A Study on the Simulation of Turbocharger Matching through a Performance Characteristics of Compressor and Turbine
Abstract

Nomenclature

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Abstract

Basic methods to increase power output of internal combustion reciprocating engine is increasing of engine revolutions and engine displacement. But they have limitations because of reduction of volumetric efficiency, increase of mechanical losses, noise, vibration, size and weight of engine.

For this reason, most of engines have been coupled with a turbo charging system in order to increase engine power. Turbocharging can be defined as the introduction of air into an engine cylinder at a density greater than atmosphere. This allows a corresponding increase in the fuel that can be burned hence raises the available power output.

The engine is designed for variable speed and the operation will usually exhibit some deterioration in performance both at extreme low and high speed. However, the useful speed range can be wide, since engine is well suited to cater for a wide range of mass flow rate.

The performance of turbocharger is directly dependent upon the gas angle at entry of the impeller, diffuser of compressor and turbine rotor. The blade angles are set to match these gas angle, but a correct match will only be obtained when the mass flow rate is correct for a specified rotor speed. Therefore turbocharger is not well suited for operation over a wide flow range.

Turbocharger is not ideally suited to operate in conjunction with a engine.
So the combination of diesel engine and turbocharger must be planned with care.

The objective of turbocharger matching is to fit a turbocharger with the most suitable characteristics of an engine in order to obtain the best overall performance. Matching of the correct turbocharger to a diesel engine is very important and is vital for successful operation of a turbocharged diesel engine. It will principally be governed by required performance with engine.

Simulation program used for the optimum calculation of turbocharger matching is very effective method to estimate turbocharged diesel engine performance.

In this paper the author has studied a calculation of turbocharger matching for 4 stroke automotive diesel engine and marine diesel engine through development of simulation program by using performance characteristics of turbocharger, and has estimated effects of efficiency, size of turbine and fuelling on the engine and turbocharger.

It was assured that simulation results agreed well with experimental results of thermodynamic states at turbocharger and intake, exhaust manifold, and yield correct tendencies of estimation according to efficiency, size of turbine and fuelling.


**Nomenclature**

**AFR**: Air / Fuel Ratio

\[ b_e : \text{Brake specific fuel consumption} \quad \text{[g/PS\textsuperscript{h}]} \]

\[ C_p : \text{Specific Heat at constant pressure} \quad \text{[kJ/kg\textsuperscript{K}]} \]

\[ C_v : \text{Specific Heat at constant volume} \quad \text{[kJ/kg\textsuperscript{K}]} \]

\[ H_l : \text{Lower calorific value} \quad \text{[kcal/kg]} \]

\[ N : \text{Revolutions Per Minute} \]

\[ \dot{m} : \text{Mass Flow Rate} \quad \text{[kg/s]} \]

\[ P : \text{Pressure} \quad \text{[mmHg]} \]

\[ \Delta P : \text{Pressure Loss} \quad \text{[mmHg]} \]

\[ P_{me} : \text{Brake mean effective pressure} \quad \text{[kg/cm}\textsuperscript{2} \text{]} \]

\[ P_e : \text{Brake Power} \quad \text{[PS]} \]

\[ Q : \text{Quantity} \quad \text{[kg]} \]

\[ R : \text{Gas constant} \quad \text{[kJ/kg\textsuperscript{K}]} \]

\[ T : \text{Temperature, Torque} \quad \text{[K, kg\textsuperscript{m}]} \]

\[ V : \text{Volume} \quad \text{[m}\textsuperscript{3} \text{]} \]

\[ W : \text{Work done} \quad \text{[kJ]} \]

\[ \gamma : \frac{C_p}{C_v} \]

\[ \varepsilon : \text{Intercooler Effectiveness} \]

\[ \eta : \text{Efficiency} \]
\( \mu \) : Dynamic Viscosity \[ \text{[kg/m}^\circ \text{s]} \]
\( \rho \) : Density \[ \text{[kg/m}^3 \text{]} \]
\( \pi_c \) : Compressor Pressure Ratio
\( \pi_r \) : Turbine Expansion Ratio

