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Modeling the Effect of Inlet Manifold Pipes Bending Angle on SI Engine Performance

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***Abstract*—**the intension in this work is to investigate the effect of different bending manifold pipes on engine performance for different engine speed. Power, Torque, and BSFC were calculated and presented to show the effect of varying bending pipes angles on them for all cases considered. A special program used to carry out the calculations. A simulation model for 4-cylinders spark ignition engine with turbocharger has been built and calculated. The analysis of the results shows that for 120^o angle the torque increases about 40% at 3000 rpm and 25% at 4000 rpm without changing in fuel consumption. For 90^o angle the increment in torque is about 10 %. For the same bending angle the increment in brake power is around 40% at 3000 rpm and 25% at 4000 rpm. The increment in fuel consumption is about 12% for 60^o and 30% for 90^o between (6000-7000) rpm.

***Keywords*—**bending pipes, inlet manifold, spark ignition engines, performance

I. Introduction

THE control of engine performance and gas emissions has begun to add to the numerous constraints that vehicle manufacturers have to satisfy. The reduction of engine fuel consumption becomes a primary requirement as well as meeting current and future emission legislations. Naturally, talking about reduction of engine fuel consumption means to keep unvaried, sometimes improved, the performance level of current engine production [1-3]. Dealing with engine topics exclusively to improve fuel economy means improving the engine thermal efficiency [4] .This target can be met following different routes, each of them could be an effective way with different cost-to-benefit ratio. Often, it could be observed, it is helpful to adopt numerous solutions contemporaneously. As an example, fast combustion, lean burn, variable valve timing and actuation, manifold bending angle, gasoline direct injection and so long may be reminded [5-7]. It is known that load reduction in spark-ignition engines is traditionally realized by introducing additional losses during the intake stroke by means of manifold and throttle valve. The optimization of intake manifold length and bending angels can provide significant increase in engine performance [8, 9]. The total intake mass flow rate and the effective compression ratio of the engine can be controlled by adjusting the manifold pipes length and angles [10].

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The objective of this paper is to contribute towards the development and pursuing, of variable manifold pipes bending angles for improving the engine performance to optimize engine torque, power and fuel consumption. The effect of engine speed has also been considered.

II. Theoretical analysis

A. Modeling the Flow through a Valve

The valve lift profile is specified by a polynomial consisting of four coefficients and four exponents. The nature of the polynomial is such that the sum of the coefficients is -1.When gas flows through a valve the development of separation and recirculation regions gives rise to a vena-contract where the actual cross-sectional area of the gas stream (effective area) is less than the geometric area of the orifice. This phenomenon cannot be simulated directly using a one-dimensional model and has to be characterized using empirical data. Data giving measured effective valve areas, or flow coefficients (C_f), are required as input values to Lotus Engine Simulation. There are several other boundary features which require similar information or data giving the variation of pressure drop with mass flow rate across the device (for example throttles).

The effective area of a valve is a hypothetical concept which enables the mass flow through the valve to be evaluated for a given pressure difference across it. A mathematical model of the flow through the valve is developed, from which the effective' area of the valve throat can be derived from the measured values of pressure across the valve and the mass flow rates. The value of effective area obtained is dependent on the particular mathematical model Woods and Khan and therefore if the data is to supplied to a wave-action simulation program it is imperative that the model used to analyze the steady-flow data matches that employed in the boundary model of the computer program. In this way the use of effective flow area measured using a steady-flow rig enables the mass flow rate obtained in the experiments, for a particular valve lift and pressure difference across it, to be reproduced by Lotus Engine Simulation.

B. Combustion Process

The combustion process employed a single zone combustion model. The heat release rate can be defined using empirical heat release functions or to be defined explicitly by the user in the form of an angle verses heat release rate curve. The empirical heat release functions are derived from the Wiebe equation. Dissociation effects (CO generation) were modeled through curve fits to the Eltinge diagram, which relates combustion products of CO and O2 to user specified parameters of air-fuel ratio and mal-distribution [11].

This approach avoids the computationally expensive chemical rate calculations.

