Thermal design of a natural gas - diesel dual fuel turbocharged V18 engine for ship propulsion and power plant applications

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Abstract. A detailed method is presented on the thermal design of a natural gas - diesel dual fuel internal combustion engine. An 18 cylinder four stroke turbocharged engine is considered to operate at a maximum speed of 500 rpm for marine and power plant applications. Thermodynamic, heat transfer and fluid flow phenomena are mathematically analyzed to provide a real cycle analysis together with a complete set of calculated operation conditions, power characteristics and engine efficiencies. The method is found to provide results in close agreement to published data for the actual performance of similar engines such as V18 MAN 51/60DF.

1. Introduction
For more than a century, internal combustion engines (ICEs) have revolutionized both human transportation and habits. Spark (Otto) and compression (Diesel) ignition engines have spanned everyday life applications from cars, trucks and small airplanes to agriculture and manufacturing equipment, trains, ships and power plants. In view of the future increase of both global population and energy demand, internal combustion engines are currently attracting a renewed research interest aiming at greater thermal efficiencies and lower fuel consumptions. Diesel engines, especially, are found to play a critical role in the high power range from 0.5 to 100 MW. Such engines typically operate below 1000 rpm, have up to 18 cylinders in line or V configurations, and find applications in ship propulsion, driving of machinery and power generation plants [1-3].

Thermal evaluation of ICEs has always been one of the fundamental criteria for engine design and performance. Theoretical and experimental methods on this field have been documented through the years in numerous books and reports [3-7]. In theory, the accuracy of a thermal design depends on the number and strength of the applied assumptions. Ideal air cycle analysis is commonly employed as fast and easy to provide the maximum thermal efficiency of an engine at a given pressure ratio. More accurate but complex methods need to account for real cylinder compositions, real valve timing, heat transfer, fluid flow dynamics, real geometry considerations and real combustion phenomena. Such real cycle methods are able to closely simulate actual performance and are valuable for preliminary modeling, cost effective engine development and energy management improvements.
In the present work, a detailed method is presented for the thermal design of a large turbocharged compression ignition engine. Real cycle analysis is accomplished based on fundamental science and empirical relations and is found able to accurately predict engine performance in comparison to actual data documented for the MAN 51/60DF (dual fuel) engine [2, 8]. The study was undertaken to formulate a basis for future work on internal combustion engine simulations and exhaust heat recovery. The method employed provide a set of engine operation conditions exploitable for energy management and efficiency improvements in diesel engine power plants.

2. Engine and Fuel Characteristics
The MAN 51/60DF engines are offered in 6-to-9 in-line and V12 to 18-cylinder models covering a power range up to 18000 kW at 500 rpm with a brake mean effective pressure rating of 1.9 Mpa [2]. As an actual example the four stroke V18 cylinder MAN 51/60DF engine has been considered. A cross section of the engine is shown in Fig. 1. This is a natural gas-diesel dual fuel engine equipped with a MAN TCA77 (four stroke version) turbocharger. Engine and turbocharger characteristics are summarized in Table 1 and Table 2, respectively [8, 9]. Engine speed is set at the maximum value of 500 rpm and operation is considered steady at full load.

![Figure 1. Cross section of the V18 MAN 51/60DF engine [8].](image)

| Table 1. 18 cylinder MAN 51/60DF engine characteristics [8] |
|------------------------------------------------------------|
| Parameter       | Value |

2
Number of cylinders       | 18         
Cylinder bore             | 510 mm     
Piston stroke             | 600 mm     
Displacement per cylinder | 122.5 lt   
Compression ratio         | 13.3       
Vee angle                 | 50°        

Table 2. MAN TCA77 (four stroke version) turbocharger characteristics [9]

| Parameter                          | Value         |
|------------------------------------|---------------|
| Turbine type                       | Axial flow turbine |
| Pressure ratio                     | up to 5.5     |
| Maximum permissible temperature    | 650°C         |
| Maximum permissible engine power   | 20900 kW      |
| Air flow rate                      | about 15-35 m³/s |

The engine may operate with either diesel oil or natural gas. In the course of the analysis two modes of operation have been considered, one with heavy oil with an approximate composition of 87 wt. % carbon and 13 wt. % hydrogen, and one with natural gas with an approximate composition of 95% mol CH₄, 4% mol C₂H₆ and 1% mol C₃H₈. Lower heating values (LHV) were calculated to be 40 MJ/kg and 49.8 MJ/kg, respectively. Combustion typically takes place with lean fuel mixtures and the analysis was conducted assuming a lambda λ ratio equal to 2.

