HEAT TRANSFER CHARACTERISTICS
OF A
FLUIDIZED BED - STIRLING ENGINE SYSTEM

A Thesis
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by
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A fluidized bed combustion (FBC) system was designed to provide heat energy to the head of a Stirling cycle engine. Preliminary testing with a simulated heat exchanger head was done to investigate the heat transfer rate from the fluidized sand bed to the cooling air inside the simulated head. A Stirling Technology, Inc. model ST-5 Stirling engine was tested in a custom built FBC. The heat transfer coefficients between the fluidized sand bed and the engine head (outside) and between the engine head and the engine working fluid (inside) were calculated along with an overall efficiency for the engine tests. The results show that the heat transfer rate into the Stirling engine was limited by the inside heat transfer coefficient and the overall efficiency of the system was 4.5 percent. A analysis on improving the overall efficiency was also done.
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LIST OF SYMBOLS

A  heat transfer surface area on one side of an exchanger, $m^2$

$c_p$  specific heat of a fluid, kJ/kg K

$E_a$  overall absolute error

$D$  diameter of a tube, m

$D_h$  hydraulic diameter of flow passages, m

$DT$  temperature difference, $^\circ C$, K

$DTLM$  log mean temperature difference, $^\circ C$

$EAIN$  air energy in, W

$EAOOUT$  air energy out, W

$EFIN$  fuel energy in, W

$EFOUT$  fuel energy out, W

$ELOSS$  energy lost through insulation, W

$EW$  energy lost in engine cooling water, W

$F$  log-mean temperature difference correction factor, dimensionless

$f$  apparent Fanning friction factor, dimensionless

$H$  heat transfer coefficient, W/m$^2$ K

$h$  enthalpy of air, kJ/kg

$k$  thermal conductivity, W/m K

$L$  characteristic length, m

$ln$  natural logarithm

$Mu$  viscosity, kg/s m

$m$  mass flow rate, kg/s

$N$  computed value in error analysis

$NF$  fin efficiency, dimensionless

$Nu$  Nusselt number, dimensionless

$PI$  constant, 3.1416

$Pr$  Prandtl number, dimensionless

$Q$  heat transfer rate, W

$R$  thermal resistance based on surface area A, K/W

$Re$  Reynolds number based on the hydraulic diameter, dimensionless

$Rc$  fluid density, kg/m$^3$

$r$  radius of a tube, m

$s$  standard deviation in error analysis

$T$  temperature, $^\circ C$ or K

$TC$  thermocouple

$U$  overall heat transfer coefficient, W/m$^2$ K

$UA$  constant for insulation tests, W/K
\( u \) independent variable in error analysis

\( V \) velocity, m/s

Subscripts

\begin{align*}
a & \quad \text{air} \\
c & \quad \text{cold fluid side} \\
f & \quad \text{fin} \\
h & \quad \text{hot fluid side} \\
i & \quad \text{inside or inlet} \\
o & \quad \text{outside or outlet} \\
p & \quad \text{prime} \\
s & \quad \text{sand} \\
tot & \quad \text{total} \\
w & \quad \text{wall}
\end{align*}
CHAPTER I

INTRODUCTION

With the recent decline in the value of many agricultural products, alternative uses for crops have been sought to help open up new markets. One area that is being investigated is power generation using crop residues and grains such as corn for a fuel source. If this technology can be developed for large scale operations, such as electric power generation, or small scale systems, such as home heating and on-farm drying of grain crops, potentially large new markets for these renewable resources could develop.

The purpose of this project was to combine a commercially available stirling engine with a fluidized bed combustor (FBC) as the engine heat source. This combination was selected to take advantage of the high heat transfer rates, multifuel capabilities, and the ability to control the combustion process that the FBC offers along with the reliability and adaptability of the stirling engine.
Stirling engines and FBC systems have existed separately for many years and in recent years both have been increasingly researched but for very different applications.

The engine chosen for this project was an ST-5 Stirling engine from Stirling Technology in Athens, Ohio (Appendix E). This engine was chosen because of its availability, rated power output and the proximity of the company to the research located at the Ohio Agricultural Research and Development Center (OARDC) in Wooster, Ohio.

A custom FBC was designed and built at OARDC to directly heat the head of the Stirling engine in a sand bed (Appendix E). The design was based on previous FBC work at OARDC. Powdered corncobs were used as the fuel in the FBC.
CHAPTER II
OBJECTIVES

This investigation focused on the operation of an ST-5 model stirling engine coupled with a custom built atmospheric fluidized bed combustor heat source. The objectives of this research were to:

1) Experimentally determine the overall, outside, and inside heat transfer coefficients for the head of the stirling engine immersed in an atmospheric fluidized bed combustor.

2) Experimentally determine the overall system efficiency, defined as the ratio of the power output to the fuel energy input.

3) Synthesize and theoretically analyze new systems to improve the overall efficiency.

4) Experimentally determine the overall efficiency of the improved system.
CHAPTER III
LITERATURE REVIEW

3.1 STIRLING ENGINE

The Stirling cycle was patented in 1816 by the Scottish minister Robert Stirling. The earliest applications of the stirling engine were low power output prime movers such as water pumps and kerosene fans. Because of their reliability and ease of use, thousands of stirling engine powered machines were built in the second half of the 19th century before the advent of the internal combustion engine. However, the stirling engine could not compete with much smaller internal combustion engines on a power-to-weight ratio basis because of limited materials and technology available. By the 1920's, stirling engines had disappeared from use.

With recent advances in materials for high temperature applications and with fossil fuel shortages, stirling engines may be making a comeback in specialized areas. Currently the largest expenditures for stirling engine development are being made in the field of vehicle propulsion and small free piston systems (Richmond, 1985).
Fig. 1. Theoretical representation of Stirling cycle.

Theoretically, Stirling engines operate on a Stirling cycle in which the enclosed working fluid undergoes a series of four internally reversible processes (See Figure 1): isothermal compression from state 1 to state 2 at temperature $T_c$, constant volume heating from state 2 to state 3, isothermal expansion from state 3 to state 4 at temperature $T_h$, and constant volume cooling from state 4 to state 1 to complete the cycle. A regenerator which is 100% effective would allow the heat rejected during process 4-1 to be used as the heat input during process 2-3. All the external heat added to the working fluid would take place in the isothermal process 3-4 and all the heat rejected to the surroundings would occur in the isothermal process 1-2. From the description of the cycle it is noted that the thermal efficiency of the ideal Stirling cycle is the same.

A thermodynamic analysis of stirling engines has been attempted by many researchers. But, it is difficult to apply any of the information gained from a particular engine to other stirling engines in general. This occurs because: 1) Stirling engines lack structure or unified methods of thermodynamic design, and 2) There is a wide variety of engine configurations. For a comprehensive listing of stirling engine models see Organ, (1987).

Since the stirling engine is an external combustion engine, it has several advantages over internal combustion spark-ignition or compression-ignition engines. The stirling engine offers the opportunity for high efficiency, reduced emissions from the combustion products, multifuel capabilities, maximum reliability, and quiet operation.

3.2 FLUIDIZED BED COMBUSTION

Recent FBC research and development has been mostly geared toward large scale fossil fuel power generation systems. Anderson, et al. (1987a) summarizes the principles of fluidized bed combustors. Researchers at OARDC (Henry, et al. 1983, Keener, et al. 1986, Anderson, et al. 1987b) have developed fluidized bed combustion systems for home heating in the range of 22 kW (Prototype I) and 44 kW (Prototype II). Corncobs, shelled corn and
coal were the primary fuels used while testing these prototypes.

The technology realized from these systems was used to design a fluidized bed combustor to directly heat the head of a stirling engine. The construction of this custom stirling fluidized bed combustor (SFBC) took place at the Department of Agricultural Engineering machine shop at OARDC.

3.3 COMBINED SYSTEM

Many systems have been developed to transfer heat into stirling engines. These systems have demonstrated the multifuel capabilities of the stirling engine. A partial list of the fuels that have been used includes gasoline, kerosene, diesel fuel, natural gas, LP gas, various solid fuels such as firewood and coal, various chemical reactions, and solar energy.

There has been very little published research which relates directly to the engine - burner combination used in this project. It is mentioned as a possibility by Uherka, et al. (1979) in a comparison with other heat transport systems for stationary power generation. Nakajima, et al. (1985) developed a multi-purpose stirling engine to use biomass as a fuel. This system was intended to be a heat and CO₂ supplier for a greenhouse as well as a 2 kW electric generator. The study compared normal burning
of wood chips, with the hot gas from the combustion heating
the stirling heater tubes, versus burning the wood chips in
a FBC and directly heating the heater tubes in the bed.
The results show that the engine heated in the FBC gained
25% more indicated work when the maximum temperature on the
heater tubes was kept the same. The total efficiency of
the normal burner system was stated to be 8.8% and no
efficiencies were given for the FBC system.
CHAPTER IV
EQUIPMENT

Experiments were conducted at The Department of Agricultural Engineering, Ohio Agricultural Research and Development Center (OARDC), Wooster, Ohio. The equipment and instrumentation used in this research were:

1) A model ST-5 stirling engine (No. 704) from Stirling Technology, Inc. in Athens, Ohio. This engine was a reconditioned engine to be used for testing only. The normal production model ST-5 engine is rated at 3.7 kW with 5 bar (gage) internal air pressure (Appendix E). The manufacturer indicated that No. 704 produced 1.25 kW of electric power while powering a belt driven water pump with 3.8 Bar (gage) internal air pressure and a head temperature of 640°C.

2) A custom built atmospheric fluidized bed combustor (FBC) which was labeled Stirling Fluidized Bed Combustor (SFBC). The design was based on technology acquired from earlier FBC work done at OARDC. The SFBC was constructed of 11 gage mild steel. The bed material was Q-ROK Graded Quartzite between No. 16 and No. 24 mesh screens with a
density of 1500 kg/m³.

3) A fuel feeding system (patent pending) to feed powered corncobs to the SFBC. This 0.142 m³ tapered hopper was equipped with a metering device and an SCR motor speed controller to control the fuel feed rate. A stirrer and a vibrator ensured continuous feeder operation.

4) OARDC's Prototype II FBC. This included electronic controls, Dresser Industries Inc. Whispair Max roots blower (No. 7413372), and Reliance Electric Company Duty Master AC motor (No. C56X0776M-SL).

5) Instruments for airflow measurement of the combustion air. A thin-plate orifice with flange taps was used. The static and differential pressures along with the air temperature at the orifice were measured. A Leeds & Northrup differential pressure transmitter (No. 76-64591-3-1) and a Setra 0-50 psig pressure transmitter (model 205-2) were used to send the pressure measurements to the datalogger.

6) Thermoelectric Type K 1/8 in. dia. thermocouples. These thermocouples were used for all temperature measurements. Accuracy is ± 2.22°C between 0-277°C and ± 0.75% of the reading between 277-1260°C. (At 800°C maximum error is ± 6°C)

7) Instruments for measurement of air properties. This consisted of an EG&G model 880-C1 dew point hygrometer (No. 1189) and a Bruel and Kjaer Copenhagen barometer.
8) A GSE Inc. model 5341, 0-50 pound load cell (No. 359) with Newport A/D meter and a magnetic rpm sensor with a Newport A/D meter. These were used in the power output measurements of the engine.

9) A Dywer Instruments Inc. water flow meter model VYC-142 (No. M828). This was used to measure the flow rate of the cooling water through the engine cooling jacket.

10) A Detecto Scales Inc, model 2321AF3 (No. 590766) scale. This scale was used for fuel measurement.

11) A data collection system. This included a digital data collector, a KAYE Instruments System 8000 (No. 1877) connected to a magnetic tape recorder, a MPE tape deck model 2500 (No. 10023) used to load the data into the HP 3000 mainframe computer.
CHAPTER V
THEORY OF HEAT TRANSFER COEFFICIENT CALCULATIONS

5.1 HEAT EXCHANGER HEAD - FLUIDIZED BED

The fluidized bed heat transfer coefficient was evaluated by inserting a heat exchanger within the bed. For the case of a cylindrical heat exchanger, with the internal fluid flow perpendicular to the gas flow in the fluidized bed, the system is represented as a crossflow heat exchanger (Fig. 2). Using results presented in Incropera, et al. (1985), the overall heat transfer coefficient was calculated by rearranging the equation for the heat transfer rate through a cross flow heat exchanger;

\[ U = \frac{Q}{(A \times F \times DT)}. \]  \hspace{1cm} (1)

For this case, the heat transfer rate, \( Q \), is given by;

\[ Q = m \times (h_{co} - h_{ci}). \] \hspace{1cm} (2)

The temperature difference, \( DT \), is given by the log mean temperature difference, DTLM, for a counter flow heat exchanger;

\[ DTLM = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \left[ \frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})} \right]} \hspace{1cm} (3) \]

and the correction factor, \( F \), for a single pass, cross flow heat exchanger with one fluid mixed and the other unmixed,
was determined by using Figure 11.13 in Incropera, et al. (1985).

The inside heat transfer coefficient, $H_i$, was found from the internal flow correlation:

$$H_i = Nu \times \frac{k}{D_h}$$

For airflow through the inside annulus (Shah, 1988):

the Nusselt number, $Nu$, is given as:

$$Nu = \frac{(f/2) \times (Re - 1000) \times Pr}{1 + 12.7 \times (f/2)^{1/2} \times (Pr^{2/3} - 1)}$$

the Reynolds number, $Re$, is given as:

$$Re = \frac{4 \times m}{\pi \times (D_o + D_i) \times \mu}$$

and the friction factor, $f$, is given as:

$$f = (1.58 \times \ln(Re) - 3.28)^{-2}$$

The relation used to calculate the outside heat transfer coefficient, $H_o$, is given by:
\[ U \]
\[ H_0 = \frac{1}{1 - U \left( \frac{L}{k} + \frac{1}{H_1} \right)} \quad (8) \]

This equation was derived from the equation for total resistance, R_{tot}, which is given by;
\[ R_{tot} = \frac{1}{U \times A_0} = R_l + R_w + R_o \quad (9) \]

where, \[ R_l = \frac{1}{(H_i \times A_i)} \quad (10) \]
\[ R_w = \frac{\ln (r_o/r_i)}{(2 \times \pi \times k \times L)} \quad (11) \]
\[ R_o = \frac{1}{(H_0 \times A_0)}. \quad (12) \]

This assumes the outside surface area, A_0, equals the inside surface area, A_i, since the head is thin.

