Research Report

Transient Performance Prediction for Turbocharging Systems Incorporating Variable-geometry Turbochargers

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可変容量ターボ過給性能シミュレーション

内田博

Abstract

Turbocharging technologies are important to improving the fuel consumption of engines by enabling downsizing and lean boost. A variable-geometry turbocharger with motor assist provides an effective means of improving the low-speed torque and response of an engine. On the other hand, the surge limits of the compressor restrict the charging pressure at low engine speeds.

We developed one-dimensional performance prediction models for a compressor featuring variable inlet guide vanes and a variable-nozzle turbine. Using these models, we went on to produce simulation code for predicting the transient performance of a turbocharged engine. An advantage of this simulation code is that the accuracy with which the transient characteristics of a turbocharging system can be calculated is independent of the calculation time step. Accordingly, the transient characteristics after an arbitrary time can be predicted with a relatively quick calculation.

We used this simulation to evaluate the effects of the variable inlet guide vane, the variable nozzle, and motor assist on the characteristics of a turbocharging system. Our results showed that significant improvements in both the boost pressure at low engine speeds and the transient response of the turbocharger can be achieved by taking advantage of the synergy between the variable inlet guide vane and the casing treatment.

Keywords Turbocharger, Compressor, Turbine, Turbocharged engine, Transient response, Simulation, Variable nozzle, Variable inlet guide vane

要 旨

エンジンの小型・高出力化とリーン過給により燃費を向上するためにはターボ過給技術が重要である。モータアシスト付の可変容量ターボチャージャはエンジンの低速トルクと過渡レスポンスの向上に有効であるが，コンプレッサのサージ限界が低速域の過給圧上昇を妨げるという課題もあり，過給エンジンシステムの特性を予測した技術開発が望まれる。本研究では，可変容量ターボチャージャの性能予測モデルを開発し，簡易的なエンジンモデルと組合せることにより，過渡特性も含めた性能予測プログラムを開発した。これは過渡性能計算時の精度が計算時間ステップに依存しないという特徴を有し，任意の時間ステップで迅速に高精度で過渡性能を予測できる。

過給エンジンの特性におよぼす可変入口案内翼，可変ノズルおよびモータアシストの影響をシミュレーションで検討した。その結果，可変入口案内翼とケーシングトリートメントの相乗効果によるサージ限界の拡大がエンジンの低速トルクおよび過渡レスポンスを大幅に向上できることを明らかにした。

キーワード ターボチャージャ，コンプレッサ，タービン，過給エンジン，過渡応答，シミュレーション，可変ノズル，可変入口案内翼

1. Introduction

Recently, with the goal of reducing CO_2 emissions, high-boost turbocharged engines have been developed to enable downsizing and lean boost and thus improve automobile fuel consumption. To improve low-end torque and the transient response of these turbocharged engines, several different turbocharging systems have been developed. These include the Variable Nozzle (VN) turbocharger, the electric-assist turbocharger, the electric-boost compressor, variable-geometry compressor, and two-stage turbocharging system. Unsurprisingly, engine-turbocharger matching and the control of these advanced turbocharging systems are more complicated than with conventional systems. Therefore, it is important to be able to predict both the steady-state characteristics and transient characteristics of these advanced turbocharging systems. The development of these systems and their control systems demands the ability to perform calculations at high speed.

We developed a wide flow range compressor incorporating both a Variable Inlet Guide Vane (VIGV) and a casing treatment. We found that the synergy between the VIGV and the casing treatment had a marked effect on the surge limits, and this allowed us to improve the surge limits much more than would have been possible with the VIGV alone. Accordingly, we were able to raise the boost pressure at low engine speeds which produced a significant improvement in the low-end torque. We also expected the transient response of the engine torque to improve due to the optimum control applied by the VIGV. To predict the effect of the variable-geometry turbocharger on the performance of a turbocharged engine, we developed a one-dimensional simulation model of the turbocharging system. The accuracy with which the transient characteristics of a turbocharging system are calculated is independent of the time step for the calculation within a range of Δt = 0.1 to 0.8 sec. Accordingly, the transient characteristics after an arbitrary time can be predicted in a very short time. The simulation evaluates the effects of the VIGV on the characteristics of the turbocharging system. The results show that a significant improvement in both the boost pressure at low engine speeds and the transient response of the turbocharger are achieved thanks to the synergy effect between the VIGV and the casing treatment.

