ISSN 1392-1134
Aleksandro Stulginskio Universiteto mokslo darbai, 2012, 44 (1-3), 131-143
Research papers of Aleksandras Stulginskis University, 2012, vol 44, no 1-3, 131-143

INSIDE CYLINDER PROCESSES MODELING IN AN INTERNAL COMBUSTION ENGINE WITH HEAT REGENERATION

ŠILUMĄ REGENERUOJANČIO VIDAUS DEGIMO VARIKLIO CILINDRE VYKSTANČIŲ PROCESŲ MODELIAVIMAS

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Gauta 2012-04-30 , pateikta spaudai 2012-09-07

Works on construction of a piston internal combustion engine with heat regeneration modeled on Stirling engine are carried out in Marine Power Plant Department of Polish Naval Academy. The essence of the project is that the engine will have characteristics of both diesel and Stirling engines. This engine is structurally similar to β-type Stirling engine that implements the open cycle. This engine is characterized in that the compression and heat delivery from the regenerator performs as in the Stirling cycle, while providing heat from the combustion and expansion as it pursues in diesel cycle. The exhaust gases dissipate heat to the regenerator before they are removed from the cylinder and replaced with cold air. Heat stored in the regenerator is used to the compressed air in the next cycle. The main advantage of the proposed solution is that: heat delivery occurs in an internal combustion process without losses of exchange and allows for a short pulse of higher temperature than in the Stirling engine. High temperature working medium, as in diesel engine and heat recovery as in Stirling engine provides high engine performance that is the subject of the project. Current research is limited to the engine construction model studies. Theoretical cycle of the engine actually shows the efficiency close to Carnot cycle efficiency at the same temperatures. In practice there are deviations from the theoretical cycle, hence to implement the necessary studies, were modeled the processes taking place inside the cylinder. Both the existing models of internal combustion engines and Stirling engines fired models allow direct modeling of the proposed workflow engine. A model which is a combination of typical elements in the modeling process in the engines of both types is presented in the paper.

Diesel engine, Stirling engine, heat recovery.

Introduction

Currently, heat engines with higher thermal efficiency are very large units of low-speed compression ignition engines, commonly known as diesels. Along with the dimensions reduction the slot-wall effect is dominated and negative relation of heat exchange surface to volume ratio impact on efficiency of diesel engines. Therefore, engine efficiency drops dramatically. The air near the wall cools the combustion chamber so that the flame is extinguished in the immediate vicinity of the cold parts of the combustion chambers and cold cracks, and this not only results in engine performance reduction but also in emission of unburned hydrocarbons into the atmosphere. Because of this there is justified that the construction of a diesel engine with volume less than $350cm^3$ is not advisable. In addition, to ensure the required strength needed to move large gas forces resulting from high compression pressure needed to create the conditions for the self-ignition robust engine design is required and with a much greater mass than in spark-ignition engines. Engines, where the wall effect exerted beneficial effects on the combustion process, were so-called semi engines.

An engine having two hot and cold chambers is a Stirling engine, in which the chamber is cold during compression stroke to the extent possible, cooler, and hot during expansion stroke. The gas (which is the working medium) parameters during pumping between the chambers are dependent on the direction of heating or cooling in heat exchangers. In this type of engine heat recovery is possible and the efficiency of the engine is close to theoretically achieve efficiency in Carnot cycle. However, providing heat through the heat exchanger reduces the temperature of the upper heat source to the allowable operating temperature by structural material. For example for heat-resistant steel H25N2052 it is reduced to 1425K. These results in lower efficiency compared to internal combustion engines, which in the impetus could reach a temperature of 2500K. So far, Stirling engines have been used as engines utilizing the heat from engine combustion chamber where permitted burning of any fuels including solid fuels have occurred. Currently, they mainly use waste heat or solar energy. The main feature of the Stirling engine is that it pursues a closed cycle without changing the working medium, which is usually an inert gas, mainly helium or nitrogen.

