 300°F or more. For working fluid inlet temperatures of 300°F or higher, condensation should not be a significant problem. When it is necessary to operate with low fluid inlet temperatures, it may be difficult to avoid condensation. In this case, more frequent washing to avoid difficult-to-remove soot buildups may be required.

As previously discussed, the use of straight gas flow passages with moderately high velocities and avoidance of changes in gas flow direction and velocity minimizes soot deposition. The proposed design concept should be resistant to fouling in this regard. In any event, while steps may be taken to limit soot deposition, it cannot be avoided entirely, and provisions must be made for cleaning. The proposed design concept should be highly amenable to cleaning by any of the available methods, including washing, blowing, and wire brushing. While wire brushing may be more time consuming than washing or blowing, the easy accessibility and limited number of straight-through passages should make this practical using a brush shaped to fit the rectangular cross section of the gas-side passages.

3.0 DESIGN OF PROTOTYPE HEAT EXCHANGER

3.1 Selected Design Conditions

For the purpose of demonstrating the feasibility of the heat exchanger design concept it was desirable to select a set of target design conditions which would be representative of the intended application area and within the scope of the Phase I effort. Based on these considerations and discussions with the Army (Reference 1), it was decided to design the prototype heat exchanger for use with a typical diesel engine for a 10 kW generator. The resulting nominal design conditions selected for the heat exchanger are given in Table 3.1. The exhaust gas conditions were determined from general performance information based on past experience with engines of the selected type and size. The temperature level and flow rate of the working fluid were selected as representative of thermally powered cooling applications of the appropriate capacity range. The working fluid is a commercially available organic fluid commonly used in high temperature heat transfer applications.

3.2 Parametric Design Analysis

For the selected set of design conditions, the highest theoretical value of heat recovery rate is approximately 27,700 Btu per hour. The primary goal in conducting the design of the prototype heat exchanger for the specified application was to maximize the waste heat recovery rate within reasonable constraints on size, weight, and pressure drop. To accomplish this purpose, a semi-empirical calculation model of heat exchanger performance was derived. The model was based on conventional heat transfer and

Table 3.1 NOMINAL HEAT EXCHANGER DESIGN CONDITIONS

Exhaust Gas Stream:	
Source	20 hp Diesel Engine (1800 RPM, 4 Cycle, 2 Cylinder)
Temperature	800 °F
Flow Rate	250 lbm/hr
Maximum Engine Backpressure (Muffler + HX)	15 in W. C.
Working Fluid Stream:	
Fluid	Syltherm-800 (Ref. 2)
Inlet Temperature	375 °F
Flow Rate	800 lbm/hr

fluid flow relationships available from standard text books (for example, References 3 and 4) as applied to the specific geometry of the proposed heat exchanger concept. The model was used to conduct a series of parametric calculations investigating the effect on performance of variations in overall core dimensions, fin geometry, and materials of construction. The results obtained were reviewed to select the specific prototype configuration for fabrication and testing.

3.2.1 Heat Exchanger Materials

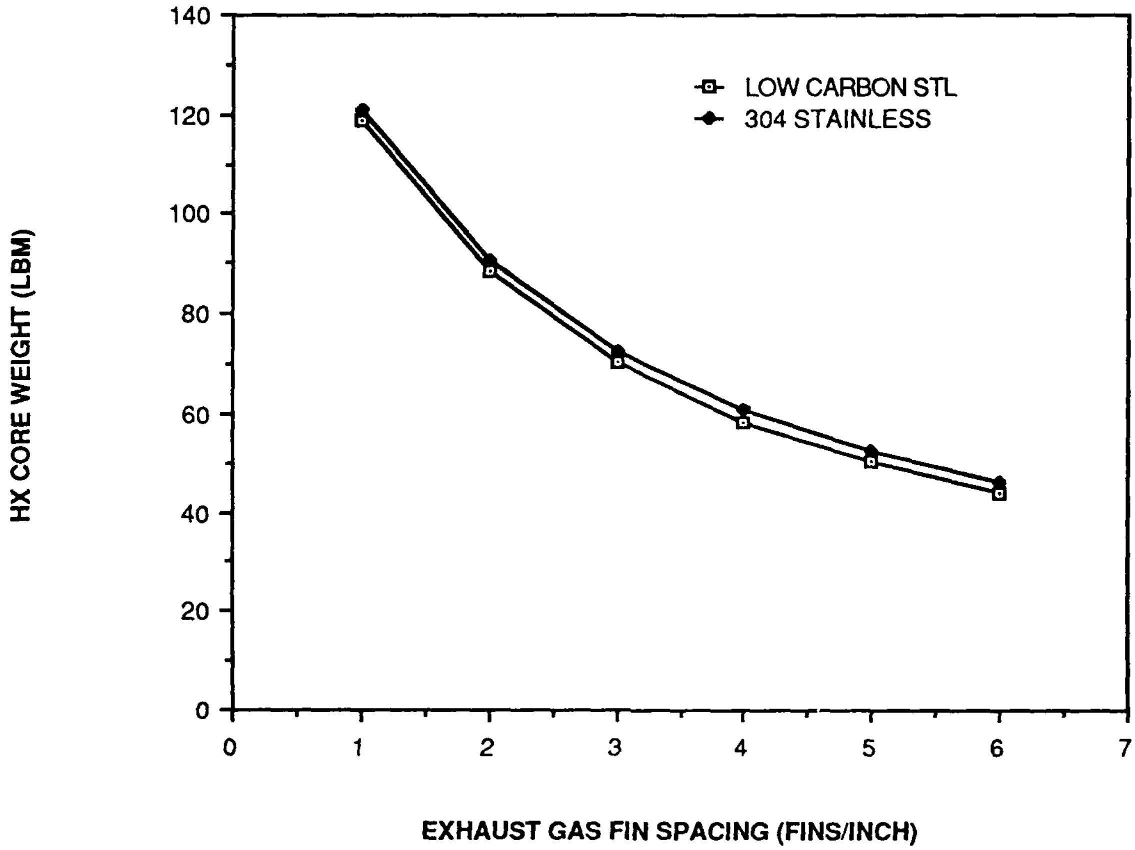
Choosing the materials of construction for the prototype heat exchanger involved consideration of several factors. Thermal stresses, for example, could be important as the applications envisioned for these heat exchanger will likely result in temperature differentials of several hundred degrees. Even without large temperature differentials, use of materials having different thermal expansion coefficients would give rise to large thermal stresses. Stainless steel would be desirable for use in the pressure containing tubing on account of its corrosion resistance. However, stainless steel has relatively low thermal conductivity, so a higher conductivity material such as low carbon steel would be preferable for the fins.

Available options included use of stainless steel throughout, which would reduce fin efficiency and require a somewhat heavier design. Another alternative was to use aluminized steel, which offers some corrosion protection, but without the durability of stainless steel. A third alternative was to use a corrosion resistant alloy such as a 12% chrome steel (for example, 410 stainless), which has a thermal expansion coefficient close to that of low carbon steel. This would have permitted the use of a corrosion resistant tubing with carbon steel fins. The effects of construction material on the design of the heat exchanger were evaluated analytically. The typical results given in Figure 3.1 show a comparison between 304 stainless steel and low carbon steel for the gas-side fins. As can be seen, there is very little difference in the required heat exchanger size for the two different materials. After reviewing the various options it was decided to use 304 stainless steel throughout for fabrication of the prototype heat exchanger.

3.2.2 Working Fluid Side

As was expected, the results of the parametric calculations showed that most of the thermal resistance is on the exhaust gas side of the heat exchanger. Consequently, varying the annulus height and fin dimensions on the working fluid side has very little effect on the design. This is illustrated by the results plotted in Figure 3.2 which show the effect of varying the fin spacing on the working fluid side. These results also show that the fin efficiency is quite low due to the high heat transfer coefficient associated with the liquid phase working fluid. This demonstrates that the fins on this side of the heat exchanger serve primarily as baffles to direct the flow rather than as extended surface to enhance heat transfer.

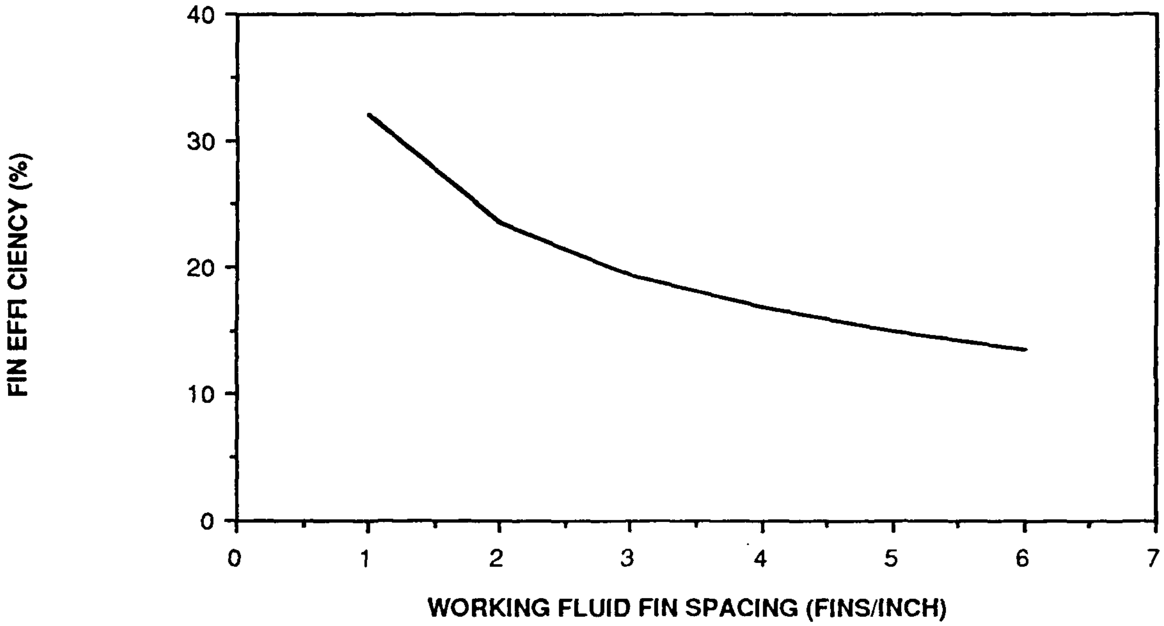
Because of the minimal impact of the working fluid fins on thermal performance, a design concept which simplifies the fabrication process was chosen. The selected approach consisted of using small-diameter stainless steel tubing (closed off at both ends) rather than sheet metal strip material for the "fins" (that is, baffles) on the working fluid side. The advantage of this approach is that tubing is readily available in a wide range of sizes and can be wound easily and controllably with simple tooling.

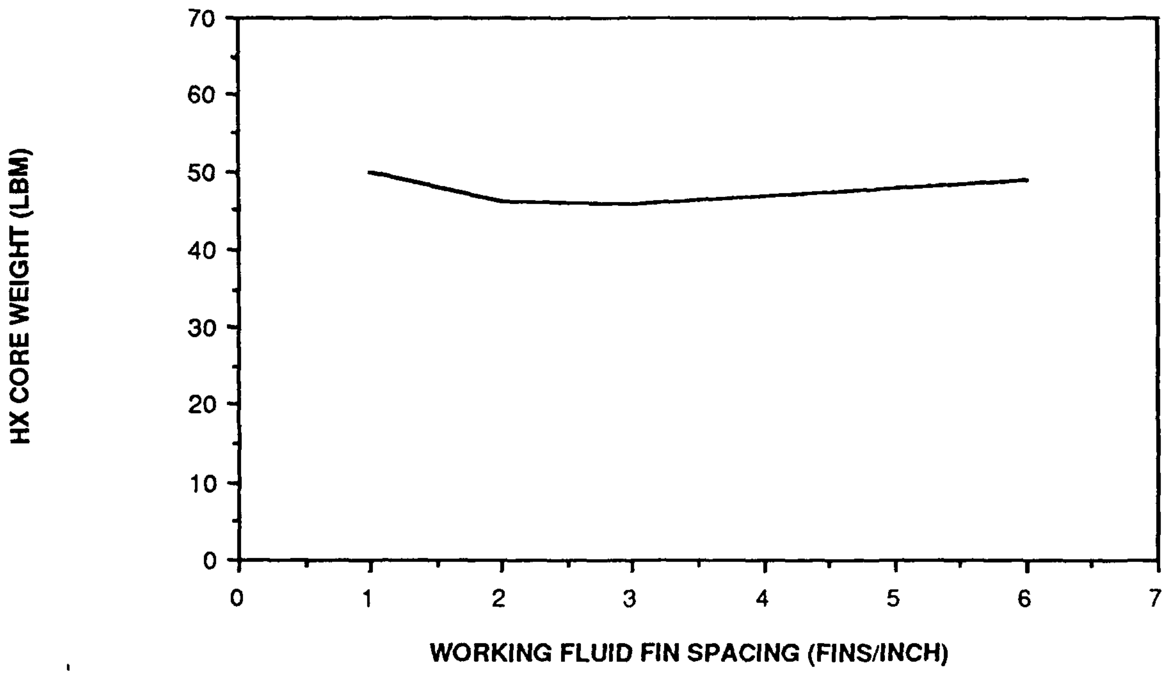


NOTES:

1. Gas Side: U-Shaped Fins; 0.5 in High; 0.020 in Thick.
2. Working Fluid Side: 0.5 in High Helical Fins; 0.02 in Thick; 2 Fins/Inch.
3. Heat Recovery Rate = 18,000 Btu/hr; Core ID = 4 in.

