Abstract

This paper explains what SCW (Super Critical Water) diesel is, what sort of combined system it is, how effective it can be, why the authors develop it and how the system can be realized. One may, in a word, consider it as a kind of parallel combined systems by means of comparatively large diesel engines. This paper clarifies the reason why the parallel combined system for diesel engines is regarded as “SCW diesel system”.

This paper introduces Super Critical Water (SCW) diesel as a parallel combined system and explains its efficacy and output focusing on the difference between parallel combined systems and series combined systems, referring to GT (gas turbine) parallel combined system and GT series combined system. One often calls the former system GT-Cheng cycle and the latter system GTCC (gas turbine combined cycle). This paper points out that SCW diesel is not a simple regenerative or economizing system but a remarkable power & efficiency amplifier for prime movers.

This paper also details how a parallel combined system for diesel engines could be materialized. Consequently, how to estimate the system equilibrium is studied. Here the system equilibrium means a balance between prime movers and waste heat exchangers and a balance between the air-gas cycle work and the SCW cycle work in the cylinders. In solving the equilibrium (especially as to the latter balance), the authors introduced the concept of “designing the cycle process of reciprocators”. Without this concept, it is almost impossible even to work out SCW diesel system and/or to make a plan of confirmation tests.

Concept verification tests are still in progress. This paper touches on the test preparation. However, the expected outcomes for power increase, efficiency improvement and emission reduction are discussed.

Key Words: Diesel Engine Technology, Gas Turbine Cheng Cycle, Diesel Cheng Cycle or Super Critical Water Diesel, Energy Saving and Low Emission Technology, Thermodynamics and Heat Transfer, Power and Efficiency Amplification (Feedback Amplification)

1. Introduction

Each one of energy economy, efficiency improvement, global environment protection and better cost-eff ectiveness is a primary concern of the engineers and scientists who have relation to heat engines. On and after the late twentieth century, the significance of cogeneration has been acknowledged. With the advancement of the world, people seem to have looked forward to the innovation technology in isolated propulsion plants and distributed generating-sources. Especially from the view point of heat power (or electricity) ratio, the power-weighted cogeneration with high total-efficiency is expected.

In 1976, D.Y.Cheng [1] invented GT(Gas Turbine)-Cheng cycle system, and not a few plants of the system have been built in Japan since the middle of eighties. As early as 1987, a committee[2] entrusted by Japanese government started a study on “Variable Heat and Power Balanced Engine System” for reciprocating engines. The study intended to realize very the Cheng-cycle system of reciprocator version. Following up the history, this paper clarifies necessary ideas and hardware elements to realize “Diesel-Cheng cycle system” or “SCW diesel system”.

One of the most important conclusions in this report is that SCW diesel is not a mere hybrid system of diesel engines and steam engines but a system of strong feedback-amplifier as to power and efficiency (especially as to power).

The authors touch on the concept-verification-tests now in preparation as far as possible.

2. GT-Cheng Cycle as a Reference

2.1 GT Combined Cycle

GT-series combined system (GTCC) and GT-parallel combined system (GT-Cheng cycle) are illustrated in “Fig. 1(a)” and “Fig. 1(b)” respectively. While a series-combined system requires expensive steam turbines on the down-stream side, a parallel combined system dispenses with steam turbines. Therefore GT-Cheng cycle system is more cost-effective for smaller gas turbines.

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2.2 Feedback Amplification

D.Y. Cheng et al. [3] gave a plain illustration regarding the comparison between GT series system and GT-Cheng cycle system by means of T-s diagrams. The authors quote it in “Fig. 2”. It is very clear that GT-Cheng cycle is different from regenerative or economizing cycle.

