A REVIEW OF PHYSIO-THERMODYNAMIC FACTORS INFLUENCING FUEL CONSUMPTION IN INTERNAL COMBUSTION ENGINES FOR AUTOMOTIVE APPLICATIONS

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ABSTRACT

The development of automotive engine and subsequent traction have a long history. From the gunpowder piston engine of Christian Huygen (1629-1695), Jean Lenoir steam engine (1822-1900), the Nicolaus Otto compressed gas-air mixture engine to the introduction of compression ignition engine by Rudolf Diesel (1858-1913)[1]. The humble development was for development of a more efficient engine for the automotive industry. To date, one recognizes further development in the state of affairs as turbo, supercharged and even hydrogen propelled engines are introduced to improve performance.

The engine, the vehicle and operating conditions affect the performance of the automotive engines. Consequently they affect fuel consumption. In this work, physio-thermodynamic factors affecting fuel consumption are explored. This forms the platform for ways to improve fuel consumption in internal combustion engines. As automotive vehicles propelled by engines burning petroleum fuels presents environmental impacts, these should be considered alongside the gains. It is estimated that road transport releases over $1x10^9$ tones of carbon dioxide (CO_2) annually which represents 15% of the World's total discharge [2] Efficiently operating engines could reduce this amount.

INTRODUCTION

The hypothetical Otto and Diesel cycle engines are ideal extremes for a practical engine showing parameters governing cycle efficiency. Respectively; the cycle efficiency are given by

$$\eta_{Otto} = 1 - \frac{1}{r_v^{\gamma-1}} \tag{1}$$

$$\eta_{Diesel} = 1 - \frac{1}{r_v^{\gamma-1}}\left[\frac{\alpha^{\gamma}-1}{\gamma(\alpha-1)}\right] \tag{2}$$

The Otto cycle efficiency depends both on γ and r_V. γ is dictated by the thermodynamic state of the working fluid. The compression ratio can be increased to the limitation of the engine block material. On the other hand, the Diesel cycle efficiency depends on r_V, γ and α.

The load ratio lies in the range $1 < \alpha < r_V$, consequently, the expression in the square brackets is always greater than unity making Diesel cycle efficiency less than Otto cycle efficiency of the same compression ratio. However, the compression ratios of compression ignited (CI) engines are usually greater than those of spark ignited (SI) engine. This fact eventually makes the cycle efficiency of Diesel engine greater than its Otto counterpart.

For Diesel cycle efficiency, $\alpha \rightarrow 1, \eta_{Diesel} \rightarrow \eta_{Otto}$. This can be shown by writing the term in the square brackets, to a Binomial expansion;

$$\frac{\alpha^{\gamma}-1}{\gamma(\alpha-1)} = \left[\frac{\{1+(\alpha-1)\}^{\gamma}-1}{\gamma(\alpha-1)}\right] = \tag{3}$$
$$\left[\frac{1}{\gamma(\alpha-1)}\left\{1+[\gamma(\alpha-1)]+\frac{\gamma(\gamma-1)}{2!}(\alpha-1)^2+\ldots-1\right\}\right]$$

As $\alpha \rightarrow 1$, then $(\alpha - 1) \rightarrow 0$ and the $(\alpha - 1)^2$ and higher terms can be neglected, hence the term in the square brackets tends to unity

If one extends the analysis for a cycle modelling combustion at isochoric, isobaric and isothermally, the expression becomes:

$$\eta_{th} = \frac{\alpha\beta^{\gamma}\sigma^{\gamma-1}-1}{r_v^{\gamma-1}[\alpha-1+\gamma\alpha(\beta-1)+(\gamma-1)\ln\sigma]} \tag{4}$$

For process in the actual engine, the situation is even complicated and assumptions have to be invoked to enable calculation of the cycle parameters. In addition to these thermodynamic parameters afecting cycle efficiency, physical factors also affect the vehicle performance. Preventive and planned maintenance are effective in vehicle upkeep as well as breakdown maintenance. Breakdown maintenance related problems mostly originate from drivers behaviour in relation to road condition. Poor road condition accelerate wear and tear of vehicle parts. Also roads determine speed, rolling resistance and so operating characteristics of vehicles. Fuel consumption of motor vehicles are favoured with mid-speed of travel and part loads.

PHYSICAL AND THERMODYNAMIC FACTORS AFFECTING FUEL CONSUMPTION

Vehicle design

The effect of vehicle design on fuel consumption is described by the fuel consumption equation[3]:

$$B_e = \frac{\int b_e . \frac{1}{\eta_u}\left[\left(m.f.g.\cos\alpha + \frac{\rho}{2}.C_w.A.v^2\right) + m(a + g.\sin\alpha) + B_r\right].v.dt}{\int v.dt} \quad (5)$$

In the fuel consumption equation, three groups of influencing resistance variables are realized; the engine, transmission and the external driving resistance.

The influence of engine is in its specific fuel consumption (sfc),

$$b_e = \frac{B}{P_{eff}} = \frac{v_B.\rho_B.3600}{t_B.P_{eff}} \quad (6)$$

The effect of transmission depends on both the transmission losses, which are to be kept as low as possible, and on selected transmission ratios (i). Transmission losses are minimized by increasing transmission eficiency of the power train, η_u. Transmission ratios determine the operating point on the engine consumption map for a given road speed. '"Longer" transmission ratios, where the speed reduction ratio is small, generally move the operating point into a low sfc value on the map. It is advantageous to use overdrive units in engine transmission as they allow to drive at high speeds at low engine RPM. Nearly 17% energy saving is realised by making use of these transmissions [2]. Exemplifying this, Fig. 1 shows fuel consumption map for two engines, A and B, with transmission ratios iA and iB respectively. Transmission ratio B (iB) is '"longer" than transmission ratio A (iA) such that iB = 0.8 iA. sfc of these engines are compared through ratio A (iA) such that iB = 0.8 iA. sfc of these engines are compared through curves of respective driving resistance against the curve of constant engine power for a road speed of 120 km/h. It is found that iB shows a reduced fuel consumption of the order 8%.

