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A NEW EQUATION TO DESCRIBE COOLING LOSS IN HYDROGEN COMBUSTION ENGINES WHICH WAS DEVELOPED FROM THE EQUATION FOR TURBULENT HEAT TRANSFER OF PIPE FLOWS

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ABSTRACT

Thermal efficiency of a hydrogen combustion engine is sometimes lower than that of conventionally fueled engines, because of a larger heat transfer from burning gas to the combustion chamber wall due to unique combustion characteristics for hydrogen. The empirical correlation is often used to calculate the heat transfer in engines. However, previous research by the author has shown that the widely used equation cannot be properly applied to the hydrogen combustion. This research tries to develop a new equation to describe the cooling loss in hydrogen combustion engines from a turbulent heat transfer equation for pipe flows.

Keywords *: Internal Combustion Engine, Hydrogen, Cooling Loss, Heat Transfer, Combustion*

1. INTRODUCTION

A higher burning velocity and a shorter quenching distance for hydrogen as compared with hydrocarbons bring a larger heat transfer from burning gas to the combustion chamber wall in internal combustion engines [1]. Because of a high cooling loss fraction by the large heat transfer, thermal efficiency of a hydrogen combustion engine is sometimes lower than that of a conventionally fueled engine [2].

Empirical equation for total heat transfer from burning gas to the combustion chamber walls are often used to calculate the cooling loss in engines [3-7]. The equations have been developed from turbulent heat transfer equations for pipe flows by correlating with experimental data of hydrocarbon combustion. However, engines fueled with hydrogen have larger changes in the composition and the thermophysical properties of the in-cylinder gas due to its larger burning velocity than hydrocarbon [8-9]. Therefore, it is necessary to reassess the process of the correlation. Moreover, previous researches by the author et al has shown that adjusting the velocity term enables the widely used equation by Woschni calculate better results for hydrogen combustion [10-12].

This research tries to develop a new equation to describe the cooling loss in hydrogen combustion engines based on a turbulent heat transfer equation for pipe flows. The process especially focuses on treatments of thermophysical properties and representative velocity of the in-cylinder gas.

2. EXPERIMENT

Figure 1 shows the experimental results of hydrogen combustion. The results were obtained in a 4-cylinder 4-stroke spark-ignition engine (bore x stroke: 85 x 88mm, compression ratio: 8.5). Hydrogen was measured with a mass-flow meter (Oval F203S) and continuously supplied to the intake pipe. Engine speed was fixed at 1500rpm, excess air ratio was at 1.0 and volumetric efficiency was at 35% including the fuel gas. In-cylinder pressure data were measured with a piezoelectric type pressure transducer (AVL GM12D) installed in the cylinder head as shown in Figure 2. Pressure data for 50 cycles were averaged and used to calculate the apparent rate of heat release and the in-cylinder gas temperature. The apparent rate of heat release $dQ/d\theta$ was calculated with the following equation.

$$dQ/d\theta=(VdP/d\theta+\gamma PdV/d\theta)/(\gamma-1)-PV/(\gamma-1)^2 d\gamma/d\theta \qquad (1)$$