Fig. 2 T-S diagram comparison (GT) (series vs. parallel)

“Fig. 2” also suggests that a state-variable vector \( x_i \) of Cheng cycle system at a stage \( (i) \) is described as a function of vector \( x_{i-1} \) at the previous stage \( (i-1) \). Namely,

\[
x_i = f(x_{i-1})
\]

When we simplify the problem, we often think of linear operator for \( f \). By limiting manipulation \( (i \rightarrow \infty) \), “Eq.(1)” becomes the equilibrium equation

\[
x_\infty = f(x_\infty)
\]

Let us consider, for instance, a state-variable vector \( x_\infty \) as to heat balance and suppose a heat balance model as shown in “Fig.3 & Fig.4”.

\[
x_\infty = [\alpha_\infty \ \beta_\infty \ \delta_\infty]^T
\]

Where \( \alpha \), \( \beta \), and \( \delta \) are the ratios against heat input as to power, exhaust heat energy and cooling-mechanical loss respectively. Further we assume that the regenerative energy from the wasted-heat exchanger is proportional to the exhaust (emitting) energy \( \beta_i \) of the prime mover: let the proportional constant be \( \gamma \). Also we assume that the additional heat input is proportional to the regenerative energy \( \gamma \beta_i \) (proportional const. \( \kappa \) ) because the typical or maximum temperature in the prime mover has to be kept constant in general. In this simple linear model, the system matrix can be a stable matrix, with physically pertinent constants \( (\gamma \ \& \ \kappa) \). Hence, the system can be stable and the equilibrium point \( x_\infty \) can exist.

\[
x_\infty = [\alpha_\infty \ \beta_\infty \ \delta_\infty]^T
\]

The final heat balance becomes different from the original one, as shown in “Fig. 4”.

Fig. 3 Heat balance (original system)

Fig. 4 Heat balance (parallel combined system)

The power amplification factor \( K_1 \) and efficiency amplification factor \( K_2 \) are defined as follows.

\[
K_1 = \frac{\alpha_\infty}{\alpha_0}
\]

\[
K_2 = \frac{\alpha_\infty}{\alpha_0 (1+\kappa \ \gamma \beta_\infty)}
\]

If we, bearing the actual GT-Cheng cycle plants in mind, assume pertinent values for \( \gamma \) and \( \kappa \), we are able to find that \( K_1 \) and \( K_2 \) are remarkable. Even this simplified model can explain GT-Cheng systems’ actual experience such that \( K_1=1.45 \) and \( K_2=1.3 \). Here, the authors note that

\[
K_1 > K_2 > 1
\]

In case of series combined system,

\[
K_1 = K_2
\]

The difference between “Eq.(7)” and “Eq.(8)” distinguishes parallel combined systems from series combined systems. Accordingly, it is important that a parallel combined system offers surprising power increase and cost effectiveness besides omission of steam turbines.

Some engineers may mention that, as parallel combined systems spend vast amount of water, they are not a proper combined system for isolated plants. However, when water generating systems or desalination systems can supply the amount, parallel combined systems are fairly attractive.

Discussing the convergence of heat-balance variables is not the only goal of the authors. We also need to estimate the balance of the state variable such as material amounts, pressures and temperatures and so on.
2.3 Two Cycle Processes in a Combustor

Sketching the outline, we may state that there are an air-gas cycle process and a Rankine cycle process in the combustion chamber of a parallel combined prime mover. The combustion chamber also acts as a boiler for the Rankine cycle. This boiler can heat up water (H\textsubscript{2}O) up to the temperature which the prime mover can endure. In the usual boilers of steam turbine plants, the maximum temperature of H\textsubscript{2}O is around 600°C, while a parallel combined prime mover heats-up H\textsubscript{2}O to very high temperatures: we can expect approximately 1000°C for small gas turbines and ~2000°C for large diesel engines.

Thus, a Rankine cycle of very high temperature is included in a parallel combined system, bringing a remarkably high efficiency, although the cycle does not depend on such a degree of vacuum as a condensing steam turbine’s Rankine cycle depends on. That is, the effective work (expansion enthalpy drop) in the Rankine cycle is increased by enhancing the maximum enthalpy value rather than by lowering the enthalpy value at the end of expansion. Anyhow, the Rankine cycle’s efficiency pulls up the original air-gas cycle’s efficiency surprisingly, accompanying also remarkable power increase.