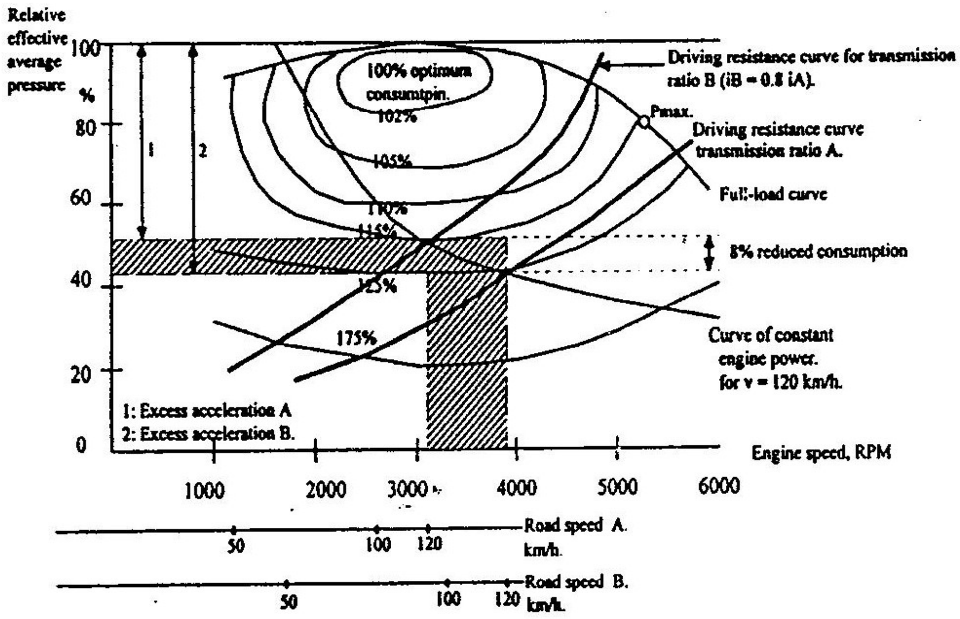

Fig. 1: Engine fuel consumption map fordifferent transmission ratios[3]

External driving resistance are represented by parameters in the square brackets of Eq. (5). They constitute the rolling resistance, aerodynamic drag, acceleration resistance, incline resistance and braking resistance. These have a direct impact over vehicle fuel consumption rate. Incline resistance increases uphill and has no effects on level roads, and it reduces consumption downhill. Rolling resistance is maximum on level roads. External driving resistance can be reduced by lowering the vehicle height, reducing the rolling resistance and improving vehicle aerodynamics as air resistance increases with the square of the vehicle speed. In addition, it depends on the shape of the vehicle. Designers should strive to simulate the shape of the vehicle body to approach that of a drop as it ofers best results aerodynamically. It is reported that in a series-manufactured vehicle, a reduction of 10% in vehicle weight, aerodynamic drag and rolling resistance improved fuel consumption by approximately 6%, 3%, and 2% respectively [3].

In Eq. (5) a distinction is made between acceleration resistance [m(a +

gsinα] and braking resistance (Br). Acceleration increases consumption particularly when subsequent deceleration is affected by the service brake. Otherwise the kinetic energy of the vehicle is used to move it forward and consumption is somewhat reduced.

Road condition

Road conditions affect operating characteristics of motor vehicles by influencing the speed of travel and rolling resistance. Rolling resistance increases with increasing road roughness by an order of 100% from smooth to very rough roads [4]. Poor road surface condition causes slippage between road surface and rubber tread. Slippage is even more pronounced in unpaved to uncompacted roads. Poor road condition accelerate wear and tear of the automotive engine and other inter-connected parts of body. An increase of up to 35% in parts consumption can result from increasing road roughness by a range of 1000 mm/km [5].

Congested roads plays a negative role in motor vehicle fuel consumption. Highly congested roads forces automotive engines to run at low speeds and idling. Fuel enrichment necessary for idling consumes more fuel.Time factor comes in congested roads as more time is spent to travel a specific distance. As an example, about ECU (European Currency Unit) 2.5 billion worth of fuel is being wasted per year in Europe due to trafic congestion while the cost of traffic inefficiency is put over ECU 100 billion [6]. Congestion has the effect of elevated rate of exhaust gases emissions, consequently, less congestion could reduce pollution up to 50%.

Vehicle maintenance and fuel consumption

Vehicle maintenance reduces vehicle downtime in case of breakdown and planned maintenance. In either case, technical and financial matters come in. The personnel doing the maintenance must have acquired the technical skill so as to assure the workability and life of the system. Skills shows better in good working environment and specialized equipment.A properly done maintenance will avail the vehicle at its optimal operating condition. As an example, a well maintained vehicle free from oil leakage into the clutch plate prevents power loss via clutch slippage. The carburettor or injector pump needs special care. Poor performing carburettors or injector pumps result into high rates of fuel consumption.

Lubricants are supposed to reduce friction and wear, and protect against corrosion. These are also required to rinse and cool engine parts. Lubrication oil contamination occurs primarily due to mechanical impurities by dust and worn-off particulate, crankcase leakage (in form of fuels and combustion products), and water through leaking cylinder head liners. About 5% of fuel in oil lessens the ability to lubricate by 30%. The oil itself ages with time and the ageing process is favoured by high operating temperatures. Because of the deterioration of oil it is recommended to be changed at specific periods of time. This varies between 2,000 to 6,000 km for engine oil, and between 10,000 to 20,000 km. for transmission and axle oils. Soiled oil must be drained at operating temperature. This eases flow ability and dirty is rinsed off better. Inadequate engine lubrication increases wear and tear to the engine liner and all other rotating parts. Worn-out piston rings results in excessive loss of compression and hence power. With poor compression fuel consumption is exaggerated and becomes difficult to start diesel engines. Properly lubricated rotating parts reduces power loss through friction and thereby increasing mechanical efficiency. Greasing is usually done after 1,500 to 9,000 km depending on the location.

