Design and development of a diesel engine computer simulation program

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ABSTRACT: This paper concerns the development of a diesel engine mathematical model and a suite of computer simulation programs which would allow the effects of various design and operational changes to be reliably and accurately predicted with the ultimate aim of producing cleaner engines and/or more efficient power units. The model has been tested against the experimental results of the Paxman engine at Newcastle University and earlier against the Atlas engine at Ricardo, Brighton, UK. The predicted results and the experimental data are in good agreement.

1 INTRODUCTION

The mathematical model developed has one main program and two auxiliary programs. The main program is called, "Element Mixing Program" which calculates cylinder pressure and hence engine performance parameters from combustion equations that leads to the calculation of heat release. The first auxiliary program is known as "Heat Release", which calculates heat released in one engine cycle using experimentally measured cylinder pressure data. The second auxiliary program is called "Rate of Injection". In this program the total mass of fuel injected into the cylinder in one engine cycle is obtained. The output for rate of injection is used as input data to the main program.

The main issue for the engine modelling is to calculate the fuel-air mixing process which is the fundamental part of the heat release. After calculating and understanding the heat release characteristic of the diesel engine, other parameters can be calculated easily.

The main aim of this paper is to design and develop a clean diesel mathematical model and computer simulation program to predict the overall engine performance for changing load and speed parameters. The model should be able to respond to the changes in various parameters according to already established experimental trends and observations. The initial computer programs were developed by Ziarati (1991). These programs were translated into FORTRAN programming language and several changes were carried out.

In order to test the computer programs against experimental data, a computer controlled engine rig was set up at TUDEV, Turkey.

2 DESCRIPTION OF THE MODEL

The programs use the modified air standard cycle as shown in Figure 1.

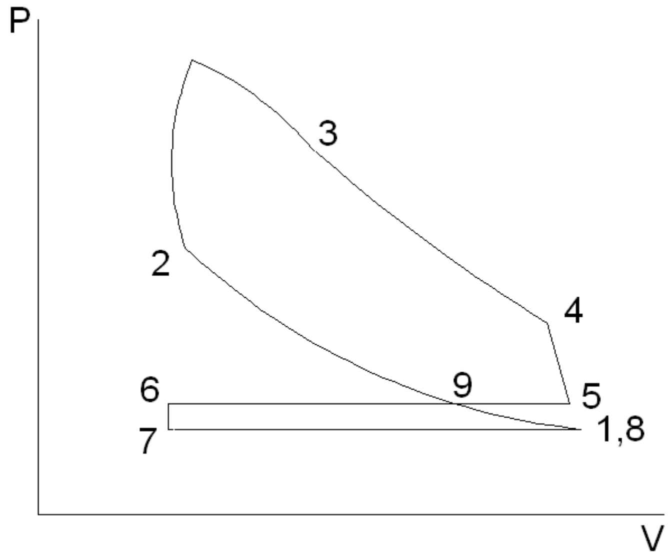

Figure 1. Air Standard Cycle

The cycle is subdivided into:

a) Closed Cycle Considerations:

- Compression period (1-2)
- Combustion period (2-3)
- Expansion period (3-4)

b) Open Cycle Considerations:

- Blow down period (4-5)
- Exhaust period (5-6)
- Overlap period (6-7)
- Suction period (7-8)
- Pre-compression period (8-9)

The model can be divided into two calculation parts: Analysis and Synthesis. The analysis part has its own program named “Heat Release Analysis Program”. The synthesis part is the “Element Mixing Program”.

2.1 *Element Mixing Program*

The model assumes that after injection the air entrainment is controlled by an elemental fuel jet until the wall impingement. After the injected fuel impinges to the wall the air entrainment is controlled by an elemental wall jet and close to this jet the intimate fuel air mixing within the jet is controlled by turbulent diffusion. The entrainment of the fuel jet and wall jet is controlled by entrainment ratio and micro mixing of the remaining fuel by diffusivity constant. These values are dependent on engine load, speed and boost depending on the application. The model can be used to predict heat release qualitatively for any direct injection engine using estimated values of the diffusivity constant and entrainment ratio.