The Wiebe function defines the mass fraction burned as:

$$m_{frac} = 1.0 - \exp^{-A\left[\frac{\theta}{\theta_b}\right]^{M+1}} \tag{1}$$

Where

A = coefficient in Wiebe equation = 10.0 for gasoline

M = coefficient in Wiebe equation = 2.0 for gasoline

Θ = actual burn angle (after start of combustion) calculated by the program

Θb = total burn angle (0-100% burn duration)

C. Heat Transfer

Heat transfer was modeled in all elements. Within cylinders the empirically derived heat transfer correlation proposed by Annand [12] was employed.

The connective heat transfer model proposed by Annand is defined as;

$$\frac{hD_{cyl}}{k} = A\,\mathrm{Re}^B \tag{2}$$

Where

h = heat transfer coefficient [W/m^2 K]

A = Annand open or closed cycle coefficient = 0.2

B = Annand open or closed cycle coefficient = 0.8

k = thermal conductivity of gas in the cylinder [W/m K]

Dcyl = cylinder bore

Re=Reynolds number based upon mean piston speed and the engine bore. The density is that calculated for the cylinder contents at each crank angle.

Thus the heat transfer per unit area of cylinder wall is defined as:

$$\frac{dQ}{A} = h\,\left(T_{gas} - T_{wall}\right) + C\,\left(T_{gas}^4 - T_{wall}^4\right) \tag{3}$$

Where:

dQ /A = heat transfer per unit area [W/m^2]

C = Annand closed cycle coefficient.

The first part of the heat transfer equation is the convective heat transfer and the second part is the radiative heat transfer.

D.Simulation tool

For the purpose of analyzing the engine characteristics the dimensions were considered with Lotus Engineering Software. The Lotus Engine Simulation and analysis program is an in-house code developed by “LOTUS ENGINEERING” since the late 1980’s. The aim of the program is to predict the gas flows, combustion and overall performance of internal combustion engines. There is a wide range of engine types and features which can be simulated using this program such as: two-stroke or four-stroke engines; arbitrary cylinder arrangements and firing intervals; DI or IDI diesel, or SI combustion systems; combustion rates via 1 or 2 part Wiebe functions or user profiles; and so on.

Validation of global performance parameters of power, torque, volumetric efficiency and fuel consumption has been performed on a wide range of current production engines. Detailed validation of many of the sub-models for predicting cylinder pressure, combustion, heat transfer, and inlet and exhaust system gas dynamics has also been performed. The solution of the equations represents the physical processes to predict the flows between the elements of the model. It is designed to solve the energy, momentum and continuity equations as appropriate within each element to obtain the thermodynamic state variables and flow velocity at each crank angle throughout the engine cycle. The solution procedure is ‘time marching’ and a number of engine cycles are simulated in order to obtain a converged (cyclically repeatable) solution.

In this analysis, a model of four cylinders gasoline engine with turbocharger has been built and investigated. Two simulations models one with simple manifold pipes and the second with bending manifold pipes for comparison are shown in figure (1).

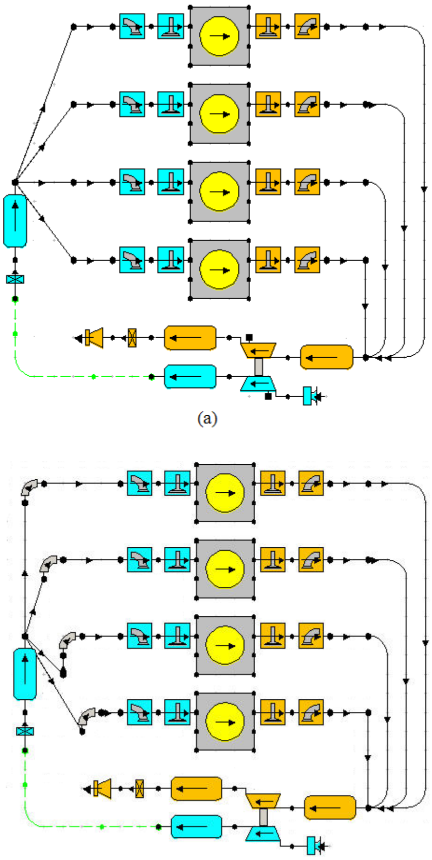

Fig. 1 Simulations models with simple (a) and bending (b) manifold pipes

The engine geometry, running conditions and valves data are shown in Table (1). All parameters were kept constant through the simulation. The turbocharger parameters and geometrics are shown in Table (2). The efficiency maps for compressor and turbine are shown in figures (2) and (3) respectively. The bending pipes geometry is shown in table (3). The test is carried out for bending angles (0, 60, 90, and 120 degree). The brake power, torque, and fuel consumption were calculated and compared with results obtained for simple pipes.

