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Method for Turbocharging Single Cylinder Four Stroke Engines

DETC2014-35044

DRAFT: METHOD FOR TURBOCHARGING SINGLE CYLINDER FOUR STROKE ENGINES

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ABSTRACT

This paper presents a feasibility study of a method for turbocharging single cylinder, four-stroke internal combustion engines. Turbocharging is not conventionally used with single cylinder engines because of the timing mismatch between when the turbo is powered, during the exhaust stroke, and when it can deliver air to the cylinder, during the intake stroke. The proposed solution involves an air capacitor on the intake side of the engine between the compressor and the intake valves. The capacitor acts as a buffer and would be implemented as a new style of intake manifold with a larger volume than traditional systems.

In order for the air capacitor to be practical, it needs to be sized large enough to maintain turbo pressure, cause minimal turbo lag and significantly increase the density of intake air. By creating multiple flow models of air through the turbocharged engine system, we found that the optimal size air capacitor is between four and five times the engine capacity. For a capacitor sized for a one-liter engine, the lag time was found to be approximately two seconds, which would be acceptable for slowly accelerating applications such as tractors, or steady state applications such as generators. The density increase that can be achieved in the capacitor, compared to air at standard ambient temperature and pressure, was found to vary between fifty percent for adiabatic compression and no heat transfer from the capacitor, to eighty percent for perfect heat transfer. These increases in density are proportional to, to first order, the anticipated power increases that could be realized with a turbocharger and air capacitor system applied to a single cylinder, four-stroke engine.

NOMENCLATURE

A   Cross sectional area of connecting tube
D   Diameter of connecting tube
F   Friction factor
K   Minor losses
L   Length of connecting tube
m_c Mass of gas inside the capacitor
$\dot{m}_c$ Mass flow rate of gas into the capacitor
$\dot{m}_t$ Mass flow rate of gas at turbocharger
P_C Pressure inside the capacitor
P_e Pressure in the engine at the end of the intake stroke
P_t Pressure at the turbocharger
P_0 Initial pressure inside the capacitor
$\dot{P}_C$ Rate of pressure change inside the capacitor
R   Specific gas constant
T_C Temperature inside the capacitor
T_t Temperature at the turbocharger
V_C Volume of the capacitor
V_e Volume of the engine
v_s Velocity of air in connecting tube
γ Heat capacity ratio (1.4 for air)
ρ_C Density of air inside the capacitor
ρ_t Density of air at the turbocharger

INTRODUCTION

A system to turbocharge single cylinder engines would be useful for a number of reasons. Adding a turbocharger to an

engine increases its specific power and requires less additional mass than adding more cylinders. The primary application for the technology provided in this paper is low-cost agricultural equipment in developing world economies. A presentation to the Indian Parliament by the Indian department of agriculture presented a direct positive correlation between farm power (in kw/Ha) and food yield (in tons/Ha) [1]. This presentation also cites that small scale farmers, which make up approximately eighty four percent of Indian farmers, cannot afford mechanization due to economies of scale. This presentation goes on to conclude that there is a need for development of technologies for these small scale farmers. Turbocharging small tractors will allow lower weight, lower cost machines to be made, which could be affordable to small-scale farmers who could not afford a conventional tractor.

In addition to farm equipment, turbocharged single cylinder engines could be used on generators, motorcycles, and anything else that needs a small engine. Another benefit of this system is increased fuel efficiency. Turbocharging four stroke diesel engines has been found to decrease mechanical losses by between eight and forty percent depending on the application [2]. The reduction in mechanical losses and the resulting fuel efficiency boost comes from the fact that the cylinder in a smaller engine has less contact area and as a result less frictional loss than a naturally aspirated engine with the same power output [3, 4]. This means that turbocharging could reduce the carbon footprint of machines in the developed world while also increasing the access to powerful machines in the developing world.