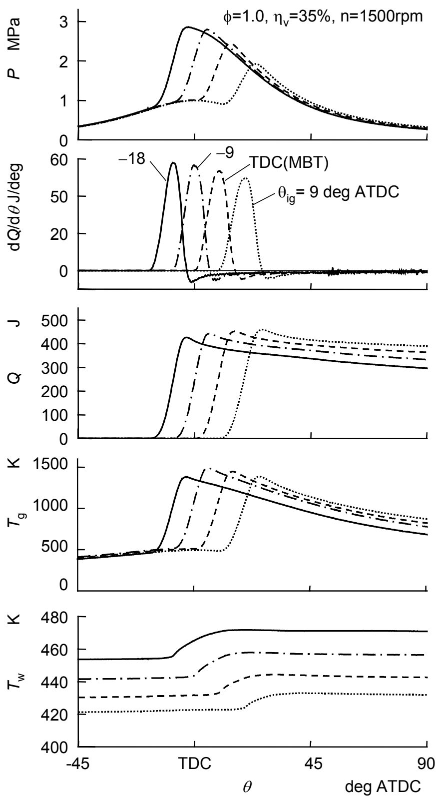

Fig.1 Stoichiometric combustion of hydrogen with different ignition timings

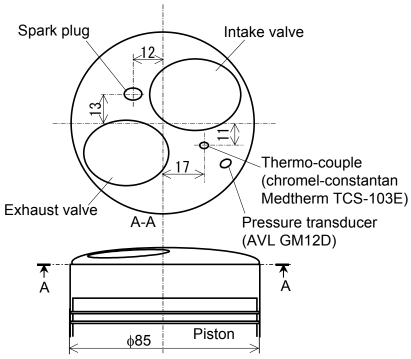

Fig.2 Combustion chamber of the tested engine

The cumulative apparent heat release Q was obtained by the integration of the apparent rate of heat release $dQ/d\theta$. Instantaneous temperature on the combustion chamber wall surface was measured with a thin-film type thermocouple (Medtherm TCS103E). By using these data of pressure and temperature, this research tries to develop a new heat transfer equation for hydrogen combustion.

3. DISCUSSION

3.1 Heat transfer analysis by heat release rate

The apparent rate of heat release $dQ/d\theta$ calculated with the measured in-cylinder pressure is influenced by the cooling loss and can be described with the real rate of heat release $dQ_B/d\theta$ and the rate of cooling $dQ_C/d\theta$ as follows.

$$dQ/d\theta = dQ_B/d\theta - dQ_C/d\theta \quad (2)$$

For a cycle, the cumulative cooling loss Q_C is described with the cumulative real heat release Q_B and the cumulative apparent heat release Q.

$$Q_C = Q_B - Q \quad (3)$$

The cumulative real heat release Q_B corresponds to a product of supplied fuel heat Q_{fuel} and combustion efficiency η_u. The combustion efficiency can be calculated from the exhaust gas composition. Therefore, the cumulative cooling loss Q_C for a cycle can be obtained from experiments [2]. On the other hand, there is a following relation among the rate of cooling $dQ_C/d\theta$, the heat transfer coefficient α, the engine speed n, the combustion chamber wall surface area S, the gas temperature T_g and the wall temperature T_w.

$$dQ_C/d\theta = 6^{-1} n^{-1} S \alpha (T_g - T_w) \quad (4)$$

$$\alpha = 6n S^{-1} (T_g - T_w)^{-1} dQ_C/d\theta \quad (5)$$

Figure 3 shows the rate of cooling $dQ_C/d\theta$ estimated from the experimental results shown in Figure 1 by the method using the Wiebe function [2,13]. The figure also shows the heat transfer coefficient α calculated by Equation (5) using the rate of cooling along with the other values in the equation. Here, the in-cylinder gas temperature calculated from pressure analysis was used as the gas temperature T_g, the instantaneous wall temperature measured with the thermo-couple was as the wall temperature T_w. The figure shows the rate of cooling and the heat transfer coefficient for after the beginning of combustion, because the accuracy of introducing them decreases at conditions with smaller temperature differences $T_g - T_w$.

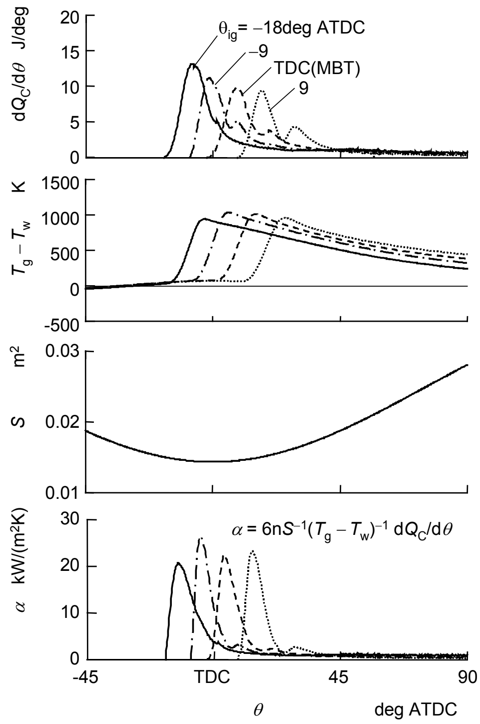

Fig.3 Rate of cooling and heat transfer coefficient derived from experiment by heat release analysis

3.2 Turbulent heat transfer equation

Some empirical equations have been proposed to estimate the cooling loss by the heat transfer from burning gas to the combustion chamber wall in internal combustion engines. Many of those are based on the turbulent heat transfer equations for pipe flows and derived through some assumptions and engine experiments using conventional hydrocarbon fuels. However, it may be necessary to reassess the introduction process of the equation for calculating the cooling loss in hydrogen combustion, because the composition and thermophysical properties of in-cylinder gas change more rapidly in hydrogen combustion than in combustion of the conventional hydrocarbon fuels [8-9].