Air cleaners are used to clean the air and muffles the suction noise. When dirty or dust enter the engine cylinder they act as abrasives which increases the rate of wear. Dust increases the amount of carbon deposits and in two-stroke SI engines it bridges the gap at the electrodes of spark plugs. The air sucked by the engine contains 0.001 to 0.002 g of dust per cubic metre of air on asphalt-paved roads and highways. On dirt roads, construction sites and cross-country the amount of dust may come up to 0.2 g per cubic metre air [7]. Dry type air cleaners of felt, fabric or paper element be cleaned before 25,000 km. and changed thereafter. Clogged cleaners restrict the flow of air. The engine loses power and consumes more fuel as the mixture becomes rich.

The primary importance of vehicle electrical system is in cranking the engine. In SI engines, electrical energy is used to initiate combustion by igniting the compressed air-fuel mixture. Approximately 0.2 MJ. of energy per individual firing is required to ignite the mixture at stoichiometric conditions. Lean and rich mixtures may require over 3.0 MJ. This energy is only a fraction of the total energy in the spark. Combustion misses occurs in cases of insufficient energy. The generator is usually examined every after 30,000 km. It is checked for proper belt tension, fastening terminals, brushes and commutator's working condition. The high voltage side of the ignition coil reserves the electrical energy in form of magnetic field. This needs changing at appropriate time as a malfunctioning system reduces the high voltage reserves, resulting in misfiring and combustion misses. Consequently, engine performance is reduced and fuel consumption increases. In severe cases, the engine may stall or fail to start, especially when cold.

Engine design

Effects of combustion chamber

Various combustion chambers are in use for both SI and CI engines. Their primary importance is vested in governing the energy conversion process. It is an agreed technology nowadays that an optimum SI engine combustion chamber design is the one that favours faster-burning characteristics. A chamber design where the fuel burning process takes place faster, i.e. occupies a shorter crank angle interval at a given engine speed, produces a more robust and repeatable combustion pattern that provides emission control and efficiency gains simultaneously. A faster-burning chamber with its shorter burn time permits operation with substantially higher amounts of exhaust gas recycle (EGR), or with very lean mixtures, within the normal constraints of engine smoothness in operation and response. Thus greater emissions control within the engine can be achieved, and at part load at this higher level of dilution a faster burning chamber shows an improvement in fuel consumption due to the reduced pumping work, reduced heat transfer (due to lower burned gas temperatures), and reduced amount of dissociation in the burned gases. These chambers are preferred for a reduced fuel consumption in SI engines. Characteristics of fast burn chambers are compact shape, spark plug centrally located within the chamber, constituting two plugs, and increased in-cylinder gas motion by swirl during the induction process or during the latter stages of compression.

In SI engine, fuel consumption and efficiency for different mixture quality, ϕ, depend on engine combustion chamber design. This is exemplified on the performance of 1.6 dm^3 four-cylinder engine with conventional chamber compared to a compact fast-burning high-compression ratio chamber as shown in Fig. 2. For rich mixtures, $\phi > 1$ the performance is comparable.

characteristics of the chamber. The faster burning compact-high compression ratio chamber shows decreasing bsfc until the lengthening burn duration and larger cycle-by-cycle variations cause bsfc to increase. For the slower-burning combustion chamber, this deterioration in combustion starts to occur almost immediately on the lean side of stoichiometric, and fuel consumption is highly noticed for $\phi \leq 0.9$.

Diesel engines utilizes either Open-chamber (termed Direct Injection, DI) or divided-chambers (termed Indirect Injection, IDI). They are designed to avoid excessive maximum cylinder pressure and excessive rate of pressure rise, in terms of crank angle during the combustion process. In DI the mixing of fuel and air depend on spray characteristics and air motion, it is not affected by the combustion process itself. Here, once the compression ratio, maximum speed and operating temperature are selected, the delay angle is determined chiefly by the fuel characteristics. The engines are sensitive to spray characteristics, which must be carefully worked out to secure rapid mixing. Sub-division of the spray and the use of high injection pressures are usually required. Air-fuel mixing is assisted by swirl, induced by directing the inlet air tangentially, or by squish.

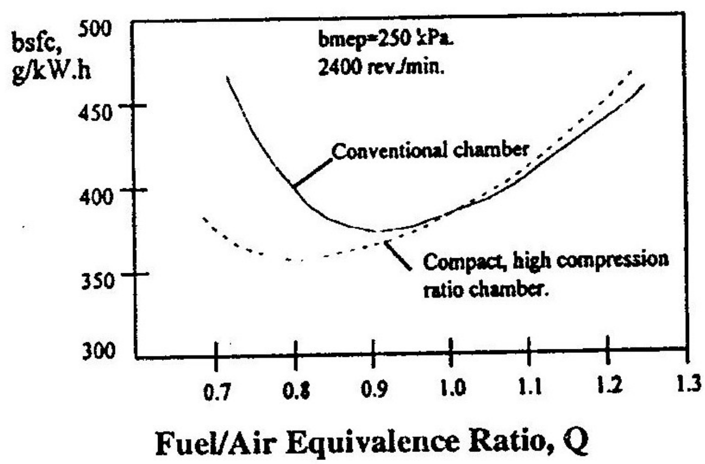

Fig. 2: Effect of combustion chamber design on spark-ignition engine brake specific fuel consumption; Comparing the 1.6 dm^3 four-cylinder engine with conventional chamber to a compact fast-burning high-compression ratio chamber with compression ratio = 13, both at bmep of 250 kPa and 2400 rev/min [8].