The essence of the internal combustion engine with heat regeneration is to create a cylinder head complex of the hot part of the cylinder and the cylinder embedded in a refrigerated cold body of the cylinder, inside which move synchronously two pistons, one sealed in the cylinder, the second called displacer, unsealed. The space above a displacer crown creates a hot cell, which occurs at burning, and between its lower part and a crown of the piston, cold chamber, in which compression occurs. Displacer outer diameter is at least one millimeter smaller than the inner diameter of the cylindrical part of the head, which creates circumferential flow channel for air flow from the cold chamber to a hot cell, which is accompanied by heat downloads from walls elements which crating the channel. As directed flow continues until they reach the minimum volume of cold chamber, at which point the fuel is injected and as a result spontaneous combustion occurs. Flow direction changes from that point to the opposite, resulting in exhaust gas pressure on the piston crown and at the same time, during the flow through the peripheral channel, heat transfer to the walls of it. Cylinder head with a hot cylindrical portion and the upper and side displacer walls are made of steel, while its lower part and the piston are made of aluminum alloy in such a way as to increase their surface area to maintain a constant low temperature during compression. In addition, the head of the outside cylindrical cylinder head part is thermally insulated from the environment. The upper central part of the head contains a cooled seat in which injector is mounted, also thermally insulated in the ceramic insulating sleeve channel. The engine working chamber is shown in Figure 1.

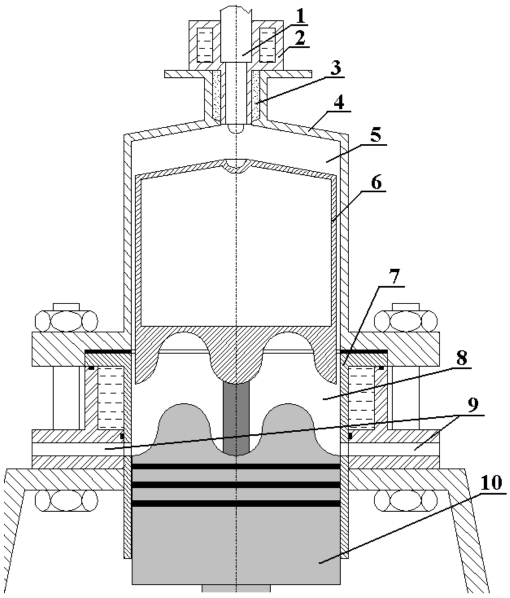

Fig. 1. The working chamber of the internal combustion Stirling engine: 1 - injector, 2 – injector cooling slot, 3 – insulating sleeve, 4- head of the cylindrical part, 5 – hot combustion chamber, 6 - displacer, 7 –cooled cylinder part, 8 – cold compression chamber, 9 –charge exchange channels, 10 - piston

1 pav. Vidaus degimo Stirlingo variklio darbinė kamera: 1 - purkštuvas; 2 – purkštuko aušinimo anga; 3 – izoliuojanti įvorė; 4 – cilindrinės dalies galvutė; 5 – karšta degimo kamera; 6 – stūmiklis; 7 - aušinamoji cilindro dalis; 8 – šaltoji suspaudimo kamera; 9 - dujų apykaitos kanalai; 10 - stūmoklis

This engine would be working in the two-stroke cycle. During the one rotation due to not simultaneous movements of piston and displacer five phases, as shown in Figure 2, can be distinguish.

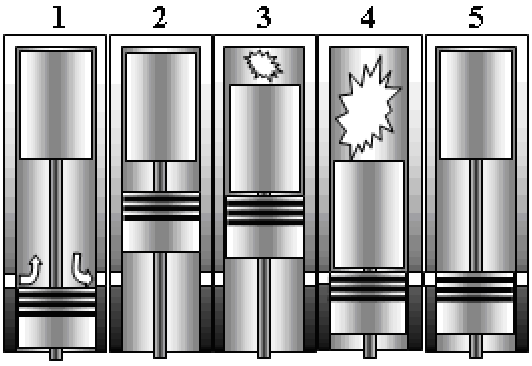

Fig. 2. The cycle of the engine: 1 – charge loading, 1-2 - compression,2-3 - heat extract from regenerator, 3 - fuel injection, 3-4 - work, 4-5 - heat putting to regenerator

