DEVELOPMENT OF A MARINE DIESEL ENGINE MEAN-VALUE MODEL FOR HOLISTIC SHIP ENERGY MODELLING

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Abstract
Ships are complex energy structures. The energy flows and transformations are diverse and numerous. In order to optimize a ship’s fuel consumption, one has first to correctly model her. This has successfully been done with the Bureau Veritas SEECAT modelling platform but with some limitations [1][2]. The development of a new diesel engine model should bring more accuracy, particularly outside the “propeller curve”, complete thermal balance calculations and allow future developments of ship exergy analysis. This paper will present the work done in the development of a marine diesel engine model. The model shall rely only on public data available and will follow the mean-value approach. It shall achieve fast time performance and good accuracy for fuel consumption and thermal balance prediction.

The best type of modelling approach and data pre-processing techniques will be discussed. The main architecture and equations of the model will be presented. Finally, results will be presented for model validation discussion.

Keywords: Marine diesel engine; mean-value approach; energy modelling; thermal balance prediction; Modelica

N.B.: In this paper, the method and the equations presented are generic: they can be applied to any marine turbocharged diesel engine. Nevertheless a specific engine has been chosen for modelling, it will be called engine A. This engine is a very large, two-stroke, slow speed, turbocharged, marine diesel engine (see Table 3 for more details). All numerical values produced below apply to this particular engine.

Introduction
Holistic ship energy modelling is an emerging topic in the maritime industry and academic world. New tools have appeared on the market [3][4]. The Bureau Veritas SEECAT tool is one of them [1]. It models and simulates all the energy flows and transformations on board ships. The complex nature of the ship is handled by following a systemic approach [5]. A first simple approach based on power conversion and transport has already been used. It showed good prediction capabilities. Nevertheless it did not make exergy analysis possible [6]. To answer this limitation, a new approach based on mass and specific enthalpy flow is developed [2]. The engine model presented in this paper is the first step of this new approach.

The purpose of this new model is to represent the behaviour of two or four stroke marine diesel engines. This model will be written in the Modelica language [7] using the simulation software SimulationX*. This will allow full compatibility with existing SEECAT library and provide full independence from market and non-open source software solutions. It must be

* http://www.simulationx.com/
able to return the fuel consumption as well as the complete thermal balance all over the engine’s operational field. Other ship energy modelling tools rely on complex and proprietary engine models fed by manufacturers’ private data [3] or simple lookup tables also fed by manufacturers’ private data. Others rely on mean value model such as the one developed in this paper [4] (see section 2 - for engine modelling state of the art).

This model will be calibrated using only public data, that is to say, data provided by engine manufacturers. Now, it so happens that these data are most of the time provided only along the propeller curve. This propeller curve corresponds to heavy running conditions (fouled hull, nominal displacement, heavy seas). However, one objective of the energy simulations is to represent real-life operating conditions, where the displacement is not always nominal and with variable sea state conditions. Moreover, it must be possible to model variable pitch propellers and even investigate the energy saving potentiality of replacing the propeller. Real-life conditions, variable pitch propellers and changing the propeller implies different propeller curves and hence engine operating point outside this propeller curve. The main goal of this engine model is therefore to extrapolate outside this propeller curve.

1 - Data available

The main marine diesel engine manufacturers provide documentations with engine data and also online tools. The typical set of data available is: specific fuel consumption, exhaust gas mass flow, exhaust gas temperature after turbine, scavenge air mass flow, scavenge air pressure, scavenge air temperature after and before scavenge air cooler (SAC), SAC heat, jacket water cooler heat and main lubrication oil heat. As previously mentioned, this set of data is usually given along the propeller curve. For engine A, this set of data is also available along the controllable pitch propeller (CPP) curve (i.e. with constant engine speed). These two sets will be used for identification.

2 - Modelling diesel engines

Diesel engine modelling is a vast field of research. One can distinguish five main types of modelling approach. In order of increasing mathematical complexity these are: performance maps, transfer functions, mean value models [8], filling & emptying [9] and multi-dimensional models. Their pros and cons are summarised in Table 1.

