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# DESIGN AND DEVELOPMENT OF DOUBLE HELIX FUEL INJECTION PUMP FOR FOUR STROKE V-16 RAIL TRACTION DIESEL ENGINE

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## ABSTRACT

The diesel fuel-injection system of ALCO DLW 251 engine consists of single cylinder injection pumps, delivery pipes, and fuel injector nozzles. Fuel injection into the combustion chamber through multi-hole nozzles delivers designed power and fuel efficiency. The two most important variables in a fuel injection system of a diesel engine are the injection pressure and injection timing. Proper timing of the injection process is essential for satisfactory diesel engine operation and performance. Injection timing needs to be optimised for an engine based on requirements of power, fuel economy, mechanical and thermal loading limitations, smoke and emissions etc. Since each of these requirements varies with the operating conditions, sometimes contrary to the requirements of the other parameters, the map of optimised injection timing can be very complex.

The ALCO DLW 251 engine's fuel injection pump is jerk type to permit accurate metering and timing of the fuel injected. The pump has a ported barrel and constant-stroke plunger incorporating a bottom helix for fuel delivery control with constant injection timing. From the point of view of good power and fuel economy, combustion should take place so that the peak firing pressure occurs at about 10-15° after TDC and is usually a few degrees after combustion starts. For this to happen, fuel should be injected at an appropriate time, depending on Injection delay and Ignition delay. Both these factors are dependent on the speed and load. Changing the operating point of the engine may change either one or both types of delay, altering the moment of start of combustion.

Various researchers have shown that both the Injection and the Ignition delay are reduced as the engine speed is decreased resulting in advancement of injection timing at lower speeds (and loads). This condition will be corrected by varying the static injection timing, which can be achieved by providing a modified helix on the plunger to delay the start of fuel injection, for the lower speeds and loads.

A new double helix (upper and lower helix) fuel injection pump for the ALCO DLW 251 16 V engine has been designed. The new fuel injection pump has been tested on the engine test cell at Research Designs & Standards Organisation and has shown an improvement of 1.2% in locomotive duty cycle fuel consumption. This paper describes the design & development of double helix fuel injection pump and discusses the engine tests completed to verify the projected improvements in fuel efficiency.

 

## INTRODUCTION

The diesel fuel-injection system of ALCO DLW 251 engine consists of single cylinder injection pumps, delivery pipes, and fuel injector nozzles. Fuel injected into the combustion chamber through multi-hole nozzles provides designed power and fuel efficiency. The two most important variables in a fuel injection system of a diesel engine are the injection pressure and timing. Proper timing of the injection process is essential for satisfactory diesel engine operation and performance.

Injection timing needs to be optimised for an engine based on requirements of power, fuel economy, mechanical and thermal loading limitations, smoke and emissions etc. Since each of these requirements varies with the operating conditions,sometimes running contrary to the requirements of other parameters, the map of optimised variable injection timing can be very complex. For example it is possible to achieve good fuel economy by suitable advancement of fuel injection timing; however this can have an adverse impact on NOx emissions.

Similarly reduction of NOx emissions requires the fuel injection timing to be retarded with consequent increase in the particulate emissions. High firing pressures and temperatures are required for proper combustion and lower brake specific fuel consumption (bsfc) but are detrimental for the reliability of the engine and require robust engine structural design.

Literature indicates [1] that from the point of view of good power and fuel economy, combustion should take place so that the peak firing pressure occurs at about 10-15° after TDC, this usually occurs a few degrees after combustion starts. For this to happen, fuel should be injected at an appropriate time, depending on the following factors

a) Injection delay
b) Ignition delay

While injection delay is primarily a function of engine speed, nozzle opening pressure and tubing length, ignition delay depends on the temperature and pressure in the cylinder, droplet size and velocity, mixing characteristics, initial droplet temperature etc.

### Injection delay

The governing equation for calculating the injection delay (implies the period between spill port closure and the start of injection) is [2]

$$\theta_{injd} = (6N * L)/ V_o \text{ °CA where]}$$

N is the rotational speed of the engine, rev/min
L is the length of high pressure tubing, m
$V_o$ is the velocity of the pressure wave in high pressure tubing, m/s

This equation is clearly speed dependent, and would be the same for all speeds if expressed in time units. The equation assumes constant injection pressure and is independent of the plunger diameter.

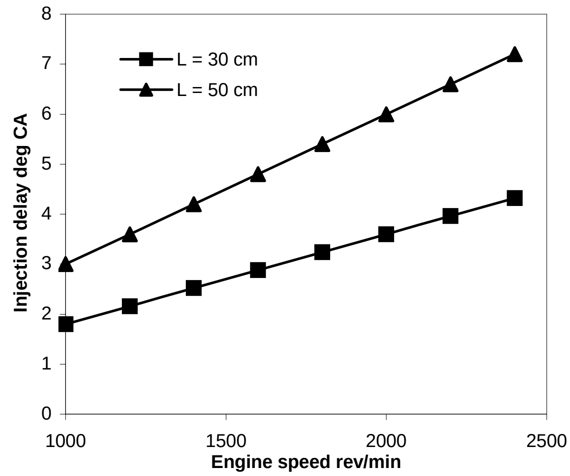


Figure 1: Effect of engine speed & length of tubing on injection delay

Figure 1 shows the theoretical values of delay for different tubing lengths. But in actual practice, the function lines are neither straight nor as evenly spaced, for the following reasons: -

- a finite nozzle lifting time, independent of tubing length
- residual pressures which vary for different speeds
- retraction action of the delivery valve

However the relative values are of the same order. Figure 1 amply demonstrates the need for short, equal length delivery tubes and indicates the order of compensation, with speed, required to be made while arriving at the actual dynamic timing.

### Ignition delay

The governing relationship for calculation of ignition delay, as proposed by different researchers is of the general form: - [2]

$$\theta_{igd} = 6N*(A/p^B)*e^{C/T} \text{ °CA, where}$$

N – engine speed, rev/min
p – mean pressure in the cylinder between injection and ignition
T – mean temperature in the cylinder between injection and ignition
A,B - constants

A typical correlation proposed by Wolfer is [2]

$$\theta_{igd} = 6N*(0.429/p^{1.19})*e^{4650/T} \text{ °CA, where}$$

N – engine speed, rev/min
p – mean pressure in the cylinder between injection and ignition
T – mean temperature in the cylinder between injection and ignition

Another relation due to Shipinski [2] takes into account the cetane number of the fuel in addition to the temperature and pressure in cylinder when ignition takes place

$$\theta_{igd} = 6N*(0.0097/p0.386)*(40/CN)*0.69*e^{4644/T} \text{ °CA, where}$$

N – engine speed, rev/min
p – mean pressure in the cylinder between injection and ignition
T – mean temperature in the cylinder between injection and Ignition
CN – Cetane number

Although these correlations take into account only pressure and temperature in the cylinder, other factors like number of spray holes, diameter of spray holes, Air fuel ratio, heat transfer rate from walls (dependent also on speed), swirl etc. also affect the delay.