2 pav. Variklio ciklo eiga: 1 – mišinio įsiurbimas; 1-2 – suspaudimas; 2-3 – šilumos paėmimas iš generatoriaus; 3 – degalų įpurškimas; 3-4 – darbo eiga; 4-5 – šilumos perdavimas į generatorių

For hot engine in steady state condition, they would look as follows:

1. The charge exchange - displacer turning in an external position (TDC) and the piston in the inner lower position (BDC). The cylinder is fed with fresh air that cools the walls of the compression chamber.
2. Compression - the piston moves to TDC and compress the air. The gas transformation will be intermediate between adiabatic and isothermal. During compression stroke generated heat is carried away to cold walls of the compression chamber.
3. Heat recovery from the regenerator - after reaching the TDC displacer moves from TDC to the surface of the piston. Air is pumped from the cold to hot chambers without changing the volume. Passing through the gap gas is heated in the isochoric transformation by displacer hot wall, head and cylinder. The increase in temperature would cause an increase in pressure. Pumping air after the hot combustion chamber should have at or above 700 ^{o}C, so there are conditions to fuel self-ignition.
4. Work stroke - when displacer contacts the plunger all air will be in the hot chamber and then fuel is injected as in diesel engine. Self-ignition occurs and raised temperature and pressure of working medium. This moves piston in the BDC direction. Displacer must follow the piston so that the volume of the cold chamber in this part of the engine operation cycle is minimal.
5. Regenerator heating - before piston reaching the BDC charge exchange channels will be unveiled just before the displacer moves towards the TDC. This will pump hot exhaust gases from combustion chamber to cold chamber. Exhaust gases flowing through the gap between cylinder and displacer are cooling cylinder head, displacer and hot parts of the cylinder.

6. Charge exchange - when displacer reaches TDC and piston is continuing move toward the BDC charge exchange will occurred through open channels. The exhaust gases of relatively low temperature and low pressure will be removed from the engine working space to the atmosphere by the incoming fresh air. The engine reaches its initial state and the whole cycle will repeat.

Theoretical cycle of regenerative heat engine

The theoretical considerations for mapping of five phases of engine operation assumes that the piston and displacer move with uniform motion so as to reach the final position of the characteristic phases introduced in the low temperature characteristic points of cycle. Assumed initial cycle temperature of 300K and pressure of 0.1 MPa, the maximum regenerator temperature, the head temperature, the hottest parts of the cylinder and the outer displacer part depend on the type of steel used in their construction. In the case of structural steel, it cannot exceed 540˚C, with alloy tool steel - 700˚C, while for heat resistant steels - 1000˚C, for steel types H23N13 or H25N2052 - 1150˚C. The discussion assumed the worst way of heat isobaric delivering as in diesel combustion cycle. The amount of supplied heat is limited to a value that causes an increase in temperature such that at the end of the expansion to obtain a temperature not higher than the allowable operating temperature of the material plus the possible loss of heat.

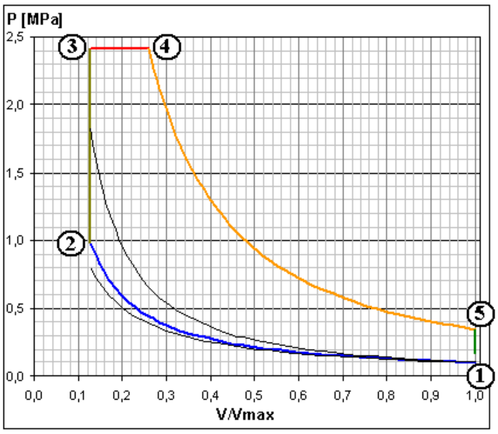

Fig. 3 Theoretical engine cycle in coordinates of P-V. 1-2 - polytrophic compression, 2-3 - isochoric heat providing from regenerator, 3-4 - isobaric heat supply from combustion,4-5 - polytrophic expansion, 5-1 - isochoric heat transfer to regenerator

