GT-power simulation report

Honda GX 35, ECOmarathon Vera 2009.

SEMCON

den 2 april 2009

Skriven av: David Willermark och Fredrik Dunert, Engine Test and Optimisation, Semcon

GT-power simulation report

Honda GX 35, ECOmarathon Vera 2009.

The GT-power engine model

An engine model of the GX35 engine was built using GT-power version 6.2 build #10.

Geometrical data

The following data in Tabell 1 was used as model input.

Bore [mm]	39
Stroke [mm]	30
Conrod length [mm]	51
Compression ratio [-]	8.2
Intake valve diameter [mm]	15.5
Exhaust valve diameter [mm]	14
Intake port diameter/length [mm]	10/24
Exhaust port diameter/length [mm]	12.5/40
Exhaust runner diameter/length [mm]	13/325
Intake runner diameter/length [mm]	12/400
Intake valve opening [CAD]	10 BTDC
Intake valve closing [CAD]	57 ABDC
Exhaust valve opening [CAD]	48 BBDC
Exhaust valve closing [CAD]	28 ATDC
Max lift [mm]	2.754

TABELL 1

Valve lift

The lift curves from the GT-power “RunMeFirst.gtm” model was used and scaled to fit the measured lift and duration. Lift vs. CAD tables are shown in Appendix.

Flow data

The intake and exhaust port flow data was measured in Semcons flow bench. The test was performed without inlet and exhaust runners. Clay was used to get a smooth cone-like boundary to the surrounding environment. Reverse flow was not tested. Discharge coefficients are shown in Tabell 2 and Tabell 3 below.

L/d	Cd
0	0
0,032258	0,070618
0,064516	0,141237
0,096774	0,200991
0,129032	0,244448
0,16129	0,260745
0,193548	0,282474
0,225806	0,293338
0,258065	0,304202
0,290323	0,309635
0,322581	0,309635

TABELL 2CD VALUES FOR INTAKE PORT

L/d	Cd
0	0
0,035714	0,077398
0,071429	0,174145
0,107143	0,251543
0,142857	0,303141
0,178571	0,32894
0,214286	0,354739
0,25	0,367639
0,285714	0,374089
0,321429	0,380539
0,348571	0,386988

TABELL 3 CD VALUES FOR EXHAUST PORT

Heat transfer settings

The Woschni model was used for cylinder heat transfer with head and piston to bore area ratio of 1. The convection multiplier was set to X. The cylinder wall temperatures was set to XXX. The other parameters in the cylinder model used default values.

The pipes had constant wall temperature which was set according to the list below:

Intake runner: 293K

Intake port: 450K

Exhaust port: 550K

Exhaust runner: 1000K

Combustion

The Wiebe model was used and the 50% burned point and durations was set to the assumed values shown in

Engine speed [rpm]	50% burned point [CAD]	Burn duration 10-90% [CAD]
1800	8	40
2400	8	40
3000	8	40
3600	8	40
4200	8	40
4800	8	40
5400	8	40
6000	8	55
6600	8	60
7200	8	60
7800	8	60
8400	8	60

TABELL 4 WIEBE COMBUSTION PARAMETERS

Friction

The standard “EngFric” model was used with parameter values according to

Constant FMEP factor [bar]	1
Peak Cylinder Pressure factor	0.008
Mean piston speed factor	0.25
Mean piston speed squared factor	0

TABELL 5 FRICTION MODEL PARAMETERS

Model validation

The friction parameters, intake pressure drop and port wall temperature was adjusted to make the simulated torque, lambda and fuel flow was correspond to the measured curves. The test data was produced 2007, no documentation available. A PI regulator was added to the model to control the injected fuel to achieve the same exhaust lambda as the measurement.

Plots of measured vs. simulated data are shown below, red curve is measurement and blue for simulation. The accuracy of the model is acceptable.