The mixing of air with the injected fuel in the combustion chamber is directly proportional to the air entrainment by the fuel jet at any instant quantities. Micro mixing of fuel and air depends on a simple consideration of turbulent diffusion. This expression includes an Arrhenius type function which controls the rate of burning (micro mixed fuel and air). The following assumptions are made:

- The molar change of the cylinder content is negligible before, during and after combustion

- During combustion, that is, during the period of heat release, the increase in the internal energy is based on the data (Gilchrist, 1947) which takes into account the effect due to changes of specific heat and dissociation. Hence, in considering internal energy and specific heat, dissociation has to be taken into account although the effect of its molar change on pressure and volume may be neglected.

2.1.1 *Object of the Program*

The total fuel injected is divided into a number of elements and the individual fuel-air mixing history of each element is calculated independently using the model. From these histories an aggregate mixing condition can easily be calculated and from this latter the cylinder pressure and heat release are determined (Ziarati, 1991).

2.1.2 *Pre-Compression Consideration and Compression Period*

The compression period starts from the initial atmospheric conditions. The compression process is assumed to obey the rule:

$$PV^n = \text{constant} \tag{1}$$

Assuming compression starts at bottom dead centre, the crank angle dependent volume can be calculated as follows (Heywood, 1988):

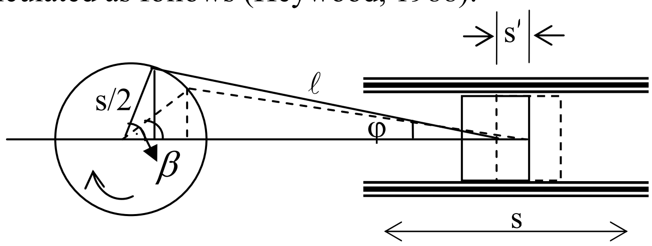

Figure 2. Piston Travel

$$s' = l + (s/2) - (l\cos\phi + (s/2)\cos\beta) \tag{2}$$

$$V' = s'.\,piston\,area \tag{3}$$

Then the trapped mass must be calculated at point 1. The following two methods can be used:

- Simplified perfect displacement model:

$$M_T = \left(s.P_A.\eta_v + V_{cl}.\frac{T_1}{T_{exh}} \right) \frac{P_1}{R.T_1} \tag{4}$$

- Simplified perfect mixing model:

$$M_T = (s_{eff}.P_A + V_{cl}) \frac{P_1}{R.T_1} \tag{5}$$

The air flow through the clearance volume during the scavenge period is given by:

$$\dot{V}_a = 2.\pi.\bar{d}.\bar{l}.\sqrt{\left(\frac{P_{in} - P_{ex}}{\rho_{in} - \rho_{ex}} \right)} \tag{6}$$

For perfect mixing the scavenge efficiency:

$$\eta_s = 1 - e^{-SR} \tag{7}$$

After calculating scavenge efficiency, the loss of air to exhaust and trapped residual gases can be calculated. The inlet temperature can be modified using the universal gas equation;

$$T_1 = \frac{P_1.V_1}{M_T.R} \tag{8}$$

So the temperature at start of injection;

$$T_2 = T_1 \cdot \left(\frac{V_1}{V_2} \right)^{m-1} \quad (9)$$

Compression work,

$$W_{1-2} = P_1 \left[V_1 - V_2 . \left(V_1 / V_2 \right)^m \right] .10^{-6} \quad (10)$$

Other parameters calculated at this stage are as follows:

Air-fuel ratio;

$$R_T = \frac{M_T}{M_F} \quad (11)$$

Total heat input per unit mass of air,

$$H_i = \frac{LHV}{R_T} \quad (12)$$

Fractional air utilization:

$$U' = \frac{R_{stoich}}{R_T} \quad (13)$$

Applying the momentum equation (Bernoulli's equation) the overall mean injection pressure is also calculated as follows:

Volumetric flow rate,

$$\dot{Q} = KA \sqrt{\frac{2\Delta P}{\rho_f}} \quad (14)$$

Therefore;

$$M_F = \dot{Q} \rho_f \frac{T_I}{6RPM} \quad (15)$$

Thus

$$\Delta P = 71{,}0433.10^{-6} \left[\frac{M_F \ RPM}{h_n \, T_I \, d_n^{\ 2}} \right]^2 \quad (16)$$

Before the start of combustion, air zone temperature is equal to product zone temperature and is given by:

$$TAZ = TBZ = T_1 \left(\frac{V_1}{V_2} \right)^{m-1} \quad (17)$$

This leads to the evaluation of the internal energy of the compressed air using the given Gilchrist Table (Gilchrist, 1947) and the gas constant function. There are two periods: The period of pre-mixed combustion and rate of pressure rise (and therefore initial rate of heat release) is governed by the quantity of fuel injected during the delay period. In the second period, the burning is controlled by the mixing rate which has an empirical expression.