2. Simulation method for turbocharged engines

We developed a one-dimensional performance prediction program for turbocharged engines so as to investigate both engine-turbocharger matching and the transient response of variable-geometry turbocharging systems, both with a short calculating time. The model consists of a simple engine model and turbocharger model, the latter of which consists of a compressor model and turbine model.

2. 1 Turbocharger model

The turbocharger model was developed to enable the prediction of the performance of a compressor with the VIGV and the VN turbine simply by inputting the turbochargers' dimensions. The velocity triangles for the compressor and turbine are calculated using a one-dimensional flow calculation with conservation of the angular momentum and pressure loss models. The pressure loss models were developed based on a one-dimensional flow analysis of test data obtained in our laboratories for many kinds of turbochargers. The VIGV angle is related to the impeller inlet velocity triangle, while the VN setting angle is related to the turbine rotor inlet velocity triangle. So, we can predict the effects of the control of both the VIGV and the VN on the turbocharging characteristics. The mechanical loss is in proportion to the square of the rotational speed. **Figure 1** shows the symbols used in the velocity triangles for both the compressor and the turbine.

2. 1. 1 Compressor model

The compressor input power Lc is given as follows.

$$Lc = Ga \cdot Cpa(T3 - T0) \quad (1)$$

Ga : air flow rate(kg/s)

Cpa : air specific heat (kcal/kgK)

$T0$, $T3$(K) : temperatures at inlet and outlet

The enthalpy rise in the compressor is given by the following equation, which is based on the angular momentum theory.

$$J \cdot Cpa(T3 - T0) = (U2 \cdot Cu2 - U1 \cdot Cu1)/g \quad (2)$$

$U1$, $U2$: circumferential velocity of impeller at inlet and exit (m/s)

$Cu1$, $Cu2$: circumferential velocity of the flow at impeller inlet and exit (m/s)

J : mechanical equivalent of heat (kgm/kcal)

g : gravitational acceleration (m/s^2)

The adiabatic pressure at the compressor outlet under conditions of equal entropy is as follows.

$$P3ad = P0\{(T3/T0)^{\kappa/(\kappa-1)} - 1\} \cdots\cdots\cdots\cdots\cdots\cdots (3)$$

$P0$: compressor inlet pressure (kPa)

κ : specific heat ratio

The compressor outlet total pressure is given as follows as a result of using the total pressure loss ΔP.

$$P3 = P3ad - \Delta P \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (4)$$

The total pressure loss is modeled as follows.

$$\Delta P = \Delta Pvigv + \Delta Pinc + \Delta Pfric + \Delta Pch + \Delta Pdif \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (5)$$

- Pressure loss in the VIGV

$\Delta Pvigv$ = func. (mach number, static pressure, vane setting angle)

- Impeller incidence loss

$\Delta Pinc$ = func. (incidence angle, relative mach number, static pressure)

- Impeller friction loss

$\Delta Pfric$ = func. (relative mach number, diffusion factor, static pressure)

- Impeller choke loss

ΔPch = func. (mach number at the throat, incidence angle, static pressure)

- Pressure loss in the diffuser

$\Delta Pdif$ = func. (mach number, inlet flow angle, static pressure)

The compressor performance is calculated by inputting the air flow rate, the rotational speed, and the dimensions. The flow chart for the calculation is shown in **Fig. 2**. A feature of the compressor model is that the characteristics under conditions of negative efficiency can be predicted. This negative efficiency condition occurs when a turbocharged engine is allowed to idle. The predicted compressor performances were compared with the results obtained by experiment, as shown in **Fig. 3**. The predicted compressor characteristics for both the