ECO_validation

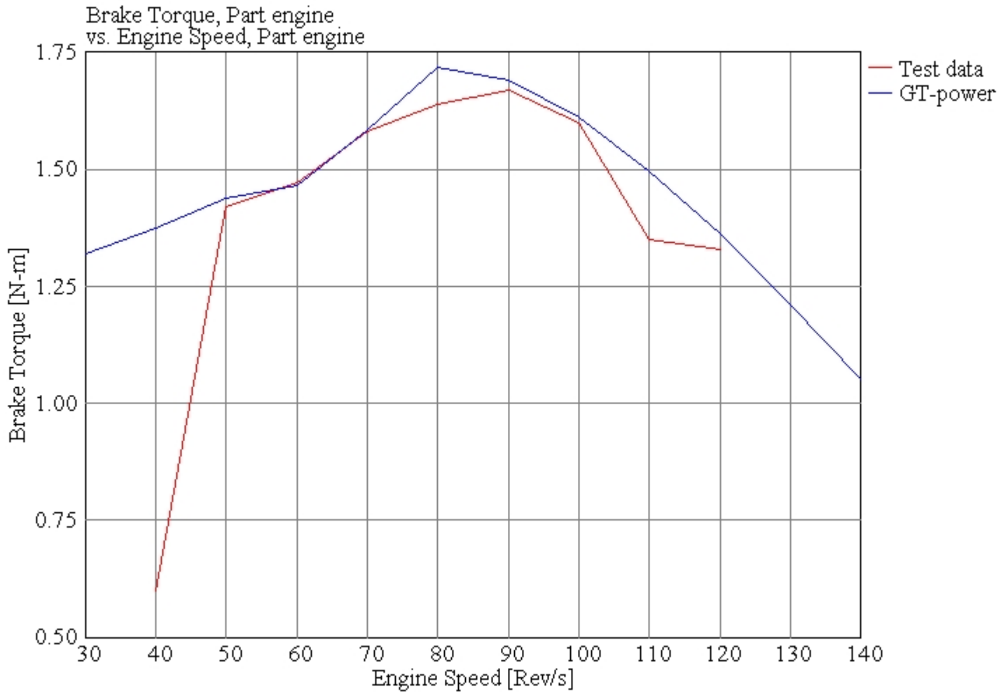

Figure 1 Simulated vs. measured brake torque

ECO_validation

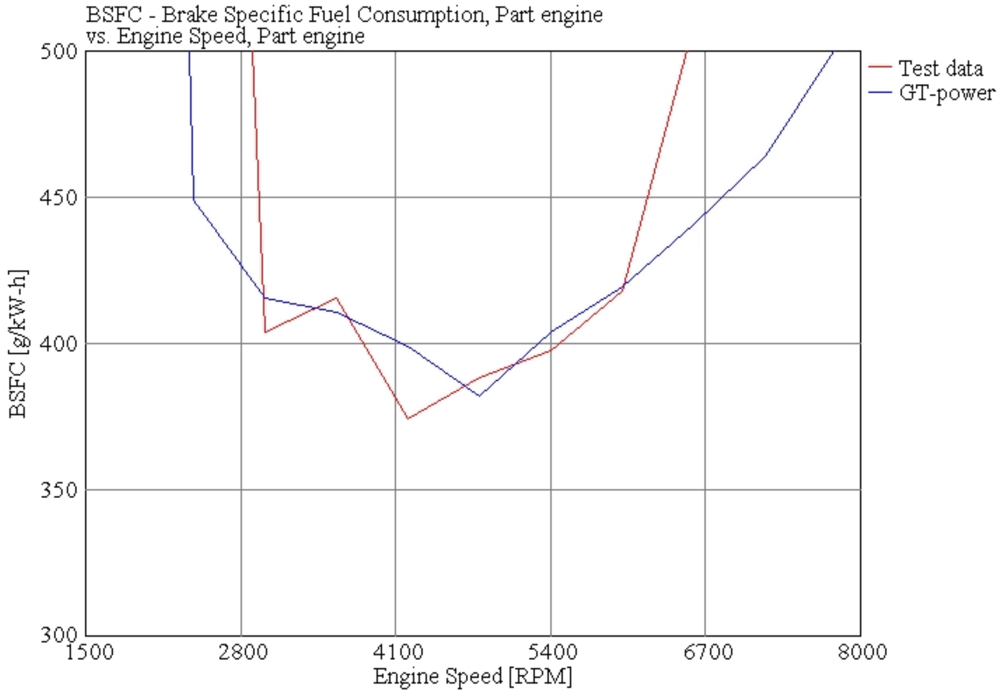

Figure 2 Simulated vs. measured bsfc

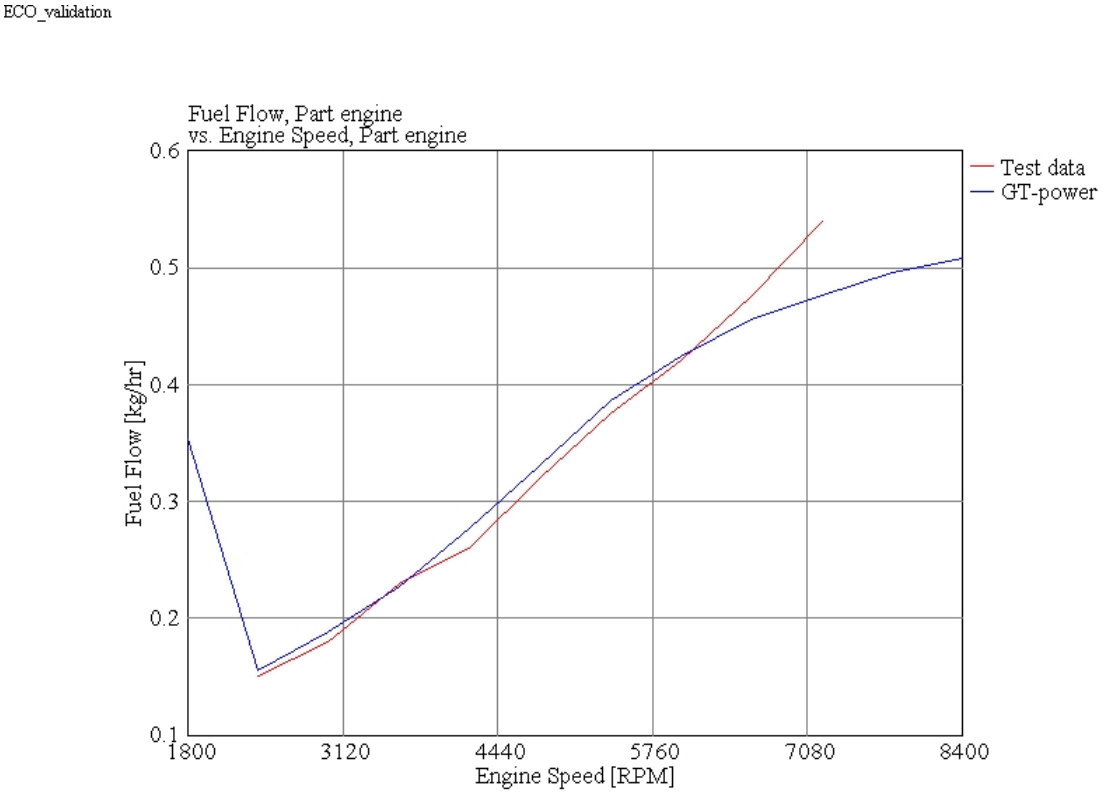

FIGURE 3 SIMULATED VS. MEASURED FUEL FLOW

Camshaft optimization

The objective of this study was to optimize the camshaft with respect to specific fuel consumption between 2000-6000rpm.

The inlet and exhaust runner lengths was optimized with the standard camshaft with respect to BSFC in the before mentioned speed range. Then the camshaft duration was varied by applying a different amount of valve lash. This method was used to avoid increased valve accelerations and to eliminate the need to produce a new camshaft. The valve lash could likely be increased without causing damage to the lifters and valve seats considered the moderate engine speed and low operation time of the engine.

The inlet and exhaust runner lengths from earlier years ECO project appeared to be well optimized. Inlet: 400mm and exhaust 325 mm.

A DOE was run in GT-power varying inlet and exhaust lash. DOE-post was used to optimize the lash with respect to BSFC between 2000-6000rpm. The optimal lashes turned out to be 0.16mm on inlet and 0.13mm on exhaust.

The compression ratio was increased from 8.2 to 11 which increased BSFC at engine speed lower than 5000rpm. The reason for this is that engine friction increases due to higher peak cylinder pressure. Though the friction model settings are estimated based on recommendations from Gamma, it is likely that the peak cylinder pressure factor is overestimated. BSFC and torque curves are shown in Figure 4 and Figure 5.