The above consideration is one of main reasons why the authors intend to develop SCW diesel as a parallel combined system.

The T-s diagrams in “Fig.5a and 5b” illustrate the two cycles in a parallel combined system. In “Fig.5a”, the cycle 1-2-3-4 and a-b-b’-c-c’ correspond to the original air-gas cycle and regenerative Rankine cycle respectively. Without additional heat input, the air-gas cycle shrinks toward the cycle 1-2-3'-4', while the Rankine cycle grows the cycle a-b-c-d-ex-c’. Accordingly, the maximum temperature and the expansion temperature in the parallel combined system decrease. This is undesirable because the efficiency of the air-gas cycle is lessened generally and because the expansion temperature (\(T_4\)) is lowered (to Point 4’). We have to evade the smaller amount of recovered steam or H\textsubscript{2}O due to the lowered temperature in general, as we are aiming at not a shrunk equilibrium of state variables but a stably enlarged one.

Thus, we are able to understand that parallel combined system requires additional fuel input as illustrated in “Fig.5b”. With some pertinent amount of additional fuel which energy corresponds to the area 3'-3-4-4’ plus the area d-d’-ex-ex’, the final Rankine cycle would be completed. The fundamental final-equilibrium of many state-variables is roughly based on the two thermal cycles, that is, the cycle 1-2-3-4 and the cycle a-b-b’-c-d-ex-c’. However, the authors have to note that the T-s modeling of “Fig.5a” and “Fig.5b” is insufficient to discuss fuel quantity and H\textsubscript{2}O quantity. The balance of the two quantities must be estimated by the equilibrium between the prime mover and the heat exchanger.

In addition, the authors have their doubt about whether the Rankine cycle in “Fig.5” is an authentic one or not. The authors’ conclusion is that, as the air-gas cycle and the Rankine cycle affect each other during being mixed and heated, the Rankine cycle is different from a usual Rankine cycle which the mono-material of H\textsubscript{2}O makes. Therefore we need to deal with the two cycles in “Fig.5” carefully.

2.4 An Additional Cycle rather than Rankine Cycle

Let us consider the discussion of the previous section in the P-V diagram as shown in “Fig.6” which describes GT-Cheng cycle system.

In the combustion chamber, H\textsubscript{2}O obtains the heat corresponding to the cycle 3-d-ex-ex’ from the air-gas cycle. Therefore, the original Brayton cycle 1-2-3-4 shrinks so that the point 4 moves toward the point 1 even when the additional heat \(\Delta B\) is given. Hence we have difficulties in separating the two cycle processes. Consequently, we had better name the cycle 3-d-ex-ex’ as an additional cycle rather than as a part of Rankine cycle.

Now it is easy to separate the additional work \(W_B\) from the original work \(W_A\) as well as the additional heat input \(Q_B\) from the original heat input \(Q_A\). Thus we are able to consider the original Brayton cycle’s efficiency \(\eta_A\) and the additional cycle’s efficiency \(\eta_B\).

\[
\eta_A = \frac{W_A}{Q_A} \tag{9}
\]
\[
\eta_B = \frac{W_B}{Q_B} \tag{10}
\]

Fig.5a Parallel cycles without additional fuel

Fig.5b Parallel cycles with additional fuel

Fig.6 P-V diagram (GT Cheng cycle)
The parallel combined efficiency $\eta_{com}$ is

$$\eta_{com} = \frac{W_A + W_B}{Q_A + Q_B} = \eta_B + \left(\frac{Q_B}{Q_A}\right)\eta_R$$

(11)