In divided combustion chamber during compression stroke, high air velocities are realized through the throat into pre-chamber results into intense turbulence and swirl. In the pre-chamber, being at high fuel-air ratios, combustion is incomplete because there's insufficient air. The high pressure developed projects the unburned fuel, together with the early combustion products into the other part of the chamber with high velocities, causing rapid mixing. This makes it possible to use fuels of poor ignition quality, ability to use single-hole injection nozzles and moderate injection pressures while tolerating greater degrees of nozzle fouling, and the ability to employ higher fuel-air ratios without smoke. The drawbacks to this design include; The requirement of complicated (more expensive) cylinder construction, more difficult starting due to the heat loss through the throat and poor fuel economy due to greater heat losses as a result of increased irreversibility through the throat which causes lower thermal eficiency and higher pumping loss.

Open-chambers tend to be used for engines designed to run at high piston speeds, as in road-vehicles, because heat and friction losses in divided-chambers become more important as speed increase.They are also generally used in case of large cylinders, which are limited to low speed. Having less irreversibility in their thermodynamic process, open chambers are preferred on the basis of having lower fuel consumption than divided chambers.

Effects of Ignition

The combustion efficiency in engines is determined by the degree and rate of combustion. It depends on the timely evolution of heat ensuring an expansion of combustion products close to the geometrical compression ratio. All other conditions being equal, the maximum work, maximum power and efficiency of an engine results when the main fuel combustion commence and terminates almost symmetrical with respect toTop Dead Centre (TDC). This is possible by tuning the moment of ignition, the ignition advance angle, α_z, Fig. 3. The ignition advance angle of SI engines determines the course of the combustion process with respect toTDC and, accordingly, the degree of heat utilization characterized by the indicated efficiency. A change in the ignition advance angle alters the temperature, pressure and turbulence of the charge at the initial moment of combustion.

A prematurely combustion or over advanced may result into severe damage to spark plugs and or engine in extreme cases when the engine knocks.

Exhaust gases becomes toxic. If combustion starts too late the peak cylinder pressure is reduced and the expansion work transfer decreases. Engine power gradually drops (Fig. 3) and fuel consumption increases. At a given speed, mixture composition and flow rate, the optimal timing, Maximum-Brake-Torque timing (MBT) gives minimum brake specific fuel consumption. To achieve this the ignition point is set to the instantaneous engine operating conditions by centrifugal and vacuum spark advance mechanisms. Wrongly set ignition angle change the intended working condition of the engine, the effect on fuel consumption is to increase the consumption. Shown in Fig. 4 are results from computer simulation of engine operating cycle indicating the effects of retarded timing on bsfc. Curves for different operating conditions and burn duration follow the same trend. As high as five degrees of retard, the rate of change of bsfc/bsfc (MBT) with angle of retard is negligibly small, less than 1.5% bsfc/bsfc (MBT) per degree of retard. Above 10 degrees there's a significant increase in bsfc, over 15% bsfc/bsfc (MBT) per degree of retard. Note should be taken for bsfc on both sides of MBT

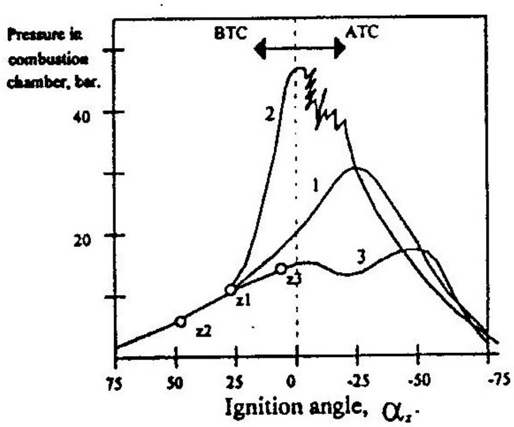

Fig. 3: Pressure curve in the combustion chamber for various ignition points.
1: Correct ignition point setting, Z1.
2: Ignition over-advanced, Z2.
3: Ignition over-retarded, Z3.

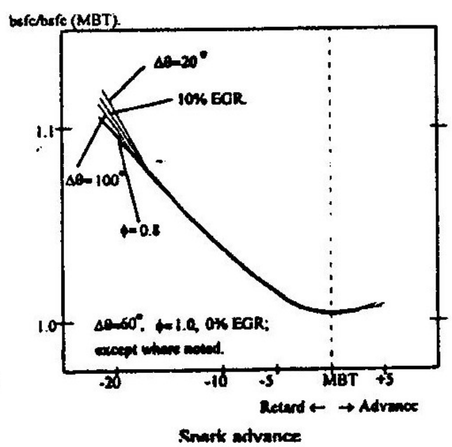

Fig. 4: Effects of spark timing on bsfc of an engine for different part load operating conditions and burn duration, Δθ [9]

Effects of fuel injection parameters

Fuel injection timing in CI engines control the crank angle at which combustion starts. The state of air into which the fuel is injected changes as injection timing is varied, consequently, varying ignition delay. The fuel injection rate, fuel nozzle design and fuel-injection pressure afect the characteristics of the diesel fuel spray and its mixing with air in the combustion chamber.