Figure 3.1 TYPICAL EFFECT OF GAS SIDE FIN MATERIAL ON HEAT EXCHANGER WEIGHT





NOTES:

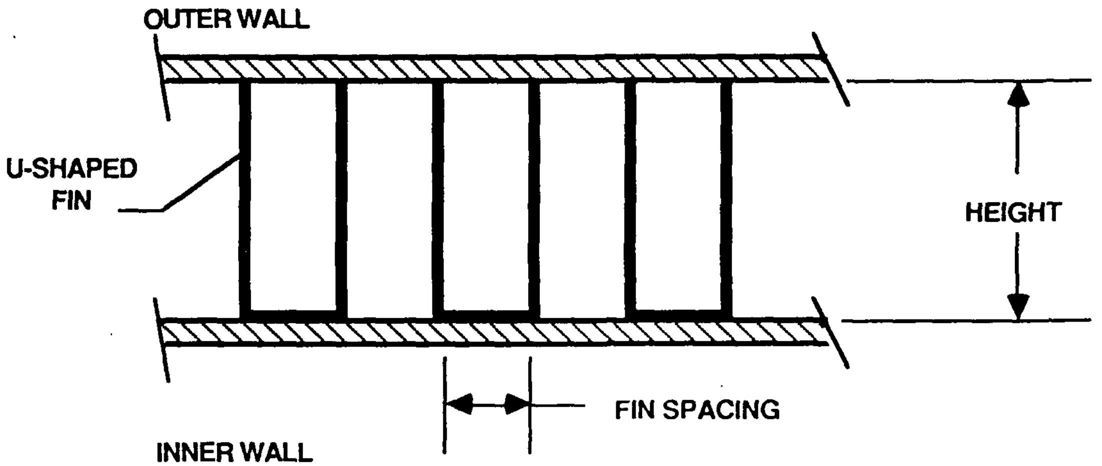
1. Gas Side: U-Shaped Fins; 0.5 in High; 0.02 in Thick; 6 Fins/Inch.
2. Working Fluid Side: 0.5 in High Helical Fins; 0.02 in Thick.
3. Heat Recovery Rate = 18,000 Btu/hr; Core ID = 4 in.

Figure 3.2 EFFECT OF WORKING FLUID SIDE FIN SPACING ON HEAT EXCHANGER DESIGN

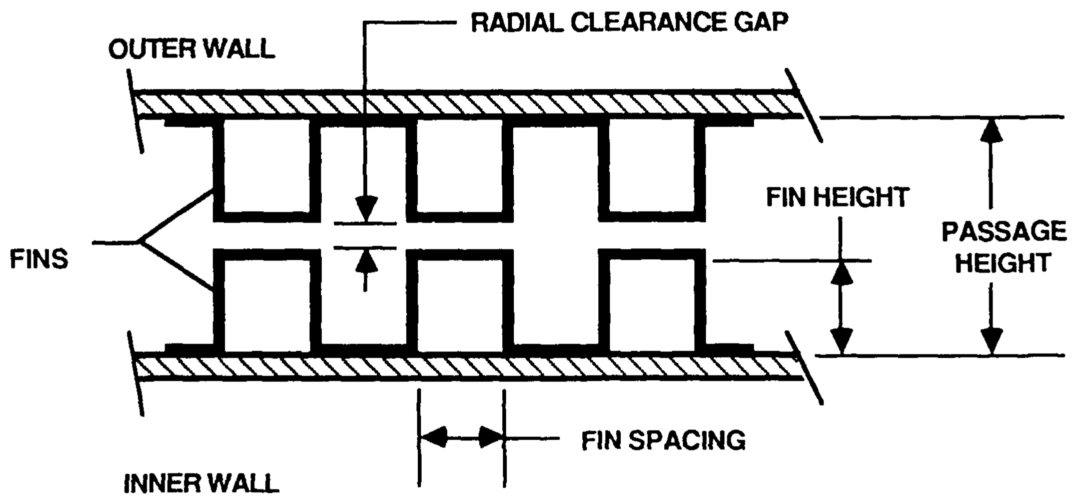
3.2.3 Exhaust Gas Side

The parametric calculations were performed for two different fin geometries on the exhaust gas side. These geometries are shown schematically in Figure 3.3. In the case of the channel fin geometry, individual U-shaped fins would be bonded (by welding or brazing) to the inner wall of the exhaust gas annulus and would extend across the full height of the annulus to the outer wall. A potential disadvantage of this configuration is that a significant contact resistance could exist at the outer wall where the fins are not firmly attached. This problem is eliminated with the corrugated fin geometry since the fins, consisting of corrugated sheet metal, are bonded to both the inner and outer walls of the exhaust gas annulus. Another advantage is that the amount of surface area per unit volume is higher due to the additional area provided by the corrugations. This increased compactness reduces the heat exchanger size required to achieve a given heat transfer capacity. Because of the various advantages, the corrugated fin geometry was selected for the gas side of the heat exchanger in designing the prototype configuration.

Since the thermal resistance on the exhaust gas side is controlling, the required size of the heat exchanger for a given recovery rate is strongly dependent on the gas-side fin area as determined by the corrugation spacing and height (Figure 3.3). This is illustrated by the results plotted in Figure 3.4 which show the variation in heat exchanger weight and gas-side pressure drop with corrugation dimensions for the case of a square corrugation geometry where spacing equals height. The results also demonstrate the basic trade-off between size and pressure drop involved in any heat exchanger design. As would be expected, the size of the heat exchanger decreases at the expense of increasing pressure drop as the gas-side fin geometry is made more compact. The gas-side

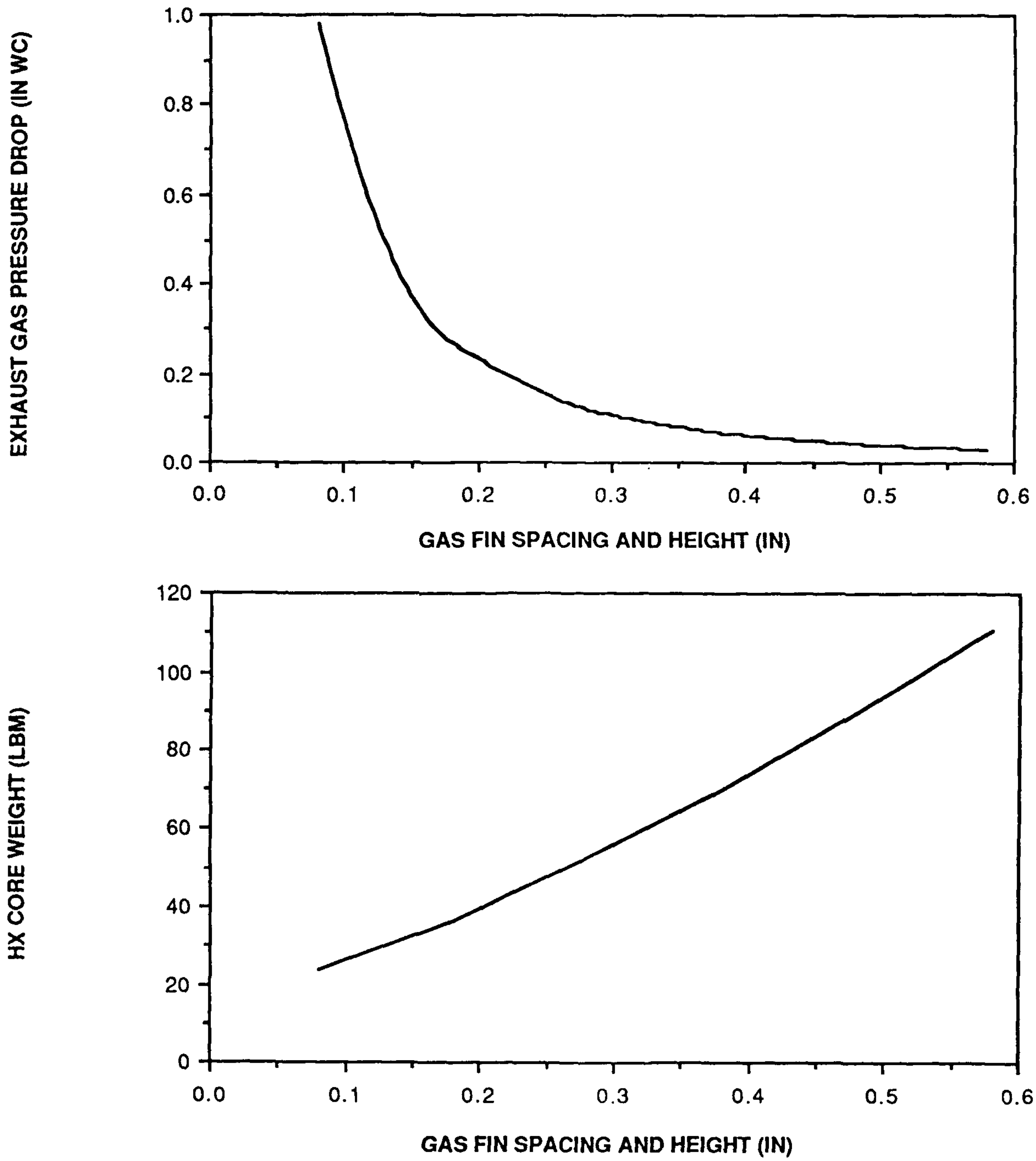


(a.) Channel Fin Geometry



(b.) Corrugated Fin Geometry

Figure 3.3 CANDIDATE GAS-SIDE FIN GEOMETRIES



NOTES:

1. Gas Side: Square Corrugated Fin Geometry (Spacing = Height); 0.02 in Thick.
2. Working Fluid Side: 0.25 in Helical Baffle; 1.0 in Pitch.
3. Heat Recovery Rate = 18,000 Btu/hr; Core ID = 4 in.

Figure 3.4 EFFECT OF GAS SIDE FIN DIMENSIONS ON HEAT EXCHANGER WEIGHT AND PRESSURE DROP pressure drop is quite reasonable even at very close fin spacings. The minimum allowable fin spacing is then primarily governed by practical considerations with regard to ease of physically cleaning the gas-side flow passages. Based on these considerations, a spacing of 1/4 inch was chosen as the minimum practical value in the prototype design.

