INTERCOOLING-SUPERCHARGING PRINCIPLE: Part II—— High-Pressure High-Performance Internal Combustion Engines With Intercooling-Supercharging

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ABSTRACT

Since Brayton, Beau de Rochas, and Otto (1876) first put forth the importance of **compression before ignition/combustion**, simple cycle internal-combustion engines have evolved into three successful types: gasoline engine, diesel engine, and gas turbine. Raising compression ratio leads to simultaneous increases in thermal efficiency and engine specific power output. A large part of the continuous improvement in performance of these three types of engines has been achieved with increasing peak cycle pressure.

Even designed at optimal peak cycle pressure, however, significant heat loss remains in the exhaust of internal combustion engines. In Part 1 of this two-part paper the **intercooling-supercharging principle** was introduced as a means that reduces exhaust heat loss. In Part 2, the proposed principle is reformulated by representing the thermal efficiency and the (mass) specific power in terms of their natural parameters: peak cycle pressure, peak cycle temperature (or, equivalent ratio), a newly-introduced intercooling-supercharging parameter, and a newly-introduced composite-engine parameter. A multi-variable search optimization procedure is then used to find the optimum designs. The proposal shares with the seminal concept of Brayton and Otto the unique characteristic, as an engineering solution, in producing simultaneous increases in efficiency and engine specific power output. Intercooling-supercharging should raise the performance of IC engines to new levels at peak cycle pressure higher than that required for the Otto cycle, the diesel cycle, and the Brayton cycle—meeting the challenge of the new generation of industrial/utility powerplants and the vehicle propulsion engines.

1. INTRODUCTION

In Part 1 (Wang 1995) of this two-part paper, the original formulation of the intercooling-supercharging principle was introduced as a means for reducing exhaust enthalpy loss from "simple-cycle" internal combustion engines. The focal point was the thermal efficiency improvement in engine systems—without requiring high temperature heat exchanger.

In reviewing the simulation-study results, it was realized that the optimization procedures used were awkward or ineffectual. In Part 2 the optimization procedure is reconsidered again by looking for the logical (natural) variables that characterize the intercooled-supercharged cycle engines as the direct extension of the "simple cycles." In terms of these variables, a multi-variable univariate search procedure is then formulated for finding the optimum design. The advantage of the "simple cycles" is their performance. The focal point of Part 2 will be the performance of internal combustion engines. The intercooled-supercharged cycle engines will be presented as logical development of simple cycles, achieving a higher performance level.

2. PERFORMANCE

Performance is used in this paper as the engine specific power (the reciprocal of engine specific weight or engine specific volume) ***and*** the engine's fuel consumption (the specific fuel consumption; alternatively, its reciprocal divided by fuel heating value—the thermal efficiency). Since the engine specific power depends on its mass specific power and the density of the working fluid, we shall consider performance as principally a function of the thermal efficiency; the (mass) specific power; and the working fluid density—at a particular (reference) point of the process, e.g., density at the bottom dead center before the compression stroke of the piston-cylinder component of a composite engine.

Performance of the "simple cycle" internal combustion engines

Since it was independently pointed out by Brayton, Beau de Rochas, and Otto (and later by Diesel), the concept of optimal compression before ignition/combustion became the single most important concept after Carnot's idea of achieving the highest combustion temperature in a combustion heat engine. As outlined by Beau de Rochas, his third and fourth conditions under which maximum efficiency could be achieved are (Heywood 1988:xx):

3. The greatest possible expansion ratio

4. The greatest possible pressure at the beginning of expansion

(Beau de Rochas's first two conditions hold heat loss from the charge to a minimum.)