In diesel engines the injection advance angle, θ_{inj} is varied between 14-25oC. When there's early injection the ignition lag increases as air temperature at the start of injection is comparatively low The combustion process occurs early causing maximum pressure beforeTDC. This causes a sharp increase in the work of compression, decrease in the work of expansion with the consequence of deterioration of performance parameters. Retarded fuel injection causes a slowly rising pressure which adversely influence performance parameters.The optimum injection advance angle is dependent on the type of mixture formation and the speed and load of an engine. It is a precise selection for each engine. Fig. 5 exemplify the effect on performance of varying injection timing in (a) a medium-swirl DI diesel engine and (b) an IDI engine. At a given speed and constant fuel delivery per cycle, the DI engine shows an optimum bsfc and thus bmep at a specific start of injection.The IDI engine experiments conducted at fixed bmep shows bsfc at full load and fuelling rate at idling. In Fig. 5b minimum values at specific injection timings are shown. Injection timing which is more advanced than this optimum results in combustion starting too early before top dead centre while retarded injections results in too late combustion with the obvious known effects.

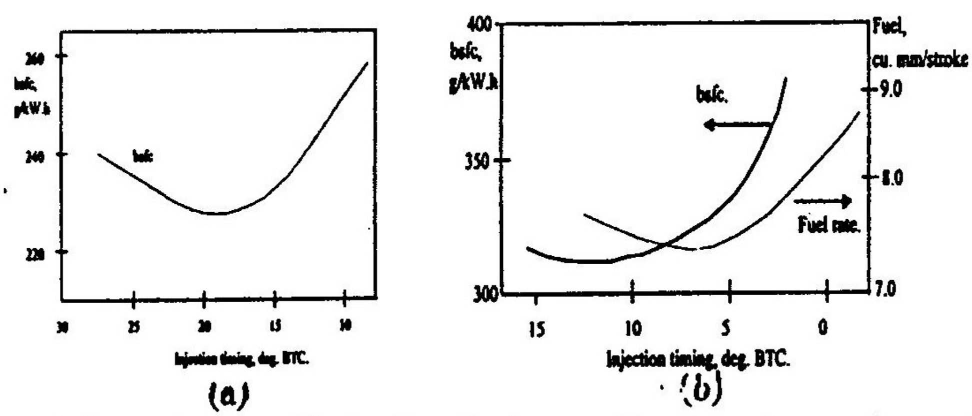

Fig. 5: Effect of start of injection timing on diesel engine performance. (a) medium-swirl DI diesel engine with deep combustion bowl and 4-hole injection nozzle, 2600 RPM, fuel delivery 75 mm3/cycle, fuel/air equivalence ratio 0.69. (b) Swirl-chamber IDI engine, 2500 RPM and 100% load[10]

Injection rate depends on the fuel-injection nozzle area and injection pressure. Higher injection rates result in higher fuel-air mixing rates, and higher heat-release rates. For a particular amount of fuel injected per cylinder per cycle as the injection rate is increased the optimum injection timing moves closer to top dead centre. Fig. 6 shows the effect of injection rate and timing on bsfc in a naturally aspirated DI diesel engine. the higher heat-release rates and shorter overall combustion process that result from the increased injection rate decrease the minimum bsfc at optimum injection timing; however, a limit to these benefits is eventually reached.

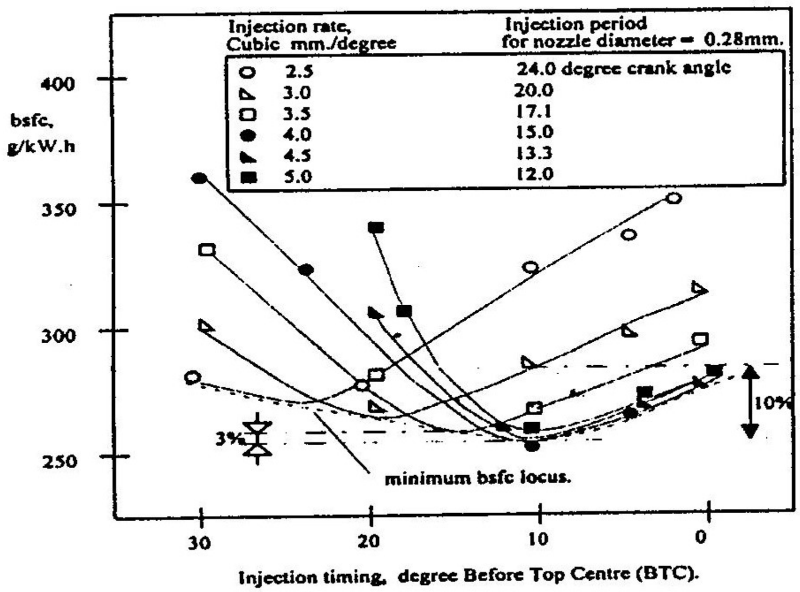

Fig. 6: Effects of injection timing and injection rate on bsfc. 0.97 dm^3 single cylinder naturally aspirated DI diesel engine with swirl. 2000 RPM, 60 mm^3 per stroke fuelling rate [11].

Both injection timing and injection rate are optimum at a specific location. Injector pumps and nozzles can be designed for the requirement of minimum fuel consumption while maximizing brake power Fuel injection parameters if not designed for fuel economy result in over consumption even under normal service condition. Indicative in Fig. 6 is that at the same injection timing of 10 degrees before top centre, BTC, a higher injection rate of 4 mm.3/degree shows an improved bsfc of 10% compared to an injection rate of 3 mm.3/degree. Similarly, a gain of the order 3% is realized by improving injection timing and rate from 15° to 10° BTC and 3.5 to 4 mm^3/degree respectively.

Compression Ignition engines have low limits on minimum fuel-air ratio due to burning requirements within the spray envelope or in evaporated portion of the charge. The practical low limit is set by mechanical rather than combustion considerations. Poor spray characteristics at suficiently low fuel quantities may set low limit due to misfiring. In general this low limit is set by fuel quantity required to overcome friction during idling. The practical high limit on fuel-air ratio is set by smoke and deposits. Smoke-free combustion is seldom obtained above fuel/air = 0.8, and most diesel engines are rated below this point for continuous operation.

