INTERCOOLING-SUPERCHARGING PRINCIPLE: Part I—Reduction of Exhaust Heat Loss From Internal Combustion Engines by Intercooling-Supercharging

Lin-Shu Wang
Department of Mechanical Engineering
SUNY, Stony Brook

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ABSTRACT

"Simple-cycle" internal combustion engines (gasoline engine, diesel engine, and simple-cycle gas turbine) are designed at or near optimal peak cycle pressure for maximum power output and thermal efficiency. Significant heat loss remains, however, in the exhaust of these optimally-designed internal combustion engines. Various methods for improving fuel economy have been proposed based on the straight-forward principle of exhaust heat recovery. Though successfully used for stationary powerplant and marine propulsion, these often bulky advanced systems are unsuitable for applications that require compactness and high power density.

Alternative cycles that directly reduce exhaust heat loss were proposed recently. As reduction means rather than recovery means, the new cycles may lead to compact engines of high efficiency. This paper discusses the common underpinning of the new cycles—the intercooling-supercharging principle—and its theoretical origin, and reviews the simulation-study results of two types of engines, as examples of the application of the new principle. While these results confirm the advantage of the intercooling-supercharging means, a critical review reveals that the original formulation of the principle has drawback as an optimization principle. A reformulation of the intercooling-supercharging principle—to be carried out in Part 2 of this two-part paper—will be required in order to find the optimum designs of intercooled-supercharged internal combustion engines, and to appreciate the relationship of the intercooled-supercharged cycle engines with the simple cycle engines.

1. INTRODUCTION

Internal-combustion engines have evolved, since Brayton, Beau de Rochas, and Otto (1876) first put forth the importance of **compression before ignition/combustion**, into three successful types: the spark-ignition displacement piston engine, the compression-ignition displacement piston engine, and the continuous-combustion dynamic gas turbine—commonly known as gasoline engine, diesel engine, and gas turbine. The development of internal-combustion engines over the past 118 years was foreshadowed by Sadi Carnot, who recognized the advantage of air as the working fluid, as well as the oxidizer for fuel combustion. Internal-combustion eliminates the need for the boiler or heat exchanger and enables high peak cycle temperature of the working fluid. "The use of atmospheric air for the development of the motive power of heat," he wrote, "presents in practice very great but perhaps not insurmountable difficulties. If we should succeed in overcoming them, it would doubtless offer a notable advantage over vapor of water" (Carnot 1824; Wilson 1981:143). Carnot's prediction was borne out in modern internal-combustion engines; the long progress of raising peak cycle temperature continues today in the advanced gas turbines being developed.

The second fundamental feature of an ideal heat engine according to Carnot is that heat should be rejected, or exhaust should be discharged, at the temperature of the surroundings. Raising compression ratio or peak cycle pressure reduces the temperature of the exhaust. However, even at the optimum peak cycle pressure the burned gas exiting from a "simple-cycle" internal-combustion engine is still at an elevated temperature. Realizing that situation, Carnot also outlined the case for a combined cycle. "We might conceive even the possibility of making the same heat act successively upon air and vapor of water. It would only be necessary that the air have, after its use, an elevated temperature, and instead of throwing it out immediately into the atmosphere, to make it envelop a steam boiler, as if it issued directly from a furnace" (Carnot 1824; Wilson 1981:144). In the 1920's a combined cycle system, in the form of diesel engine, exhaust boiler, and a reciprocating steam engine, was tested, but failed to catch on. In the recent years combined cycle systems in the form of gas turbine topping cycle, heat recovery steam generator (HRSG), and steam turbine bottoming cycle have become the most popular fossil-fuel fired stationary powerplants, some 170 years after Carnot proposed them.

Other means (cycles) to recover the exhaust heat loss—some of them are of more recent origin—have been proposed and developed. All these cycles require gas-to-air heat exchanger or gas-to-steam heat exchanger operating at elevated temperature—the higher the peak cycle temperature, the higher the heat exchanger operating temperature. Most, like the combined cycles, also require two working fluids. The resulting size and complexity of the systems limit their application to stationary plants and to marine propulsion. The application of exhaust heat recovery to produce compact internal-combustion engines remains unfulfilled.