3 pav. Teorinis variklio ciklas P-V koordinatėse: 1-2 – politropinis suspaudimas; 2-3 – izochorinis šilumos tiekimas iš generatoriaus; 3-4 – izobarinis šilumos tiekimas degimo proceso metu; 4-5 – politropinis išsiplėtimas; 5-1 – izochorinis šilumos perdavimas į generatorių

Difference between the temperature of air leaving the regenerator and the temperature of exhaust gas fed to the regenerator is assumed as 100 K. This difference depends on the efficiency of the regenerator and is necessary to cover the losses of heat exchange. Theoretical cycles of the engine with compression ratio of 8 which lighten these limitations are shown in Figure 3 in coordinates of P-V and in Figure 4 in coordinates of T-S.

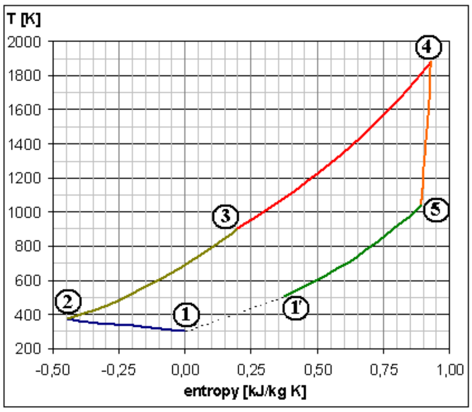

Fig. 4 Theoretical engine cycle in coordinates of T-S (symbols as in Figure 3)
4 pav. Teorinis variklio ciklas T-S koordinatėse: (Žymėjimai kaip 3 paveiksle)

Several simulations were performed. Cycle efficiency was calculated as a function of compression ratio for several polytrophic compression exponent values starting with isothermal compression adiabatic compression. In addition, the chart shows the Carnot and Sabathe cycle's efficiency under identical conditions. In the preliminary discussion, adopted polytrophic expansion exponent equal to 1.4 and the regenerator temperature equal to 970K. The results of this simulation are shown in Figure 5.

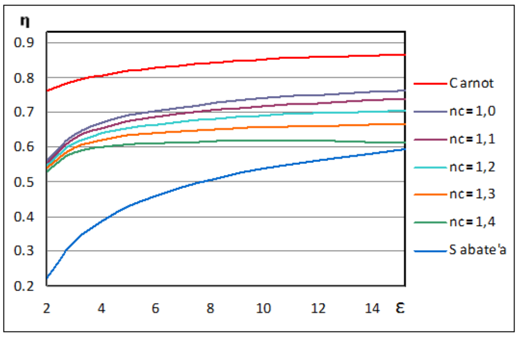

Fig. 5 Engine efficiency versus compression ratio compared to the Carnot and Sabathe cycle efficiency with different polytrophic exponent of compression and expansion constant polytrophic exponent equal 1.4
5 pav. Variklio efektyvumo priklausomybė nuo suspaudimo laipsnio, lyginant duomenis su Karnot ir Sabathe ciklų efektyvumu esant skirtingoms politropinio suspaudimo proceso eksponentėms ir pastoviai išsiplėtimo proceso eksponentei lygiai 1.4

On the basis of simulations, it was found that the efficiency of the proposed engine is between the Carnot and Sabathe cycle efficiency. At least favorable adiabatic compression (green) efficiency is close to 60% even at compression ratio = 4. While in the Sabathe cycle comparable efficiency can be achieved with compression ratio = 15. In isothermal compression efficiency value over the entire range is much higher than in the Sabathe cycle, and even at compression ratio 6 exceeds 70%.

Although it would be most beneficial isothermal compression, another simulation was conducted assuming polytrophic compression exponent equal to 1.1. This polytrophic exponent can be achieved with an efficient piston, cylinder and the cold bottom of the displacer cooling. To improve heat transfer piston crown should be performed with possibly large heat exchange surface. A relatively long time for heat exchange should also be provided which could be achieved only in low-speed engine. During this simulations polytrophic expansion exponent varied in the range of 1.2 to 1.6. The calculations were repeated for different compression ratios ranging from 2 and ending at 15.19 as the highest, in which the proposed engine cycle converges with Carnot cycle. The discussion does not include the initial pressure change as a result of molecular transformation.