Despite efforts made to produce a uniform mixture for SI engines, non-uniformity remain. For a given engine cycle, the fuel, air, EGR and residual gas are not completely mixed, and the composition non-uniformity across the charge may be significant. These produce variations in composition at the spark plug location (the critical region since the early stages of flame development influence the rest of the combustion process) which can be of order ±10% peak-to peak. In addition, in multi-cylinder engines, the average air, fuel, and EGR flow rates to each cylinder are not identical. Typical cylinder-to-cylinder variations have standard deviations of ±5% of the mean for air flow rate and fuel flow rate (giving a ±7% variation in the air/fuel ratio) for steady-state engine operation[10]. At unsteady engine operating conditions these variations are reported to be higher

Spark ignition engines attain maximum power at excess air ratio ranging from 0.9 to 1.0, minimum fuel consumption is attainable at 10% excess air ($\lambda \approx 1.1$). In practice, ignitable mixtures are kept at λ between 0.7 and 1.25. The rich mixture are for starting to overcome the well known efficiencies. On the other hand a comparatively richer mixture (full load and Idle enrichment) is required at full load and idle as well as in the lowermost part of the part-load range. A lean mixture is desirable in the middle of the part-load range (normal driving) for fuel consumption reasons.

Fuel metering systems in both SI and CI engines have been primarily designed to produce specific engine characteristics. It is important that fuel consumption reasons be among criterion in such designs. In as far as bsfc of engines varies with mixture composition, fuel metering systems should be maintained at their optimal designs by taking appropriate measures, especially servicing. Any deviations introduced in the fuel systems increases fuel consumption.

Effects of exhaust gas recycle, EGR.

Exhaust Gas Recycle (EGR) can control NO_x emissions in SI engine. At part load, EGR act as an additional diluent in the unburned gas mixture, thereby reducing the peak burned gas temperature and NO formation rates. The absolute temperature reached after combustion varies inversely with burned gas mass fraction. The contents of residual gas is influenced by load and valve timing in particular the extent of valve overlap.To a lesser degree it is also affected by the air/fuel ratio and compression ratio.The effect of exhaust gas recycle on engine performance and eficiency for mixtures with equivalent ratios,$\phi \leq 1.0$, is similar to the addition of excess air. Both EGR and excess air dilute the unburned mixture. At constant burn duration, bsfc and exhaust temperatures decreases with increasing EGR. Fuel consumption improves with increasing EGR as there is a reduced pumping work as EGR is increased at constant brake load. Fuel and air flow remain almost constant, This is thus increasing intake pressure. In addition, reduced heat loss to the walls as the burned gas temperature decrease. Furthermore, as a consequence of EGR dilution, a reduction in the degree of dissociation is favoured. This allows for more of the fuel's chemical energy to be converted to sensible energy near TDC. The effects of EGR on reduced pumping work and heat loss to walls is comparable whilst that due to dissociation is 50% of either of the two.

Save for the omission of experimental values, Fig. 7 shows experimental bsfc versus EGR data for two combustion chambers: a combustion chamber with a moderate burning rate and a faster-burning chamber with open geometry and induction generated swirl. Addition of EGR lengthens the flame development and propagation process. However the faster-burning chamber shows a pattern of significant bsfc reduction until, at about 20% EGR, when the combustion quality deteriorates. For slower-burning combustion chamber, the tolerance to dilution with EGR is much less. The delayed combustion chemistry in slower burning chambers affects turbulence and allows more heat loss to walls.

Effects of gas exchange process

Volumetric efficiency is an indication of the effectiveness of the gas exchange process. This is required to admit fresh charge to the cylinder and expel much of the exhaust gases. This process influence the performance characteristics of the engine. Well designed natural aspirated engines have volumetric efficiency over 90%, and over 100% efficiency for tuned induction systems. Several ways to improve the gas exchange process is possible as in supercharged and turbocharged engines. The output of an engine is related to its Volumetric efficiency by:

$$\overline{P_b} = \eta_v . \eta_b . \rho . H_u / AFR \qquad (7)$$

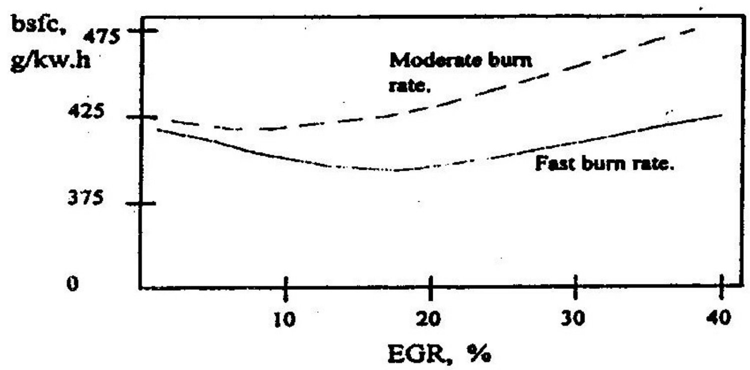

Fig. 7: Brake specific fuel consumption at MBT spark advance as a function of percent recycled exhaust, for four-cylinder spark-ignition engine with a moderate burn rate combustion chamber and a fast burn rate combustion chamber: 1400 rev./min, 324 kPa bmep, equivalence ratio 1.0 [12]

Admission and exhaust processes are related to one another and occur more or less simultaneously. The drop of a pressure in the intake system, ΔPa, depends on the speed of the engine, the hydraulic resistance of the flow system, the cross-section areas through which the fresh chage moves and the density of charge. ΔPa is directly proportional to the square of the speed n^2 and inversely proportional to square of area of the intake system, a_{is}^2. In modern four-stroke engines with overhead valves, the possibility of increasing the area is limited by the cylinder head. The total area of the cross-section of the inlet valves can be increased by using four valves (two inlet and two exhaust valves). The volumetric efficiency for the same capacity of SI engines is lower than that of diesel engines. This is due to flow losses in the carburettor and throttle, intake manifold heating, the presence of fuel vapour, and a higher residual gas fraction associated with the SI intake system.