All these approaches serve different purposes but in fact only very few models were found with the intent of thermal balance calculation [10]. Given the objectives and constraints of the model, a mean value approach has been chosen as it seemed the best solution. The limited available data do not make it possible to use a filling and emptying approach. Transfer function models cannot extrapolate outside the propeller curve. And finally performance maps are simply not available. Mean value approach is hence the only approach possible given the data set available. But even with all the desired data available, the mean value approach would have remained a good choice as its time scale (several engine cycles) is well suited to holistic ship energy modelling.
Table 1: Comparison of specifications and requirements between various diesel engines modelling approaches. The “insight” criterion represents the model’s capacity to give adequate understanding into the various engine processes by identifying the effect of key operating parameters. The “adaptability” criterion represents the model’s capacity to easily adapt to different engines/operating conditions. This table is adapted from Rakopoulos et Giakoumis [11]

<table>
<thead>
<tr>
<th>Performance map</th>
<th>Transfer function</th>
<th>Mean Value</th>
<th>Filling &amp; Emptying</th>
<th>Multi-dimensional</th>
</tr>
</thead>
<tbody>
<tr>
<td>Linear</td>
<td>No</td>
<td>Yes</td>
<td>Quasi</td>
<td>No</td>
</tr>
<tr>
<td>Transient</td>
<td>Quasi-static</td>
<td>Quasi-static</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Math Complexity</td>
<td>Very Low</td>
<td>Very Low</td>
<td>Low</td>
<td>High</td>
</tr>
<tr>
<td>Insight</td>
<td>Very Low</td>
<td>Very Low</td>
<td>Low</td>
<td>Adequate</td>
</tr>
<tr>
<td>Computer time</td>
<td>Negligible</td>
<td>Negligible</td>
<td>Limited</td>
<td>Very large</td>
</tr>
<tr>
<td>Adaptability</td>
<td>Very Low</td>
<td>Medium</td>
<td>Low</td>
<td>Medium</td>
</tr>
</tbody>
</table>

3 - Data analysis and cleaning
The mean value model relies mostly on physical equations, but also partly on empirical equations. These equations are built using the manufacturer’s data. These data must be verified and eventually corrected before usage. Many engine manufacturers’ datasets have been studied. Their thermal balances were calculated using the equations of Table 2 and considering the system boundary and flows of Figure 1. In each case the data showed inconsistent thermal balance (usually more output power than input). Some data “cleaning” process was necessary before starting modelling. This pre-processing work was an important phase of the development process and guaranteed that the values were coherent, especially regarding the first law of thermodynamics.

Table 2: Engine thermal balance: equations of output and input power flows [12]

<table>
<thead>
<tr>
<th>Input powers</th>
<th>Output powers</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P_{\text{combustion}} = \dot{m}_{\text{fuel}} \cdot LHV )</td>
<td>( P_{\text{crankshaft}} )</td>
</tr>
<tr>
<td>( \dot{Q}<em>{\text{air}} = \dot{m}</em>{\text{air}} \cdot \Delta h_{\text{air}} = \dot{m}<em>{\text{air}} \cdot c</em>{\text{air}} \cdot \Delta T_{\text{air}} )</td>
<td>( \dot{Q}<em>{\text{SAC}} = \dot{m}</em>{\text{air}} \cdot \Delta h_{\text{air}} = \dot{m}<em>{\text{air}} \cdot c</em>{\text{air}} \cdot \Delta T_{\text{air}} )</td>
</tr>
<tr>
<td>( \dot{Q}<em>{\text{fuel}} = \dot{m}</em>{\text{fuel}} \cdot \Delta h_{\text{fuel}} = \dot{m}<em>{\text{fuel}} \cdot c</em>{\text{fuel}} \cdot \Delta T_{\text{fuel}} )</td>
<td>( \dot{Q}<em>{\text{exhaust}} = (\dot{m}</em>{\text{air}} + \dot{m}<em>{\text{fuel}}) \cdot \Delta h</em>{\text{exhaust}} )</td>
</tr>
<tr>
<td>Thermal power of cooling water</td>
<td>( \dot{Q}<em>{\text{exhaust}} = (\dot{m}</em>{\text{exhaust}}) \cdot c_{\text{exhaust}} \cdot \Delta T_{\text{exhaust}} )</td>
</tr>
<tr>
<td>( \dot{Q}<em>{\text{water}} = \dot{m}</em>{\text{water}} \cdot c_{\text{water}} \cdot \Delta T_{\text{water}} )</td>
<td></td>
</tr>
<tr>
<td>Thermal power of lubricating oil</td>
<td>( \dot{Q}<em>{\text{oil}} = \dot{m}</em>{\text{oil}} \cdot c_{\text{oil}} \cdot \Delta T_{\text{oil}} )</td>
</tr>
<tr>
<td>( \dot{Q}_{\text{radiation}} )</td>
<td>3</td>
</tr>
</tbody>
</table>
4 - Presentation of the model