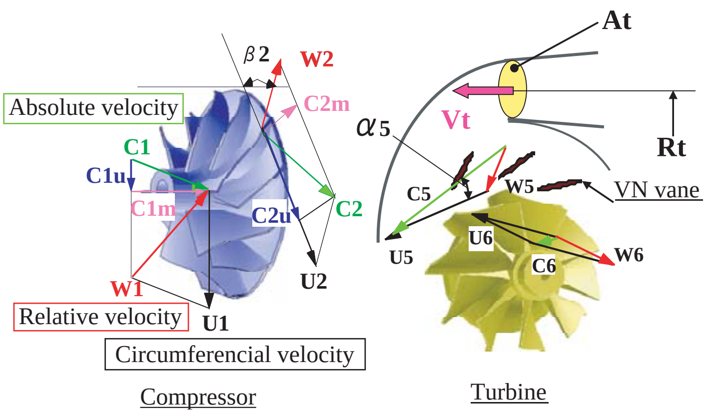

Fig. 1 The velocity triangles of the compressor and turbine.

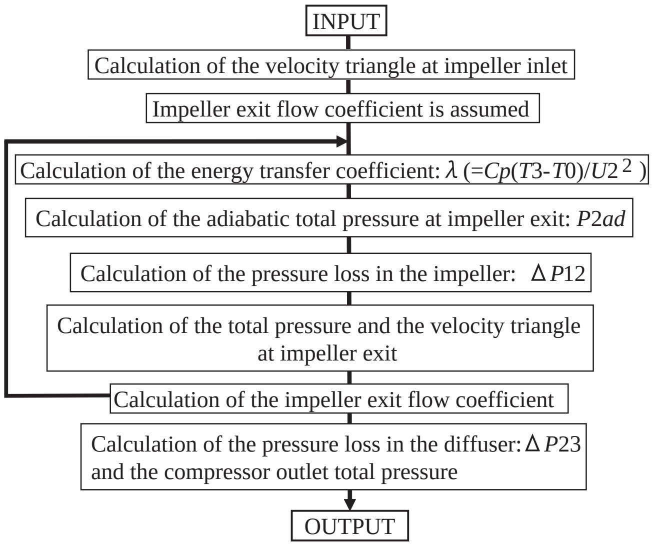

Fig. 2 Calculation flow of the compressor model.

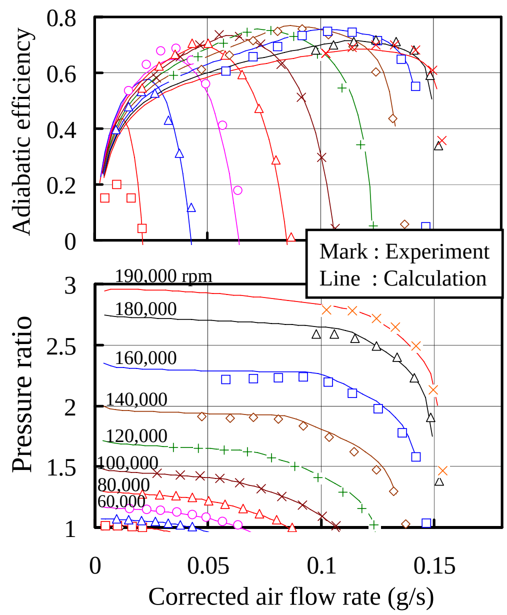

Fig. 3 Comparison of the compressor performance between calculation and experiment.

pressure ratio and efficiency agree well with the experimental results.

2. 1. 2 Turbine model

The turbine input power $Ltad$ is given as follows.

$$Ltad = Gg \cdot Cpg \cdot T4\{1-(P4/Ps6)^{-(\kappa g-1)/\kappa g}\} \cdots (6)$$

Gg : gas flow rate
Cpg : gas specific heat
$T4$: turbine inlet temperature
$P4$: turbine inlet total pressure
$Ps6$: turbine outlet static pressure

The enthalpy drop in the turbine is given by the following equation, and conforms to the angular momentum theory.

$$J \cdot Cpg(T4-T6) = (U5 \cdot Cu5 - U6 \cdot Cu6)/g \cdots (7)$$

$U5$, $U6$: circumferential velocity of turbine rotor at inlet and exit (m/s).
$Cu1$, $Cu2$: circumferential velocity of the flow at the turbine rotor inlet and exit (m/s).