A turbocharger uses pressurized exhaust gas from the engine to spin a turbine, which is connected to a compressor [5, 6]. The compressor pressurizes the ambient air, increasing its density. The dense air goes into the engine cylinder during the intake stroke. Since there is more oxygen as a result of the increased density of air in the cylinder, the engine can combust more fuel. This means that the engine gets more power out of every stroke. Turbocharged engines are traditionally multi-cylinder because they can be timed in such a way that when one cylinder is exhausting, and thus powering the turbo, another cylinder is in the intake process [5, 6]. In a single cylinder engine the exhaust and intake strokes are out of phase. A previous experimental study showed that the mass flow output a turbocharger under pulsating inlet conditions is peaky and that it is influenced by many variables including frequency and pulse shape [7]. We propose to turbocharge single cylinder engines by adding a volume between the turbocharger and the engine intake that would act as a buffer and smooth out the peaky nature of a turbocharger operating under pulsing inlet conditions.

A block diagram of this system is shown in Fig. 1 The scope of the project involves using an off-the-shelf small turbocharger and adapting it to work with a single cylinder engine. This involves designing an appropriate intake manifold. We will refer to this large volume intake manifold as the air capacitor. The
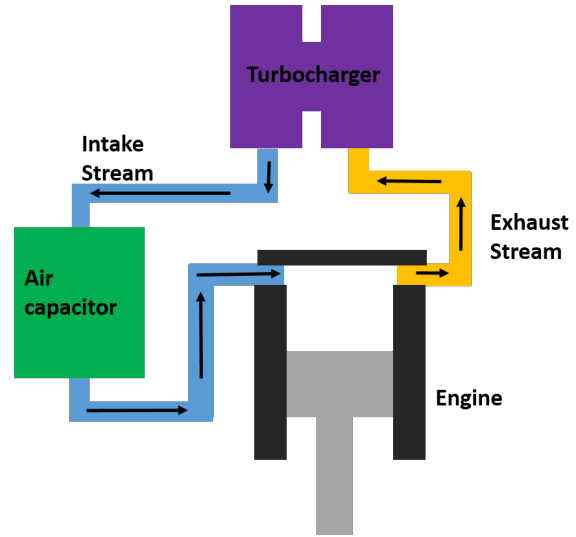

FIGURE 1. BLOCK DIAGRAM OF ENGINE FLOW WITH AIR CAPACITOR SYSTEM

shape and size of the air capacitor is important [8]. It needs to have an appropriate volume; a volume too large will cause excess turbo lag due to the pressurization time of the capacitor, where as a volume too small will induce a large pressure drop during the intake stroke, negating the benefits of the turbo. The shape is important to minimize pipe losses and resonances in the system. In this paper we analyze the theoretical feasibility of this system, focusing on fill time of the capacitor, the optimal volume of the capacitor, the density gain that can be achieved by this system, and thermal effects due to adiabatic compression of air.

AIR CAPACITOR SIZING

The goal of this analysis was to find the fill time and appropriate size for the air capacitor in order to asses the feasibility of the concept. The air capacitor has to be reasonably sized and cannot take too long to pressurize. The most basic model of the system is shown in Fig. 2. The model involves treating the turbocharger as a constant pressure source that is filling the capacitor through a connection which has frictional loss. The volume has to be as small as possible to minimize cost, incorporated easily into the engine, and minimize turbo lag. But it has to be large enough not to experience a significant pressure drop when the engine intakes air. Our target is to design the system such that there is no more then a twenty five percent pressure drop in the capacitor during the intake stroke when it is running in steady state. Modeling the system as isotropic expansion we can find what the capacitor volume should be relative to the engine volume; this is shown in Eqns. 1-6 [9].
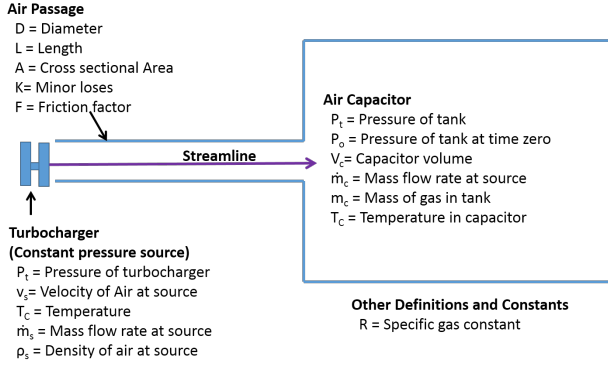
Air Passage
D = Diameter
L = Length
A = Cross sectional Area
K = Minor loses
F = Friction factor