We shall refer to the Brayton-cycle gas turbine, and the Otto-cycle and diesel-cycle naturally aspirated piston engines as simple cycle engines. Performance—thermal efficiency, mass specific power, and density—of simple cycle engines depends principally on two variables (three variables, if one also consider the cylinder temperature and the possibility of raising it). For the gas turbine,

$$\eta_{th} = \eta_{th}(P_{peak};\ T_{peak}) \tag{6}$$

$$p_{mass} = p_{mass}(P_{peak};\ T_{peak}) \tag{7}$$

$$\rho_{ref} = \rho_{ref}(P_{peak};\ T_{peak}) \tag{8}$$

where η_{th}, p_{mass}, ρ_{ref} represent thermal efficiency, mass specific power, and charge density at some reference point. For piston engines,

$$\eta_{th} = \eta_{th}(P_{peak};\ \phi,\ T_{cylinder}) \tag{9}$$

$$p_{mass} = p_{mass}(P_{peak};\ \phi,\ T_{cylinder}) \tag{10}$$

$$\rho_{ref} = \rho_{ref}(P_{peak};\ \phi,\ T_{cylinder}) \tag{11}$$

where ϕ is the equivalent ratio of charge. Standardized ignition and combustion processes for the gasoline engine and for the diesel engine are assumed in the above equations. It should be noted that ϕ for both Otto and diesel engines are at their respective maximum, and $T_{cylinder}$ is also at the maximum—unless high-temperature material and lubrication advance make it possible for raising its value.

(Only principal design thermodynamic variables are considered in the above equations. Other design and operating variables, which are also important to engine performance such as engine/combustion-chamber geometry, ignition timing, valve timing, speed, flow field characteristics, ... , are assumed at their "optimal" standard design-values.)

At the maximum ϕ, $T_{cylinder}$, or at the current gas turbine T_{peak} (again depending on the material technology) the single remaining principal variable is peak cycle pressure. For each type of simple cycle engine there is a corresponding optimal P_{peak} range. The optimal P_{peak} values for the thermal efficiency and for the specific power are in general distinct, but not far apart; these two values define

Nomenclature

p_{engine} = engine specific power
p_{mass} = mass specific power
P = pressure ratio
r = intercooling supercharging parameter defined by equation (12a)
s = composite engine parameter, defined by equation (15a)
T = temperature
α = intercooling supercharging parameter, defined by equation (12)
β = (non-integer) exponent in equations (15), (15a)
η_{th} = thermal efficiency
ρ = density
ϕ = equivalent ratio

Subscript

back = piston back pressure, or temperature
cylinder = cylinder wall temperature
peak = peak cycle values
ref = reference point at which the charge density is evaluated as the most relevant to p_{engine}
super = supercharging pressure ratio

the optimal range. In each case, raising P_{peak} toward its optimal value range increases thermal efficiency, specific power, and charge density—leading to simple cycle engines of good performance. It is important to note that the main reason for the increase in thermal efficiency by raising P_{peak} is the lowering of exhaust charge temperature.

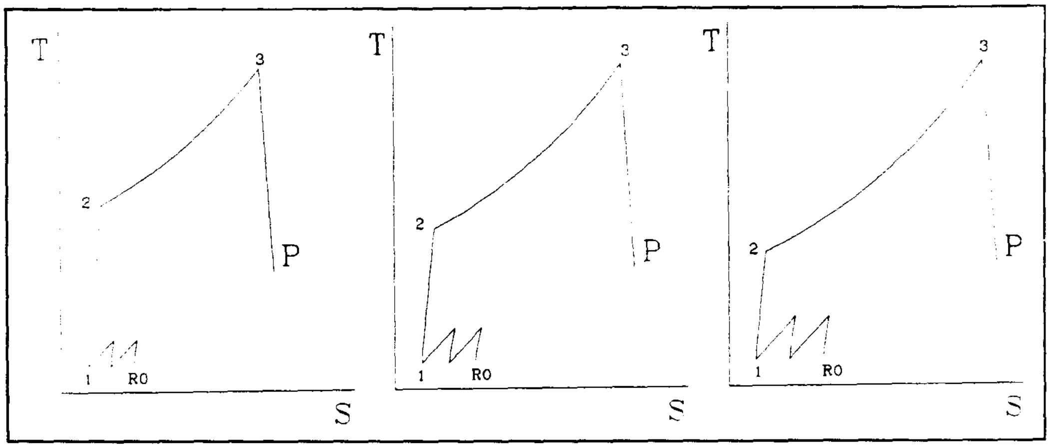

Figure 7 Temperature versus entropy for various r values under constant P_{peak}. The middle diagram represents the "optimal" placement of intercoolers.

