Design strategy for six-cylinder stationary diesel engine exhaust systems

PAULDURAY SEENI KANNAN[1*],
VELAYUTHAMPALAYAM MUNIAPPAN PERIASAMY[2],
PERIA SAMY NAGARAJ[3]

Overall engine performance of an engine can be obtained from the proper optimized design of engine inlet and exhaust systems. This paper presents the influence of an exhaust manifold pressure on the performance improvement of a reciprocating four-stroke engine. The application of computational methods for the development of high performance of a four-stroke stationary engine has been evaluated. A one-dimensional fluid dynamic code has been employed to simulate engine performance at full load, and data predicted from computer simulation have been compared with experimental results. To do this aforementioned validation process, computer simulation techniques were applied to develop a new exhaust system so as to optimize volumetric efficiency over a wider range of speed. These techniques proved to be powerful and effective in identification of the modifications required to obtain the engine performance targets. Findings from these studies are used to derive exhaust system design guidelines that define the optimum exhaust system geometry to give effective scavenging and fine cylinder charging.

Key words: stationary diesel engine, mass flow rate, engine performance, exhaust manifold, computer simulation

1. Introduction

Compression ignition engine operates with a much higher compression ratio than spark ignition engine, thus it has higher efficiency. The effect of heat transfer was analysed in terms of compression ratio as design parameter. The energy analysis with respect to exhaust systems has been carried out by Parlak [1]. Breathing of an engine is largely dependent on the design of the intake and exhaust systems.

[1] Department of Mechanical Engineering, Sethu Institute of Technology, Kariapatti, Virudhunagar District-626106, Tamilnadu, India
[2] Crescent Engineering College, Chennai, T. N., India
[3] Department of Mechanical Engineering, Mepco Schlenk Engg. College, T. N., India
[*] Corresponding author, e-mail address: seenipkannan@yahoo.co.in

The use of computer simulation techniques for the improvement of engine development process has been rapidly expanding over the last decade. Computational methods predict the influence of geometrical modification on engine performance, thus reducing the number of prototypes and experimental tests required for engine optimization, but also allow a better understanding of the unsteady flow phenomena that occur during the gas exchange processes. Shuhn-Shyurng Hou studied the heat transfer effects and pointed to an importance to provide good guidance for the performance evaluations of practical diesel engines [20].

Instantaneous exhaust, residual fraction, mass flow rates of engine system can be obtained from the computer simulation techniques, while these engine physical parameters are difficult to be measured experimentally. To achieve optimum cylinder charging, it was observed that during the exhaust event, the phasing of compression and rarefaction waves in the exhaust port was of critical importance. These design strategies have been studied extensively and clearly by Bush [17].

Volumetric efficiency is defined as the ratio between the air mass flowing into the cylinders from the intake manifold and the air mass theoretically contained in the cylinders at the manifold temperatures. In general, it represents a measure of the effectiveness of the air pumping system composed by the intake manifold, the inlet port, valve and the cylinders. The flow through the exhaust system is also considered for the effectiveness of the air pumping system, by making the cylinders empty before the scavenging process indirectly. The value of the volumetric efficiency mainly depends on the engine parameters like crankshaft speed, the intake and exhaust pressures, air-fuel ratio, the design geometry of the system and many others. Modelling the volumetric efficiency of internal combustion engines with various techniques is demonstrated in the study by De Nicolao [8].

Computer simulation has been used extensively in the development of intake and exhaust systems [6]. However, considerable effort can still be required to identify systems that will achieve optimum cylinder charging and scavenging characteristics with minimum pumping losses. Recognizing this, various authors have presented simplified design criteria for intake systems [7, 12]. Similarly the design of an exhaust manifold to improve transient performance of a high speed diesel engine has been presented by Galindo [9]. Since engine breathing is very sensitive to the design of the intake system and exhaust system, various methodologies have been presented specifically for its design [4, 14].

Design methodologies for exhaust systems are of varying complexity. Some are based solely on the application of acoustic theory [5] whereas others require the use of models of successively increasing complexity and ultimately the use of a full gas-dynamics simulation to identify an optimum design [16]. This paper presents the study of the gas exchange process and the effect of exhaust system design geometry on this process. From these results design criteria are presented for six-cylinder, four-stroke engine which when satisfied, give the optimum phasing of pressure waves at the exhaust manifold to assist cylinder emptying and filling.

The aim of this paper is therefore to provide an example of the integrated use of computational and experimental techniques for the development of high performance four-stroke stationary diesel engine. Base engine characteristics features are described in section 2. In section 3 the general base engine computer simulation model is described and the results of the validation process for matching the simulation model with the real engine are presented. Details of the experimental measurements and computational results are provided in section 4. Section 5 presents the discussion of enhancement of engine performance of the simulation analysis. The paper concludes with a summary of findings in section 6.

