In accordance with the specified requirements, there is an active transition to fuels with ultra-low sulfur content. The use of these fuels in marine diesel engines is associated with a number of difficulties, because these engines are usually designed for operation on fuels with high viscosity and lubricity. The viscosity values for ultra-low sulfur fuels are close to the permitted minimums for diesel engines at normal engine room temperature. The greatest difficulties occur when the viscosity values fall below the specific range when the fuel temperature before the engine increases. For reliable operation of the engine, the fuel temperature must be constantly maintained at a range in which the fuel viscosity will have the required values. For this purpose the engine design provides presence of fuel cooling system with a water cooler and a chiller for heat removal from water. In this paper the efficiency of chiller refrigeration plant was investigated using new perspective refrigerant mixtures R125/R290 and R134a/R290 as working fluids in comparison with basic R134a and R22. The values of composition for both mixtures are chosen such that they are closest to the azeotrope. It is possible for azeotrope mixtures to minimize the temperature difference between heat exchanging medias in condenser and evaporator of refrigeration plant. During the investigation it was revealed that the values of refrigeration coefficient of refrigerating plant when using mixtures as working fluids were somewhat lower when operating on R134a and R22. But the values of volumetric refrigeration capacity with mixtures as working fluids are significantly higher.

**Key words:** marine diesels; fuel viscosity; chiller; refrigeration plant; mixtures R125/R290 and R134a/R290; energetic efficiency; specific refrigeration capacity.

**Introduction**

Correct simulation of the turbocharger parameters is an urgent requirement for the internal combustion engine (ICE) operating cycle simulation software. The most common way to obey this condition – is to use specially prepared maps for compressor and turbine performance.

It is vital to provide fast and stable calculations and to consider important peculiarities of the turbocharger operation: compressor choke, rotating stall and surge, turbine partial gas supplying configurations, gas flow inertia, etc. For the transient engine operation the effect of thermal inertia is also significant [1]. Among mentioned cases, the compressor surge problem is, probably, the most complicated.

This paper aims to develop adaptive adjustable centrifugal compressor maps treatment to provide both stable and correct calculations.

**Literature review**

The problem of the correct centrifugal compressor behavior simulation has been discussed widely. The leading ICE simulation software, such as Ricardo Wave, AVL Boost, GT-Power, use the compressor maps to define the compressor parameters during calculations. These maps are obtained from the experimental tests and should pass the preprocessing procedures, which include interpolation and extrapolation routines, before been used for calculations. Experimental compressor maps have to be extrapolated to the choke region, compressor zero speed region and have
to provide an ability of rotating stall and surge simulation. Many papers suggest different approaches of how to consider compressor operation in the surge mode [2-4]. Most of papers and algorithms are based on the Moore-Greitzer model for the compressor unsteady behavior [10, 11]. The compressor surge consideration always makes calculations more unstable, sensitive to the input data and requires much longer time for execution. However, it’s always unnecessary to consider accurately compressor rotating stall and surge during calculations. In some cases – for example on the turbocharger matching procedure, or when the calculations are focused on other aspects – it’s much more convenient to ignore this mode and to get significantly faster and more stable calculations.

Centrifugal compressor performance maps
treatment

The ICE operating cycle synthesis generally is implemented as a numerical solution of the set of differential equations.

Blitz-PRO – the online computation service for ICE static and transient operation simulation – utilizes combination of the quasi-steady (single and two-zone) and one-dimensional unsteady approaches to describe the processes in interrelated open thermodynamic systems (OTS), which are parts of the general thermodynamic system – the engine [5]. The equations sets include the first law of thermodynamics, mass balance and state differential equations for quasi-steady OTS and energy, pulse and mass conservation differential equations for one-dimensional OTS.

The solution for the combination of equations sets for all interrelated OTS is numerical: the user can choose between the simple Euler and second order implicit Runge-Kutta method [6]. Numerical solution, obviously, requires the set of initial conditions and then the iterative calculations to obey the target level of calculations accuracy. The calculations accuracy is defined as the relative difference in densities of the working gas at the operating cycle start and finish. The number of such iterative approaches generally is between 20…300 iterations.

Fig. 1a shows the deviation of averaged by operating cycle values of compressor pressure increase ratio $\Pi_{\text{comp}}$, air mass flow $G_{\text{air}}$ and turbocharger speed $n_{TC}$ by the iteration number. It is to be noted, that the variations of instantaneous compressor parameters could also be significant as it is illustrated in Fig. 1b.