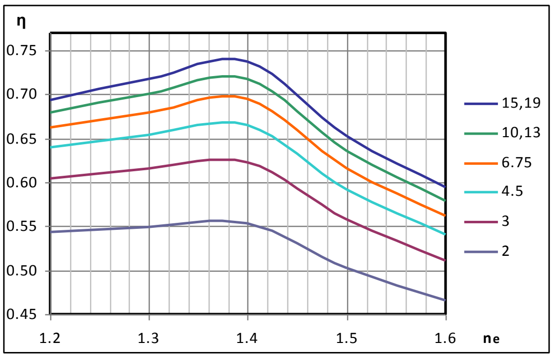

Fig. 6 Engine efficiency versus polytrophic expansion exponent with different degrees of compression ratio - and constant polytrophic compression exponent 1.1
6 pav. Variklio efektyvumo priklausomybė nuo politropinio išsiplėtimo eksponentės esant skirtingiems kompresijos laipsniams – ir pastoviai politropinio suspaudimo eksponentei 1.1

Regardless of the compression ratio the best performance could be achieved by an adiabatic expansion (polytrophic exponent close to 1.4). To get close to the adiabatic expansion heat exchange surface should be limited and provide possible high temperature of combustion chamber walls, so that its difference with respect to the average gas temperature was minimal. For compression close isothermal cooling of the piston and the bottom of the displacer should be applied, and their surfaces should be as large as possible. The upper part of displacer and head should have the smallest possible space to reduce heat transfer.

Figure 7 shows the effect of regenerator temperature limit for the engine efficiency versus compression ratio.

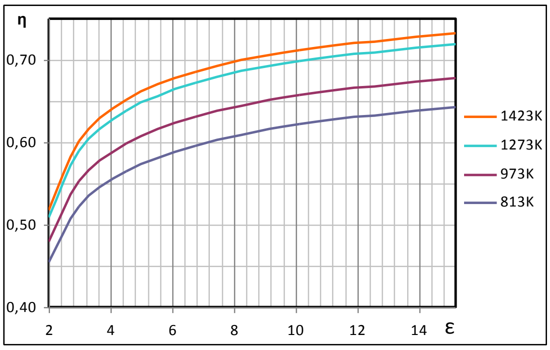

Fig. 7 Effect of allowable operating regenerator temperature on the engine efficiency versus compression ratio

7 pav. Variklio efektyvumo kitimas, esant leidžiamai generatoriaus darbinei temperatūrai, priklausomai nuo suspaudimo laipsnio

Higher regenerator temperature could provide higher temperature of the expansion end. Then, more heat will be isochoric supplied from the regenerator increasing cycle efficiency. However, to obtain a higher temperature at the end of expansion the higher maximum temperature should be provided. Values of the maximum cycle temperature for several regenerator temperatures versus compression ratio are shown in Figure 8.

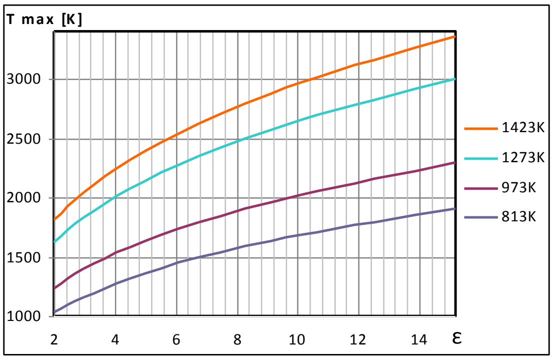

Fig. 8. Maximum cycle temperature for different temperatures of the regenerator versus compression ratio

8 pav. Maksimalios ciklo temperatūros kitimas, esant skirtingai generatoriaus darbinei temperatūrai, priklausomai nuo suspaudimo laipsnio

Crank system and deviations from the ideal cycle

To ensure optimum operating conditions for the engine operating mechanism must satisfy the following conditions:

- During the charge exchange and compression, from point 5 to point 2 displacer should remain in the TDC and the volume of hot space should seek to 0,
- At heat recovery time from point 2 to point 3 the total volume should be constant because the plunger should be gently moved to the TDC to compensate the increase in volume caused by active volume reducing by the piston rod,
- At work stroke from point 3 to point 4 the volume of the cold space between the piston and displacer should strive to 0.