2. Base engine characteristics

The experimental investigation was conducted on a six-cylinder diesel engine, 4.75 litre, direct injection, stationary-type diesel engine. The technical data of the engine are given in Table 1. Initially, an extensive experimental program was undertaken, during which key engine performance indicators together with exhaust and intake system manifold pressure and temperature data were recorded on the base engine. Two configurations of exhaust system were installed and the measured data from each test were used to validate predictions from the GT-Power6.2 engine simulation code [10] that uses the method of characteristics for the calculation of unsteady flow in the manifold systems.

The principal dimensions and features of this base engine are presented in this section. Figures 1, 2 show the brake power and brake torque output at full load for this engine and the pressure-volume diagram of the base engine, respectively. The basic concepts of pressure-volume diagram for the thermodynamic processes performed by ideal air standard diesel cycle have been presented and some guidance

Table 1. Main design features of the test engine

Type	Six-cylinder diesel engine
Type	Vertical diesel engine
Bore	104 mm
Stroke	100 mm
Compression ratio	16 : 1
Max speed	2400 rpm
Maximum BHP	110 hp
Type of cooling	Air-cooled
Firing order	1, 5, 3, 6, 2, 4

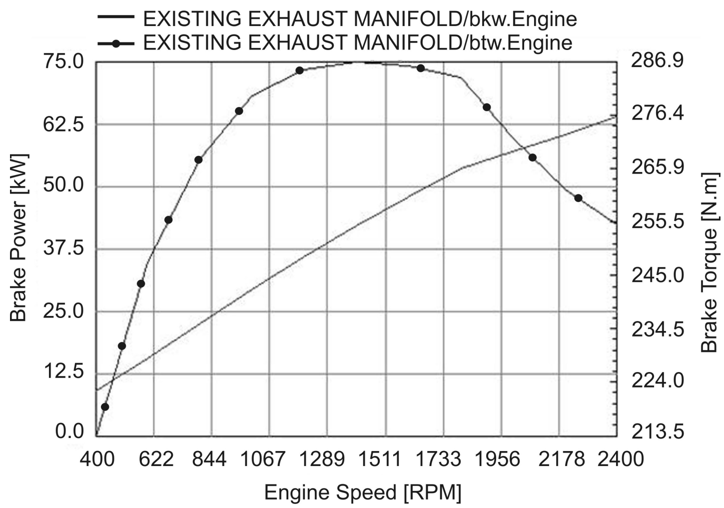

Fig. 1. Brake power and brake torque output at full load.

for design improvement and effect of heat transfer on the performance of diesel engines has been provided by Akash [2].

The pressure record at the exhaust manifold has a significant influence on cylinder scavenging and pumping losses. The effect of the phasing of pressure waves at the exhaust manifold was systematically studied using the validated model. It is not possible to generate a low pressure at the exhaust port throughout the exhaust

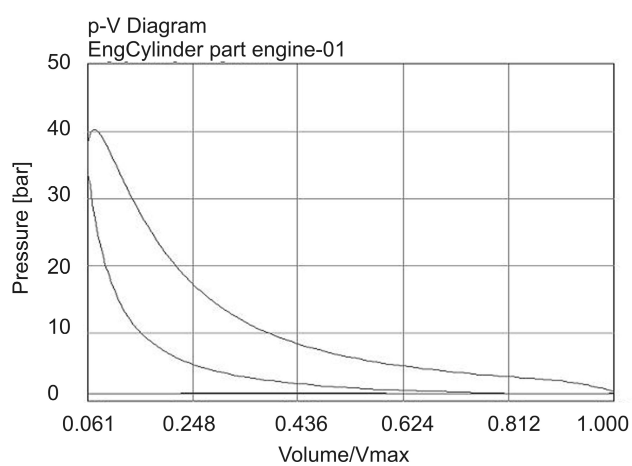

Fig. 2. Pressure-volume diagram of the base engine.