Other assumptions in these calculations were: 1) Steady state conditions. 2) No internal heat generation in the head. 3) One dimensional conduction. 3) Negligible heat loss to surroundings. 4) Negligible kinetic and potential energy changes. 5) Constant properties. 6) Fully developed conditions for the air (U is independent of position). 7) Smooth pipes.

5.2 STIRLING ENGINE - FLUIDIZED BED

The stirling engine - FBC can not be readily modeled as a cross flow heat exchanger as in section 5.1 since the internal fluid flow is reciprocating. Evaluation of the heat transfer into the engine, \( Q \), can be done, however, by writing an energy balance for the system.

\[ Q = E_{FIN} + E_{AIN} - E_{FOUT} - E_{AOUT} - E_{LOSS}. \quad (13) \]

For calculation of the heat transfer coefficients, the system is modeled as an annulus with a constant heat
Fig. 3. Schematic of stirling engine head model.

transfer rate at the outer surface (Fig. 3). The overall and outside heat transfer coefficients were found by rearranging the equation for the heat transfer rate at steady state;

\[ U_o = \frac{Q}{(A_o \times DT_{tot})} \]  \hspace{1cm} (14)

where, \( A_o \) = the outside surface area of the head

\( DT_{tot} \) = temperature difference between sand bed and inside working fluid \((T_s - T_a)\)

and,

\[ H_o = \frac{Q}{(A_o \times DTo)} \]  \hspace{1cm} (15)

where, \( DTo \) = temperature difference between sand bed and head \((T_s - Thead)\).

The calculation for the inside heat transfer coefficient was complicated by the presence of fins on the inside of the head. The equation used to calculate the inside
coefficient was obtained by equation (14) with equation (9) and with:

\[
R_i = \frac{1}{H_i \times (NF \times Af + Ap)} \tag{16}
\]

\[
R_w = \ln \left( \frac{r_o}{r_i} \right) \left/ \left( 2 \times \pi \times L \times k \right) \right. \tag{17}
\]

\[
R_o = \frac{1}{(Ho \times Ao)} \tag{18}
\]

Inserting this into equation (14) and rearranging:

\[
H_i = \frac{1}{\frac{D_{Ttot}}{\ln(r_o/r_i)} + \frac{1}{(NF\times Af + Ap)\times Q \left( \frac{1}{2\pi L k} + \frac{1}{Ho \times Ao} \right)}} \tag{19}
\]

An assumed fin efficiency, NF, was used to calculate Hi. Hi was then used to calculate a value for NF using the method from Shah, (1988). Iterations were done on the heat transfer calculation until the calculated value of NF converged to within 0.01 of the last value of NF.

The assumptions for these calculations were: 1) Steady state conditions. 2) One-dimensional radial conduction. 3) Constant properties. 4) Uniform volumetric heat generation in the sand bed which was equal to the total energy input. 5) The temperature of the sand bed and the head remained constant. 6) The conditions inside the engine were considered to be constant at their average values. 7) Uniform heat transfer coefficients. 8) Negligible changes in kinetic and potential energy. 9) No internal heat generation in the head.

Additional assumptions for the inside fin calculations were: 1) The gap was approximated as being flat with plain
triangular fins. 2) The fin conduction length is the full length of the fin because heat transfer occurs on one side, and the thermal resistance between the fin and the base is negligible. 3) The other end of the fin was treated as having a finite heat leak with the same heat transfer coefficient as the rest of the fin. 4) Radiation heat transfer for the fin was neglected.

The inside heat transfer coefficient could also be calculated using the equation for Q;

\[ Q = U \times A \times DT \]  

(20)

but the temperature difference between the inside of the head and the working fluid must be calculated first. Both methods can be reduced to the same set of equations.

Three separate efficiencies were calculated for the system. The definitions of each are as follows;

\[ \text{overall efficiency} = \frac{\text{power output}}{\text{fuel energy rate in}} \]  

(21)

\[ \text{transfer efficiency} = \frac{\text{energy rate into engine}}{\text{total energy rate in}} \]  

(22)

\[ \text{engine efficiency} = \frac{\text{power output}}{\text{energy rate into engine}} \]  

(23)
CHAPTER VI
PRELIMINARY SYSTEM CONFIGURATION AND TESTING

Construction of the mild steel SFBC began in January 1988. It was designed to have the head of the stirling engine protrude horizontally into the fluidized sand bed to take advantage of the high heat transfer rates and uniform temperatures of the fluidized sand bed.

6.1 HEAT EXCHANGER HEAD

Initial testing of the SFBC was done with a simulated finless head. A cylinder with approximately the same outside diameter as an actual engine head was constructed of 16 gage galvanized sheet metal. The end of the simulated head was flat which would offer more resistance to the fluidization of the sand than an actual domed head. The simulated head was inserted into the uninsulated SFBC for initial testing.

The clean cooling jacket air from Prototype II was used as fluidizing air in the SFBC to avoid purchasing and assembling more controls and air pumping equipment. This arrangement allowed for the use of the clean hot air from Prototype II for initial fluidization tests and to preheat the SFBC when it was used as a burner in later tests.
The first test was to determine whether the sand would fluidize around the simulated head. The top of the bed was removed and several different levels of sand were visually inspected for agitation with cold and then hot fluidizing air. The results showed that agitation of the sand increased with increasing fluidizing air temperature. Uniform agitation of the sand around the head was observed for sand depths varying from below to above the head. The best agitation occurred with the static sand level at the top of the head. Since the flat end of the simulated head caused more obstruction than an actual domed head, it was concluded that the SFBC design would allow adequate fluidization of the sand.

Next, the burner was insulated and an insert was made to fit inside the sheet metal head to simulate the air flow through the actual head (Fig. 4 and 5). The inlet and outlet temperatures and the flow rate of the air inside the heat exchanger head were measured along with the sand bed temperatures outside the head. This data was used to calculate the heat transfer coefficients between the sand and the head (outside, Ho) and between the head and the air flowing through the heat exchanger (inside, Hi) using the method outlined in Section 5.1. For the two airflow rates utilized, the limiting heat transfer coefficient was on the air side of the head. (See Table 1.)
Fig. 4. Actual stirling engine head (left) and simulated heat exchanger head (right).

Fig. 5. Sketch of simulated heat exchanger head.
Table 1. Summary of heat transfer coefficient calculations for heat exchanger head. Airflow rate is for inside exchanger. All values are experimental except for wall value.

<table>
<thead>
<tr>
<th>Airflow Rate (kg/hr)</th>
<th>Heat Transfer Coefficients (W / m² K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>outside</td>
</tr>
<tr>
<td>115</td>
<td>366</td>
</tr>
<tr>
<td>151</td>
<td>372</td>
</tr>
</tbody>
</table>

The temperature uniformity of the bed was also investigated with the heat exchanger head by placing thermocouples inside the heat exchanger at the top, bottom, left side, and right side to measure the temperature of the heated air at these places. The temperature was found to be nearly uniform around the head (within 15°C) except at the top which ran 50°C cooler. (See Fig. 6.)

6.2 STIRLING ENGINE

The ST-5 stirling engine was obtained from Stirling Technologies in June 1988. A 0.76 cubic meter concrete mounting slab for the engine was constructed and the SFBC insulation jacket was modified to accept the stirling engine.

The water for the cooling jacket on the stirling engine was taken straight from the OARDC well-water tap. An inline flow meter was placed in the inlet line and thermocouples were placed in the line at the inlet and outlet connections to the cooling jacket. The heated
outlet water was dumped to a drain as opposed to a closed cooling water system. This increased engine efficiency by allowing the inlet water to remain as cool as possible. The equation for the theoretical efficiency of the stirling cycle is given by \(1 - \frac{T_c}{T_h}\). Therefore, the greater the temperature difference between the two ends of the engine, the more efficient the engine will operate.

The engine was connected to a 5 kW generator via double drive belts and a jackshaft for a 5 to 1 speed increase. The generator was connected to a controllable electrical resistance heater unit for loading the engine. A wattmeter was placed between the generator and heater to record the electric power output. Since the ST-5 has no built-in speed control, the load on the engine was gradually increased from zero during the initial start-up.
as the engine gained power. The engine increased in power as the internal air pump slowly increased the pressure inside the engine to 3.5 bar and as the head temperature reached its maximum allowable temperature of 650°C. The temperature limit placed on the head is restricted by the limits of the material used in the construction of the head. The heater unit was adjusted to maintain the specified engine rpm as the power increased.

A special seal was designed and constructed to fit between the SFBC and the stirling head. This high temperature seal contained the combustion air and sand in the SFBC and absorbed the vibration of the stirling engine. A three piece design was used in which one ring was attached to the SFBC, one was attached to the stirling head, and a third ring, with its inside diameter slightly larger than the outside diameter of the stationary rings, was placed between the end rings. A high temperature fiberglass type rope was used for the sliding seal at both ends of the center ring. The assembly slides axially and also allows for a small amount of radial movement.

The first two tests of the fluidized bed - stirling engine system were performed using only the heated air from Prototype II for fluidizing and heating the sand bed. The engine was started by hand cranking the flywheel when the sand bed temperature reached 550°C. Using only the heated air from Prototype II, no measurable output was recorded
because the engine did not develop enough power to operate the generator.

These preliminary tests proved the system would work and the observations made from these tests were valuable in deciding what changes were needed to improve the operation of the system. Thermocouples were added to measure the actual stirling engine head temperature so the fuel could be adjusted to keep the head at a constant temperature of 650°C.

The generator proved to be too large for loading and measuring the output of the engine, thus a prony brake dynamometer was designed and constructed to measure engine output directly at the flywheel. This was an improvement over the generator system since belt slippage and generator losses were eliminated. The generator was left in place as a backup system to avoid engine overspeeding if the prony brake assembly failed. The belts are disengaged from the flywheel during normal operation but could be quickly tightened to engage the generator.

Since the heated air from Prototype II was inadequate to run the engine at rated power, a fuel inlet was added to the air inlet pipe to provide additional heating. In this configuration, the sand bed is preheated using the hot clean air from Prototype II. Once the sand bed reaches 550°C the fuel at the SFBC inlet is turned on so burning takes place in the bed around the engine head.
Simultaneously, the fuel is shut off to Prototype II so the inlet air to the stirling bed is no longer being heated. Now the SFBC operates on its own using only the air pump and controls from Prototype II.

A few short tests were run with this set up, shown in Fig. 7 and 8. The engine was started when the stirling head temperature reached 500°C. It quickly reached the rated speed of 720 rpm. The load on the engine was gradually increased to maintain the rated rpm as the engine increased in power. The fuel (powdered corncobs at 5% moisture) was adjusted to maintain the head temperature at 650°C.

The temperatures in the system during this test are shown in Fig. 9 and 10. Several points can be made about the temperature profile.

1) The SFBC air inlet temperature is decreasing throughout the test. The temperature of the inlet air coming through the cooling jacket of Prototype II decreases because burning in Prototype II was stopped after the SFBC was preheated enough to start burning on its own.

2) The above head and exhaust temperatures were quite high. As a result, much heat energy is lost in the exhaust. The amount of fluidizing air may need to be decreased to help reduce this loss.
Fig. 7. Current connection of Prototype II to SFBC. Stirling engine is shown removed from SFBC.

Fig. 8. Sketch of current system shown in Fig. 7.
Fig. 9. SFBC temperature profile.

Fig. 10. Stirling engine cooling jacket temperatures and mechanical power output.
3) The temperature of the stirling head was maintained at a fairly constant temperature of 650°C. This is due to the precise control over the combustion process that the SFBC system offers.

4) The power of the stirling engine gradually increased throughout the test indicating steady state had not been reached. The fluctuations in the power were due to difficulties in reading the scale on the dynamometer which fluctuated due to engine flywheel imbalance. The average maximum power at the end of the test was 2.6 kW, which is less than the rated power of a production ST-5 engine (3.7 kW) but greater than the manufacture's specifications for our derated prototype engine. The overall efficiency was 5%.

5) The energy lost in the cooling water remained relatively constant throughout the test.
CHAPTER VII
INITIAL TESTING

Instrumentation was added to aid in determining the power output of the engine, the air temperature inside the engine, and SFBC insulation losses. Fortran 77 programs were used to read the data from the datalogger and calculate averages over the length of the test. These values were written to external files so they could be read into a Basic program for numerical calculations.

The dynamometer was upgraded by replacing the manually read spring scale with a 0-50 pound load cell and the manually read tachometer with an engine flywheel magnetic pickup tachometer. Both outputs were recorded on the datalogger. These two additions were needed to improve the accuracy of the flywheel power output calculations.

Because the temperature of the working fluid inside the engine was unknown in the preliminary testing, thermocouples were installed in the stirling engine to measure the temperature of the working fluid inside the engine head (Fig. 11). In a stirling engine, the working fluid is shuffled back and forth inside the engine by the displacer. The temperature changes according to the
Cross sectional view of engine head

temperature measurement points

regenerator

base

top

cut

in

airflow direction

Fig. 11. Location of temperature measurement inside stirling engine head.

stopped
maximum
stopped

maximum
average
minimum

maximum
average
minimum

TIME ——>

Fig. 12. Airflow rate and temperature inside the stirling engine head as a function of time.
direction and velocity of the working fluid (Fig. 12). Since the engine operates at 720 rpm, the instantaneous temperature cycles too fast for measurement with a thermocouple, therefore, the base and top temperatures recorded were mean temperatures. The average of these two temperatures was used as the inside air temperature ($T_a$ in Fig. 3).

