International Journal of
Emerging Research in Management &Technology
ISSN: 2278-9359 (Volume-2, Issue-10)

Research Article

October 2013

Friction Analysis and Reduction in a Single Cylinder Four Stroke Diesel Engine

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***Abstract*—**

The diesel engines are inherently more fuel efficient and are widely used in the transport sector i.e. passenger cars, buses, trucks, trailers, locomotives, boats and ships. Thus, it is useful to improve further the fuel efficiency by various means. Improving thermal efficiency through reduction in engine friction is one of the options which reduce the frictional power. Reduction of engine friction is desirable for effecting fuel economy in a diesel engine. The reported data on engine friction is traditionally obtained by direct motoring, Willan's line and Morse tests. Many researchers have given relations for estimating friction losses attributable to various components based on motoring tests and then accounting for various design variables and operating conditions of a particular system. Detailed studies on component friction indicate that piston assembly friction accounts for 58-75 percent of the total friction. According to a further estimate, the piston rings may be contributing 75 percent of the total piston friction. To carry out the work, a four stroke diesel engine has been used to determine experimentally total engine friction using Willan's Line method. Next Bishop's, detailed model has been adopted for calculating components of engine friction with minimum of experimental data and using the design data available from manufacturer. The total engine friction is calculated from this model and compared with the Willan's Line data. Further components have been identified which contribute maximum friction amount. Finally modifications have been suggested to parameters of piston rings and piston

***Keywords*— FMEP, Oil rings, Piston skirt, specific fuel consumption, Willan's line method**

I. Introduction

The four stroke diesel engine is one of the most versatile prime movers. Small diesel engines are used for pumping sets, construction machinery, air compressor and drilling rigs. Diesel engine is used for both stationery as well as mobile electric generators. One of the major applications of diesel engine is in the transport sector. i.e. passenger cars, busses, trucks, trailers, locomotives, boats and ships. All use diesel engines for various capacities.A positive feature of a diesel engine is its low specific fuel consumption and high thermal efficiency. A diesel engine has high compression ratio. Thus the chemical energy of the fuel is converted efficiently into heat. However, the whole of this energy is not utilized for driving the piston, as there are losses to the exhaust, to the coolant and some losses by radiation. The remaining energy converted to power is called the indicated power (I.P.). This is utilized to drive the piston. The piston motion through the connecting rod is transmitted to the crankshaft. In this transmission, wherever energy losses due to friction, pumping etc. are taking place, the sum of all the losses is termed as friction power and the useful mechanical energy at the crankshaft is called brake power (B.P.).To calculate brake output from the cycle simulation results of indicated output, a correlation for engine friction is needed. To obtain such a correlation, it is required to accurately measure I.M.E.P and B.M.E.P over a wide range of operating conditions. Unfortunately, it is extremely difficult to measure I.M.E.P with sufficient accuracy using the commonly available indicators and pressure transducers. It has been customary to obtain the engine friction data by simple motoring type tests which include direct motoring, Willan's line test.

II. Problem Formulation and Objectives

The general approach is towards analytical analysis for engine friction.The following objectives were set forth for the proposed analytical method:

- A detailed model for calculating components of engine friction with minimum of experimental data and used for design data available from manufacturer.
- To identify the components which contribute maximum amount to the engine friction through calculations.
- To develop a procedure for calculating the friction of piston and piston rings and to study the effect of important parameters on friction of the above components.
- To formulate a procedure for determining the wall pressure exerted by piston rings and to study the effect of changes in ring parameters on wall pressure.