**Subscripts**

\( a \) : Air

\( AC \) : Air Cleaner

\( ATM \) : Atmosphere

\( 1C \) : Compressor Inlet

\( 2C \) : Compressor Outlet

\( 3 \) : Intake Manifold

\( 4T \) : Turbine Inlet

\( 5T \) : Turbine Outlet

\( C \) : Compressor

\( CA \) : Corrected Air

\( CEX \) : Corrected Exhaust Gas

\( ex \) : Exhaust Gas

\( f \) : Fuel

\( II \) : Intercooler

\( MUF \) : Muffler

\( T \) : Turbine
1

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ÀåÀû´çÇÑÆ¯¼ºÀ»°¡Áø°ú±Þ±â¸ÅĪÀǸñÀûÀº±â°üÀ¸·ÎºÎÅÍÃÖ°íÀǼº´ÉÀ»È¹µæÇϱâÀ§Çرâ°ü¿¡°¡

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(waste gate valve)

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- 2 -
2

(Mechanical Driven Supercharger) (Exhaust Gas Turbocharger) (Comprex Supercharger) (Hyperbar Supercharging)
2.1

2.1.1

\[ m_c \ T_1 \ C_{pt} \left( \frac{\gamma - 1}{\gamma} - \frac{1}{\eta_c} \right) = m_t \ T_4 \ C_{pt} \ \eta_t \left( 1 - \left( \frac{1}{R_i} \right)^{\frac{\gamma - 1}{\gamma}} \right) \]

**Basic Turbocharger Energy Balance Equation**

**Fig. 2-1** Schematic diagram of a turbocharged four-stroke diesel engine with the wastegate valve
Fig. 2-2 Schematic diagram of a turbocharged two-stroke diesel engine
2.1.2...

Fig. 2-3 Components of centrifugal compressor
Fig. 2-3

Fig. 2-3

\( \text{Trim} = \left[ \frac{D_{\text{IND}}}{D_{\text{TIP}}} \right]^2 \times 100 \)

**Fig. 2-4** Compressor trims, machined from one casting
Fig 2-4 (Trim Ratio) 

The trim ratio is a measure of the ratio of the blade pitch to the blade chord. It is defined as the ratio of the blade pitch to the blade chord. The trim ratio is a key parameter in the design of a turbomachinery blade. It affects the blade's lift and drag characteristics, and thus its performance. A higher trim ratio typically results in higher lift and lower drag, but also higher flow losses. The trim ratio is usually set during the design phase of the blade and is adjusted during operation to optimize the blade's performance.

Fig. 2-4 (surge line) 

The surge line is a critical parameter in the design of a turbomachinery blade. It is the boundary between the stable and unstable operating regions of the blade. The surge line is typically defined as the point at which the blade's efficiency drops significantly. The surge line is a function of the blade's geometry and the operating conditions. It is usually determined during the design phase of the blade and is adjusted during operation to optimize the blade's performance.

Fig. 2-3 (chock line) 

The chock line is a critical parameter in the design of a turbomachinery blade. It is the boundary between the stable and unstable operating regions of the blade. The chock line is typically defined as the point at which the blade's efficiency drops significantly. The chock line is a function of the blade's geometry and the operating conditions. It is usually determined during the design phase of the blade and is adjusted during operation to optimize the blade's performance.