An alternative method to minimize the exhaust heat loss in gas turbine engines and gas generator engines (Wang and Jeng 1992; Jeng and Wang 1992a, 1992b; Guo and Wang 1992) without the high-temperature heat exchanger has been proposed. The proposed engines represent two examples of the application of a general principle: *intercooling-supercharging principle*. This paper (part 1 of a two-part paper) and the follow-up part-2 will examine the general principle: its theoretical origin, its formulation, its applicabilities to other engine configurations, and the general characteristics of the resulting engines in relationship with the simple cycle engines. Part 1 reviews the simulation-study results obtained in earlier studies and examines critically the principle as an optimization procedure.

We shall first retrace the steps that led to the intercooling-supercharging principle.

2. EXHAUST HEAT/EXERGY LOSS

a. Energy Balance Analysis

Air and fuel enter an internal-combustion engine system, undergoing compression-combustion-expansion process with possible cooling, and exit as burned gas. Energy balance of the working fluid may be expressed as

$$({}_PH_R)_0 = W_{shaft} + (-Q) + (H_P - H_{P0}), \tag{1}$$

where terms are defined in **Nomenclature**. Energy balance analysis points out the possibility of increasing shaft work output by reducing engine cooling or/and recovering exhaust heat loss. The following are examples of cycles based on the exhaust heat recovery principle:

- combined cycle, with steam bottoming cycle
- steam-injected gas turbine (STIG or the Cheng cycle)
- regenerative gas turbine
- InterCooled Regenerative (ICR) gas turbine
- Air Bottoming Cycle (ABC)
- Humid Air Turbine (HAT) cycle
- combined cycle, with Kalina bottoming cycle

While energy balance analysis suggests the possibility and correlates the possible results, it does not determine the real benefit of a particular proposed method. For

Nomenclature

H = enthalpy,
$({}_PH_R)_0$ = fuel heating value of the air-fuel mixture,
$H_P - H_{P0}$ = exhaust enthalpy (heat) loss.
P = pressure ratio,
$-Q$ = heat rejection by the system, e.g., engine block cooling, or turbine blade cooling,
T = temperature,
S = entropy,
W_{shaft} = shaft work output,
η_{th} = thermal efficiency

Subscripts

back = pressure or temperature in the exhaust manifold of cylinder
M = main compressor
P (or, Ex) = exiting product or exiting burned gas,
P0 = exhaust gas (exiting burned gas) after its cooling to the surroundings temperature,
Peak = peak cycle temperature or pressure ratio
R = fuel air (reactant) mixture,
S = each supercharging compressor,
Super = supercharging unit (compressors & intercoolers)
0 = the surroundings condition

example, the reduction of engine block cooling should be correlated with increase in the sum of work output and exhaust heat loss, according to the energy balance equation. The equation itself does not predict work output increase and exhaust heat loss increase individually. In the case of low-heat-rejection diesel engine, reduction in the engine block cooling leads to large increase in the exhaust heat loss, with little change in the work output. Nor does energy balance analysis reveal the maximum possible shaft work to be recovered.

b. Energy/Exergy Analysis

The more restrictive possibility of maximum useful work is of course imposed by the exergy balance analysis.

The energy analysis or exergy analysis is a misnomer. A complete thermodynamic analysis always involves both energy and entropy variables. In this sense, a complete thermodynamic analysis is energy/exergy analysis. Results from such analyses can be presented in terms of energy balance, as well as in terms of exergy balance. The "exergy (balance) analysis" stresses the importance of presenting result of thermodynamic analysis in term of exergy balance.

Only complete energy/exergy analyses of the above cycles can determine the real benefits of the proposed systems. Exergy balance values then disclose the maximum of the remaining possible work to be extracted, thus suggesting further improvement possibilities.

A traditional energy/exergy analysis and its energy balance and exergy balance summary are passive exercises. Though capable of predicting engine system performance and revealing further theoretical possibilities, they alone do not offer explicit means for achieving such possibilities. The various cycles and their improvement are devised by inventors and innovators, rather than simple outgrowth of energy/exergy analysis.

c. Operational Energy/Exergy Analysis

It has been suggested that the approach taken by Carnot in founding thermodynamics is not the prevailing passive approach of today, but an approach that incorporates an engineering design concept—the operational means. It was demonstrated (Wang et.al. 1992; Wang and Capobianchi 1993) that understanding of thermodynamic processes requires the inseparable and explicit consideration of operational means in applying thermodynamic laws to analyze processes.

