TEACHING THE THEORY AND OPTIONS FOR IMPROVING THE EFFICIENCY OF PISTON CYLINDER INTERNAL COMBUSTION ENGINES

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ABSTRACT

It is now common practice to use a heat recovery bottoming cycle on internal combustion gas turbines with a resulting 50% increase in power and fuel efficiency. While power is not now typically recovered from the exhaust of piston cylinder engines, a similar but somewhat more complicated potential exists for a comparable increase in power and efficiency. While the exhaust of a gas turbine engine contains availability or exergy in the form of elevated temperature, a piston cylinder engine exhaust has availability associated with the surplus pressure at the end of the power stroke along with the resulting elevated temperature. The potential also exists to increase the efficiency of such engines by 50% and the corresponding miles per gallon performance of motor vehicles by 50%. This means a 30 mpg car could be a 45 mpg car and a 65 mpg car could be a 100 mpg car.

1. Introduction

Combustion products release heat over the entire temperature range from the maximum flame temperature down to the ambient temperature. Internal combustion engines have the thermodynamic advantage of excellent conversion of the high temperature heat into work because the combustion products are also the working fluid, but they have the disadvantage of exhausting at moderately high temperatures and thus poor utilization of the moderate to low temperature heat.

In recent years it has become common practice to install a power producing bottom cycle on gas turbine engines. The resulting combined cycle efficiency is typically over 50% as compared to 35% without the combined cycle.

If a similar improvement could be made by extending existing automobile type piston cylinder engines to combined cycles a 30 mpg car would achieve 45 mpg, and a 65 mpg car might achieve 100 mpg. This defines the motivation for teaching the theory and suggesting options for extending automotive type engines to combined cycles.

We start by defining the availability or exergy that exists in the exhaust of a gas turbine vs a piston cylinder engine. Since a gas turbine exhausts at atmospheric pressure but elevated temperature, all of the exergy is associated with the elevated temperature of the exhaust. In contrast, a piston/cylinder engine has exergy associated with both excess pressure and excess temperature at the end of the power stroke.

While both gas turbine and piston/cylinder engines are heat engines there is a fundamental difference in how the heat is converted to work. Since the gas turbine is a flow device, and thus not positive displacement, the net work is the result of repressurizing in the turbine at a higher specific volume than for the pressurization process in the compressor. Heat is used to increase the specific volume.

In contrast the positive displacement piston cylinder engine produces net work because the burning of the fuel increases the pressure. Thus the expansion is performed at a much higher pressure than the compression. Since the piston is normally connected to a crank shaft, the displacement volume for the expansion is the same as for the compression which results in the wasteful elevated pressure at the end of the expansion stroke.

Thus, the complete conversion of the exhaust availability of a piston cylinder engine into work requires a two stage bottoming cycle. The first stage should produce work from the full expansion down to atmospheric pressure and the second stage should produce work from the elevated temperature that still exists after the full expansion process.

The need for a two stage bottoming cycle for piston/cylinder engines can now be compared with techniques that have been proposed and/or implemented but that recover only the surplus pressure or only the elevated temperature. The former is demonstrated by various full expansion concepts such as the Miller cycle which shortens the effective compression stroke by modifying the valve action, or by extending the expansion with a supplemental exhaust cylinder. The latter is demonstrated by using the elevated exhaust temperature to boii steam for a Rankine cycle.

We develop the analysis for an idealized spark ignition Otto cycle and Diesel engine in terms of work from the basic cycle, additional work from full expansion and then the extra work that can be extracted by a thermal bottom cycle.

A potentially practical, but less than ideal, two stage bottom cycle is then proposed and evaluated. This proposed two stage bottom cycle consists of an exhaust turbine to implement the full expansion and then an external combustion gas turbine cycle as a light weight and compact technique for the thermal energy recovering stage of the bottom cycle.