These examples show, that while the calculations are in progress, there is a possible situation, when turbocharger operates in a surge or choke region (see Fig. 2), and the corresponding routines should provide fast and stable calculations under these conditions.

Simulation of two-stroke ICE, marine low-speed ICE particularly, is one of the most challenging in terms of turbocharger performance calculations, as the sensitivity of two-stroke engines to the gas exchange
processes is well known. This paper considers the simulation of the MAN 8G70ME-C operating cycle as an example of the possible turbocharger performance maps treatment, focusing on the centrifugal compressor maps.

The engine is equipped with two MAN TCA77 turbochargers, which work in parallel. The extrapolated and interpolated flow and efficiency maps of the turbocharger, prepared according to [7] are presented on Fig. 3. The initial data of compressor performance was taken from the manufacturer’s official project guide [8]. The maps, presented on Fig. 3, are to be considered as the emulation of the real turbocharger performance maps and are not to be considered as the actual turbochargers characteristics.

As it is clear from Fig. 3 the experimental compressor performance maps are extended up to the choke limit as well as to the regions of reverse air flow and the pressure increase ratio below 1 ($\Pi_{cwrpr} < 1$).

Let's consider in more detail the constant-speed line extrapolation and treatment as it is shown on Fig. 4. The constant-speed line consists of 3 parts: positive air flow part (line $1$), negative air flow part (line $2$) and the conjunction part (line $BD$ on Fig. 4). The conjunction part describes the compressor behavior under unstable conditions. The positive flow line is generated as the extrapolation of the experimental flow lines to the choke limit, while the negative flow and conjunction parts are calculated according to the Moore-Greitzer method [10]. Point $D$ is given as following [12]:

$$\Pi_{cwrpr}^D = \left[1 + \frac{(\omega_{cwrpr}/2)^2}{c_{p,T_0}} \left(\frac{D_2^2}{D_{1,av}} - 1\right)\right] \cdot \frac{k}{\kappa - 1} \cdot C_{ref}^D = 0,$$

where $\omega_{cwrpr}$ – compressor speed, $T_0$ – air temperature at the compressor inlet; $D_2$, $D_{1,av}$ – compressor wheel exducer and average inducer diameters.

The surge process modeling approach can be described as follows. At the periods of time when the intake receiver pressure is smaller than the maximum pressure at point $B$, the right brunch of map is used, so the compressor pumps air into the intake receiver. If receiver restriction is too high for given compressor speed, compressor isn’t able to operate at stable mode, so it runs into surge mode with oscillations of air flow and charge air pressure. When the intake receiver pressure exceeds the maximum pressure at point $A$, the routine switches to the left brunch of the map, using conjunction part $BD$, and the air flow reverses its direction (line $CD$). The left brunch is used until the intake receiver pressure reaches the minimal value at point $D$, and then the routine switches back to right brunch through the conjunction $BD$, creating the surge loop. The described picture is common to the full-scale surge of the compressor.
For the mild surge simulation it is proposed to scale the surge loop ABCDA applying the multiplier \( \mu_{G_{s}} < 1.0 \). Scaling gives new loop BC’D’A’, which gives reasonably smaller oscillations of the air flow and pressure increase ratio.

\[
G_{\text{ref}} = G_{\text{ref}}^B - \mu_{G_{s}}(G_{\text{ref}}^B - G_{\text{ref}}) ; \\
\Pi_{\text{cmp}} = \Pi_{\text{cmp}}^B - \mu_{G_{s}}(\Pi_{\text{cmp}}^B - \Pi_{\text{cmp}}) .
\]

If the computation tasks are not related to the turbocharger surge problems, it is proposed to turn the calculations into the “stable” mode, applying the alternative extension of the constant-speed line. The line 3 on Fig. 4 illustrates the basic approach, when the constant-speed line is extended to the air flow rates smaller than the flow at the point B with the following equation:

\[
G_{\text{ref}} = \left[ \frac{G_{\text{ref}}^B}{2(\Pi_{\text{cmp}} - \Pi_{\text{cmp}}^B)} \right] .
\]

Using the “stable” mode provides much faster and stable calculations as the compressor surge never occurs during simulation process. It helps to match the turbocharger faster, or to provide brief calculations neglecting the surge phenomena, albeit the researcher should always be aware about possible surge.