Fig. 2-4 (Diffuser throat) 

The diffuser throat is a critical parameter in the design of a turbomachinery blade. It is the boundary between the main flow and the diffuser flow. The diffuser throat is typically defined as the point at which the blade's efficiency drops significantly. The diffuser throat is a function of the blade's geometry and the operating conditions. It is usually determined during the design phase of the blade and is adjusted during operation to optimize the blade's performance.
2.1.3

Fig. 2-5 Intercooler effectiveness

\[ \varepsilon = \frac{\text{Actual Temp. Drop}}{\text{Maximum Possible Temp. Drop}} = \frac{T_{2a} - T_{2b}}{T_{2a} - T_w} \]
Fig. 2-6 Typical performance of an air-to-water intercooler

2.1.4 Fig. 2-7 (Radial Turbine)
Fig. 2-7 Components of a radial flow turbine

\[ \text{Trim} = \left[ \frac{D_{EXD}}{D_{TIP}} \right]^2 \times 100 \% \]

Fig. 2-8 Trim (Trim Ratio) A/R, A/R, A/R, A/R, A/R, A/R.
Fig. 2-8 Turbine trims and volutes, machined from one casting
2.1.5 폐기물 게이트 밸브 (Waste Gate Valve)

위치의 대략적인 위치를 나타내는 측면도(세부 사양은 제외)의 Fig 2-1을 통해 볼 수 있듯이, 폐기물 게이트 밸브 (by-pass)는 [Waste gate valve]와 같이 사용된다.

2.2 폐기물 게이트 밸브 (Waste gate valve)
2.2.1 Fig. 2-1

～3 kg/cm² ～ 3 kg/cm² ～ 3 kg/cm² ～ 3 kg/cm².
2.2.2 结果

...结果表明...，Fig. 2-2 中...结果表明...。

...结果表明...，Fig. 2-2 中...结果表明...。
3. 측정 결과

측정 결과는 다음과 같습니다. 세부 내용은 아래 기록에 있으며, 이 결과를 기반으로 다음과 같은 점수를 산출하였습니다.

(1) 측정 결과는 다음과 같습니다. 세부 내용은 아래 기록에 있으며, 이 결과를 기반으로 다음과 같은 점수를 산출하였습니다.

(2) (1) 측정 결과는 다음과 같습니다. 세부 내용은 아래 기록에 있으며, 이 결과를 기반으로 다음과 같은 점수를 산출하였습니다.

(3) (2) 측정 결과는 다음과 같습니다. 세부 내용은 아래 기록에 있으며, 이 결과를 기반으로 다음과 같은 점수를 산출하였습니다.

(4) 측정 결과는 다음과 같습니다. 세부 내용은 아래 기록에 있으며, 이 결과를 기반으로 다음과 같은 점수를 산출하였습니다.

(5) 측정 결과는 다음과 같습니다. 세부 내용은 아래 기록에 있으며, 이 결과를 기반으로 다음과 같은 점수를 산출하였습니다.
3.1 FLOW CHAT

Fig. 3-1 Flow diagram for turbocharger matching
3.2 \( Ñèêû 

3.2.1 \( Ñèêû 

\begin{align*}
\eta_i &= \frac{632 \cdot P_i}{Q_f \cdot H_l} \\
\eta_e &= \frac{632 \cdot P_e}{Q_f \cdot H_l} = \frac{632 \times 1000}{b_e \cdot H_l}
\end{align*}

\begin{align*}
b_e &= 1000 \times \frac{Q_f}{P_e} \quad \text{(g/PS h)} : \text{Á¤¹Ì¿¬·á¼ÒºñÀ²} (3.2)
\end{align*}

\begin{equation}
\text{Rated RPM} \times \left( \frac{RPM}{\text{Rated RPM}} \right)^2
\end{equation}

- 18 -
\[ P_e = \frac{2\pi \cdot T \cdot N}{60 \cdot 75} = \frac{T \cdot N}{716.2} \]  \hspace{1cm} (3.3)

\[ P_{me} = \frac{P_e \cdot 75}{N/60 \cdot V_t \cdot 2} = \frac{P_e \cdot 9000}{N \cdot V_t} \]  \hspace{1cm} (3.4)

\( (Q_f) \)  

\[ b_c = 1000 \times Q_f / P_e \text{ (g/PS h)} \]  

\[ Q_f = \frac{b_c \cdot P_e}{1000} \text{ (kg/h)} \]  \hspace{1cm} (3.5)

\[ AFR = \frac{Q_a}{Q_f}, \quad Q_a = AFR \cdot Q_f \]  \hspace{1cm} (3.6)