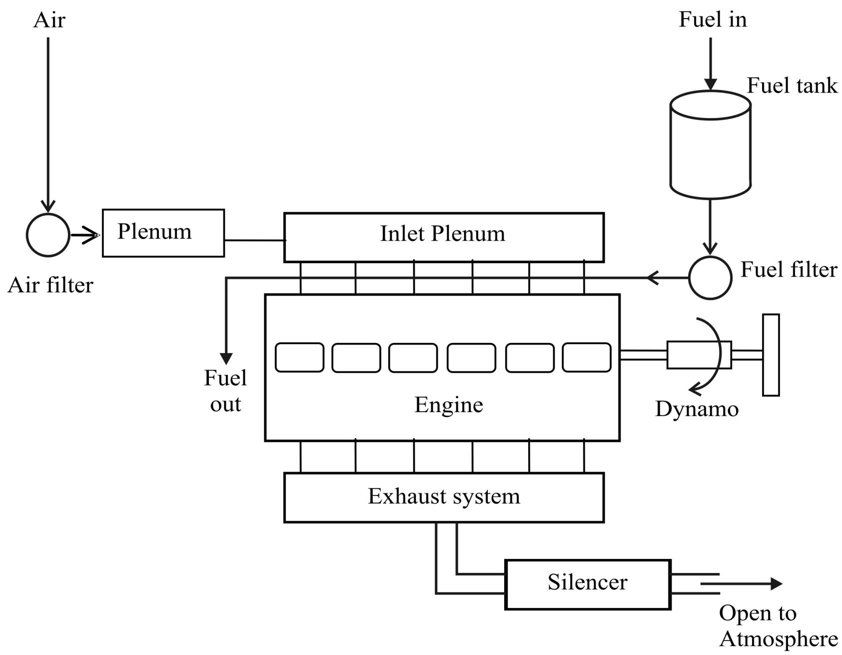

Fig. 3. Base engine experimental test rig.

event and so this phase of the study identified critical periods when engine performance was most sensitive to compression and expansion pressure waves. Figure 3 shows a diagram of the base engine experimental test rig.

Further exhaust system gas dynamics was studied. The effect of exhaust system junctions and components on the transmitted and reflected characteristics of pressure waves was demonstrated, as was the pressure wave resonance characteristics of pipes. It was shown that the pressure history at the exhaust manifold can be well represented by the superposition of a limited number of incident pressure waves originating from key exhaust system components.

Using the findings from the above processes, design criteria for exhaust system manifolds were proposed that, when satisfied, gave the optimum phasing of pressure waves at the exhaust port to assist cylinder gas exchange and reduce pumping losses. The original application of an existing acoustic model was used to assist with the design process; full gas-dynamic simulations were not required until the final stage of this process.

3. Base engine computer simulation model

A one-dimensional fluid dynamic code developed by Gamma Technologies Inc. for engine performance prediction has been used for computer simulation. The general features of the code GT-Power6.2 are described in [10], while a brief overview is reported in this paper. In GT-Power6.2, the intake and exhaust systems are represented as a network of ducts connected by junctions. The junctions represent either physical joints between the ducts, such as area changes or volumes, or subsystems such as the engine cylinder. 1-D calculation methods based on simple analytical solution methods of computational fluid dynamics for design purposes of intake and exhaust systems in internal combustion engines are presented by Rosello [19].

The solution of the equations governing the conservation of mass, momentum and energy for each element of the network is then obtained using a finite difference technique. Pressure losses in ducts due to wall friction are automatically computed through a simple approach based on the Reynolds number, whereas losses in the junction are governed by discharge coefficients, which can be either automatically computed by the code, according to junction geometry, or specified by the user, according to the experimental data obtained on a steady state flow bench.

As far as the ducts are concerned, the heat transfer coefficient is modelled proportional to the friction using the Colburn analogy, whereas the heat transfer coefficient for the cylinder model is calculated using the Woschni correlation [21]. The combustion model for diesel engines uses a Wiebe relationship or the complete combustion heat release profile, which has to be specified by the user. Finally, the engine friction is modelled using a modified form of the Chen-Flynn correlation, an empirical relationship that determines the total engine friction as a function of the peak cylinder pressure, of the mean piston speed and of the mean piston speed squared.

The base engine model construction requires a careful and detailed schematization of the engine, intake and exhaust system geometry, as well as accurate and extensive experimental data, which have to be carefully analysed to properly set the intake and exhaust valve discharge coefficients, the combustion model and the engine friction estimation. Kandylas evaluated the engine exhaust system based on heat transfer computation. During the study and investigation the steady state and transient heat transfer measurements in automotive exhaust systems were analysed with computer model covering all exhaust piping configurations [13].

After the construction of the model, a detailed validation process is required to access the accuracy and reliability of the engine model. The simulation results should therefore be compared with the experimental results over the whole engine operating range. The main results of the validation process, which is required to access the engine model accuracy and reliability, are presented.