The *inseparableness* of thermodynamic laws and operational means also suggests the possibility that understanding gained by operational energy/exergy analysis may directly lead to energy conversion device design

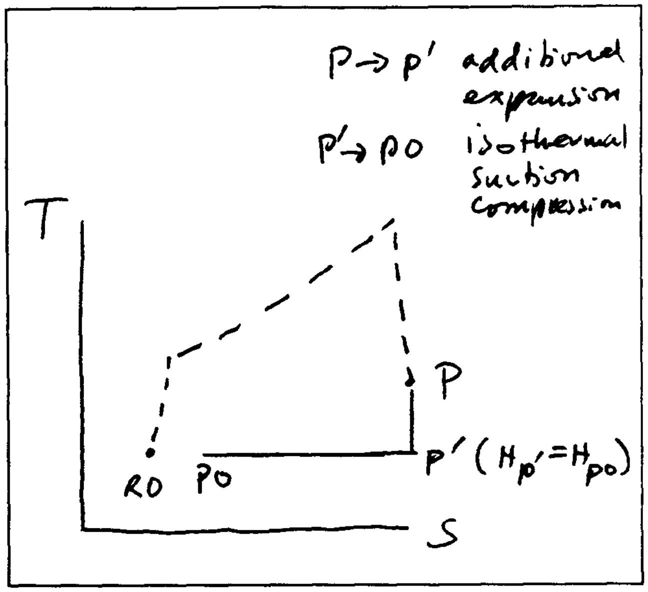

Figure 1 Operational analysis of the exhaust gas exergy from simple-cycle internal combustion engines.

concept. The following is a description of such example.

Consider a simple cycle internal-combustion engine, made up of compression, combustion, and expansion steps, as shown in **Fig.1**. State of the exhaust gas is designated by subscript P; when the exhaust gas eventually comes to thermal and mechanical equilibrium with the surroundings, its state is designated by subscript P0. The exergy (availability) of the exhaust gas is

$$(H_P - H_{P0}) - T_0(S_P - S_{P0}) \tag{2}$$

How can one "recover" this exhaust exergy operationally? Expression of the exhaust exergy suggests immediately the following theoretical construction: An ideal isothermal suction-compressor of work requirement, $T_0(S_P - S_{P0})$, is attached to engine exit to maintain engine exit condition at P'. Additional expansion stages are added to the engine so that the combustion product gas now expands further isentropically through the additional stages to the specific lower pressure at P'. The extra work output from the additional expansion stages is exactly $H_P - H_{P0}$. The net useful work gain in this theoretical construction is exactly the exhaust exergy of the simple cycle internal-combustion engine.

The same theoretical thermodynamic advantage can be achieved by using the operation of isothermal supercharging rather than isothermal suction-compression. In fact, isothermal supercharging, which eliminates the exhaust exergy/heat loss, is a more attractive mechanical configuration than isothermal suction-compression, which recovers the loss. The pressure ratio of the isothermal supercharging compressor should be at a specific value so that the exhaust gas exiting exactly at T_0. In a real engine, the ideal isothermal supercharging is replaced by *intercooling-supercharging*.

3. INTERCOOLING-SUPERCHARGING PRINCIPLE

In a real engine the exhaust gas will exit at a temperature above T_0. Intercooling-supercharging will *reduce* (not eliminate) the exhaust heat/exergy loss, which is in contrast to the use of various methods listed in section 2a for *recovering* the exhaust heat/exergy loss.

> The optimal pressure ratio of the intercooling-supercharging compressor should be at a specific value producing the optimally lowest exhaust gas temperature, defined by the maximization of thermal efficiency over the domain of specific constraints.

