Efficiency Improvements by Passive Control and Optimization of the Combustion Process and Engine Cooling

FINAL

Cary Chiu, Jeremy Bonifacio, Sruthi Nadimpalli, and Reza Toossi
Department of Mechanical and Aeronautical Engineering,
California State University, Long Beach

and,

Michael Sheppy, and Kambiz Vafai
Department of Mechanical Engineering,
University of California, Riverside

METRANS contract number 07-03

September 5, 2011

FINAL REPORT

Disclaimer

Abstract

An innovative passive thermal management cooling system deploying expandable thin film assembly is proposed. The system is sensitive to small changes in temperature and adjusts through thermal expansion of a stagnant gas. The expansion of the stagnant gas accommodates a sufficient increase in flow dimensions to control the flow rate. The proposed system is ideal for passive control of density and air/fuel ratio of fresh charge of an internal combustion engine. Other applications include thermal heat management of high-intensity microchips and other electronic systems, and design of a variety of biosensors.

A dual-layer thin film device has been designed that could prove effective as an alternative to carburetors. The device is capable to passively regulate the air/fuel ratio between 14.6 and 14.7 resulting in substantial reduction in pollutants emitted by internal combustion engines.

Table of Contents

List of Figures

List of Tables

Introduction

The internal combustion engine (ICE) is most commonly used for motor vehicles such as automobiles and motorbikes. It is indeed one of the most important inventions, yet it has some down sides, which must be controlled. Two major drawbacks of these systems are fossil-fuel consumption and air pollution. These negative effects are more pronounced especially due to increasingly large number of ICEs running daily. Today, limitation of fossil fuel reserves and also destructive damages of air pollution are two serious challenges. Significant effort has been made to improve the performance of ICEs and minimize their deficiencies. In any combustion reaction, theoretically there is a required volume of oxygen to react with a given amount of fuel. In a typical combustion process a fuel mixes with air, which contains only 20.9% oxygen. The remaining 79.1% is not required for combustion and detract from the combustion process by having to be heated and causing lower efficiency.

Air/fuel ratio and volumetric efficiency are the two most important parameters affecting engine performance. Richer mixtures require additional fuel and produce higher concentrations of unburned hydrocarbon and carbon monoxide. Running the engine with too much air causes engine misfiring and an increased level of nitrogen oxide emissions. Therefore, the process of adjusting fuel/air mixture inflow of an ICE according to its operating condition is of great importance. As the engine temperature increases, the fuel/air inflow rate must be controlled and cooling rate must be increased in order to protect the engine components from overheating.

Volumetric efficiency - defined as the mass of the fuel and air inducted into the cylinders divided by the mass that would occupy the displaced volume at the fresh charge density – also affects engine performance. For two engines with equal thermal efficiency, the one with higher volumetric efficiency will provide more power. In other words, increasing the volumetric efficiency allows higher overall system efficiency and reduced fuel consumption. Volumetric efficiency is, however, directly proportional to density at the intake manifold, therefore cooling the intake air fuel mixture and delaying the time when fuel evaporates is beneficial to overall engine performance. At lower engine speeds, the air flow rate is slower and the air remains in the intake system for a longer time. Additionally viscous drag increases the temperature with significant loss of efficiency at higher speeds [1].

In another effort to improve the thermal efficiencies of modern automobiles, some exhaust gas is recycled through an exhaust gas recirculation (EGR) valve back into the intake manifold. This reduces the air/fuel ratio and thus the amount of nitrogen that must be heated up to combustion temperatures, but also raises the intake density and reduces the volumetric efficiency. Cooling the intake air temperatures is particularly important when exhaust gas recirculation is employed.

Thus, it is crucial to design a control system that regulates the fuel/air inflow rate and cooling passively. The level of criticality of thermal issues and the inadequacy of the current systems thus call for a novel and sophisticated solution. The present research proposal suggests two innovative combustion thermal management systems, which will result in higher overall efficiency, lower fuel consumption, and reduced air pollution. In this study, we will investigate the applicability of an innovative design consisting of a double-layer expandable thin-film system; recommendations for future experimental investigation are also made.

Thin films have been investigated in numerous studies due to their effectiveness in a wide range of applications including cooling heat transfer. Recent studies demonstrate a number of advanced features of thin films that show substantial expansion/contraction as pressure and temperature are varied. This led to the novel idea of employing thermal expansion to enhance passive thermal management of combustion process. The idea, like some other inventions was mainly inspired by nature. For instance, the organs of the circulatory system such as blood vessels expand/contract on a regular basis to accommodate changes in temperature and pressure. The proposed system mimics the same behavior via thermal expansion of a stagnant fluid contained in a complex elastic sealing assembly. The sealing material must be soft enough to expand and at the same time it should be able to function well in a desired range of operating pressures and temperatures. Several candidates for the sealing material have been identified and will be considered. The proposed system consists of soft seals placed between special guiders as shown in Figure 1. As such, side expansion of the seals is minimized, whereas the transverse thin film thickness is allowed to expand as much as possible.

Unlike ordinary heat transfer assemblies, which are usually designed for a specific thermal load capacity, the proposed expandable systems possess variable capacities and therefore can accommodate variations in the external thermal load passively with no need for external bulky cooling or mechanical controlling devices. These systems will open new avenues for control of heat transfer in a wide variety of applications such as internal combustion and thermal insulation, to name a few. Specific applications include reducing pollution emissions of ICEs with and without catalytic converters and/or fuel injection systems. Example of ICE's without catalytic converters are lawn mowers, dirt bikes, and carbureted automobiles. The results of the proposed research will provide better heat transfer characteristics and improved combustion efficiency, which directly translates into substantial energy savings, and reduced air pollution. A brief description of our proposed systems is presented in what follows.

Double layer expandable system for passive control of flow and thermal exit conditions in combustion processes.

Double-layer thin films were found to possess favorable features as illustrated in the work of Vafai and Zhu [2]. Khaled and Vafai [3] applied the idea of volumetric thermal expansion to double layer thin films to regulate the flow rate as well as exit thermal conditions. In their work, a design for alleviating thermal load and internal pressure fluctuations was established for combustion processes. It consists of three parallel plates between which two separate streams flow. A complex sealing assembly supports the plates. A three-dimensional schematic of the design is illustrated in figure 1.