4. Experimental measurements and computational results

After completing the base line engine validation, the existing exhaust manifold and Y-section exhaust manifold were modelled and then simulated with the base engine model. The second manifold was manufactured identical to the existing one. This second manifold has been designed with Y junction geometry design

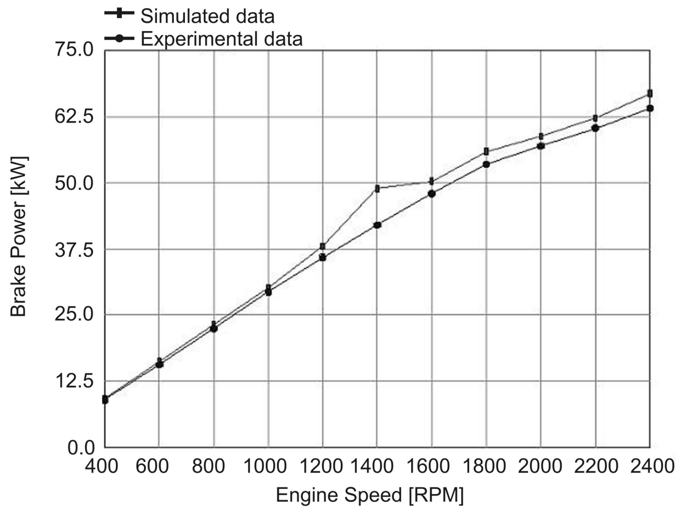

Fig. 4. Base engine model validation (brake power at full load).

characteristics and attributes. Engine performance tests were performed with these two manifolds, the results obtained have been plotted versus engine speed between the speed ranges of 400 rpm to 2400 rpm, both the experimental measurement tests and the computer simulation were carried out under full throttle open. The engine was fuelled with a rich mixture and a relative air fuel ratio from 9.3 to 10.4 was used. Figures 4 and 5 show the comparison between the experimental

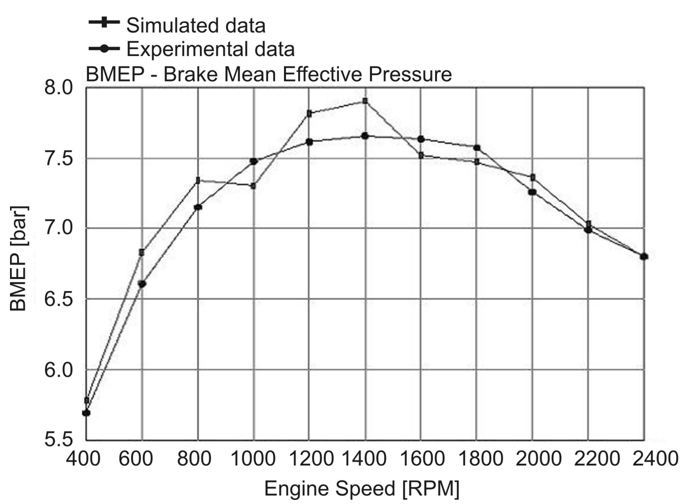

Fig. 5. Base engine model validation (BMEP at full load).

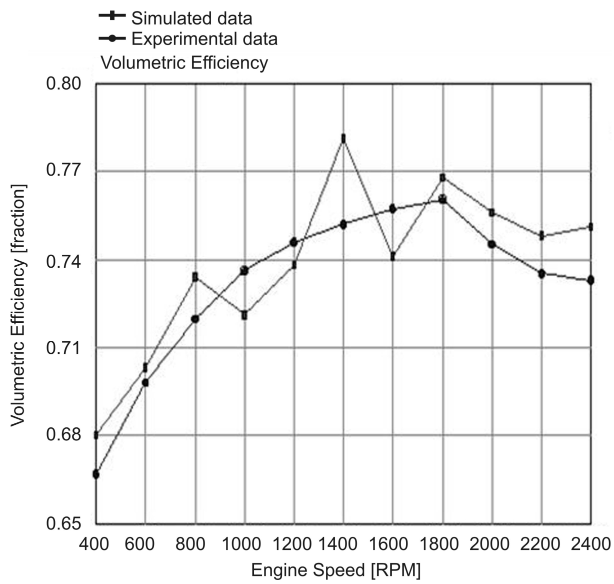

Fig. 6. Base engine model validation (volumetric efficiency at full load).

measurements and calculated curves for engine power and brake mean effective pressure, respectively. The calculated data are quite close to the experimental data, with differences between 2 % and 4 % on average. The computer simulation seems to slightly overestimate the engine performance over the whole speed range, although the differences between the computed and experimental data are relatively modest.

High values of brake mean effective pressure of the value 7.6 bar are obtained in a speed range of 1200 rpm to 1400 rpm, although a sudden fall of brake means effective pressure after a value from 1600 rpm. Thus a base engine is compared and correlated to make confidence in the engine model's predictions.