ECO

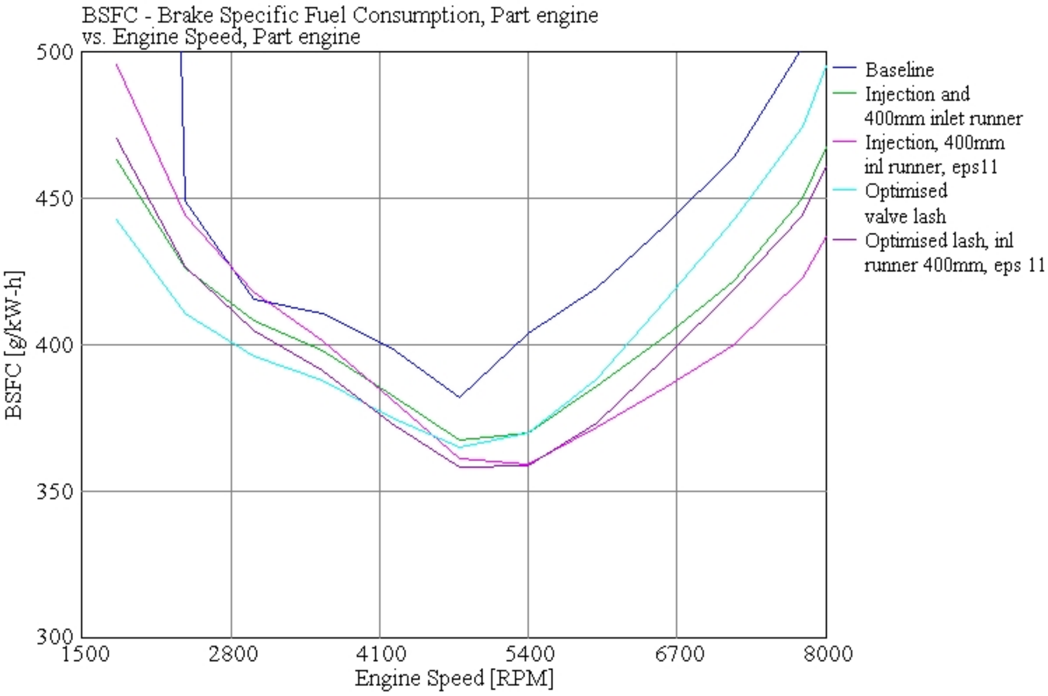

FIGURE 4 BSFC COMPARISON

ECO

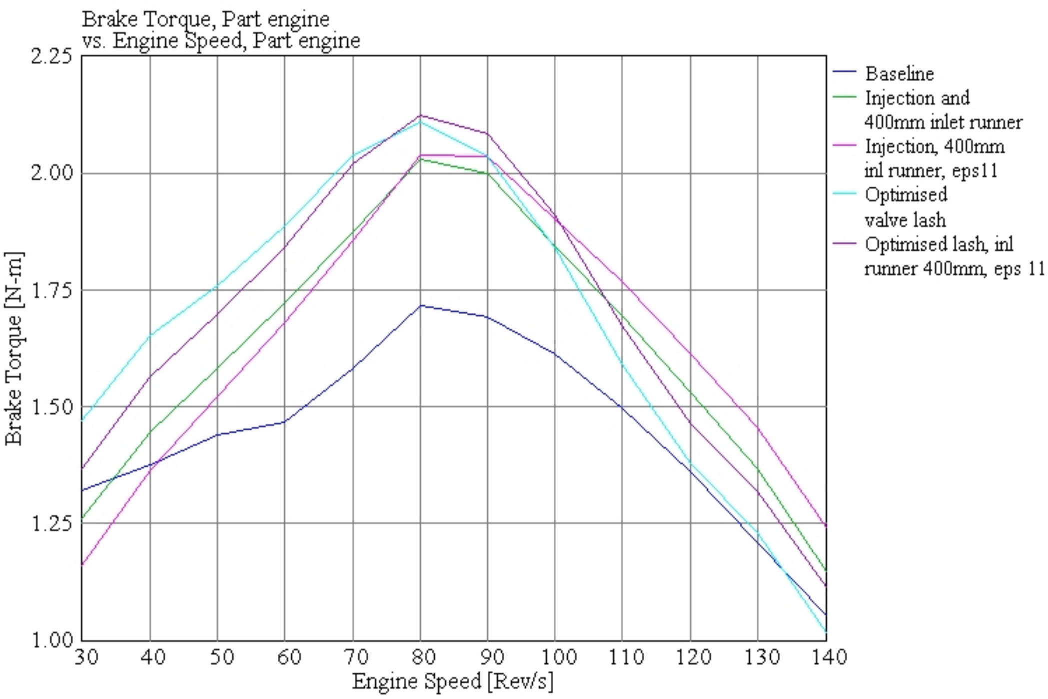

FIGURE 5 TORQUE COMPARISON

Conclusion

The simulations shows that a inlet and exhaust runner length of 400mm and 325mm respectively is optimal with respect to BSFC in the range 2000-6000rpm. A valve lash of 0.16mm on inlet and 0.13 on the exhaust valve reduces BSFC in the same speed range. An increase in compression ratio to 11 may reduce BSFC further, but its effect on engine friction is not known. Engine testing is necessary in order to see the friction effect. Further it is recommended to take measures to reduce engine friction.