2.1.3 Free Jet

The fuel injected first forms a free jet and may later form an axi-symmetric wall jet. For a given step, rate of injection is assumed constant, and the position of free jet at time ‘t’ since the beginning of injection is assumed to be governed by the following equation (Ziarati et al., 1988):

$$y = 1020{,}8 \left(\frac{d_n \, P_i^{1/2} \rho_{stpt}}{\rho_c} \right)^{1/2} \quad (18)$$

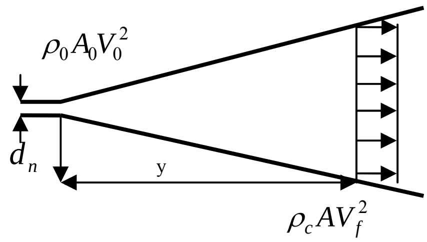

Figure 3. Spray Presentation (Ziarati, 1991)

Application of momentum equation leads to:

$$\Delta E = \int_t^{t+\Delta t} \frac{\pi}{3} \tan^2 \theta \, y^3 \rho_a \, dt \quad (19)$$

Integrating equation (19) and using equation (18) lead to:

$$\Delta E = \frac{\pi}{3} \rho_a \tan^2 \theta \left[\frac{1{,}042.10^6 \, d_n \, P_i^{1/2} \rho_{stp}}{\rho_c} \right]^{3/2} \left[(t + \Delta t)^{3/2} - t^{3/2} \right] \quad (20)$$

2.1.4 Wall Jet

Once t is greater than T_Y (impingement time) the jet front changes from a free jet to a wall jet (Ziarati et al., 1988). Transition time, air entrainment during transition and loss of kinetic energy in the direction of flow for the transition are neglected. The initial conditions for the wall jets are:

$$r_0 = Y \tan \theta \quad (21)$$

and since the velocity and flow areas immediately before and immediately after transition are assumed to be the same, initial volume flow at wall jet can be found:

$$Q_0 = \pi \, r_0^2 \, W_0$$
$$Q_0 = \pi \, y^2 \tan^2 \theta W_0 \quad (22)$$

The equation of Glauert (Glauert, 1956) is now can be used to describe the velocity W, jet thickness and volume flow Q (of the wall jet) assuming again a square velocity profile for the jet front,

$$\Delta E_{wall} = \frac{Q_0 \, \rho_a}{1{,}459 \, t_0^{0{,}459}} \left[(t_w + \Delta t)^{1{,}459} - t_w^{\ 1{,}459} \right] \quad (23)$$

After the end of the injection the net increment in air entrainment is then:

$$\Delta E = \Delta E_{jet\ front} - \Delta E_{jet\ back} \quad (24)$$

Further, the jet is assumed to expand or contract with cylinder volume.

2.1.5 Micro Mixing

While the air entrained by the gas jet at any instant quantifies the larger scale mixing of fuel and air in the chamber, intimate mixing of fuel and air is represented by a simple consideration of turbulent diffusion. The macro mixed quantity of air within the jet boundaries, as determined by air entrained E is assumed to micro mix with injected fuel according to the equation:

$$\frac{dM_a}{dt} + DV_f\, M_a = DV_f\, E \tag{25}$$

Multiplying both sides of the equation by $e^{DV_f\, t}$ and integrating:

$$M_a\, e^{DV_f\, t} = E\, e^{DV_f\, t} + c \tag{26}$$

Heat release therefore;

$$H_{release} = \frac{M_a\, LHV}{15} \tag{27}$$

2.2 Combustion Work Done

The combustion work can be obtained from the 1st law of thermodynamics (Heywood, 1988, Ferguson, 1986) as follows:

$$W_{2-3} = W_{combustion} \tag{28}$$

2.3 Delay Period

With the following empirical expression the delay period can be calculated (Ziarati, 1990):

$$DEL = 15{,}375.10^{-3}\; RPM(C_n)^{-a}\,(P)^{-0{,}38}\, e^{AE/T} + 39{,}04\Delta P^{-0{,}35}\, d_n^{\;0{,}4} \tag{29}$$

Where a=0,7 to 1,0 and AE is the activation energy.