Streamline

Air Capacitor
P_t = Pressure of tank
P_o = Pressure of tank at time zero
V_c = Capacitor volume
$\dot{m}_c$ = Mass flow rate at source
m_c = Mass of gas in tank
T_c = Temperature in capacitor

**Turbocharger
(Constant pressure source)**
P_t = Pressure of turbocharger
v_s = Velocity of Air at source
T_c = Temperature
$\dot{m}_s$ = Mass flow rate at source
ρ_s = Density of air at source

Other Definitions and Constants
R = Specific gas constant

FIGURE 2. DIAGRAM OF CONSTANT PRESSURE SOURCE SYSTEM

$$P_C V_C^{\gamma} = (0.75P_C)(V_C + V_e)^{\gamma} \tag{1}$$

$$V_C^{1.4} = 0.75(V_C + V_e)^{1.4} \tag{2}$$

$$\frac{V_C}{V_C + V_e} = .814 \tag{3}$$

$$1.228V_C = V_C + V_e \tag{4}$$

$$.228V_C = V_e \tag{5}$$

$$V_C = 4.4V_e \tag{6}$$

$$\frac{P_t - P_c}{\rho_t} = \frac{v_s^2}{2} + \frac{FL}{D}\frac{v_s^2}{2} + k\frac{v_s^2}{2} \tag{7}$$

$$\dot{m}_s = \rho_t v_s A \tag{8}$$

$$\dot{m}_C = \frac{\dot{P}_C V_C}{RT_C} \tag{9}$$

$$\dot{m}_C = \dot{m}_S \tag{10}$$

$$\rho_t v_s A = \frac{\dot{P}_C V_C}{RT_C} \tag{11}$$

$$v_s = \frac{\dot{P}_C V_C}{\rho_t A R T_C} \tag{12}$$

$$\frac{P_t - P_c}{\rho_t} = \left(\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2}\right)v_s^2 \tag{13}$$

$$\frac{P_t - P_c}{\rho_t} = \left(\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2}\right)\left(\frac{\dot{P}_C^2 V_C^2}{\rho_t^2 A^2 R^2 T_C^2}\right) \tag{14}$$

$$P_t - P_c = \left(\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2}\right)\left(\frac{\dot{P}_C^2 V_C^2}{\rho_t A^2 R^2 T_C^2}\right) \tag{15}$$

$$C = \frac{\rho_t A^2 R^2 T_C^2}{\left(\frac{1}{2} + \frac{FL}{2D} + \frac{k}{2}\right)V_C^2} \tag{16}$$

$$P_t - P_c = \frac{1}{C}\dot{P}_C^2 \tag{17}$$

$$\dot{P}_C^2 + CP_C - CP_t = 0 \tag{18}$$

Equation 1 is the equation for isotropic expansion of gas where the product of pressure and volume raised to the heat capacity ratio is constant. The left hand side of this equation represents the state before the intake stroke where the pressure is the pressure of the capacitor and the volume is the volume of the air capacitor. The right hand side of 1 represents the state at the end of the intake stroke where the pressure is seventy five percent of the initial capacitor pressure and the volume is equal to the combined volume of the capacitor and the engine. Equations 2-6 solve Eqn. 1 step by step in order to find the capacitor volume as a function of engine volume. The analysis shows that the capacitor volume should be a bit under four and a half times the engine volume. This means that for a 0.8 liter engine the volume would be approximately 3.5 liters.