It was decided to design and implement a research engine of a cylinder capacity equal 50 cm^3. Attempt was made to calculate the typical diamond crank system for Stirling engines [Fig. 9] but despite system optimization to achieve the assumed motion geometry of the diamond system was impossible. In an exemplary system the piston crank runs the road obtained as in Figure 10. Particularly pernicious deviation from expectation is that at the end of charge exchange displacer remains at about half-way to TDC. Additionally hot gases in the cylinder have significant deviation from the ideal gas and theoretical cycle.

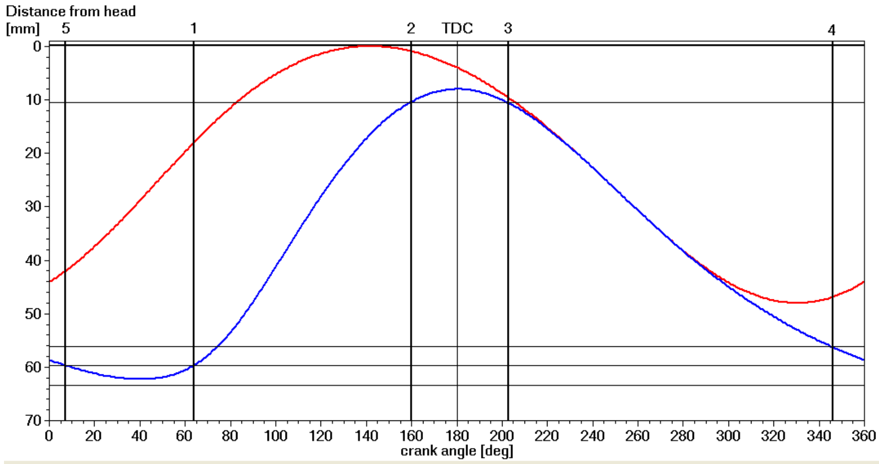

Fig. 9. Graph of the piston and displacer position versus crank angle in the diamond system

9 pav. Maksimalios ciklo temperatūros kitimas, esant skirtingai generatoriaus darbinei temperatūrai, priklausomai nuo suspaudimo laipsnio

Therefore it was implemented patent procedure for constructed engine crank angle system which we could not disclose in this paper. We can only show in the Figure 10 the graph of the piston and displacer position versus crank angle for this new crank system.

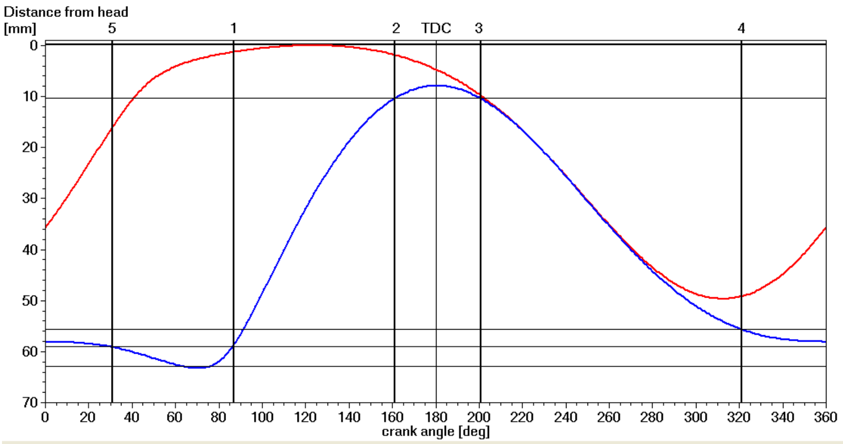

Fig. 10. Graph of the piston and displacer position versus crank angle in the constructed system