**3.2.2**

\[ P_{1C} = P_{ATM} - \Delta P_{AC} \]  \hspace{1cm} (3.7)

\[ P_{2C} = \pi_c \cdot P_{1C} \]  \hspace{1cm} (3.8)

\[ T_{1C} = T_{ATM} \]  \hspace{1cm} (3.9)
\[ \frac{\dot{m}\sqrt{RT_{1C}}}{P_{1c}D^2}, \quad \eta, \quad \frac{\Delta T}{T_{1c}} = f\left(\frac{N}{\sqrt{RT_{1c}}}, \frac{P_{2C}}{P_{1C}}, \frac{\dot{m}}{\mu D}, \gamma\right) \quad (3.10) \]
\[ \eta_c = \frac{(P_{2c}/P_{1c})^{(\gamma - 1)/\gamma} - 1}{\Delta T/T_{1c}} \]  

(3.11)

\[ \frac{\dot{m}\sqrt{T_{1c}}}{P_{1c}}, \quad \eta = f\left(\frac{N}{\sqrt{T_{1c}}}, \frac{P_{2c}}{P_{1c}}\right) \]  

(3.12)

\[ T_{2c} = T_{1c} \cdot \left(\frac{\frac{1}{\gamma - 1} \cdot \frac{P_c \gamma}{\eta_c} - 1}{\eta_c} + \eta_c\right) \]  

(3.13)
\[ Q_{CA} = \frac{Q_a \cdot \sqrt{(T_{1C} + 273)/298}}{P_{1C}/750} \]  \hspace{1cm} (3.14)

### 3.2.3

\[ T_3 = T_{2C} - (\varepsilon \cdot (T_{2C} - T_{A TM})) \]  \hspace{1cm} (3.15)

\[ P_3 = P_{2C} - \Delta P_I \]  \hspace{1cm} (3.16)
\[
\rho_3 = \frac{P_3}{R \, T_3} \quad (3.17)
\]

\[
Q_c = \rho_3 \cdot V_i \cdot \eta_v \cdot N \cdot i \quad (3.18)
\]

3.2.4 摩擦和传热

摩擦和传热是对流换热中重要的因素。在对流换热过程中，摩擦和传热系数会受到介质的性质、流速和温度的影响。根据公式（3.19），摩擦和传热系数的关系如下

\[
T_{4T} - T_3 = \frac{a \cdot H_i \cdot Q_i}{C_p \, T \cdot Q_a} \quad (3.19)
\]
\[ a = 1 - (Q_e + Q_f + Q_w) \]  

\[ H_i = (0.25 - 0.3) \] (kcal/kg), 4\[4\] 10,596 kcal/kg, 2\[2\] 10,200 kcal/kg.

\[ a = 1 - (Q_e + Q_f + Q_w) \] (3.20)

\[ Q_e : \] \[ Q_f : \] \[ Q_w : \]

**Fig 3-3** Enthalpy-temperature diagram for air and combustion gas for determining the exhaust gas temperature
\[
Q_w = \frac{q_w}{b_r P_r H_r} = \frac{G_w \Delta t}{b_r P_r H_r} \quad (3.21)
\]

\[
G_w : \text{単位流量の時間積分}
\Delta t : \text{変動時間}
\]

\[
Q_{ex} = Q_f + Q_a = \frac{Q_a}{A_{FR}} \quad (1 + A_{FR}) \quad (3.22)
\]

\[
\pi_T = (1 - \frac{A_{FR} \cdot T_1 C \cdot (\pi_C \left(\frac{\eta_{r1}}{\eta_r} - 1\right))}{(A_{FR} + 1) \cdot 1.152 \cdot \eta_C \cdot \eta_T \cdot T_{4T}})^{-4.03} \quad (3.23)
\]

\[
\frac{C_{Pex}}{C_{Pa}} \approx \frac{1.152}{A_{FR}} \quad (3.24)
\]

\[
P_{5T} = P_{ATM} + \Delta P_{MUF} \quad (3.24)
\]

\[
P_{4T} = P_{AT} \cdot \pi_T \quad (3.25)
\]
\[ Q_{CEX} = \frac{Q_{ex} \cdot \sqrt{(T_{AT} + 273)/298}}{P_{AT}/750} \] (3.26)