The adiabatic pressure at the turbine outlet under conditions of equal entropy is given by the following.

$$P6ad = P4\{1-(T4/T6)^{-\kappa g(\kappa g-1)}\} \cdots\cdots\cdots\cdots (8)$$

κg : gas specific heat ratio

The turbine outlet pressure is given by the following, using total pressure loss ΔP.

$$P6 = P6ad - \Delta P \cdots\cdots\cdots\cdots (9)$$

The total pressure loss is modeled as follows.

$$\Delta P = \Delta Psc + \Delta Pvn + \Delta Pinc + \Delta Pfric + \Delta Pex \cdots\cdots\cdots\cdots (10)$$

- Scroll pressure loss
  ΔPsc = func. (mach number at throat, static pressure)
- Pressure loss in the VN
  ΔPvn = func. (mach number, vane setting angle, static pressure)
- Rotor incidence loss
  $\Delta Pinc$ = func. (incidence angle, relative mach number, static pressure)
- Rotor friction loss
  $\Delta Pfric$ = func. (relative mach number, diffusion factor, static pressure)
- Discharge loss at rotor exit
  ΔPex = func. (relative mach number, velocity angle, static pressure)

As $P6$ approaches the given value, the turbine performance is calculated repeatedly by inputting the pressure ratio, the rotational speed, and the dimensions. The flow chart for this calculation is shown in **Fig. 4**. Another feature of this turbine model is that the characteristics under conditions of negative efficiency can be predicted. **Figure 5** shows the predicted performance for a VN turbine in comparison with the values obtained by experiment. The predicted turbine characteristics for both the flow rate and efficiency agree well with the values obtained by experiment regardless of the conditions at the VN.

2. 2 Transient model for turbocharged engine

We developed a performance prediction program in order to predict the characteristics of turbocharging systems while requiring little time for

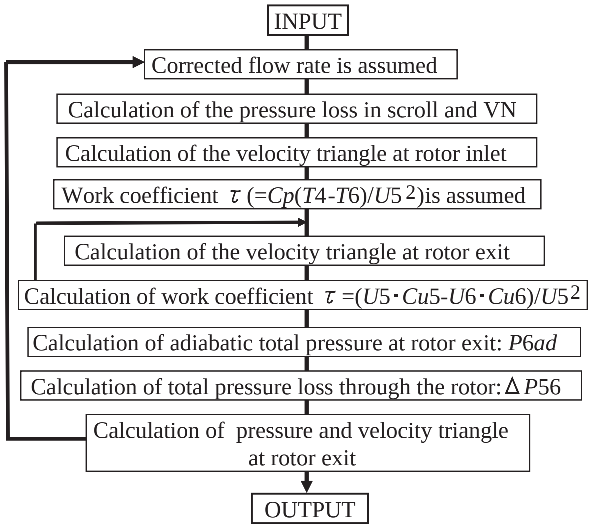

Fig. 4 Calculation flow of the turbine model.

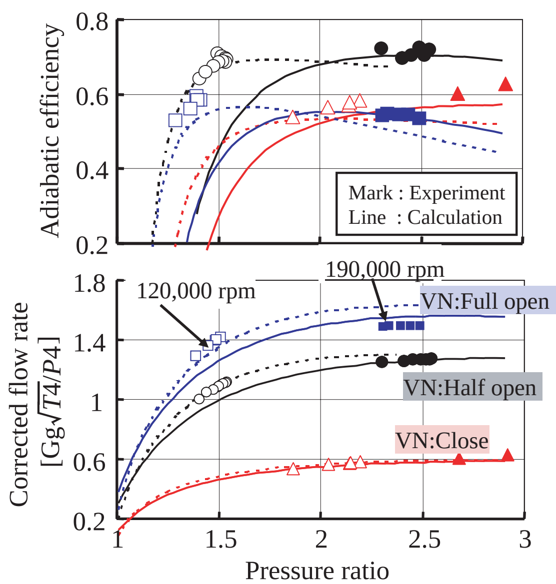

Fig. 5 Comparison of the turbine performance between calculation and experiment.