The model is coded in Modelica. It is composed of four main components: the compressor, the scavenge air cooler (SAC), the engine cylinders and the turbine. The components are organized and linked as shown in Figure 2. The main links represent either fresh air or exhaust gases and transport the gas pressure, temperature, mass flow and excess air ratio.

Since modelling the transient response of the engine, that adds complexity and required input data difficult to obtain, was not the primary focus, a quasi-static model has been developed.
(no differential equations). As a consequence of the quasi-static approach the input mass flow of each component equals its output mass flow:

\[ \dot{m}_{in} = \dot{m}_{out} \]  

(1)  

Moreover, except for inside the cylinders, the fresh air and exhaust gases are considered as perfect gases and therefore:

\[ \Delta h = c_p \cdot \Delta T = c_p \cdot (T - T_{ref}) \]  

(2)  

The specific heat capacity is calculated using the Keenan and Kayes empirical formulas [13]. The turbocharger is first of all represented by its shaft mechanical efficiency:

\[ \eta = \frac{P_{compressor}}{P_{turbine}} \]  

(3)  

This efficiency is identified using the manufacturer’s data and calculating the shaft friction losses. In a first approach, this efficiency is considered constant. The compressor and turbine are both considered adiabatic. The quality of the compression is characterized by the pumps isentropic efficiency:

\[ \eta = \frac{T_{in}}{T_{out} - T_{in}} \cdot \left( \frac{\gamma - 1}{\gamma} \right) \]  

(4)  

This efficiency is identified using the manufacturer’s data. For engine A, values are found to vary between 80.1% and 81.6% for loads higher than 30%. Given this narrow range of variation and the lack of further information concerning the compressor, based on the work of Hendricks [8] and Jankovic [14], it has been decided to consider a constant compressor efficiency throughout the engine operational range. The compressor pressure ratio usually varies with the air mass flow and speed. The speed is not available and therefore a polynomial law, function of the mass flow only has been identified with the manufacturer’s data:

\[ pr = c_1 \cdot m^2 + c_2 \cdot m + c_3 \]  

(5)  

It is often possible to access turbine and compressor maps. If so, these maps can be used instead of the previous equation. The turbine power is calculated thanks to eq. (3). This power makes it possible to calculate the output temperature:

\[ P_{turbine} = \dot{m} \times \Delta h = \dot{m} \times \Delta T \]  

(6)  

The output temperature and turbine isentropic efficiency makes it possible to calculate the pressure ratio:

\[ T_{out} = T_{in} \cdot \left( 1 - \eta \cdot \left( 1 - pr \right) \right) \]  

(7)  

In a first approximation, the turbine isentropic efficiency is chosen identical to the compressor one. The scavenge air cooler is considered to have perfect energy efficiency (no heat loss) and therefore:

\[ \dot{Q}_{SAC} = \dot{m} \times \Delta h \]  

(8)  

The fresh air output temperature is calculated using the SAC effectiveness:

\[ T_{out} = T_{in} \cdot (1 - \varepsilon) + \varepsilon \cdot T_{coolant} \]  

(9)  

Heat exchangers effectiveness usually varies with temperature difference and mass flows. In a first approach, the effectiveness will be estimated by a linear function of the air mass flow:

\[ \varepsilon = s_1 \times \dot{m}_{air} + s_2 \]  

(10)  