The SFBC insulation jacket was experimentally calibrated for heat energy losses. From equation (20), it is seen that $Q$ depends only on the temperature difference, $DT$ for a constant $U$ and $A$. The jacket was calibrated at three different airflow rates by calculating the energy lost ($EOSS$), measuring the temperature difference across the insulation, and by assuming that $U$ and $A$ for the insulation remain constant.

The SFBC was operated at normal temperatures for engine operation but the engine was not started. The cooling water flow rate and inlet and outlet temperatures were measured to calculate the energy lost through the engine by conduction ($EW$). The inlet and outlet energy in the fuel ($EFIN$, $EFOUT$) and air ($EAIN$, $EAOUT$) were calculated using the method developed by Dr. Keener for the FBC system. This was programmed in the Fortran programs "FBCANZ" and "ORFICE". (See Lin, (1988) and Spink, (1972) for description of equations and assumptions). The energy lost was found from the equation,
ELOSS = EFIN + EAIN - EFOUT - EAOUT - EW. \hspace{1cm} (24)

The UA constant for each test was found by dividing the energy lost by the temperature difference across the insulation. A Basic program "INSUL" was written to perform the calculations (Appendix A). The UA values were plotted versus airflow rates and a best fit regression line \( R^2 = .999 \) was obtained to give UA as a function of airflow. The equation is in program "STC" in Appendix B.

Seven initial tests of the Stirling engine were made to determine the heat transfer coefficients and efficiencies described in Section 5.2. A Basic computer program "STC" was written to perform the calculations (Appendix B). The energy in the fuel and air were found as in the insulation tests. The energy lost through the insulation was calculated using the measured temperature difference across the insulation and the results from the insulation loss tests to find UA.

The results of these initial tests are presented in Table 3 and also in Figures 16-24 along with the results from the final set of tests so the two can be compared and discussed.
CHAPTER VIII

ANALYSIS TO IMPROVE OVERALL EFFICIENCY

Using the data obtained from previous tests, ways to improve system efficiency were investigated. From equation (21), it is noted that to increase the overall efficiency, either the power output must be increased or the fuel energy input must be decreased.

8.1 POWER OUTPUT

In order to increase the power output of the engine, without changing the engine itself, the heat transfer rate into or out of the engine must be increased or the fluid friction losses must be decreased. There are three ways to increase the heat transfer rates. From equation (20), it is seen that either U, A, or DT must be increased to increase Q. For our testing the heat transfer rate out of the engine is limited by the cooling water inlet temperature. This is assumed to be fixed since the temperature of the water could not be lowered enough to make a large difference and it would take energy to cool the water. Changing the water to a fluid with a higher specific heat would not be economical because of the large amounts needed during testing.
From equations (9), (16), (17), and (18) it is noted to increase the overall heat transfer coefficient, $U$, into the engine, $H_i$ or $H_o$ must be increased or $R_w$ must be decreased. Since $R_w$ is much less than $R_i$ and $R_o$, changes in $R_w$ will not have a significant effect on $U$. The inside heat transfer coefficient, $H_i$, is the limiting factor in $U$, therefore it will have the greatest effect on $U$.

On the inside of the engine, the properties of the working fluid are important in two ways, heat transfer and friction losses. Changing the internal working fluid of the engine from air ($c_p = 1.051 \text{ kJ/kg K at 600K, 1 atm}$) to a fluid with a higher specific heat (such as helium, $c_p = 5.193 \text{ kJ/kg K}$ or hydrogen, $c_p = 14.55 \text{ kJ/kg K at 600K, 1 atm}$) will increase $H_i$. Increasing the engine internal pressure will also increase the specific heat of the working fluid. The density of the working fluid is important in relation to the flow friction losses. These losses are directly proportional to $1/2 \times \rho \times R \times V^2$, where $\rho$ is the density and $V$ is the gas velocity. This equation also points to the use of hydrogen ($\rho = 0.04 \text{ kg/m}^3 \text{ at 600K}$) or helium ($\rho = 0.098 \text{ kg/m}^3 \text{ at 600K}$) as the working fluid instead of air ($\rho = 0.58 \text{ kg/m}^3 \text{ at 600K}$) to reduce the friction losses inside the engine. (Michels, 1976.)

In the ST-5, sealing becomes a problem when dealing with the less dense fluids and the internal pressure is limited to 3.5 bar. Also, since it operates at such a low
rpm the fluid friction loss savings of a less dense fluid would not be great.

On the outside of the engine, the flow rate of the sand/air mixture may affect the heat transfer coefficient but the limiting factor is on the inside. Small changes in the outside heat transfer coefficient will not make a large difference in the overall heat transfer coefficient.

In order to increase the temperature differential across the head, the temperature of the sand must be increased or the inside air temperature must be decreased. Since the objective is to get the inside air as hot as possible the only alternative is to increase the sand bed temperature. This temperature is limited by the maximum allowable head temperature (650°C) and by the ash fusion temperature of corn cobs (800°C) which was used as the fuel. Therefore, increasing the DT across the head is not an option in this case given the constraints placed on maximum temperatures in the system.

To increase the surface area, the addition of fins on the outside of the engine head was investigated theoretically. Closely spaced straight fins that run from the base to the top of the head are used on normal production ST-5 engines because it is designed for air as the working fluid on the outside of the head. These fins are in the wrong direction for flow in the SFBC and would plug with sand. This investigation looked at the design
Fig. 13. Production ST-5 head with longitudinal fins and proposed head with circular fins.

of circular fins spaced on the head which would not plug nor impede air/sand flow around the head (Fig. 13). The equations used in this analysis were obtained from Figure 3.19 and Example 3.8 in Incropera, et al. (1985). A Basic computer program was written to perform the calculations and find the five best combinations of number of fins, fin height, and fin thickness for actual test data and head geometry. Fin thickness ranged from 1mm to 10mm in 1mm increments. Fin height ranged from 10mm to 50mm in 5mm increments. The number of fins ranged from 1 to 50 and the minimum fin spacing was set before the program was run.

The assumptions made in the calculations for the circular fins were: 1) Steady state conditions. 2) One-
dimensional radial conduction in the fins. 3) Constant properties. 4) No internal heat generation. 5) Negligible radiation exchange with surroundings. 6) Uniform convection coefficient over the outer surface (with or without fins).

Calculations were made using the results from four engine tests (Appendix C) with the increase in transfer efficiency versus outside heat transfer coefficient plotted in Fig. 14. From this plot it can be seen that an increase in the heat transfer coefficient results in a lower percent increase in the transfer efficiency. This can be explained by looking at equation (20) for the heat transfer rate.

---

Fig. 14. Increase in transfer efficiency vs. outside heat transfer coefficient.
As $H$ increases, changes in $A$ have a reduced effect on $Q$ and therefore a reduced effect on the transfer efficiency.

The following designs, presented in Table 2 from Basic program "SFIN", were chosen on the basis of 330 W/m$^2$K being the more representative value of the outside heat transfer coefficient.

Table 2. Theoretical external fin designs for optimum heat transfer into the Stirling engine head.

<table>
<thead>
<tr>
<th># FINs</th>
<th>SPACING</th>
<th>HEIGHT</th>
<th>THICKNESS</th>
<th>TRANSFER</th>
<th>OVERALL % increase</th>
</tr>
</thead>
<tbody>
<tr>
<td>12</td>
<td>10</td>
<td>45</td>
<td>7</td>
<td>29</td>
<td>29.5</td>
</tr>
<tr>
<td>7</td>
<td>20</td>
<td>45</td>
<td>8</td>
<td>20</td>
<td>20.8</td>
</tr>
</tbody>
</table>

The overall efficiency was obtained by assuming the engine efficiency and total energy rate into the SFBC will remain constant. Testing needs to be done to find the minimum fin spacing before plugging with sand occurs.

8.2 FUEL ENERGY REQUIREMENTS

As seen in Table 2, the overall efficiency increases with the addition of fins to the outside of the head. This is because the power output of the engine increased for a given fuel input. The fins could also be used to reduce the fuel input requirements for a given power output. The energy into the engine could be maintained at the same level as without fins but with a lower bed temperature,
resulting in the same power output but with reduced fuel energy input. This should result in the same increased overall efficiency as in the case of increased power output in section 8.1.

Preheating the combustion air with exhaust heat energy is a commonly used method to improve the overall efficiency of a burner system. By adding heat energy to the incoming combustion air, less fuel energy is required to maintain a constant total energy input. This can be accomplished by installing a heat exchanger in the exhaust of the SFBC to preheat the combustion air. The selection of the construction type, flow arrangement, and surface geometry of the heat exchanger was dependent on the fluids used on each side of the exchanger, operating pressures, temperatures, corrosiveness of the fluids, fouling, leakage, current system configuration (if any), required effectiveness, ducting requirements, allowable thermal stresses, energy losses, available technology, maintenance requirements, reliability, cost, safety, and other criteria.

A counter flow tube-in-tube heat exchanger (Fig 15) was chosen for this application for several reasons: 1) Existing straight 3.35 m span of exposed exhaust piping lends itself well to a tubular heat exchanger. 2) Ease and inexpense of manufacture. 3) Thermal expansion could be handled with a moving seal at one end of the exchanger.
Fig. 15. Counter flow tube-in-tube heat exchanger.

4) A counter flow arrangement with the outside fluid being the preheated air maximizes heat transfer and minimizes heat losses. 5) The allowable pressure drop is low and surface area must be high with air as the working fluid on both sides.

The methods and equations used in the thermal and hydraulic design of the exchanger were obtained from Shah, (1988). These equations were programmed in Basic to calculate the exchanger length required to meet energy and tube diameter requirements. (Appendix D.)

The assumptions made in the calculations were as follows: 1) Negligible heat transfer between exchanger and surroundings. 2) Negligible kinetic and potential energy changes. 3) Tube and annulus flows and thermal conditions are fully developed. 4) Negligible fouling effects. 5) Smooth pipes. 6) Constant properties evaluated at the mean temperature on each side. 7) The viscosity and thermal
conductivity of the exhaust gas are the same as air at the same temperature.

Requirements set for the design to meet were: 1) Heat transfer rate to be at least 7.3 kW. 2) Maximum allowable pressure drop for the combustion air side to be 3.5 kN/m².

The final design chosen was a 3.35 m long counter flow tube-in-tube heat exchanger. The inner tube had a 50.8 mm inside diameter and 60.3 mm outside diameter. The gap between the inner and outer tubes was 9.5 mm. This design was chosen for several reasons. 1) The preset requirements were met by this exchanger. 2) The pressure drop was lowest for this design, allowing for a margin of error in the pressure drop calculations. 3) The arrangement would be economical and easy to manufacture. Because this simple design would meet the requirements, fins were not used.

Another way to decrease the fuel requirements would be to increase the insulation around the system to decrease the losses from the system. Also the design of the burner could be changed to make it more fuel efficient. From the temperature profiles of actual tests, it was noted that the temperature is greater above the head than below the head for all airflow rates. This indicates the need for a deeper sand bed below the head so the fuel will completely burn before it leaves the bed. The annular gap of sand between the engine head and the wall of the SFBC might also
be investigated to find if there is an optimum width of the gap for heat transfer into the head. The gap was 50 mm in the SFBC tested.
CHAPTER IX

FINAL SYSTEM CONFIGURATION AND TESTING

Several designs to improve system efficiency were considered for construction and testing. The heat exchanger in the exhaust to preheat the incoming air was the only system that was constructed because of time and safety considerations. The flow rate of the combustion air was varied to analyze its effects on the heat transfer into the engine.

The heat exchanger design presented in Section 8.2 was constructed in early August, 1989. The tubing material was mild steel with inner tube diameters of 51 mm inside diameter and 60.4 mm outside diameter and an annulus tube diameter of 79.4 mm inside diameter. The exchanger was 3.35 m long, with the hot end fixed, and a bellows at the cold end to allow for thermal expansion of the tubes.

The pressure drop through the exchanger annulus was measured before the outlet was connected to the SFBC. The experimental value for the pressure drop test was 1.49 kN/m\(^2\) which was below the design pressure drop of 3.5 kN/m\(^2\). This value included the piping between the pump and the inlet to the exchanger. The calculated pressure drop
through the annulus was 1.21 kN/m² and additional piping losses were calculated to add 0.43 kN/m² for a total of 1.64 kN/m² calculated pressure drop at the measured airflow rate of 130 kg/hr. The calculated pressure drop was 0.15 kN/m² (9.5% error) higher than the actual pressure drop.

From the five tests run on the heat exchanger, 26.3% of the exhaust heat energy was recovered by the heat exchanger, increasing the overall efficiency for these tests by 13.2%. The energy recovered was less than the required 7.3 kW in most cases. This was due to the fact that the exhaust temperature at the inlet of the heat exchanger was less than the outlet temperature of the SFBC and the flow rates were different than the values used in the design of the exchanger.

The heat transfer coefficients were well matched and averaged 60.2 W/m² K on both sides of the exchanger. There was a slight increase in the heat transfer coefficients as the flow rate increased. This was expected because of the increased Reynolds number (turbulence) caused by the higher airflows.