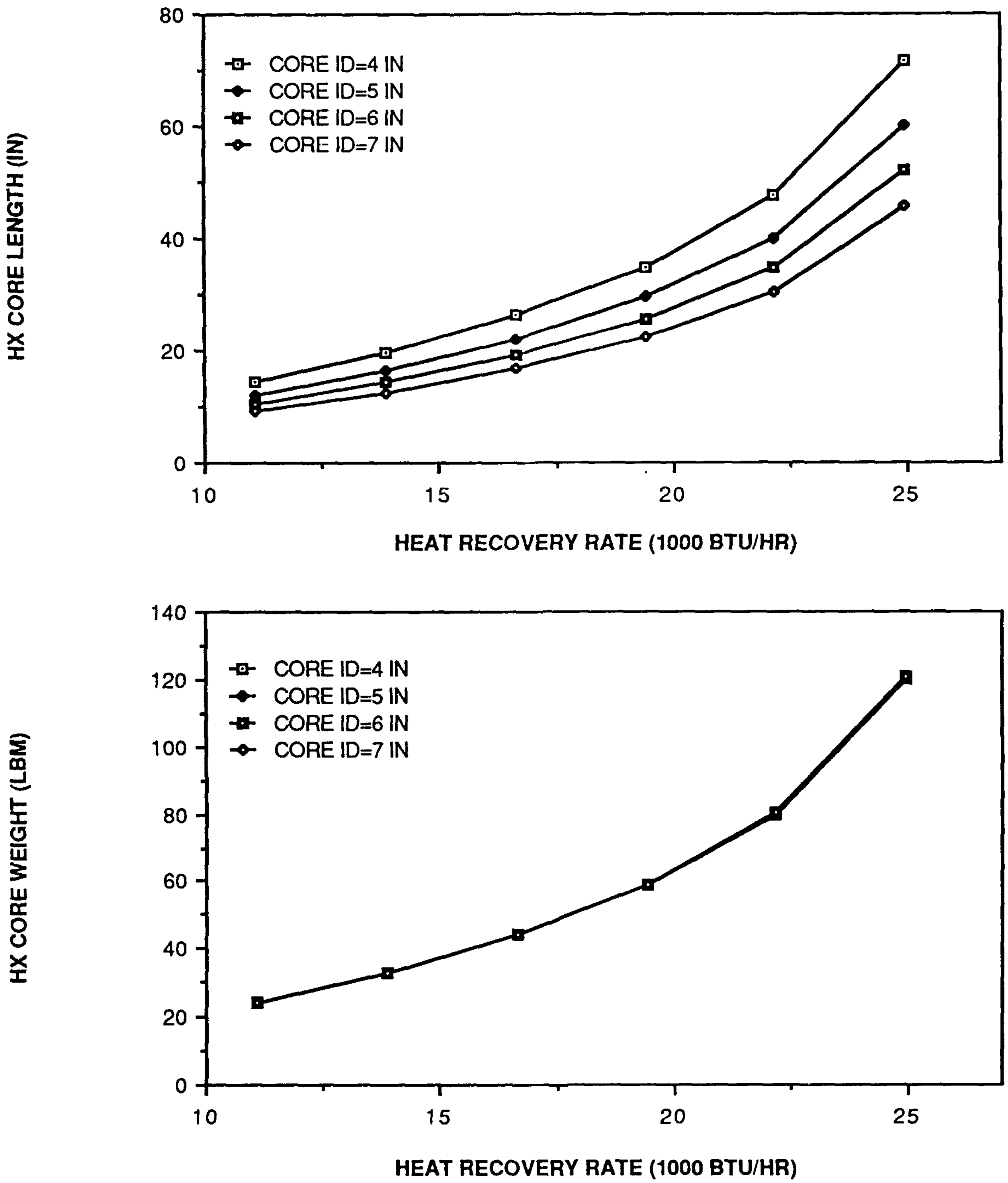
3.2.4 Overall Dimensions

Given the detailed dimensions of the fins on the exhaust-gas and working-fluid sides, the achievable waste heat recovery rate increases as the overall size of the heat exchanger is increased. This is demonstrated by the results plotted in Figure 3.5 which show the variations in heat exchanger weight and length with recovery rate for specific values of inside diameter. At a particular value of recovery rate, the core length of the heat exchanger varies inversely with inside diameter. As a result, the heat exchanger weight is essentially independent of inside diameter. This provides flexibility in choosing the value of inside diameter to obtain a more desirable overall package shape without co promising the total weight of the heat exchanger.

3.3 Selected Configuration

The details of the prototype configuration were chosen based on a critical review of the results obtained in the parametric design calculations. A schematic showing two views of the selected configuration is given in Figure 3.6. Additional detailed information on the selected design and performance specifications are given in Table 3.2.

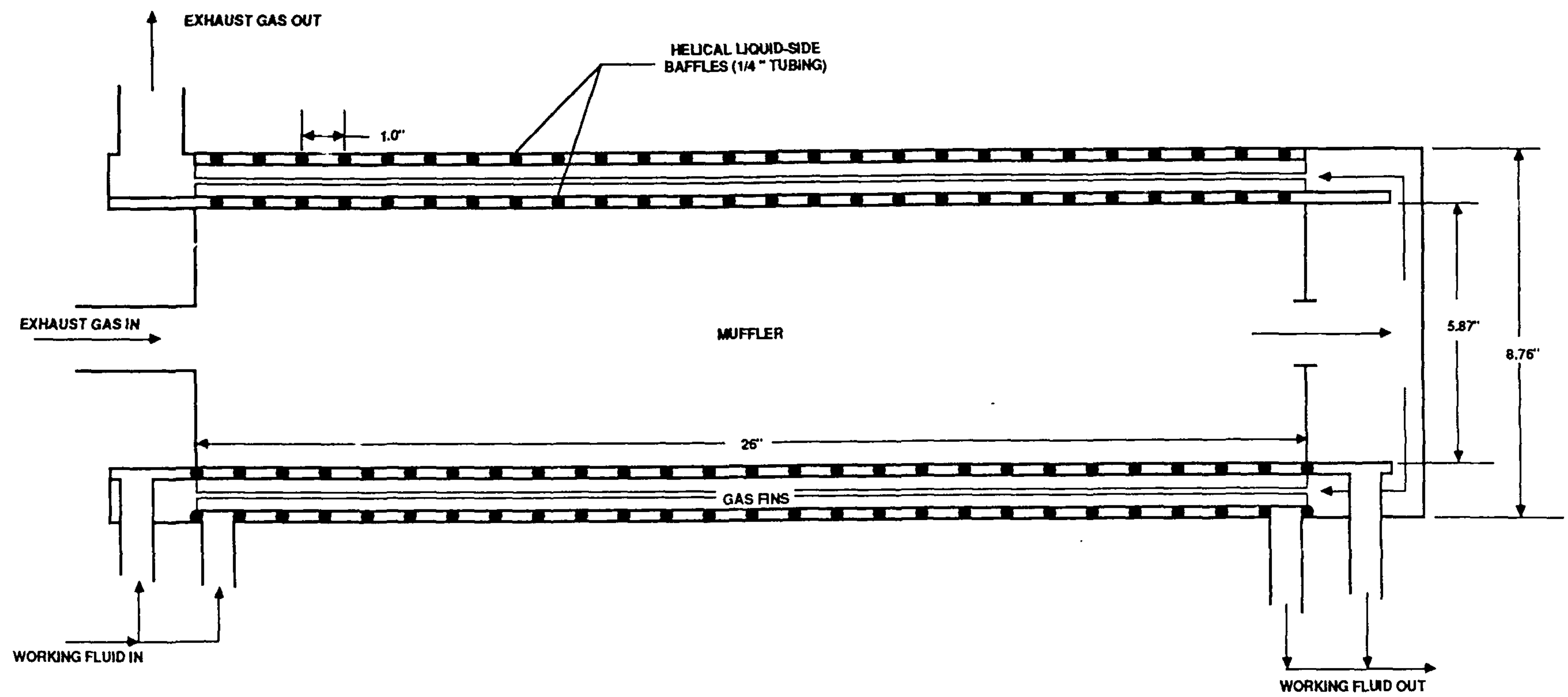
The exhaust gas stream from the engine enters the heat exchanger assembly along the center line and flows through a



NOTES:

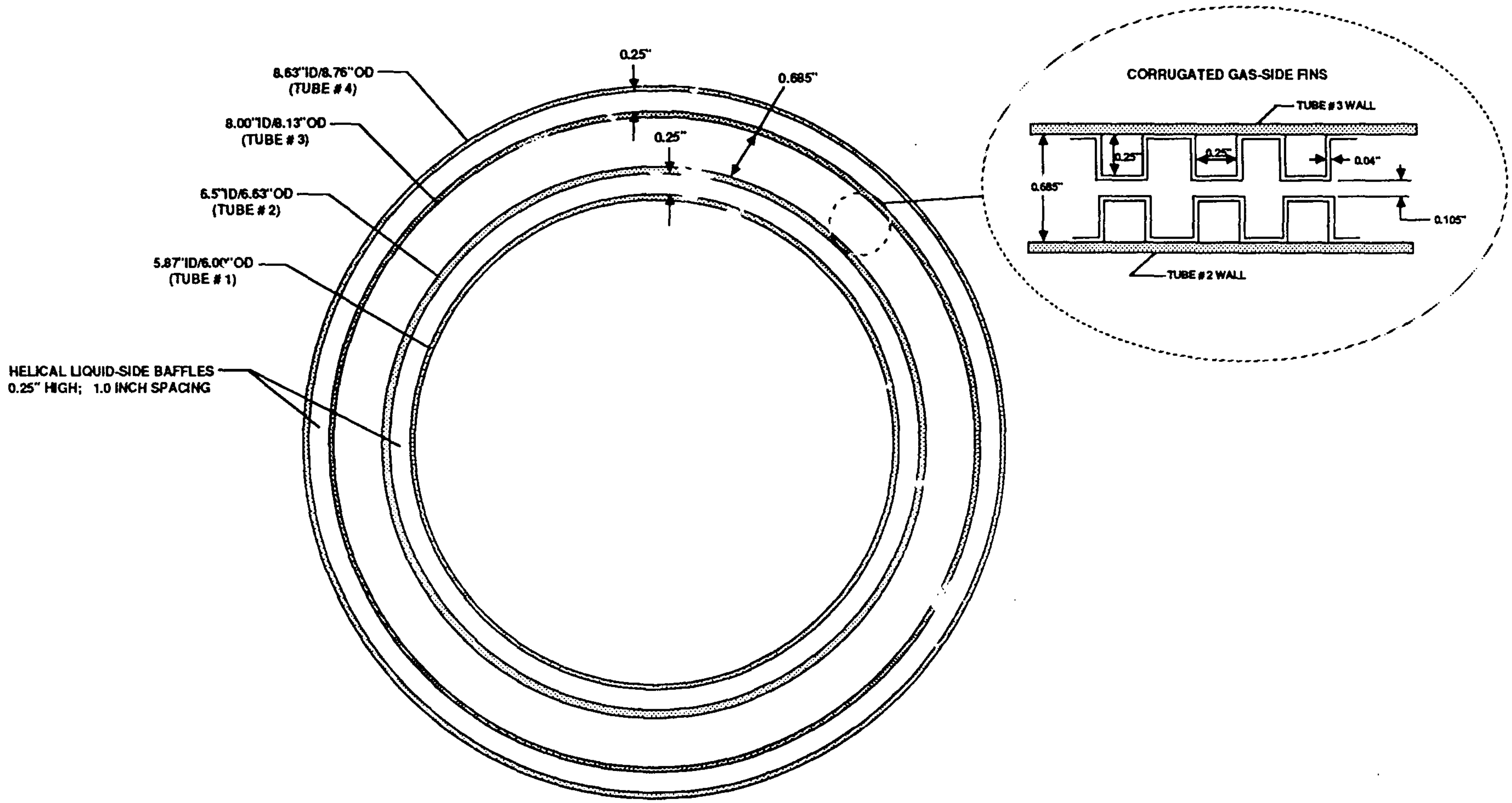
1. Gas Side: Corrugated Fins; 0.25 in Spacing and Height; 0.04 in Thick.
2. Working Fluid Side: 0.25 in High Helical Baffle; 1.0 in Pitch.