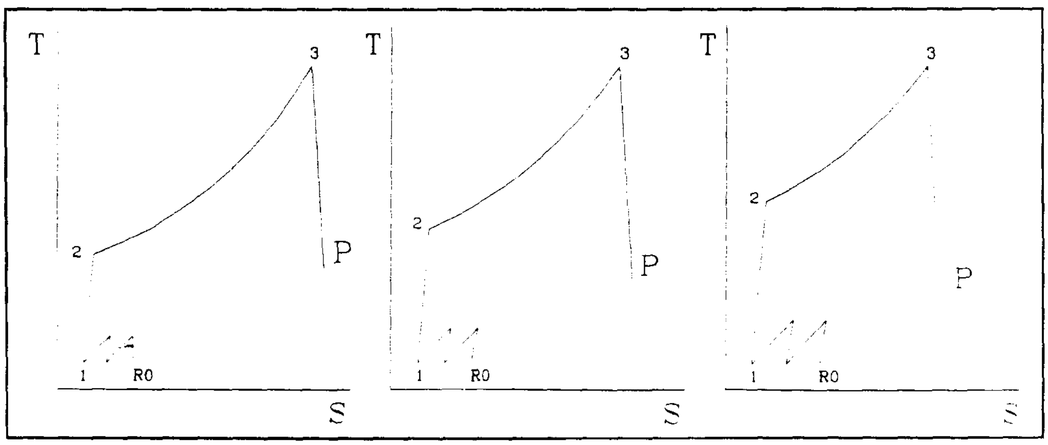

Figure 8 Temperature versus entropy for increasing P_{peak} with "optimal" placement of intercoolers.

3. INTERCOOLING SUPERCHARGING PARAMETER

Even at the optimal P_{peak}, significant enthalpy loss remains in the exhaust of a simple cycle engine. Intercooling-supercharging was proposed in Part 1 as a means to further reduce the exhaust enthalpy loss. It is noted that equation (3) in Part 1—representing the old optimization procedure—is not an explicit function of P_{peak}.

We shall begin by expressing performance of the intercooled-supercharged gas turbine in terms of P_{peak}, thus establishing a direct link with the simple-cycle gas turbine. A second variable, the *intercooling supercharging* parameter, α or r, is defined as

$$P_{super} = P_{peak}^{\alpha} \tag{12}$$

or, alternatively

$$r \equiv \tfrac{1}{2}\ln P_{super}/\ln(P_{peak}/P_{super}) \tag{12a}$$

$$1/r \equiv 2\{[\ln P_{peak}/\ln P_{super}] - 1\}$$

Either α or r can be used for characterizing the placement of the two intercoolers. They are simply related as

$$\alpha^{-1} - 1 = \tfrac{1}{2} r^{-1}$$

Instead of equation (3), we have now

$$\eta_{th} = \eta_{th}(P_{peak}, r; T_{peak}) \tag{13}$$

$$p_{mass} = p_{mass}(P_{peak}, r; T_{peak}) \tag{14}$$

r is defined between 0 and 1; α defined between 0 and 2/3. With r = 0, equations (13),(14) reduce to equations (6),(7), the simple cycle case. With r = 1, equations (13),(14) become the thermal efficiency function and the specific power function for the conventional intercooled cycle turbine, where the two intercoolers are placed between compressors of equal pressure ratios.

4. THE TWO-VARIABLE UNIVARIATE SEARCH OPTIMIZATION FOR INTERCOOLED-SUPERCHARGED GAS TURBINE

With a constant T_{peak} for the gas turbine, equations (13),(14) suggest a two-variable univariate search method. Instead of the procedure represented by Fig.3 in Part 1, the two-variable univariate search is represented by the T-S diagrams in Fig.7 and Fig.8. Fig.7 shows schematically the T-S diagrams of increasing r (from left to right) under constant P_{peak}. Fig.8 shows schematically the T-S diagrams of increasing P_{peak} (from left to right) under constant r.