There was a total of 12 tests conducted on the stirling engine. Seven tests were run before the addition of the heat exchanger while five tests were run after the heat exchanger was installed. For each test the engine head was limited to 650°C, engine rpm was limited to 720 rpm, and the internal engine pressure was set at 3.5 bar.
gage. The combustion airflow rate was the only variable in each test. The results of these tests are shown in Figures 16-21 and Table 3. The average inside, Hi, and outside, Ho, heat transfer coefficients are 76 W/m²K and 350 W/m²K, respectively. The average overall heat transfer coefficient, U, is 150 W/m²K and is based on the outside head surface area. The standard deviations of Hi, Ho, and U are 7.7, 17, and 6.5 W/m²K. These are within the limits of experimental error which are presented in Appendix E. It is noted that in tests 6, 7, and 8 the engine efficiency (Fig. 16) was low and even though all other conditions remained constant. Examination of the engine (both internal and external) after test 8 revealed a broken spring inside the engine and a broken engine mounting bolt. Several inconsistencies are noted in these three tests.

The power output (Fig. 17) did not increase with increasing airflow as expected even though the outside heat transfer coefficient shows a small increase (Fig. 18). This was because the temperature of the sand bed decreased as the airflow rate increased. Therefore, the temperature differential across the head decreased (Fig. 19), offsetting the slight increase in Ho.

Another major reason for the constant power output was the low heat transfer coefficient on the inside of the head. The system was so limited on the inside of the head that it could not take advantage of the potential for an
increased heat transfer rate. This was clearly noted in the preliminary testing (Table 1) where all the coefficients are based on the same surface area.

The average heat transfer rate into the engine was 17.5 kW. This is lower than the 20.9 kW the manufacturer claims for their burner system (Appendix B). This can be explained by noting that the internal pressure in our engine was limited to 3.5 bar while a production engine is set at 5 bar. This again points to the limitation to heat transfer on the inside of the engine.

The two major differences in the test results before and after the installation of the heat exchanger were the input fuel energy requirements (Fig. 20) and the insulation losses (Fig. 21). As expected, the fuel requirements were lower for the tests run with the heat exchanger as discussed above, but the insulation losses also decreased with the addition of the heat exchanger to the system.

The insulation jacket was recalibrated after the installation of the heat exchanger to see if the calibration had changed. The addition of hot air at the burner inlet changed the temperature profile of the SFBC and the burning characteristics of the fuel. Also, since less fuel was required with the heat exchanger, less burning was taking place inside the SFBC. The combination of these factors resulted in a decrease in insulation losses for a given airflow rate.
Fig. 16. Engine efficiency vs. power output.

Fig. 17. Power output vs. airflow rate.
Fig. 18. Outside heat transfer coefficient vs. airflow rate.

Fig. 19. Temperature change across head vs. airflow rate.
Fig. 20. Fuel energy in vs. airflow rate.

Fig. 21. Insulation energy losses vs. airflow rate.
Table 3. Results of stirling engine tests.

<table>
<thead>
<tr>
<th>TEST #</th>
<th>EFFICIENCY (%)</th>
<th>HEAT TRANSFER COEFF. (W / m² K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>TRANSFER ENGINE</td>
<td>OVERALL</td>
</tr>
<tr>
<td>Initial tests without heat exchanger</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>33.6</td>
<td>8.2</td>
</tr>
<tr>
<td>2</td>
<td>40.4</td>
<td>4.5</td>
</tr>
<tr>
<td>3</td>
<td>43.6</td>
<td>9.6</td>
</tr>
<tr>
<td>4</td>
<td>42.7</td>
<td>10.2</td>
</tr>
<tr>
<td>5</td>
<td>35.1</td>
<td>11.5</td>
</tr>
<tr>
<td>6</td>
<td>39.8</td>
<td>6.9</td>
</tr>
<tr>
<td>7</td>
<td>39.4</td>
<td>6.7</td>
</tr>
<tr>
<td>Final tests with heat exchanger</td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>34.3</td>
<td>7.3</td>
</tr>
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<td>9.6</td>
</tr>
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<td>12.0</td>
</tr>
<tr>
<td>12</td>
<td>31.6</td>
<td>11.5</td>
</tr>
</tbody>
</table>

Average heat transfer coefficients: 350 76 150
Standard deviation (±) 17 7.7 6.5

Note: Averages do not include tests 6-8.
CHAPTER X

CONCLUSIONS

The results of the tests lead to several conclusions.

1) The FBC transferred enough heat energy into the finless stirling engine head to run the engine at power output levels above the manufacturer's specifications for the engine.

2) The heat transfer coefficients from the stirling engine tests compared well with the heat transfer coefficients obtained from the heat exchanger head tests. These results show that high heat transfer coefficients can be obtained on the outside of a large cylindrical object immersed in a fluidized bed of sand particles.

3) The system was limited by the heat transfer coefficient on the inside of the engine head. Changing the engine working fluid to helium and increasing the engine internal pressure are two recommendations for improving the inside heat transfer coefficient.

4) The heat exchanger in the exhaust was a successful way to economically improve the overall efficiency.

5) The addition of heat energy to the incoming
combustion air also changed the SFBC temperature profile and the combustion process resulting in lower insulation energy losses.

6) Since the engine tested was a reconditioned engine, it was not very efficient or reliable. The operation of the engine required continuous monitoring because the power output would fluctuate even at steady state burner conditions. If this particular system was to be further investigated, a production engine should be tested.

In summary, the FBC - Stirling engine system developed in this research has proved the concept that the systems could operate together successfully. The FBC kept the head temperature uniform and had a high heat transfer coefficient to the engine, both of which were reasons for combining these systems. This research should be continued with other engines (Stirling or not) that have an inside heat transfer coefficient greater than or equal to the heat transfer coefficient of the fluidized sand bed.
APPENDIX A

INSULATION LOSS CALCULATIONS
The results of the Basic program "INSUL" for three tests at different airflow rates are shown (Tables 4, 5, 6) with an insulation temperature profile (Fig. 22). The input data was read from an external file that was created by a Fortran program that averaged the experimental data over the length of the test. This data is shown at the top of the sample output in numbers 1) - 17). The program output is shown below the input data for each test. The program is similar to the program "STC" which is presented in Appendix B.

Fig. 22. Insulation loss test temperature profile.
Table 4. Results of an insulation test at the lowest airflow rate tested.

<table>
<thead>
<tr>
<th>PROGRAM INSULATION TEST CALCS</th>
<th>TEST FILE NO : 11</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>BED:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (°C) of inlet combustion air</td>
<td>(TIN) 1): 76.2</td>
</tr>
<tr>
<td>Temp (°C) of exhaust</td>
<td>(TEX) 2): 687</td>
</tr>
<tr>
<td>Temp (°C) change across insulation</td>
<td>(DTL) 3): 621</td>
</tr>
<tr>
<td>Temp (°C) change across head</td>
<td>(DTAV) 4): 123</td>
</tr>
<tr>
<td>Temp (°C) change outside head</td>
<td>(DTO) 5): 24.6</td>
</tr>
<tr>
<td>ORIFICE:</td>
<td></td>
</tr>
<tr>
<td>Temp (°C) of air at orifice</td>
<td>(TAR) 6): 39.4</td>
</tr>
<tr>
<td>Orifice static pressure (mmHg)</td>
<td>(SP) 7): 70</td>
</tr>
<tr>
<td>Orifice differential pressure (mmH20)</td>
<td>(DP) 8): 52.1</td>
</tr>
<tr>
<td>AMBIENT:</td>
<td></td>
</tr>
<tr>
<td>Temp (°C) of ambient air</td>
<td>(TAMB) 9): 36</td>
</tr>
<tr>
<td>Dew point temperature (°C)</td>
<td>(TDPT) 10): 20.8</td>
</tr>
<tr>
<td>Atmospheric pressure (mm Hg)</td>
<td>(PATM) 11): 740.5</td>
</tr>
<tr>
<td>FUEL:</td>
<td></td>
</tr>
<tr>
<td>Flow rate of corn cobs (lb/hr)</td>
<td>(FR) 12): 10.8</td>
</tr>
<tr>
<td>Moisture content of corn cobs (%wb)</td>
<td>(MC) 13): 9.8</td>
</tr>
<tr>
<td>ENGINE:</td>
<td></td>
</tr>
<tr>
<td>Flow rate of cooling water (gpm)</td>
<td>(CW) 14): 4</td>
</tr>
<tr>
<td>Temp (°C) change of cooling water</td>
<td>(DTCW) 15): 1.42</td>
</tr>
<tr>
<td>Temp (°C) of head</td>
<td>(THO) 16): 621</td>
</tr>
<tr>
<td>Power output of stirling (hp)</td>
<td>(P) 17): 0</td>
</tr>
</tbody>
</table>

**PROGRAM INSULATION TEST CALCS** **TEST FILE NO : 11**

| FLOW RATES | FUEL | 10.8 | lb/hr |
| AIR | 154 | lb/hr |

**ENERGY CALCULATIONS**

| IN | FUEL | 72132 | BTU/hr | 21139 | V |
| AIR | 2720 | 797 |

| OUT | FUEL | 92 | 24 |
| AIR | 51944 | 15223 |
| WATER | 5102 | 1495 |
| LOST | 17723 | 5194 |

**UA CONSTANT** : 8.36 W/C
Table 5. Results of an insulation test at the middle airflow rate tested.

<table>
<thead>
<tr>
<th>PROGRAM INSULATION TEST CALCS</th>
<th>TEST FILE NO : I2</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>BED:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of inlet combustion air</td>
<td>(TIN) 1: 71.6</td>
</tr>
<tr>
<td>Temp (C) of exhaust</td>
<td>(TEX) 2: 681</td>
</tr>
<tr>
<td>Temp (C) change across insulation</td>
<td>(DTL) 3: 593</td>
</tr>
<tr>
<td>Temp (C) change across head</td>
<td>(DTAV) 4: 98.6</td>
</tr>
<tr>
<td>Temp (C) change outside head</td>
<td>(DTO) 5: 24.8</td>
</tr>
<tr>
<td><strong>ORFICE:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of air at orifice</td>
<td>(TAIR) 6: 44.2</td>
</tr>
<tr>
<td>Orifice static pressure (mmHg)</td>
<td>(SP) 7: 62.1</td>
</tr>
<tr>
<td>Orifice differential pressure (mmH20) (DP)</td>
<td>8: 74.6</td>
</tr>
<tr>
<td><strong>AMBIENT:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of ambient air</td>
<td>(TAMB) 9: 32.1</td>
</tr>
<tr>
<td>Dew point temperature (C)</td>
<td>(TDPT) 10: 17.2</td>
</tr>
<tr>
<td>Atmospheric pressure (mm Hg)</td>
<td>(PATM) 11: 739.5</td>
</tr>
<tr>
<td><strong>FUEL:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of corn cobs (lb/hr)</td>
<td>(FR) 12: 12.8</td>
</tr>
<tr>
<td>Moisture content of corn cobs (%wb)</td>
<td>(MC) 13: 9.8</td>
</tr>
<tr>
<td><strong>ENGINE:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of cooling water (gpm)</td>
<td>(CW) 14: 1.5</td>
</tr>
<tr>
<td>Temp (C) change of cooling water</td>
<td>(DTCW) 15: 4.07</td>
</tr>
<tr>
<td>Temp (C) of head</td>
<td>(THO) 16: 602</td>
</tr>
<tr>
<td>Power output of stiring (hp)</td>
<td>(P) 17: 0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>PROGRAM INSULATION TEST CALCS</th>
<th>TEST FILE NO : I2</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>FLOW RATES</strong></td>
<td></td>
</tr>
<tr>
<td>FUEL</td>
<td>12.8 lb/hr</td>
</tr>
<tr>
<td>AIR</td>
<td>182 lb/hr</td>
</tr>
<tr>
<td><strong>ENERGY CALCULATIONS</strong></td>
<td></td>
</tr>
<tr>
<td>IN FUEL BTU/hr</td>
<td>V</td>
</tr>
<tr>
<td>AIR</td>
<td>85490</td>
</tr>
<tr>
<td>OUT FUEL BTU/hr</td>
<td>V</td>
</tr>
<tr>
<td>AIR</td>
<td>3148</td>
</tr>
<tr>
<td>WATER</td>
<td>5484</td>
</tr>
<tr>
<td>LOST</td>
<td>22111</td>
</tr>
<tr>
<td>UA CONSTANT</td>
<td>10.93 W/C</td>
</tr>
</tbody>
</table>
Table 6. Results of an insulation test at the highest airflow rate tested.