For loads below 40%, the effectiveness is considered constant and equal to 95%.
The engine cylinders are the most important parts of the engine but at the same time are the most complicated ones to model because of physical phenomena such as combustion and heat transfer. In the mean value approach several phenomena are described by mean value empirical equations. In this case four empirical equations are used to calculate the air mass flow, the engine global efficiency, the jacket water thermal cooling power and the lubrication oil thermal cooling power. These equations are all identified from the manufacturer’s data. A specific tool is used to calculate polynomial interpolations. These interpolations are well adapted to engine A. For a different engine model, the equation form might change. Several mean value models use fluid flow equations such as the equation of Barré de Saint-Venant [15][16] to determine the air mass flow. This approach considers the cylinders as an expansion valve [17] but requires geometric dimensions which are generally not available. By analysing the manufacturer’s data, the air mass flow has been found to be mainly a function of torque and engine speed. The following interpolation has been found to fit correctly the corrected manufacturer’s points:

\[ \dot{m}_{\text{air}} = a_1 + a_2 \cdot \omega + a_3 \cdot \tau + a_4 \cdot \omega \cdot \tau + a_5 \cdot \tau^2 \] (11)

All mean value models found in the literature use empirical equations to assess the global engine efficiency. Different equation forms exist. The following equation has been found to fit best:

\[ \eta = b_1 + b_2 \cdot \omega + b_3 \cdot \tau + b_4 \cdot \omega^2 + b_5 \cdot \omega \cdot \tau + b_6 \cdot \tau^2 \] (12)

The efficiency is used to calculate the fuel mass flow:

\[ \eta_{\text{global}} = \frac{W}{m_f \cdot PCI} \] (13)

And the fuel mass flow calculates the air-fuel equivalence ratio, which is used for heat capacity calculations:

\[ \lambda = \frac{\dot{m}_{\text{air}}}{\dot{m}_f \cdot C_{\text{stoich}}} \] (14)

Engine cooling powers are rarely calculated in engine modelling. Some models calculate the heat transfer through cylinder walls and the friction losses. But these calculations are too complex for a mean value approach. Concerning the jacket water cooling power, a polynomial with engine speed to the power two and engine torque to the power one was found to fit best the corrected manufacturer’s data:

\[ \dot{Q}_{\text{water}} = c_1 + c_2 \cdot \omega + c_3 \cdot \tau + c_4 \cdot \omega^2 + c_5 \cdot \omega \cdot \tau \] (15)

Concerning the engine lubrication oil cooling, a polynomial with engine speed and torque to the power two has been found to fit best the corrected manufacturer’s data:

\[ \dot{Q}_{\text{oil}} = d_1 + d_2 \cdot \omega + d_3 \cdot \tau + d_4 \cdot \omega^2 + d_5 \cdot \omega \cdot \tau + d_6 \cdot \tau^2 \] (16)

Finally, all engine thermal and mechanical powers are known except the exhaust power. The exhaust power can hence be deduced from the first law of thermodynamics:

\[ P_{\text{combustion}} + \dot{Q}_{\text{fuel}} + \dot{Q}_{\text{air}} = P_{\text{crankshaft}} + \dot{Q}_{\text{cooling}} + \dot{Q}_{\text{exhaust}} + \dot{Q}_{\text{radiation}} \] (17)

Moreover, following the mass conservation law, the mean exhaust mass flow \( \dot{m}_{\text{exh}} \) is equal to:

\[ \dot{m}_{\text{exh}} = \dot{m}_{\text{air}} + \dot{m}_{\text{fuel}} \] (18)

Finally, the exhaust receiver temperature \( T_{ER} \) is deduced from the exhaust gas thermal power, as:
The method and the equations described above are used to build the mean value model of engine A. The models are coded in the Modelica language and executed with SimulationX.

5 - Results and discussion
Simulations were run to check the validity of the model. More than thousand points were plotted along the propeller curve and compared to the corrected manufacturer’s data (Figure 3). The initial manufacturer’s data are also displayed to indicate the corrections applied.

As it can be observed, for most variables, the data that were used to identify empirical equations are correctly predicted, that validates the identification process. Concerning the jacket water cooling and lub. oil cooling powers, the accuracy is very good. Concerning the brake specific fuel consumption (BSFC), the SAC heat and the exhaust mass flow, the accuracy is considered as very satisfactory with relative differences below 5%. The manufacturer’s exhaust mass flow curve shows a small “bump” around 30% which is supposed to be caused by an auxiliary blower stopping. As auxiliary blowers are not represented in the current model, this bump is not predicted. This bump can also be observed on the exhaust gas temperature curve. As far as exhaust gas temperature is concerned, for loads above 50% the average temperature calculated by the model, 225.5 °C, is very close to the corrected data set, 222.3 °C and the maximum error is under 8 °C. For loads under 50%, discrepancies are higher but remain under 22 °C.