An example of the computed two-step results (first step represented in Fig.7, second step represented in Fig.8) for the thermal efficiency and the mass specific power using GATE/CYCLE software (see Wang and Pan 1994 for details) is shown in Fig.9. A complete performance curve map is given in Wang and Pan (1994), where the respective performances of the simple cycle, the intercooled-supercharged cycle, and the conventional intercooled cycle are shown.

The first step of the univariate search shows the existence of optimal r value for the maximum thermal efficiency. Increasing r thus simultaneously increases specific power, thermal efficiency, and charge density. The second step shows largely increase in thermal efficiency—associated with the lowering of exhaust enthalpy—as well as increase in charge density, of course. With the additional variable r in equations (13),(14), performance of an optimum designed intercooled-supercharged cycle gas turbine is capable of reaching a level significantly higher than that of the simple cycle case. The optimal peak cycle pressure for maximum performance is at a value much higher than that of simple cycle gas turbine.

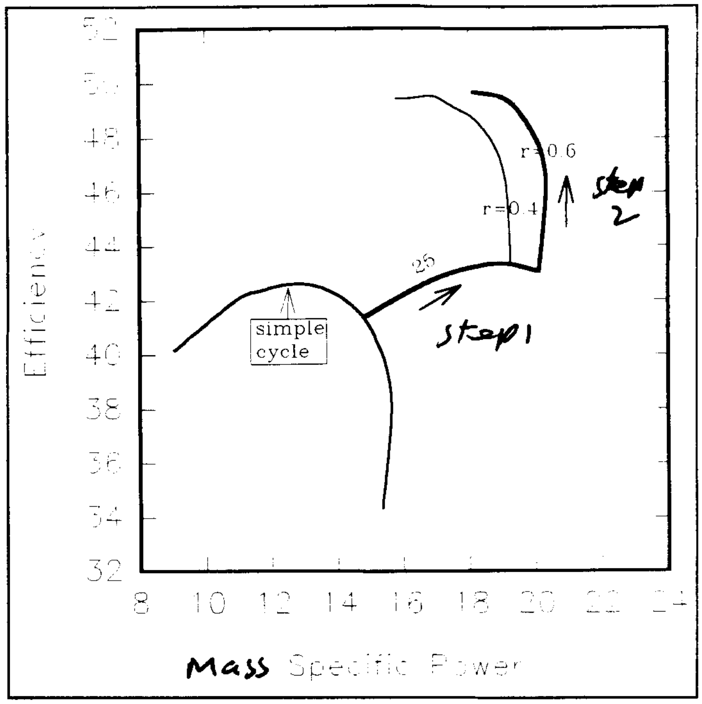

Figure 9 The two-step optimization procedure.

5. COMPOSITE ENGINES

> "The piston engine is eminently suitable to deal with relatively small volumes at high pressure and temperature and the turbine, by virtue of its high mechanical efficiency and large flow areas, to deal with large volumes at low pressures. Clearly the logical development is to combine the two in series to form a compound unit."
>
> *H. R. Ricardo*
> [quoted in Smith (1955:279-280)]

With the exception of the utility and large industrial powerplants, there is a compelling logic in consider the **composite** internal combustion engines that combine the piston engine and the turbine/compressor. Their potentials, as envisioned by Ricardo, have not been realized. We will make the case that intercooling is the key in realizing the full advantage of the composite engines, and will outline the reformulated methods in applying intercooling-supercharging principle to "combine" the piston engine and the turbine.

Without intercooling, the concept of optimal r and the benefit of higher optimal P_{peak} do not exist. (The optimal P_{peak} of the simple cycle applies.) Supercharging increases charge density in the cylinder, thus increases engine specific power. However, there will be no increase in mass specific power and no increase in thermal efficiency—no simultaneous increases in power and efficiency.

With intercooling, simultaneous increases in power and efficiency may lead to extraordinary gain.