The lower and the intermediate plates are fixed while the upper plate, which is supported by the soft sealing assembly, is free to move vertically. The lower thin film is named the main layer and it carries a flow of combustion residuals. The upper thin film is named the secondary layer and it carries a flow of liquid fuel. The sealing assembly of the secondary layer also contains closed voids filled with a stagnant fluid having a relatively large volumetric thermal expansion coefficient. As heat is transferred to the stagnant gas compartments in the secondary layer, the stagnant gas expands and the height of the secondary layer increases. This increases the flow rate of fuel to the engine.

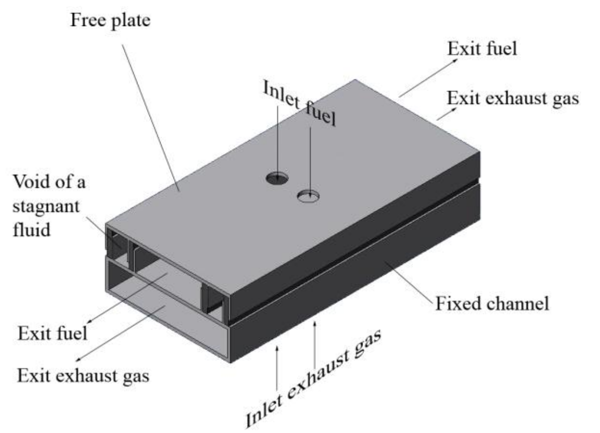

Figure 1. Three-dimensional view of proposed device.

Design Constraints

The goal of this work is to demonstrate a proof-of-concept. It will be shown that the proposed device allows an engine to perform stoichiometric combustion at all steady state engine speeds. An engine with well known operating variables was chosen as a platform to design the proposed device around. The engine that was chosen is a 250 cubic inch, V8 made by American Motor Corporation. The reason this particular engine was selected was due to the availability of data for this particular engine. This engine has an 8:1 compression ratio [7]. A schematic of the engine is shown in figure 2; this figure is also a graph of the engine's power and brake specific fuel consumption versus revolutions per minute.

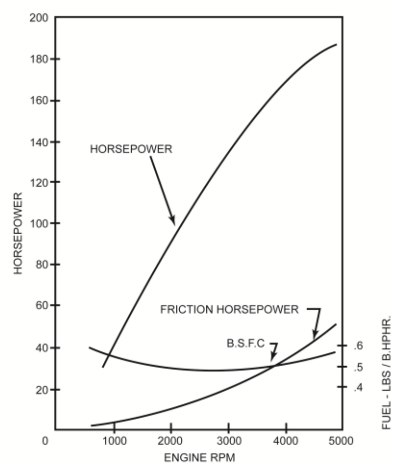

Figure 2. Brake specific fuel consumption and horsepower versus engine speed [Adapted from reference 8]

By manipulating the horsepower and brake specific fuel consumption data, the required volumetric flow rate of fuel is obtained as a function of engine speed. This relationship, shown in figure 3 below, is positive, linear and used as the required output of the proposed device. Multiplying the fuel flow rate data by the stoichiometric air-fuel ratio (accounting for the density of air at the intake manifold) yields the engine's required air flow rate.

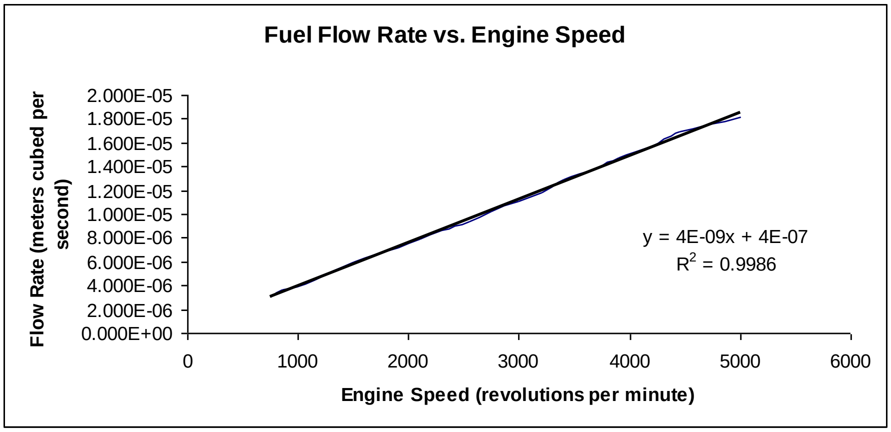

Figure 3. Fuel Flow Rate vs. Engine Speed [Adapted from Reference 8]

The following parameters will need to be designed to work with the given engine: the dimensions of the device, the type of stagnant gas, the type of fuel, the materials in the assembly and the flow rate of combustion residuals.

As stated earlier, the expansion of the proposed device will rely on the temperature of the engine's combustion residuals. Figure 4 is a graph of exhaust gas temperature versus rotational speed of a typical IC engine. The graph shows a positive linear relationship between these two engine variables. Thus an increase in the rotational speed of the engine will increase the exhaust gas temperature, thus increasing the heat transfer through the intermediate plate; which in turn will increase the height of the secondary layer.

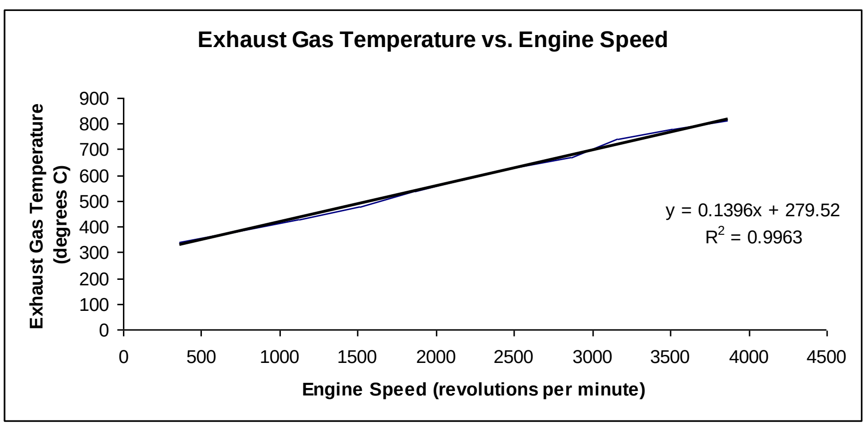

Figure 4. Exhaust gas temperature versus engine speed [5].