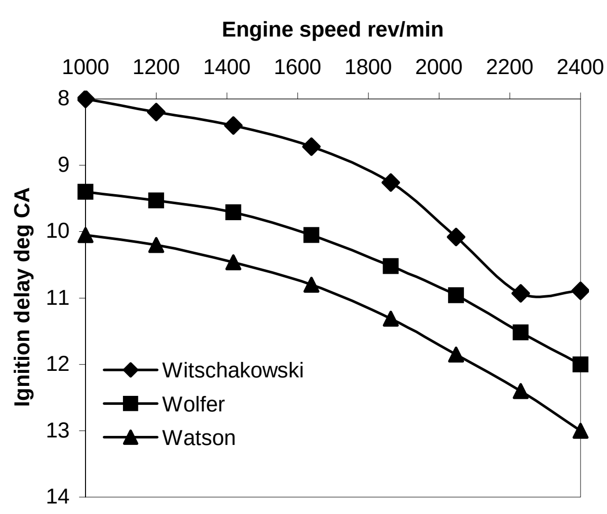


Figure 2: Ignition delay using different correlations – ignition at 4° BTDC

It clearly shows that the correlations have limited applicability, as each correlation is true only for a certain set of injection system and combustion chamber design. This would not be good enough for determining the map of optimum timing for a certain engine. However figure 2 shows that ignition delay increases with speed of the engine. Thus at lower speeds the start of combustion (SOC) shall be early as compared to higher speeds. Since for the maximum brake torque (MBT) we need half of the pressure rise to be at top dead centre (TDC) and the balance after TDC, at lower speeds we need to inject fuel closer to TDC to get the desired pressure rise characteristics. This condition translates into the need to retard the start of injection (SOI) as the engine speed decreases.

**Fuel Injection system of ALCO engines**

The existing fuel injection system of ALCO engine consist of three main components, i.e. the fuel injection pump, the high pressure tubing connecting the fuel injection pump to the nozzle and the fuel injection nozzle. The fuel injection pump is mounted on the fuel pump support which is mounted on the side of the engine crankcase. The pump is actuated by the fuel cam lobe of the camshaft through a lever arm and roller. The ALCO fuel injection pump is a jerk type plunger pump to permit accurate metering and timing of the fuel injected. The pump has a ported barrel and constant-stroke plunger incorporating bottom helix for fuel delivery control. The pump consists primarily of a housing, delivery valve and spring, delivery valve holder, element(plunger and barrel assembly), plunger spring, a geared control sleeve and control rack(rod) assembly. The pump element comprises a barrel and plunger, which are matched, assembled to a very close tolerance. The fuel injection pump has three functions:[4]

· To raise the fuel supply pressure to a value which will efficiently atomise the fuel.
· To supply the correct quantity of fuel to the injection nozzle commensurate with the power and speed requirements of the engine.
· To accurately time the delivery of the fuel for efficient and economical operation of the engine.

The fuel injection pump has a plunger diameter of 17 mm with a bottom helix for proper fuel metering. The pump is capable of producing fuel injection pressures up to 1000 bar. The high-pressure tubing is made of special alloyed steel and its internal diameter is shot peened to provide compressive stress. The tubing is capable of handling the required fuel injection pressures. The injector is fitted into the cylinder head and consists of a body, the nozzle holder and nozzle. The nozzle is a low sac design with nine fuel injection holes. The fuel is injected into a quiescent combustion chamber; therefore the penetration of the injected spray is largely dependent on the injection characteristics of the injector nozzle and the pump injection pressure.

**Determination of a theoretical 2-dimensional map of desired timing**

A change in speed or load (compression temperature and pressure) may change either injection or ignition delay or both, altering the moment of start of combustion. This condition would need to be corrected by varying the static injection timing. A two dimensional bsfc map was produced for the ALCO DLW 251 16-cylinder engine by using the electronic fuel injection system. At every notch the injection timing was varied to find out the most optimum injection timing for the lowest bsfc. This is shown in figure 3.

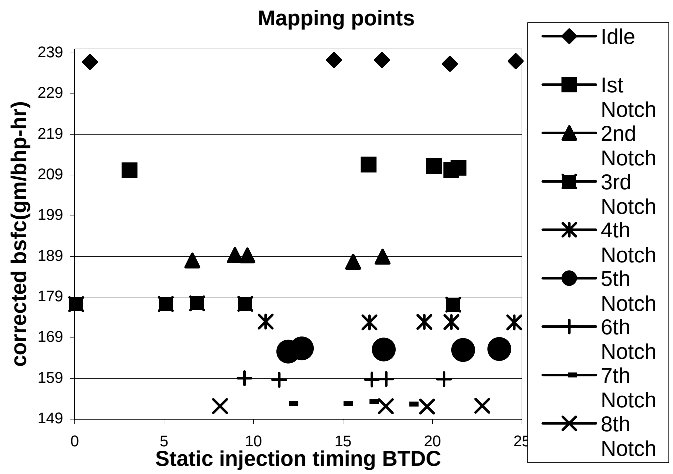


Figure 3: Mapping points with the Electronic Fuel Injection system

retarded as the speed decreases. Based on above results, optimum injection timings for each notch setting have been found out and the trend line added to these values as shown in figure 5. This shows a trend of retardation of injection closer to the TDC as the engine speed and load is decreased.