10 pav. Grafikai atspindintys stūmoklio ir stūmiklio padėties kitimą priklausomai nuo sukonstruotos sistemos veleno posūkio kampo

An attempt was made to modify the system as a result of rhomboid supplemented by an additional element of self-aligning. We cannot publish the scheme of created innovative system because of the ongoing patent application procedure. This system has 13 parameters affecting the movement geometry of the piston and displacer. These parameters are the dimensions of the elements, the points of arms support, and offset angles between the axes. Such a number of parameters and a specially designed program enabled the optimization of arbitrary chosen shape displacer and piston displacement, despite the limitations imposed by design nature. System is restricted by the minimum angle value of the connecting rod, the minimum pins diameter and, consequently, the minimum distance between the pivots and pins. Obtained crank system kinematics was examined in the Autodesk Inventor program.

Conclusions

The prepared engine is similar in construction to the Stirling engine, however, differs from the Stirling engine in way of heat delivering, which is made by heater and charge cooling replacing by the internal combustion process. Engine loses virtually all the features and advantages of Stirling engine, however, remains the possibility of regeneration of heat from the working medium pumped between the chambers. Heating and cooling elimination will reduce the harmful volume and eliminate heat transfer losses.

Both disadvantages of Stirling engines such as outside combustion chamber separated from working medium and the working medium separated from the refrigerant as the main reasons for reducing the efficiency are practically solved in the project. This will ensure that:

1. The high thermal engine efficiency according to the calculations above 60%.
    a. Higher than in Stirling engine due to: smaller harmful volume, no heat exchange losses and ability to obtain a pulse temperature of about 2500K as in diesel engines,
    b. Higher than in diesel engine due to: lower compression work, the possibility of heat recovery and not to receive heat during the expansion.
2. Work on low-quality liquid fuels by high temperature combustion chamber walls, as in the medium pressure engines with incandescent bulb Lanz Bulldog tractor.
3. Reducing emissions of harmful substances into the atmosphere. CO and HC - because of better combustion in the hot combustion chamber and burning in the gap between displacer and hot cylinder at a temperature above the ignition temperature. NOx - due to lower pressures and temperatures than in diesel engine. CO_2 - due to better heat utilization.
4. The deposits generation through high entire combustion chamber wall temperature and significant excess of air.
5. Reduced oil consumption because the oil lubricates only the cold chamber and do not enters the combustion chamber so there is no oil combustion in the engine like in a diesel.
6. Limited thermal field. Engine parts, head, part of the cylinder and displacer heat up to a temperature of the "white heat" and shall be thermally insulated from the environment. Gases prior to exhaust while giving heat to the regenerator reaches a temperature much lower than in diesel engine.
7. Limited acoustic emission due to less energy - cooled exhaust and lower engine speed.
8. A smaller mass of the engine due to lower compression ratio and lower maximum pressure than in diesel engines.

High efficiency would be ransomed significant complication of the engine crankshaft system. Nevertheless, even with the loss of performance due to deviations from theoretical cycle can be expected that this engine project will be economically justified in many applications.

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МОДЕЛИРОВАНИЕ ПРОЦЕССОВ В ЦИЛИНДРЕ РЕГЕНЕРАЦИОННОГО ДВИГАТЕЛЯ ВНУТРЕННЕГО СГОРАНИЯ