The comparison between the experimental measurement and calculated curves for the volumetric efficiency is shown in Fig. 6. It can be seen that the simulation results are quite close to the experimental data, with differences, which are, on average, below 3.5 %, even though some larger discrepancies can be observed at maximum speed and at 1400 rpm. The volumetric efficiency drop may be due to a less effective scavenging process during the valve overlap interval. The experimental test results and the computer simulation results emphasize decrease of volumetric efficiency after 1800 rpm, which is clearly responsible for the corresponding torque decrease, the numerical simulation can provide more detailed information and it was found that this is due to the less effective induction process during the engine speed 1400 rpm.

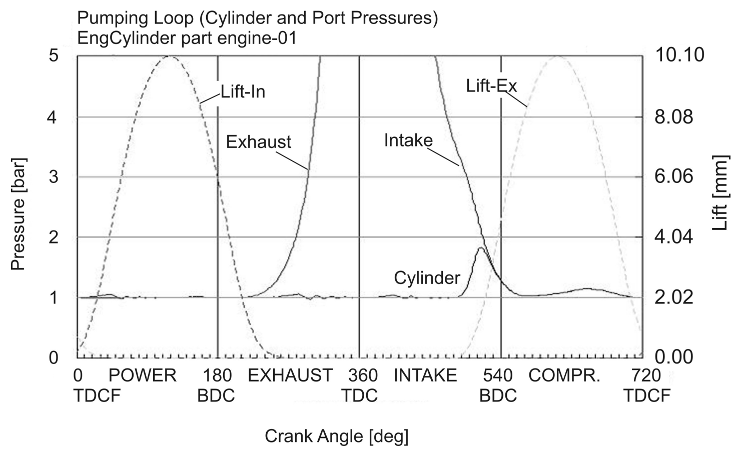

Fig. 7. Valve lift profiles and calculated pressure traces at 2200 rpm.

It has been discussed in the previous work that all the above-represented results for validation process should agree with the engine modelling [4]. The evolutions of the valve lift profiles and calculated pressure traces at various speeds are recorded and the results for 1400 rpm and 2200 rpm are plotted. The computer simulation results are obtained for the calculated pressure traces, which are presented in Fig. 7. After the first exhaust pressure peak that follows the blowdown phase, a second pressure pulse, which is due to pressure wave propagation

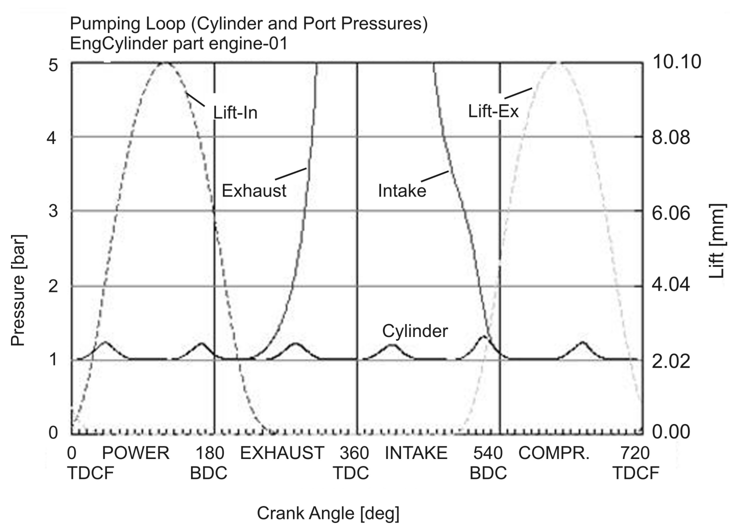

Fig. 8. Valve lift profiles and calculated pressure traces at 1400 rpm.

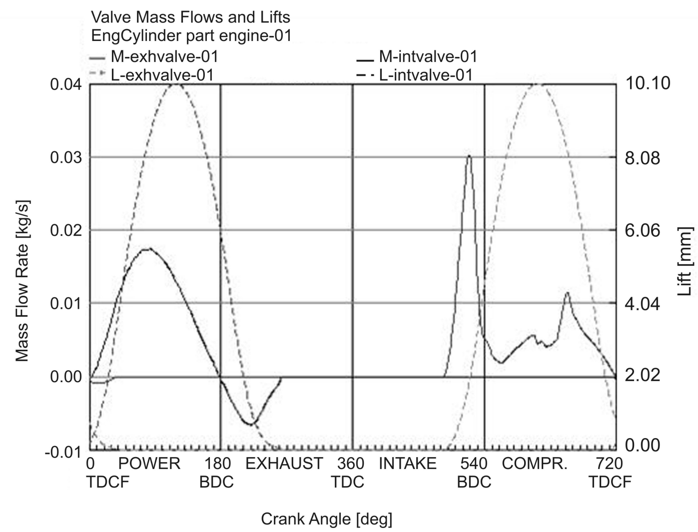

Fig. 9. Valve lift profiles and calculated pressure traces at 2200 rpm.

