Supercharging possibilities for motorcycle Suzuki GSX-R 750

Możliwości doładowania w motocyklu Suzuki GSX-R 750

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In the paper, possibility for the power increase of motorcycle Suzuki GSX-R 750 is proposed. The turbocompressor available in the market was selected, and its characteristics were checked. Calculations proved that theoretical improvement of 33 kW, which is ca. 29%, was possible, which meant the ratio power to mass could be increased up to 1.1 kW/kg. This way, the discussed motorcycle achievable power could be similar to the ones with engines of stroke capacity 1000 cm³. 3KEYWORDS: supercharger, turbocompressor, internal combustion engine

W artykule omówiono możliwość zwiększenia mocy motocykla Suzuki GSX-R 750. Wybrano dostępną na rynku turbosprężarkę i sprawdzono jej właściwości. Obliczenia wykazały, że była możliwa teoretyczna poprawa o 33 kW, czyli ok. 29%, co oznaczało, że stosunek mocy do masy można było zwiększyć do 1,1 kW/kg. Tym samym moc osiągalna motocykla mogłaby być zbliżona do pojazdów z silnikami o objętości skokowej 1000 cm³.
SŁOWA KLUCZOWE: doładowanie, turbosprężarka, silnik spalinowy

Introduction

The global fleet of motor vehicles of all types, including two-wheelers, is now around 1.5 billion (of which 1 billion are cars) and is expected to reach 2 billion shortly after 2020 [1]. It can be expected that in the nearest future, the development of internal combustion engine vehicles may focus on low-cost stop-start vehicles with engine downsizing supported by turbocharging and electric supercharging.

Supercharging was developed as a tool to extract maximum performance from a given engine swept volume [2]. There are many families of air chargers, dependent on the charging principle [3]. The supercharging mainly consists on a mechanical driving of the charger: roots blower, sliding vane, screw compressor. Moreover, reciprocating and centrifugal compressor belong to this group, too [4]. In the present study, supercharging possibility for the particular model of motorcycle is analyzed.

Technical data

In the Table, there are technical data of the motorcycle Suzuki GSX-R 750 (Fig. 1) produced in 2005 with double overhead camshaft and high compression. This type of engines leave limited room for modification, but calculation of turbocompressor presented below demonstrate some improvement possibilities.

Fig. 1. Suzuki GSX-R 750 [5]

TABLE. Motorcycle Suzuki GSX-R 750 k5 technical data [5]

Model	Suzuki GSX-R 750 k5
Engine, capacity	Four-stroke, transverse four-cylinder, DOHC, 4 valves per cylinder, 749 cm³
Max power	114.3 kW (155 hp)/12,800 rpm
Max torque	86.3 Nm/ 11,200 rpm
Bore × stroke	72.0 × 46.0 mm
Compression ratio	12.3:1
Ignition	Digital DC-CDI
Induction	Fuel injection, 46 mm throttle bodies
Dimensions and mass	
Length	2075 mm
Width	715 mm
Height	1145 mm
Wheelbase	1396 mm
Dry weight	163 kg

For the calculations of supercharging compressor, the following parameters were assumed:

- air excess coefficient: $\lambda_c = 1$
- pressure of the surrounding air: $p_{ot} = 101.325$ [kPa]
- temperature of the surrounding air: $T_{ot} = 25$ [°C] = 298 [K]
- supercharging pressure: $p_d = 100$ [kPa]
- heating up of the fresh charge: $\Delta T = 5$ [K]
- four-stroke engine: $i = 4$

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For the fuel gasoline, the following data were taken:
- calorific value: W_o = 43.5 · 10^3 kJ/kg
- density: $\rho_{pb} = 0{,}74 \left[\frac{kg}{dm^3}\right]$
- theoretical air consumption for burning of 1 kg gasoline: L'_t = 14.9 kg/kg

Compressor calculations

Generally, there are three methods of increasing air capacity: increasing the piston displacement, running the engine at high speed, and increasing the density of the charge [6]. The latter is done with a supercharger. Supercharging ratio can be calculated from the equation (1) as follows:

$$S_{td} = \frac{N_d}{N_e} = \frac{p_{ed}}{p_e} \tag{1}$$

where:
N_d – power of a supercharged engine [kW],
N_e – power of a naturally aspirated engine [kW],
p_{ed} – effective pressure of a supercharged engine [MPa],
p_e – effective pressure of a naturally aspirated engine [MPa].