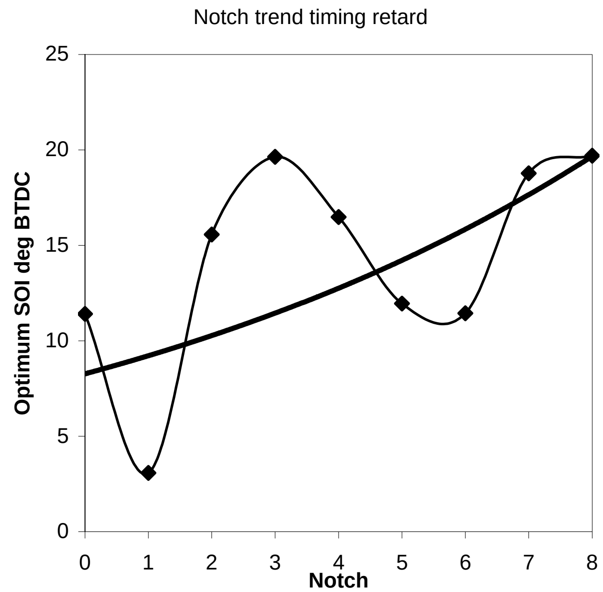


Figure 5: Trend line of the optimum Start of Injection notch-wise

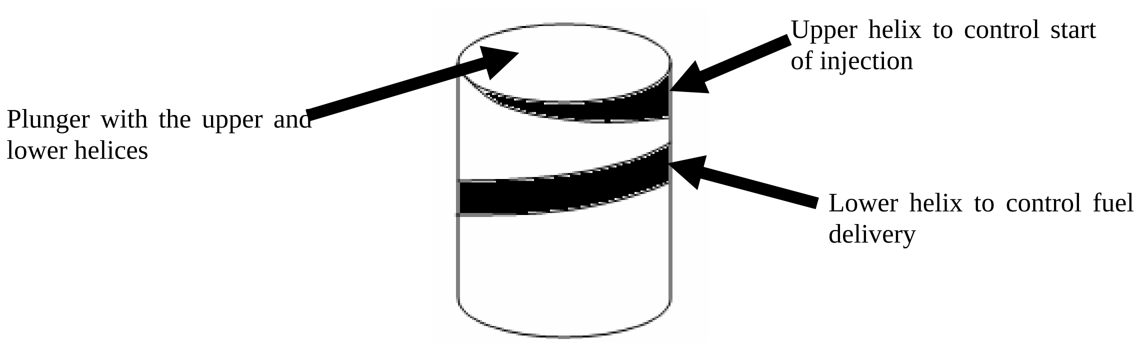


Figure 6: Concept of double helix plunger

To find the optimum injection timing, fuel consumption and sfc measurements were done at various injection timings. Figure 3 is a two dimensional map of the engine sfc vis-à-vis the injection timings shows the spread of the readings. Notch optimisations of injection timings are presented in figure 4 as shown in Annexure ‘A’.

It can be observed from figure 4 that the optimum injection timings for each notch (different engine speed and load) can vary. In general the optimum injection timing needs to be from this experiment is that an advance / retard of 13-14 deg CA is required over the full working range of the ALCO 251 16 cylinder engine. A change in design of the plunger timing helix is required to accomplish this retardation with engine speed.

## DOUBLE HELIX CONCEPT

The ALCO 251 engine fuel injection pump is a single acting, constant stroke and plunger type with the effective working stroke being adjustable. The barrel and plunger element of the fuel injection pump are shown in figure 7.

Figure 7: Barrel and plunger of Double Helix Fuel Injection pump, double helix and single helix plungers

The upper section of the plunger has been provided with just one helical cut to obtain the injection timing and delivery as required. The ALCO 251engine fuel injection pump element has a right hand single helix. The barrel has delivery ports spaced 180° apart and the delivery period is during the time the ports remain covered by the plunger element. The fuel injection stops as soon as the ports come in contact with the bottom helix.

On the current pump, the fuel injection starts at the same CA irrespective of the engine load and speed. To change the dynamic injection timing it is somehow required to delay the covering of the ports for different time intervals at different

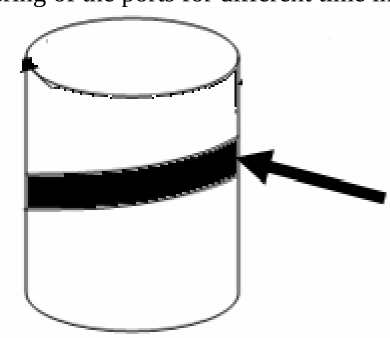


Figure 8 : Concept of single helix plunger

speeds. Literature has shown that provision of an upper helix has been successfully used for obtaining such injection delays [3]. Therefore it was planned to incorporate another helix at the top of the plunger element for variable injection timing whereas the bottom helix was to ensure the required fuel delivery. The concept is illustrated in figure 7.

As the plunger moves up to complete the stroke, depending on the rack position (or the engine notch) the barrel ports remain uncovered initially for varying time intervals. Thus at lower notches, the delivery starts closer to TDC as compared to the higher engine notches and the desired SOI retardation with decrease in engine speed and load is obtained. Since the plunger element now has two helices, at the top and bottom, the fuel injection pump incorporating the same is called a double helix pump. Design of this element would take inputs from the fuel-mapping diagram as shown in figures 4 and 5.

Wartsila has used similar concept on its Model 64 engine, the largest engine type in its series of medium-speed engines. The engine has a twin-plunger concept of the injection pump, where one plunger helix sets the start of injection and the other meters the amount of injected fuel.[5]

**Design of the double helix plunger element**

As shown in figure 5, a retardation of 14 CA° in fuel injection is required at the Idle engine notch, as compared to the engine full load and speed. The position of the helix relative to the delivery ports is changed by rotation of the plunger element, which in turn is controlled by the control rod assembly. The control rod assembly position at idle engine notch is 9 mm and at 8th engine notch is 30 mm, thus the control rod assembly moves a distance of 21 mm over the full working range of the engine. Considering a proportional fuel injection retard at intermediate engine notches, a retard of 0.7 CA° /mm of the control rack is required to achieve the optimum injection timings.

The camshaft fuel lobe is responsible for providing lift to the fuel injection plunger. Figure 9 shows the fuel injection pump plunger lift plotted against the crank angle for a conventional camshaft. It can be seen that the lift of the plunger starts at 125° and the plunger reaches the centre of the dwell period at TDC at 246.5°. Shown in figure 10 below is the fuel cam lobe and its radius of curvature at different angles reckoned from the dwell centre line. Thus 246.5° of CA correspond to the centre line of the dwell portion of the fuel cam lobe, which is taken as 0° cam angle for the fuel cam lobe.

 

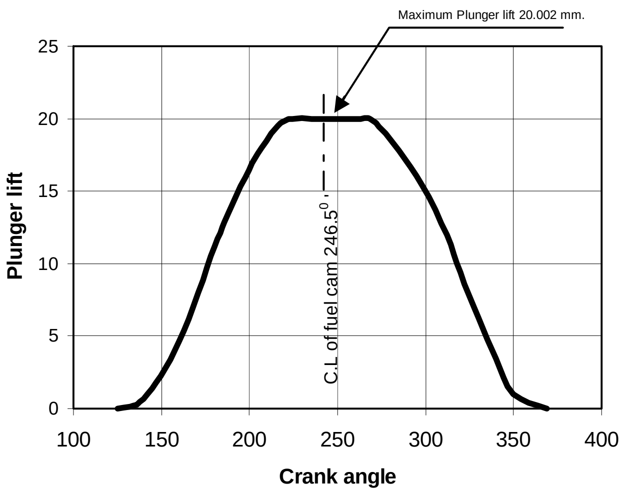


Figure 9: Lift diagram for the Fuel Cam

Based on the lift diagram of the fuel cam the period for lifting of the fuel injection pump plunger is calculated as 60.75°. Thus the lift of the plunger starts at 60.75° cam angle of the fuel cam lobe.