III. Experimental Setup

The compression ignition engine set-up along with rope brake dynamometer, spring balance, fuel input measuring system, air intake measuring system, digital panel board, thermocouples for temperature measurement, digital tachometer and arrangement for measuring heat carried away by cooling water from engine jacket was supplied by K.C. Engineers Pvt Ltd., Ambala Cantt, Haryana, India. The set-up shown in the Fig.1, consists of a variable speed 661cc, single cylinder, 4-stroke, Kirloskar make, DI Diesel Engine coupled to rope brake dynamometer. The calibrated temperature sensors were used for temperature measurement. The rope brake dynamometer was used to apply the load on the engine. The control panel has digital meters to display the water temperature (inlet and outlet) for engine and calorimeter, exhaust gas temperature to and from the calorimeter. The set-up enables the study of engine brake power, fuel consumption, air consumption, heat balance, thermal efficiency, volumetric efficiency etc.


Fig.-1 Single cylinder CI Engine Test Rig

IV. EXPERIMENTAL PROCEDURE

- Checked for all electrical connections and proper earthlings for the equipment.
- Ensured water in the main water supply tank.
- Ensured selected fuel about 2 liters in quantity in the fuel supply tank.
- Filled the manometer up to half of the height of manometer with water.
- Filled the burette with diesel by opening the valve provided at the lower side of burette.
- Supplied the diesel to the engine by opening the valves provided in the fuel supply line.
- Opened continuous cold water supply to the engine jacket.
- Started the engine and allowed it to run for five minutes on the minimum load so that engine got stabilized.
- When engine started running smoothly, firstly loaded the engine to desired value with the help of brake rope dynamometer and as load increase, R.P.M decrease, so ran the engine for two minutes so that it became stabilize.
- Noted the readings of brake rope dynamometer and noted the R.P.M of engine with the help of hand tachometer.
- Closed the diesel supply valve and opened the valve of burette. Noted down the time taken for consumption of 10 ml of fuel to know fuel consumption rate.
- Now opened the diesel supply valve which refilled the burette.
- After refilling the burette, closed the burette valve and continued the diesel supply by opening the diesel supply valve.
- Noted down the reading of manometer to calculate the air intake by the engine.
- Noted the temperature of inlet and outlet of water circulating through the engine jacket, displayed digitally on control panel
- Noted the temperature of exhaust gas expelled from the engine, on the digital display control panel.
- Measured the flow rate of water with the help of water meter and stop watch.
- Repeated the procedure for loads of 3.7, 5.8, 6, 6.9, 7.2, 8.2 kg maintaining the R.P.M constant.
- When experiment was over, reduced the load on engine and put off the engine.
- Closed the fuel supply and water supply to the engine

V. Results and Analysis

TABLE I PERFORMANCE TEST OBSERVATIONS ON DIESEL FUEL

S.No	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Load, kg	No load	3.7	5.8	6	6.9	7.2	8.2
RPM	1500	1548	1546	1542	1540	1537	1535
Fuel Consumption, ml	10	10	10	10	10	10	10
Time, sec	78	73	60	58	55	53	48
Air consumption (h),Cm	10	9.6	9.5	8.9	8.4	8.2	9
Engine inlet water temp, T_1 ^0C	30	29.4	30	29.4	29.4	29.4	29.4
Engine outlet water temp, T_2 ^0C	35	56	35.4	35	80	82	47
Exhaust gas inlet temperature, T_3 ^0C	135	163	180	217	243	259	227
Exhaust gas outlet temperature, T_4 ^0C	66.5	76.3	81.6	89	92.7	99	92.4
Calorimeter inlet water temperature, T_5 ^0C	30	29.4	30	29.4	29.4	29.4	29.4
Calorimeter outlet water temperature, T_6 ^0C	31.6	34.5	31.5	30.2	33.1	34	35