3. Method

Atmospheric conditions were assumed at $T_0 = 293$ K and $p_0 = 0.1$ MPa. Turbocharger pressure ratio was set at the maximum value of $\Pi = 5.5$, and compressor outlet temperature and pressure were calculated as

$$p_C = \Pi p_0$$

$$T_C = T_0 \left( 1 + \frac{\Pi^{(\gamma-1)/\gamma} - 1}{\eta_C} \right)$$

where $\gamma = c_p / c_v$ is the average specific heat capacity ratio of air during compression (equal to 1.39) and $\eta_C$ is the isentropic efficiency of the compressor taken 0.8. Air density may also be taken as

$$\rho_a = \frac{10^6 p_C}{RT_C}$$

where $R = 287$ J/kgK is air gas constant.

For a turbocharged engine at the nominal speed of maximum brake power, the pressure of the residual gases inside the cylinder is typically 75-98% of the $p_C$. Assuming a mean percentage of 85% this was calculated as

$$p_r = 0.85 p_C$$

Then the temperature of the residual gases was assumed equal to $T_r = 825$ K. The assumed value will later be calculated and corrected if necessary.

Air exiting from the turbocharger typically undergoes a temperature change inside the intake manifold before entering the cylinder. This is of the order of -5 to +10 K and an average value of
\( \Delta T = 5 \text{ K} \) was considered. Moreover, pressure loss in the intake manifold can be estimated according to Bernoulli equation as

\[
\Delta p = 10^{-6} (\beta^2 + \xi_{in}) \left( \frac{V_{in}^2}{2} \right) \rho_a
\]

Here \( \beta \) is a coefficient of velocity decrease inside the manifold, \( \xi_{in} \) is a coefficient of local pressure loss due to intake valve port restriction and \( V_{in} \) is the mean velocity of air at the valve port restriction. Empirical values of \( (\beta^2 + \xi_{in}) \) vary between 2.5 and 5, while \( V_{in} \) typically vary between 50 and 130 m/s. In the present study, assumed values were taken equal to 3 and 70 m/s, respectively. Then, air pressure at the end of intake stroke is

\[
p_1 = p_C - \Delta p
\]

The mass ratio of residual gases to fresh air inside the cylinder is given as

\[
x = \left( \frac{T_C + \Delta T}{T_r} \right) \left( \frac{p_r}{\gamma p_1 - p_r} \right)
\]

where \( r \) is compression ratio, and the fraction of the residual gases in the total mass trapped is

\[
x_r = \frac{x}{1 + x}
\]

Air temperature at the end of intake stroke is approximately given by

\[
T_1 = \frac{T_C + \Delta T + xT_r}{1 + x}
\]

Cylinder gas composition during compression is calculated according to the relations in Table 3 [5, 6]. Based on this composition, the specific heat of the reactants may be estimated as a function of temperature through correlations given in literature [5, 6, 10]. During compression, a mean polytropic exponent \( n_c \) is considered. This is calculated through an iterative converging scheme following the relations

\[
n_c = \gamma + \Delta \gamma
\]

\[
T_2 = T_1 r^{n_c - 1}
\]

\[
T_{av} = \frac{T_1 + T_2}{2}
\]

\[
\gamma = 1 + \frac{8.314}{c_{v,R@T_{av}}}
\]

where \( c_{v,R@T_{av}} \) is the reactant mixture specific heat in the average temperature \( T_{av} \) of the compression process, and \( \Delta \gamma \) is assumed equal to -0.02. For Diesel engines \( \Delta \gamma \) is typically in the range of -0.02 to +0.02 [7]. This take into consideration the overall heat transfer between the reactants and the cylinder walls during compression. Negative \( \Delta \gamma \) values correspond to heat transfer from the gases to the cylinder wall while positive values correspond to heat transfer in the opposite direction. After
convergence, temperature at the end of the compression stroke is calculated and pressure at the same state is given by

\[ p_2 = p_1 r^{n_c} \]

Compression is followed by fuel injection, combustion, and rapid increase in temperature and pressure. Cylinder gas composition after combustion is calculated according to the relations in Table 4 [4, 5]. Given the compositions of the reactants and products one can calculate the coefficient of molar multiplication \( \mu \) as

\[ \mu = \frac{N_p}{N_R} \]

where \( N_R, N_p \) are the total number of moles of the reactants and products, respectively.