Another important parameter is to figure out the time it takes to pressurize the capacitor. Using conservation of mass, the ideal gas law, and the Bernoulli equation along the streamline that runs from the pressure source to the center of the air capacitor where the velocity of air is zero, we are able to derive a nonlinear first order partial differential equation that describes how the capacitor gets pressurized. Note that it is assumed that the flow is steady and incompressible [5, 10]. The derivation is shown in Eqns. 7-18.

The basis for this derivation is Eqn. 7 which is a simplified version of the Bernoulli equation along the streamline shown in Fig. 2, which accounts for pipe losses. Equation 8 is the equation for mass flow from the pressure source as a function of density, velocity, and the cross sectional area of the connecting tube. Equation 9 is the equation for mass flow into the air capacitor as defined by the ideal gas law. Equation 10 shows that due to mass conservation the mass flow rate out of the constant pressure source will be equal to the mass flow rate into the air capacitor. Equation 11 substitutes Eqns. 8 and 9 into Eqn. 10. Equation 12 rearranges equation 11 in order to isolate the velocity term. Equation 13 rearranges Eqn. 7. Equation 14 combines Eqns. 12 and 13 to eliminate the velocity term. Equation 15 simplifies Eqn. 14. In Eqn. 16 we define a variable C that includes the friction factor and velocity, which vary with the pressure differential. However, the variations in these factors are small enough that we can treat C as a constant. In Eqn. 17 we substitute the constant C into Eqn. 15. Finally in Eqn. 18 we rearrange Eqn. 17 to get a nonlinear first order differential equation that describes the pressure inside the capacitor as a function of the turbo pressure and the and the constant C.

We created a matlab script that numerically solves for the time to fill the air capacitor to eighty percent of the constant pressure source's pressure. The script looked at a series of tube di-
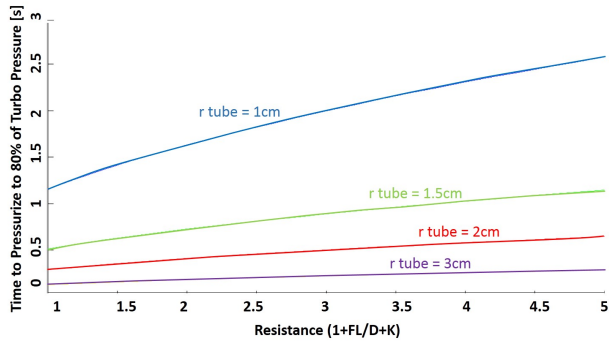
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FIGURE 3. GRAPH SHOWING FILL TIME AS A FUNCTION OF THE RESISTANCE AND TUBE RADIUS

FIGURE 4. DIAGRAM OF THE FULL INTAKE MODEL

ameters and tube resistances. In this analysis the turbo pressure is two atmospheres, the initial capacitor pressure is one atmosphere, and that the capacitor has a volume of four liters. The result of this script is shown in Fig. 3. The y axis of this plot shows the time it takes the capacitor to pressurize to eighty percent of the turbocharger pressure. The x axis of the plot shows the sum of the dimensionless resistances: the major losses, the minor losses, and the velocity losses. The fill time is heavily dependent on the tube diameter. Fig. 3 shows that the fill time is significantly less then a second for a tube diameters greater than two centimeters. This fill time corresponds to lag time, which is how long it takes for the system to reach steady state, which is how long it takes to get the power boost from the turbo charger. If lag time is too large then this system will not be feasible.