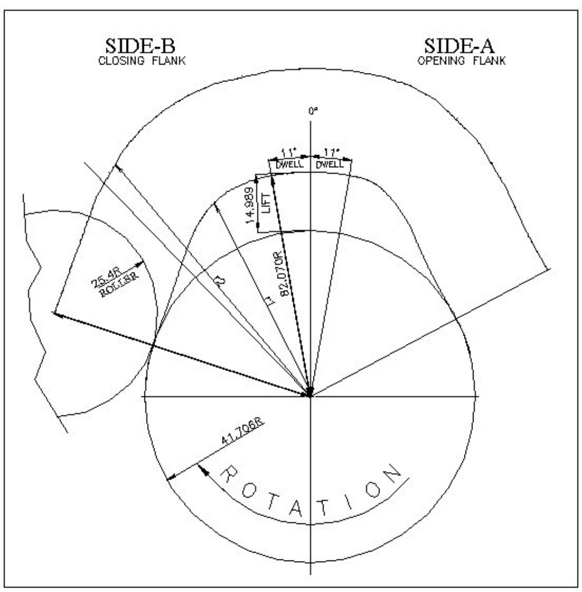


Figure 10: Details of the Fuel cam lobe

In a single helix element the fuel injection starts from 60.75° fuel cam lobe angle irrespective of the load and speed. In variable injection timing we need to retard the fuel injection at idle by 7° fuel cam (or 14° CA). Thus the fuel injection should start only at 53.75° fuel cam lobe angle. Taking into account the average radius of curvature of the fuel cam lobe at 60.75° and 53.75° the empty movement of the plunger at idle engine notch to provide the required injection retard of 14° CA works out to be 4 mm. Similarly the empty movement of the plunger at another intermediate notch is calculated and the helix is designed as explained below.

**Plunger helices**

After calculation of the cam lift the construction of the plunger helices is taken up. For convenience in the graphical procedure, the plunger position can be considered fixed and barrel ports moved relative to it.

The first step was to calculate the port positions. The plunger diameter was laid out in developed position and the upper helix line is drawn based on the empty plunger movement at different engine notches. The bottom helix line is drawn based on the fuel delivery required at different engine notches. The pitches of the two helices are then calculated geometrically. The arrangement of the fuel rack, the control sleeve and the fuel injection pump plunger is shown in figure 11.

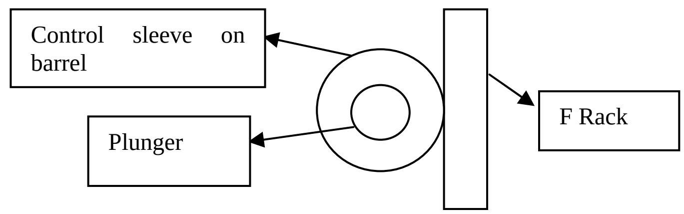


Figure 11: Arrangement of the fuel rack with plunger, barrel & control sleeve

The port closing depends on the port size, the engine rpm and the ratio of the plunger stroke to the plunger lift required for port closing. The effective stroke of the plunger for fuel delivery is based on the fuel delivery requirements at each engine notch. After the port positions are marked on the plunger development diagram, tangents are drawn to the circles (port) at the top and the bottom position. These give the top and the bottom helix lines. Finally pitch of the top and bottom helix is calculated. The double helix geometry is laid out as shown in figure 12.

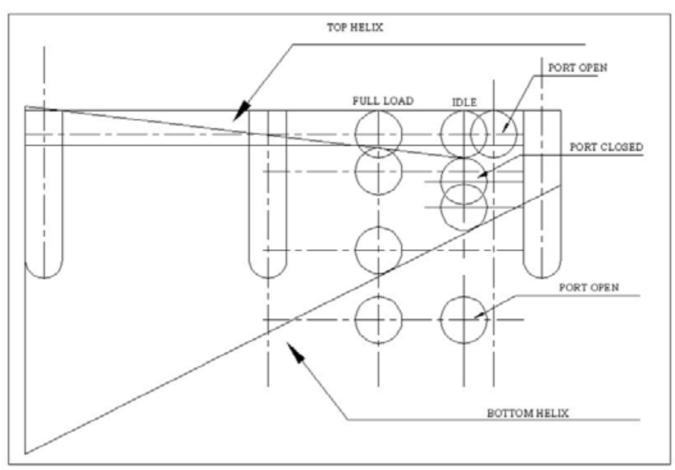


Figure 12: Layout of the plunger helix

**Design verification tests**

After the design & development of the double helix fuel injection pump, it was planned to carry out design verification tests to find out whether the design objectives of reduction in fuel consumption have been met or not. One engine set of double helix pump were manufactured at MICO Bosch India Ltd. and delivered to the Engine test laboratory at the Research Designs & Standards Organisation under the Ministry of Railways.

Engine configuration as given in table 1 was used for evaluating the performance of double helix pumps.

Table 1: Engine configuration for testing

| | |
|---|---|
| 1 | 16 Cylinder ALCO 251 engine |
| 2 | Bore 228.6 mm, Stroke 254 mm |
| 3 | Displacement – 10.4 liter per cylinder |
| 4 | Rated power – 2312 kW |
| 5 | Rated speed – 1050 rpm |
| 6 | Large after cooler (16 row) |
| 7 | Modified water piping |
| 8 | Stiffer unit cam shaft |
| 9 | 17 mm fuel injection pumps, Double helix and Single helix |
| 10 | 11.75 CR deep bowl steel cap Piston with barrel type ring pack |
| 11 | 0.35 mm fuel injection hole with 157º spray angle |
| 12 | RR 606 multigrade engine oil of Indian Oil Corporation |
| 13 | 10º impeller water pump |
| 14 | Plate type Lube Oil Cooler 290 kW heat dissipation capacity |
| 15 | New design cylinder heads |

**Experimental load points**

Figure 13 shows the load points at which the engine was tested. In all nine load points of the locomotive operation were chosen with adequate thermal stabilisation at each point. The engine was first taken to the ninth load point and the measurements were done in notch reduction mode.

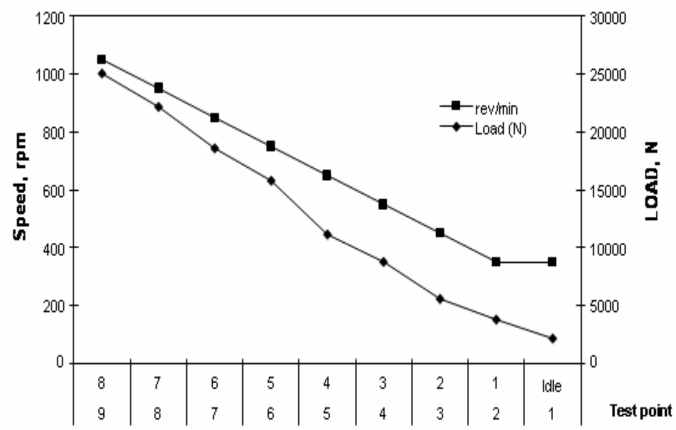


Figure 13: Load and speed test points

## DISCUSSION ON RESULTS

As a first step, it was desirous to find out the actual fuel injection timings with the double helix fuel injection pumps and compare with theoretical predictions of engine mapping in figure 4. Figure 14 shows the Start of Injection (SOI) obtained with single helix and double helix fuel injection pumps at different engine notches. It can be seen that a double helix pump with 22° static injection retard demonstrates conformance to the theoretical predictions based on engine mapping carried out with the electronic fuel injection system. SOI does not vary with engine notch (load and speed) when the single helix pump is used as anticipated. Thus 22° static injection timing (the flywheel timing mark) with double helix fuel injection pump is expected to show the lowest brake specific fuel consumption (bsfc).