the calculation. We established a steady-state model for the engine, like that for a gas generator without considering any cyclic phenomena. The turbocharger model is connected to the engine model via the energy balance equation between the compressor and the turbine. The energy balance equation for the turbocharger is given as follows.

$$Lt + Lea = Lc + Lm + Ip \cdot (\omega/2\pi) \cdot (\Delta\omega/\Delta t) \quad \cdots\cdots (11)$$

Lt : turbine output power
Lea : electric assist power
Lc : compressor input power
Lm : mechanical loss
ω : angular velocity of the rotor
Ip : inertia momentum of the rotor
Δt : time required for calculation

Under transient operating conditions, factors Lt, Lea, Lc, Lm, and ω are all functions of time. Accordingly, the calculation time step Δt has to be very short to ensure a high level of accuracy when conventional cycle simulation is applied, because the values of Lt, Lea, Lc, Lm, and ω after Δt are not known. In our model, the steady-state operating condition after Δt is calculated first, after which the transient operating condition is calculated by using an energy balance equation with the mean values of Lt, Lea, Lc, Lm, and ω. The values of Lt, Lea, Lc, Lm, and ω are calculated repeatedly by averaging the values at time t and $t+\Delta t$. Therefore, the calculation accuracy for the transient characteristics is almost independent of the calculation time step Δt. The transient characteristics after time $n\Delta t$ are calculated by integrating Eq. (11) from $t = 0$ to $n\Delta t$. **Figure 6** shows the results for the transient performance prediction under the conditions for each of the time step and indicates that the calculation time step does not affect the calculation results.

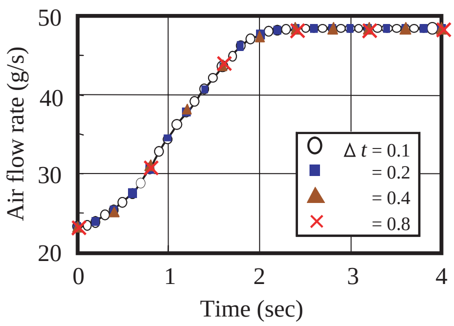

Fig. 6 The effect of the calculation time step on the prediction accuracy.

Figure 7 compares the predicted boost pressure and the turbine inlet pressure with the values obtained by experiment for a VN turbocharged engine operating under full load. The figure shows that the predicted values agree well with the measured values.

3. Performance prediction for turbocharging systems

We investigated the performance of variable-geometry turbocharging systems by means of simulation under steady-state conditions and transient operation. The time required for calculation was about 20 minutes for a transient simulation for a two-stage turbocharging system when using a late-model personal computer.

3. 1 Variable-geometry turbocharging

We developed performance prediction models for compressors and turbines to evaluate the effects of the VIGV, the casing treatment, and the VN on the performance of a turbocharged engine. We assumed the use of a gasoline engine with a high power density. The maximum turbine inlet temperature was 1130 K at the maximum power operating point. **Figure 8** shows the effects of the VIGV and the casing treatment on the boost pressure under full-load operating conditions. The boost pressure at low engine speeds is significantly improved by the use of the casing treatment and the VIGV because