As a conclusion, the overall accuracy of the model along the propeller curve is considered as good, which is very satisfactory for our intended purpose, given the limited data and
knowledge that were needed (and available) as input. The same analysis has been made along the controllable pitch propeller curve and has led to similar conclusions.

In order to investigate the engine model response outside these two curves in the engine diagram, The good model accuracy along the propeller and the CPP curve increases confidence for values between and outside these two curves.

Figure 4: Contour maps built using the sensitivity analysis data

Of course, a definitive validation of the model for values outside the propeller and the CPP curves is not strictly possible. Nevertheless a sensitivity analysis can be made and has been carried out in order to generally study the response of the model to a variation of its inputs and notably assess its robustness and check for “strange” values and variations. 17 simulations were run at constant speed, from 110% of SMCR speed to 30%. For each simulation, the engine load was changed, from a maximum value on the “overload curve” down to 5% load (below the model usually stops converging). In total, more than 1 400 engine operating points were simulated. For each point, the brake specific fuel consumption (BSFC), the air cooler heat, the engine jacket and lubrication oil cooling heat as well as the exhaust mass flow and temperature (after turbine), were recorded. These values are presented as contour maps in Figure 4.

The jacket water cooling, lub. oil cooling, SAC cooling and exhaust mass flow signals show similar trends. They vary mainly with engine load. The jacket water and lub. oil cooling show a small dependency towards engine speed, which is reasonable. The BSFC map shows a traditional “mussel” shape with a minimum at approximately 70% load and 90% speed which is very close to propeller curve. The exhaust gas temperature appears to be a complex function of both speed and power. Nevertheless, most of the domain corresponds to

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*The overload curve corresponds to a technical limitation for engine damage prevention. Engines cannot exceed this limit.*
temperatures between 200 °C and 240 °C. Moreover, at constant speed, the temperature appears to be mainly increasing with power which is sensible. Qualitatively speaking, no abnormal responses are observed in the sensitivity analysis range. On this basis, and given the purpose of this model, the model can be considered as robust. A proper validation of the model results, outside the two propeller curves used for identification, would require comparison with reference data, and engine manufacturers will be approached for this. The validity of the points outside the two curves only depended on the structure and the quality of the interpolation equations as well as on the pertinence of the different assumptions (e.g.: perfect gas hypothesis, quasi-static approach, etc.). Most assumptions are traditional assumptions in mean value and zero dimensional modelling and their pertinence have already been discussed above. Concerning interpolations, their structure have been chosen to fit best physics or literature equations and manufacturer’s data. The adjusted R-square indicators obtained for the four three-dimensional interpolations performed were larger than 0.975, indicating very good interpolation quality. Moreover, this model is time efficient. On a laptop running on a 2.80 GHz dual core processor, calculating 120 engine operating points along the propeller curve takes approximately 1 second.

**Conclusion**

In this paper, a computer model for marine diesel engines has been presented. The modelling approach is based on a quasi-static mean value approach. Physical and empirical equations are used to describe the behaviour of the engine. The empirical equations are calibrated with manufacturer’s data that are usually available for a specific engine but appeared to require a thorough verification and corrections based on the first law of Thermodynamics. The developed model computes the complete thermal balance of an engine for any engine operating point. Several observations can be made:

- The mean value approach is not initially meant for thermal balance calculations. In fact, all modelling approaches included, only very few papers were found in the literature for thermal balance calculations. The rare papers found were fairly simple and relied heavily on empirical data. The mean value approach introduces more physical basis and insight and more flexibility.
- This model is fast to execute and its equations remain accessible.
- It is reminded that given the quantity and the quality of the data available, the mean value modelling is considered by the authors as the best compromise for the intended purpose of holistic ship energy modelling. If more data is available (for example turbocharger maps), the model components can easily be changed, keeping the same structure but modifying the equations, hence producing a more complex but more accurate model.