MODEL OF FLOW THROUGH INTAKE INTO ENGINE

The model described in this section accounts for the air leaving the capacitor and entering the engine. We built on the constant pressure source model using the same differential equation to describe flow going from the air capacitor to the engine (Eqn. 18). The air capacitor is treated as a varying pressure source and the engine as a varying volume of air whose volume is based on the phase angle of the crank shaft (Fig. 4). We consider two ways of modeling the turbocharger [5,6]. The first is to treat it as a constant pressure source, as presented in the previous case. We refer to this model as the infinite inertia model, as once the turbine is spinning it does not slow down. The second is to treat the turbo as an intermittent pressure source that only acts like a pressure source when the engine is going through the exhaust stroke. We refer to this as the zero inertia model, because the turbine spools up instantly and then spools down instantly with the exhaust stroke. In reality the turbine will act somewhere in between these two models. Our analysis considers a four liter capacitor, a two atmosphere turbo pressure, a 0.8 liter engine moving at 2000 RPM, and an engine with a compression ratio of 12:1. Since we
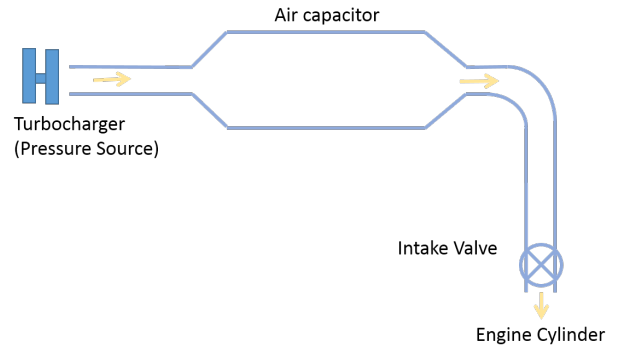
assume that the turbocharger operates at a fixed pressure due to the fact that it has a waste gate when you increase the RPM you will find that the maximum boost will decrease and the fill time will increase. This is due to the fact that the turbocharger provides a set pressure gradient that is independent of the engine speed while the engine intakes more air as its speed increases. The goal of these models is to check if the fill time of the capacitor is reasonable and to see how long it takes the capacitor to reach steady state.

The plot of the pressure inside the capacitor which uses the infinite inertia model is shown in Fig. 5. The capacitor reaches steady state in about half a second with a pressure almost equal to the turbo pressure. Figure 6 shows the infinite inertia model response during a single cycle (note that the engine pressure is only plotted during the intake stroke). During the intake stroke the pressure in the capacitor drops until it becomes equal to the engine pressure. Then the engine pressure and the capacitor pressure increase together for the rest of the intake stroke (the spikes in the engine pressure during the intake stroke are due to a conditional statement in the matlab code that does not allow the pressure in the engine cylinder to exceed the pressure in the capacitor). Finally, during the other three strokes the pressure in the capacitor increases rapidly. The main issue with this model is that it does not take into account the spool-up time for the turbocharger to reach its operating angular velocity.