The total work $W_{com}$ and the power amplification factor $K_1$ are

$$W_{com} = W_A + W_B$$

(12)

$$K_1 = 1 + \frac{W_A}{W_B} = 1 + \left(\frac{Q_B}{Q_A}\right)\left(\frac{\eta_B}{\eta_A}\right)$$

(13)

The efficiency amplification factor $K_2$ is

$$K_2 = \frac{\eta_{com}}{\eta_A} = \frac{1 + \left(\frac{Q_B}{Q_A}\right)\left(\frac{\eta_B}{\eta_A}\right)}{1 + \left(\frac{Q_B}{Q_A}\right)}$$

(14)

"Eq.(11)" denotes that the higher the additional cycle's efficiency is, the higher the combined cycles efficiency is, and that the larger the additional heat input ratio is, the higher the combined cycle's efficiency is. In general, however, it is difficult to get higher $\eta_B$ and $\left(\frac{Q_B}{Q_A}\right)$ at the same time, because the higher $\eta_B$ comes from the small enthalpy difference between Point (d) and Point (c) in "Fig.5" while the larger $\left(\frac{Q_B}{Q_A}\right)$ ratio comes chiefly from the small enthalpy value of Point (c) in the same figure. Thus we might see a composition of an optimization problem.

The important point to be noted here is that the efficiency $\eta_B$ is far greater than the corresponding Rankine efficiency $\eta_R$. It is no wonder if $\eta_B$ exceeds 1. The Rankine efficiency $\eta_R$ is defined as

$$\eta_R = \frac{i_{c'} - i_{c}}{i_{e'} - i_e}$$

(15)

Where $i_{c'}$, $i_{c}$ and $i_e$ are the H2O's enthalpy value corresponding to Point (d'), Point (ex') and Point (a) of "Fig.5b" respectively. While $\eta_B$ is regarded as

$$\eta_B = \frac{i_{c'} - i_{c}}{i_{e'} - i_e}$$

(16)

Where $i_c$ is the enthalpy value for Point (c) in "Fig.5".

Accordingly,

$$\eta_B = \frac{i_{c'} - i_c}{i_{e'} - i_e}$$

(17)

Therefore $\eta_B$ can exceed 1 with large $\eta_R$ and large $i_c$.

In addition to the above context, the authors had better show the total efficiency $\eta_{cs}$ for the series combined system which is made of the cycle 1-2-3-4 plus the cycle a-b-b'-c-c' in "Fig.5a". The representation is

$$\eta_{cs} = \eta_A + k (1 - \eta_A)\eta_{RS}$$

(18)

Where $\eta_{RS}$ is the efficiency of the Rankine cycle a-b-b'-c-c'.

There are three chemical kinetic effect of the steam-injected combustion besides physical effects due to flame temperature reduction. This way of thinking was verified by means of numerical calculations by D.Zhao et al. [4].

The authors also expect this chemical kinetic effects in case of SCW diesel. Concerning physical effects, the authors' anticipation is placed on the almost-unchanged maximum temperature (on the spatial average) and decreased diffusion flame temperatures (reduction of locally higher temperature). In case of diesel engines, it is inevitable that the diffusion flame temperature exceeds the water-gas-reaction's freezing temperature (approx. 1400°C). However, if the diffusion flame temperature can be reduced and if the cycle's maximum temperature (space average) is kept at the original level, then the not-worsened efficiency and the reduced NOx formation are expected. Further, the authors think that the NOx formation is very sensitive to the excess temperature-increment over the freezing temperature.

3. SCW Diesel System

3.1 System Skeleton

"Fig.7" shows the constitution of the SCW diesel system which the authors have planned. Why not steam but SCW, why 35 MPa for the water pressure etc. are explained in the following sections.

$$\eta_{RS} = \frac{i_{c'} - i_c}{i_{e'} - i_e}$$

(19)

And the constant $k$ is approximately the product of recovery rate of waste exhaust gas energy and the efficiency of heat exchanger. When we imagine a back pressure steam turbine for the series combined system, then

$$\eta_B > \eta_R > \eta_{RS}$$

(20)

The designers of combined systems have to analyze and estimate the comparison between "Eq.(11)" and "Eq.(18)".