Figures 5 and 6 show how stoichiometric combustion is related to production of NOx, hydrocarbons and CO. Also, it is clearly shown that the efficiency of a catalytic converter is at a maximum within a narrow range of air/fuel ratios. Based on these two figures, the proposed device will limit the air/fuel ratio to between 14.6 and 14.7 (assuming air is metered into the engine to proportionately match the fuel flow rate. It is clear that engine using the proposed device will operate more efficiently with significantly reduced polluting emissions.

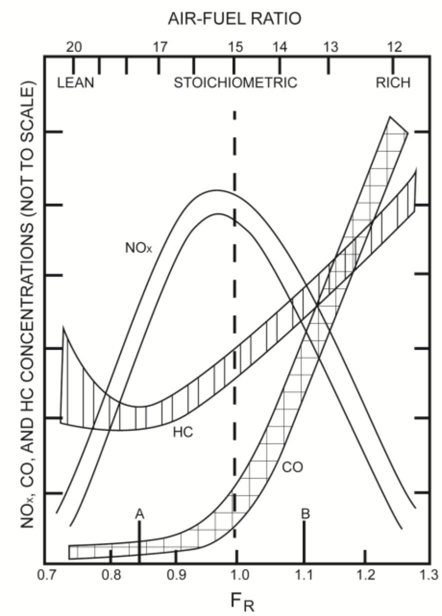

Figure 5. Effect of fuel-air ratio on exhaust emission in spark-ignition engines. A=normal lean limit; B= normal for maximum bmep (torque) [Adapted from reference 6].

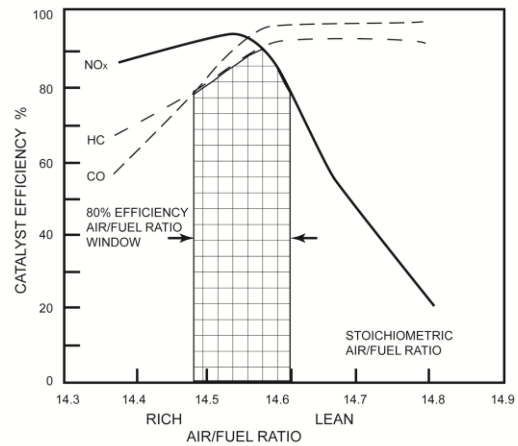

Figure 6. Conversion efficiency for NO, CO, and HC for a three-way catalyst as a function of the air-fuel ratio [Adapted from reference 7]

Methodology

The objective is to study and analyze the performance of a passive thermal and flow control systems. The research draws upon expertise in the fields of fluid dynamics and heat transfer. It consists of several experimental, theoretical, and computational parts, which will advance parallel to one another. Existing results [2-4], obtained from preliminary theoretical investigations, are promising and require further investigation.

Mathematical Formulation

In this work, a numerical analysis was performed for the transport processes in the proposed passive expandable heat transfer systems. The governing equations were solved for various pertinent parameters to determine the optimum fluid flow and heat transfer characteristics for different designs. The numerical simulation was carried out using MATLAB.

Modeling the Double Layer System

Figure 7 is the schematics of the simple model representing the behavior of the secondary layer in the proposed double layer system. The expansion of the stagnant gas is modeled as an isobaric process with respect to air. A balancing weight opposes the average pressure of the secondary flow. The temperature of the main flow is modeled as a uniform temperature at the intermediate plate. As the temperature of the intermediate plate the air undergoes thermal expansion according to the following equation:

$$V_2 = V_1 \frac{T_{s2}}{T_{s1}} \qquad [1]$$

where V_1 and V_2 are the volumes of the stagnant gas before and after the temperature increase of the intermediate plate from T_{s1} to T_{s2}, respectively. However, this analysis does not account for the elastic forces from the seals. A more accurate model of the double layer system, where the pressure of the stagnant gas increases linearly with volume, is shown in figure 8.

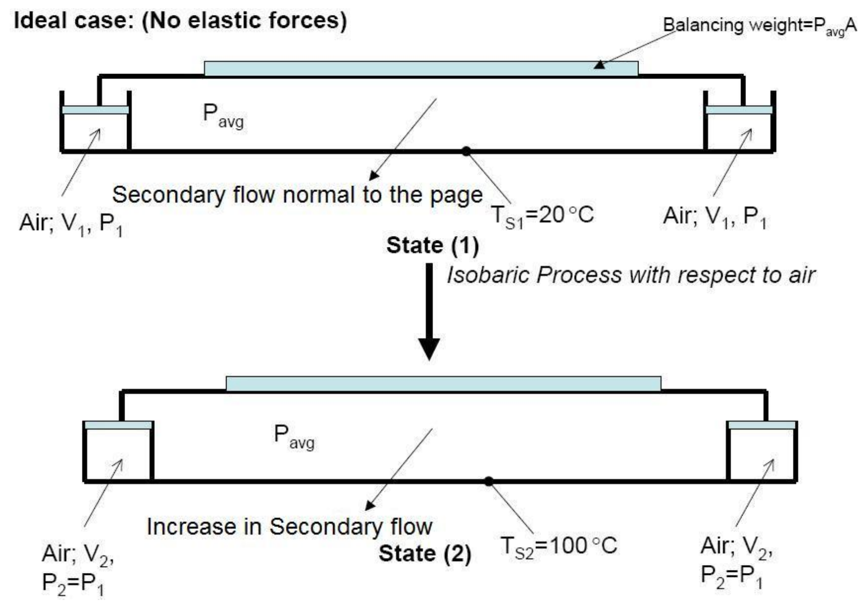

Figure 7. Double Layer System Ideal Gas Expansion.