Assuming that the power of supercharged engine is N_d = 147 kW, and having from technical data N_e = 114 kW, calculated ratio S_{td} = 1.29.

The power increase is:

$$\Delta N = N_d - N_e = 147 - 114 = 33 \text{ [kW]}$$

Supercharger compression value can be calculated as follows:

$$\pi_s = \frac{p_2}{p_1} = \frac{201.325}{101.325} = 1.987 \tag{2}$$

where: p_2, p_1, – values of pressure after and before supercharger, respectively [kPa].

Since $p_1 = p_{ot}$ and p_2 denotes the pressure after being supercharged, its value was $p_2 = p_1 + p_d$. To calculate the increase of density φ_d of loaded compressed air, formula (3) can be employed as follows:

$$\varphi_d = \frac{\rho_2}{\rho_1} = \frac{p_2}{p_1}\cdot\frac{T_1}{T_2} = \pi_s \cdot \frac{1}{1 + \frac{1}{\eta_{iz-s}}\cdot\left(\pi_s^{\frac{k-1}{k}} - 1\right)} = $$
$$= 1.987 \cdot \frac{1}{1 + \frac{1}{0.74}\cdot\left(1.987^{\frac{1.4-1}{1.4}} - 1\right)} = 1.537 \tag{3}$$

where:
k = 1.4 – adiabatic exponent,
η_{iz-s} – assumed value of the supercharger's isentropic efficiency, which should belong to the range from 0.60 to 0.77.

Correlations between compression of a turbocharger π_s and density increase φ_d, considering temperature ratio and isentropic efficiency of the device are presented in Fig. 2.

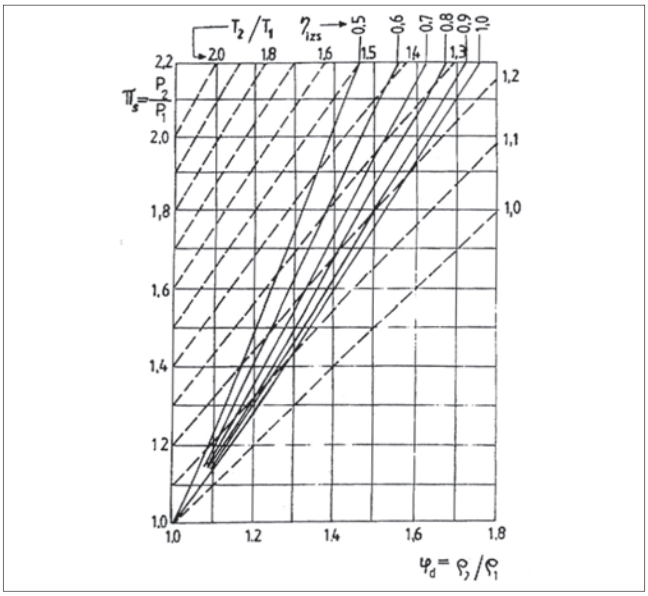

Fig. 2. Dependence of the compression π_s and temperature ratio after and before the compressor T_2/T_1 on the density increase φ_d at different values of isentropic efficiency η_{iz-s} [7]

Next, from the equation (4) it is possible to calculate temperature ratio after and before the compressor:

$$\frac{T_2}{T_1} = \frac{p_2}{p_1}\cdot\frac{\rho_1}{\rho_2} = \frac{1}{\varphi_d}\cdot\pi_s = \frac{1}{1.537}\cdot 1.987 = 1.292 \tag{4}$$

The equation (4) demonstrates the increase of air temperature after compression in supercharger of ca. 30%. Then, the specific air demand can be obtained from the following formula:

$$\Gamma = \frac{3600\cdot \dot{m}_s}{N_d} = \lambda_c \cdot g_e \cdot L'_t \cdot 10^{-3} \tag{5}$$

where:
$\dot{m}_s$ – overall air mass flow through the engine $\left[\frac{kg}{s}\right]$,
N_d = 147 [kW] – effective power of supercharged engine,
λ_c = 1 – air excess coefficient,
g_e = 320 [g/kWh] – specific fuel consumption by the engine,
$L'_t = 14.9 \left[\frac{kg_{air}}{kg_{fuel}}\right]$ – theoretical air needed to burn 1 kg of fuel.