A. *Test Results of Engine Performance*

TABLE II TEST RESULTS OF ENGINE PERFORMANCE ON DIESEL FUEL

S. No.	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Brake Power, (BP) KW	-	0.9222	1.44072	1.48848	1.70845	1.7792	2.0191
Fuel Consumption, Kg/hr	0.3876	0.417	0.503	0.511	0.547	0.559	0.599
BrakeSpecific Fuel Consumption, (BSFC) Kg/ KW-hr	-	0.4492	0.3498	0.3503	0.322	0.321	0.312
HeatSupplied , KW	4.5899	4.9054	5.9682	6.1742	6.51078	6.7568	7.4602
Air consumption m3/sec	0.0057	0.0056	0.00551	0.00534	0.00519	0.0051	0.0053
Swept Volume, m3/sec	0.0072	0.0072	0.00714	0,00713	0.00712	0.0071	0.0071
Volumetric Efficiency, %	78.61	77.08	77.21	74.895	72.893	72.011	75.63
Brake Thermal Efficiency, %	-	18.799	24.14	24.11	26.24	26.33	27.066

B. *Determination of Friction Power by Willan's Line Method*

This method is used to determine the power lost in overcoming friction. The power developed by engine is greater than what is available at output shaft as some part of the total power developed by engine is lost due to friction. Frictional resistance can be reduced by proper lubrication but it cannot be completely eliminated. In this method fuel consumption Vs brake power at a constant speed has been plotted and the graph extrapolated back to zero fuel consumption. The point where this graph cut the brake power axis is an indication of frictional power of the engine. This negative work represents the combined loss due to mechanical friction and pumping losses.

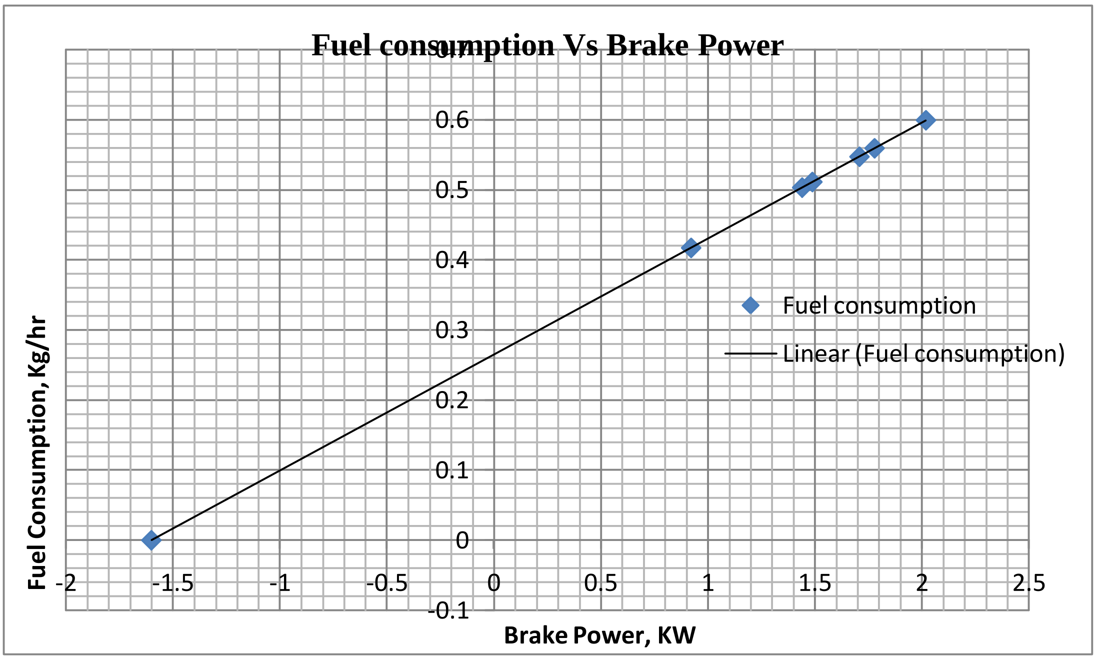

Fig.2 Determination of Frictional power of engine

Friction power = 1.60 kW

$$\text{Friction power} = \frac{(FMEP)*L*A*\frac{N}{2*60}}{1000}$$

$$1.60\ \text{kW} = \frac{(FMEP)*10^5\, 0.110*\frac{3.14}{4}*(.0875)^2*\frac{1500}{2*60}}{1000}$$