2.5 NOx Reduction

The NOx emission level of the gas turbines which are fueled with city gas is low from the first. The gas turbines, the output of which ranges several mega watts, emit exhaust gas of around 200ppm NOx (at O2 = 0%) in general. Once GT-Cheng cycle system is built by means of those gas turbines, NOx level is reduced roughly half. Here turbine inlet and outlet temperatures may be considered unchanged. This fact suggests there are some chemical kinetic effect of the steam-injected combustion besides physical effects due to flame temperature reduction. This way of thinking was verified by means of numerical calculations by D.Zhao et al. [4].

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The authors are now preparing the theory verification test by means of one cylinder engine. The authors consider that this principle-confirmation test is indispensable before the development of series-engine system is planned and started.
because the design of SCW diesel needs rather different approach from the conventional procedure. Here the theory or principle means the ground on which the designers could plan the system-detail.

The principal items of the diesel engine to be used for the theory verification are listed in Table 1.

<table>
<thead>
<tr>
<th>Item</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>bore/stroke</td>
<td>30[cm] / 42[cm]</td>
</tr>
<tr>
<td>speed</td>
<td>750min⁻¹</td>
</tr>
<tr>
<td>output</td>
<td>450kW / cyl.</td>
</tr>
<tr>
<td>firing press.</td>
<td>up to 25 MPa</td>
</tr>
<tr>
<td>mean effective press.</td>
<td>up to 2.5MPa</td>
</tr>
</tbody>
</table>

### 3.2 SCW-Fuel Ratio

In planning SCW diesel systems, the first thing to be deliberated is to estimate the equilibrium between input fuel and water. There are two viewpoints: how vast amount of water could be thrown into the cylinders and how minimum amount of water could be supplied from the heat-exchanger.

The answer clue for the former question seems to lie in the experience of NOx reduction technologies. “Table.2” shows a comparison of three typical methods: emulsified fuel, water injection and charge air humidification.

<table>
<thead>
<tr>
<th>No</th>
<th>Item</th>
<th>H₂O-fuel ratio (weight)</th>
<th>expected reduction</th>
<th>elements to be developed</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>fuel emulsification</td>
<td>approx. 0.5</td>
<td>~40%</td>
<td>fuel injection pump</td>
</tr>
<tr>
<td>2</td>
<td>water injection in cylinders</td>
<td>approx. 0.5</td>
<td>~40%</td>
<td>water injection system</td>
</tr>
<tr>
<td>3</td>
<td>charging air humidification</td>
<td>approx. 4~6</td>
<td>~60%</td>
<td>saturate humidity adding system</td>
</tr>
</tbody>
</table>

The water amounts of items (1) & (2) are rather small due to the mechanical restriction of hardware elements. The item (3) suggests considerably large amount of water could be thrown into the cylinders without flame quenching and fatal ignition delay provided that the humid-adding process is pertinent. Actually, the charge air of 50°C-60°C could contain a good deal of water. From this point of view, the authors aim at several times of water-fuel ratio, taking account NOx reduction effect into consideration.

In the next place, we have to pay attention to the recovery steam amount from the exhaust gas economizer of usual diesel plant. When utility steam (0.8MPa, ~200°C, h ≈ 2840 kJ/kg, v =0.247 m³/kg) is extracted from the exhaust gas of the diesels under consideration, the steam fuel ratio level is at most around 2.0-2.5. In planning SCW system, this ratio is a little bit unsatisfactory for the authors, but the volume amount is too large to be pushed into cylinders. Therefore the authors gave up steam injection from the beginning and adopted critical-water injection. It seems important for the SCW diesel designers to make the H₂O cycle process locus, in the cylinders and in the heat exchanger, pass away from the critical point and the gas-liquid phase-zone, because the H₂O properties such as specific heat, volume, enthalpy and so on vary steeply around the critical points, bringing potentially undesirable effects on the steadiness of the SCW system. Here, the steadiness means the stable state between the prime movers and the heat exchanger and/or between the air-gas cycle and the SCW cycle in the cylinders. Thus, adopting not steam injection but SCW injection with sufficiently high pressure was concluded. “Fig.8a” and “Fig.8b” illustrate P-V-T and P-T phase diagram (H₂O) respectively.
3.3 Design of Cycle Process