2.4 Heat Transfer

The heat transfer formula used in the model is Annand's based on the actual cylinder piston surface areas and is calculated step by step throughout the calculations (Annand, 1963). This equation can be expressed as follows:

$$\dot{q} = A\frac{a\,K}{D}(\mathrm{Re})^b\,(T - T_w) + c(T^4 - T_w^{\;4}) \tag{30}$$

2.5 Work Done

Expansion work: The expansion process starts from the condition at the end of combustion and this period is the last sequence of closed cycle calculations. The expansion work (Heywood, 1988, Ferguson, 1986):

$$W_{3-4} = \frac{(P_3 V_3 - P_4 V_4)}{n-1}.10^{-6} \tag{31}$$

Closed period work done: Closed period work done is the total work of the processes 1-2, 2-3 and 3-4:

$$W_{1-4} = \int_1^4 Pdv = W_{1-2} + W_{2-3} + W_{3-4} \tag{32}$$

Blown-down period work done:

$$W_{4-5} = \frac{P_4 - P_5}{2}(V_5 - V_4) \tag{33}$$

Exhaust period work done**:**

$$W_{5-6} = -P_6\,(V_6 - V_5) \tag{34}$$

Overlap period work done:

$$W_{6-7} = -P_5(V_6 - V_{cl}) + P_7(V_7 - V_{cl}) \tag{35}$$

Suction period work done:

$$W_{7-8} = P_7(V_8 - V_7) = P_7(V_5 - V_7) \tag{36}$$

Pre-compression work:

$$W_{4-8} = \int_4^8 Pdv = W_{4-5} + W_{5-6} + W_{6-7} + W_{7-8} \tag{37}$$

2.6 Overall Cycle Parameters:

$$W_{ind} = W_{closed\ period} + W_{open\ period} \tag{38}$$

Since the friction mean effective pressure is generally known, the brake work is given by:

$$W_{brake} = W_{ind} - FMEP.V_{swept} \tag{39}$$

The power output indicated and brake can be calculated respectively as:

$$P_{ind} = W_{ind}\,.RPM \tag{40}$$

$$P_{brake} = W_{brake}\,.RPM \tag{41}$$

Similarly specific fuel consumption is given by:

$$SFCI = \frac{RPM\;\; M_F}{P_{ind}} \tag{42}$$

And

$$SFCB = \frac{RPM\;\; M_F}{P_{brake}} \tag{43}$$

The computer program represents equations (1) to (43) with some exceptions and additional complications and facilities (Ziarati, 1991).

3 HEAT RELEASE PROGRAM

The basis of this "Heat Release" program depends on the 1st law of thermodynamics (Heywood, 1988, Ferguson, 1986):

$$\begin{gathered} Q - W = U \qquad \textit{Therefore}: \\ Q = \int_B^{AB} Pdv + U \end{gathered} \tag{44}$$

4 RATE OF INJECTION PROGRAM

This program can be used only when practical injection diagrams are considered. Rate of injection program calculates the rate of fuel injected at specified crank angle. The annular area between the needle and the seat and total nozzle holes area are assumed to be two orifices in series. Knowledge of the line pressures P_L, cylinder pressures C_P and needle lifts L for specified crank angles for any given step enables the rate to be calculated:

$$Q = K_e\, A_e'\,\sqrt{(P_{diff} / RPM)} \tag{45}$$

Q can therefore be calculated for each element. The equation (45) can only be correctly determined if A_s and A_h can be calculated as shown:

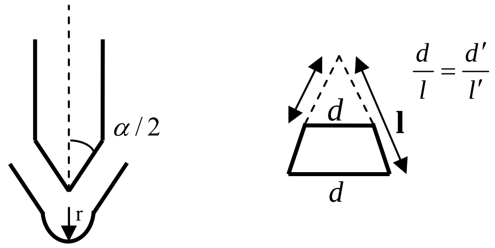

Figure 4. Valve Seat Presentation (Ziarati, 1991)