Three types of arrangements are possible according to shaft power characteristics. The case of both crank shaft and turbine shaft producing power is known as the turbocompounding engine. The remaining two:

- intercooled-supercharged gas generator engine—composite engine with single (turbine) power-shaft
- intercooled-turbocharged piston engine—composite engine with single (piston crank) power-shaft

will be considered in this paper, due to their advantage over the turbocompounding engine by avoiding the complexity of dual power shafts.

Due to their single power-shaft nature, important advantage can be gained by "shifting" charge expansion enthalpy into the device producing power, away from the non-net-power-producing component. A third principal variable for the performance functions is to be introduced in the form of the ratio of piston back pressure (ratio), P_{back}, to the supercharging pressure (ratio),

$$P_{back}/P_{super}.$$

A best selection of P_{back}/P_{super} may be made according to maximum thermal efficiency at a given r and P_{peak}. This value does not remain constant, however, for variable r or/and variable P_{peak}. From the limited results obtained, the best ratio is approximately proportional to $P_{super}^{-constant}$. The new variable, the **composite engine parameter**, s, is defined accordingly

$$P_{back}/P_{super} = sP_{super}^{-\beta} \quad (15)$$

i.e.,

$$s \equiv (P_{back}/P_{super})(P_{super})^{\beta} \quad (15a)$$

where β is a constant exponent, to be selected for the gas generator engine, and for the turbocharged engine respectively. It should be repeated that this definition of s is not unique; its definition is chosen for facilitating the optimization procedure without excessive iteration required. β is expected to be a constant for the best ratio only in a finite range of r's and P_{peak}'s. For the gas generator engine β may also be selected according to thermal load (instead of maximum thermal efficiency), represented by constant T_{back}. In this case, β is found to be ½ in our previous study.

Performance of the composite engines is now represented by functions of three pressure-related variables.

6. THREE VARIABLE UNIVARIATE SEARCH METHOD FOR INTERCOOLED-SUPERCHARGED GAS GENERATOR ENGINE

For gas generator engines the performance functions are:

$$\eta_{th} = \eta_{th}(P_{peak}, r, s; T_{cylinder}) \quad (16)$$

$$p_{mass} = p_{mass}(P_{peak}, r, s; T_{cylinder}) \quad (17)$$

$$p_{ref} = p_{ref}(P_{peak}, r, s; T_{cylinder}) \quad (18)$$

For a constant $T_{cylinder}$, the optimum design of the gas generator engine is determined by an iterative three step univariate search. The engine schematic diagram is shown in Fig.5 in Part 1. P_{back} (pressure maintained in the main Ex/in Manifold) is typically greater than P_{super}. A new additional feature, variable (late) inlet valve closing timing, is introduced in the present (Part 2) consideration. The iterative steps are

- At preset r and P_{peak} values, consider the **matching of different two-stage turbines** (see Fig.5) **with the piston component** to vary the back pressure, P_{back}. The supercharging pressure ratio and the compression ratio of the piston component may have to be adjusted within narrow ranges to maintain constant r and P_{peak}. Change P_{back} until maximum thermal efficiency is found; this defines the optimal s. (If the corresponding T_{back} exceeds the thermal load limit, a constant T_{back} limit will be used instead of maximum thermal efficiency to define the optimal s.)

- At the same preset P_{peak}—defining the mechanical load—and the same s as determined, change r value by **varying supercharging pressure ratio and inlet valve closing timing in combination** maintaining the constant P_{peak}. Different matching two-stage turbines may be needed to maintain constant s. The index constant β is to be determined in this step by repeating step 1 so that constant s defines approximately the best matching turbine for maximum thermal efficiency at each individual r case. Change r until maximum thermal efficiency is found. Thermal efficiency and specific power results are recorded for each r.

- At the same s and r as determined, consider **increasing piston compression ratios** for higher and higher P_{peak} values. Other system parameters (turbine-piston matching, supercharging pressure, inlet valve closing timing, ...) again may need to be adjusted to maintain constant s and r. Change P_{peak} until thermal efficiency and specific power beginning to decline.

- In the region of near optimal r and P_{peak} after the first three steps, consider β and s values again. Change β value if necessary, and revise s value if necessary.