The change in pressure of the stagnant gas is as follows:

$$(P_2 - P_1)A = KdH \qquad [2]$$

where A is the area of the seals that the pressure acts on (cross sectional area) and is twice the cavity width, w, times the cavity length, L.

$$A = 2wL$$

K is the stiffness coefficient of the seals (cavity wall) and is determined from a dynamics equation:

$$K = \frac{EA}{H} = \frac{E(2.2t)L}{H} = \frac{4EtL}{H} \qquad [3]$$

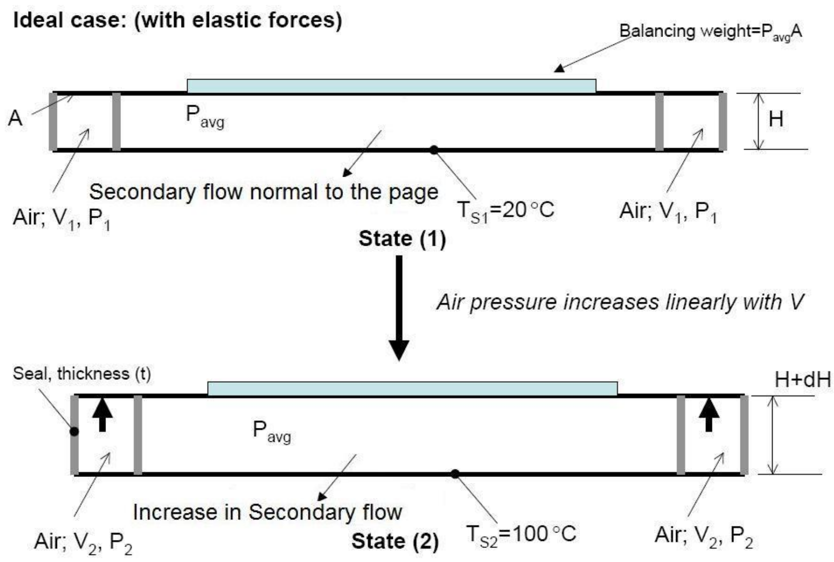

Figure 8. Double Layer System Ideal Gas Expansion with Elastic Forces

Here E is the elastic modulus and t is the thickness of the seals.

$$(P_2 - P_1) = \frac{4EtL}{AH} dH$$
$$(P_2 - P_1) = \frac{4EtL}{(2w * L)H} dH$$
$$(P_2 - P_1) = \frac{2Et}{wH} dH$$

From the ideal gas relationship:

$$(P_2 - P_1) = mR\left(\frac{T_{s2}}{V_2} - \frac{T_{s1}}{V_1}\right) \quad [4]$$

$$(P_2 - P_1) = mR\frac{T_{s1}}{V_1}\left(\frac{T_{s2}/V_2}{T_{s1}/V_1} - 1\right)$$

$$(P_2 - P_1) = mR\frac{T_{s1}}{V_1}\left(\frac{T_{s2}}{T_{s1}\left(\frac{V_2}{V_1}\right)} - 1\right)$$

In these equations, m is the mass of the stagnant gas; R is the universal gas constant for air. Therefore,

$$(P_2 - P_1) = P_1\left(\frac{T_{s2}}{T_{s1}\left(1+\left(\frac{dH}{H}\right)\right)} - 1\right) \qquad [5]$$

Thus,

$$\left(\frac{2Et}{w}\right)\left(\frac{dH}{H}\right) = P_1\left(\frac{T_{s2}}{T_{s1}\left(1+\left(\frac{dH}{H}\right)\right)} - 1\right) \qquad [6]$$

If we use the values provided in figure 8 then we can see the behavior of the model as a function of elastic modulus, E:

Table 1. Elastic Modulus of Seals affect on System Behavior

E (Pa)	dH/H
0	0.27
1.00E+04	0.266
1.00E+05	0.232
1.00E+06	0.189

The temperature of the intermediate plate was determined earlier in this work as given in Fig.4 as a function of engine speed:

$$T_s = .1396(RPM) + 279.52 \qquad [7]$$

Where T_s is in degrees Celsius and RPM, the speed of the engine, is in revolutions per minute. Plugging equation [7] into equation [6] we get an equation for the secondary layer height as a function of engine speed. This resulting equation can then be used to solve for the average velocity of the secondary fluid via the following relationship [10]:

$$\dot{V} = VwH = \frac{H^3 w}{12\mu}\left(\frac{\rho V^2}{2} \cdot \frac{f}{2H}\right) \qquad [8]$$

Where V is the average velocity in the secondary layer, ρ is the density of gasoline, μ is the dynamic viscosity of gasoline and f is the friction factor which is taken from figure 9:

The model of the proposed system must be constrained in the following way: fuel consumption as a function of engine speed must follow the relationship determined earlier in this paper. For the V-8 engine that was selected, volumetric flow rate of fuel is as follows:

$$\dot{V} = \overline{w}wH = 4x10^{9}(RPM) + 4x10^{-7} \qquad [9]$$

Where $\dot{V}$ is in meters cubed per second.

Equations 4 and 6-9 are used to solve for the unknowns of the system at the expanded state, i.e., the height of the system, H_2, the pressure P_2, and modulus of elasticity, E. Details of this operation are given below assuming intermediate plate remains as the same temperature as the exhaust gases.

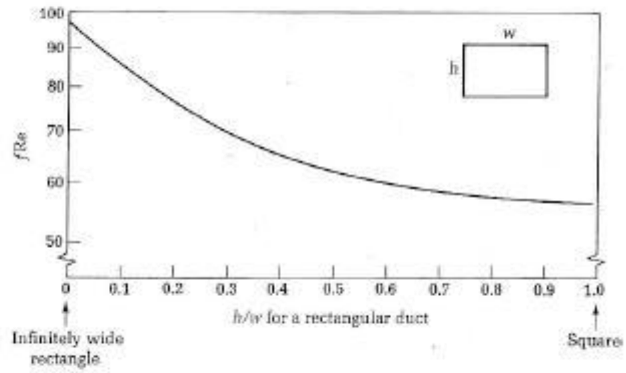

Figure 9. Friction factor for laminar flow in a rectangular duct [11]

Numerical procedure

The procedure for the numerical solution is summarized as follows:

1. First, assume an idling engine speed.
2. Using equation Eq. (9) and (7) find the actual flow rate of fuel and exhaust gas temperature of the engine.
3. Plug a range of H values into Eq. (8) and compare the predicted flow rate to the actual flow rate. An acceptable H value is one that satisfies
   $$\left|\frac{H_{predicted} - H_{actual}}{H_{actual}}\right| < 10^{-2}$$
4. Using H, the volume of stagnant gas V_2, the change in height of the secondary layer dH and the quantity $\frac{dH}{H}$ are determined.
5. Eq. (4) then yields the pressure of the stagnant gas at the new temperature and volume.
6. Eq. (6) then yields the elastic modulus E.
7. Repeat steps (2)-(6) to find the operating conditions of the proposed device at all other engine speeds.