If the seat angle is assumed to be 60^0, the truncated cone A_s' equals to:

$$A_s' = \frac{\pi}{2} d\,l - \frac{\pi}{2} d'\,l' = \frac{\pi}{2}(d\,l - d'\,l') \tag{46}$$

Thus,

$$A_s' = \pi L \sin(\alpha/2)\big[d - L\sin(\alpha/2)\cos(\alpha/2)\big] \tag{47}$$

Assuming $d = d'$ and $\alpha = 60^0$, A_s' expression reduces to:

$$A_s' = \pi\, r_{sac} \,.\, L \tag{48}$$

5 VALIDATION OF THE PROGRAMS

The developed model was applied to a single cylinder direct injection diesel engine (Ricardo, Atlas) for a number of nozzle configuration, plunger sizes, engine speeds and engine loads. The experimental heat release data was available. Engine dimensions are below:

Table 1. Research Engine Specifications

No. of Cylinders	1
Bore	215.9 mm
Stroke	241.3 mm
Swept Volume	8833 mm^3
Compression Ratio	11
Inlet Valve Opening (ABCD)	48 deg CA

Table 2. Theoretical and Experimental Results

					Experimental				Theoretical			
Plunger (mm)	Nozzle config.	Boost (bar)	Scavenge ratio	Speed	Bmep	Bsfc	Isfc	Pmax	Bmep	Bsfc	Isfc	Pmax
22	8x0,40	3,0	1,05	1500	17,91	261,5	193,8	146	17,88	263	195	144,3
22	9x0,35	2,12	1,05	1500	10,84	274,9	174,4	115	10,83	276	174,7	113,3
22	9x0,45	2,12	1,05	1500	10,83	291	184,5	115	10,65	296	187	114,2
22	9x0,35	3,0	1,05	1200	19,48	240,6	194,6	145	18,77	250	201	144,9
20	9x0,40	3,0	1,05	1500	17,78	274,7	203,5	146	17,90	274	203	142,7
20	9x0,40	3,6	1,1	1500	17,78	281,5	208,3	145	18,03	277	206	143,0
20	9x0,40	3,0	1,0	1000	20,56	249,4	213,1	146	20,04	256	218	144,2
20	9x0,40	3,6	1,05	1000	20,56	244	208,5	147	20,66	243	207	145,2
20	8x0,35	3,6	1,05	1200	19,54	242,9	196,6	149	19,56	243	196	148,6
20	8x0,35	3,6	1,05	1200	5,11	329,5	173,5	73	5,1	331	174	76,5
18	8x0,45	3,0	1,05	1500	17,67	282,7	208,8	147	17,90	282	209	143

Table 2 above summarizes the theoretical and experimental results for various conditions.

The screenshots of outputs from "Element Mixing Program", "Heat Release Program" and "Rate of Injection Program" can be seen in Figures 5, 6 and 7 respectively:

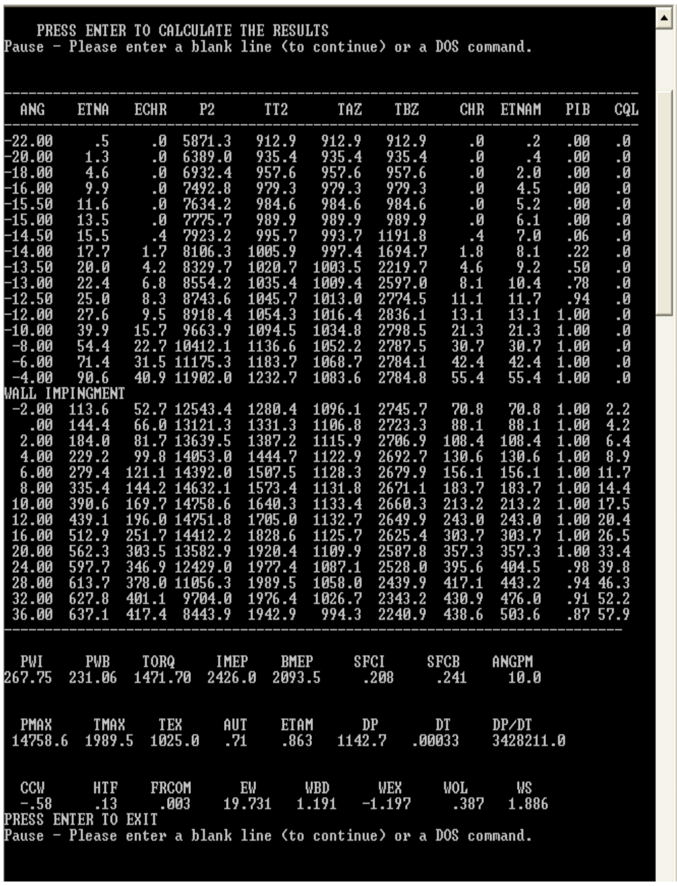

Figure 5. Output Screenshot of Element Mixing Program

RESULTS

HEAT RELEASE CALCULATION CYCLE= 9

CA	CYL P	VOL	TEMP	HEAT R	WORK D	DQ	DW
161.0	977.0	1109.6	941.0	.00	.00		
162.0						.611	-1.910
163.0	1063.0	1046.7	965.8	1.22	-3.82		
164.0						-1.013	-1.839
165.0	1132.9	990.4	973.9	-.80	-7.50		
166.0						.108	-1.730
167.0	1214.6	940.9	992.0	-.59	-10.96		
168.0						.843	-1.597
169.0	1299.8	898.2	1013.5	1.10	-14.15		
170.0						5.710	-1.453
171.0	1432.7	862.5	1072.6	12.52	-17.06		
172.0						.374	-1.255
173.0	1500.7	833.7	1086.1	13.27	-19.56		
174.0						5.693	-1.007
175.0	1619.2	812.0	1141.4	24.65	-21.58		
176.0						6.000	-.727
177.0	1729.1	797.5	1197.0	36.65	-23.03		
178.0						5.850	-.395
179.0	1822.8	790.0	1250.0	48.35	-23.82		
180.0						7.469	-.018
181.0	1916.5	789.7	1313.7	63.29	-23.86		
182.0						7.957	.396
183.0	1993.2	796.5	1378.1	79.20	-23.06		
184.0						7.435	.836
185.0	2036.6	810.4	1432.7	94.07	-21.39		
186.0						8.435	1.284
187.0	2065.8	831.5	1491.0	110.94	-18.82		
188.0						11.068	1.742
189.0	2100.0	859.6	1566.9	133.08	-15.34		
190.0						10.935	2.198
191.0	2107.3	894.7	1636.6	154.95	-10.94		
192.0						12.227	2.635
193.0	2105.1	936.7	1711.7	179.40	-5.67		
194.0						9.659	3.031
195.0	2061.3	985.6	1763.6	198.72	.39		
196.0						10.255	3.372
197.0	2010.2	1041.2	1816.9	219.23	7.13		
198.0						9.809	3.668
199.0	1945.5	1103.5	1863.6	238.85	14.47		
200.0						8.246	3.902
201.0	1863.7	1172.4	1896.7	255.34	22.27		
202.0						7.220	4.072
203.0	1773.4	1247.6	1920.6	269.78	30.42		
204.0						6.992	4.192
205.0	1683.1	1329.1	1941.9	283.76	38.80		
207.0						4.862	4.274
209.0	1488.1	1510.3	1950.9	303.21	55.90		
210.0						4.639	4.265
211.0	1398.3	1609.6	1953.7	312.49	64.43		
212.0						4.370	4.232
213.0	1313.5	1714.5	1954.7	321.23	72.89		
214.0						3.319	4.173
215.0	1230.0	1824.7	1948.2	327.87	81.24		
216.0						1.424	4.080
217.0	1144.8	1940.2	1928.1	330.72	89.40		