Figure 3.5 VARIATION OF WASTE HEAT RECOVERY RATE WITH OVERALL HEAT EXCHANGER SIZE



(a.) LONGITUDINAL CROSS SECTION

Figure 3.6 SCHEMATIC OF PROTOTYPE DESIGN CONFIGURATION



(b.) CORE CROSS SECTION

Figure 3.6 (continued) SCHEMATIC OF PROTOTYPE DESIGN CONFIGURATION

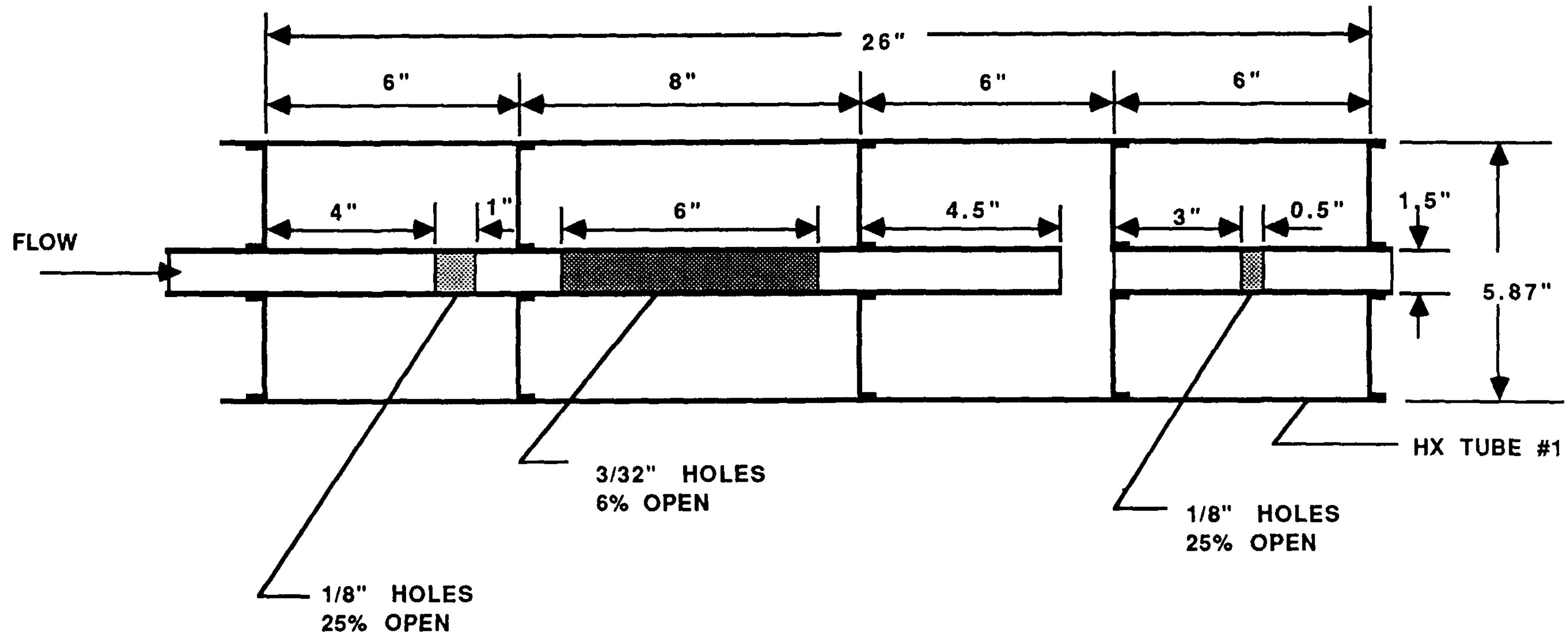
Table 3.2 DETAILED DESIGN AND PERFORMANCE SPECIFICATIONS FOR PROTOTYPE CONFIGURATION

Overall Heat Exchanger:	
Heat Recovery Rate	18,950 Btu/hr
HX Effectiveness	0.683
Overall Heat Transfer Conductance (UA)	81.1 Btu/hr-°F
Number of Transfer Units (NTU)	1.24
Material of Construction	304 Stainless Steel
Tube Wall Thickness	0.065 in
Core Inside Diameter	5.87 in
Core Outside Diameter	8.76 in
Core Length	26 in
Core Weight	74 lbm
Exhaust Gas Side:	
Flow Rate	250 lbm/hr
Inlet Temperature	800 °F
Outlet Temperature	510 °F
Fin Corrugation Height	0.25 in
Fin Corrugation Spacing	0.25 in
Fin Thickness	0.04 in
Radial Gap	0.105 in
Annulus Height	0.685 in
Reynolds Number	977
Heat Transfer Coefficient	4.19 Btu/hr-sq ft-°F
Fin Efficiency	86.6 %
Pressure Drop	0.30 in w.c.
Working Fluid Side:	
Flow Rate	800 lbm/hr
Inlet Temperature	375 °F
Outlet Temperature	428 °F
Helical Baffle Height	0.25 in
Helical Baffle Pitch Spacing	1.0 in
Reynolds Number	3,190
Heat Transfer Coefficient	86.9 Btu/hr-sq ft-°F
Pressure Drop	0.53 psi

muffler which fits within the inside tube of the heat exchanger. The muffler, shown schematically in Figure 3.7, is a simple straight-through geometry designed to fit within the available space envelope and provide a maximum back pressure of 10 in W.C. or less. This allows an exhaust gas pressure drop in the heat exchanger of as much as 5 in W.C. (see Table 3.1) which is well above the estimated value of 0.3 in W.C. and consequently provides a significant margin for error. After exiting from the muffler, the exhaust gas stream reverses direction and flows through the gas-side annulus (between tubes 2 and 3). The exhaust gas is then discharged readily upward through a tube in the outer wall of the heat exchanger.

The working fluid enters the heat exchanger through an inlet manifold and is divided into separate streams by the two inlet tubes leading to the inner and outer annuli. Each fluid annulus contains a helical baffle formed by winding a continuous length of 1/4" tubing around the large inner tube of the annulus. The fluid stream in each of the two annuli is directed by the baffle so as to flow helically along the length of the annulus between adjacent turns of the baffle in an overall direction counter to the flow direction of the exhaust gas stream. The two fluid streams exit through two small diameter tubes and rejoin into a single stream in an outlet manifold.

The core of the prototype heat exchanger is 5.87 inches inside diameter, 8.76 inches outside diameter, and 26 inches in length. The weight is estimated to be approximately 74 pounds. As indicated in Table 3.2, the waste heat recovery rate achievable with this configuration is projected to be around 19,000 Btu per hr at the stated exhaust gas and working fluid conditions. The remaining portions of the Phase I effort were aimed at verifying this projected performance by fabricating and testing a prototype heat exchanger based on the selected design configuration.



NOTE: Designed for Maximum Backpressure of 10 in W.C.

Figure 3.7 SCHEMATIC OF MUFFLER DESIGN

4.0 PROTOTYPE FABRICATION AND ASSEMBLY

The specific design details of the prototype configuration were finalized. Fabrication and assembly drawings were prepared for the various components and the overall heat exchanger. The detailed design drawings were issued to selected vendors, as well as AMTI's machine shop for fabrication of the necessary components.

The four concentric tubes which serve as the framework for the overall configuration (see Figure 3.6) were fabricated by rolling and seam-welding 304 stainless steel sheet metal of the appropriate thickness. The seam-weld on each tube was then ground down to provide an acceptably smooth surface on the inside and outside of the tube.

The helical baffles on the working fluid side were obtained by wrapping 1/4 inch O.D. by 0.035 inch wall thickness stainless steel tubing around Tubes #1 and #3 with a one inch pitch. The winding was accomplished by mounting the large tube in a lathe as shown in Figure 4.1 for Tube #1. The 1/4 inch tubing was tack-welded to the large tube every 90° of rotation to hold it in place. When the wrapping was completed, the cut ends of the 1/4 inch tubing were sealed by inserting and welding end plugs. A 1/16 inch hole was drilled in each end of the 1/4 inch tubing to relieve any pressure buildup which might occur.

The corrugated stainless steel material for the exhaust gas-side fins was obtained from a commercial vendor specializing in fabrication of fin material. The material was bonded to the O.D. of Tube #2 and the I.D. of Tube #3 to provide the fin surface for the exhaust gas annulus. Since the material was supplied in a 13