<table>
<thead>
<tr>
<th>PROGRAM INSULATION TEST CALCS</th>
<th>TEST FILE NO: 13</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>BED:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of inlet combustion air (TIN) 1:</td>
<td>120</td>
</tr>
<tr>
<td>Temp (C) of exhaust (TEX) 2:</td>
<td>716</td>
</tr>
<tr>
<td>Temp (C) change across insulation (DTL) 3:</td>
<td>655</td>
</tr>
<tr>
<td>Temp (C) change across head (DTAV) 4:</td>
<td>101</td>
</tr>
<tr>
<td>Temp (C) change outside head (DTO) 5:</td>
<td>12.8</td>
</tr>
<tr>
<td><strong>ORFICE:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of air at orifice (TAIR) 6:</td>
<td>51.9</td>
</tr>
<tr>
<td>Orifice static pressure (mmHg) (SP) 7:</td>
<td>80.5</td>
</tr>
<tr>
<td>Orifice differential pressure (mmH2O) (DP) 8:</td>
<td>119</td>
</tr>
<tr>
<td><strong>AMBIENT:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of ambient air (TAMB) 9:</td>
<td>36.9</td>
</tr>
<tr>
<td>Dew point temperature (C) (TDPT) 10:</td>
<td>16.8</td>
</tr>
<tr>
<td>Atmospheric pressure (mm Hg) (PATM) 11:</td>
<td>740</td>
</tr>
<tr>
<td><strong>FUEL:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of corn cobs (lb/hr) (FR) 12:</td>
<td>16.3</td>
</tr>
<tr>
<td>Moisture content of corn cobs (%wb) (MC) 13:</td>
<td>9.8</td>
</tr>
<tr>
<td><strong>ENGINE:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of cooling water (gpm) (CW) 14:</td>
<td>1.5</td>
</tr>
<tr>
<td>Temp (C) change of cooling water (DTCW) 15:</td>
<td>5.71</td>
</tr>
<tr>
<td>Temp (C) of head (THO) 16:</td>
<td>692</td>
</tr>
<tr>
<td>Power output of stirling (hp) (P) 17:</td>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>PROGRAM INSULATION TEST CALCS</th>
<th>TEST FILE NO: 13</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>FLOW RATES</strong></td>
<td></td>
</tr>
<tr>
<td>FUEL : 16.3 lb/hr</td>
<td></td>
</tr>
<tr>
<td>AIR : 230 lb/hr</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ENERGY CALCULATIONS</th>
<th>BTU/hr</th>
<th>W</th>
</tr>
</thead>
<tbody>
<tr>
<td>IN FUEL : 108866</td>
<td>31904</td>
<td></td>
</tr>
<tr>
<td>AIR : 8349</td>
<td>2447</td>
<td></td>
</tr>
<tr>
<td>OUT FUEL : 129</td>
<td>38</td>
<td></td>
</tr>
<tr>
<td>AIR : 80892</td>
<td>23706</td>
<td></td>
</tr>
<tr>
<td>WATER : 7693</td>
<td>2255</td>
<td></td>
</tr>
<tr>
<td>LOST : 28500</td>
<td>8352</td>
<td></td>
</tr>
</tbody>
</table>

| UA CONSTANT | 12.75 W/C |
APPENDIX B

STIRLING ENGINE TEST CALCULATIONS
The Basic program "STC" is presented so the equations and constants used in the calculations on the stirling engine tests can be seen. The results of the program for a test with the heat exchanger in the inlet air is shown (Table 7) with the temperature profiles and power output for the run (Fig. 23, 24, and 25). The input data is read from an external file that was created by a Fortran program that averaged the experimental data over the length of the test. This data is shown at the top of the sample output in numbers 1) - 17). The program output is shown below the input data in the table. The results for all stirling engine tests are shown in Table 3.

![Temperature Profile](image)

**Fig. 23.** SFBC temperature profile with heat exchanger in the inlet air.
Fig. 24. Stirling engine inside air and head temperature.

Fig. 25. Stirling engine cooling water temperatures and mechanical power output.
Table 7. Results of a stirling engine test with the heat exchanger in the inlet air.

<table>
<thead>
<tr>
<th>PROGRAM STIRLING TEST CALCS</th>
<th>TEST FILE NO - S818</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>BED:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of inlet combustion air</td>
<td>(TIN) 1): 259</td>
</tr>
<tr>
<td>Temp (C) of exhaust</td>
<td>(TEX) 2): 745</td>
</tr>
<tr>
<td>Temp (C) change across insulation</td>
<td>(DTL) 3): 667</td>
</tr>
<tr>
<td>Temp (C) change across head</td>
<td>(DTAV) 4): 341</td>
</tr>
<tr>
<td>Temp (C) change cuteside head</td>
<td>(DTO) 5): 145</td>
</tr>
<tr>
<td><strong>ORIFICE:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of air at orifice</td>
<td>(TAIR) 6): 42.8</td>
</tr>
<tr>
<td>Orifice static pressure (mmHg)</td>
<td>(SP) 7): 59.2</td>
</tr>
<tr>
<td>Orifice differential pressure (mmH2O)</td>
<td>(DP) 8): 87.9</td>
</tr>
<tr>
<td><strong>AMBIENT:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of ambient air</td>
<td>(TAMB) 9): 30.2</td>
</tr>
<tr>
<td>Dew point temperature (C)</td>
<td>(TDPT) 10): 15.2</td>
</tr>
<tr>
<td>Atmospheric pressure (mm Hg)</td>
<td>(PATM) 11): 744</td>
</tr>
<tr>
<td><strong>FUEL:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of corn cobs (lb/hr)</td>
<td>(FR) 12): 20.9</td>
</tr>
<tr>
<td>Moisture content of corn cobs (%wb)</td>
<td>(MC) 13): 9.8</td>
</tr>
<tr>
<td><strong>ENGINE:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of cooling water (gpm)</td>
<td>(GW) 14): 4</td>
</tr>
<tr>
<td>Temp (C) change of cooling water</td>
<td>(DTCW) 15): 13.4</td>
</tr>
<tr>
<td>Temp (C) of head</td>
<td>(THO) 16): 652</td>
</tr>
<tr>
<td>Power output of stirling (hp)</td>
<td>(P) 17): 2.45</td>
</tr>
</tbody>
</table>

**PROGRAM STIRLING TEST CALCS**  **TEST FILE NO : S818**

| FLOW RATES | FUEL | 20.9 | lb/hr |
| AIR | 199 | lb/hr |

**ENERGY CALCULATIONS**

| IN | FUEL | 139569 | BTU/hr | 40907 | W |
| AIR | 19627 | 5811 |

| OUT | FUEL | 175 | BTU/hr | 51 |
| AIR | 77276 | 22646 |
| LOST | 25841 | 7573 |
| ENGINE | 56124 | 16448 |
| WATER | 48144 | 14109 |

**HEAT TRANSFER COEFS.**

<table>
<thead>
<tr>
<th>BTU/(hr ft² F)</th>
<th>W/(m² K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>OVERALL</td>
<td>24.2</td>
</tr>
<tr>
<td>OUTSIDE</td>
<td>56.9</td>
</tr>
<tr>
<td>INSIDE</td>
<td>12.3</td>
</tr>
</tbody>
</table>