For economy fuel consumption, it is a practice to extend the valve timing phases for opening the valve wide during the period of admission and exhaust so as to make the best use of the inertia of the gases in the admission and exhaust systems in order to improve scavenging and chaging of the cylinders. The process of exhaust begins 40 to 60° before Bottom Dead Centre, BDC. Thereafter to TDC the exhaust gas discharge freely owing to the difference between the cylinder pressure and the outlet.The cylinder is then scavenged by the piston which expels the gases as it moves towards TDC. The exhaust valve closes 15 to 30° afterTDC, as a result, both valves remain open for a certain time.As the spent gases flow through the exhaust valve, their ejecting action induces a rarefaction in the zone of the cylinder under the inlet valve. Hence, when both valves are open, a fresh chage is admitted into the cylinder while the exhaust gases are being expelled.

Diesel engines are more frequently turbocharged. These engines have higher maximum torque and power than equivalent natural aspirated engines. Compared to naturally aspirated engines the fresh charge enters the cylinder at higher temperatures and pressures. With a high supercharging ratio of more than 2.0, a cooler to reduce the temperature is installed in the manifold behind the compressor before being admitted to the engine cylinder

The output of naturally aspirated Diesel engines is limited by the maximum tolerable smoke emission levels, which occur at overall equivalence ratio values of about 0.7 to 0.8. Turbocharged Diesel engine output is usually constrained by stress levels in critical mechanical components. As boost pressure is raised, maximum pressures and thermal loading will increase. In practice, the compression ratio is often reduced and the maximum fuel/ air equivalence ratio and thus fuel is reduced in turbocharged engines (relative to naturally aspirated engines) to maintain peak pressures and acceptable thermal loading. The fuel flow rate increases at a much lower rate than the air flow rate as boost pressure is increased as it encounters an increased load.

Small high-speed high-swirl turbocharged DI diesel engines (e.g. suitable for automobile or light-truck applications) have similar performance maps to those of equivalent IDI engines. Maximum bmep values are closely comparable. Usually slightly higher boost is required to ofset the lower volumetric efficiency of the high-swirl-generating port and valve of the DI engine. Best bsfc values for DI engine are usually about 15% lower than of comparable IDI engines.

Load and speed

Part load behaviour of engines are analyzed using performance maps, Fig. 8 and 9. Maximum bmep contours occurs in mid-speed range. Minimum bsfc island is located at slightly lower speed and at part-load. At very low speeds fuel-air mixture combustion quality deteriorates and dilution with exhaust gas becomes remarkable. On the other hand, high speeds increases bsfc. Here, the already good fuel conversion eficiency is outweighed by friction losses which increase almost linearly with increasing speed. Other contributing factors are variations in volumetric efficiency, η_v, and the marginally increase in indicated fuel conversion efficiency, η_f. η_f increases slowly due to the decreasing importance of heat transfer per cycle with increasing speed.

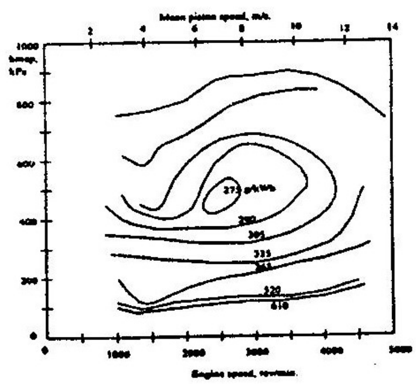

Fig. 8: Performance map for 2 dm^3 four-cylinder fast-burn spark-ignition engine showing contours of constant bsfc in grams per kilowatt-hour [13].

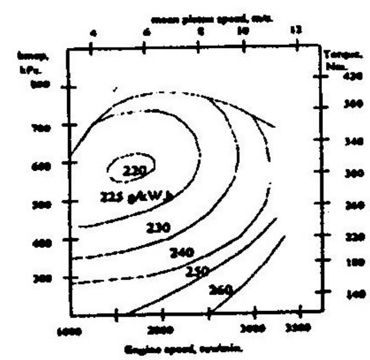

Fig. 9: Performance map for 6.54 dm^3 8-cylinder air cooled naturally aspirated medium--swirl DI diesel engine. Contours of constant bsfc in g/kWh. shown. Bore = 102 mm, stroke = 100 mm, compression ratio = 18. Multi-hole fuel nozzle [14].

Comparisons between naturally aspirated DI and IDI diesel engines of closely comparable design and size indicate that the DI engine is always more efficient, though the benefit varies with load. At full load, differences of up to 20% bsfc have been noted. At part load, the gain is less of the order 10%. The full load difference is due in large part to the retarded timing of the IDI combustion process and its long, late-burning, heat-release profile. At light load, about 300 kPa bmep (A/F = 50) these combustion effects are small and the indicated efficiency penalty of the IDI (of about 5 to 7%) is due to the higher heat losses associated with the lager surface area and high-velocity flow through the connecting nozzle of the divided-chamber geometry and due to the pumping pressure loss between the main and auxiliary chambers. The percentage improvement for various factors in DI versus IDI is indicated in fig. 10.