and reflection phenomena in the exhaust system, reaches the exhaust valve during the overlap interval, maintaining pressure levels that are higher than the intake pressure until exhaust valve closing. This unfavourable pressure gradient causes a negative intake mass flow during overlap, which is responsible for the high trapped

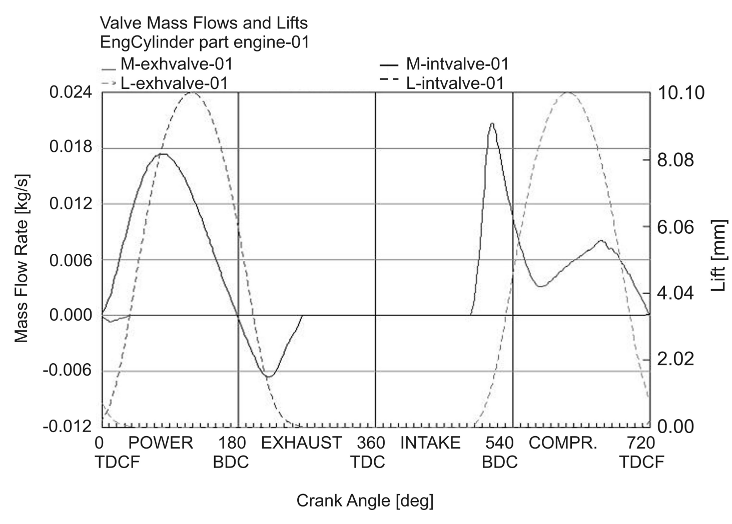

Fig. 10. Valve lift profiles and calculated pressure traces at 1400 rpm.

residual fraction. This could be noted from the valve lift profiles and calculated pressure traces at 1400 rpm (Fig. 8).

An almost ideal situation could be observed by analysing the pressure traces and the mass flow rates in Figs. 9 and 10, calculated at 2200 and 1400 rpm, where the volumetric efficiency and the trapped residual fraction reach its maximum and minimum values, respectively.

5. Discussion of enhancement of engine performance

Power output characteristics of various types of exhaust pipe collection with comparison of power output characteristics of the base engine and developed engines are described by Parlak [15]. The objective of exhaust system tuning is to optimize the dynamic pressure history in the exhaust port during the exhaust event in order to improve the gas exchange process. Conventional exhaust tuning theory indicates that volumetric efficiency is influenced by the exhaust port pressure during valve overlap but that cylinder pumping losses are influenced by the port pressure during the period BDC (exhaust) to EVC. This theory was tested simulating the performance of an engine fitted with two different exhaust systems by Bush [17].

Evaluation of engine models for volumetric efficiency with real engine measurements and development of new models under various pressure values are derived

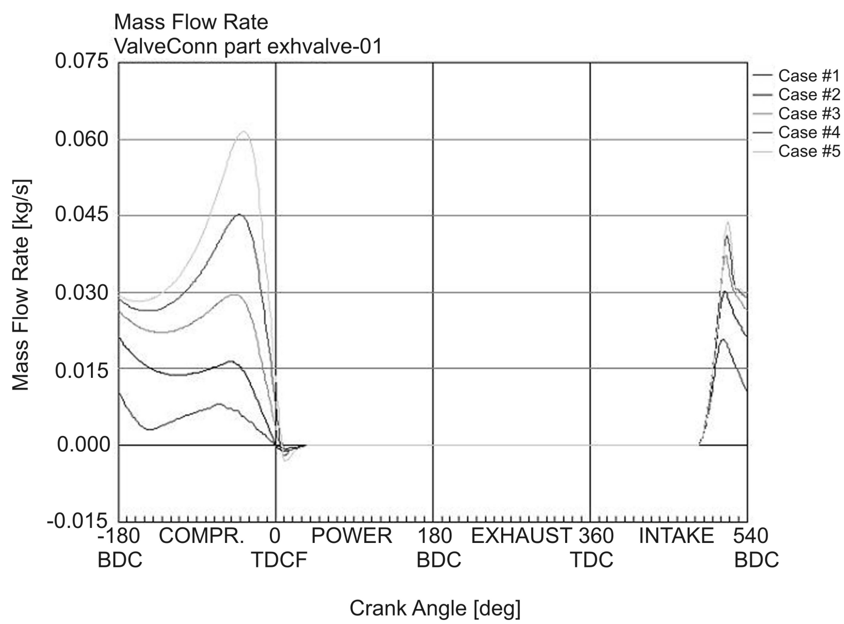

Fig. 11. Crank angle vs. mass flow rate (existing manifold).