| Reactant | moles/moles of O\(_2\) |
|----------|------------------------|
| Fuel \(4(1 - x_r)(1 + 2\kappa)/\lambda\) | \(\frac{M_f}{N_R}\) |
| CO\(_2\) | \(x_r\kappa/\lambda\) |
| H\(_2\)O\(_{\text{g}}\) | \(2x_r(1 - \kappa)/\lambda\) |
| CO | 0 |
| H\(_2\) | 0 |
| O\(_2\) | \(1 - x_r/\lambda\) |
| N\(_2\) | 3.773 |

Moles of reactant mixture: \(N_R\)

\[ (1 - x_r) \left[ \frac{4(1 + 2\kappa)/\lambda}{M_f} + 4.773 \right] + x_r N_p \]

| Table 4. Composition of the burned mixture in moles per mole of O\(_2\) reactant for \(\lambda \geq 1\). For a hydrocarbon fuel C\(_a\)H\(_b\) it is \(\kappa = 4/[4 + (b/a)]\). |
| Product | moles/moles of O\(_2\) |
|---------|------------------------|
| CO\(_2\) | \(\kappa/\lambda\) |
| H\(_2\)O\(_{\text{g}}\) | \(2(1 - \kappa)/\lambda\) |
| CO | 0 |
| H\(_2\) | 0 |
| O\(_2\) | \((\lambda - 1)/\lambda\) |
| N\(_2\) | 3.773 |

Moles of product mixture: \(N_p\)

\[ [(1 - \kappa)/\lambda] + 4.773 \]

3.1. Diesel fuel mode analysis

In the Diesel fuel mode of operation, the temperature of the combustion products \(T_3\) at the end of the combustion process may be calculated by the first law of thermodynamics, written as

\[ \zeta \frac{\text{LHV}}{N_R} + (c_{p,R@T_2} + 8.314\alpha)T_2 + 2270(\alpha - \mu) = \mu c_{p,P@T_3}T_3 \]

Here, \(T_2\) and \(T_3\) are set in Celsius degrees, \(\zeta\) is a factor of heat utilization, LHV is in kJ/kg, \(N_R\) is the number of moles of the reactants in kmol/kg of fuel, and \(\alpha\) is the combustion pressure ratio. For a
turbocharged Diesel engine at full load operation $\zeta$ is about 0.86 and $\alpha$ is about 1.5 [7]. The energy balance is solved iteratively to provide $T_3$ and the maximum pressure of the cycle is given as

$$p_3 = \alpha p_2$$

The ratio of pre-expansion is given by

$$\varepsilon = \frac{\mu T_3}{aT_2}$$

and the ratio of main expansion is

$$\delta = \frac{r}{\varepsilon}$$

During the expansion stroke, a mean polytropic exponent $n_e$ is considered. This is calculated through an iterative converging scheme according to the relations

$$n_e = \gamma + \Delta \gamma$$

$$T_4 = \frac{T_3}{\delta^{n_e-1}}$$

$$T_{av} = \frac{T_3 + T_4}{2}$$

$$\gamma = 1 + \frac{8.314}{c_{v,p@T_{av}}}$$

where $c_{v,p@T_{av}}$ is the product mixture specific heat in the average temperature $T_{av}$ of the expansion process, and $\Delta \gamma$ is assumed equal to -0.02. After convergence, temperature at the end of the expansion stroke is calculated and pressure at the same state is given by

$$p_4 = \frac{p_3}{\delta^{n_e}}$$

Now the temperature of the residual gases inside the cylinder may be calculated and compared to the value assumed before during the analysis. The new $T_r$ value is taken by

$$T_r = \frac{T_4}{3} \left( \frac{p_4}{p_r} \right)^{\frac{p_3}{3}}$$

and an iterative calculation can be applied until the convergence of $T_r$ value.

3.2. Natural gas fuel mode analysis

In the natural gas mode, the engine operates according to a lean-burn Otto combustion cycle. The pre-mixed lean gas–air mixture is ignited by the compression ignition of a small quantity of marine diesel pilot fuel injected into the main combustion space. A pilot injection of less than 1% of the normal liquid fuel quantity ensures very low NOx emissions in gas mode [2].
In an Otto cycle the temperature of the combustion products $T_3$ at the end of the combustion process may be calculated by the first law of thermodynamics, written as

$$\frac{\text{LHV}}{N_R} + c_{v,P@T_2}T_2 = \mu c_{v,P@T_3}T_3$$

where $\zeta$ is again a factor of heat utilization set at 0.86 and $T_2$, $T_3$ are set in Celsius degrees. The energy balance is solved iteratively and give $T_3$. For a gas engine this typically varies between 2200 and 2500K. The maximum pressure of the cycle is given as