TABLE I
BASE ENGINE GEOMETRY

No. of cylinders	4
Bore	87 mm
Stroke	84 mm
Connecting rod length	130 mm
compression ratio	8
Heating value	43000 kJ/kg
H/C molar	1.8
molecular mass	114.23 kg/k.mol
Max. valve lifts	8.5 mm
IVO angle	15° bTDC
IVC	60° aBDC
EVO	40° bBDC
EVC	20° aTDC

TABLE II
BASE TURBOCHARGER DATA

Inlet Dia. (mm)	50.00
Outlet Dia. (mm)	40.00
Rot. Inertia (kg.m²)	4.0000e-006
Gear Ratio to Shaft	1.00
Drive Gear Mech Eff. (0-1)	1.000
Inlet Dia. (mm)	40.00
Outlet Dia. (mm)	60.00
Rot. Inertia (kg.m²)	4.0000e-006
Gear Ratio to Shaft	1.00
Drive Gear Mech Eff. (0-1)	1.000

TABLE III
MANIFOLD PIPES GEOMETRY

Total length [mm]	30
Start diameter [mm]	39
End diameter [mm]	39
Bend angle [deg.]	0, 60, 90, 120
Bend radius [mm]	20
Cooling type	Air cooled

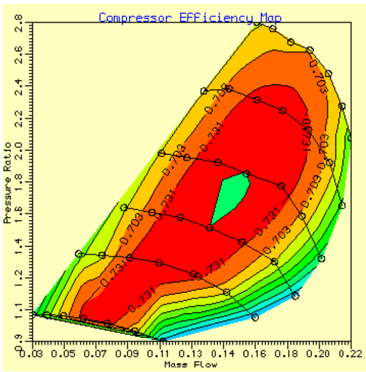

Fig. 2 Compressor efficiency map

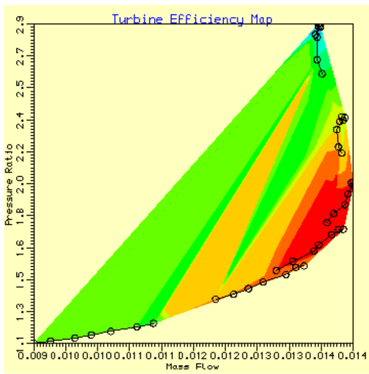

Fig. 3 Turbine efficiency map

III. RESULTS AND DISCUSSION

For the engine geometry and running conditions shown above, all parameters were kept constant except the pipe's bending angle. It was varied from the original value 00 to 120o in steps. Figure (4) shows the torque versus bending angle for different engine speeds between (1000 - 7000 rpm). It shows slight change in torque for low engine speed less than

2500 rpm and high speed more than 5000. This effect is more recognized with engine speeds between (2500-5000 rpm). The maximum torque is obtained for bending angle 1200.

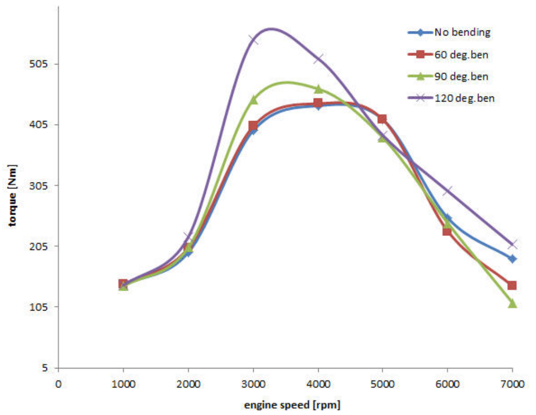

Fig. 4 Brake torque for different bending angle and engine speed.

Figure (5) shows the variation of brake power for different bending angle and speed. It shows an increase of brake power for all engine speed range with 90o bending angle. The maximum power is obtained for 60o angle at 5000 rpm.