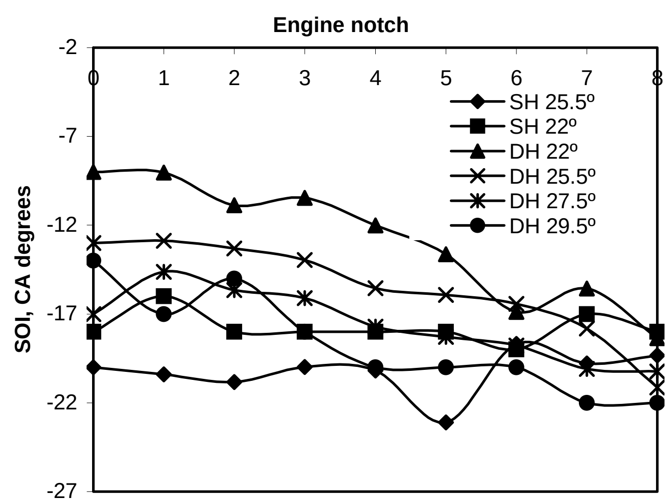


Figure 14: SOI timings obtained with different static injection timings

Optimum SOI timing shall reflect in lesser fuel being consumed for a lower engine notch as compared to the single helix fuel injection pump since horsepower produced at an engine notch is fixed. Thus a modification of the lower helix is also required to cater to lower fuel delivery requirements at lower notches. Accordingly this phenomenon should be reflected in the injected fuel quantity per stroke of the fuel injection pump. Figure 15 depicts the difference in fuel injection quantity with different combinations for four engine notches. The lowest fuel injection is seen with double helix pump with 22° BTDC static injection timing. It is therefore expected that this combination of the fuel injection pump and the static injection timing should result in the lowest brake specific fuel consumption.

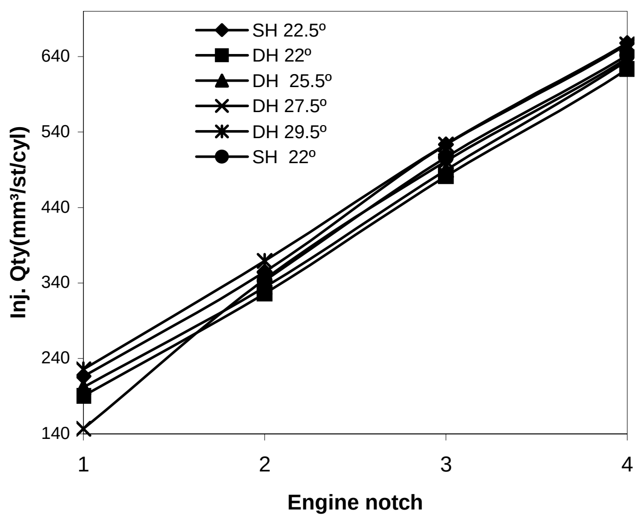


Figure 15: Fuel injection quantity with different static injection timings

Reduction in the brake specific fuel consumption (bsfc) was the primary design objective. Figure 16 shows the bsfc obtained at engine notches with single helix (SH) and double helix (DH) fuel injection pumps at different static fuel injection timings. It can be observed that the least bsfc is obtained with double helix fuel injection pump at 22° BTDC static fuel injection timing. The highest bsfc is seen with double helix pump at 29.5° BTDC static fuel injection timing at lower engine notches and with double helix pump at 27.5° BTDC static fuel injection timing at higher notches. The single helix pump with 25.5° BTDC static fuel injection timing exhibits the second highest bsfc at lower notches, an ample indication of non-optimisation of the single helix at lower notches. Single helix pump with 22° BTDC and 25.5° BTDC injection timings show higher bsfc at lower notches. Thus the engine with the double helix fuel injection pump is able to achieve the lowest bsfc at lower notches verifying the design objective. Plotting of brake specific fuel consumption against static engine timing for each engine notch would have further supported the design validation but could not be done due to time constraints.

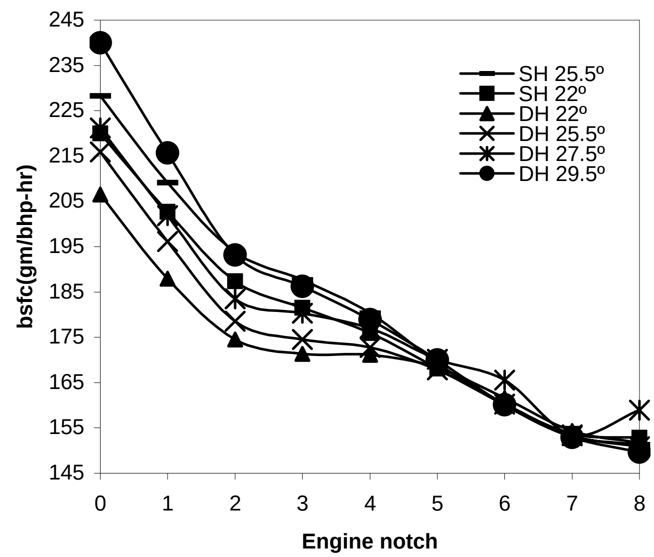


Figure 16: bsfc with different static injection timings

The next most important parameter to consider is the duty cycle fuel consumption of the ALCO 251 engine. Indian Railways have two separate duty cycles for the freight and the passenger train operation. The duty cycle fuel consumption with these two different duty cycles are presented in figures 17 and 18. It can be seen that the engine with the double helix fuel injection pumps have outperformed the one fitted with the single helix pumps. Choice of particular static injection timing shall depend on other factors like the firing pressures, turbine inlet temperatures etc. An improvement of about 1.2% fuel consumption over duty cycle is calculated with use of DH pump.