The authors have already stated that SCW diesel systems are not a mere hybrid system made of diesel engines and steam engines. If so, the next subject is to predict the cycle diagram in the cylinder of SCW engines. It is easy to chase the air-gas cycle process in the cylinder provided that necessary data (boost, heat release rate, polytropic indexes, air-gas characteristic data etc.) are supplied. Then the firing pressure, the indicated mean effective pressure, the expansion temperature and so on are calculated. Even in case that there is a superposing of \( \text{H}_2\text{O} \) injection (cf. "Fig.9") over the above air-gas cycle, it is not difficult to calculate the cycle process.

However, in solving the SCW cycle process, we have to control the maximum temperature and maximum pressure so as not to exceed those of the original air-gas cycle in principle. When the peak pressure exceeds the original one, the designers of the engine ought to recheck the strength and stiffness of all the engine parts and take measures if necessary. As the objectives of the development are not to design a fully new diesel, such work must be avoided. Furthermore, the higher the peak pressure the more the mechanical efficiency and the damage risks of bearings. This must be evaded too.

In the next place, the higher peak temperature makes the designers examine the thermal load endurance of the engine. In case of SCW diesels, however, the thermal load is not be derived from the increased mean-effective pressure which is enhanced by SCW and additional fuel, but the thermal load is to be derived from the original cycle itself.

By the above reasons, the peak pressure and temperature must not exceed the original values.

The authors here would like to emphasize that they are seeking SCW diesel not for the enhanced degree of constant volume but for the lateral expansion of P-V diagrams (cf. "Fig.10"). Here, the authors define the word "lateral expansion" as an enlargement of P-V diagrams not by enhancing the maximum pressure and/or the compression ratio but by increasing the degree of constant pressure or the cut-off ratio of the P-V diagrams.

Certainly a technology which is capable of affecting heat release rate becomes prerequisite. Although the research on the relation between heat release rate and injection rate/mode is still one of the modern and uncompleted studies in general, the authors has developed and are developing the electric fuel-injection valves of "variable rate shaping". That is, the authors are possessing a tool which can influence the heat release rate to some extent.

In this way, the subject becomes how the heat release rate of the original cycle process should be modified with SCW injection under the constraint conditions of peak pressure, peak temperature and expansion temperature. This cycle process calculation is not so easy as the conventional diesel cycle simulations which are mentioned in the above context. The simple and self-made calculation, the algorithm of which is clear and understandable, is desirable as it is most important to know the relative comparisons between the air-gas original cycle and SCW combined cycle.

The authors contrived an algorithm to solve the due change of heat release rate for the SCW diesel cycle process. In order to realize and estimate the lateral expansion of the P-V diagram as shown in "Fig.10", it is convenient to introduce two coordinate systems: \((P, V, T)\) coordinate system and \((\pi, \nu, \tau)\) coordinate system. The former coordinate is used for the original air-gas cycle, while the latter coordinate is for the mixed-binary-combined cycle. This image is illustrated in "Fig.11".

The variables in the two coordinates are summarized in "Table.5".