Following relation among the Nusselt number Nu, the Reynolds number Re and Prandtl number Pr is often used to describe the heat transfer in pipe flows.

$$Nu = C\, Re^{m} Pr^{n} \quad (6)$$

The Nusselt number Nu and the Reynolds number Re can be written using the representative length d, the thermal conductivity λ, the representative velocity u, the kinetic viscosity ν as follows.

$$Nu = \alpha \mathrm{d} / \lambda \quad (7)$$

$$Re = u \mathrm{d} / \nu \quad (8)$$

By introducing these to Equation (6), the heat transfer coefficient α is described as follows.

$$\alpha = \mathrm{C}\, \mathrm{d}^{\mathrm{m}-1} \lambda\, \nu^{-\mathrm{m}} u^{\mathrm{m}} Pr^{\mathrm{n}} \quad (9)$$

Here, the influence of changes in the Prandtl number Pr on the heat transfer coefficient α is quite small even in hydrogen combustion [9]. The Prandtl number is treated as constant.

$$\alpha = \mathrm{C}\, \mathrm{d}^{\mathrm{m}-1} \lambda\, \nu^{-\mathrm{m}}\; u^{\mathrm{m}} \quad (10)$$

Turbulent heat transfer equations for gas flows in pipes, such as the equation by Kays et al, usually employ m=0.8 for the exponent for the Reynolds number.

$$\alpha = \mathrm{C}\, \mathrm{d}^{-0.2} \lambda\, \nu^{-0.8} u^{0.8} \quad (11)$$

While most of the heat transfer equations for internal combustion engines use the mean piston velocity $\mathrm{C_m}$ for the representative velocity u, the mean piston velocity is constant for a constant engine speed condition. The representative length d is also constant when the cylinder bore is employed as the length. Therefore, trends of calculated heat transfer coefficients are dominated by trends of thermal conductivity λ and the kinetic viscosity ν.

$$\alpha = \mathrm{C}\, \lambda\, \nu^{-0.8} \quad (12)$$

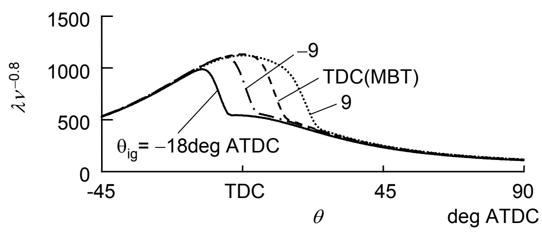

Fig.4 $\lambda\nu^{-0.8}$ of in-cylinder gas during combustion

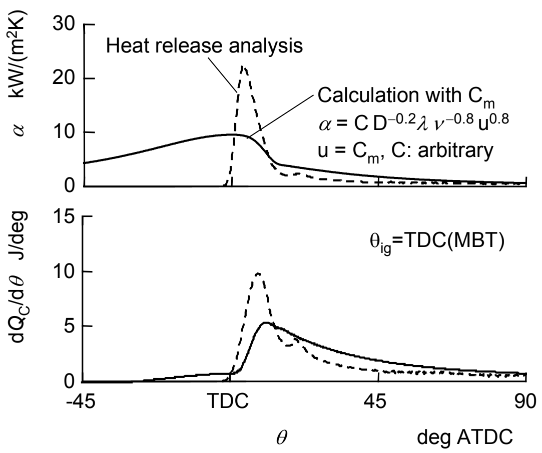

Fig.5 Heat transfer coefficient and rate of cooling calculated by turbulent heat transfer equation with mean piston speed