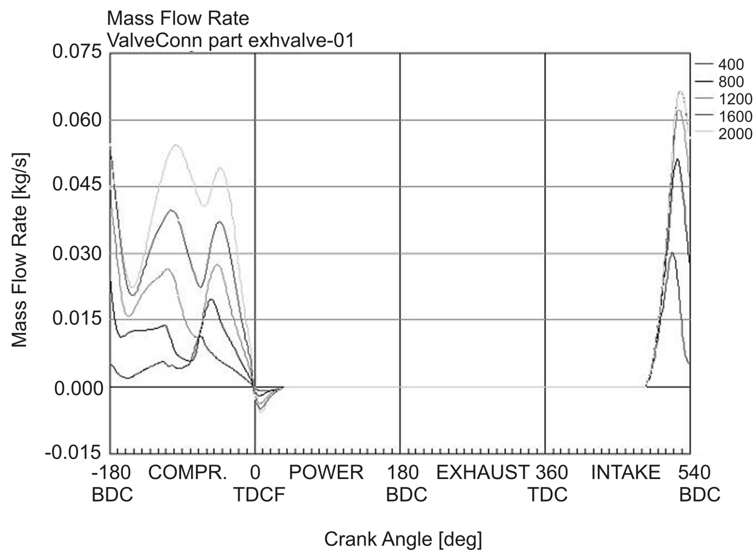

Fig. 12. Crank angle vs. mass flow rate (Y-section manifold).

and demonstrated by Bengtsson [11]. From the above study and validation of the paper, the overlap and residual gas model shows an overall good stable behaviour. Crank angle versus mass flow rate for existing and Y-section manifolds plots is presented in Figs. 11 and 12, respectively.

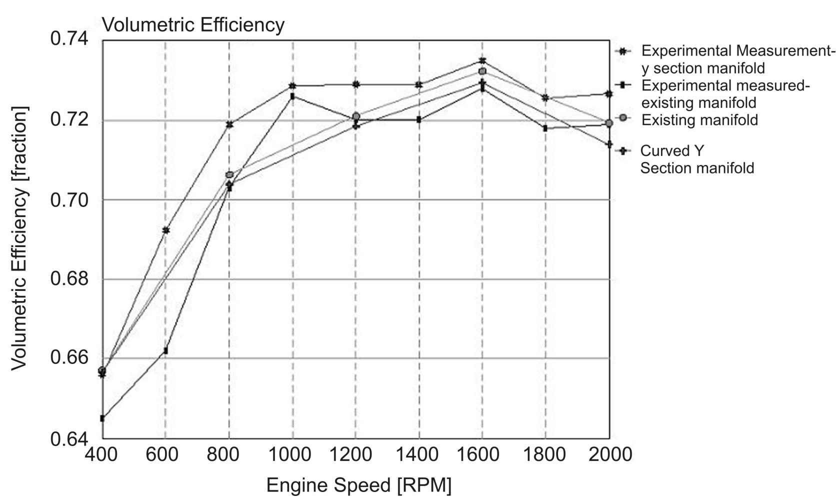

Fig. 13. Comparison between volumetric efficiency of existing and Y-section manifold.

(a) (b)

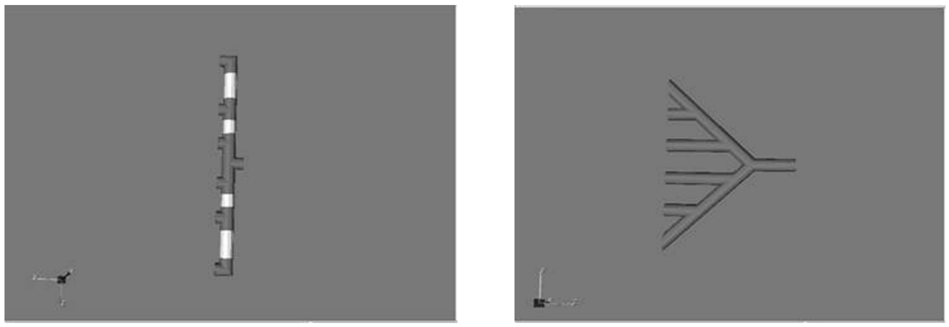

Fig. 14. Shape comparison between existing (a) and Y-section (b) manifolds.

For comparison, the mass flow rates for both manifolds at 400 rpm are 0.024 $kg \cdot s^{-1}$ and 0.030 $kg \cdot s^{-1}$ during the intake process of the base engine. Similarly, the mass flow rates for both the manifolds at 2000 rpm are 0.042 $kg \cdot s^{-1}$ and 0.065 $kg \cdot s^{-1}$.