There is also a potential synergism between the development of the adiabatic piston/cylinder engine which requires no cylinder cooling and the suggested combined cycle. The basic adiabatic engine will be somewhat more efficient because there is no net loss of heat through the cylinder walls, but this also means the exhaust will have more thermal energy in terms of higher pressure and temperature. Thus more power can be recovered from this exhaust stream with each of stage of the proposed bottom cycle.

While the basic efficiency of an adiabatic engine is somewhat improved, absence of cylinder cooling means the exhaust will be much higher pressure and temperature. Thus much more potential efficiency improvement can be recovered from the exhaust.

2. Analysis of Otto Cycle with Two Stage Bottoming Cycle

This section analyzes the basic idealized Otto cycle, and also identifies the availability and the requirements for recovering the availability at the end of the power stroke.

The Otto and full expansion Otto cycles are analyzed on the assumption of an 8 to 1 compression ratio, 19 lbs of air for 1 lb of fuel and the combustion products are an ideal gas with a constant pressure heat capacity of .25. (Btu/lbm R) and a ratio of 1.4 between the constant pressure and constant volume heat capacities. The cycle also starts at 40 F or 500 R.

Figure 1 shows the basic Otto cycle as the F occsses 1-2-341 and the full expansion Otto cycle as 1-2-3-5-1.

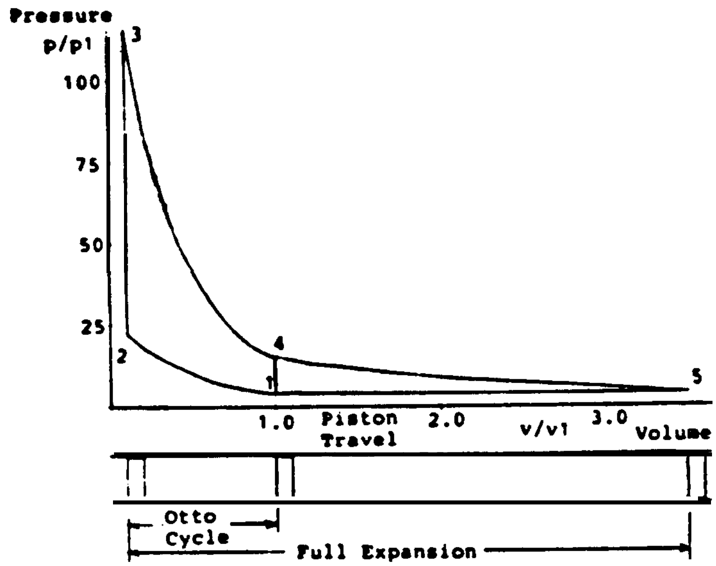

Figure 1. P/Pi vs V/Vi for Otto and Full Expansion Otto Cycle

Figure 2 shows the same Otto and Full Expansion Otto Cycle on an absolute temperature vs entropy diagram.

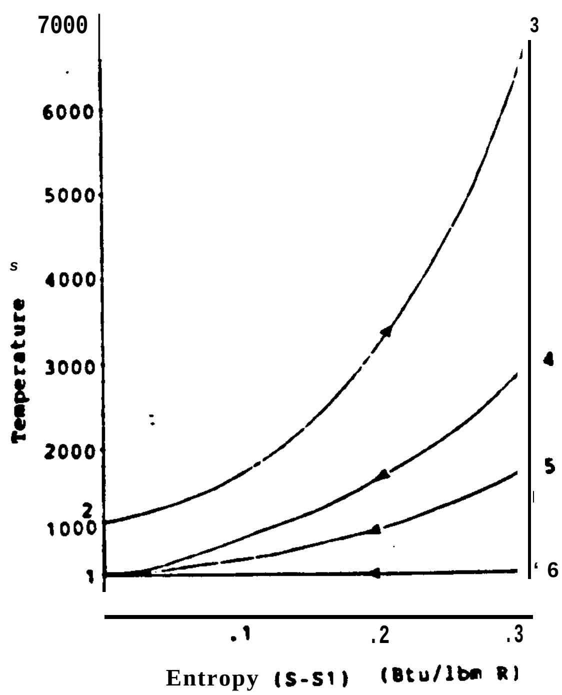

Figure 2 T vs S-S1 for Otto and Full Expansion Otto

The property Values for each point in Figure 1 and 2 is presented in Table I.