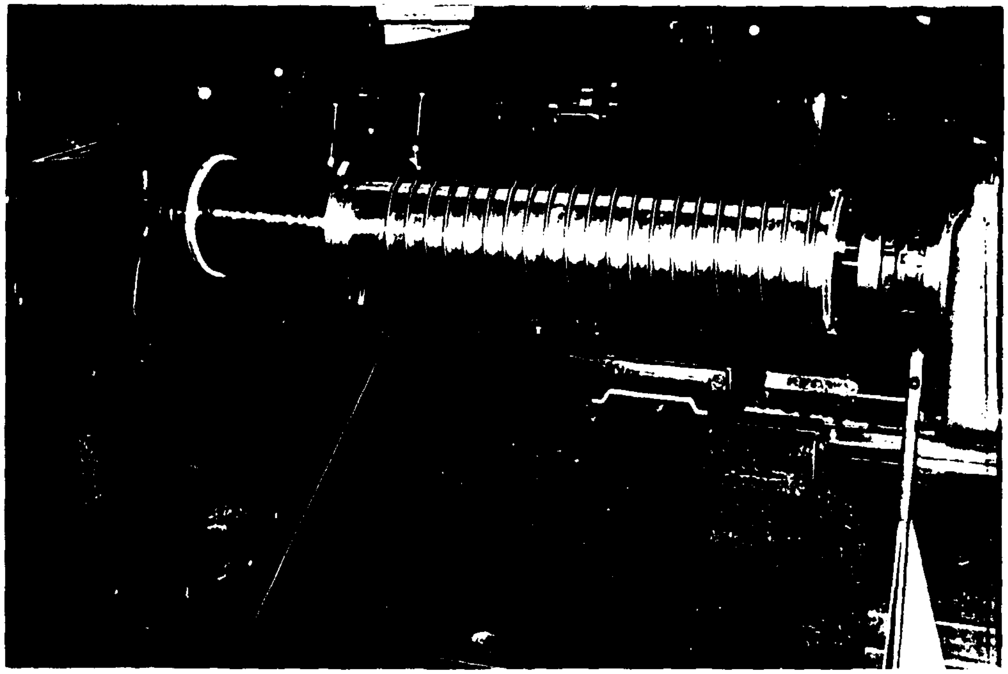


Figure 4.1 WINDING OF HELICAL BAFFLES ON TUBE #1

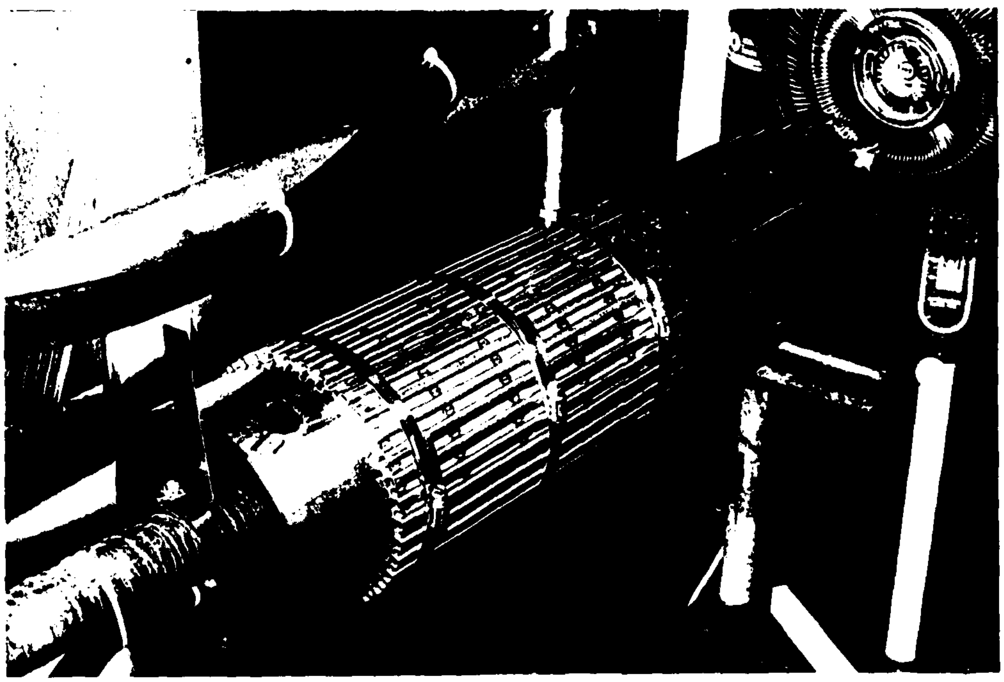


Figure 4.2 SPOT WELDING OF CORRUGATED FIN MATERIAL TO TUBE #2

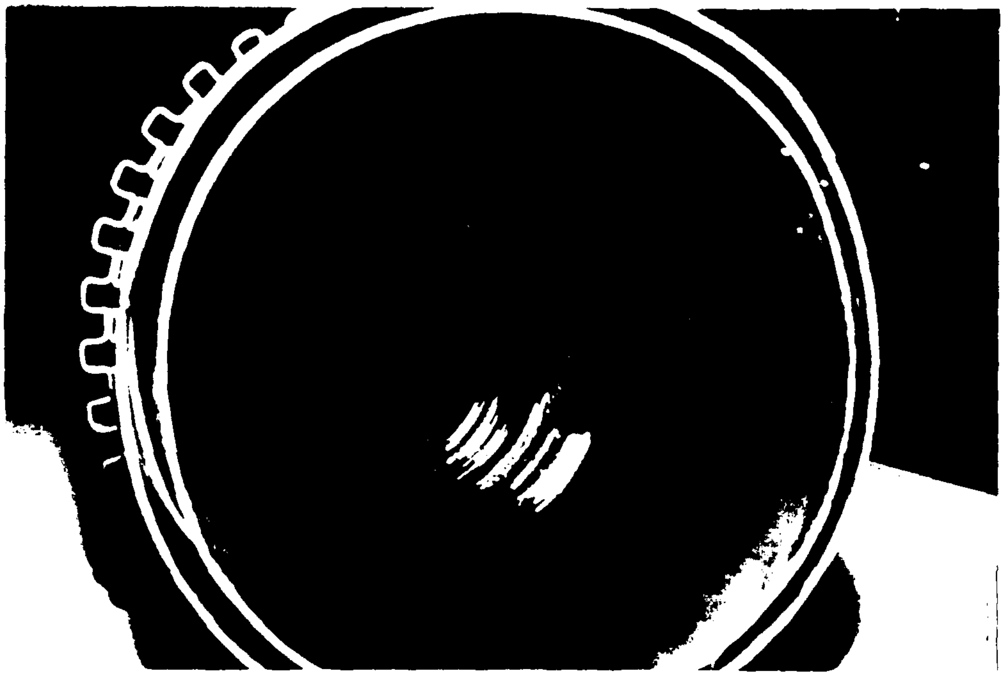
inch width, two pieces were necessary for each tube to give the 26 inch long finned section called for in the design. In production, the material would be bonded to each tube by continuous welding along the length of each corrugation in contact with the tube wall. Since the required tooling was beyond the scope of the Phase I prototype development effort, an alternative approach was adopted. The selected method consisted of using a heavy duty spot-welder to bond the fin material to each tube. This was accomplished by making a large number of spot welds along each corrugation with the spots spaced closely enough together to approximate a continuous weld. The procedure is illustrated in Figure 4.2 which shows Tube #2 mounted in the spot-welder for attachment of the second piece of fin material. As indicated, the material is wrapped around the tube and oriented so that the corrugations align with the first piece which has already been attached. Three band clamps temporarily hold the material in position until the spot-welding is completed. Essentially the same overall procedure was used to attach the corrugated fin material to the inside wall of Tube #3. Once this was accomplished, the helical baffle for the outer working fluid annulus was obtained by wrapping 1/4 inch tubing around the O.D. of Tube #3 in the same manner as described previously for Tube #1.

The most difficult and time-consuming step in the overall assembly procedure was the insertion of Tube #1 (with its helical baffle attached) within Tube #2 (with its fin material attached). After considering various alternatives, a shrink fitting approach was selected. This consisted of shrinking Tube #1 by filling it with dry ice and expanding Tube #2 by heating it with an open flame burner. During this cooling and heating of the tubes, they were mechanically pressed together using a long one-inch diameter threaded rod, two end caps, and a large nut which was turned manually by means of a crescent wrench with an extension to provide leverage. The completed assembly is shown in Figure 4.3 where the upper photograph shows the two tubes immediately following the shrink fitting procedure and the lower photograph shows the front of the assembly following welding of sealing rings which close off the annulus at both ends. The threaded studs seen in the lower photograph are used to attach a cover plate in the final assembly of the heat exchanger.

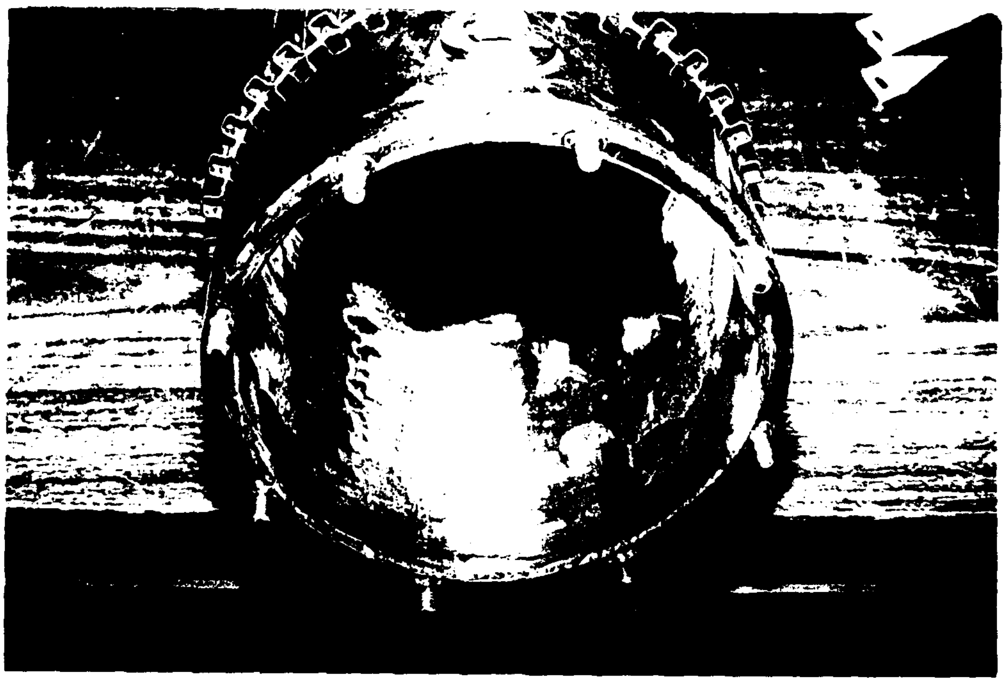
The fabricated components of the muffler designed to fit within the I.D of the heat exchanger are shown in Figure 4.4. These were installed inside the assembly of Tubes #1 and #2, inserting the two separate sub-assemblies from both ends. The outermost muffler baffles were then fuse-welded to the I.D. of Tube #1. The resulting configuration as viewed from the exhaust gas inlet end is shown in Figure 4.5.

The next step in the assembly procedure consisted of inserting Tube #3 (with the corrugated fin material attached to the I.D. and the helical baffle wound around the O.D.) into Tube #4. The difficulty encountered in this step was that Tube #4 was slightly over-sized resulting in an unacceptably large radial gap. Resolution of this problem required cutting Tube #4 lengthwise and wrapping it tightly around Tube #3. The resulting overlap was scribed and then removed with a saw. Tube #4 was then refitted around Tube #3 and the seam in Tube #4 was butt-welded. The O.D. was ground in the weld area to smooth out the outer surface. The assembly of Tubes #3 and #4 was completed by installing and welding rings to seal off the annulus at both ends and flanges for attachment of the cover plates. An interior view from one end of the Tube #3/#4 assembly is shown in Figure 4.6.

The final assembly of the overall prototype heat exchanger configuration consisted of installing the Tube #1/#2 assembly



(a.) Prior to Sealing of Annulus



(b.) With Sealing Ring Installed and Welded
(View from Exhaust Gas Inlet End)