Figure 4 shows changes in $\lambda\nu^{-0.8}$ of the in-cylinder gas for the experimental results shown in Figure 1 for different ignition timings. Figure 5 shows the heat transfer coefficient α calculated by Equation (11) with the mean piston velocity and the cylinder bore for the optimum ignition timing. The rate of cooling $\mathrm{d}Q_\mathrm{C}/\mathrm{d}\theta$ by Equation (4) is also shown in the figure. The broken lines are the heat transfer coefficient α and the rate of cooling $\mathrm{d}Q_\mathrm{C}/\mathrm{d}\theta$ obtained by the heat release analysis of the experimental data shown in Figure 3. The thermal conductivity λ and the kinetic viscosity ν were estimated from the mean temperature and the composition of the in-cylinder gas [7,14]. Figure 5 indicates that the heat transfer coefficient, which was calculated by the turbulent heat transfer equation with the mean piston velocity as the representative velocity, is apparently inadequate. This can be due to that an increase in the convection generated by combustion cannot be expressed by the mean piston velocity which is constant even at the combustion period. Influence of the convection by combustion on the heat transfer can be larger for the engines fueled with hydrogen which has the higher flame propagation velocity than hydrocarbons [15]. It is especially important to properly treat the representative velocity during combustion when calculating the cooling loss in hydrogen engines.

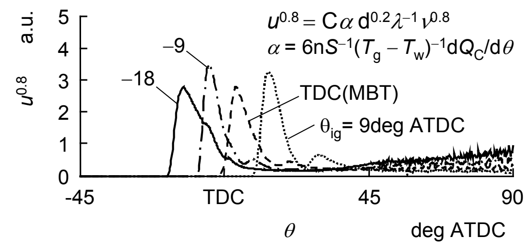

Fig.6 Representative velocity of in-cylinder gas derived from experiment

3.3 Mean gas velocity term

From Equation (11) the velocity term $u^{0.8}$ in the pipe flow heat transfer equation can be described as follows.

$$u^{0.8} = \mathrm{C}\, \mathrm{d}^{0.2} \lambda^{-1} \nu^{0.8} \alpha \quad (13)$$

This equation and the heat transfer coefficient α from the rate of cooling in Equation (5) give the following relation.

$$u^{0.8} = \mathrm{C}\, \mathrm{d}^{0.2} \lambda^{-1} \nu^{0.8}\, \mathrm{n}\, S^{-1} (T_\mathrm{g} - T_\mathrm{w})^{-1}\, \mathrm{d}Q_\mathrm{C} / \mathrm{d}\theta \quad (14)$$

Figure 6 shows the velocity term $u^{0.8}$ obtained by Equation (14) from the experimental data. The velocity term derived from experiments, which rapidly increases with combustion and then decreases, is largely different from the trend of the mean piston velocity. The influence of the increased representative velocity during the combustion should be included in calculating cooling losses of hydrogen combustion. For this reason, this paper tries to describe the increase in the velocity by employing the heat release rate that corresponds to the burned mass per time.

The rate of heat release $\mathrm{d}Q/\mathrm{d}t$ can be described with a heating value of a unit mass mixture Hu_m and a mass burning rate of mixture dm_b/dt.

$$dQ/dt = Hu_m\, dm_b/dt \tag{15}$$

By using a volumetric burning rate dV_b/dt and the density ρ of the mixture, above equation is rewritten as follows.

$$dQ/dt = Hu_m\, \rho\, dV_b/dt \tag{16}$$

Here, this paper defines the characteristic velocity of the in-cylinder gas motion by combustion u_b with the volumetric burning rate dV_b/dt and the cylinder bore D as follows.

$$dV_b/dt = 0.25\, \pi\, D^2\, u_b \tag{17}$$

Equations (16) and (17) give Equation (18).

$$dQ/dt = 0.25\, \pi\, D^2\, Hu_m\, \rho\, u_b \tag{18}$$

And the characteristic velocity becomes the following form.

$$u_b = 4\, \pi^{-1}\, D^{-2}\, Hu_m^{-1} \rho^{-1}\, dQ/dt \tag{19}$$

On the other hand, the in-cylinder gas motion is influenced not only by the combustion but also by the piston motion. This paper describes the representative velocity with the characteristic velocity by combustion u_b and the mean piston velocity C_m as follows.