**EFFICIENCIES**

| OVERALL | 4.5% |
| TRANSFER | 35.2% |
| ENGINE | 11.1% |
CL5
CLS
151 DIM X(20),Y(20),C(20),A(5,7)
152 KEY OFF
154 NUM = 17 : TK = 459.9 : AMH=0 : PI = 3.14159
156 PRINT: "PROGRAM STIRLING TEST CALCS"
158 INPUT "Is data available from the data logger? (y,n) " AS
160 IF (AS="y") AND (AS="y") AND (AS="n") AND (AS="n") THEN LOCATE 1: PRINT"
162 IF (AS="y") OR (AS="n") THEN GOSUB 3000
168 IF CHECK = 1 THEN GOTO 190
185 INPUT "Enter the test number. (ex. S714T1)"($)5
190CLS
194
t print the format for inputting the data
198
201 LOCATE 1.8: PRINT "PROGRAM STIRLING TEST CALCS TEST FILE NO = ";$H
205 LOCATE 3.1: PRINT "BED:"
207 LOCATE 4.10: PRINT "Temp (C) of inkt combustion air (TIN) 1:";
220 LOCATE 4.10: PRINT "Temp (C) of exhaust (TEX) 2:";
225 LOCATE 5.10: PRINT "Temp (C) change across insulation (DTH) 3:";
230 LOCATE 6.10: PRINT "Temp (C) change across head (DT(H) 4:";
235 LOCATE 7.10: PRINT "Temp (C) change outside head (DTO) 5:";
237 LOCATE 8.1: PRINT "ORIFICE:"
238 LOCATE 8.10: PRINT "Temp (C) of air at orifice (TAIR) 6:";
243 LOCATE 9.10: PRINT "Orifice static pressure (mmHg) (SP) 7:";
248 LOCATE 10.10: PRINT "Orifice differential pressure(mmH2O)(SF) 8:";
252 LOCATE 11.1: PRINT "AMBIENT:"
253 LOCATE 11.10: PRINT "Temp (C) of ambient air (TAMB) 9:";
257 LOCATE 12.10: PRINT "Dew point temperature (C) (TDPT) 10:";
258 LOCATE 13.10: PRINT "Atmospheric pressure (mm Hg) (PATHM) 11:";
256 LOCATE 14.1: PRINT "FUEL:"
257 LOCATE 14.10: PRINT "Flow rate of corn cobs (lbs/hr) (FR) 12:";
259 LOCATE 15.10: PRINT "Moisture content of corn cobs (%wb) (MC) 13:";
260 LOCATE 16.1: PRINT "ENGINE:"
261 LOCATE 16.10: PRINT "Flow rate of cooling water (gpm) (CW) 14:";
262 LOCATE 17.10: PRINT "Temp (C) change of cooling water (DTCW) 15:";
263 LOCATE 18.10: PRINT "Temp (C) of head (THO) 16:";
264 LOCATE 19.10: PRINT "Power output of stirling (hp) (P) 17:";
265
270 ' INPUT DATA LOCATIONS
275
290 FOR I = 1 TO NUM
291 X(I) = 1 + 2
292 Y(I) = 58
295 LOCATE X(I),Y(I)
300 IF CHECK = 1 THEN PRINT C(I): GOTO 320
310 INPUT C(I)
320 NEXT I
350 ' CHECK DATA AND MAKE CORRECTIONS
360
380 LOCATE 23: PRINT "
382 LOCATE 23: PRINT "ANY CORRECTIONS (Y,N)"
390 LOCATE 23.24: INPUT AS
400 IF (AS="y") AND (AS="y") AND (AS="n") AND (AS="n") THEN LOCATE 23.20:PR INF"
410 IF (AS="n") OR (AS="n") THEN GOTO 475
415 LOCATE 23: INPUT "How many corrections are there (limited to 5)"; NUMCOR
417 LOCATE 23: PRINT "
420 LOCATE 23: PRINT "The numbers to correct are:
425 FOR J = 1 TO NUMCOR
430 LOCATE 23,(J*5+25):INPUT COR(J)
435 NEXT J
440 FOR J = 1 TO NUMCOR
445 NO = COR(J)
450 LOCATE X(NO),Y(NO):INPUT C(NO)
455 NEXT J
460 GOTO 380
465 LOCATE 23: PRINT "Is the printer ready? If o.k. press any key to continue"
470 AS = INKEYS: IF AS = "" THEN 470
475
480 * SET VARIABLES AND CHANGE UNITS
490
491 TIN=CC(1)+.8+32 : TEX=CC(2)+1.8+32 : DTL = C(3) : DTAV = C(4)
492 DDO = C(5) : TAIR=C(6)+1.8+32 : TDPT=CT(1)+1.8+32+TK
493 X = CC(8)/25.4 : PC=C(17) : THO = C(18)
494 DTCW=C(15) : TAMC=B(11)+.8+32 : PATH=C(11)+.019337 : FR = C(12) : CW = C(14)
495 OD = orifice diameter (in), PD = pipe diameter (in), ECOMB = gross heating value of corn cobs (btu/lb), TCOND = thermal conductivity of 321SS (W/mK)
496 MO = 1 : ECOMB = 7000 : PD = 1.908 : OD = 1.276 : TCOND = 22.5
497 * for air : MW=molecular weight,XW% in air,1=nitrogen,2=oxygen,3=water
498 4=carbon dioxide,5=argon,6=sulfur dioxide,7=carbon monoxide,8=ash
500 (7)=28.011:MW(8)=1
501 WXY(1)=.7552 :WXY(2)=.2315 :WXY(3)=.0004 :WXY(4)=.0128 :WXY(5)=.0073 :WXY(7)=.0004
502 * for fuel : MFW=molecular weight,WF% in fuel,1=nitrogen,2=oxygen,3=hydrogen
503 4=carbon,5=argon,6=sulfur,7=carbon,8=ash
504 MFW(1)=28.016 :MFW(2)=32:MFW(3)=2.016 :MFW(4)=12.011 :MFW(5)=39.944 :MFW(6)=32.0
505 WFM(7)=12.011 :MFW(8)=1
506 WFX(1)=.003089 :WFX(2)=.44105 :WFX(3)=.05982 :WFX(4)=.4d :WFX(5)=0.0 :WFX(6)=.00061 :WFX(7)=
507 WFX(8)=.01483
508 * assign constants for specific heat evaluation
509 FOR I = 1 TO 7
510 FOR J = 1 TO 5
511 READ A(J,I)
512 DATA 9.470,0.0,-3470,1160000
513 DATA 11.515,0.0,-175,1530.0
514 DATA 19.866,0.0,-597,7500.0
515 DATA 18.2,0.0,-6530,1410000
516 DATA 4.953,0.0,0.0
517 DATA 9.48,0.0,-3290,1070000
518 DATA 0.0,0.0,0.0
519 NEXT J
520 NEXT I
521 * CALCULATIONS
522 * CALCULATE AIR AND FUEL FLOW RATES
523 * method for air flow and page numbers are from "Principles and
524 * Practice of Flow Meter Engineering" by L.R. Spink, 8th ed, 1972
525 527 * find actual molecular weight of air (AMW)
528 FOR I = 1 TO 5
529 AMW = MFW + XW(I) * MW(I)
530 NEXT I
531 533 * absolute static pressure of flowing gas (FF) in psi
534 SP = PATH + 60
535 * find the vapor pressure of the water in the air (VAP)
536 IF (TDPT > 0) THEN GOTO 580
537 VAP = EXP(23.3924-11286.5489#*TDPT-.46057#*LOG(TDPT))
538 GOTO 585
539 VAP = EXP(54.8229-12301.688#*TDPT-5.18923#*LOG(TDPT))
540 * ratio of orifice diameters (B)
541 B = OD/PD
542 * constant found from B (6) page 527
543 S =.598 * .01 + S + 1.9470-05 = (10 + B)4.425 = B2
544 * Z = compressibility ratio at TAIR and PP from page 373-6
545 FA = correction for temp expansion from page 156
546 FM = manometer correction factor from page 330
547 FC = reynolds number correction from page 225
548 * Y = expansion factor from chart on page 356
549 L1 = FA 1 : FFH1 = FCs1
550 Y = 1 - (FM * FA) / DF / FF ) .0125
551 GF = (AMW - PP - VAP + MW(3) * VAP) / (10.73 + TAIR + Z)
552 LBAH = rate of air flow in lb air/hour page 332 eq 45
553 LBAH = 1.65 * S + PD + DP * FA * FM * FC * Y = (GF + DF) .5
554 DFR = dry fuel flow rate, WFR = water in fuel flow rate (lb/hr)
555 DFR = (1 - MC) = FR
556 WFR = MC = FR
557 calculate the amount of water in the air
558 XY(3) = MW(3) = VAP / (AMW - (PATH - VAP))
559 Z = XY(3) / (1 + XY(3))
560 DLABH and WLABH = dry air flow rate and water in air flow rate (lb/hr)
561 DLABH = (1 - Z) = LBAH
562 WLABH = Z = LBAH
563 calculate the weight of each component in the air
564 FOR I = 1 TO 7
565 ZW(I) = XY(I) * DLABH + MW(I) * DF / MW(F)
566 NEXT I
567 NEXT J
568 J = 3 TO 7
569 ZT(2) = ZW(2) - (MW(J) - MFW(J)) = WJ / MFW(J)
'CALCULATE HEAT TRANSFER RATES
701'
702 'temp of fuel in = TFIM, reference temp = TREF
703 TFIM = TAMB : TREF = TAMB
704 'energy in fuel and air EFIN and EAIN (btu/hr) or EFIN and EAIM (W)
705 EFIN = (ECOMB + .36*(TFIM-TREF))*DFR+1*(TFIM-TREF)*EFW-1059.3*(Z(W3)-WLAB)
706 EFIN = EFIN / 3.4123
707 'energy in fuel and air EDMIN and EDAIM (W)
708 EDMIN = .24*(TMIN-TREF)*DLBHA + .45*(TMIN-TREF)*WLAB
709 EDAIM = EAIN / 3.4123
710 'energy out fuel and air EFOUT and EAOOUT (btu/hr) or EFOOUTH and EAOOUTH (W)
711 EFOUTH = .38 * (TEX - TREF) / .02 * DFR
712 EFOUTH = EFOUTH / 3.4123
713 'evaluate integrals for specific heats
714 TC = TEX + TK : TR = TREF + TK
715 P(J) = RC(TC - TR)
716 P(J) = RC(TC - TR)
717 P(J) = P(J)
718 FOR I = 1 TO 7
719 H(I) = 0
720 FOR J = 1 TO 5
721 H(I) = H(I) + (H(I) = H(I) + A(U(I) = P(J)
722 NEXT J
723 NEXT I
724 EAOUT = EAOOUT + H(I) = Z(W3) / W(W3)
725 NEXT I
726 NEXT I
727 EAOOUTH = EAOOUTH / 3.4123
728 'energy lost through insulation (from experimentation) ELOSS and ELOSSM (W)
729 ELOSS = ELOSS / 3.4123
730 'energy into stirling engine Q (btu/ib), QM (W)
731 QM = EFIM + EAIN - EFOUTH - EAOOUTH - ELOSS
732 NEXT I
733 NEXT I
734 NEXT I
735 'CALCULATE MEAN HEAT TRANSFER COEFFICIENTS
736 'since the inside flow rate is sinusoidal in nature the coeff.
737 'calculated are mean coefficients
738 'outside (RO) and inside (RI) radius (m), length of cylinder (CYLEN)(m)
739 'end surface area (ESA) (m2), total outside surface area (AREA)(m2)
740 RO = .16 : RI = .157 : CYLEN = .218 : ESA = .132
741 AREA = 2*PI*RO*CYLEN*ESA
742 'outside heat transfer coefficient U(btu/hrft2F), UM (W/m2K)
743 'based on outside surface area
744 UM = QM / (AREA = DTAV)
745 U = UM * .17612
746 'outside heat transfer coefficient H0(btu/hrft2F), HOM(W/m2K)
747 HOM = QM / (AREA = DTO)
748 HO = HOM * .17612
749 'assumed fin efficiency NF
750 NF = .75
751 'fin thickness FINTH(m), fin width FINW(m), fin area AP(m2), prime area AP(m2)
752 FINTH = .001 : FINW = .012 : AP = 1.28 : AP = .16 : ESA
753 'inside heat transfer coefficient H1(btu/hrft2F), H1M(W/m2K)
754 H1M = 1/(NF*AP*AP)/(DTAV*QH*(LOG(RO/RI)/(2*PI*CYLEN*TCND)+1)/(HOM*AREA)))
755 H1 = H1M * .17612
756 'calculate fin efficiency NFC, method from Shah pp202-212
757 'compare calculated fin efficiency to last eff. and iterate if necessary
758 IF (NFC > NF+.01) OR (NFC < NF-.01) THEN NFC = NFC : GOTO 1060
759 '
760 'CALCULATE EFFICIENCY
761 '
762 'overall efficiency EFF = power out / energy in fuel (%) 
763 EFF = 100 * (P = 2544) / EFIN
764 'transfer efficiency TREFF = energy into engine / energy available (%) 
765 TREFF = 100 * Q / (EFIN + EAIN)
766 'engine efficiency EFF = power out / energy into engine (%) 
767 EFF = 100 * (P = 2544) / Q
768 '
' CALCULATE ENERGY LOST IN COOLING WATER
1200 ' EW = energy lost in cooling water (BTU/hr). EWH (W)
1205 ' FW = mass flow rate of water (lb/hr)
1206 ' CPW = specific heat of water (BTU/lb F) at 300K
1207 ' GAMA = specific weight of water (lb/ ft3) at 300K
1208 ' SSHA = 62.25 : CPW = .998
1210 ' FW = CW * 60 = GAMA / 7.47
1215 ' EW = FW * CPW * 1.4 = DTCW
1220 ' EWH = EW / 3.412
1500 ' PRINT RESULTS
1501 :
1502 :
1503 CLS
1510 LOCATE 1,40: PRINT "TEST FILE NO : ":H=
1511 LOCATE 1,6 : PRINT "PROGRAM STIRLING TEST CALC'S"
1520 LOCATE 3,1 : PRINT "FLOW RATES"
1525 LOCATE 3,23: PRINT USING "FUEL: ":****.ss lb/hr:";FR
1530 LOCATE 4,23: PRINT USING "AIR: ":****.ss lb/hr:";LBAH
1535 FOR I = 1 TO PRINT "ENERGY CALCULATIONS BTU/hr":W"
1540 LOCATE 7,10: PRINT USING "IN: ":****.ss FUEL :****.ss
1545 LOCATE 9,10: PRINT USING "AIR :****.ss
1550 LOCATE 3,10: PRINT USING "OUT: ":****.ss FUEL :****.ss
1555 LOCATE 5,10: PRINT USING "OUT: ":****.ss
1560 LOCATE 10,10:PRINT USING "AIR :****.ss
1565 LOCATE 11,10:PRINT USING "OUT: ":****.ss
1570 LOCATE 12,10:PRINT USING "ENGINE :****.ss
1575 LOCATE 13,10:PRINT USING "WATER :****.ss
1580 LOCATE 15,10:PRINT USING "HEAT TRANSFER COEFS. BTU/(hr ft2 F) W/(m2 K)
1590 LOCATE 16,10:PRINT USING "OVERALL :****.ss
1700 ' .U.UH
1585 LOCATE 17,10:PRINT USING "OUTSIDE :****.ss
1590 LOCATE 18,10:PRINT USING "INSIDE :****.ss
1595 LOCATE 20,8 :PRINT USING "EFFICIENCIES OVERALL :****.ss".OFF
1600 LOCATE 21,10:PRINT USING "TRANSFER :****.ss".TREFF
1605 LOCATE 22,10:PRINT USING "ENGINE :****.ss".EFF
1900 ' SAVE DATA AND RUN AGAIN IF DESIRED
1902 :
1905 LOCATE 23:PRINT " 
1910 LOCATE 23:INPUT "Would you like to save data on disk (y,n)":A$ 
1915 LOCATE 23:PRINT " 
1920 IF (A$="y") AND (A$="y") AND (A$="n") AND (A$="n") THEN LOCATE 23:PRINT "GOTO 1910 
1925 IF (A$="y") OR (A$="n") THEN GOSUB 2000 
1948 LOCATE 23:PRINT " 
1950 LOCATE 23:INPUT "Would you like to run program again (y,n)":A$ 
1955 IF (A$="y") AND (A$="y") AND (A$="n") AND (A$="n") THEN GOTO 1950 
1960 IF (A$="n") OR (A$="n") THEN END 
1985 RUN 
2000 REM SUB TO SAVE NUMBERS 
2100 LOCATE 23 
2200 INPUT "Enter name of file to be saved ";NAMS 
2300 OPEN "O","S",NAMS+.dat" 
2355 FOR I = 1 TO NUM 
2500 PRINT #1, C(I) 
2600 NEXT I 
2700 CLOSE #1 
2800 RETURN 
3000 REM SUB TO LOAD FILE 
3100 INPUT "Enter name of the file to input from the disk ";R$ 
3200 ON ERROR GOTO 5000 
3255 OPEN "1","S",R$+.dat" 
3300 FOR I = 1 TO NUM 
3400 INPUT #1,C(I) 
3450 NEXT I 
3500 CLOSE #1 
3600 CHECK = 1 
3700 RETURN 
5000 REM SUB IF FILE NOT FOUND 
5010 IF ERR = 53 THEN CLOSE : GOTO 5020 
5015 ON ERROR GOTO 0 
5020 PRINT "The file ";R$: was not found on disk 
5030 PRINT "Please try another file name" 
5040 GOTO 156
APPENDIX C

EXTERNAL HEAD FINS CALCULATIONS
The Basic program "SFIN" was written to perform the calculations for external fins on the engine head. The parts of the program that differ from program "STC" are presented in this appendix. The fin calculation results using the data from two engine tests as input are presented in Table 8. The calculations were performed for fin spacings of 10 mm and 20 mm. The input data is read from an external file that was created by a Fortran program that averaged the experimental data over the length of the test. This data is shown at the top of the sample outputs in numbers 1) - 6).
### Table 8. Results of external fin calculations.

<table>
<thead>
<tr>
<th>PROGRAM CIRCULAR FINS</th>
<th>TEST FILE NO - F1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temp (C) at base of fin.</td>
<td>(TB) 1: 633</td>
</tr>
<tr>
<td>Temp (C) change across head</td>
<td>(DT) 2: 357</td>
</tr>
<tr>
<td>Outside heat transfer coef. (w/m²k)</td>
<td>(HO) 3: 330</td>
</tr>
<tr>
<td>Inside heat transfer coef. (w/m²k)</td>
<td>(HI) 4: 89.9</td>
</tr>
<tr>
<td>Inside fin efficiency</td>
<td>(NFI) 5: .75</td>
</tr>
<tr>
<td>Thermal conductivity (w/mk) of fin.</td>
<td>(K) 6: 22.5</td>
</tr>
</tbody>
</table>

The heat transfer rate with out fins is compared to the 5 best calculated heat transfer rates with fins.

<table>
<thead>
<tr>
<th>Q with out fins = 19693 W</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q (W)</td>
</tr>
<tr>
<td>0.007</td>
</tr>
<tr>
<td>0.008</td>
</tr>
<tr>
<td>0.008</td>
</tr>
<tr>
<td>0.008</td>
</tr>
<tr>
<td>0.007</td>
</tr>
</tbody>
</table>

Minimum fin spacing = .01 m

<table>
<thead>
<tr>
<th>Q with out fins = 18690 W</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q (W)</td>
</tr>
<tr>
<td>0.008</td>
</tr>
<tr>
<td>0.008</td>
</tr>
<tr>
<td>0.009</td>
</tr>
<tr>
<td>0.009</td>
</tr>
<tr>
<td>0.008</td>
</tr>
</tbody>
</table>

Minimum fin spacing = .02 m
program calculates heat transfer rates for a circular cylinder with and without fins. The fin thickness, height, and number of fins are found and printed for the 5 largest finned heat transfer rates. The ranges are set in lines 513, 515, and 528. The fin data can be read and stored in an external file. The minimum fin spacing is set in line 194. The equations used are obtained from Fig. 3.19 in "Introduction to heat transfer" by Incropera and Dewitt.