Table 1 and Fig. 5 display the results of calculations comparison for different surge simulation modes. The engine operates at 12140 kW, 82 rpm by the propeller law curve. It is clear, that the “stable” mode provides the fastest calculations, which take only 36 iterations to obey the set level of accuracy. For the full-scale surge mode (\( \mu_{G_{s}} = 1.0 \)) even 351 iterations are not enough to find the solution – the values of achieved calculations errors are much beyond the set level. The mild surge adjustment mode gives the results, which are very close to the “stable” mode – calculations took 40 iterations and the simulated engine performance is pretty similar. The examination of the Fig. 5 shows that during the mild surge mode still there are oscillations of the compressor mass flow rate and efficiency comparing to the “stable” mode, while the intake receiver pressure diagrams are very close. The full-scale surge mode has the high level of instability, so the oscillations of compressor performance are significant.

How to deal with the calculations results? Fig. 6 shows scaled compressor map with the calculated operation point (for the mild surge and “stable” modes). It is clear, that the operating point is far enough from the surge line and lies on the constant-speed operating curve, which indicates the stable compressor operation. It could be reasonable assumed, that the calculations results are rather correct and the inability of the routine to find proper solution under the full-scale mode is explained mostly by numerical issues.

From the above it seems to be reasonable to recommend calculations in “stable” mode for initial model setup and then turn into the surge mode to check if it could occur under current conditions. These recommendations could be useful for turbocharger continuous monitoring system application [13].

| Table 1. Results of calculations |
|----------------------------------|
| Surge simulation set-up, \( \mu_{G_{s}} \) | “stable” mode | 0.6 | 1.0 |
| Calculations accuracy estimation (target < 0.05 %) |  |
| In-cylinder density accuracy \( \delta_{\rho_{\text{cyd}}} \), % | 0.0068 | 0.0009 | 9.78 |
| Receiver density accuracy \( \delta_{\rho_{\text{rcv}}} \), % | 0.0127 | 0.0014 | 0.604 |
| Exhaust manifold density accuracy \( \delta_{\rho_{\text{em}}} \), % | 0.0041 | 0.0305 | 2.145 |
| Turbocharger power accuracy \( \delta_{P_{\text{tc}}} \), % | 0.0337 | -0.22 | -4.82 |
| Calculated parameters (extraction) |  |
| Brake power \( p_{\text{b}} \), kW | 12139 | 12138 | 12369 |
| Air excess ratio \( \alpha \) | 2.2295 | 2.2293 | 2.2745 |
| Injected fuel \( q_{\text{fuel}} \), g | 71.914 | 71.913 | 72.930 |
| Volumetric efficiency \( \eta_{\text{v}} \), % | 73.72 | 73.71 | 73.05 |
| Scavenging factor, \( \phi_{\text{scav}} \) | 1.709 | 1.71 | 1.866 |
| Receiver average pressure \( p_{\text{av}} \), kPa | 219.8 | 219.8 | 230.7 |
| Compressor air mass flow, \( G_{\text{inst}} \), kg/s | 32.23 | 32.24 | 34.26 |
| Compressor adiabatic efficiency \( \eta_{\text{ad,cmpr}} \), % | 88.35 | 88.14 | 87.59 |
| Supercharger speed \( n_{\text{tc}} \), rpm | 8252 | 8258 | 8542 |
| Calculations log |  |
| Computation time, s | 41.96 | 46.66 | 406.05 |
| Number of iterations | 36 | 40 | 351 |
Conclusions
Turbocharger compressor performance maps treatment should provide the ability to adjust the simulation modes according to the calculations tasks. Proposed three calculation modes – “stable” (neglects the compressor surge), full-scale surge and adjustable mild surge – allow to get faster and stable calculations and to overcome some numerical issues that cause the non-physical instability of the compressor operation.

The problem of the correct compressor surge simulation need further research to provide better recommendations for the mathematical model adjustment.

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Як приклад розглядається режим роботи двигуна MAN 8G70ME-C90.

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ЗАСТОСУВАННЯ КАРТ ХАРАКТЕРИСТИК ВІДЦЕНТРОВОГО КОМПРЕСОРА ПРИ МОДЕЛЮВАННІ РОБОЧОГО ЦИКЛУ ДВИНУГІВ ВНУТРІШНЬОГО ЗГОРОЖЕННЯ

Д.С. Мінцев, Р.А. Варбанець

Моделювання робочого циклу двигуна внутрішнього згоряння з наддувом неможливо без коректної оцінки параметрів роботи агрегату наддуву. Стандартним є підхід, при якому використовуються спеціально підготовлені карти характеристик компресора, отримані на основі експериментальних даних або даних, представлених виробниками. Інтерполяція, екстраполяція та обробка карт характеристик відцентрового компрессора є досить непростим завданням, так як необхідно забезпечити коректне впливання всього більшого, як поблизу точки виникнення збурення. Як приклад розглядається режим роботи двигуна MAN 8G70ME-C90.