$$p_3 = \frac{0.85p_2\mu T_3}{T_2}$$

During the expansion stroke, a mean polytropic exponent $n_e$ is considered. This is calculated through an iterative converging scheme according to the relations

$$n_e = \gamma + \Delta \gamma$$

$$T_4 = \frac{T_3}{r^{n_e-1}}$$

$$T_{av} = \frac{T_3 + T_4}{2}$$

$$\gamma = 1 + \frac{8.314}{c_{v,P@T_{av}}}$$

where $c_{v,P@T_{av}}$ is the product mixture specific heat in the average temperature $T_{av}$ of the expansion process, and $\Delta \gamma$ is assumed equal to -0.02. After convergence, temperature at the end of the expansion stroke is calculated and pressure at the same state is given by

$$p_4 = \frac{p_3}{r^{n_e}}$$

Then, the temperature of the residual gases inside the cylinder may be calculated again and iteration may be followed until the convergence of the $T_r$ value.

### 3.3. Cycle evaluation

Volumetric efficiency is given for both fuel modes as

$$\eta_v = \frac{T_C}{T_C + \Delta T} \frac{1}{r^{n_e}} (rp_1 - p_r)$$

Indicative mean effective pressure is given for diesel cycle by

$$imep = \frac{0.95p_C}{r - 1} \left[ \frac{a\epsilon}{n_e - 1} \left( 1 - \frac{1}{\delta^{n_e-1}} \right) - \frac{1}{n_c - 1} \left( 1 - \frac{1}{r^{n_c-1}} \right) + a(\epsilon - 1) \right]$$

and for natural gas (Otto) cycle by

$$imep = \frac{0.95p_C}{r - 1} \left[ \frac{a\epsilon}{n_e - 1} \left( 1 - \frac{1}{\delta^{n_e-1}} \right) - \frac{1}{n_c - 1} \left( 1 - \frac{1}{r^{n_c-1}} \right) + a(\epsilon - 1) \right]$$
\[ imep = \frac{0.96P_c}{r - 1} \left[ \frac{P_3}{P_2(n_e - 1)} \left( 1 - \frac{1}{r^{n_e-1}} \right) - \frac{1}{n_e - 1} \left( 1 - \frac{1}{r^{n_e-1}} \right) \right] \]

Friction mean effective pressure was calculated for both fuel cases by the correlation given by Kolchin and Demidov [7] for direct injection four stroke diesel engines as

\[ f_{mep} = 0.089 + 0.0118 \bar{u}_p \]

where \( \bar{u}_p = 2N_s \) is piston mean velocity at engine speed \( N \). Hence, brake mean effective pressure is given as,

\[ b_{mep} = imep - f_{mep} \]

Indicative power is given for both fuel modes in kW as

\[ P_I = \frac{imepV_DN}{n_R} \]

where \( imep \) is indicative mean effective pressure in MPa, \( V_D \) is overall engine displacement volume in liters, \( N \) is crankshaft rotational speed in 1/s and \( n_R \) is 2 for four stroke engines.

Indicative cycle efficiency is given for diesel cycle as

\[ \eta_I = \frac{\lambda \left( \frac{A}{F} \right)_{st} imep}{\rho_f \eta_v LHV} \]

where \( \lambda \) is lamda ratio, \((A/F)_{st}\) is stoichiometric air-fuel mass ratio, \(\rho_f\) is diesel fuel density, and LHV is diesel fuel lower heating value in MJ/kg. For the natural gas cycle \( \eta_I \) is given by

\[ \eta_I = \frac{\lambda \left( \frac{A}{F} \right)_{v, st} T_c imep}{\rho_f \eta_v LHV} \]

where \((A/F)_{v, st}\) is stoichiometric air-fuel molar or volume ratio and LHV is natural gas lower heating value in MJ/m³.

Mechanical efficiency is given by

\[ \eta_m = \frac{b_{mep}}{imep} \]

and brake mean effective pressure is given by

\[ P_b = \frac{b_{mep}V_DN}{n_R} \]

The thermal efficiency of each cycle is calculated as

\[ \eta_{th} = \eta_I \eta_m \]

Finally, brake specific fuel consumption is calculated by the expression
4. Results and Discussion

The cycle analysis of both fuel modes has been conducted according to the method described. Operation pressure and temperature conditions have been estimated in all salient cycle points. These are summarized in Table 5. As can be observed, both modes provide similar operation conditions and therefore are expected to give about the same power and efficiency characteristics. Since both fuels provide similar cycle operation the selection criteria of engine materials are about the same in both cases. However, one can observe that maximum engine temperature is 2430.93 K when operation takes place with natural gas. This temperature is about 150 K higher than the maximum temperature of the Diesel cycle.