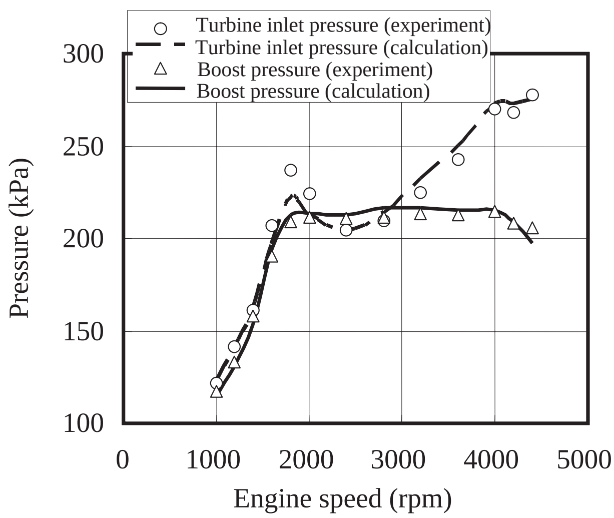

Fig. 7 The predicted boost pressure and the back pressure compared with the experiment.

compressor surge limits are drastically improved thanks to the synergy effect described above. We believe that the low-end torque and the transient response of the turbocharged engine are improved by the use of a compressor with the VIGV and the casing treatment. On the other hand, the wide flow range turbine is necessary to achieve a high boost pressure over a wide operating range. Figure 8 also shows the turbine inlet pressures that are needed to obtain the boost pressure for "VIGV+Casing Treatment" as shown in the figure. For a conventional turbine with a waste gate valve, the turbine inlet pressure increases as the engine speed increases due to the increase in the waste gas flow. At high engine speeds, the turbine inlet pressure leads to an unacceptably high value. The high turbine inlet pressure leads to an increase in the pumping loss and the occurrence of knocking. Therefore, the fuel consumption of the turbocharged engine increases. In the case of the VN turbine, on the other hand, the turbine inlet pressure at low engine speeds is higher than that of the conventional turbine. This is caused by the turbine efficiency falling as a result of closing the VN. The pressure is lower than that of a conventional turbine at mid-range and high engine speeds because the turbine efficiency is comparatively high. Additionally, it is difficult to apply a VN turbine to a gasoline engine due to its complicated structure and because the turbine inlet temperature is very high. As a result, we believe that it is necessary to develop a variable-geometry turbine offering both high levels of efficiency and a wide flow range for application to turbocharged gasoline engines.

The effect of the VIGV control on the transient response of a turbocharging system was predicted by a turbocharged engine simulation. **Figure 9** shows the transient characteristics of the boost pressure. When we apply control to the VIGV, the vane is maintained at a constant angle of 80 degrees for the initial 0.6 sec, and is then linearly reduced to zero degrees at the 2-sec point. It is subsequently held at 0 degrees. The transient response of the boost pressure is improved by controlling the VIGV angle as mentioned above. The VIGV can change the rotational speed of the turbocharger despite the same pressure ratio of the compressor being maintained. Accordingly, we believe that the transient response of a turbocharged engine can be improved by controlling the VIGV so as to cancel the turbo-lag. The operating line on the compressor map is shown in **Fig. 10**. The turbo-lag causes the operating line to shift towards the lower flow rate as the turbocharged engine decelerates. So, the VIGV angle has to be controlled to prevent surge from occurring.

3. 2 Two stage turbocharging

A two-stage turbocharging system is effective for improving the low-end torque and the transient response of a turbocharged engine. We investigated the steady state and transient characteristics of the system through an engine simulation. The main role of the small turbocharger is to charge air at low engine speeds, while the large turbocharger is mainly responsible for charging at high engine

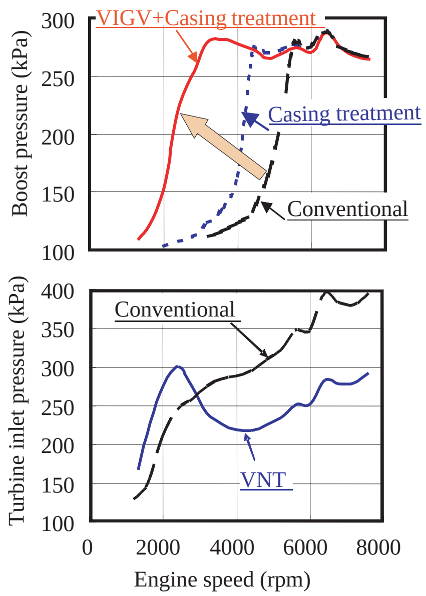

Fig. 8 The effect of VIGV on the boost pressure.