The plot of the pressure inside the capacitor that uses the zero inertia model is shown in Fig. 7 (note that the engine pressure is only plotted during the intake stroke). The capacitor reaches steady state in approximately two seconds and maintains a steady state pressure which is about seventy five percent of the turbo pressure. Figure 8 shows the pressure response in the engine and capacitor over a single cycle. During the intake stroke the pressure in the capacitor drops until it becomes equal to the engine pressure and then drops at a lower rate for the rest of the intake cycle (the spikes in the engine pressure during the intake stroke are due to a conditional statement in the matlab code that does not allow the pressure in the engine cylinder to exceed the
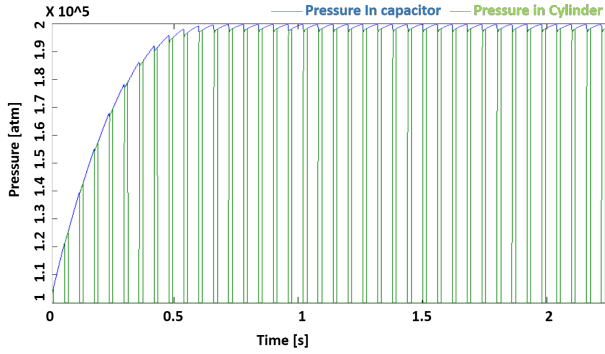
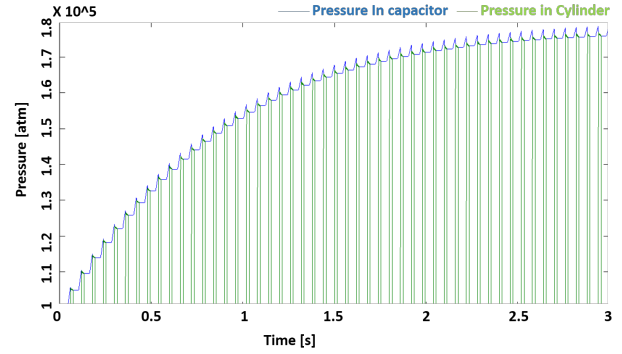**FIGURE 5**. PLOT OF ENGINE AND CAPACITOR PRESSURE PROFILE USING THE INFINITE INERTIA MODEL

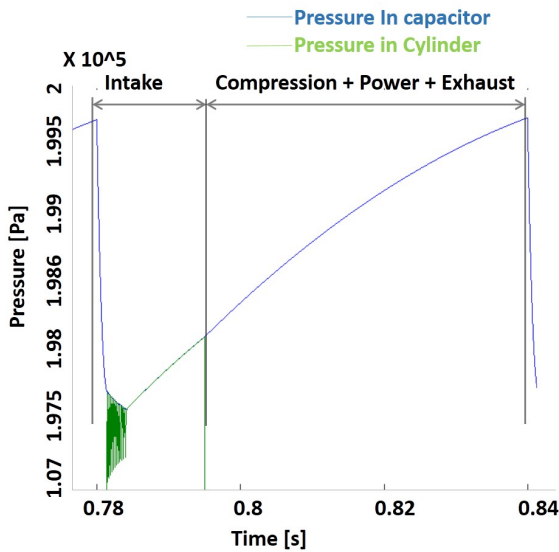**FIGURE 7**. PLOT OF ENGINE AND CAPACITOR PRESSURE PROFILE USING THE ZERO INERTIA MODEL

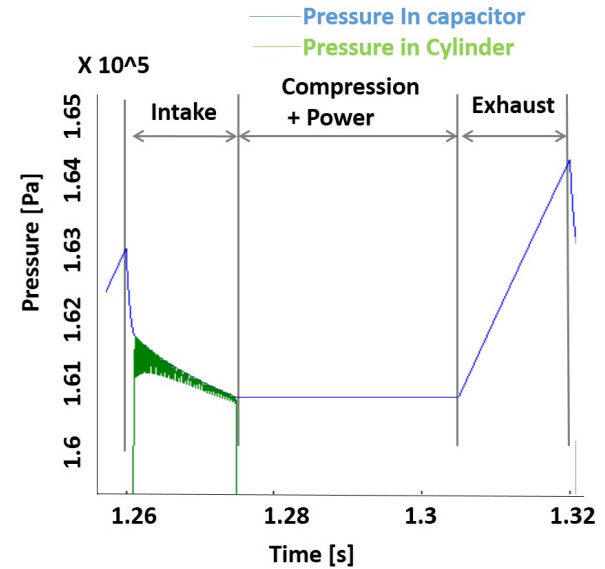**FIGURE 6**. INFINITE INERTIA PLOT ZOOMED IN ON A SINGLE CYCLE

FIGURE 8. ZERO INERTIA PLOT ZOOMED IN ON A SINGLE CYCLE

pressure in the capacitor). Then the capacitor pressure stays constant for the compression and power stroke. During the exhaust stroke the pressure in the capacitor increases rapidly.