<table>
<thead>
<tr>
<th>variable</th>
<th>Point or infinitesimal vector</th>
<th>meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>(P/\pi)</td>
<td>(A/a)</td>
<td>pressure</td>
</tr>
<tr>
<td>(V/\nu)</td>
<td>do.</td>
<td>volume</td>
</tr>
<tr>
<td>(T/\tau)</td>
<td>do.</td>
<td>temperature</td>
</tr>
<tr>
<td>(\Theta/\Theta)</td>
<td>do.</td>
<td>crank angle</td>
</tr>
<tr>
<td>(N_a)</td>
<td>(A)</td>
<td>mol. (air)</td>
</tr>
<tr>
<td>(N_b)</td>
<td>(a^*)</td>
<td>mol. (H(_2)O)</td>
</tr>
<tr>
<td>(dP/d\pi)</td>
<td>(\Delta A/A ) (\text{aa}^2)</td>
<td>infinitesimal press.</td>
</tr>
<tr>
<td>(dV/d\nu)</td>
<td>do.</td>
<td>infinitesimal vol.</td>
</tr>
<tr>
<td>(dT/d\tau)</td>
<td>do.</td>
<td>infinitesimal temp.</td>
</tr>
<tr>
<td>(dN_a)</td>
<td>—</td>
<td>0 (mol)</td>
</tr>
<tr>
<td>(dN_b)</td>
<td>—</td>
<td>infinitesimal mol. (H(_2)O)</td>
</tr>
<tr>
<td>(dQ_a)</td>
<td>—</td>
<td>infinitesimal heat input (original)</td>
</tr>
<tr>
<td>(dQ_a)</td>
<td>—</td>
<td>infinitesimal heat input (SCW cycle)</td>
</tr>
</tbody>
</table>
All the variables of the original air-gas cycle can be assumed to be known: $P$, $V$, $T$, $\theta$, $N_a$ etc. And the heat release rate $dQ/d\theta$ for the original cycle is also known. The mass amount $m_a$ and/or molecular amount $N_a$ is kept constant after the inlet valves are closed. There are, for the time being, five variables ($\pi$, $v$, $\tau$, $dQ/d\theta$, $N_b$) to be considered for the mixed binary cycle.

Among these five variables, the molecular amount $N_b$ as to SCW is known because SCW's timing/duration and injection rate are planned values. The infinitesimal heat input $dQ$ remains an unknown variable to be solved. We need four equations to determine four unknown variables: $d\pi$, $d\tau$, $d\tau$ and $dQ$. Two of the equations are brought in by the mass conservation law (equation of state) and the energy conservation law (equation of the second law) as follows.

\[ \nu d\pi + \pi d\nu = (N_a+N_b)R_u \rho d\tau + R_u dN_b \]  
\[ (N_a\nu_a+N_b\nu_b) d\tau + \pi d\nu = dQ + P_a \rho dN_b - (\tau-T_a) c_{v_a} dN_b \]  

Where $R_u$, $c_{v_a}$ and $c_{v_b}$ are universal gas constant, specific heat at constant volume (air-gas) and specific heat at constant volume (H$_2$O) respectively. And $P_a$, $\nu_a$, $T_a$ are the pressure, specific volume and temperature of SCW supplied into this cylinder, respectively.

In “Eq.(22)”, the compressibility factor for the SCW gas is omitted because the authors assume that the SCW gas diffused into the cylinder-space under sufficiently high temperature-field can be treated approximately as an ideal gas.

It should be noted that the authors are paying attention to the pushing-in energy of SCW, but are neglecting the flowing-in velocity-energy of SCW. Therefore the momentum conservation law (equation of motion) does not appear here.

The two remaining equations are given by regarding the original air-gas cycle process as a template. For the sake of simplicity, let us consider the process after the top dead center. If, during heat release, the following relations are kept, then the peak pressure and temperature of SCW cycle never exceed those of the original cycle.

\[ d\pi = dP \]  
\[ d\tau = dT \]  

We may assume more general relations such as follows, if we want to control or adjust the resulting peak pressure and temperature.

\[ d\pi = f_{unc1}(dP) \]  
\[ d\tau = f_{unc2}(dT) \]  

Here, the authors note that the linear type of functions for $f_{unc1}$ and $f_{unc2}$ are convenient to manage the peak pressure and temperature. Thus, the unknown variables $d\nu$ and $dQ$ can be solved, giving a laterally expanded $\pi$-$\nu$ (pressure-volume) diagram as well as a new heat release rate $dQ/d\theta$.