The use of intercooling-supercharging to reduce the energy/exergy loss should be applicable to all internal-combustion engines, as long as efficient compression means and efficient additional expansion means are available. In practice efficient additional expansion means require the inclusion of turbine expansion stages. The following three examples—each involves turbine expansion means—will be considered as illustrations of the application of intercooling-supercharging principle:

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

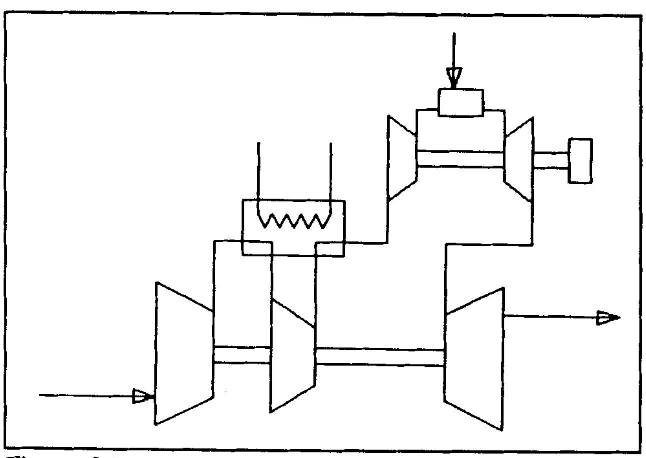

Figure 2 Intercooled-supercharged gas turbine.

Definition of composite engines will be given in Part 2. A schematic diagram for the case of gas turbine is shown in **Fig.2**.

4. INTERCOOLED-SUPERCHARGED GAS TURBINE

The constraint against which the optimization in thermal efficiency (and specific power) is sought for the intercooled-supercharged gas turbine is constant *turbine inlet temperature*, T_{peak}. A schematic temperature-vs-entropy diagram showing various supercharging pressure ratios, and the corresponding thermal efficiency plotted against the supercharging pressure ratio,

$$\eta_{th} = \eta_{th}(P_{super}, P_M; T_{peak}) \tag{3}$$

are shown in Fig.3 and Fig.4 respectively. η_{th} is the thermal efficiency. P_{super} is the pressure ratio of the supercharging unit. P_M is the pressure ratio of the main compressor. The peak cycle pressure, P_{peak} is then

$$P_{peak} = P_{super}P_M \approx P_S^2 P_M \tag{4}$$

(P_{super} is approximately equal to the square of P_S, the pressure ratio of each supercharging compressor, as two supercharging compressors are shown in Fig.2—assuming small intercooler pressure head loss.) The maximum thermal efficiencies at the optimal supercharging pressure ratios are shown in Fig.4—where the reduction of exhaust heat loss by intercooling-supercharging is balanced with the increasing intercooling loss, as shown in Fig.3.

The optimization procedure represented by equation (3) and in Fig.3 and 4 are carried out under constant P_M—as suggested by the operational energy/exergy analysis. P_{peak}, therefore, increases with increasing supercharging pressure ratio. In order to study the effect of P_M or the peak cycle pressure, one has to repeat the above optimization procedure for different P_M. Examples of such procedure can be found in Guo and Wang (1992). Only after comprehensive results obtained, one can obtain, by indirect construction, the thermal efficiency (and the specific power) as a function of P_{peak}, and determine the optimal P_{peak}. Such systematic study was not carried out in Guo and Wang (1992). In hindsight, it was not carried out because the optimization procedure represented by equation (3), as suggested directly by the original operational energy/exergy analysis, is not the most logical formulation.

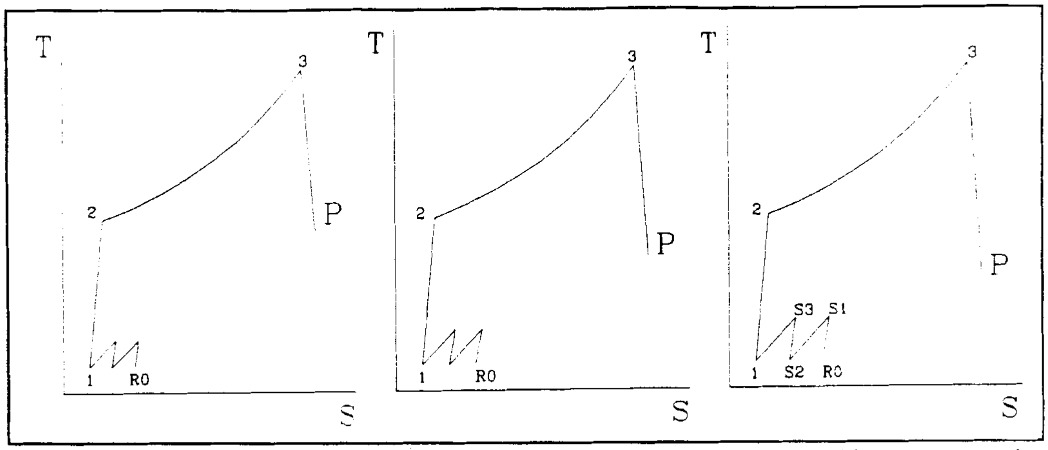

Figure 3 Optimal pressure ratio of intercooling-supercharging for the gas turbine——constraint is constant turbine inlet temperature.