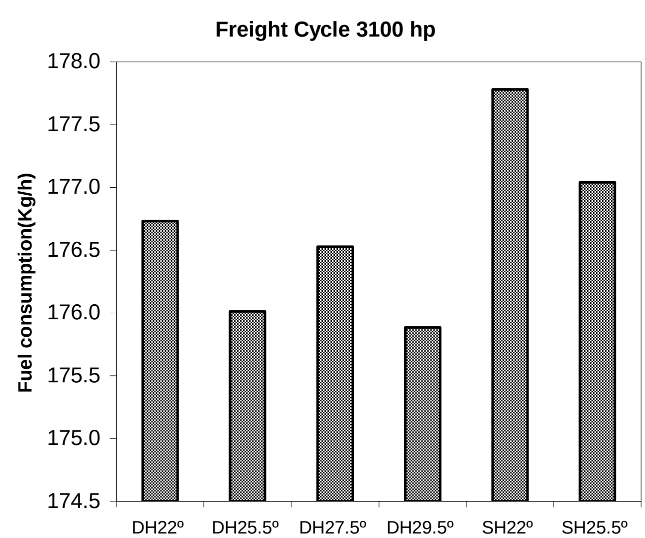


Figure 17: Comparison of Freight duty cycle fuel consumption

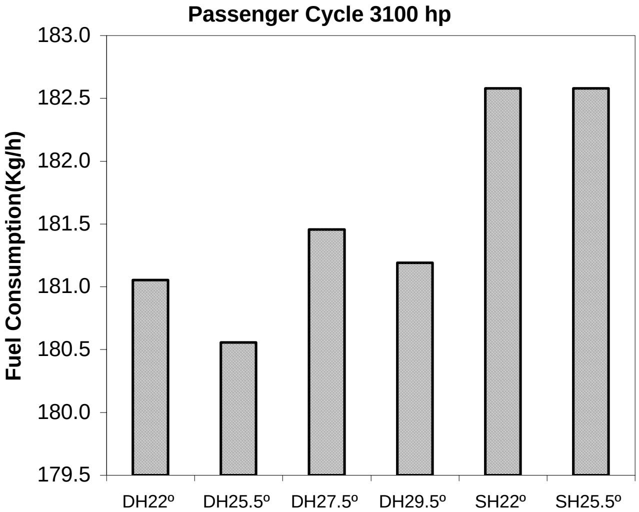


Figure 18: Comparison of Passenger duty cycle fuel consumption

The cylinder peak firing pressures and temperatures determine the mechanical and thermal loading of the engine. Changes in the SOI affect the start of combustion and the peak firing pressures and temperatures in the cylinder. The cylinder inline pressures and temperatures also affect reliability of engine. Figure 19 above shows the peak firing pressures obtained with different fuel injection pumps. With the double helix pumps and static injection timing of 22° BTDC, the second lowest peak firing pressures are obtained. The allowable peak firing pressures are 1800 psi; therefore the DH pump with static injection timing of 22° BTDC meets the criteria.

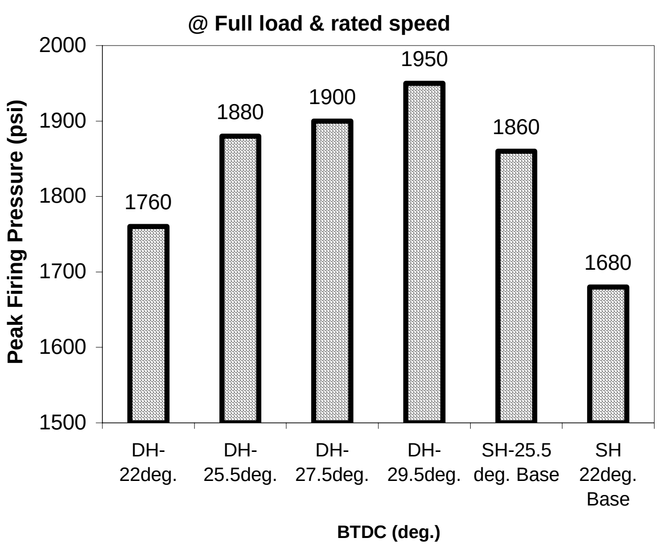


Figure 19: Firing pressures with different static injection timings at full load and speed

Turbine gas inlet temperatures are representative of the cylinder peak firing temperatures and combustion efficiency and also determine the efficiency of the turbocharger. In case of higher combustion efficiency, the amount of unburnt HC passing into & burning in the exhaust manifold is less leading to lower turbine gas inlet temperatures. These temperatures at different notches with different static injection timings are shown in figure 20. At higher notches the lowest temperatures are obtained with the 22° static injection timing with double helix fuel injection pump. This correlates well with the lowest brake specific fuel consumption (highest thermal efficiency) obtained with the DH 22° static injection timing. Another aspect which can be examined is the residual HC in the exhaust manifold which is generated at lower notches. With single helix fuel injection pump, at lower notches, because of non-optimised SOI, a significant amount of unburnt HC passes to the Exhaust manifold and resides in the manifold because of the constant pressure turbocharger exhaust manifold. At higher notches as the gas flow rate increases along with the peak firing temperatures, this HC burns leading to higher turbine gas inlet temperatures. At lower notches the lowest turbine gas inlet temperatures are obtained with 25.5° and 29.5° static injection timings and double helix fuel injection pumps. The temperatures show a drop after 5th engine notch with the DH pump and 22° static fuel injection timing. This aspect shall be discussed further during combustion analysis.

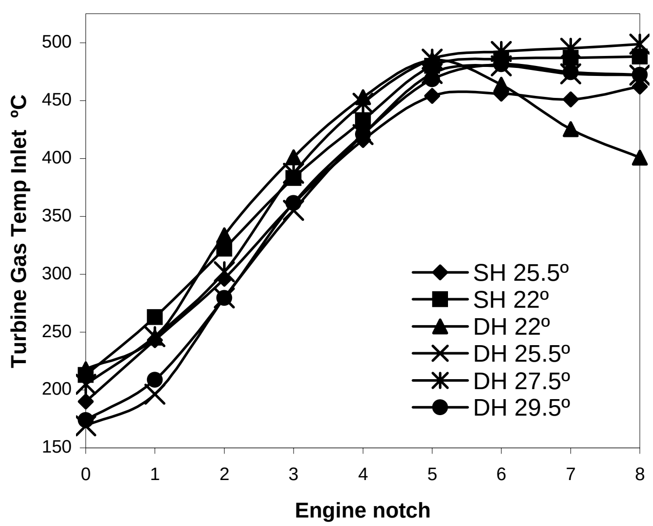


Figure 20: Turbine gas inlet temperature with different static injection timings

Although the engine was able to obtain the required horsepower with all the combinations of the fuel injection pumps and the static injection timings, the air boost pressures were also measured to find out any differences due to the different pumps and static injection timings. Figure 21 shows boost pressure obtained with different configurations at different engine notches. Type of fuel injection pump and static injection timing do not have any appreciable effect on the boost air pressure, although the double helix pump with 22° CA BTDC static injection timing shows the highest boost pressures.

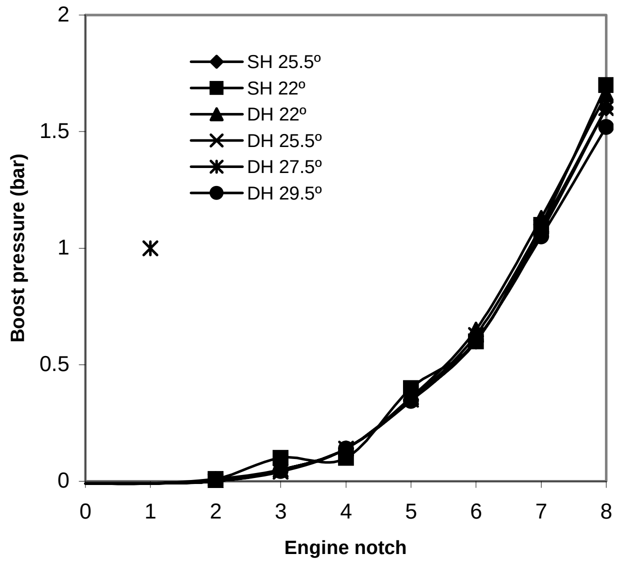


Figure 21: Boost pressure with different static injection timings

Figure 22 & 23 bring out the difference in start of injection and start of combustion with single helix and double helix fuel injection pumps. Similar results have been obtained at other lower notches. Due to the use of the double helix fuel injection pump, the pmax is higher than the single helix pump. The θpmax for double helix is about +5º ATDC whereas for single helix pump this maximum in pressure occurs at about +1º ATDC. This is the reason for lower bsfc observed with the double helix pump. Higher peak pressures are accompanied with higher in-cylinder temperatures and are the cause for higher gas turbine inlet temperatures at lower notches with DH pumps.