Results

Fuel metering performance

Figure 10 shows a plot of the height of the secondary channel versus the speed of the engine. The width of the channel was fixed at 6.31 cm; this provided the lowest error between the known demands of the engine for fuel and the equations describing flow in a rectangular channel. The height of the channel is shown to increase linearly with increasing engine speed. An equation describing the height of the channel as a function of engine speed is displayed next to the trend line in the figure.

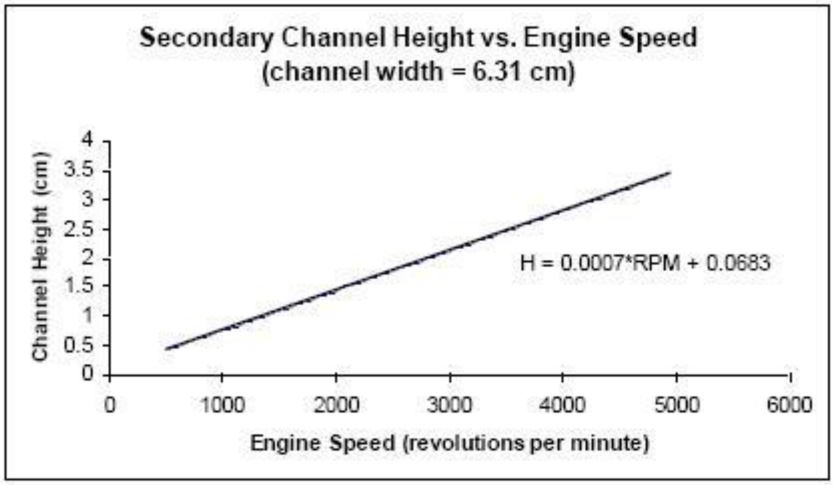

Figure 10. Secondary layer height versus engine speed; channel width fixed at 6.31 cm

Using equation 6 and the results from figure 10, the pressure of the stagnant gas in the secondary layer as a function of engine speed was determined. Figure 11 shows a plot of the stagnant gas pressure versus engine speed. The three trend lines show the effect that the elastic modulus of the seals has on the pressure of the stagnant gas. As the elastic modulus of the seals increases, the pressure of the stagnant gas increases for all values of engine speed.

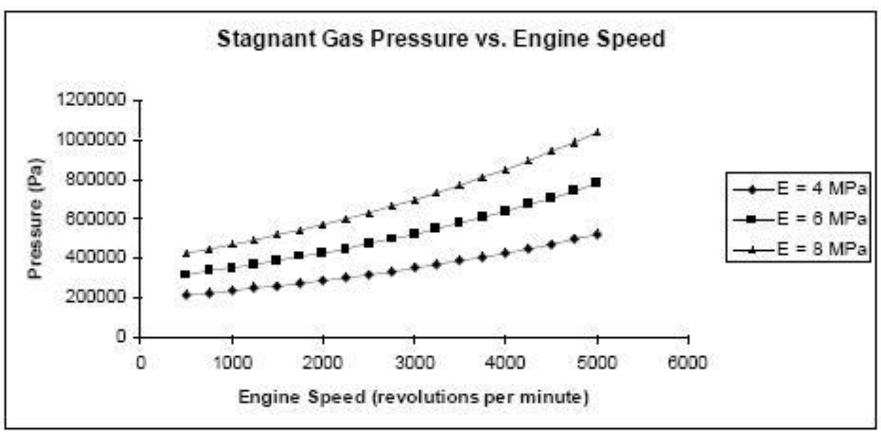

Figure 11. Effect of seal elastic modulus on the expansion of the stagnant gas.

Device Dimensions and Operating Variables

Based on the results described above, all of the pertinent device dimensions and operating variables have been specified in Table 2. The width, length and height dimensions of the device are no larger than the dimensions of other common devices that fit under the hood of a conventional automobile. Also, the elastic modulus of the seals is in the range of many common sealing materials.

Table 2. Device Dimensions and Operating Variables

elastic modulus of the seals	4 MPa
overall width	11.3 cm
overall length	5 cm
secondary flow width	6.3 cm
height range of secondary layer	0.41-3.44 cm
cold start height of secondary layer	0.1 cm
pressure range of stagnant gas	0.213-0.52 MPa
cold start pressure of stagnant gas	276 kPa
temperature range of stagnant gas	419-977^oC
cold start temperature of stagnant gas	20^oC

The result presented above shows a very stringent requirement on the seal as it must accommodate expansion in the wide range 0.41-3.44 cm. A possible solution is using flexible diaphragms, or at smaller temperature ranges of operation.

Sealing Materials

A sealing material must be selected that will not fail when in contact with very high temperatures of the exhaust gas (via the intermediate plate). From the results, the elastic modulus of the seal is in the range 4 MPa and 349 – 964 degrees Celsius, respectively. Table 3 below lists some common sealing materials and maximum service temperature.

Table 3. 100% modulus and maximum service temperature of sealing materials [10]

Material Name	100% Modulus (Gpa)	Maximum Service Temperature, Air (^oC)
Silica Aerogel	.00100 - .0100	500
Thermoset Fluoroelastomer	.000600 - .140	157 - 330
Thermoset Polyurethane Foam	.000138 - 2.00	70 - 121
Thermoset Polyurethane, Elastomer	.000241 - .276	82 - 100
Chlorosulfonated Polyethylene Rubber	.00100 - .0189	125
Silicone, RTV, Adhesive/Sealant Grade	.000621 - .00459	117 - 316
Silicone Rubber	.000896 - .00585	200 - 260
Du Pont Elastomers Kalrez FFKM Perfluoroelastomer	0.00720	316
Greene Tweed CHEMRAZ Perfluroelastomer	0.00276-0.00689	324
Precision Polymer Engineering Perlast Perfluoroelastomer	0.00370-0.0074	310

A good choice of sealing material for the proposed device is silica aerogel since it works well with the parameters of the proposed device and can accommodate temperatures close to the exhaust temperature. The Aerogel can be made hydrophobic by means of fluorinating the surface of the material during the manufacturing process, therefore suggesting that liquids can be used as an expandable fluid but great care must be taken into consideration in deciding which fluid would work with this particular application especially when high heat and potentially high pressures are present in the system. A more suitable sealing material is Grafoil® which is made by GrafTech International. Grafoil® is a gasketing material essentially made of about 95% graphite which is self lubricating and resists temperatures of up to 850 degrees Fahrenheit in oxidizing atmospheres.