Figure 4.3 COMPLETED ASSEMBLY OF TUBES #1 AND #2

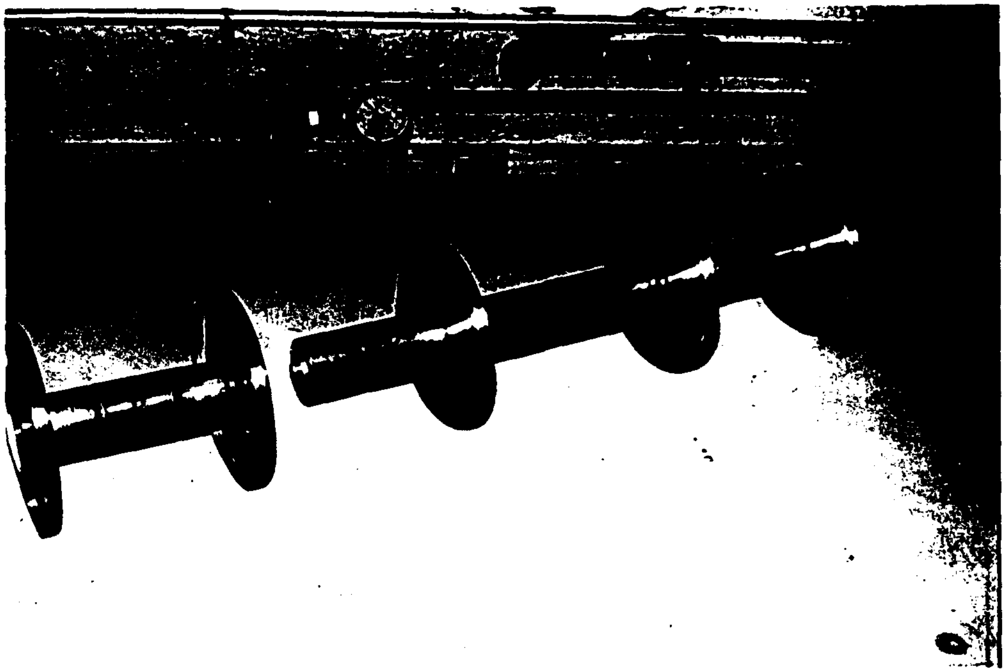


Figure 4.4 MUFFLER COMPONENTS

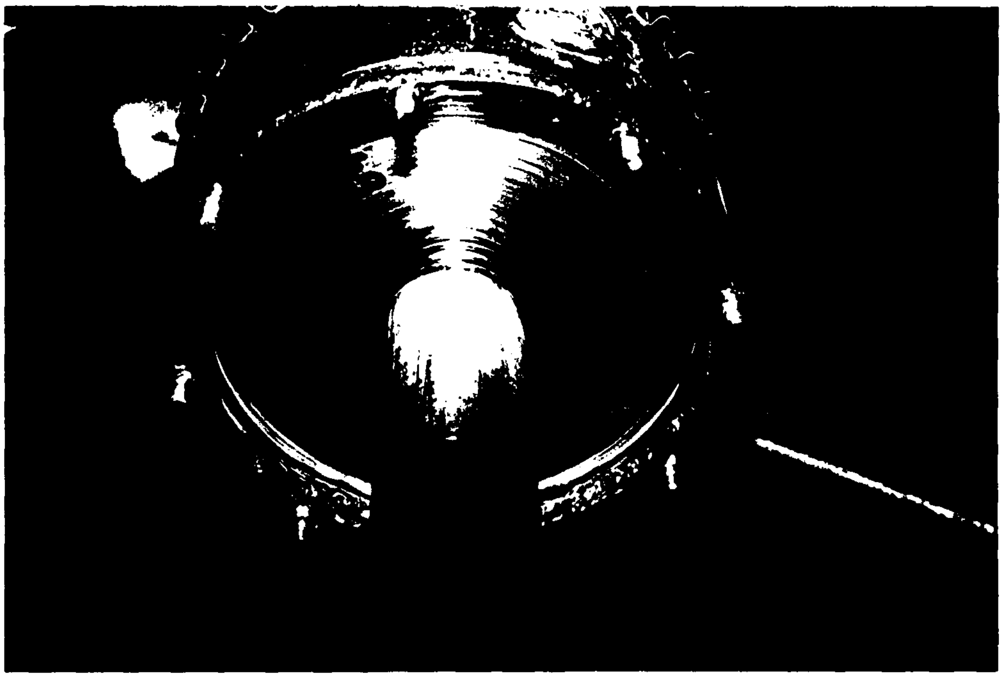


Figure 4.5 MUFFLER INSTALLED IN TUBE #1/#2 ASSEMBLY

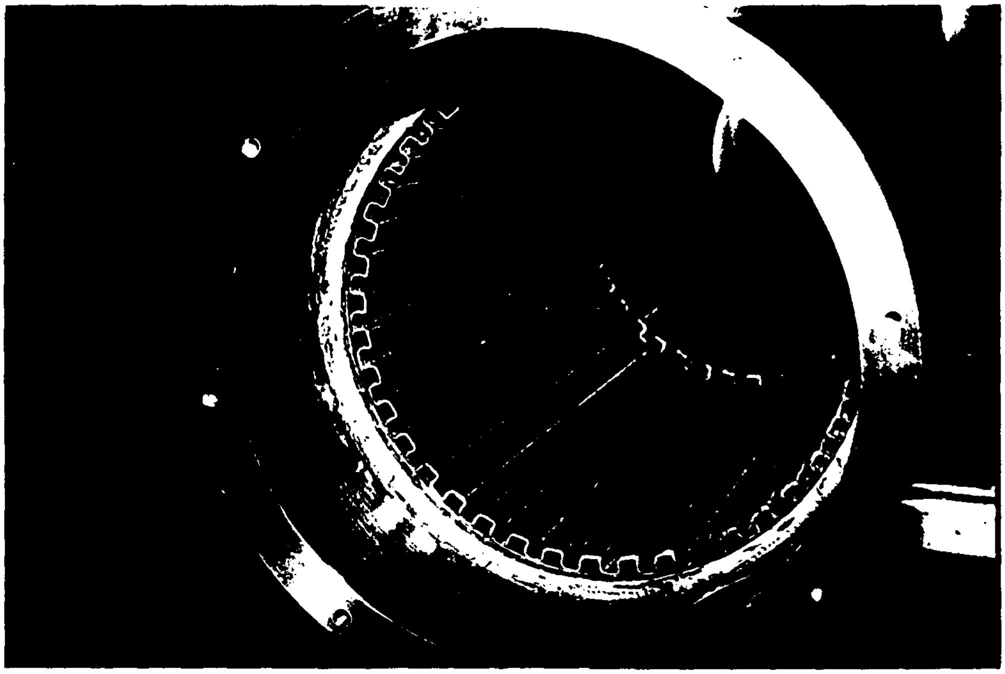


Figure 4.6 COMPLETED ASSEMBLY OF TUBES #3 AND #4

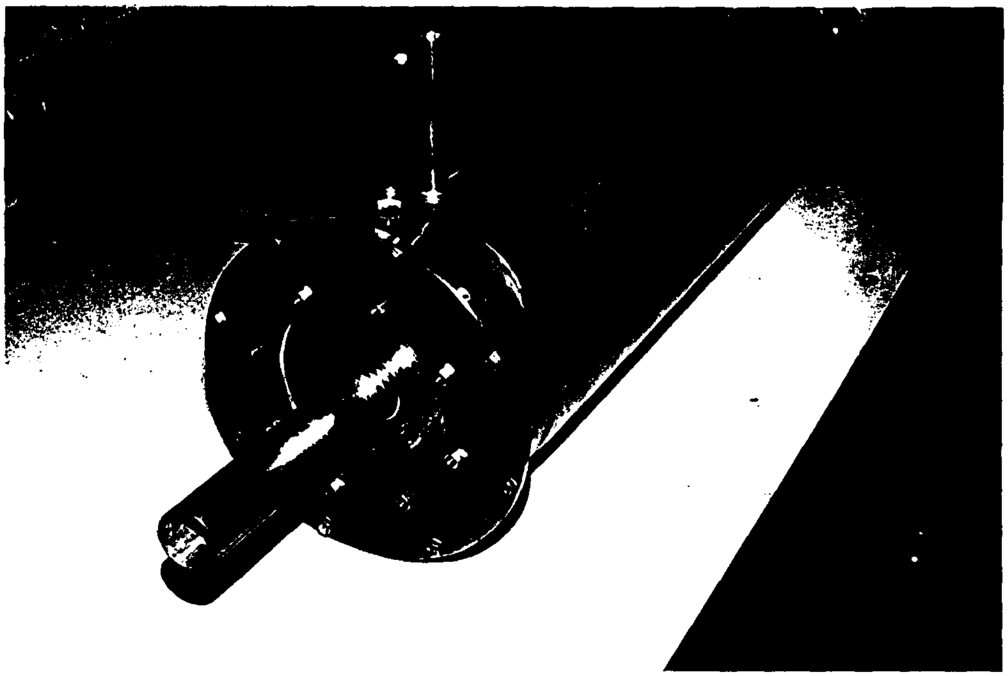


Figure 4.7 PROTOTYPE HEAT EXCHANGER VIEWED FROM EXHAUST GAS INLET END

within the Tube #3/#4 assembly, installing and brazing the working fluid inlet and outlet tubes, and attaching the cover plates at both ends of the heat exchanger. Figure 4.7 shows a view of the overall configuration from the exhaust gas inlet end. The configuration viewed from the opposite end (that is, the muffler outlet end) is shown in Figure 4.8. The prototype heat exchanger shown in these photographs is complete except for the manifold fittings which would be installed to connect the two working fluid tubes together at both the inlet and at the outlet of the heat exchanger. A closeup of the muffler outlet end of the heat exchanger with the access cover plate removed is given in Figure 4.9. This illustrates the manner in which the interior of the exhaust gas passages would be accessed for cleaning purposes in actual use of the waste heat recovery heat exchanger.

Figure 4.8 PROTOTYPE HEAT EXCHANGER VIEWED FROM MUFFLER OUTLET END

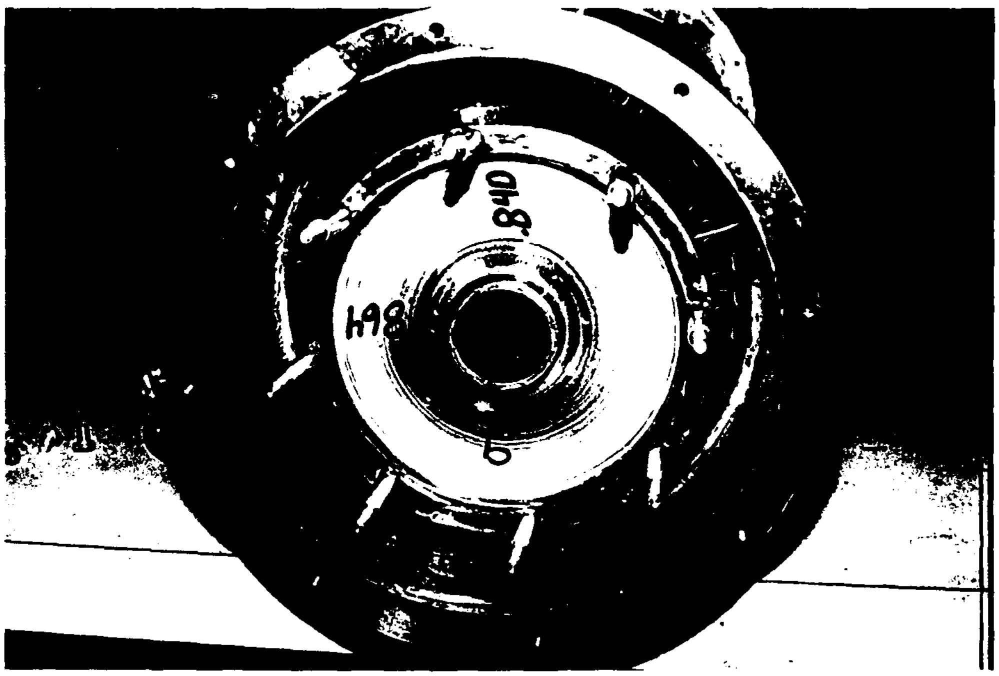


Figure 4.9 PROTOTYPE HEAT EXCHANGER WITH ACCESS COVER REMOVED

5.0 PERFORMANCE TESTING OF PROTOTYPE HEAT EXCHANGER

5.1 Description of Test Program

The major goals of the testing program were to demonstrate the feasibility of the heat exchange design approach and confirm the anticipated performance of the prototype configuration. To accomplish this purpose the thermal performance and pressure drop of the prototype heat exchanger were measured in a series of parametric tests over a representative range of operating conditions. The tests were conducted using water as the working fluid and an available open-flame diesel burner as the source of hot gas simulating engine exhaust. The experimental parameters consisted of flow rate and temperature of the hot gas stream and the flow rate of water on the working fluid side of the heat exchanger. The test values of these parameters corresponded to anticipated conditions under actual operation of the heat exchanger. The experimental results were used to calibrate the heat exchanger design model and refine the projected performance of the heat exchanger.

5.2 Overall Test Loop

The test loop used in the experimental measurement of prototype performance is shown schematically in Figure 5.1. The required flow of hot gas was induced by an industrial blower connected to the outlet of the heat exchanger. Since the available blower did not have sufficient pressure rise capability to overcome the design pressure drop through the muffler, all testing was conducted with a ducting arrangement which bypassed the muffler. This consisted of capping the muffler inlet pipe and replacing the access cover

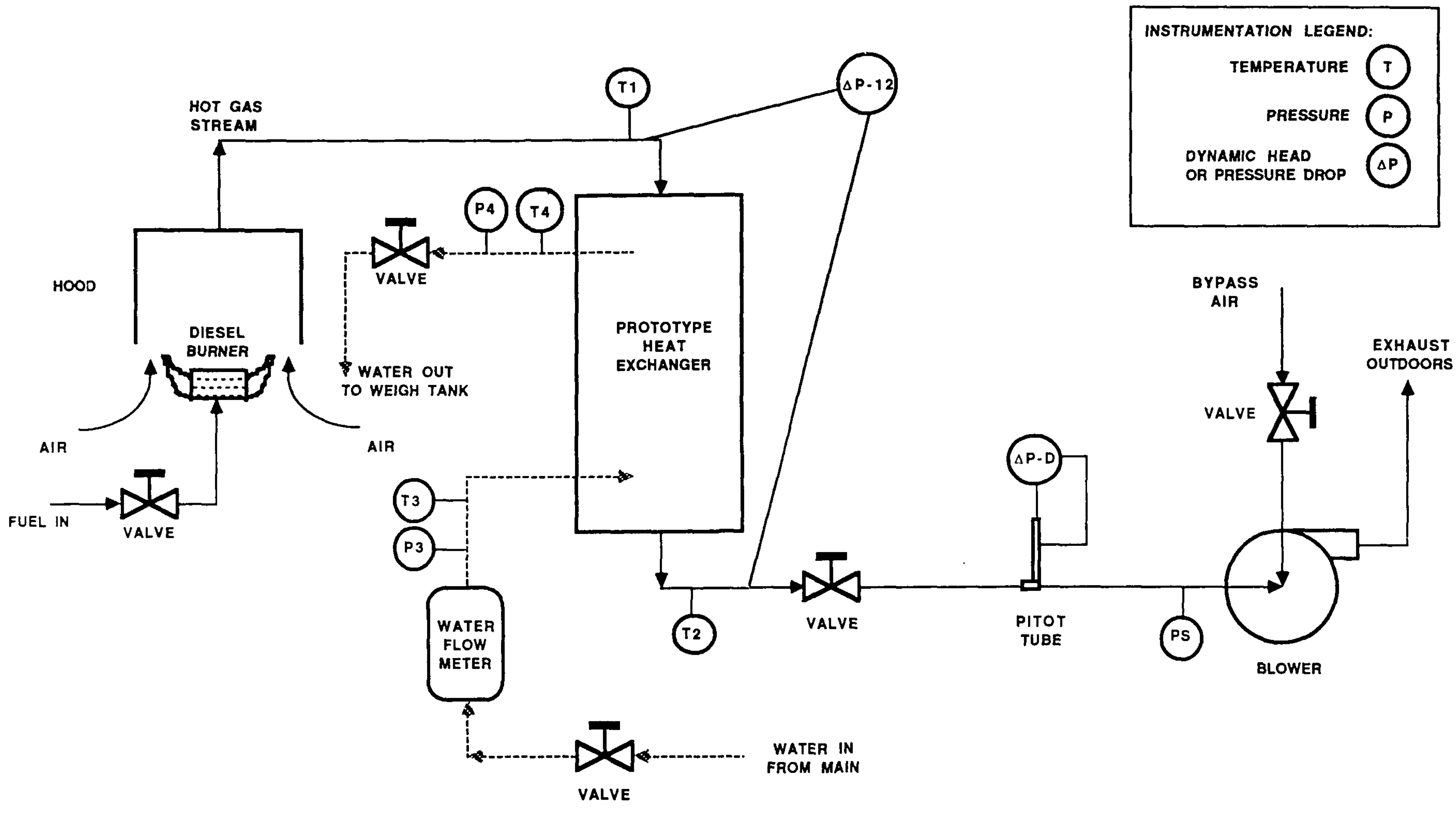


Figure 5.1 HEAT EXCHANGER TEST LOOP SCHEMATIC

(at the muffler outlet) with a temporary cover containing a 1.5 inch pipe. The hot stream was then introduced through this pipe directly into the heat exchanger at the inlet of the exhaust gas annulus. With this approach, the test results would be expected to be somewhat conservative in terms of thermal performance, since they would not include any heat transfer which would occur in the muffler during actual operation.