The evolution of the volumetric efficiency and overall engine performance of the base engine fitted with both the manifolds can be observed in Fig. 13. The developed design of Y-section manifold is enough to increase the volumetric efficiency with enhancing the engine power while comparison is made with existing manifold. A considerable increase in volumetric efficiency has been observed from the plot. The analysis of these calculated variations may be still improved by adopting exhaust energy saving systems. The shape comparison between the existing and Y-section manifolds is shown in Fig. 14a,b.

6. Summary and conclusions

The experimental work was supplemented by a comprehensive theoretical analysis that included a detailed simulation code of design and optimization of the engine, which is specially developed, taking into account the engine performance.

GT-Power6.2 computer simulation techniques were adopted not only to gain an in-depth understanding of the one-dimensional unsteady flow character that occurs during the scavenging and exhaust processes, but also for development of the Y-section exhaust system so as to enhance engine volumetric efficiency over a wider speed range.

The higher volumetric efficiency obtained with the Y-section exhaust manifold could allow a substantial enhancement of brake mean effective pressure in the low and medium speed range, with a significant improvement.

Comparison and careful observation of the experimental and theoretical results facilitate to investigate the phenomenon further, proving insight into many interesting aspects of mass flow rates and valve lift when the engine operating conditions are concerned.

REFERENCES

[1] PARLAK, A.: Energy Conversion and Management, *46*, 2004, p. 167.
[2] AKASH, B. A.: Int. Comm. Heat Mass Transfer, *28*, 2001, p. 87.
[3] BADAMI, M.—MILLO, F.—GIAFFREDA, G.: Experimental and computational analysis of a high performance four-stroke motorcycle engine equipped with a variable geometry exhaust system. SAE Paper 2002-01-0001, 2002.
[4] BENAJES, J.—REYES, E.—GALINDO, J.—PEIDRO, J.: Pre-design model for intake manifold in internal combustion engines. SAE paper 970055, 1997.
[5] BENAJES, J.—REYES, E.—BERMUDEZ, V.—SERRANO, J. R.: Pre-design criteria for exhaust manifolds in internal combustion engines. SAE paper 980783, 1998.
[6] BENSON, R. S.—GARG, R. D.—WOOLLATT, D.: International Journal of Mechanical Science, *6*, 1964, p. 114.
[7] BINGHAM, J. F.: In: I. Mech. Engg. Conference, No. C25/87, 1987, p. 111, ISBN-10: 0852986254.
[8] DE NICOLAO, G.—SCATTOLINI, R.—SIVIERO, C.: Control Engg. Practice, *4*, 1996, p. 1405.
[9] GALINDO, J.—LUJAN, J. M.—SERRANO, J. R.—DOLZ, V.—GUILAIN, S.: Experimental Thermal and Fluid Science, *28*, 2004, p. 863.
[10] GAMMA TECHNOLOGIES Inc., GT-POWER User's Manual, 2007.
[11] BENGTSSON, H.: Modelling of Volumetric Efficiency on a Diesel Engine with Variable Geometry Turbine. [Master Thesis]. Linköping, Linköpings University 2002.
[12] HALL, K. G.: Design optimization of engine induction systems. Autotech new engines, components and materials seminar, C399/20, 1989.
[13] KANDYLAS, I. P.—STAMATELOS, A. M.: Energy Conversion & Management, *40*, 1999, p. 1057.
[14] LIU, J. P.: Development of Design Guidelines for Engine System Design from Fundamental Gas-dynamics Studies. [Ph.D. Thesis]. Glasgow, The University of Strathclyde, Dept. of Mech. Engg. 1995.
[15] PARLAK, A.—YASAR, H.—SAHIN, B.: Energy Conversion & Management, *44*, 2003, p. 163.
[16] PAYRI, F.—DESANTES, J. M.—CORBERAN, J. M.: I. C. Engine intake and exhaust manifold design procedure. SAE paper 905234, 1990.
[17] BUSH, P.—TELFORD, C.—BOAM, D.—BINGHAM, J.: A design strategy for four cylinder SI automotive engine exhaust systems. Journal of SAE, 2000-01-0913, 2000.
[18] RASIHHAN, Y.: Heat Recovery Systems and CHP Journal, *15*, 1990, p. 1.
[19] ROSELLO, M. D.—SERRANO, J. R.—MARGOT, X.—ARNAU, J. M.: Mathematical and Computer Modelling, *36*, 2002, p. 33.
[20] SHUHN-SHYURNG HOU: Energy Conversion and Mangement, *45*, 2004, p. 3003.
[21] WOSCHNI, G.—SPINDLER, W.—KOLESA, K.: In: Proceeding of Congress International SIA Moteurs Diesel, Society of Automotive Engineers. Warrendale, PA, 1988, p. 142.

Received: 13.8.2007
Revised: 17.12.2007