Fuel Controller

Schematics of two different designs of the fuel controllers are shown in the Figures 12 and 13 below. In proposed design shown in Figure 12, the fuel controller operates by means of a vertically shearing gate valve. Exhaust gases from the engine passes through the exhaust chamber of the controller and gives its heat through the chamber wall into a working fluid. The working fluid is housed in a chamber which is capped off with a piston that actuates according to the change in volume of the fluid in the working fluid chamber. The sides of the piston that come into contact with the piston housing assembly will be lined with a 0.003" thick Graffoil gasket material to provide a good self-lubricating non-expanding seal that will resist high temperatures.

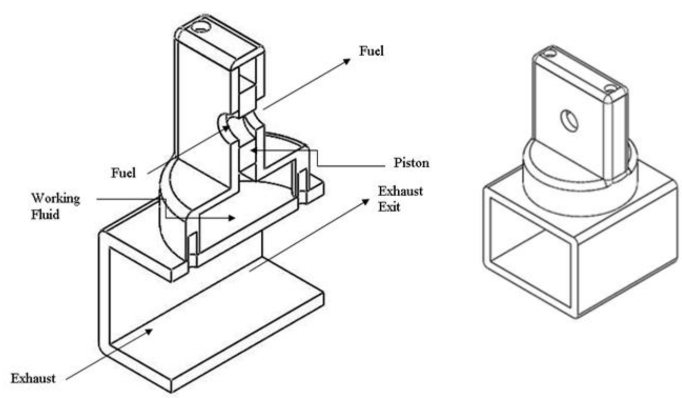

Figure 9. Fuel controller using piston assembly

An alternate design which incorporates the use of an expandable seal can also be adapted for use as a fuel controller (See Figure 13). In comparison to the previous non-expandable sealed design, it will work in a similar fashion but instead of an internal sliding piston to regulate fluids, the fuel controller housing will act as an external piston while the internal slug is statically fixed to the exhaust chamber. The cylindrical flexible seal will be in place of the expandable fluid chambers cylinder wall. Ideally, lead could still be used as an expandable fluid to actuate the fuel regulation valve while the flexible seal will elastically stretch to contain the molten metal

Ultimately, the ideal application for thermal-mechanical devices would be used for less extreme temperatures. Though the fuel controller was designed to regulate fuel flow, this technology can be adapted for use for other applications such as bleeder valves for high pressure fluid or gas tanks that are exposed to heat causing over-pressurization. Water can be used as an expandable fluid for lower temperature applications ranging from 4^{o}C to nearly100^{o}C, but if a higher operational temperature is desired, molten salts will also work for temperatures reaching upwards of 800^{o}C.

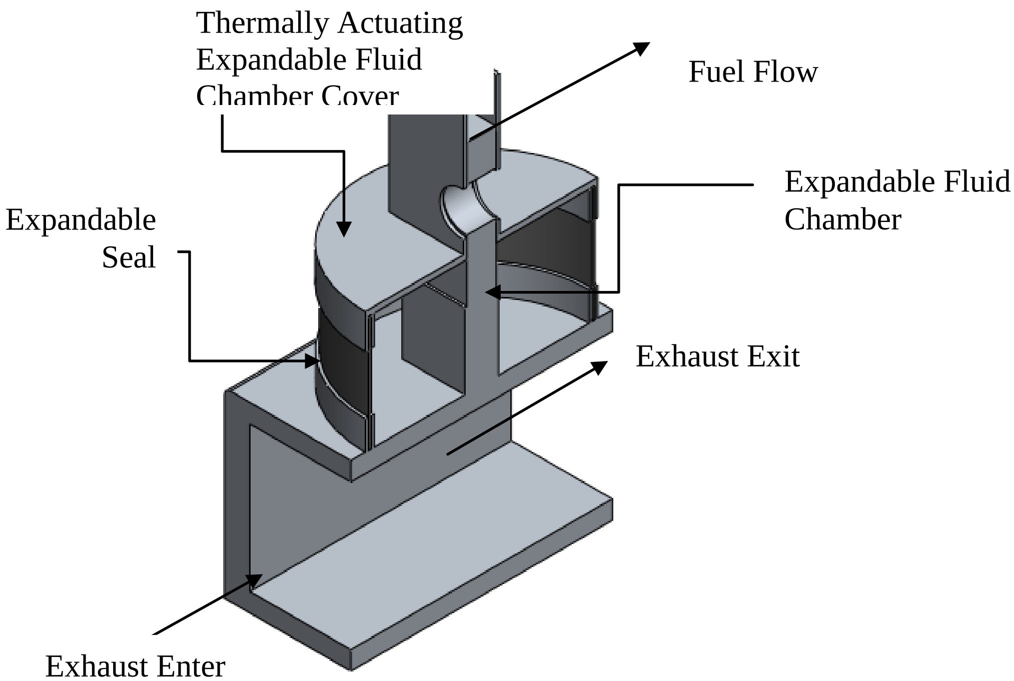

Figure 10. Fuel controller using expandable seal

Feasibility and Conclusion

The proposed device meets the fuel flow demands of the engine and it can fit under the hood of a conventional automobile. The sealing material that will be used is silica aerogel which is well suited for the high operating temperatures. Air is used as the stagnant gas and it expands to a maximum pressure of about 5 times atmospheric. Static friction between the seals and sliding parts should be taken into consideration when using air as an expandable fluid when attempting to manufacture the fuel regulator. If static friction is present in the unit, the device would not operate smoothly and may not be able to regulate the fuel/ air ratio as efficiently as intended. Taking these factors into consideration the proposed device will feasibly operate in an ICE.

Specific applications include reducing pollution emissions of ICEs with and without catalytic converters and/or fuel injection systems. Example of ICE’s without catalytic converters are lawn mowers, dirt bikes, and carbureted automobiles. Since the proposed device operates passively, it is ideally suited for carbureted ICEs or ICE applications where it is not feasible to use fuel injection. The proposed device would meter fuel into the engine much like a fuel injection system and it would avoid all of the common problems associated with carburetors. If a lawnmower had improved fuel metering then it could potentially have reduced emissions. This could be especially important for lawnmowers used around school yards where many children's lungs are susceptible to pollution.

Two different designs for the fuel controller are presented. The high temperature demand makes the expandable seal design more challenging. The device should work for low temperature applications when temperature differences do not exceed 200^{o}C.