$$u = C_m + C\, u_b \tag{20}$$

$$u = C_m + 4\, \pi^{-1}\, C\, D^{-2}\, Hu_m^{-1}\, \rho^{-1}\, dQ/dt \tag{21}$$

By using this velocity for the representative velocity u and the cylinder bore D for the representative length d in Equation (11), the heat transfer coefficient α is described as follows.

$$\alpha = C_1\, D^{-0.2}\, \lambda\, \nu^{-0.8}\, (C_m + C_2\, D^{-2}\, Hu_m^{-1} \rho^{-1}\, dQ/dt\,)^{0.8} \tag{22}$$

Figure 7 shows the new velocity term by Equation (21) for the experimental results in Figure 1. Here, the apparent rate of heat release was used for dQ/dt in Equation (21). The rate of cooling by the heat release analysis in Section 3.1 is also shown for a comparison in the figure.

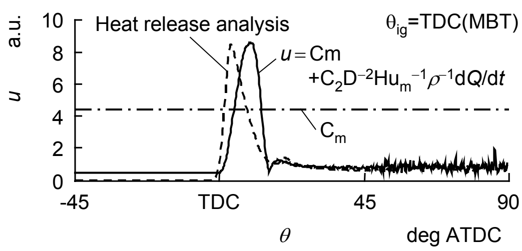

Fig.7 New velocity term employing heat release rate

Figure 8 shows the heat transfer coefficient α and the rate of cooling $dQ_C/d\theta$ which were calculated by Equation (22) with the new velocity term. Trends of these are similar to those in Figure 3 by the heat release analysis of experimental data. The calculated results in Figure 8 using the new velocity term including the effect of combustion is better than the results in Figure 5 using just the mean piston velocity.

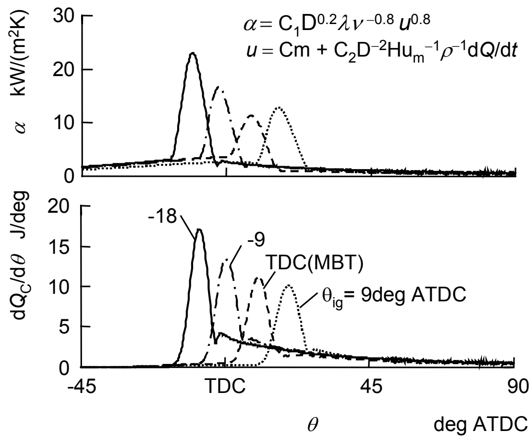

Fig.8 Heat transfer coefficient and rate of cooling calculated by turbulent heat transfer equation with the new velocity term

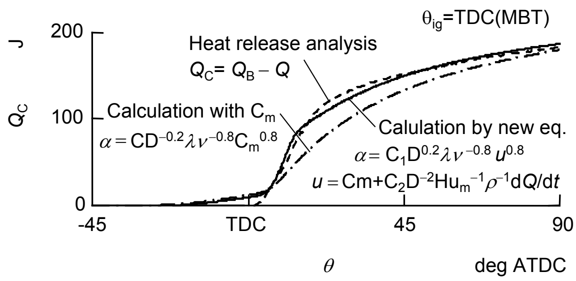

Fig.9 Cumulative cooling loss calculated by the new heat transfer equation

Figure 9 shows calculated cooling losses by the turbulent heat transfer equation with just the mean piston velocity C_m and the new velocity term including the effect of combination. The cooling loss by the heat release analysis is also shown in the figure. Here, constants in Equations (11) and (22) were determined so to satisfy a following relation at the exhaust valve opening timing.

$$Q + Q_C\ = \eta_u Q_{fuel} \tag{23}$$

The figure indicates that the calculation with the new velocity term is closer to the heat release analysis of the experimental data.

Figure 10 shows cumulative real heat releases calculated by the new equation for different ignition timings. Results by Woschni's equation are also shown in the figure. The cooling losses calculated by Woschni's equation were multiplied by a correction constant to satisfy Equation (23) at the exhaust valve opening timing. Regardless the ignition timing, the new equation brings more proper results for hydrogen combustion engine compared to Woschni's equation.