Figure 6. Output Screenshot of Heat Release Program

RESULTS

FUEL INJECTON CALCS CYCLE= 9 CORRECTION FACTOR= 1.0806

CA	FLOW R	ACC R	PN	CYL P	LINE P	LIFT
161.0	.1465	.0000	.05	977.00	3857.08	.0011
163.0	8.4172	8.5637	176.86	1063.03	5106.50	.0555
165.0	23.9945	40.9755	1437.23	1132.88	4632.05	.2224
167.0	34.3157	99.2856	2939.60	1214.65	4910.09	.5550
169.0	40.3377	173.9390	4061.87	1299.83	6270.72	.5994
171.0	41.2289	255.5057	4243.33	1432.70	6625.67	.5994
173.0	43.2576	339.9921	4671.19	1500.68	7217.25	.5994
175.0	48.5308	431.7805	5879.47	1619.25	8814.50	.5994
177.0	52.5062	532.8176	6882.16	1729.13	10151.47	.5994
179.0	52.6565	637.9803	6921.61	1822.82	10293.45	.5994
181.0	53.0989	743.7357	7038.41	1916.52	10530.08	.5994
183.0	53.9499	850.7845	7265.80	1993.18	10885.03	.5994
185.0	53.2027	957.9371	7065.95	2036.62	10683.89	.5994
187.0	52.3603	1063.5000	6843.96	2065.75	10441.34	.5994
189.0	50.3854	1166.2460	6337.41	2099.99	9855.68	.5994
191.0	45.4504	1262.0810	5156.77	2107.32	8418.15	.5994
193.0	40.4307	1347.9630	4080.61	2105.10	7098.93	.5994
195.0	27.9891	1416.3820	1955.60	2061.32	4454.58	.5994
197.0	16.7099	1461.0810	697.03	2010.22	2863.24	.5994
199.0	10.5102	1488.3020	275.76	1945.48	2307.15	.4995
201.0	9.0551	1507.8670	204.69	1863.71	2336.73	.2331
203.0	.0000	1516.9220	.00	1773.42	1478.94	.0011

PRESS ENTER TO EXIT
Pause - Please enter a blank line (to continue) or a DOS command.

Figure 7. Output Screenshot of Rate of Injection Program

The predicted cylinder temperature against the experimental cylinder temperature calculated from "Heat Release Program" is shown in Figure 9 below:

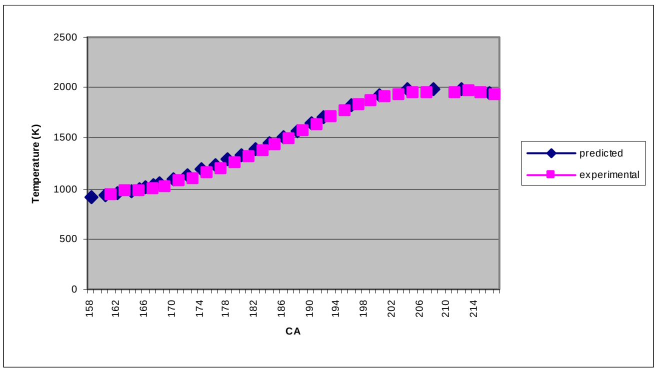

Figure 9. Predicted and Experimental Cylinder Temperature

The graphical comparison of predicted heat release and experimental heat release data for a given engine condition is shown in Figure 8.

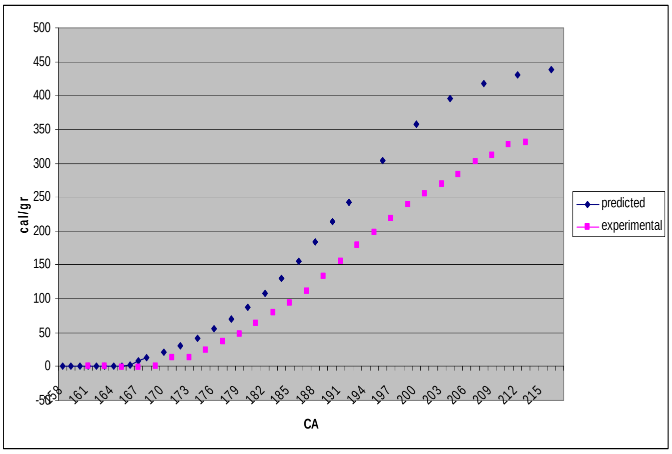

Figure 8. Predicted and Experimental Heat Release

6 CONCLUSIONS

As can be seen the predicted theoretical values are in good agreement with the experimental results. The ability of the model to predict engine performance parameters to within the experimental error band for different test conditions must be considered very encouraging and a proof of the model. The Element Mixing Program can now be used reliably in establishing the effects of changes to running conditions or testing of changes to engine design parameters or to fuel injection equipment as well as to turbocharger configuration, and indeed to any other engine sub-system, without resorting to running engine unnecessarily. This would reduce the cost of engine design and development to a minimum.

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