HEAT TRANSFER OUT OF THE AIR CAPACITOR

This section describes the thermal effects on the capacitor. The pressure inside a fixed volume increases while the density stays constant when temperature increases. As a result, the temperature increase that happens inside the capacitor during isotropic compression results in a pressure gain that is proportionally larger than the density gain. If the air coming out of the turbine is be cooled to atmospheric temperature, the density gain would match the pressure gain. Since the improvement in engine

performance that the capacitor could provide is due to air density increase, this density reduction due to the temperature gain that happens during compression can reduce the power that comes out of the system. This effect is shown in Eqn. 1, which is for isotropic compression [10]. Using the matlab zero inertia turbo model, we calculated how the density in the capacitor varies relative to pressure in the capacitor if there is no heat exchange (Fig. 9). This model shows that with no heat exchange the system can still result in a fifty percent density gain in the air going into the engine, which translates to a first order power gain of fifty percent.

An intercooler type system could cool and densify the air coming out of the turbo. Using an intercooler could increase the
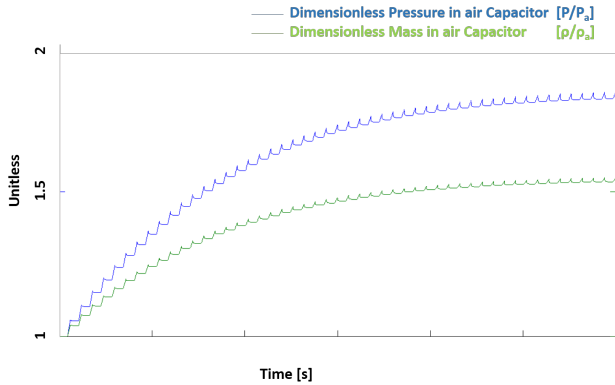
FIGURE 9. PLOT SHOWING DENSITY DECREASE DUE TO THERMAL EFFECTS OF ISOTROPIC COMPRESSION

power output of the engine by an additional twenty percent if the air in the capacitor is cooled to ambient temperature. This effect is caused by the fact that when you cool a fixed mass of air in a fixed volume you decrease the pressure while maintaining a constant density. In this case cooling the air results in a smaller pressure gradient between the atmospheric air and the air in the capacitor per unit density. This results in the turbine having to overcome a lower pressure gradient per unit density and a larger mass of air entering the engine. Assuming eighty percent efficient adiabatic cooling the density of air inside the capacitor was found to increase about twenty percent over the case with no cooling. Since the density of air inside the capacitor is directly correlated to the power out of the engine we can safely approximate that an intercooler would increase your power output by approximately twenty percent. In this case, the dimensionless density profile of the air in the capacitor will be the same as the dimensionless pressure profile of the air in the capacitor. However, intercoolers can be expensive so the feasibility of this modification depends on the economics of the application.

CONCLUSIONS

In this paper we consider turbocharging a four-stroke, single cylinder, internal combustion engine by adding an air capacitor to the intake in order to deal with the pulsating air flow conditions. The analysis presented addresses three critical design factors: capacitor size, capacitor charge time, and density gain of the air fed into the engine. Our research shows that the capacitor is a reasonable volume of between four and five times the engine capacity, that the charge time is less than two seconds, and that the density gain would be at least 50% without cooling the air and up to 80% with ideal cooling. The next step in this project will be to modify a single cylinder diesel generator with a turbo and air capacitor to see how much additional power can be obtained and how closely performance matches our model.

Historically, turbocharging can increase the power output of an engine for less cost then adding additional cylinders while also boosting the engines fuel economy [2]. There is a real need in the Indian agricultural sector for any technology that can provide more power for minimal cost [1]. The technology presented in this paper could create a broad impact on equipment such as tractors, light vehicles, generators, and water pumps increasing their specific power and making them affordable to new users.

ACKNOWLEDGMENT

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