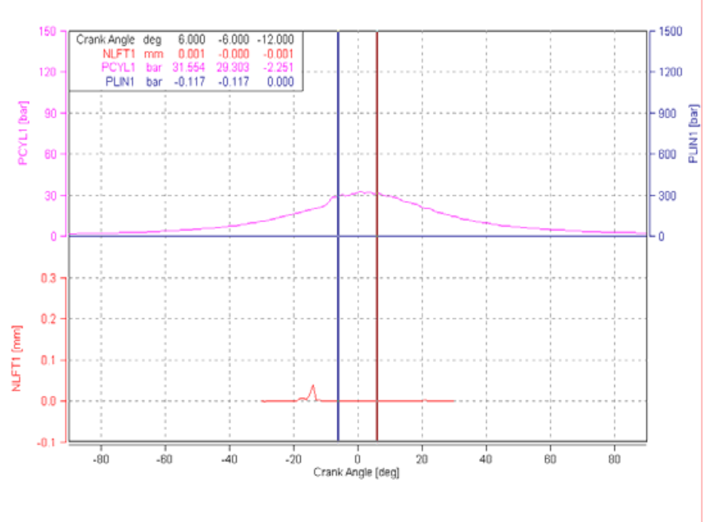


Figure 22: p-θ and lN- θ diagram for single helix pump at idle notch, SOI at -18º BTDC, SOC at -10º BTDC

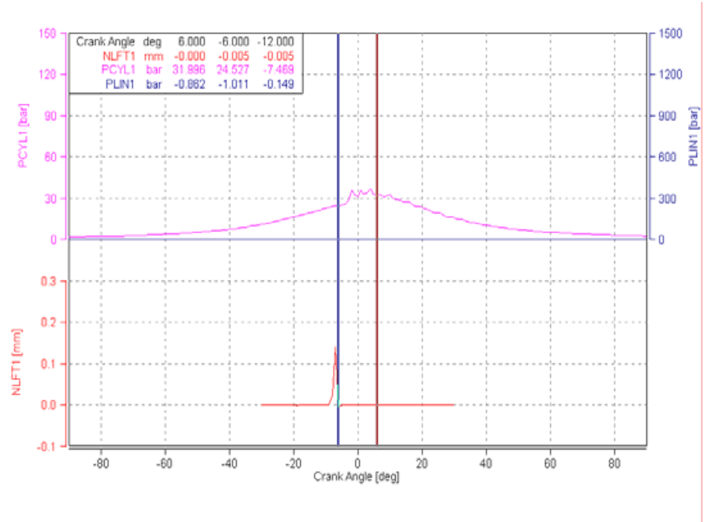


Figure 23: p-θ and lN- θ diagram for double helix pump at idle notch, SOI at -10º BTDC, SOC at -3º BTDC

**CONCLUSION**

Development of double helix fuel injection pump has been successful. The theoretical modelling of the double helix fuel injection is seen to show fair match with the experimental results with double helix pump adjusted for 22º BTDC static fuel injection timing.

The double helix pump has shown fuel savings as compared to the single helix fuel injection pump both in terms of bsfc and the duty cycle fuel consumption. The peak firing pressures with the double helix injection pumps and 22º BTDC static fuel injection timing are found to be below the stipulated limit of 1800 psi. The turbine gas inlet temperatures with double helix fuel injection pump and the 22º BTDC static fuel injection timing are found to be lowest at the higher notches. This combination has also exhibited the lowest bsfc amongst all the configurations tested. Gas turbine inlet temperatures obtained with the 22° DH and the 22° SH pumps are different mainly because of the difference in the combustion efficiencies.

## FIELD TRIALS & DESIGN SWITCH

Subsequent to the development of the double helix fuel injection pump testing on the 16-cylinder ALCO test cell at Research Designs & Standards Organisation, sixteen of these DH pumps were fitted on a diesel locomotive of Indian Railways for field tests. The fuel injection pumps were tested for a period of nine months and showed savings in fuel consumption in line with the laboratory tests. As a consequence, Indian Railways have decided to build all the new ALCO engine based diesel locomotives with double helix fuel injection pumps resulting in design switch from single helix to double helix fuel injection pumps.