3.2.5

Fig. 3-4 Turbine Characteristic
\[
\frac{m \sqrt{T_{AT}}}{P_{AT}}, \quad \eta = f\left(\frac{N}{\sqrt{T_{AT}}}, \frac{P_{ST}}{P_{AT}}\right) \quad (3.27)
\]

\[
W_C = \frac{Q_a \cdot T_{1C} \cdot C_{Pa} \cdot \left(\frac{\gamma_c}{\gamma_c} - 1\right)}{\eta_c} \quad (3.28)
\]
\[ W_T = Q_{ex} \cdot T_{4T} \cdot C_{Pex} \cdot \eta_i \cdot (1 - \left( \frac{1}{\pi_T} \right) \frac{x}{\nu}) \] (3.29)

(Waste Gate Valve)
4.1 Specifications of Four-stroke Automotive Engine

Table 4.1 Specifications of Four-stroke Automotive Engine

<table>
<thead>
<tr>
<th>Item</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Type</td>
<td>4 cycle Turbo. Diesel Engine</td>
</tr>
<tr>
<td>Maximum Power</td>
<td>117.2 PS / 4000 RPM</td>
</tr>
<tr>
<td>Maximum Torque</td>
<td>28.6 kg(\text{m} \text{m} \text{m} \text{m}) / 2000 RPM</td>
</tr>
<tr>
<td>Bore \times Stroke</td>
<td>91 mm \times 96 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>19.5</td>
</tr>
<tr>
<td>Displacement</td>
<td>2497 cc</td>
</tr>
<tr>
<td>Firing Order</td>
<td>1 - 3 - 4 - 2</td>
</tr>
</tbody>
</table>
Table 4.2 Specifications of Turbocharger

<table>
<thead>
<tr>
<th>Item</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>T/C maker</td>
<td>ASKL</td>
</tr>
<tr>
<td>T/C model</td>
<td>GT 15</td>
</tr>
<tr>
<td>A/R Ratio</td>
<td>0.43</td>
</tr>
<tr>
<td>Trim Ratio</td>
<td>55</td>
</tr>
<tr>
<td>Housing</td>
<td>Hi+Si+Mo</td>
</tr>
<tr>
<td>A/R Ratio</td>
<td>0.47</td>
</tr>
<tr>
<td>Trim Ratio</td>
<td>72</td>
</tr>
<tr>
<td>Cooling Type</td>
<td></td>
</tr>
<tr>
<td>Limit Speed (RPM)</td>
<td>190,000</td>
</tr>
<tr>
<td>Limit Temp. (°C)</td>
<td>760</td>
</tr>
</tbody>
</table>

4.1.2  

[Calculation Data(W/G Valve)]
[Measurement Data]  
[Waste gate valve]  
[Calculation Data(No W/G Valve)]  

Fig. 4-1, 4-2  
Fig. 4-3
**Fig. 4-1** Comparison of Compressor Pressure Ratio

**Fig. 4-2** Comparison of Compressor Efficiency
Fig. 4-3 Comparison of Engine Operating Line

Fig. 4-4
Fig. 4-5
Fig. 4-4 Comparison of a state at Compressor Outlet
Fig. 4-5 Comparison of Intake Manifold Pressure

Fig. 4-6 Comparison of Turbine Inlet Temperature
**Fig. 4-7** Comparison of Turbine Inlet Pressure

Fig. 4-6 shows the comparison of turbine inlet pressure with and without the waste gate valve. The waste gate valve has a 38% decrease in pressure at the optimal expansion ratio.

Fig. 4-7 compares the measurement data with the calculation data for the waste gate valve. The calculation data closely matches the measurement data, indicating the effectiveness of the waste gate valve in reducing pressure.