The model has nevertheless some limitations:

- The model does not yet take into account variations of ambient air temperature. The model is hence fixed to the reference temperature of the manufacturer’s data (20 °C for engine A).
- A propeller and a CPP curve are necessary to build most of the empirical equations. The CPP curve is not always available freely.

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This statistic measures how good an interpolation fit is in predicting the variation of the data. It can take on any value less than or equal to 1, with a value closer to 1 indicating a better fit. See the exact definition at: [http://www.mathworks.co.uk/help/curvefit/evaluating-goodness-of-fit.html](http://www.mathworks.co.uk/help/curvefit/evaluating-goodness-of-fit.html).

1 Laptop running on windows 7 64 bits, equipped with 4 GB of RAM and an Intel Core i5-3360M processor running at 2.80GHz.
• The accuracy of the model remains questionable as the results have not been quantitatively validated. However, as long as the engine is operated “close” to the propeller and CPP curve, the errors should remain limited. Along these two curves the accuracy of the BSFC, jacket and lub. oil cooling power, SAC cooling power, exhaust mass flow and temperature results are good and depend mainly on the empirical interpolations. Moreover the variations between the two curves are small, hence reducing further the risk of error. Finally, the output values calculated are coherent with literature and common knowledge.

• The model contains no differential equations and therefore is not a “true” dynamic model. It is nevertheless a quasi-static model. This limitation should have small consequences as long as the ship speed variations remain limited. This limitation should nevertheless be kept in mind when writing a ship operational profile model.

To overcome these limitations and expand the possibilities of the model, several developments are considered:

• Implement influence of ambient air temperature on engine
• Development of new components and functionalities is foreseen:
  ▪ by-passes;
  ▪ exhaust gas recirculation system;
  ▪ power turbines;
  ▪ auxiliary blowers;
  ▪ turbo-compound.

• The model shall also be adapted to describe dual-fuel and gas engines.

Finally, beyond the technical aspects, the work presented in this paper illustrates a powerful approach for modelling diesel engines in the purpose of thermal balance prediction. Its adaptability to holistic ship energy and exergy modelling will be presented in a future paper.

<table>
<thead>
<tr>
<th>Table 3: Engine A details</th>
</tr>
</thead>
<tbody>
<tr>
<td>SMCR speed</td>
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<tr>
<td>SMCR power</td>
</tr>
<tr>
<td>Cylinder diameter</td>
</tr>
<tr>
<td>Cycle type</td>
</tr>
<tr>
<td>Number of cylinders</td>
</tr>
<tr>
<td>Mean effective pressure</td>
</tr>
</tbody>
</table>
References


Nomenclature

\( a \): Empirical coefficient
\( b \): Empirical coefficient
\( c \): Specific heat capacity \((J/(kg.K))\) or empirical coefficient
\( d \): Empirical coefficient
\( c_p \): Specific heat capacity at constant pressure \((J/(kg.K))\)
\( h \): Specific enthalpy \((J/kg)\)
\( \dot{m} \): Mass flow \((kg/s)\)
\( p \): Pressure \((Pa)\)
\( P \): Power \((W)\)
\( pr \): Pressure ratio - \( pr = p_{out}/p_{in} \)
\( \dot{Q} \): Heat transfer rate \((W)\)
\( s \): Empirical coefficient
\( T \): Temperature \((K)\)
\( \dot{W} \): Work rate \((W)\)

Greek letters:

\( \gamma \): Adiabatic index
\( \varepsilon \): Effectiveness
\( \eta \): Efficiency
\( \lambda \): Air-fuel equivalence ratio or air excess ratio
\( \tau \): Torque \((N \cdot m)\)
\( \omega \): Rotary velocity \((rad/s)\)

Acronyms and abbreviations:

BSFC : Brake specific fuel consumption \((g/kWH)\)
CPP : Controllable pitch propeller
ER : Exhaust receiver
HT : High temperature
LHV : Lower heating value \((J/kg)\)
LT : Low temperature
Lub. oil : Lubrication oil
\( ref \) : Reference
SAC : Scavenge air cooler
SEECAT : Ship energy efficiency calculation and analysis tool
SMCR : Specified maximum continuous rating