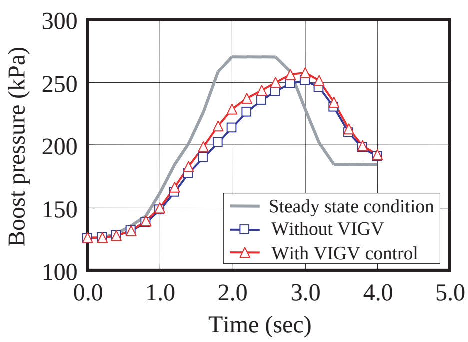

Fig. 9 The effect of VIGV on the transient response.

speeds. So, the compressor surge does not restrict the charging pressure at low engine speeds. Unfortunately, such a two-stage turbocharging system is more complicated and larger than a single turbocharging system. In many cases, it requires a bypass valve to avoid the choking of the small compressor at high engine speeds. Therefore, engine-turbocharger matching technologies are particularly important to a two-stage turbocharging system. **Figure 11** compares the transient characteristics of a two-stage system with those of a single-stage system. The inertia of the small turbocharger is 50 % of the single stage turbocharger, while the large turbocharger inertia is the same as that for a single-stage turbocharger. The boost pressures of the two systems at each calculating time step are basically the same when each system is operated under steady-state conditions. The small turbocharger of the two-stage system responds more quickly to the transient operation than the single-stage turbocharger. Accordingly, we believe that a two-stage turbocharging system improves the transient response of a turbocharged engine.

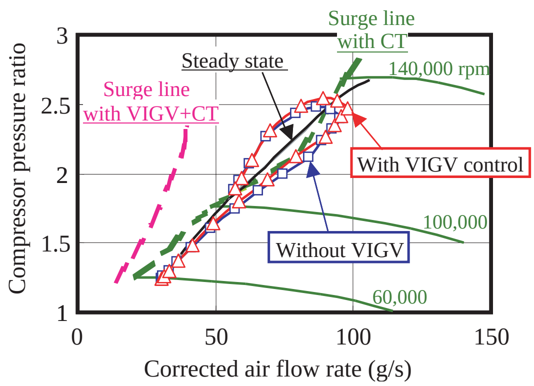

Fig. 10 The operating line on the compressor map.

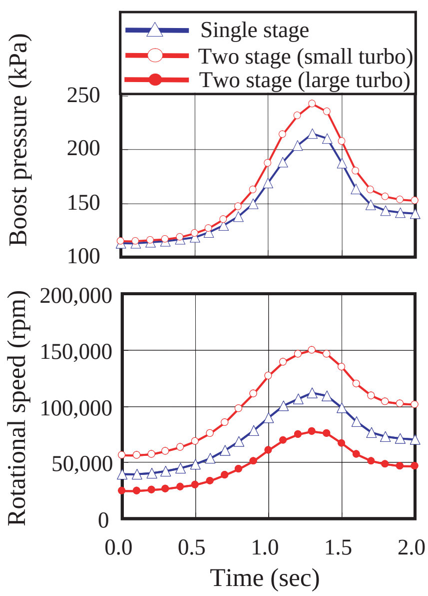

Fig. 11 The transient response of the two-stage turbocharging system.

4. Conclusion

We developed a performance prediction program for variable-geometry turbocharging systems to investigate the effect of a variable-geometry turbocharger on transient response. Then, we evaluated the steady state and transient characteristics of a variable-geometry turbocharging system by means of simulation by using a performance prediction program. A summary of these results is given below.

(1) We found that we could calculate the transient characteristics of a turbocharged engine independently of the calculation time step.

(2) The transient response of a turbocharged engine can be improved by controlling the VIGV angle so as to cancel the turbo-lag.

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