Table I
Property Values for Otto and Full Expansion Otto Cycle

#	p/pi	v/vi	T(F)	T(R)	u (Btu/lbm)	s-s1 (Btu/lbm R)
1	1	1.0	40	500	89.29	0
2	2838	.125	688.7	1148.7	205.13	0
3	107.97	.125	6288.7	6748.7	1205.13	3162
4	S.88	1.00	2447.6	2937.6	524.57	3162
5	1.00	354	1311.2	17712	31629	.3162

Using this property table and assuming 1 lbm within the control volume, a 1st Law Process and Cycle Table is developed for the Otto cycle and presented in Table 11 and for the full expansion Otto cycle in Table III.

Table II
First Law Process and Cycle Table for the Otto Cycle

Proms	Heat In (Btu)	m*(u-u) (Btu)	Work Out (Btu)
1-2	0	115.84	-115.84
2-3	1000	1000	0
3-4	0	-680.56	680.56
4-1	-435.2a	-425.28	0
Net	564.72	0	564.72

Table 111
First Law Process and Cycle Table for ● Full Expansion Otto Cycle

Process	Heat In (Btu)	m*(u-u) (Btu)	Work Out (Btu)
1-2	0	115.84	-1 15.84
2-3	1000	1000	0
3-5	0	-888.84	888.84
5-1	-317.8	-227.00	90.80
Net	6822	0	682.20

While point 5 which at the end of the power stroke of the full expansion engine is at the starting atmospheric pressure an elevated temperature of 13112 F or 1771.2 R exists. The corresponding availability can now be recovered by a thermal bottoming cycle.

This availability is defined by the area 1-5-6-1 on Figure 2. Thus the ideal engine would be a three process cycle that forms a perfect fit for this area. Such a cycle could be a gas turbine with an isothermal compressor (6-1), an ideal gas to gas heat exchanger (l-5) and an ideal *turbine* (5-l).

The efficiency of such a cycle has been developed by the ● uthor (Reference 1) and is presented in equation 1, and is evaluated for Thm = 17712 R and the ambient temperature Tc is 500 R.

$$Eff = Work/Qhot = 1-Tc*ln(Thm/tc)/(Thm-Tc) \quad (1)$$

The corresponding value of this ideal engine efficiency is 5025% . It is noted that the heat input to the cycle Qhot corresponds to the heat rejected from the full expansion engine in the 5-1 process which is 317.8 Btu as shown in Table II. Thus the corresponding work that can be recovered is 159.7 Btu which is the product of the efficiency and the heat input.

The prior analysis has identified the amount of work that can be produced as the sum of 1) work from the basic Otto cycle, 2) additional work form an ideal thermal bottoming cycle, and 3) the additional work from an ideal thermal bottoming cycle with the heat supplied after the full expansion process.

Site the total heat input from fuel was 1000 Btu in the Otto cycle, the efficiency of each cycle of process can be defined as the ratio of its work to this input.

The amount and fraction of the total energy from each of these cycles and processes is now summarized in Table IV.

Table IV

Summary of Work and Corresponding Percent for 1000 Btu *of* Fuel Input

	Cycle or Process	Work (Btu)	Work (%)	Efficiency (%)
1.	Idealized OUO Cycle	564.71	67.1	56.47
2.	Full Expansion Extra	117.49	14.0	11.75
3.	Thermal Bottom Cycle	159.70	18.9	15.97
	Total	841.90	100.0	84.19

Using this prior analysis the availability vs crank angle can be presented for the basic Otto cycle in a manner that also shows the loss of availability at the end of the normal expansion stroke when the exhaust valve opens. This information is based upon the input of 1000 Btu of fuel and is presented in Figure 3.