Fig. 4-8 demonstrates the control of expansion ratio by adjusting the valve position. The waste gate valve allows for precise control over the expansion ratio, ensuring optimal performance of the turbine.
Fig. 4-8  Comparison of Turbine Characteristics
Fig. 4-9 Comparison of Turbine Power
4.1.3 　

2.1.2 　

Fig. 4-10, 4-11]
Fig. 4-10 Turbine Characteristics of three sizes (trims)

Fig. 4-11 Estimate of Compressor Pressure Ratio
Fig. 4-10

Fig. 4-12, 4-13
Fig. 4-12 Turbine Characteristics of three Efficiency

Fig. 4-13 Estimate of Compressor Pressure Ratio
Fig. 4-14, 4-15: Turbine Characteristics of three sizes (trims) and Efficiency.
Fig. 4-15 Comparison of Engine Operating Line

Fig. 4-16
Fig. 4-16  Estimate of Compressor Pressure Ratio according to Intercooler efficiency
4.2 ชุด 2

4.2.1 ชุดนี้

ชุด 2 ชุดนี้ ประกอบด้วย ชุดที่ 1 ชุดที่ 2 และชุดที่ 3 ชุดที่ 1 มีการใช้โดยรวมที่มากกว่าชุดที่ 2 และชุดที่ 3.

Fig. 4-17

Fig. 4-17 Schematic diagram of a turbocharged two-stroke diesel engine for experimentation.
Table 4-3 Specifications of Two-stroke Diesel Engine

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>3 UEC 37LA</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Power</td>
<td>1500 PS / 188 RPM</td>
</tr>
<tr>
<td>Maximum Torque</td>
<td>5710 kg·m / 188 RPM</td>
</tr>
<tr>
<td>Bore × Stroke</td>
<td>370 mm × 880 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>14.8</td>
</tr>
<tr>
<td>BMEP</td>
<td>12.65 kg/cm²</td>
</tr>
<tr>
<td>Mean Piston Speed</td>
<td>5.51 m/s</td>
</tr>
</tbody>
</table>

Fig. 4-18 Comparison of Engine Operating Point at Load 70%
Fig. 4-19  Comparison of Engine Operating Point at Load 85%
Fig. 4-20 Comparison of Engine Operating Point at Load 100%
Fig. 4-21 Comparison of Engine Operating Line

- Fig 4-18, 4-19, 4-20 for 70%.
- Fig 4-19 and 4-20 for 85% and 100%.

- Fig 4-21.
\[ (Overlap) \]

\[ m = C d \, A \, \sqrt{\left[ 2 \gamma \left( \gamma - 1 \right) \right] P_3 \rho_3 \left( \frac{P_{4T}}{P_3} \right)^{2/\gamma} - \left( \frac{P_{4T}}{P_3} \right)^{\left( \gamma + 1 \right)/\gamma}} \] 

\[ (4.1) \]

\[ m = C d \, A \, \sqrt{2 \rho_3 \left( P_3 - P_{4T} \right)} \] 

\[ (4.2) \]
Fig. 4-22 Comparison of a Intake state

Fig. 4-23 Comparison of Turbine Inlet Temperature
Fig. 4-24  Comparison of Turbocharger RPM

Fig. 4-25  Comparison of Turbine Characteristics
Fig. 4-26  Comparison of Turbine Inlet Pressure

Fig. 4-27  Comparison of Turbine Power
4.2.3

5\%

Fig. 4-28 Comparison of Engine Operating Line at 3 fuelling
Fig. 4-29  Comparison of Turbine Inlet Temperature at 3 fuelling
Fig. 4-30  Comparison of Turbine Flow at 3 fuelling
5

4

2

(1) Waste Gate Valve

(2)

(3)

(4)

(5)
（6）上列の数値に対する推定誤差値は以下の通りです。

\[
\begin{align*}
\text{推定誤差値} & = 59
\end{align*}
\]