(FMEP) = 1.928 bar

C. Analysis of Component Friction using Mathematical Model

1) CRANKCASE MECHANICAL FRICTION

1) (a) BEARING FRICTION

The bearing friction includes the friction due to main bearing, connecting rod bearing and any other auxiliary bearing and is given by the relation:

$$(FMEP)_{bearing} = \frac{K*D*N}{S*1000}$$

Where K is a factor depending upon size and geometry of the journal, engine speed and oil viscosity. For well designed Babbitt bearing of diesel engine K=0.12086

$$= \frac{0.1232*8.75*1500}{11*1000}$$

= 0.1422 bar

1) (b)Valve Gear Friction

Valve gear friction losses vary with the engine design variables and are more difficult to predict. An approximate relation is:

$$(FMEP)_{valve\ gear} = 0.02245\left(30 - \frac{4N}{1000}\right)*\left(\frac{(d_1)^{1.75}}{(D)^2*S}\right)$$

$$= 0.22023\left(30 - 4*\frac{1500}{1000}\right)\left(\frac{3.45^{1.75}}{8.75^2*11}\right)$$

$$= \frac{5.388*3.45^{1.75}}{8.75^2*11}$$

$= 0.05587$ bar

1) (c) Pump and Miscellaneous Friction

Pump and miscellaneous losses are proportional to engine size and speed and are given by the relation:

$$(FMEP)_{pump\&misc} = 0.0273*\left(\frac{N}{1000}\right)^{1.5}$$

$$= 0.02678*\left(\frac{1500}{1000}\right)^{1.5}$$

$$= 0.02678 * 1.8371$$
$$= 0.0492 \text{ bar}$$

2) Blowby (Compression-Expansion Loop) Losses

The leakage losses during compression and expansion are accounted for in the form of "Blow-by" losses and are considered to vary as the square root of inlet pressure while tending to increase with increasing compression ratio and decreasing engine speed. Thus the estimation of blowby is given by the formula:

$$(\text{FMEP})_{\text{blowby}} = 0.069\sqrt{\frac{P_a}{0.975}}\left[1.72 * R^{0.4} - (0.49 + 0.015)\left(\frac{1500}{1000}\right)^{1.85}\right]$$

$$= 0.069\sqrt{\frac{1.01}{0.975}}\left[1.72 * 17^{.4} - (.49 + .015)\left(\frac{1500}{1000}\right)^{1.85}\right]$$

$$= 0.2863 \text{ bar}$$

3) Valve Pumping Losses

The work required to induct fresh charge during the suction and to exhaust the combustion products during the exhaust stroke is given in the form of valve pumping losses and is visible on light spring indicator diagram in the form of pumping loop. The work increases with speed and throttling and is given by formula:

$$(\text{FMEP})_{\text{valve pumping}} = 0.893\left(\frac{N}{1000}\right)^{1.7} * \left[\frac{(0.07585 * V_S)}{2.54 * d^2}\right]^{1.28}$$

$$= 0.893\left(\frac{1500}{1000}\right)^{1.7} * \left[\frac{0.07585 * 662}{2.54 * (3.45)^2}\right]^{1.28}$$

=0.3346 bar

4) Piston Mechanical Friction

Piston mechanical friction is divided into 2 categories:

4) *(a) Friction due to Ring Tension*

The friction due to ring tension is estimated from the formula (2 oil rings and 2 compression rings)

$$(\text{FMEP})_{\text{Ring tension}} = \frac{0.3681 * S * N_r}{D^2}$$

$$= \frac{0.3681 * 11 * 6}{(8.75)^2}$$

=0.31731 bar

Whereas friction resulting from gas pressure behind the piston ring is estimated from the relation:

$$(\text{FMEP})_{\text{Ring gas pr.}} = \frac{P_a}{138.52} * \frac{5.97}{D^2} * S * \left(0.088\, R + 0.182\, R^{\left(1.33 - \frac{0.0476 * S * N}{6000}\right)}\right)$$

$$= \frac{1.01}{138.52} * \frac{5.97}{8.75^2} * 11 * \left(0.088 * 17 + 0.182 * 17^{\left(1.33 - \frac{0.0476 * 11 * 1500}{6000}\right)}\right)$$

=0.0417 bar

4) *(b) Viscous Piston Friction*

Viscous friction which is an important part of total engine friction is calculated from the relation**:**

$$(\text{FMEP})_{\text{viscous-piston}} = \frac{3.826}{6000} * \frac{S_{le} * N}{D}$$

$$= \frac{3.826}{6000} * \frac{6.61 * 1500}{8.75}$$

=0.7225 bar

D. *Total Component Friction (FMEP)*

Total (FMEP) = $(\text{FMEP})_{\text{bearing}} + (\text{FMEP})_{\text{valve gear}} + (\text{FMEP})_{\text{pump \&misc}} +$
$+ (\text{FMEP})_{\text{blow by}} + (\text{FMEP})_{\text{valve pumping}} + (\text{FMEP})_{\text{Ring tension}} + (\text{FMEP})_{\text{Ring gas pr.}}$
$+ (\text{FMEP})_{\text{viscous-piston}}$

= 0.1442 + 0.05587 + 0.0492 + 0.2863 + 0.33346 + 0.31731 + 0.0417 + 0.7225

= 1.9505 bar

TABLE III COMPONENT'S FRICTION

S.No	Components Friction	Frictional Losses, FMEP (Bar)	Component contribution in Engine Friction, %
1.	Bearing Friction	0.1442	7.39
2.	Valve Gear Friction	0.05587	2.86
3.	Pump & Miscellaneous Friction	0.0492	2.52
4.	Blowby Losses	0.2863	14.68
5.	Valve Pumping Losses	0.33346	17.10
6.	Friction due to Ring Tension	0.31731	16.27
7.	Friction due to Ring gas pressure	0.0417	2.14
8.	Viscous PistonFriction	0.7225	37.04
Total Friction		1.9505	100

From the table III it is clear that the total component friction in engine is 1.9505 bar and the contribution of piston and piston rings is 53.31% of the total component friction.

E. Reduction of Friction in Piston Components

1) Reduction in Oil Ring

$$(FMEP)_{compression\ ring} = \frac{0.3681 * S * N_r}{D^2}$$

$$= \frac{0.3681 * 11 * 2}{(8.75)^2}$$

$$= 0.10577 \text{ bar}$$

1 oil ring = 2 compression rings

If one oil ring is removed then friction is reduced by

$$(FMEP)_{Ring\ tension} = 0.31731 - 0.10577$$
$$= 0.21154 \text{ bar}$$

So reduction in friction = 0.10577 bar

2) *Reduction in Skirt Area of Piston*

Since there is no oil ring on the skirt the area of the skirt can be reduced by 12%. Considering that 2% will be increase on the thrust to give support to the piston. Hence net reduction area will be 10%. So reduction in the friction will be