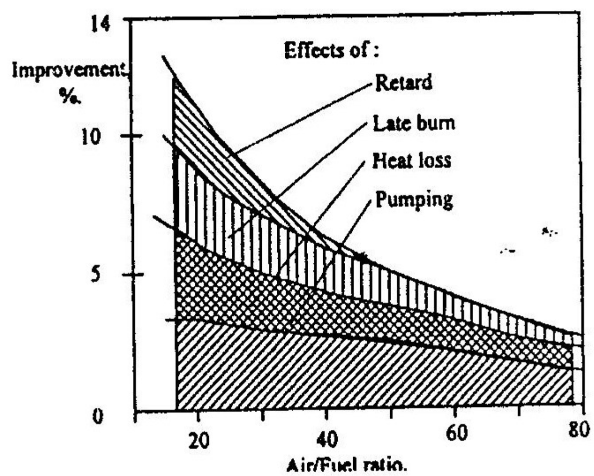

Fig. 10: Factors which improve the indicated efficiency of naturally aspirated small DI diesel combustion systems relative to IDI swirl-chamber combustion system, as a function of A/F or load [15]

Electronic and electromechanical controls

The contemporary automotive technology is making use of electronics to optimize even the once complicated timings. A combination of instantaneous variables like speed, temperature, load and toxicity can be the inputs to a single control. The centrifugal and vacuum spark advance systems have been replaced by semi or fully electronic ignition systems. As such, each ignition point is selected taking into account conditions such as exhaust emissions, knocking threshold, speed and driveability Additional variables such as engine temperature or the overrun and full load information are correction value inputs. A voltage correction ensures the availability of the highest possible voltage with minimum heating of the control unit and ignition coil. The microcomputer in the electronic trigger box uses the input variables to calculate the precise timing angle by means of a stored ignition map. The entire system is maintenance free and requires no adjustment for life contrary to the centrifugal and vacuum

advance systems.
Both CI and SI engines uses electronics in fuel metering systems. In diesel engines electronic governors have replaced mechanical units in fuel injection pumps. The imbedded microprocessor uses the accelerator position, engine speed and correction variables in conjunction with the maps stored in the memory to calculate desired quantity or the desired position of the injector pump's control rod. Electronically controlled carburettors have several controls in addition to the basic carburettor Output values of the processor control the choke and throttle valve actuators. Idle and acceleration enrichment are quickly and precisely provided through brief choke closure. During overrun, the supply of fuel is cut-of and idle system is interrupted. Similarly the supply of fuel is cut-of by closing the throttle valve when the engine is turned off. The idle speed control system holds the engine speed constant and significantly reduced. These electronic fuel metering systems results in enhanced performance and reduced fuel consumption and toxicity levels.

On the other hand, the electromechanical systems are included for comfort and safety, navigation and driver information, transmission control and suspension. In pneumatic and hydro-pneumatic suspension, the vehicle body can be maintained at the same height irrespective of the load. In many cases the height of the vehicle body can also be varied at will by the driver. Advantages of these suspension units includes reduction of fuel consumption through speed dependent control vehicle tilt and vehicle body by improved aerodynamics, increase in vehicle body level when travelling on uneven road surfaces and improved cornering stability through absolute transverse shut-off of the pneumatic springs.

Automatic-transmission control is increasingly employing combined electronic/hydraulic systems. Hydraulic systems are solely used to provide the power drive for the clutches while the choice of gear and the adjustment of the pressures to the torque to be transmitted are performed by the electronics. By means of a shift point control, a driver can choose between various driving programs, as optimum fuel economy or maximum power It is also possible to keep the transmission in low gear for special operating conditions as in towing a trailer

CONCLUSIONS

(i) Optimum vehicle designs tend to minimize drag coefficient through improved vehicle aerodynamics, low vehicle height and reduced rolling resistance. "Longer" transmission ratios having overdrive units favours fuel economy.

(ii) Compact Fast-burn combustion chambers for use in SI engines are more efficient and shows improvement in fuel consumption. The trend to CI engines is towards open-chamber for high-automotive speeds close to bsfc point thus at desired high brake thermal efficiency and economic fuel use. However, advantages of divided chambers is vested in improved emission control. In both cases higher compression ratios are desirable.

(iii) Minimum fuel consumption rate of automotive engines coincides with mid-speeds and part load. To maintain this, obstacles leading to too low speeds be minimized and drivers shouldn't drive at higher speed on clear roads. Road conditions must be maintained for vehicle passability. They should be designed so as to minimize traffic congestion and to attain less roughness. Urban authority have the duty to incorporate the minimum travel distance concept in urbanization programs. Multicentric Cities and well arranged commuter roads allows people to travel shorter distances.

(iv) Scheduled and unscheduled maintenance should be carried out properly by appropriate personnel to keep the engine in its optimum operating conditions. This saves time and avail a vehicle at its best operating conditions at required time.

(v) Electronic computerized technology introduced in automotive engines enhances performance and reduce fuel consumption.

NOMENCLATURE

A = front surface, m^2
a = acceleration, m/s^2
AFR = air/fuel gravimetric ratio
B = fuel consumption, kg/h

B_e	= fuel consumption over a specific distance, g/m
B_r	= braking resistance, N
be	= specific fuel consumption, g/kWh
c_w	= drag coefficient
f	= Coefficient of rolling resistance
g	= acceleration due to gravity, m/s^2
H_u	= calorific value
m	= vehicle mass, kg
P_{eff}	= effective/Brake power, kW
P_b	= brake power.
t	= time, s
t_B	= elapsed time for measured volume consumption, s
v	= road speed, m/s
V_B	= measurable fuel volume on test bench, cm^3

Greek letters

α	=	the load ratio; the ratio of the volume at the end of heat addition to that at the end of expansion process or angle of incline, [^{o}]
β	=	volume ratio at isobar processes
r_v	=	the compression ratio
γ	=	the gas adiabatic coefficient.
σ	=	volume ratio at isotherm processes
η_u	=	transmission efficiency of the power train,
ρ_B	=	fuel density, kg/m^3
η_b	=	brake efficiency,

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