References

[1] Taylor, C. F., "The Internal Combustion Engine in Theory and Practice, Vol. 1 and 2, The MIT Press, 1977., pp. 149-187.

[2] Vafai, K., and Zhu, L., "Analysis of a Two-Layered Micro Channel Heat Sink Concept in Electronic Cooling," *International Journal of Heat Transfer and Mass Transfer*, 42, pp. 2287-2297, 1999.

[3] Khaled A. –R. A. and Vafai K., "Control of Exit Flow and Thermal Conditions using Two-layered Thin Films Supported by Flexible Complex Seals," *International Journal of Heat and Mass Transfer*, 47, 1599-1611, 2004.

[4] Khaled, A. –R. A., and Vafai, K., "Cooling Enhancements in Thin Films Supported by Flexible Complex Soft Seals In the Presence of Ultra-fine Suspensions", *ASME Journal of Heat Transfer*, 125, pp. 916-925, 2003.

[5] Willard W. Pulkrabek, Engineering Fundamentals of the Internal Combustion Engine, Second Edition. Prentice Hall, 2004.

[6] Charles Fayette Taylor, The Internal Combustion Engine in Theory and Practice: Fuels, Materials, Design, Volume 2: Combustion, Revised Edition, The MIT Press. 1985.

[7] Eran Sher, Handbook of Air Pollution from Internal Combustion Engines: Pollutant Formation and Control, Academic Press. 1998.

[8] The New American Motors V-8. John Adamson et al. American Motors Corporation. SAE National Passenger-Car, Body and Materials Meeting, 1956.

[9] Ferguson and Kirkpatrick. Internal Combustion Engines: Applied Thermosciences, Second Edition. John Wiley & Sons, Inc. 2001.

[10] MatWeb. Material Property Data, (www.matweb.com).

[11] William S Janna, Introduction to Fluid Mechanics, 3rd edition.. PWS-KENT Publishing, 1993.

[12] Fluid Properties. The fluid flow calculations website. LMNO Engineering, Research, and Software, Ltd (www.lmnoeng.com).

Appendix A: Proposed Experimental Investigation

A schematic of the experimental setup for studying the feasibility of proposed expandable system is shown in Figure 14. It should be noted that the required set-up demands substantially more resources and is outside the scope of the tasks enumerated in the original proposal and therefore must be carried out in the future.[1] The setup is designed for measurements as well as enabling a wide range of adjustments in the relevant parameters, such as heat load, inlet temperature, and pressure. The purpose of the experiments is to assess the flow rate control of the proposed system. The flow of fuel is simulated in the experiment as the flow of water. A variable capacity heater provides the thermal load to the water (secondary flow) as it passes through the test section. The heat load applied by the heater simulates the flow of combustibles (main flow) through the proposed device. The stagnant gas in the test section will be helium. As the heat load is increased the flow rate of water will be observed and recorded at a fixed inlet pressure and temperature.

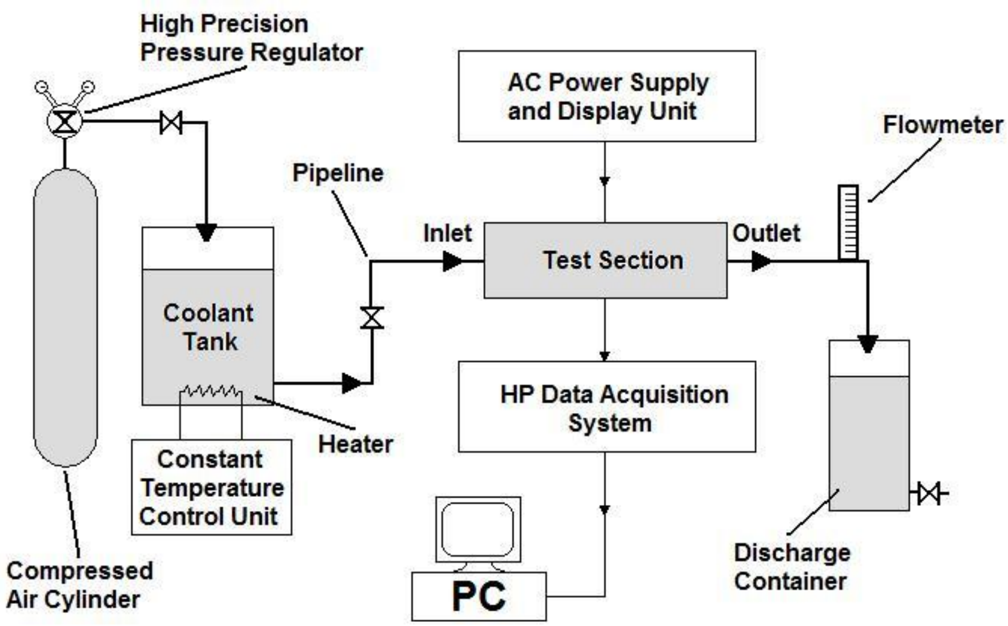

Figure 11. Schematic of Experimental Setup

The flow of the working fluid is driven by a constant pressure source. A compressed air tank pressurizes the tank containing the working fluid. The pressure is well controlled via a high precision pressure regulator. A constant temperature control unit keeps the temperature of the coolant inside the tank constant. The fluid is driven from the tank into the test-section. A valve is placed between the coolant tank and the test section. The pipeline and the valve connecting the coolant tank and the test section must be well insulated. A number of thermocouple wires are arranged in the system. A variable capacity heater provides the thermal load. A power supply unit, consisting of an AC voltage stabilizer, a voltage transformer, and a power meter with a known uncertainty,

[1] We requested a modified SOW that allowed limited experimentation. Since the modified budget required some diversion of funds for purchase of equipments and parts, the request was denied.

supplies the power. The test section is well insulated to minimize the heat transfer to the surroundings. Nevertheless, in order to collect accurate data comparable to the analytical results, the amount of heat transfer with the surroundings must be determined through a comparison between the total heating power input and the net enthalpy change across the test sections. The inlet and outlet temperatures of the coolant will be measured by high precision jacket thermocouples with a known uncertainty. The inlet/outlet pressures and the mass flow of the coolant are measured using accurate pressure gage and flow meter with known uncertainties. Since the system undergoes a variable heat load and is designed in such a way to passively increase the mass flow rate of the coolant to account for the increase in thermal load, the response times of the thermocouples, pressure gage, and the flow meter are of great importance. All temperatures, pressures, and mass flow rates are collected using an HP high-speed data acquisition system. The data is then stored, processed and displayed on a PC. An experimentation procedure must be developed to ensure repeatability of the results.