With the abovementioned values, calculated specific air demand is $\Gamma = 4.768 \left[\frac{kg}{kW\cdot h}\right]$.
The air density ρ_{ot} can be calculated as follows:

$$\rho_{ot} = \frac{p_{ot}\cdot\mu_p}{\mu\cdot R\cdot T_{ot}} \tag{6}$$

where:
p_{ot} = 101325 [Pa] – outer atmospheric pressure,
T_{ot} = 298 [K] – outer air temperature,
$\mu_p = 28.9644 \left[\frac{kg}{kmol}\right]$ – average molar mass of the air [8],
$\mu \cdot R$ = 8314.33 [J/kmol·K] – universal gas constant for the air.

Thus, the calculated air density is:

$$\rho_{ot} = \frac{101325\cdot 28.96}{8314.33\cdot 298} = 1.185 \left[\frac{kg}{m^3}\right].$$

The compressor has to ensure proper air outflow to make engine possible to work. The mass flow, and the volume flow can be calculated as follows:

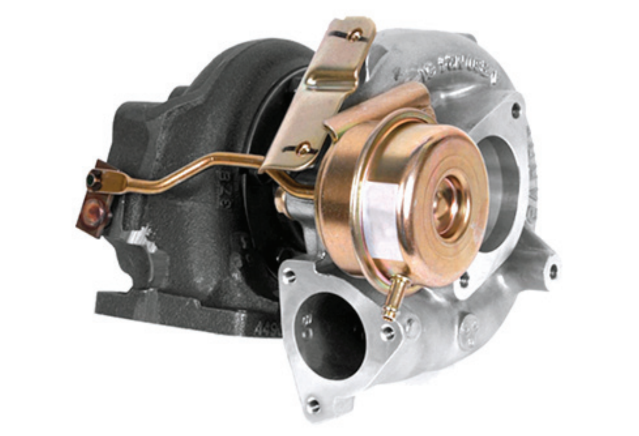

Fig. 3. GT2560R Turbocharger [9]

$$\dot{m}_s = \frac{\Gamma \cdot N_d}{3600} = \frac{4.768 \cdot 147}{3600} = 0.195 \left[\frac{kg}{s}\right] \quad (7)$$

$$\dot{V}_s = \frac{\Gamma \cdot N_d}{\rho_{ot} \cdot 3600} = \frac{4.768 \cdot 147}{1.18 \cdot 3600} = 0.164 \left[\frac{m^3}{s}\right] \quad (8)$$

When the air is leaving the compressor, its temperature T_2 can be derived from the equation (4) and known temperature of loaded air T_1 = 303 K:

$$T_2 = T_1 \cdot \frac{T_2}{T_1} = 303 \cdot 1.292 = 391.74\,[K] \quad (9)$$

However, assuming temperature accuracy of ±0.5 [K], T_2 should be rounded to 391.5 [K].

The obtained parameters $\dot{m}_s$ = 0.195 [kg/s] and π_s = 1.987 were basis for the choice of available compressor type GT2560R produced by Garrett [9].

Gas turbine calculations

Total efficiency of the turbocharger was assumed η_{cts} = 0.59, so that parameter ξ_0 was calculated from the following formula:

$$\xi_0 = \eta_{cts} \cdot \frac{T_3}{T_1} \cdot \frac{\dot{m}_t}{\dot{m}_s} = 0.59 \cdot \frac{890}{303} \cdot \frac{0.099}{0.195} = 0.88 \quad (10)$$

where:
T_3 = 890 [K] – temperature of exhaust gases before the turbine,
T_1 = 303 [K] – temperature of the air load before compressor,
$\dot{m}_t$ – mass airflow through the turbine,
$\dot{m}_s$ – total mass airflow, calculated in eq. (7).

The temperature difference ΔT_{23} before and after compressor $\Delta T_{23} = T_3 - T_2$ = 890 – 391.5 = 498.5 [K]. Since there may be a flow outlet $\Delta\dot{m}$ between compressor and turbine, this value must be checked after the turbocompressor is chosen. Thus, the calculated value of $\dot{m}_t$ is as follows:

$$\dot{m}_t = \dot{m}_s - \Delta\dot{m} = 0.195 - 0.096 = 0.099\,[kg/s] \quad (11)$$

The decompression π_t in the turbine can be found from the Fig. 4. In this graph, the flow outlet is not taken into consideration, so that parameter ξ is calculated as follows:

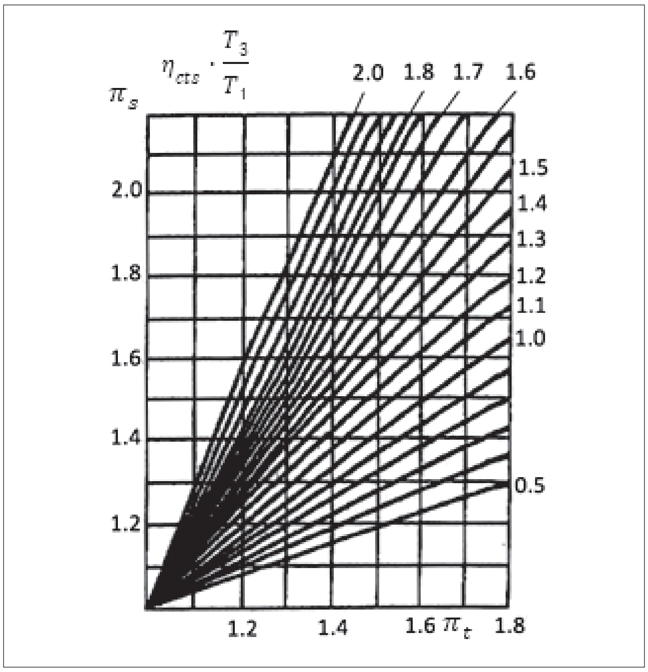

Fig. 4. Dependence between the pressure ration in turbine p_2/p_4, and its compression for different coefficients $\xi = \eta_{cts} \cdot \frac{T_3}{T_1}$ [7]

$$\xi = \eta_{cts} \cdot \frac{T_3}{T_1} \cdot = 0.59 \cdot \frac{890}{303} = 1.7 \quad (12)$$

where assumed efficiency of the turbocompressor is η_{cts} = 0.59.

Kowalewicz [7] provided the graph of dependence between pressure ratios and ξ parameter, shown in Fig. 4. Checking the obtained values, there can be found the value of ratio $\pi_s = \frac{p_3}{p_4} = 1.57$.

Since p_4 = 105 [kPa], the pressure p_3 can be calculated as p_3 = 164.85 [kPa]. Now, the adiabatic expansion in the turbine can be calculated from the formula (13):

$$H_{ad-t} = \frac{k_{sp}}{k_{sp}-1} \cdot R_{sp} \cdot T_3 \cdot \left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{k_{sp}-1}{k_{sp}}}\right] \quad (13)$$

With adiabatic exponent of the exhausted gases k_{sp} = 1.34, and their gas constant R_{sp} = 288.3 $\left[\frac{J}{kg \cdot K}\right]$, the result of the calculation from equation (3) is H_{ad-t} = 109362.41 $\left[\frac{J}{kg}\right]$. Similar equation can be applied to the compressor H_{ad-s}, but with air adiabatic exponent k = 1.4 and air gas constant R = 287 $\left[\frac{J}{kg \cdot K}\right]$. Then, the result is as follows: H_{ad-s} = 65965.03 $\left[\frac{J}{kg}\right]$.

Turbocompressor efficiency

Now, the overall efficiency of the turbocompressor can be calculated as follows:

$$\eta_{cts} = \frac{H_{ad-s}}{H_{ad-t}} \quad (14)$$

which provides the result η_{cts} = 0.6. It should be noted that the resulting efficiency is close to the value assumed in the equation (12). It can be thus derived from the turbocompressor characteristics, that for the pressure ratio π_s and mass airflow $\dot{m}_s$, its rotational speed is n_t = 121,000 rpm. Hence, if the turbine diameter d_t = 53 mm, its peripheral speed is v_p = 335.6 m/s.

Fig. 5. Isentropic efficiency of the radial turbine versus its isentropic indicator ψ_t [7]

To complete calculations, isentropic efficiency should be checked. For that purpose, isentropic indicator is calculated from the following equation:

$$\psi_{\mathrm{t}} = \frac{2 \cdot H_{\mathrm{ad-t}}}{v_p^2} \qquad (15)$$

Calculation provide the value $\psi_t = 1{,}94$, so that from the graph in Fig. 5 [7] turbine isentropic efficiency can be read. Its value is $\eta_{\mathrm{iz-t}} = 0.85$.

Conclusions

The calculations proved that the power of motorcycle Suzuki GSX-R 750 could be substantially increased with a turbocompressor. The device available in the market was selected, and its characteristics were checked, so that theoretical improvement of 33 kW, which is ca. 29%, was possible. From another perspective, indicator of the power related to the mass could be increased up to 1.1 kW/kg. This way, the discussed motorcycle is able to achieve the power similar to the ones with engines of stroke capacity 1000 cm^3.

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