| Parameter                                         | Diesel mode | NG mode | Real value |
|---------------------------------------------------|-------------|---------|------------|
| Ambient temperature, $T_0$ (K)                    | 293         |         |            |
| Ambient pressure, $p_0$ (MPa)                      | 0.1         |         |            |
| Turbocharger outlet temperature, $T_C$ (K)         | 517.63      |         |            |
| Turbocharger outlet pressure, $p_C$ (MPa)          | 0.55        |         |            |
| Fraction of residual gases, $x_r$                 | 0.0436      | 0.0425  |            |
| Temperature of residual gases, $T_r$ (K)          | 825         | 848     |            |
| Temperature at the end of intake stroke, $T_1$ (K) | 535.83      | 536.47  |            |
| Pressure at the end of intake stroke, $p_1$ (MPa) | 0.522       | 0.522   |            |
| Temperature at the beginning of combustion, $T_2$ (K) | 1332.66     | 1334.09 |            |
| Pressure at the beginning of combustion, $p_2$ (MPa) | 17.29       | 17.29   |            |
| Temperature at the end of combustion, $T_3$ (K)   | 2284.35     | 2430.93 |            |
| Pressure at the end of combustion, $p_3$ (MPa)    | 25.93       | 26.82   |            |
| Temperature at the end of expansion, $T_e$ (K)    | 1095.54     | 1128.05 |            |
| Pressure at the end of expansion, $p_e$ (MPa)     | 1.09        | 1.10    |            |

Results for power characteristics and engine efficiencies are given in Table 6 in comparison with data published for the actual V18 MAN 51/60DF engine. As shown, results for brake mean effective pressure, torque and brake power are in satisfying agreement with the values reported for the actual engine. Mean effective pressure is found around 1.9 MPa and brake power is estimated close to the reported value of 17550 kW. Overall, the method employed is found to be highly accurate and promising for implementation in future works.

| Parameter                          | Diesel mode | NG mode | Real value |
|------------------------------------|-------------|---------|------------|
| Engine speed                        | 500 rpm     |         |            |
| Fuel mass flow rate, $\dot{m}_f$ (kg/h) | 4030.66     | 3412.82 |            |
| Air mass flow rate, $\dot{m}_a$ (kg/h) | 116332.9    | 116336.2|            |
| Volumetric efficiency, $\eta_v$ (%) | 94.95       | 94.95   |            |
| Brake specific fuel consumption, bsfc (g/kWh) | 230.66     | 190.40  | 183.5 [8]  |
| Indicated mean effective pressure, imep (MPa) | 2.107       | 2.156   |            |
Friction mean effective pressure, $f_{mep}$ (MPa) | 0.207 | 0.207 |
Brake mean effective pressure, $b_{mep}$ (MPa) | 1.900 | 1.949 | 1.900$^\dagger$ |
Indicated thermal efficiency, $\eta_i$ (%) | 43.26 | 41.99 |
Mechanical efficiency, $\eta_m$ (%) | 90.17 | 90.40 |
Brake (real) thermal efficiency, $\eta_{th}$ (%) | 39.01 | 37.96 |
Torque, $\tau$ (kNm) | 333.7 | 342.3 | 335.2$^\dagger$ |
Indicated power, $P_i$ (kW) | 19376.81 | 19826.78 |
Friction power, $P_f$ (kW) | 1902.88 | 1902.88 |
Brake (real) power, $P_b$ (kW) | 17473.91 | 17923.89 | 17550$^\dagger$ |

$^\dagger$ for Diesel mode without attached pumps

5. Conclusions
This paper focuses on the thermal design and evaluation of an 18 cylinder four stroke turbocharged dual fuel reciprocating engine with the characteristics of the commercial V18 MAN 51/60DF model. Assuming constant engine speed at 500 rpm and both Diesel and natural gas modes of operation, an analytical method was presented in detail to calculate real cycle operation conditions, power characteristics and engine efficiencies. Results have shown that both fuel modes provide similar cycle operation with a maximum pressure of 26.82 MPa and a maximum temperature of 2430.93 K.

Brake mean effective pressure was estimated 1.900 MPa and 1.949 MPa at Diesel and natural gas operation, respectively. Engine power was estimated 17473.91 kW and 17923.89 kW at Diesel and natural gas operation, respectively. These results are found in close agreement with the actual values of $b_{mep} = 1900$ MPa and $P_b = 17550$ kW which are reported by the MAN manufacturer. Overall, the applied method was found to be highly accurate and promising for future application in combustion engine research.

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