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Дослідники
повинен рационально використовувати всі три режими врахування помпажа відцентрового компрессора в залежності від поставленого завдання, віддаючи перевагу режиму «стабілізації» при початковому налаштуванні математичної моделі і режиму локального помпажа для перевірки вірогідності виникнення помпажа. У той же час необхідні додаткові дослідження в області коректного моделювання роботи компрессора в умовах помпажа.

**Ключові слова:** відцентровий компрессор; помпаж; моделювання двигуна внутрішнього згоряння; характеристики компрессора

**ПРИМЕНЕНИЕ КАРТ ХАРАКТЕРИСТИК ЦЕНТРОБЕЖНОГО КОМПРЕССОРА ПРИ МОДЕЛИРОВАНИИ РАБОЧЕГО ЦИКЛА ДВИГАТЕЛЕЙ ВНУТРЕННЕГО СГОРАНИЯ**

Д.С. Мичев, Р. А. Варбанец

Моделирование рабочего цикла двигателя внутреннего сгорания с наддувом невозможно без корректной оценки параметров работы агрегата наддува. Стандартным является подход, при котором используются специально подготовленные карты характеристик компрессора и турбины турбокомпрессора, полученные на основе экспериментальных данных или данных, представленных производителями. Интерполяция, экстраполяция и обработка карт характеристик центробежного компрессора является достаточно непростой задачей, так как необходимо обеспечить корректный учет работы компрессора в таких специфических зонах его работы, как вблизи границы запирания, вращающегося крыла и в режимах помпажа. При этом необходимо обеспечить быстрое и стабильное выполнение расчетов рабочего цикла. Bliz-Pro – программный сервис для расчета рабочего цикла двигателей внутреннего сгорания, доступный онлайн. Предлагается возможность подготовки и использования карт характеристик агрегатов наддува. При этом доступны три различные режимы учета работы центробежного компрессора в условиях помпажа: 1) полномасштабный помпаж на основе подхода, предложенного F. K. Moore и E. M. Greitzer; 2) мелкомасштабный помпаж с возможностью гибкой настройки; 3) режим «стабилизации», при котором явлений помпажа компрессора исключается путем экстраполяции расходной характеристики компрессора от точки возникновения срыва в сторону уменьшения расхода воздуха с использованием гиперболического уравнения. В качестве примера рассматривается режим работы двигателя MAN 8G70ME-E (точка 12140 кВт при 82 мин-1 по винтовой характеристике) со сравнением результатов расчета для трех методов учета работы компрессора в условиях помпажа. Режим «стабилизации» обеспечивает наиболее быстрый и стабильный расчет, во время как режим учета полномасштабного помпажа может в ряде случаев характеризоваться необоснованной неустойчивостью расчетов, что объясняется высокой чувствительностью двухтактных малооборотных двигателей к частоте вращения и режиму работы центробежного компрессора. В то же время необходимо дополнительное исследование в области корректного моделювання работы компрессора в условиях помпажа.

**Ключевые слова:** центробежный компрессор; помпаж; моделирование двигателя внутреннего сгорания; характеристики компрессора.

УДК 621.436:621.43

**В.М. Бганцев**

**НАУКОВІ ПРИНЦИПИ ПІДВИЩЕННЯ ЕФЕКТИВНИХ ПОКАЗНИКІВ ДИЗЕЛІВ ПРИ ВИКОРИСТАННІ БІОПАЛИВ**

Використання палив біологічного походження на основі рослинних олій та іншої аналогічної сировини в дизельних двигунах з наддувом в розділеніх кількостях дозволяє зменшити витрати мінерального дизельного палива та покращити стан оточуючого середовища. Це підтверджує дослідження, проведені відомими російськими та німецькими вченими, що об'єднали різноманітні дослідження в області використання біопалив.

Однак, використання біопалив набуває своєї специфіки: вони не так добре згорають як традиційне дизельне паливо; потужність двигуна, відносна продуктивність двигуна, швидкісні характеристики двигуна збільшуються. У цій статті розглянуто питання про вплив швидкісних характеристик двигуна при використанні біопалив.

**Ключові слова:** двигун; відцентровий компрессор; помпаж; полномасштабный помпаж; центробежный компрессор.

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