THOMAS ANZALONE

JUNE 1989 BASIC PROGRAM "FINS"

LOCATE 3,8: PRINT "PROGRAM CIRCULAR FINS" TEST FILE NO = ";HS
LOCATE 5,10: PRINT "Temp (C) at base of fin. (TB) 1:";
LOCATE 6,10: PRINT "Temp (C) change across head (DT) 3:";
LOCATE 7,10: PRINT "Outside heat transfer coef. (w/m2k) (RO) 3:";
LOCATE 8,10: PRINT "Inside heat transfer coef. (w/m2k) (HI) 4:";
LOCATE 9,10: PRINT "Inside fin efficiency (WFI) 5:";
LOCATE 10,10: PRINT "Thermal conductivity (w/mk) of fin. (K) 6:";

TRANSFER DATA

TB = C(1); DT = C(2); RO = C(3); HI = C(4); WFI = C(5); K = C(6)
RD = .18; RI = .157; CYLEN = .218; ESA = .132
API = 1.2; AF = 1.18 + ESA

= CALCULATIONS

outside surface area without fins (AREA) .m2
AREA = 2 * PI = RO * CYLEN + ESA
heat transfer with cut fins (QWO) .w
QWO = (GMF*(API))*LOG(RO/RI)/(2*PI*CYLEN*K).w

= calculations for fins on outside
fin thickness = t (m), fin height = d (m), number of fins = N

FOR T = .001 TO .05 STEP .001
T = 1 TO 50

= calculate space between fins (s)
S = (CYLEN+RD)/(N+1)

= if spacing is less than allowable then go to next L
IF S < 0.75 THEN GOTO 563

= calculate constants needed to use fig 3.19
ROF = L + RO
RC = ROF + T/2
LC = L + T/2
AF = LC / T

= find the line on fig 3.19 to follow
LIM = RC/RO
fig 3.19 is limited to values of LIN between .5 and 6
IF LIN < .5 THEN GOTO 583
IF LIN > 6 THEN GOTO 586

= calculate the value for the x-axis of fig 3.19
X = (LC-.5) / (ROF*K*AF) .5

fig 3.19 is limited to x-axis values between 0 and 2.5
IF (X < 0) OR (X > 2.5) THEN GOTO 583

= find the y-axis value for LIN = 1 & correction for other LIN values
Y = (5.84*4.5*EXP(-2.34*1.78*X^2+.16*X-8.84*X^4)) / 100
COR = (-2.17*3.03*LIN-10.3*LIM-21.63*LIM^2-7.09*LIM^3-11.027*LIM^4)/100

= find and save the 5 best combinations
FOR I = 1 TO 5
IF ABS(QWO) > ABS(QWO) THEN GOTO 565 ELSE GOTO 578

Q(I) = QWO
TF(I) = T
L(I) = L
NOF(I) = N
GOTO 540

NEXT I
NEXT N
NEXT L
NEXT T
APPENDIX D

HEAT EXCHANGER CALCULATIONS
The Basic program "TUBE" was written to perform the calculations to size the heat exchanger to preheat the combustion air with the exhaust from the SFBC. The parts of the program that differ from program "STC" are presented in this appendix. The data used to size the exchanger is presented in numbers 1) - 14) in Table 9. Numbers 11 - 14 were varied to obtain the desired energy recovery, length of the exchanger, and pressure drop. The results of the calculations are shown below the input in Table 9.

The program was modified to calculate the outlet temperatures given the geometry of the heat exchanger. The input and output of this program for a heat exchanger test are shown in Table 10. The temperature profile of the heat exchanger inlet and outlet temperatures for the test is presented in Fig. 26. The length of the exchanger used in the calculations was the length between the inlet and outlet temperature measurement points. This is less than the design length of the exchanger.
### Table 9. Results of heat exchanger calculations.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bed: Temp (C) of inlet combustion air</td>
<td>(TIN) 1)</td>
<td>101</td>
</tr>
<tr>
<td>Bed: Temp (C) of exhaust</td>
<td>(TEX) 2)</td>
<td>764</td>
</tr>
<tr>
<td>Orifice: Temp (C) of air at orifice</td>
<td>(TAIR) 3)</td>
<td>51.6</td>
</tr>
<tr>
<td>Orifice static pressure (mmHg)</td>
<td>(SP) 4)</td>
<td>67.7</td>
</tr>
<tr>
<td>Orifice differential pressure (mmH2O)</td>
<td>(DP) 5)</td>
<td>62.4</td>
</tr>
<tr>
<td>Ambient: Temp (C) of ambient air</td>
<td>(TAMB) 6)</td>
<td>38.4</td>
</tr>
<tr>
<td>Ambient: Dew point temperature (C)</td>
<td>(TDPT) 7)</td>
<td>19.1</td>
</tr>
<tr>
<td>Atmospheric pressure (mm Hg)</td>
<td>(PATM) 8)</td>
<td>738</td>
</tr>
<tr>
<td>Fuel: Flow rate of corn cobs (lb/hr)</td>
<td>(FR) 9)</td>
<td>21.6</td>
</tr>
<tr>
<td>Fuel: Moisture content of corn cobs (%wb)</td>
<td>(MC) 10)</td>
<td>10</td>
</tr>
<tr>
<td>Heat Ex: Cooling air outlet temp (C)</td>
<td>(TCO) 11)</td>
<td>400</td>
</tr>
<tr>
<td>Heat Ex: Tube inside diameter (in)</td>
<td>(DIT) 12)</td>
<td>2.01</td>
</tr>
<tr>
<td>Heat Ex: Tube outside diameter (in)</td>
<td>(DOT) 13)</td>
<td>2.38</td>
</tr>
<tr>
<td>Heat Ex: Number of tubes</td>
<td>(NT) 14)</td>
<td>1</td>
</tr>
</tbody>
</table>

### Program Tubular Heat Exchanger Test File No. H1

<table>
<thead>
<tr>
<th>Flow Rates</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>165 lb/hr</td>
</tr>
<tr>
<td>Exhaust</td>
<td>187 lb/hr</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Outlet Temps</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Side</td>
<td>400 C</td>
</tr>
<tr>
<td>Exhaust</td>
<td>496 C</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Required Heat Transfer</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>25625 btu/hr</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Tube Diameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside</td>
<td>2.01 in.</td>
</tr>
<tr>
<td>Outside</td>
<td>2.38 in.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Number of Tubes</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1 tubes</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Gap (in)</th>
<th>HTCBE (btu/hr ft² F)</th>
<th>HNIN (ft)</th>
<th>LENGTH (ft)</th>
<th>DPTUBE (psi)</th>
<th>DPAHN (psi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.06</td>
<td>9.47</td>
<td>54.80</td>
<td>7.05</td>
<td>0.02</td>
<td>10.51</td>
</tr>
<tr>
<td>0.13</td>
<td>9.47</td>
<td>26.83</td>
<td>7.84</td>
<td>0.03</td>
<td>1.40</td>
</tr>
<tr>
<td>0.19</td>
<td>9.47</td>
<td>17.53</td>
<td>8.63</td>
<td>0.03</td>
<td>0.44</td>
</tr>
<tr>
<td>0.25</td>
<td>9.47</td>
<td>12.89</td>
<td>9.43</td>
<td>0.03</td>
<td>0.19</td>
</tr>
<tr>
<td>0.31</td>
<td>9.47</td>
<td>10.11</td>
<td>10.22</td>
<td>0.04</td>
<td>0.10</td>
</tr>
<tr>
<td>0.38</td>
<td>9.47</td>
<td>8.27</td>
<td>11.01</td>
<td>0.04</td>
<td>0.06</td>
</tr>
<tr>
<td>0.44</td>
<td>9.47</td>
<td>6.95</td>
<td>11.80</td>
<td>0.04</td>
<td>0.04</td>
</tr>
<tr>
<td>0.50</td>
<td>9.47</td>
<td>5.97</td>
<td>12.58</td>
<td>0.04</td>
<td>0.03</td>
</tr>
</tbody>
</table>
LOCATE 1.8: PRINT "PROGRAM TUBULAR HEAT EXCHANGER TEST FILE NO.': "$#
LOCATE 3.1: PRINT "BED:"#
LOCATE 3.10: PRINT "Temp (C) of inlet combustion air (TIN) 1:"#.2
LOCATE 4.10: PRINT "Temp (C) of exhaust (TEX) 2:"#.2
LOCATE 5.1: PRINT "ORIFICE:"#
LOCATE 5.10: PRINT "Temp (C) of air at orifice (TAIR) 3:"#.2
LOCATE 6.10: PRINT "Orifice static pressure (mmHg) (SP) 4:"#.2
LOCATE 7.10: PRINT "Orifice differential pressure(mmH2O)(DP) 5:"#.2
LOCATE 8.1: PRINT "AMBIENT:"#
LOCATE 8.10: PRINT "Temp (C) of ambient air (TAMB) 8:"#.2
LOCATE 9.10: PRINT "Dew point temperature (C) (TDPT) 7:"#.2
LOCATE 10.10: PRINT "Atmospheric pressure (mm Hg) (PATH) 8:"#.2
LOCATE 11.1: PRINT "FUEL:"#
LOCATE 11.10: PRINT "Flow rate of corn cobs (lb/hr) (FR) 9:"#.2
LOCATE 12.10: PRINT "Moisture content of corn cobs (Kwb) (MC) 10:"#.2
LOCATE 13.1: PRINT "HEAT EX:"#
LOCATE 13.10: PRINT "Cooling air outlet temp (C) (TCO) 11:"#.2
LOCATE 14.10: PRINT "Tube inside diameter (in) (DIT) 12:"#.2
LOCATE 15.10: PRINT "Tube outside diameter (in) (DOT) 13:"#.2
LOCATE 16.10: PRINT "Number of tubes (NT) 14:"#.2
TIN=C(1)+1.8*32: TEx=C(2)+1.8*32: NT=C(14)
TAIR=C(3)+1.8*32+TK: TDPT=C(7)+1.8*32+TK
SP=C(4)*.018337: PC=C(5)*.018337: PD=1.998+0.018337: OD=1.74E-09
TAMB=C(5)+1.8*32: PATH=C(8)*.018337: PD=1.998: OD=1.74E-09
FR=C(9): MC=C(10)/100: TCO=C(11)
TCOND=.251: DIT=C(12)/12: DOT=C(13)/12
'FIND THE FLOW RATE OF THE EXHAUST
TOTWT=0
FOR I = 1 TO 2
TOTWT=TOTWT+ZM(I)
NEXT I
'find flow rates for each tube
LBAH=LBAH/NT: TOTWT=TOTWT/NT
'*** CALCULATE PROPERTIES FOR AIR ***
TCA=TCA-TCA; deg C
TCO+TCA+2: TCA=TCA
TII=TII+TII: deg R
TII=TII+1.8*(TII)+32+TK
enthalpies of air in btu/lb (HI and HOUT)
HI=4.8203+.22239*TIIL+1.3739E-06*TIIL-2
HOUT=4.8203+.22239+TOO+1.3739E-06*TOO-2
'find specific heat (CP) in J/kg K and convert to btu/lbm F
CP=1.0495-3.4978E-04*T+8.7209E-07*T^2-5.5027E-10*T^3+1.8615E-13*T^4
CP=CP+1000*2.3886E-04
'find density (RO) in kg/m3 and convert to lb/ft3
RO=8.6125-.70952E-3*T+.3723E-04*T^2-1.0151E-06*T^3-.7232E-09*T^4+1.652E-12
RO=RO+.062428
'find viscosity (VIS) in N s/m2 and convert to lbm/ft hr
VIS=.00000001*(.2-0.0515+.78966E-6.5909E-04*T^2+3.9818E-07*T^3-1.1538E-10*T^4+4.6152E-14*T^5)
VIS=VIS/.2419.1
'find thermal conductivity (K) in W/m K and convert to btu/hr/ft
K=.001*(-.7.1009+.1.6723*T-.2.8402E-4*T^2-.2.9489E-9*T^3-.8.9919E-11*T^4+1.50
3E-14*T^5)
K=.57782
' find pertinent number (FR)
FR=69308-.001583*T+6.8874E-06*T^2-.2.0748E-08*T^3+3.9712E-11*T^4-.8.6103E-14*T^5+4.3289E-17*T^6+3.5295E-24*T^7+8.4.70340E-26*T^8+2.4631E-32*T
'IF EXHAUST = 1 THEN GOTO 904
' set the values for air
ROA = ROA * VIS: KA = K / PRA = FR
475'
**CALCULATE PROPERTIES FOR EXHAUST**

EXHAUST = 1

**Flow rate of exhaust in each tube (MEX) lbm/hr**

MEX = TOTWT

**Ratio of inlet air to exhaust**

RATIO = LBAH / MEX

**TREF = TAM**

**Energy in inlet air (EAIN) btu/hr**

EAIN = .24*(TIN-TREF)*DLBAH + .45*(TIN-TREF)*WLSBAH

**Evaluate integrals for each component to calculate energy out (EAOUT)**

TC = TEX + TK : TR = TREF + TK

P(i) = TC - TR

P(2) = (TC-2) - TR**2/2

P(3) = 2 * (TC-.5 - TR -.5)

P(4) = LOG(TC / TR)

P(5) = -(1/TC - 1/TR)

FOR I = 1 TO 7

H(i) = 0

FOR J = 1 TO 5

H(i) = H(i) + A(J, I) * P(J)

NEXT J

NEXT I

FOR I = 1 TO 5

EAOUT = EAOUT + H(I) * ZW(I) / HM(I)

NEXT I

**Calculate specific heat of exhaust (CPEX) btu/lbm F**

CPEX = (EAOUT - EAIN) / (MEX * T = (TEX - TIN))

**Find required heat transfer btu/hr**

GREQ = LBAH / (HOUT - HI)

**Find output temp of exhaust and average temp of exhaust**

THI = TEX

**Calculate properties for exhaust**

GOTO '20

ROEX = RO / RATIO : VISEX = VIS : KEX = K : PREX = PR * CPEX / CP

**Print output format**

GOSUB 1500

**Heat transfer and pressure drop calculations**

**Find tube (IT) and annulus (TA) wall temp R**

TT = (TCA + THA) / 2 - 273

**Find delta t log mean (DTLM) for the exchanger**

DTLM = DTI - (TTO - TK) : DT2 = THO - (TII - TK)

**Find flow area (TFA) ft2**

TFA = P1 * (DIT/2)**2

**Do calculations for several gap sizes from 1/16 to 1/2 step 1/16**

FOR GAP = .0625 TO .5 STEP .0625

Find dia of annulus (DIA) ft and annulus flow area (AFA) ft2

DIA = DOT + .25 * GAP/12

AFA = P1 * ((DIA/2)**2 - (DOT/2)**2)

**Calculations for heat transfer coefficient**

**For annulus**

FA = (1.58 * LOG(FA) - 3.28)**-2

NUA = (FA/2) * (REA-1000) / FRA / (1 + 12.7 * (FA/2)**.5 * (FRA**(-2/3)-1))

HANN = NUA / (DIA - DOT)

**For tube**

RENT = RET / (DIT = DI = VISEX)

**Overall heat transfer coefficient, U (btu/hrft**2)

U = 1 / (1/HTUBE + LOG(DOT/DIT)**2 + DIT(2*TCOND) + DIT(4*DHAH))

**Calculate pressure drop**

**For annulus**

**For hydraulic radius RHA(ft), velocity = UMA(ft/s), pressure drop = DPA(psi)**

RHA = (DIA - DOT) / 4

UMA = LBAH / (3600 * ROA * AFA)

DPA = FA / (LENGTH / RHA) = (RCA - UMA**2 / (2 * GC)) / 144
Fig. 26. Temperature profile of heat exchanger test.
### Table 10. Heat exchanger test results.