The hot gas stream was provided by an open-flame diesel burner located under a hood connected to the inlet of the heat exchanger. When the blower is operating, the combustion products from the burner along with secondary air are induced to flow through the exhaust gas side of the heat exchanger. By adjusting the firing rate of the burner, the temperature of the resulting hot gas stream was controlled to achieve the desired value for any particular set of test conditions.

The flow rate of the hot gas stream was measured using a single pitot tube mounted in the center of the outlet pipe from the heat exchanger. The pressure drop across the pitot tube (dynamic head) was measured using an inclined manometer with 0 to 2 inches of water column. The pitot tube was pre-calibrated against a precision ASME orifice. The orifice could not be used to measure flow during the tests because it contained plastic fittings which would have been damaged by the high temperatures.

The hot gas inlet temperature was measured using a Type K thermocouple probe; all other temperatures were measured using Type T thermocouple probes. The gas stream pressure drop across the heat exchanger and the static pressure at the blower inlet were measured using 0-5 in W.C. Magnehelic gages. The pressures on the water side were measured by 0- 15 psig Bourdon gages. Water flow rate was determined both by means of a weigh tank and by a piston type positive displacement water meter.

Each test consisted of making measurements at a given set of steady state conditions. The procedure followed in each test consisted of first setting the water flow rate and hot gas flow rate at the desired values by adjusting the appropriate flow control valves. The temperature of the hot gas stream was then set to the desired value by adjusting the burner firing rate. Once steady state was attained, all of the instruments were read and recorded every few minutes for a total test period of around twenty minutes. The average values of the various parameters over the test period were then used to define each set of test conditions and corresponding performance.

5.3 Discussion of Experimental Results

A total of sixteen tests were conducted investigating the effects of independent parametric variations in hot gas flow rate (95-256 lbm per hour), hot gas inlet temperature (410-812°F) and water flow rate (268-788 lbm per hour). The pertinent experimental results obtained are given in Table 5.1. The last few columns show the heat balance between the gas side and water side measurements for each test. This comparison provides a measure of the accuracy and consistency of the experimental data. Examination of the data shows that the two sets of values of heat recovery rate agree within approximately ± 8%. This is quite reasonable and provides confidence in the accuracy of the experimental performance results.

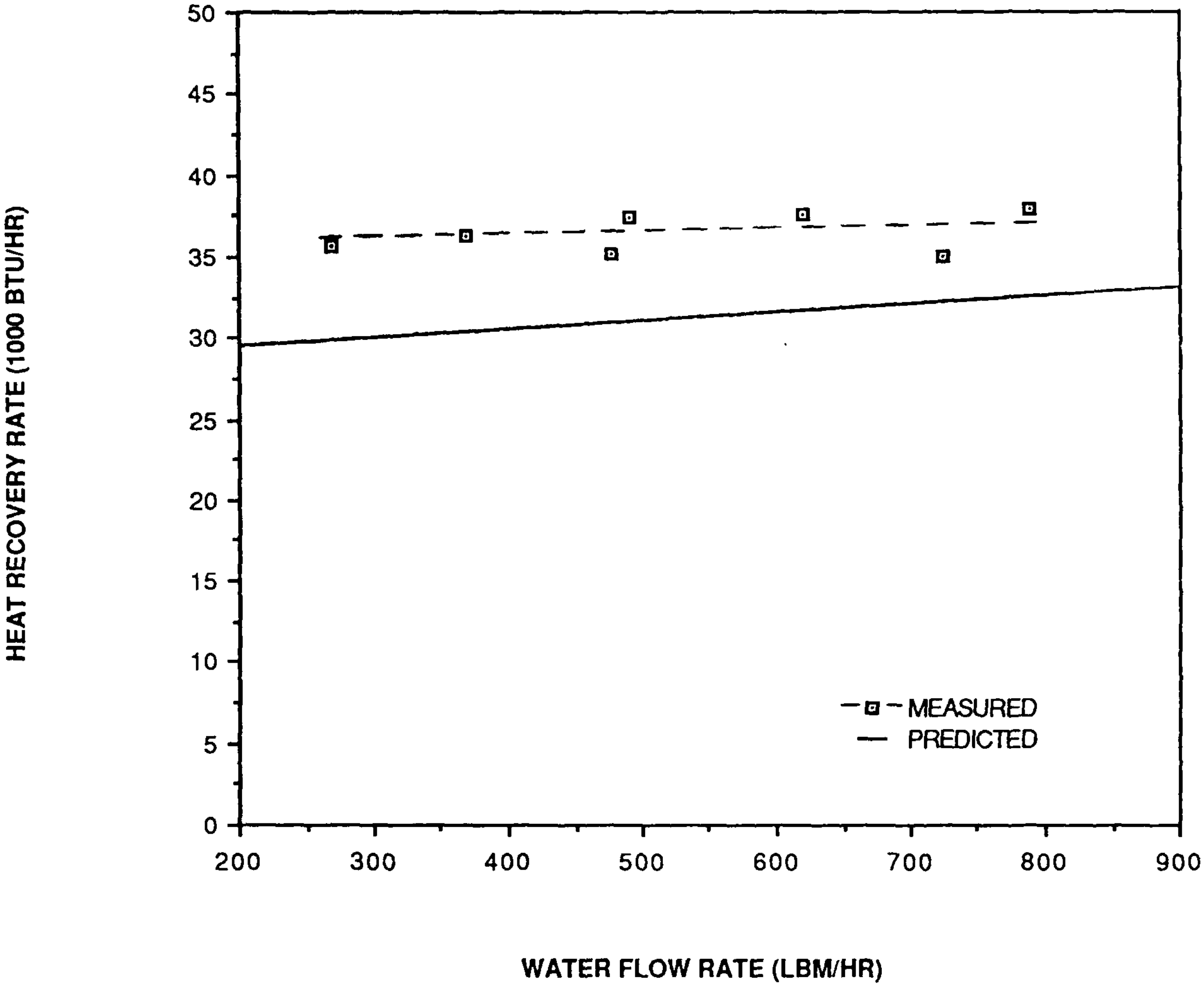
To provide a calibration of the heat exchanger design model and finalize the projected performance of the prototype configuration, the test results were compared to predicted values of heat recovery rate based on calculations performed using the analytical model.

Table 5.1 SUMMARY OF TEST DATA

Test No.	Hot Gas Side Measurements				Water Side Measurements				Heat Recovery Rate (Btu/hr) Calculated from Test Data		
	Flow Rate (lbm/hr)	Inlet Temp. (°F)	Outlet Temp. (°F)	Pressure Drop (in w.c.)	Flow Rate (lbm/hr)	Inlet Temp. (°F)	Outlet Temp. (°F)	Pressure Drop (psi)	Based on Gas-Side Data	Based on Water-Side Data	Average of Both
1	242	792	164.7	4.3	477	67.6	135.5	1.5	38,000	32,400	35,200
2	244	696	155.2	4.4	477	67.7	127.2	1.5	33,000	28,400	30,700
3	246	592	144.7	4.5	477	67.7	117.6	1.6	27,600	23,800	25,700
4	252	498	133.8	4.6	473	67.8	108.6	1.5	22,500	19,300	20,900
5	256	410	123.4	4.7	474	67.9	101.2	1.5	18,000	15,800	16,900
6	186	803	139.6	2.4	469	68.1	124.0	1.5	30,800	26,200	28,500
7	149	812	120.6	1.6	467	68.2	113.7	1.8	25,700	21,200	23,500
8	95	810	91.3	0.6	467	68.2	96.6	1.8	17,100	13,200	15,200
9	239	784	150.7	4.3	723	67.7	112.2	1.8	37,900	32,200	35,000
10	234	806	172.4	4.2	268	69.6	197.0	1.5	37,100	34,100	35,600
11	246	806	171.0	4.7	369	67.8	159.0	1.8	39,000	33,600	36,300
12	249	802	152.8	4.6	620	66.0	121.8	1.8	40,500	34,600	37,500
13	253	798	131.0	4.6	788	65.5	107.9	1.9	42,200	33,400	37,800
14	246	804	150.6	4.5	491	66.3	136.9	1.7	40,200	34,700	37,400
15	202	800	138.3	2.9	486	66.8	126.3	1.7	33,400	28,900	31,200
16	167	810	126.7	1.8	487	66.8	117.2	1.7	28,500	24,500	26,500

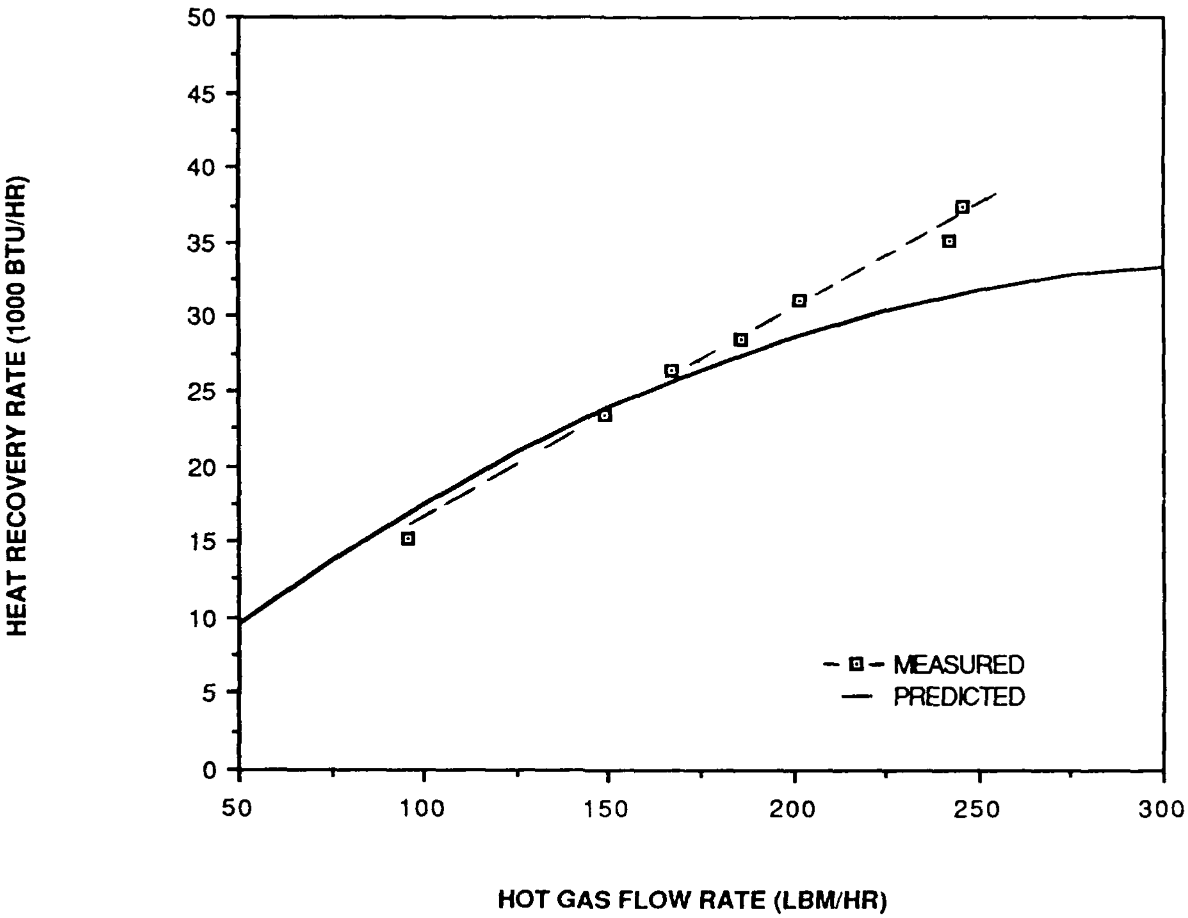
The calculations were performed using fluid properties for water and the test values of flow rates and inlet temperatures. Figures 5.2-5.4 show the comparison between measured and predicted variations in heat recovery rate with water flow rate, hot gas flow rate, and hot gas inlet temperature, respectively. The trends exhibited by the experimental data agree reasonably well with the results of the calculation model predictions. In general, however, the measured values of heat recovery rate are consistently higher than the predicted values. This is illustrated more clearly by the overall comparison shown in Figure 5.5 which summarizes all of the test data. These results then indicate that the design model is somewhat conservative in predicting thermal performance. Examination of the data shows that the opposite is true in terms of pressure loss. On both sides of the heat exchanger, the measured values of pressure drop are significantly higher than the predicted values by a factor of around 20 for the hot gas stream and 10 for the water stream. Fortunately, both of these values are within acceptable limits on pressure drop for each side of the heat exchanger.