- Repeat step 2 and step 3 for a finite range of r and P_{peak} to construct performance curves of thermal efficiency vs. specific power (or thermal efficiency vs. engine specific power) for a chosen best s value.

This map of performance curves together with thermal load and mechanical load consideration will serve as the guide for the design of the intercooled supercharged gas generator engine.

7. INTERCOOLED TURBOCHARGED PISTON ENGINE

The starting point of designing the intercooled turbocharged piston engine (*ITurbo*™ engine) is the recognition that the piston crank shaft is the power shaft. Charge expansion energy should therefore be shifted away from the turbocharger turbine into the piston-cylinder by keeping P_{back} lower than P_{super}—the piston intake pressure—and reducing blowdown loss. Performance functions are:

$$\eta_{th} = \eta_{th}(P_{peak}, r, s; \phi, T_{cylinder}) \tag{19}$$

$$p_{mass} = p_{mass}(P_{peak}, r, s; \phi, T_{cylinder}) \tag{20}$$

$$\rho_{ref} = \rho_{ref}(P_{peak}, r, s; \phi, T_{cylinder}) \tag{21}$$

For spark-ignition homogeneous charged engine, equivalent ratio, ϕ, may be chosen at a constant 0.9.

The optimization procedure including a three-step univariate search is as follows:

- Choose preset r and P_{peak}, and a particular intake volume and effective compression stroke. Consider different combinations of (longer) piston stroke and (later) inlet valve closing (IVC) timing such that the intake volume remaining constant, and then match the piston with the best turbocharger turbine unit. Repeat until thermal efficiency becomes maximum for a particular combination of piston stroke, IVC timing, and turbocharger turbine. Note the corresponding P_{back} value and s value. (For non-constant-pressure turbocharger turbine, P_{back} is determined as the effective value of an equivalent constant-pressure turbocharger turbine according to equal h_{back} value.)

- Using the same bore/stroke design and at the preset P_{peak} and the s value as determined, vary the combination of supercharging (turbocharging) pressure and IVC timing such that P_{peak} remaining constant and select turbocharger turbine unit such that s remaining constant. The index constant ß is to be determined in this step by repeating step 1 so that constant s defines approximately the best matching turbine for maximum thermal efficiency at each individual r case. Continue until thermal efficiency reaching maximum. Record η_{th} and p_{mass} as function of r.

- Consider increasing effective compression stroke ratio and the corresponding P_{peak} under constant s and r, until thermal efficiency and mass specific power beginning to decline.

- In the region of near optimal r and P_{peak} after the first three steps, consider ß and s values again. Change ß value if necessary, and revise s value if necessary.

- Repeat step 2 and step 3 for a finite range of r and P_{peak} to construct performance curves of thermal efficiency vs. specific power (or thermal efficiency vs. engine specific power) for a chosen best s value.

Again this map of performance curves together with thermal load and mechanical load consideration will serve as the guide for the design of the *ITurbo*™ engine.

8. CONCLUSION

Since it was independently pointed out by Brayton, Beau de Rochas, and Otto, the concept of optimal compression before ignition/combustion became the single most important concept—after the Carnot's idea of achieving the highest combustion temperature in a combustion heat engine. The single-variable optimization led to "simple cycle" internal combustion engines of good performance. Simple cycle IC engines remain the dominant powerplant of choice because of their performance, even with critical concern of fuel economy and emission which are not the strong points of simple cycles.

The application of the intercooling-supercharging principle unfolds the possibilities of multi-variable optimization that raise performance of internal combustion engines to new levels. These high-performance engines are characterized by high peak cycle pressure.

While detailed studies for the gas generator engine and the turbocharged engine remain to be carried out, a drastic increase in engine specific power, p_{engine}, is expected for the two composite engines, resulting in a quantum leap in performance. As a market-driven solution, which favors high-performance technology, is the preferred solution to societal well-being, the three proposed internal combustion cycles may represent the most effective solution to fuel economy and emission.

To paraphrase Carnot: The application of intercooling-supercharging for the development of the high-pressure intercooled engines presents in practice great challenges. If we should succeed in overcoming them, it would doubtless offer a notable advantage over simple-cycle engines.

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