<table>
<thead>
<tr>
<th>Program Tubular Heat Exchanger</th>
<th>Test File No - T1</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Bed:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of inlet combustion air</td>
<td>(TIN) 1: 258</td>
</tr>
<tr>
<td>Temp (C) of exhaust</td>
<td>(TEx) 2: 745</td>
</tr>
<tr>
<td><strong>Orifice:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of air at orifice</td>
<td>(TAIR) 3: 42.8</td>
</tr>
<tr>
<td>Orifice static pressure (mmHg)</td>
<td>(SP) 4: 59.2</td>
</tr>
<tr>
<td>Orifice differential pressure (mmH2O)</td>
<td>(DP) 5: 87.9</td>
</tr>
<tr>
<td><strong>Ambient:</strong></td>
<td></td>
</tr>
<tr>
<td>Temp (C) of ambient air</td>
<td>(TAMB) 6: 30.2</td>
</tr>
<tr>
<td>Dew point temperature (C)</td>
<td>(TDPT) 7: 15.2</td>
</tr>
<tr>
<td>Atmospheric pressure (mm Hg)</td>
<td>(PATM) 8: 744</td>
</tr>
<tr>
<td><strong>Fuel:</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate of corn cobs (lb/hr)</td>
<td>(FR) 9: 20.9</td>
</tr>
<tr>
<td>Moisture content of corn cobs (%)</td>
<td>(MC) 10: 9.8</td>
</tr>
<tr>
<td><strong>Heat Ex:</strong></td>
<td></td>
</tr>
<tr>
<td>Cooling air inlet temp (C)</td>
<td>(TCI) 11: 42.8</td>
</tr>
<tr>
<td>Cooling air outlet temp (C)</td>
<td>(TCO) 12: 274</td>
</tr>
<tr>
<td>Hot air inlet (C)</td>
<td>(THI) 13: 557</td>
</tr>
<tr>
<td>Hot air outlet (C)</td>
<td>(THO) 14: 398</td>
</tr>
</tbody>
</table>

### Program Tubular Heat Exchanger Test File No : T1

<table>
<thead>
<tr>
<th>Flow Rates</th>
<th>AIR: 199 lb/hr</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Outlet Temps:</strong></td>
<td>EXHAUST: 220 lb/hr</td>
</tr>
<tr>
<td>AIR SIDE:</td>
<td>245 C</td>
</tr>
<tr>
<td>EXPERIMENTAL</td>
<td>274 C</td>
</tr>
<tr>
<td>EXHAUST:</td>
<td>379 C</td>
</tr>
<tr>
<td>EXPERIMENTAL</td>
<td>398 C</td>
</tr>
</tbody>
</table>

| Required Heat Transfer               | 20155 btu/hr |

<table>
<thead>
<tr>
<th>Geometry</th>
<th>TUBE ID: 2.01 in</th>
</tr>
</thead>
<tbody>
<tr>
<td>OD: 2.38 in</td>
<td></td>
</tr>
<tr>
<td>ANNULUS ID: 3.125 in</td>
<td></td>
</tr>
<tr>
<td>LENGTH: 10.6 ft</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Heat Transfer Coefficients (BTU/hr ft2 F)</th>
<th>TUBE: 9.9</th>
</tr>
</thead>
<tbody>
<tr>
<td>PENNULUS: 9.5</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure Drops (psi)</th>
<th>TUBE: 0.034</th>
</tr>
</thead>
<tbody>
<tr>
<td>PENNULUS: 0.070</td>
<td></td>
</tr>
</tbody>
</table>

| Fuel Savings                           | 18649 btu/hr (11.8%) |
APPENDIX E

MAJOR SYSTEM COMPONENTS
The two major components of the system are the fluidized bed combustor and the ST-5 stirling engine. The uninsulated fluidized bed combustor that was used in the testing of the stirling engine is presented in Fig. 27. The information presented on the ST-5 was taken from the manufacturer's brochure on the engine (Stirling Technology, 1987).

Fig. 27. Uninsulated fluidized bed combustor used in stirling engine tests. Shown with heat exchanger head installed.
The ST-5 Multifuel External Combustion Engine

Fig. 28. The ST-5 stirling engine. Shown with production finned head. (Stirling Technology, 1987)

Energy Balance in Total Home Energy System

Fig. 29. Manufacturer's energy balance in total home energy system. (Stirling Technology, 1987)
The ST-5 operates on the temperature difference between its hot and cold ends. At one end is the heater head which is inserted in a burner; at the other, the cooling jacket and crankcase. These are the three components of the engine body which contain air, pressurized to 5 bar (approximately 72 p.s.i.) by a built-in air compressor.

When heat is applied to the heater head by burning some combustible material in the burner, the air contained in the head is caused to expand and move to the cold end through a heat sponge called a regenerator. At the cold end, the air is cooled and compressed, and is then moved back to the hot end again through the regenerator. The purpose of the regenerator is to improve efficiency by absorbing heat when the air is shuttled to the cold end and releasing heat when the air returns to the hot end.

When the air in the hot end expands, it produces work which drives the piston towards the cold end. At the end of the piston's stroke, the displacer is moved by a mechanical linkage which pushes the expanded gas through the heater, regenerator and cooler into the cold end. The piston now returns from the end of its stroke, compressing the cold gas, and the displacer moves down towards the piston, squeezing the compressed gas in a reverse direction through the heat exchanger loop to the hot end. The cycle then repeats itself. This linear movement of the piston is converted into the rotary movement of the flywheel by means of a simple linkage contained in the crankcase. The crankcase has been designed to run dry, not needing any oil or grease. The bearings in the linkage are sealed and require no lubrication.

The only moving parts in the engine are the displacer, piston and linkage. This is what makes the engine extremely reliable and easy to service and maintain.

Fig. 30. How the ST-5 works. (Stirling Technology, 1987)
The ST-5 has been designed for those who either cannot obtain grid electricity easily and economically or have realized the uncertainty of total dependence on utilities to supply their power needs. If you need electricity at remote locations where grid power is not a feasible proposition; have experienced the inconvenience of grid power failure; wish to avoid the predicted utility "rate shock"; have found the internal combustion engine an undesirable backup for your wind or photovoltaic system; actively seek alternative sources of energy for your security and peace of mind, or if you simply desire an Independent Home Energy System for your home or greenhouse, the ST-5 may be the solution you have been looking for.

The ST-5

The ST-5 is a rugged engine designed to produce over 3 hp of mechanical power reliably and on demand, directly from a variety of renewable and gaseous fuels. Its external combustion Stirling cycle eliminates spark-plugs, fuel-injection systems, mufflers, noxious fumes and, very noticeably, the noise of internal combustion engines. This lack of complexity coupled with the use of high quality sealed bearings and long life Teflon impregnated seals, eliminates the need for messy oil changes and makes possible a long, trouble-free engine life. The ST-5 has been designed to be user serviceable with a minimum of basic hand tools.

Most important of all, virtually any combustible material is a suitable fuel for the ST-5. Among various acceptable fuels are wood, wood pellets, husks and chaff, peanut shells, weeds and hay, cotton waste, other agro-byproducts, natural gas and propane. This wide variety of fuels ensures you produce your fuel on your own land.

The ST-5 System and Operation

The ST-5 system comprises two main components—the ST-5 (engine) and the burner. The function of the burner is to provide heat to the heater head of the ST-5, which is inserted into the burner. Two burner options are now available for the ST-5. One is a cyclone burner for small particle fuels such as sawdust or other shredded biomass. The second is a two stage wood burner. The fuel is fed into the burner either manually, as is the case with the wood burner or through a hopper, if one is using small particle fuel. Gaseous fuels may be used in the cyclone burner by modifying the intake orifice. The wood burner can hold enough fuel to run the engine at full output for two to four hours (depending on the wood used).

In order to obtain the high temperatures necessary to run the engine, both burners have a forced air blower. Once the engine is started, the blower is operated by the ST-5 itself. Before the engine can be started, however, the blower needs to be operated for 10 minutes either by a battery or by hand cranking until the desired heater head temperature is reached. Once the temperature is high enough (the ST-5 will start at temperatures around 95°F), an easy pull on the flywheel starts the engine. Once up to temperature this engine is an easy and sure starter, eliminating the need for multiple pulls on a recoil rope as is often experienced with conventional engines.

Wood Burner—
Heater Head Schematic

Just as the ST-5 takes a few minutes from the time you fire it up until the time you start it, it also takes a while to come to a stop even after the fuel supply has been cut off. This is because the heat retained in the heater head is sufficient for it to operate for 10 to 15 minutes, although at decreasing power output.

The cooling loop works like any automotive cooling system. A small pump, operated by the engine, circulates the cooling liquid (water or water and glycol) through an automotive radiator, while a fan mounted on the flywheel blows air across the radiator and dumps the unused heat into the atmosphere.

Fig. 31. Manufacturer's ST-5 system operation.
(Stirling Technology, 1987)
APPENDIX F

EXPERIMENTAL ERROR CALCULATIONS
Since measurements from several different instruments were used to compute the heat transfer coefficients, an analysis was performed to investigate the experimental error in the results. The method used is presented in Doeblin, (1983) and will be summarized here.

Consider a quantity, \( N \), which is a known function of \( n \) independent variables \( u_1, u_2, \ldots, u_n \):

\[
N = f(u_1, u_2, \ldots, u_n)
\]

(25)

where the \( u \)'s are the measured quantities and are in error by \( \pm \Delta u_1, \pm \Delta u_2, \ldots, \pm \Delta u_n \), respectively. The errors were considered to be \( \pm 3 \sigma \) statistical bounds. The proper method of combining such errors is the root-sum square (rss) formula

\[
E_{\text{rss}} = \sqrt{\left( \Delta u_1 \frac{\partial f}{\partial u_1} \right)^2 + \left( \Delta u_2 \frac{\partial f}{\partial u_2} \right)^2 + \ldots + \left( \Delta u_n \frac{\partial f}{\partial u_n} \right)^2}
\]

(26)

\( E_{\text{rss}} \) represents a \( \pm 3 \sigma \) limit on \( N \) and 99.7% of the values of \( N \) can be expected to fall within these limits. The \( u \)'s included values taken from tables and equations used in the calculations.

The results of the experimental errors for the heat transfer coefficients are presented in Table 11. A sample experimental error calculation is also presented.
<table>
<thead>
<tr>
<th>Heat Transfer Coefficient W/m² K</th>
<th>Experimental error +/- range +/- %</th>
</tr>
</thead>
<tbody>
<tr>
<td>overall</td>
<td>150</td>
</tr>
<tr>
<td>outside</td>
<td>350</td>
</tr>
<tr>
<td>inside</td>
<td>76</td>
</tr>
</tbody>
</table>

Note: Coefficients are averages from Table 3.

The overall heat transfer coefficient, $U$, was calculated using equation (20), therefore the experimental error in $U$ is given by:

$$E_{au} = \sqrt{\left(\frac{\partial U}{\partial DTot}\Delta DTot\right)^2 + \left(\frac{\partial U}{\partial Ao}\Delta Ao\right)^2 + \left(\frac{\partial U}{\partial Q}\Delta Q\right)^2}$$  \hspace{1cm} (27)

The quantities were found as follows:

The temperature difference, $DTot$ is given by:

$$DTot = Ts - Ta = 341 \, ^\circ C$$  \hspace{1cm} (28)

Since this equation is linear, the error in $DTot$ is:

$$\Delta DTot = \sqrt{\Delta Ts^2 + \Delta Ta^2} = \pm 6.8 \, ^\circ C$$  \hspace{1cm} (29)

The area, $Ao$, was taken from manufacturer's specifications:

$$Ao = .290 \, m$$  \hspace{1cm} (30)

$$\Delta Ao = \pm 0.002 \, m$$  \hspace{1cm} (31)

The heat transfer rate, $Q$, is given by equation (13):
\[ Q = 16000 \text{ W} \]  

(32)

with the error defined as:

\[ \Delta Q = \sqrt{\Delta E_{\text{FIN}}^2 + \Delta E_{\text{AIN}}^2 + \Delta E_{\text{FOUT}}^2 + \Delta E_{\text{AOUT}}^2 + \Delta E_{\text{LOSS}}^2} \]

\[ = 1290 \text{ W} \]  

(33)

The error term for each energy term was evaluated using the same method.

The partial derivatives are given as:

\[ \frac{\partial U}{\partial DT_{\text{tot}}} = - \frac{Q}{Ao \times DT_{\text{tot}}} = - 0.47 \text{ W}/(\text{m}^2\text{C}^2) \]  

(34)

\[ \frac{\partial U}{\partial Ao} = - \frac{Q}{Ao^2 \times DT_{\text{tot}}} = - 560 \text{ W}/(\text{m}^4\text{C}) \]  

(35)

\[ \frac{\partial U}{\partial Q} = \frac{1}{Ao \times DT_{\text{tot}}} = 0.01 \text{ l}/(\text{m}^2\text{C}) \]  

(36)

Substituting these values into equation (27):

\[ E_{\text{au}} = \pm 19 \% \]  

(37)

This represents \( \pm 3\sigma \) limits on \( Q \).
LIST OF REFERENCES