Аннотация

В отделе Морского электрооборудования Польской военной морской академии выполнены конструкционные работы по моделированию конструкции поршневого регенерационного двигателя внутреннего сгорания в соответствие с прототипом двигателя Стирлинга. Сущность проекта составляет желание иметь спроектированный двигатель, характеристики которого будут отражать положительные качества обоих дизельного и Стирлинга двигателей. Этот двигатель структурно похожий на β-типа двигатель Стирлинга, который работает по открытому циклу. Такой двигатель отличается тем, что процессы сжатия и подвода регенерационной теплоты выполняются таким же образом, как и в цикле Стирлинга, а выделение теплоты в процессе сгорания и сам процесс расширения происходит как в дизельном цикле. Теплота отработавших газов, прежде чем устранить их из цилиндра и заменить свежим воздухом, отводится в регенератор. В регенераторе накопленная теплота используется в следующем цикле вместе с сжатым воздухом. Основным преимуществом такого решения является то, что подвод теплоты в процессе внутреннего сгорания происходит без потерь в процессе газообмена и в течение короткого такта позволяет достичь большей, чем в двигателе Стирлинга, температуру газов. Высокая температура рабочей среды как в дизельном двигателе и регенерация теплоты как в двигателе Стирлинга гарантируют эффективную работу двигателя, что и составляет сущность этого проекта. В настоящее время выполняется студия моделирования конструкции такого двигателя. Теоретический цикл двигателя на самом деле показывает эффективность близкую циклу Карно при одинаковых температурах. Практически отмечаются отклонения от теоретического цикла, поэтому в целях выполнения намеченных студий, процессы моделирования выполняются в цилиндре двигателя. Оба, существующие модели двигателей внутреннего сгорания и модели рабочих процессов двигателей Стирлинга позволяют выполнить прямое моделирование предложенного рабочего процесса двигателя. Модел, который описывает типичную комбинацию элементов процесса моделирования для обоих типов двигателей, представлен в этой статье.

Дизельный двигатель, двигатель Стирлинга, регенерация теплоты.

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ŠILUMĄ REGENERUOJANČIO VIDAUS DEGIMO VARIKLIO CILINDRE VYKSTANČIŲ PROCESŲ MODELIAVIMAS

Reziumė

Lenkijos karinio jūrų laivyno akademijos Jūrų elektros įrengimų skyriuje atlikti stūmoklinio regeneratorinio vidaus degimo variklio konstravimo darbai modeliuojant pagal Sirlingo variklio prototipą. Projekto esmę sudaro tai, kad suprojektuotas variklis turės abiem dyzelinio ir Stirlingo varikliams būdingų charakteristikų. Šis variklis struktūriškai yra panašus į β-tipo Stirlingo variklį, kuris veikia pagal atvirą ciklą. Toks variklis pasižymi tuo, kad suspaudimo ir regeneruotos šilumos tiekimo procesai atliekami tokiu pat būdu kaip Stirlingo cikle, tuo tarpu šilumos išsiskyrimas degimo procese ir išsiplėtimo procesai vyksta kaip dyzeliniame cikle. Šiluma išmetamų deginių, prieš pašalinant juos iš cilindro ir pakeičiant šviežiu oru, nuvedama į regeneratorių. Regeneratoriuje sukaupta šiluma kartu su suspaustu oru yra panaudojama sekančio ciklo metu. Pagrindinis tokio sprendimo privalumas yra tas, kad šilumos tiekimas vidaus degimo procese vyksta be dujų apykaitos nuostolių ir per trumpą taktą leidžia pasiekti aukštesnę negu Stirlingo variklio temperatūrą. Aukštos temperatūros darbinė aplinka kaip dyzeliniame variklyje ir šilumos regeneravimas kaip Stirlingo variklyje užtikrina efektyvų variklio darbą, kas ir sudaro šio projekto esmę. Dabartiniu metu atliekama tokio variklio konstrukcijos modeliavimo studija. Teorinis variklio ciklas iš tikrųjų rodo efektyvumą artimą Karno ciklui, esant toms pačioms temperatūroms. Praktiškai yra pastebimi nukrypimai nuo teorinio ciklo, taigi siekiant įgyvendinti reikiamas studijas, modeliavimo procesai atliekami cilindro viduje. Abu, esami vidaus degimo variklių modeliai ir Stirlingo variklių darbo proceso modeliai leidžia atlikti tiesioginį pasiūlyto variklio darbinio proceso modeliavimą. Modelis, kuris aprašo tipinę modeliavimo proceso elementų kombinaciją abiejų tipų varikliuose, yra pateikiamas šiame straipsnyje.

Dyzelinis variklis, Stirlingo variklis, šilumos regeneracija.