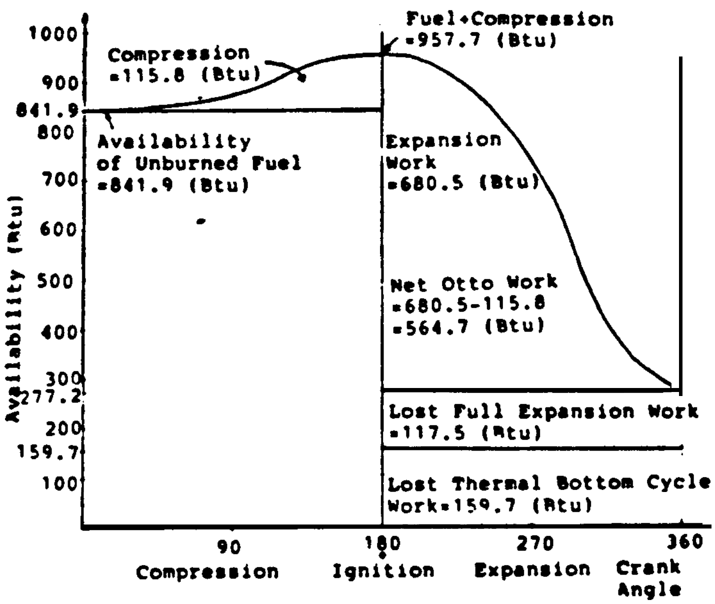

Figure 3 Availability vs Crank Angle for Basic Otto Cycle for 1000 Btu of Fuel

3. A Possible Design of a Two Stage Bottom Cycle

The prior analysis was based on an idealized Otto cycle engine and would require complete and instantaneous combustion, no heat loss through the cylinder walls and no mechanical friction on pistons or valve trains.

The corresponding idealized efficiency of the Otto cycle was about 56% and increased to 84% with the combination of full expansion and an ideal thermal bottom cycle. This also means that a 50% increase in power or ear gas mileage can be achieved.

However, the realistic Otto cycle will have a maximum efficiency of about 28% as a result of various losses. While there is no realistic method for recovering virtually all of the availability, a potentially practical method for recovering a substantial proportion is shown in Figure 4.

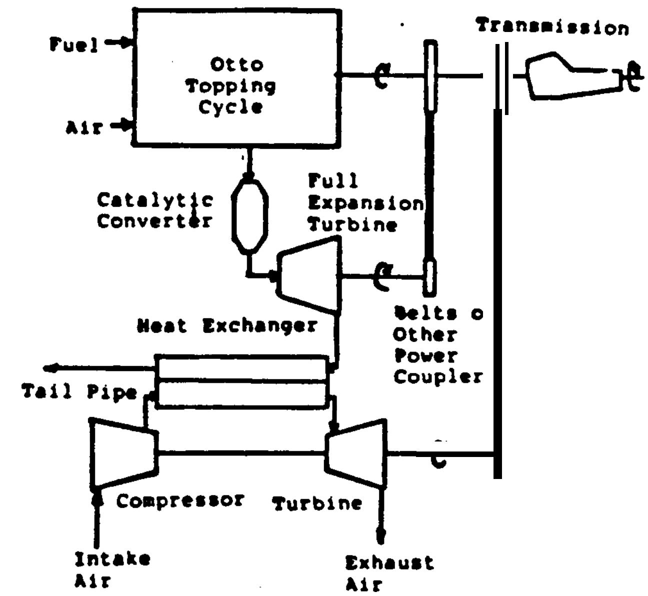

Figure 4 A Two Stage Bottoming Cycle for a Piston Cylinder Engine

This system consists of a conventional piston cylinder engine with a turbine installed on the engine exhaust, and then with • compact external combustion gas turbine cycle recovering heat from the full expansion turbine exhaust.

This system is also characterized by three power output shafts corresponding to the piston engine, the exhaust turbine and the bottoming gas turbine, which must be integrated either mechanically or electrically for transmission to the vehicle drive shaft or wheels.