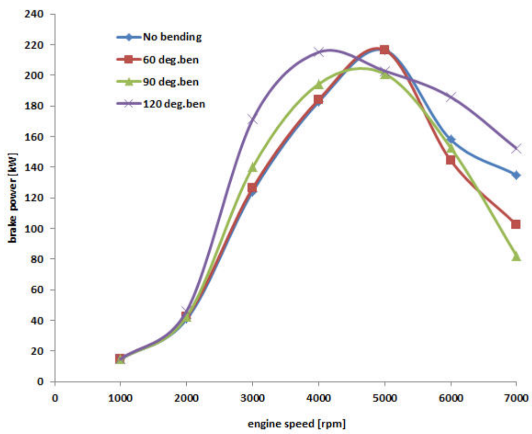

Fig. 5 Brake power for different bending angle and engine speed

Figure (6) shows the variation of BSFC versus engine speed and bending angle, this shows that BSFC is not affected by bending angle for engine speeds (<6000 rpm). But it was insensitive to bending angle for engine speeds between (6000-7000 rpm).the maximum fuel consumption is obtained at 90^o angle for 7000 rpm.

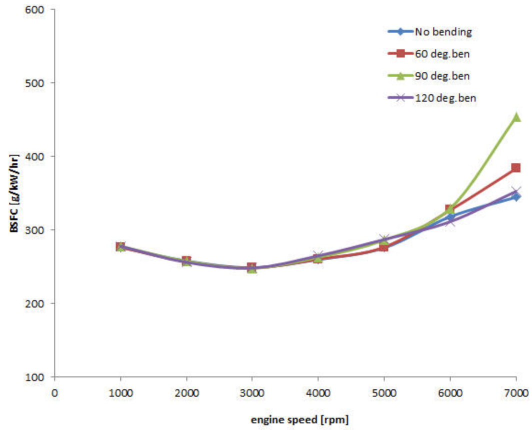

Fig. 6 Fuel consumption for different bending angle and engine speed

IV. Conclusions

1. For 120o angle the torque increases about 40% at 3000 rpm and 25% at 4000 rpm without changing in fuel consumption. For 90o angle the increment in torque is about 10 % for the same speeds.
2. The increment in brake power is around 40% at 3000 rpm and 25% at 4000 rpm for 120o.
3. The increment in fuel consumption is about 12% for 60o and 30% for 90o between (6000-7000) rpm. For speed less than 4000 rpm there no effect on the fuel consumption for all considering angles.

References

[1] Najjar, Y. and AbuKayyas, H."Performance of Automotive Reciprocating Engines with Different Fuels", Journal of Fuel Science and Technology International, Vol. 6, No. 3, pp. 315-327, 1988.
[2] Najjar, Y. and Al-Turki, A."Modeling and Simulation of Spark-Ignition Engines", Journal of Fuel Science and Technology International., Vol.14, No. 8, pp 993-1018,1996.
[3] Najjar,Y. and Al-Turki, A."Modeling and Simulation of Compression-Ignition Engines", Journal of Fuel Science and Technology International, Vol. 14, No.8, pp1019-1035, 1996.
[4] Heywood, J.B., "Internal Combustion Engines Fundamentals", McGraw Hill, 1988.
[5] Fontana, G., Galloni, E., "Variable valve timing for fuel economy improvement in a small spark-ignition engine"; Applied Energy 86 (2009) 96–105
[6] Ahmad, T. and Thobald, M., "A survey of variable valve actuation technology"; SAE, New York, SAE paper 89, 674, 1989.
[7] Gray, C., "A review of variable engine valve timing"; SAE, New York, SAE paper, 880 - 386, 1988.
[8] Ferguson, R. and Kirkpatrick, A."Internal combustion engines applied thermodynamics", 2nd Edition, John Wiley and sons inc., New York, 2001.
[9] Baur, H., "Gasoline-engine Management - Basic components"; Bosch Company, 1st edition, Stuttgart, 2001.
[10] Stone, "Introduction to Internal Combustion Engines"; McMillan Press, 3rd edition, London, 1999.
[11] Eltinge,L.," Fuel-Air Ratio and Distribution from Exhaust Gas Composition", SAE 680114. 1968.
[12] Annand, W.J.D.," Heat Transfer in the Cylinder of Reciprocating Internal Combustion Engines", Proc.I.Mech.E, 177-973, 1963.