The experimental information in conjunction with the analytical model were used to finalize the estimated heat exchanger performance with Syltherm-800 as the working fluid at the selected design conditions. This involved using the calibrated model with the appropriate values of flow rate, fluid properties and inlet temperature associated with Syltherm. The resulting final specifications for the prototype heat exchanger are given in Table 5.2. On the basis of these results, it is anticipated that the prototype heat exchanger will perform quite satisfactorily in the intended waste heat recovery application.



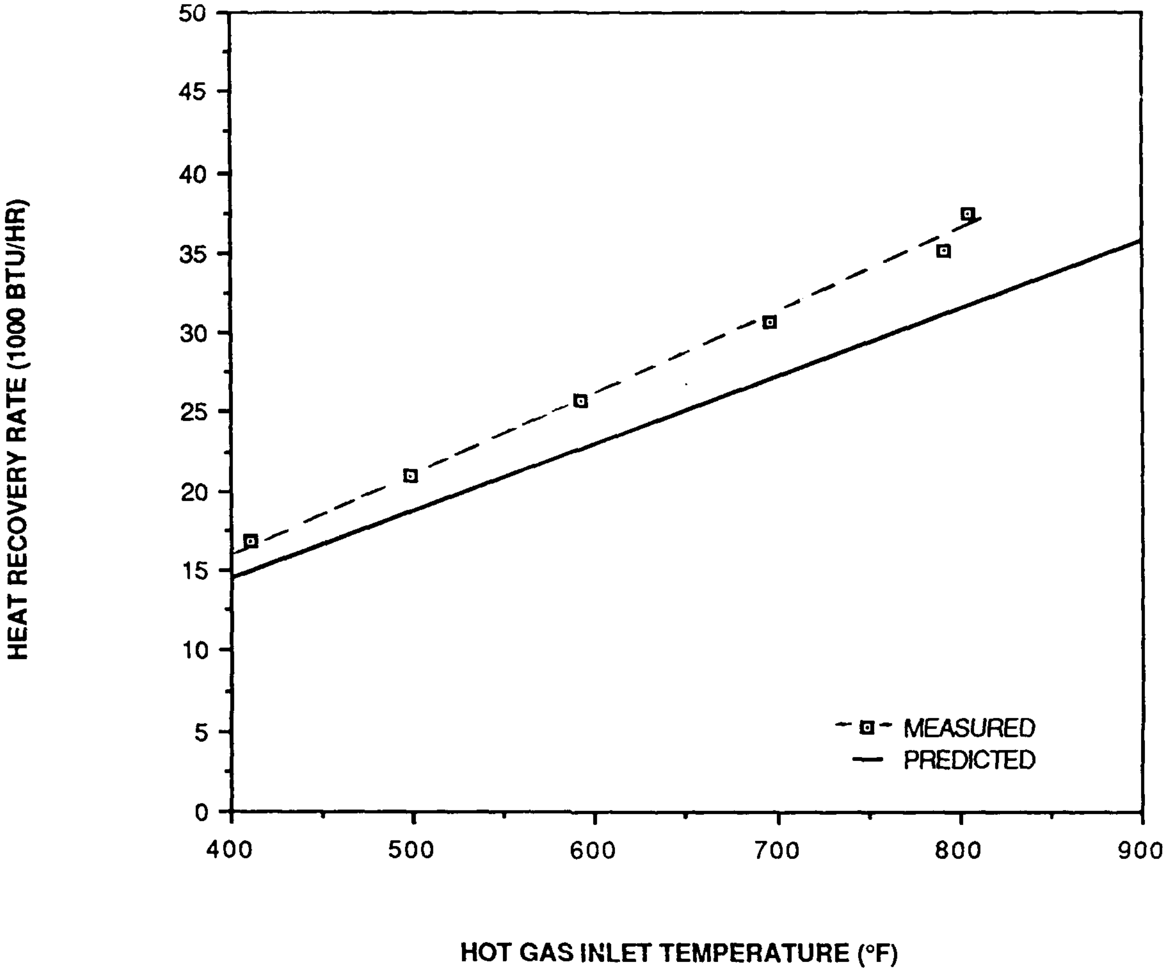
NOTE: Data from Tests 1, 9, and 10-14

Figure 5.2 VARIATION OF HEAT RECOVERY RATE WITH WATER FLOW RATE FOR PROTOTYPE HEAT EXCHANGER



NOTE: Data from Tests 1, 6-8, and 14-16

Figure 5.3 VARIATION OF HEAT RECOVERY RATE WITH HOT GAS FLOW RATE FOR PROTOTYPE HEAT EXCHANGER



NOTE: Data from Tests 1-5 and 14.

Figure 5.4 VARIATION OF HEAT RECOVERY RATE WITH HOT GAS INLET TEMPERATURE FOR PROTOTYPE HEAT EXCHANGER

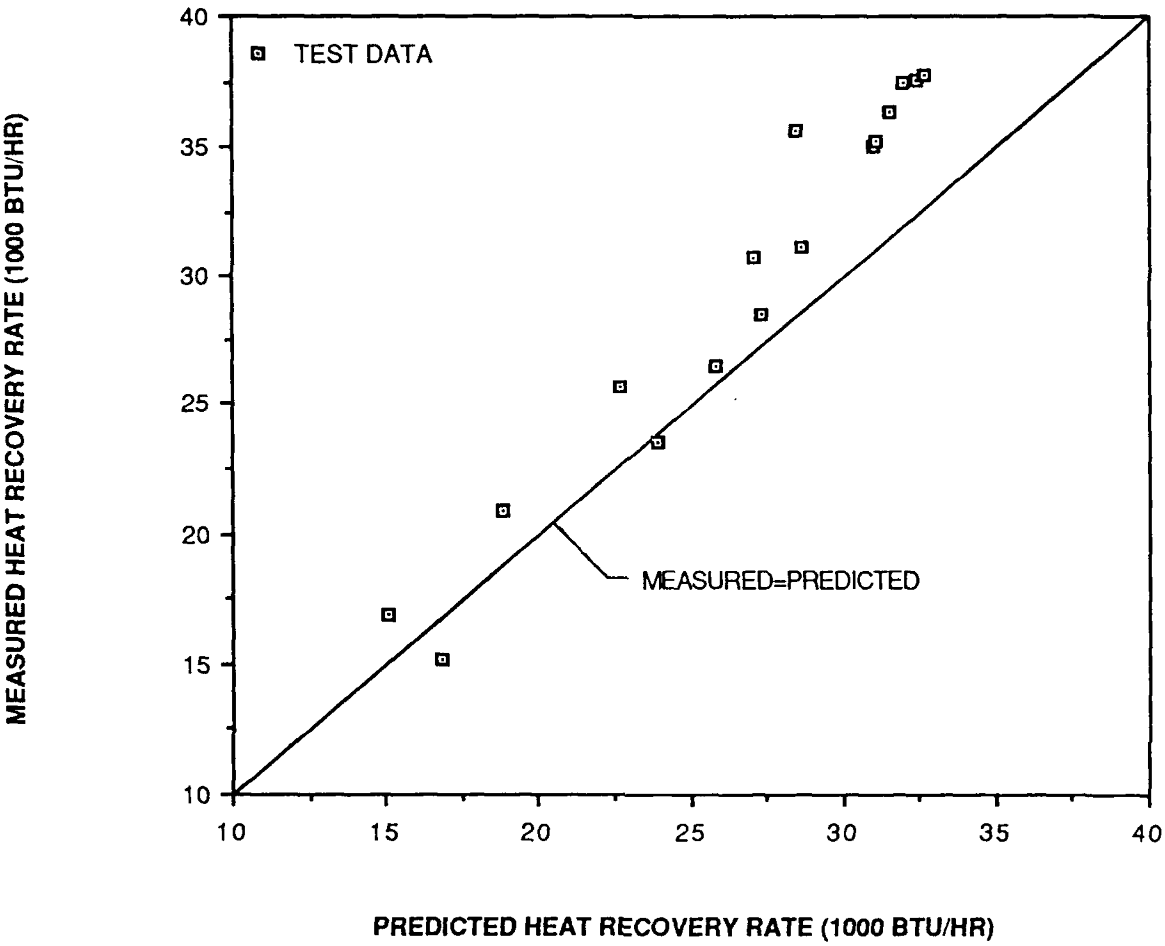


Figure 5.5 OVERALL COMPARISON OF MEASURED AND PREDICTED VALUES OF HEAT RECOVERY RATE FOR PROTOTYPE HEAT EXCHANGER

Table 5.2 FINAL PERFORMANCE SPECIFICATIONS FOR PROTOTYPE HEAT EXCHANGER

Waste Heat Recovery Rate	19,900 Btu/hr
Exhaust Gas Stream:	
Flow Rate	250 lbm/hr
Inlet Temperature	800 °F
Outlet Temperature	495 °F
Pressure Drop	4.5 in W. C.
Working Fluid (Syltherm-800) Stream:	
Flow Rate	800 lbm/hr
Inlet Temperature	375 °F
Outlet Temperature	430 °F
Pressure Drop	2 psi

6.0 CONCLUSIONS

A preliminary program to investigate the feasibility of an innovative design concept for waste heat exchangers has been successfully completed. A prototype heat exchanger based on the concept has been designed, fabricated, and tested. The major conclusions which have been reached on the basis of the overall program results are as follows:

- Utilization of the selected design concept results in a compact heat exchanger configuration which is well-suited for applications involving waste heat recovery from diesel engine exhaust streams. The concept provides distinct advantages in packaging, resistance to fouling, and ease of cleaning of the exhaust gas passages.

- Fabrication of the components of the prototype heat exchanger was relatively straightforward. Assembly of the components into the overall configuration proved to be more difficult than anticipated. Further development of the concept will need to focus on identifying suitable means of maintaining acceptable tolerances on parts of the assembly which must fit together tightly.

- Testing of the prototype heat exchanger showed that the analytical design model was sufficiently accurate in predicting heat recovery rate, but overly optimistic in predicting pressure loss on both sides of the heat exchanger. In general, however, the test results demonstrated that the prototype heat exchanger is suitable for the intended application and will provide satisfactory performance. The scope of the testing was insufficient to determine the long term reliability of the concept, but the results obtained were promising enough to warrant further investigation.

7.0 RECOMMENDATIONS

Additional experience under actual operating conditions is required to determine conclusively the long term feasibility of the concept. To accomplish this purpose, a detailed experimental investigation of the prototype heat exchanger is recommended. The investigation should consist of testing the long term fouling tendencies and serviceability of the prototype in recovering waste heat from the exhaust of an actual diesel engine.

In parallel with the testing program, a manufacturing study should be conducted using the prototype configuration as a reference design. The study should focus on identifying potential fabrication and assembly procedures suitable for low-cost manufacture of the heat exchanger in production quantities.

8.0 REFERENCES

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