Consequently we get the separation of the original cycle’s work $W_a$ from the original cycle’s work $W_a$ as well as the separation of the additional heat input $Q_b$ from the original heat input $Q_a$. “Fig.12” illustrates the image of the heat release difference between the two coordinates.

It should be noted additionally that the variables $\theta$ and $\pi$ belong to different coordinates, so they are not equal. The transformation of $\theta$ into $\pi$ and vice versa is necessary during the pursuit of infinitesimal steps.

![Fig.12 Comparison between heat release rates](image)

### 3.5 SCW-Fuel Ratio Equilibrium

In the previous section, it was discussed how the SCW cycle (the mixed binary cycle in the cylinder) should be controlled when the system goes to an equilibrium state. Without this standpoint of view, the plan, design and experiment of SCW diesel system are unmanageable, because the shortage of constraint conditions never guarantees the system’s equilibrium.

The calculation of the equilibrium between the SCW diesel cycle and the heat-exchanger is carried out in accordance with the procedure flow shown in “Fig.13”.

Original air-gas cycle and initial SCW injection timing/mode are given.

Calculation of $Q_b$ (SCW cycle’s heat release rate) attended with SCW injection.

Increase of temp. before turbine, exhaust gas amount, exhaust gas enthalpy before heat-exchanger.

By-passing excess energy before turbine toward heat exchanger.

Increase of temp. and amount before heat-exchanger

Asymptotic increasing convergence of state variables

Confirmation $\theta_b$, $\nu_{com}$, heat balance etc.
“Fig.14” illustrates the excess energy (before the turbine of super-charger) which is produced by the SCW cycle. It should be noted that the SCW researchers have to count in, from the first, the margin: what degree of superfluity (marginal) as to the turbine inlet temperature and the exhaust valve’s passage-area the engine can offer. This counting-in is necessary for the designers to minimize the modification of cylinder covers and super-charger’s specifications.

4. View and Test Preparation

As far as the authors have investigate the potential of SCW diesel systems along the various views stated above, the system seems promising especially as to power and efficiency amplification ($K_1$ & $K_2$). The authors current targets for $K_1$ & $K_2$ are 1.15 and 1.08 respectively. These moderate objectives were set in consideration of the margin around exhaust valves (passage area, armor metal’s strength etc.). Yet the targets are attractive.

Regarding NOx, the authors are expecting the chemical kinetic effect of SCW injection besides the physical effect. Although, in the SCW cycle’s design, it is a principle to keep the peak temperature, the authors are anticipating an operation mode in which the weight of air-gas work is lessened, bringing lowered (spatial) peak temperature.

With an operation mode for NOx reduction, the authors are targeting on 66% removal of NOx.

The authors have stated that the injection valves of “variable rate shaping” are necessary from the view points of “cycle process design”. These valves are of poppet type, and have the function of proportional action. The acting principle is illustrated in “Fig.15”.

5. Conclusion

Starting from the recurrence equation for system equilibrium, the authors have discussed the thermo-dynamical equilibrium of SCW diesel system or Diesel-Cheng cycle, extracting the potential capability of the system as to a power and efficiency amplifier. The principles and theories which the authors have brought-in are indispensable for the plan, design and verification. The authors have pointed out the possibility and its reasons about NOx reduction, suggesting the chemical kinetic effects as well as the physical effects. Some important items to be paid attention to in experiments have been also presented.

The authors are hoping for the next opportunity to show further views and experimental results apart from “Steam Injection Diesel (STID)” concept [5].

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7. References