5. INTERCOOLED -SUPERCHARGED GAS GENERATOR ENGINE

The drawback of the formulation (equation (3)) becomes more acute when application of the intercooling -supercharging principle to the gas generator engine is considered.

A gas generator engine consists of a gas generator unit, and a power turbine unit that is the only power producing unit. In the gas generator unit, a piston engine component drives the supercharging compressors. Together they form a gas generator producing no net power.

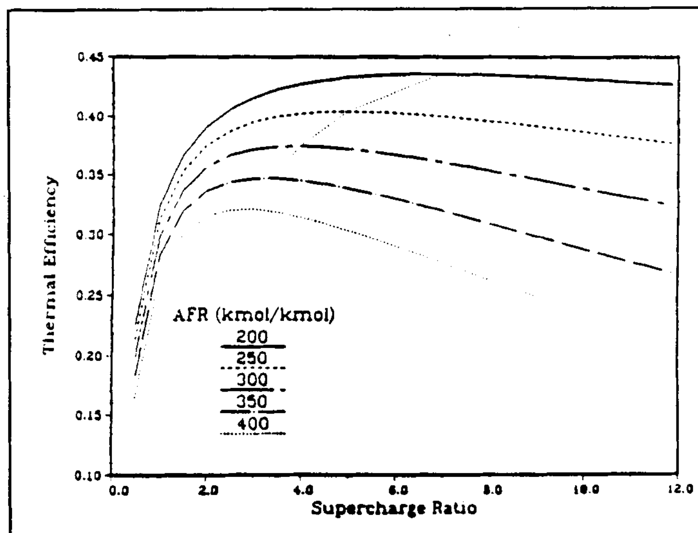

Figure 4 Maximum thermal efficiencies at the optimal supercharging pressure ratios.

Unlike the case of the gas turbine, fuel-air ratio of the gas generator engine is not an independent parameter. The peak cycle temperature is dependent on the piston back exhaust temperature, T_{back}, (which is nearly equal to the turbine inlet temperature). Control of T_{back}, hence T_{peak}, is through the control of piston back pressure by the design-matching of turbine units with the piston unit (Jeng and Wang 1992a and 1992b; Chung and Wang 1993; Wang and Chung 1994). Since the required piston back pressure is higher than the piston intake pressure, two-stage piston exhaust process is necessary (Wang 1991). A schematic diagram of the intercooled-supercharged gas generator engine is shown in **Fig.5**. Computer simulation discloses (Wang and Chung 1994) an optimal value of T_{back} around 1100K at which the thermal efficiency is maximum. This temperature is selected as the constant constraint for thermal efficiency optimization against the supercharging pressure ratio.

The "optimal" supercharging pressure ratio, P_{super}, is investigated under constant T_{back}, aa well as constant piston compression ratio. The result shown in **Fig.6** (Wang and Chung 1994) is of the form,

$$\eta_{th} = \eta_{th}(P_{super};\ T_{back},\ \text{compression ratio}). \qquad (5)$$

It should be stressed that the above equation is not of the same form as Equation (3). As supercharging pressure ratio increases, T_{peak} also increases in the present case. As a result, the exhaust temperature initially decreases, but eventually stops decreasing and instead increases moderately. The graphical interpretation, Fig.3, of the intercooling-supercharging principle is **not** followed by equation (5). As the "optimally lowest exhaust gas temperature" concept is no longer valid, a clear optimal supercharging pressure ratio is not assured any more—as in Fig. 6, only a barely visible "optimum" is shown for the metal engine (case 1) and no optimum is found for the Low Heat Rejection engine (case 2).

While the application of the intercooling-supercharging principle as formulated to the gas turbine engine is awkward, its application to the gas generator engine is ineffectual. Though the results reported in Wang and Chung (1994) shows excellent engine specific power (the reciprocal of engine specific volume), those results are not—may be far from—the optimal performance result capable of the intercooled-supercharged gas generator engine.