Appendix B: MATLAB Code

```
clear;
clc;
%Assumptions: The temperature of the intermediate plate is that of the
%combustion residuals
%PROPERTIES OF THE SECONDARY FLUID (GASOLINE)
mu = .0003128; %dynamic viscosity of gasoline kg/(m*s) or Pa*s
rho = 680; %(kg/m^3)density of the gasoline
stepsize = .0001;
w=.063;
e=1;
RPM(1) = 500;
i=1;
while RPM(i) < 5000 %loop 3
Vdot(i) = 4E-9*RPM(i)+4E-7; %initial volumetric flow rate
errorcheck(e,1)=1E100;
k=1;
h(1) = .001;
pass2=0;
while pass2==0%loop 1
A(i,k) = h(k)/w;
velactual(i,k) = Vdot(i)/(w*h(k));
Dh(i,k) = (4*h(k)*w)/(2*h(k)+2*w);
Re(i,k) = rho*velactual(i,k)*Dh(i,k)/mu;
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%Re
if Re(i,k)<2300
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%A
if A(i,k)<=1 & A(i,k)>=0
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%A
if A(i,k)>=0 | A(i,k)<=.05
fRe(i,k) = 96+(89.91-96)*(A(i,k))/(.05);
end
if A(i,k)>.05| A(i,k)<=.1
fRe(i,k) = 89.91 +(84.68-89.91)*(A(i,k)-
.05)/(.1-.05);
```

```
end
if A(i,k)>.1| A(i,k)<=.125
fRe(i,k) = 84.68 +(82.34-84.68)*(A(i,k)-
.1)/(.125-.1);
end
if A(i,k)>.125| A(i,k)<=.167
fRe(i,k) = 82.34 +(78.81-82.34)*(A(i,k)-
.125)/(.167-.125);
end
if A(i,k)>.167| A(i,k)<=.25
fRe(i,k) = 78.81 +(72.93-78.81)*(A(i,k)-
.167)/(.25-.167);
end
if A(i,k)>.25| A(i,k)<=.4
fRe(i,k) = 72.93 +(65.47-72.93)*(A(i,k)-
.25)/(.4-.25);
end
if A(i,k)>.4| A(i,k)<=.5
fRe(i,k) = 65.47 +(62.19-65.47)*(A(i,k)-
.4)/(.5-.4);
end
if A(i,k)>.5| A(i,k)<=.75
fRe(i,k) = 62.19 +(57.89-62.19)*(A(i,k)-
.5)/(.75-.5);
end
if A(i,k)>.75| A(i,k)<=1
fRe(i,k) = 57.89 +(56.91-57.89)*(A(i,k)-
.75)/(1-.75);
end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%A
f(i,k) = Re(i,k)*fRe(i,k);
%use the bisection method to find velpredict
Va=10000000;
Vb = -10000000;
error2=100;
while abs(Va-Vb)>.01
Vc = (Va+Vb)/2;
%evaluate the function
fa=((h(k)*rho*(Va^2)*f(i,k))/(48*mu))-Va;
fc=((h(k)*rho*(Vc^2)*f(i,k))/(48*mu))-Vc;
if fa*fc >0
Va=Vc;
else
Vb=Vc;
end
end
velpredict(i,k) = Vc;
error(i,k) = 100*abs(velpredict(i,k)-
velactual(i,k))/velactual(i,k);
%find w and h that have the lowest error for a
%given RPM
if error(i,k) < errorcheck(e,1)
errorcheck(e,1) = error(i,k);
errorcheck(e,2) = w;
errorcheck(e,3) = h(k);
errorcheck(e,4) = RPM(i);
```

```
errorcheck(e,5) = velpredict(i,k);
errorcheck(e,6) = velactual(i,k);
end
end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%A
end
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%Re<2300
if h(k)<.07
h(k+1)=h(k)+stepsize;
k=k+1;
else
pass2=1;
end
end%end loop 1
if RPM(i)<4950
RPM(i+1)=RPM(i)+50;
i=i+1;
e=e+1;
else
RPM(i)=5000;
end
end%end loop 3
figure(1)
plot(RPM,errorcheck(:,5),'r')
hold on
plot(RPM,errorcheck(:,6))
figure(2)
plot(RPM,errorcheck(:,3))
xlabel('Engine Speed (revolutions per minute)')
ylabel('Secondary Channel Height (m)')
title('Secondary Channel Height vs. Engine Speed; Fixed Width')
%determine the elastic modulus of the seals that agrees with the
%temperature and geometry of the device
g=1; %a counting variable
t=.001;%thickness of the seals in meters
L = .05; %length of the device in meters
wair = .05; %combined width of the stagnant gas channels in meters
R = 286.9;%ideal gas constant J/kgK
Ho = errorcheck(1,3);%initial height of the secondary layer in meters
Vair_o = L*wair*Ho; %volume of stagnant gas [meters cubed]
Pair_o = 101000; %intial pressure of air in the device in Pa
Tair_o = 273.15+20; %assumed temperature of the air in the device in K
when the engine is off
mair = Pair_o*Vair_o/(R*Tair_o);%total mass of the stagnant gas in kg,
remember Tair is in K
E = 8E6; %elastic modulus of the seals in Pa
while g<=numel(errorcheck(:,1))-1
RPM(g) = errorcheck(g,4);
Texh(g) = .1396*RPM(g)+279.52+273.15;%exhaust temperature in Kelvin
this is assumed to be the temperature of the stagnant gas at each RPM
value
H1(g)=errorcheck(g,3);
H2(g) = errorcheck(g+1,3);
dHbyH(g) = (H2(g)-H1(g))/H2(g);
V1(g) = H1(g)*wair*L; %volume of air before the change in
```

```
temperature
V2(g) = H2(g)*wair*L; %volume of air after the change in
temperature
if g==1
P1(g)=E/((((Texh(g)/Tair_o)/(1+dHbyH(g)))-
1)*(wair)/(dHbyH(g)*t*2));
else
P1(g)=-E/(wair*(((Texh(g)/Texh(g-1))*(1/(1+dHbyH(g))))-
1)/(dHbyH(g)*t*2));
end
if g<=numel(errorcheck(:,1))-1
g=g+1;
end
end
plot(RPM,P1)
```