It is demonstrated that an exhaust turbine on an Otto cycle serves virtually the same function as the less practical methods for a full expansion engine such as a secondary cylinder, a Miller cycle or some positive displacement rotary engine techniques that can have a larger expansion than compression volume.

While virtually all bottom cycles at electric utility plants are steam Rankine cycles, such a system would be too large, complicated and in need of too much condenser surface area to be practical for a bottoming cycle on a ear.

Thus, an external gas turbine cycle, which can be almost as efficient is proposed Such a cycle has been previously proposed and analyzed by the author (References 2 and 3).

The analysis of a non-idealized system is much more complicated relative to the prior idealized system, but such an analysis has been performed with the following assumptions, estimates and results.

The Otto engine was assumed to be 28% efficient with no exhaust turbine, but reduced to 24% with an exhaust turbine. The two turbines and compressor have 88% isentropic efficiencies. Heat loss through the cylinder walls is 20% of input fuel heat. The catalytic converter increases the exhaust temperature by 300 F, by completing the burn, which increases the availability for the bottoming cycle. The gas to gas heat exchanger has a hot stream to cold stream temperature difference of 200 F. The gas turbine pressure ratio is optimized as a function of temperatures and compressor and turbine efficiencies.

The corresponding output on each shaft for 1000 Btu of fuel input is summarized in Table V in a form similar to the idealized assumptions in Table IV.

Table V

Estimated Performance of Realistic Otto Cycle with Two Stage Bottoming Cycle

	Cycle or Process	Work (Bin)	Work (%)	Efficiency (%)
1.	Realistic Otto Cycle	240	57.1	24.0
2.	Exhaust Turbine	100	23.8	10.0
3.	Turbine Bottom Cycle	80	19.1	8.0
	Total	420	100.0	42.2

These results indicate that while the efficiency of a realistic Otto cycle is about half the efficiency of the ideal, the capability exists in each case to increase power output by about 50%. Thus, the subject system could represent a practical method to achieve the projected 50% improvement in car mileage, or to decrease fuel consumption by 33% relative to existing vehicles.

We also note that while the arrangement on Figure 4 is developed and presented for the purpose of thermodynamic analysis, this arrangement maybe simplified. One possibility y is to combine the two turbines into a single turbine. Another would be to replace the relatively small turbine driven compressor with an engine belt driven piston compressor similar to the car air conditioner, or usc crank case compression similar to the technique used by 2 cycle engines.

4. Historic and Future Perspective

The preceding provides a thermodynamic conceptual and analytical basis for extending a piston cylinder engine to a combined cycle. It is also useful to present these new ideas in the context of the nearly 300 years of development of heat engines. Thus Table VI has been prepared and is presented to the students to suggest the challenge for their generation should be the development of such an engine.

Table VI

Development and Evolution of Heat Engines (development dates, inventors, efficiency range)

	Piston/cylinder (non-flow)	Turbine (now)
External Combustion	1700-1820 (Newcomen, Watt) (1% to 12%)	1875-1900 (Parson, Curtis) (lo% to 3s%)
Internal Combustion	1875-1900 (Otto, Diesel) (10% to 3s%)	1930-1960 (Whittle, et al) (10% to 3s%)
internal Combustion with Power Recovery (Combined Cycles)	2000-2050 ? (large teams ?) (30% to 55%?)	1960-1995 (large teams) (4090 to 58%)

5. Conclusions and Recommendations

The author has outlined one possiiity for a two stage bottoming cycle for piston cylinder engines, but much additional synthesis and analysis in necessary for this method, and there arc many other similar but different options.

Thus, the multiple objectives are to demonstrate to engineering students the potential and importance of increasing piston cylinder engine performance by 50% with the expectation that they will continue to investigate and hopefully help develop such an engine.

While, the fundamental operation of existing car engines was first demonstrated by Nicholas Otto in 1876, our modem engine is the result of a myriad of continuous and often dramatic improvements over the last century. This paper is meant to help define similar opportunities for the next century.

Acknowledgement:

The author thanks the Niagara Mohawk Power Corporation for the energy related projects grant that helped make this work possible.

References:

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