## REFERENCES


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3. Fuel Injection and controls for Internal Combustion Engines, Burman Paul G, DeLUCA Frank, The Technical Press Ltd, London, 1962.
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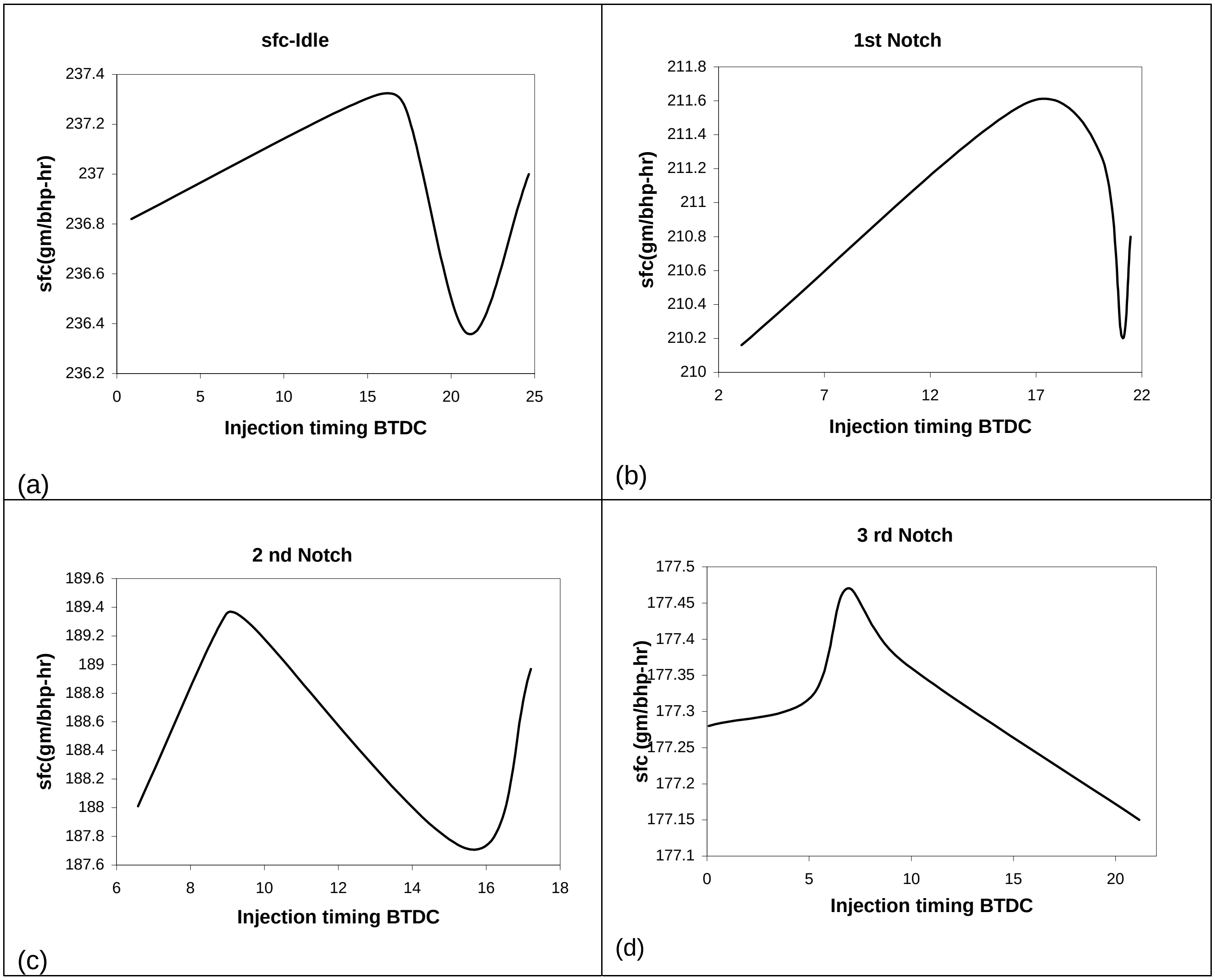
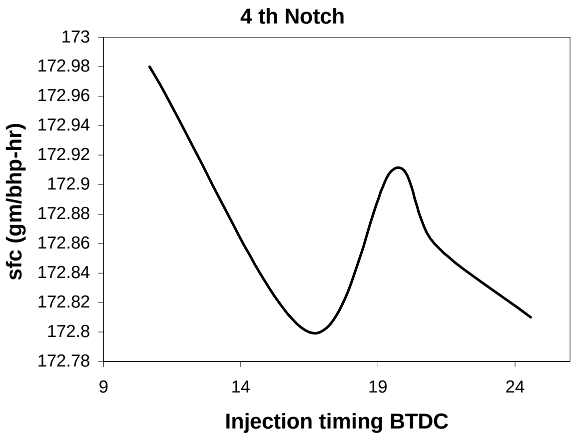
5. Twin-plunger concept for containership, Marine Engineers Review, pp. 31. Feb. 2005

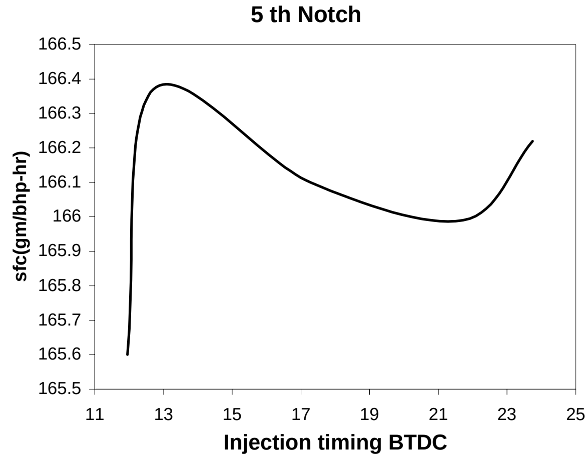
## ANNEX A

### FIGURE 4: NOTCH OPTIMISATION OF INJECTION TIMING

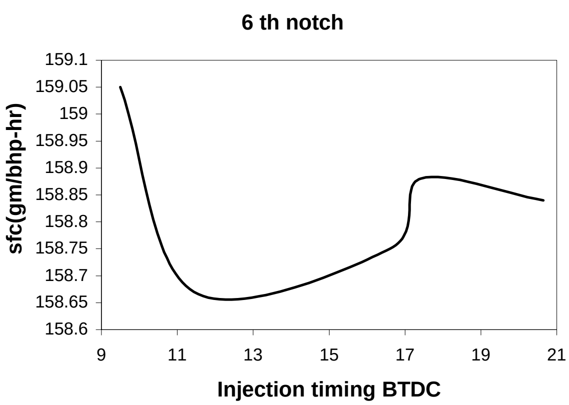



4 th Notch
173
172.98
172.96
172.94
172.92
172.9
172.88
172.86
172.84
172.82
172.8
172.78
sfc (gm/bhp-hr)
9
14
19
24
Injection timing BTDC

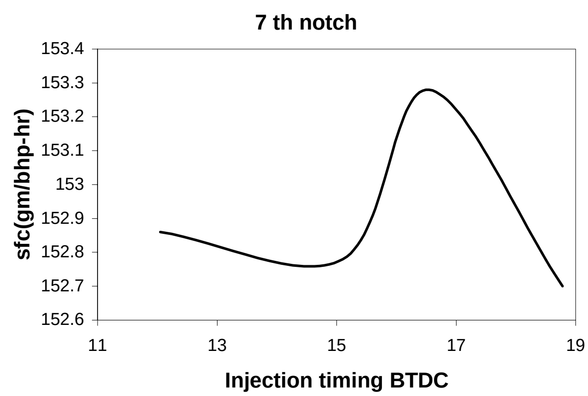
(e)


5 th Notch
166.5
166.4
166.3
166.2
166.1
166
165.9
165.8
165.7
165.6
165.5
sfc(gm/bhp-hr)
11
13
15
17
19
21
23
25
Injection timing BTDC

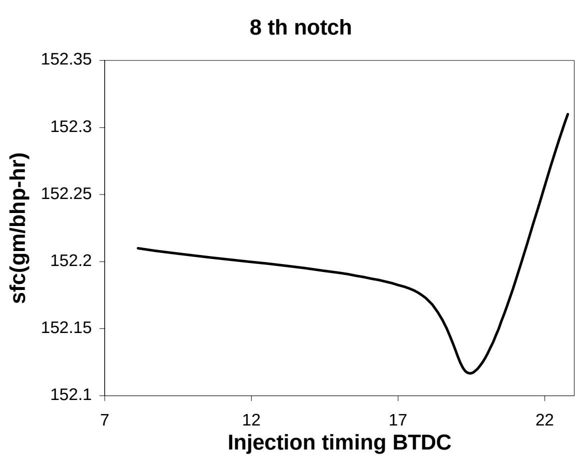
(f)


6 th notch
159.1
159.05
159
158.95
158.9
158.85
158.8
158.75
158.7
158.65
158.6
sfc(gm/bhp-hr)
9
11
13
15
17
19
21
Injection timing BTDC

(g)


7 th notch
153.4
153.3
153.2
153.1
153
152.9
152.8
152.7
152.6
sfc(gm/bhp-hr)
11
13
15
17
19
Injection timing BTDC

(h)


8 th notch
152.35
152.3
152.25
152.2
152.15
152.1
sfc(gm/bhp-hr)
7
12
17
22
Injection timing BTDC

(j)