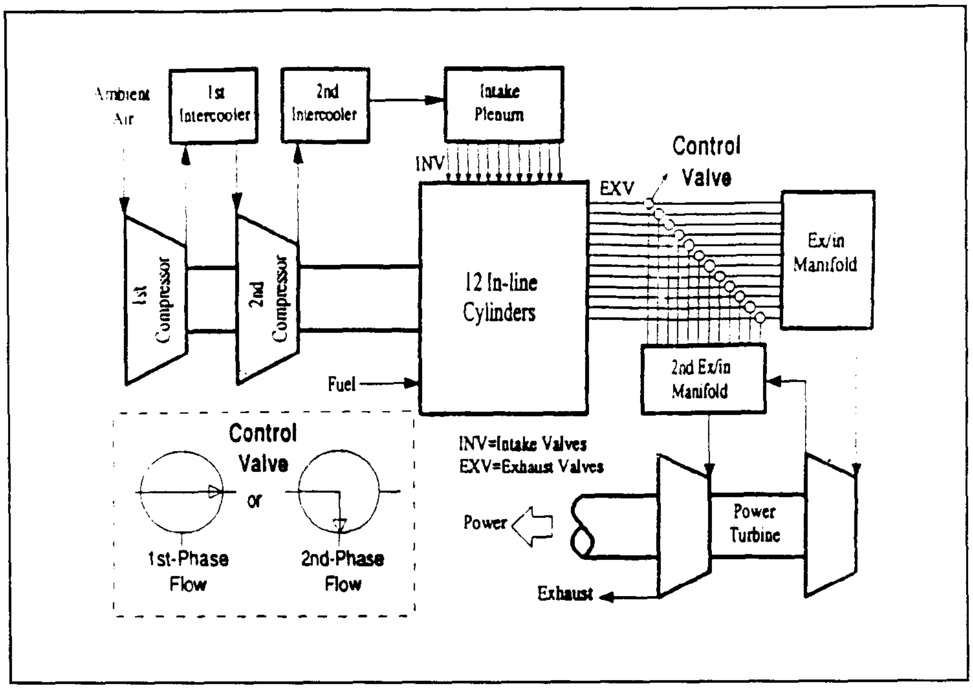

Figure 5 Schematic diagram of intercooled-supercharged gas generator engine.

6. CONCLUSION

As far back as 1824, Carnot already pointed out the possibility of recovering the exhaust heat loss of internal combustion engines and suggested the idea of cogeneration and combined cycle. Many other means of recovery have since been proposed. An alternative (intercooling-supercharging) principle has been proposed to improve the thermal efficiency of internal combustion engines by reducing the exhaust heat loss. This paper gives a critical review of the proposed principle. While the simulation-study results confirm the advantage of intercooling-supercharging, its original formulation has drawback as an optimization principle.

In the formulation as directly suggested from the operational energy/exergy analysis, the application of the principle in designing optimal engine configuration is found to be awkward for the case of the gas turbine, and ineffectual for the case of the gas generator engine. This conclusion suggests the necessity for the reformulation of the principle, which is presented in Part 2 of the two-part paper.

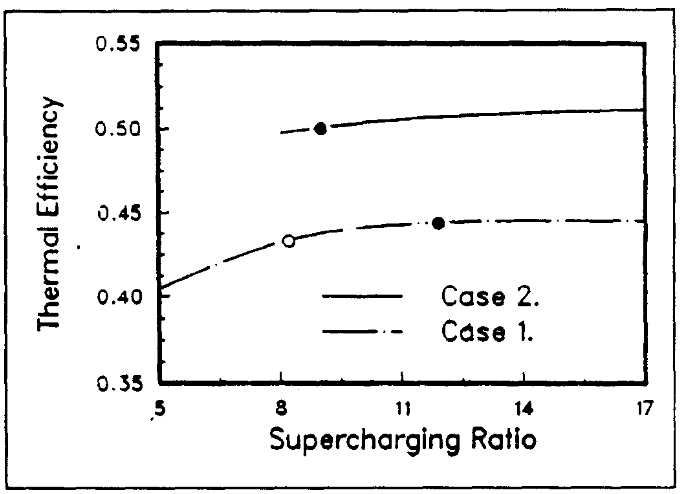

Figure 6 Maximum thermal efficiency at the optimal supercharging pressure ratio.

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