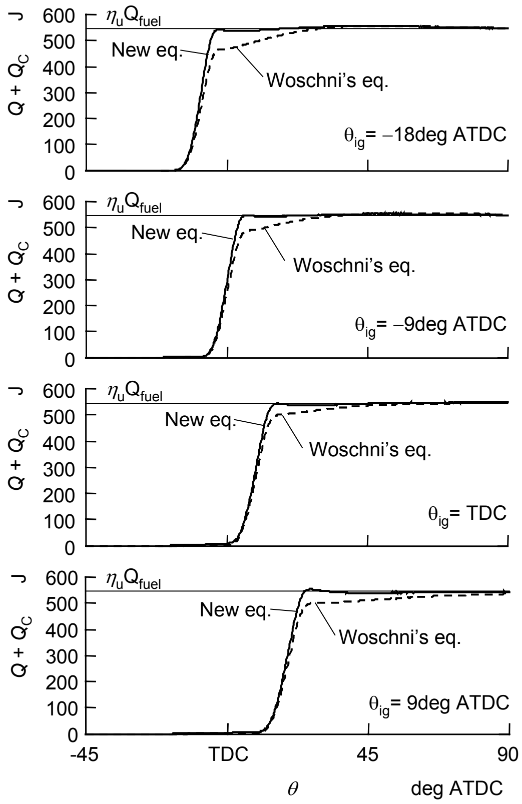

Fig.10 Cumulative real heat release calculated by the new heat transfer equation for different ignition timings

4. CONCLUSIONS

(1) This research calculated the heat transfer coefficient from the rate of cooling, which was estimated by the apparent rate of heat release and Wiebe function, in a hydrogen combustion engine. The representative velocity of the in-cylinder gas was derived from the turbulent heat transfer equation for pipe flows by using the obtained heat transfer coefficient.

(2) The turbulent heat transfer equation employing the mean piston velocity as the representative velocity cannot express an increase in the convective heat transfer during combustion. It is important to include the effect of the convection generated by combustion in calculation of cooling losses in hydrogen combustion due to its large burning velocity.

(3) A new velocity term, which includes the effect of both the piston motion and the combustion, enables the turbulent heat transfer equation to calculate proper cooling losses for hydrogen combustion.

(4) This paper proposes a following equation for calculating the cooling loss in hydrogen combustion engines.

$$\alpha = \mathrm{C_1\,D^{-0.2}}\,\lambda\,\nu^{-0.8}\,(\mathrm{C_m} + \mathrm{C_2\,D^{-2}\,Hu_m}^{-1}\rho^{-1}\,\mathrm{d}Q/\mathrm{d}t\,)^{0.8}$$

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NOMENCLATURE

ϕ	equivalence ratio
η_v	volumetric efficiency
n	engine speed (rpm)
θ	crank angle (degCA)
θ_{ig}	ignition timing (degCA ATDC)
P	in-cylinder pressure (Pa)
V	in-cylinder volume (m^3)
T_w	combustion chamber wall temperature (K)
T_g	in-cylinder gas mean temperature (K)
γ	specific heat ratio
η_u	combustion efficiency
Q_{fuel}	heating value of mixture per cycle (J)
Q	cumulative apparent heat release per cycle (J)
$dQ/d\theta$	apparent rate of heat release (J/degCA)
dQ/dt	apparent rate of heat release (J/s)
Q_B	cumulative real heat release (J)
$dQ_B/d\theta$	real rate of heat release (J/degCA)
Q_C	cumulative cooling loss per cycle (J)
$dQ_C/d\theta$	rate of cooling (J/degCA)
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
α	heat transfer coefficient ($W/(m^2 \cdot K)$)
u	representative velocity (m/s)
λ	thermal conductivity ($W/(m \cdot K)$)
ν	kinetic viscosity (m^2/s)
ρ	density (kg/m^3)
d	representative length (m)
D	cylinder bore (m)
S	combustion chamber surface area (m^2)
C_m	mean piston velocity (m/s)
u_b	characteristic gas velocity by combustion (m/s)
dm_b/dt	mass-burning rate of mixture (kg/s)
dV_b/dt	volumetric burning rate of mixture (m^3/s)
Hu_m	heating value of unit mass mixture (J/kg)
C, C_1, C_2	constants
m, n	coefficients
TDC	top dead center
MBT	optimum ignition timing for thermal efficiency