$$(FMEP)_{viscous\text{-}piston} = \frac{3.826}{6000} * \frac{S_{le} * N}{D}$$

$$= \frac{3.826}{6000} * \frac{6.61 * 1500}{8.75}$$

$$= 0.7225 \text{ bar}$$

Reduction in friction = 0.7225 * 0.10
= 0.07225 bar

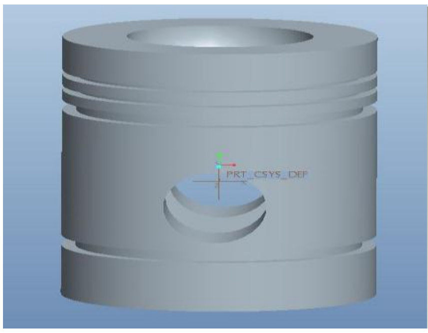

Fig. 3 Original Piston

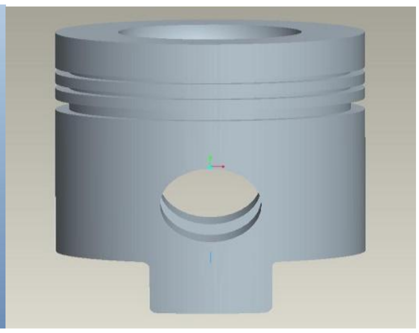

Fig. 4 Modified Piston

3) *Reduction in Axial Width and Tangential Tension*

To estimate the effect of narrow piston rings with reduction in tangential tension, the following formula proposed by McGeehan (14) has been used:

$$FMEP = (C_{max})^{1.5} \times (\mu w)^{0.5}$$

Where C_{max} is the maximum piston speed, it is the dynamic viscosity of the oil, w is the bearing load per unit transverse width and is given by

$$W = p_e \times B$$

Where b is the axial width of the ring and p_e is the effective pressure on the ring is given by

$$p_e = \frac{2 * F_t}{B * D}$$

Where F_t is the tangential tension and D is the outer dia of the ring.

Considering the tangential tension for the present rings as 30 N, width of rings as 2.5 mm and dynamic viscosity of SAE-30 oil as 13.5 × 10^{-3} N-s/m^2, and maximum piston speed of engine at 1500 rpm as 1.63 c_{mean}i.e 9.3 m/s, we obtain

$$p_e = \frac{2 * 30}{\left(\frac{2.5}{1000}\right)\left(\frac{87.5}{1000}\right)}$$

$$= 2.7429 * 10$$

$$w = \frac{2.5 * 2.7429 * 10^5}{1000}$$

$$= 6.8571 * 10^2$$

$$FMEP \propto (9.3)^{1.5}(13.5 * 10^{-3} * 6.8571 * 10^2)^{0.5} \propto (28.36)(3.0426)$$

$$\propto 86.292$$

Now, if axial width is reduced to 2 mm and tangential tension is reduced to 24 N, then FMEP α 77.18. With 2 compression rings and 1 oil ring, the FMEP is 0.21154 barand further reduction of 10.6 percent will being a reduction in FMEP of orders 0.022423 bar.

Thus reduction in friction by taking all the measures together is

$$0.10577 + 0.022423 + 0.07225 = 0.2555 \text{ bar}$$

It is reported in literature that a reduction in FMEP of the order 0.0687 bar gives a fuel economy of the order 1-2%. Thus in the present case fuel economy obtainable will be in the range 3.72 to 7.44%.

VI. CONCLUSIONS

The following conclusions can be drawn from the above discussion:-

- As piston friction both viscous and non-viscous contributes substantially to the total engine friction, special attention has to be paid to piston design.
- Reduction in skirt area of the piston will cause decrease in viscous friction as well as reduction in weight of reciprocating parts, the latter in turn affecting the side thrust. In the present case a reduction of about 10% of skirt area will reduce FMEP by 0.07225 bar. Lug type piston design is recommended.
- In view of the oil rings contributing substantially to the total ring friction (probably twice that of compression rings), one of the oil rings especially below the wrist pin can be dispensed with. This will also facilitate the removal of skirt area at that place, thereby simplifying the piston design.
- Reduction of the axial width of the piston rings and reduction in the tangential tension reduces the engine friction. In the present case reduction in axial width from 2.5 mm to 2 mm and reduction of the tangential tension from 30 N to 24 N brings about a reduction in engine friction of 10.6 %.
  - As per estimates from literature, a reduction in friction of piston and rings results in fuel economy is the measurable range of 3.7 – 7.4%.

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