МІНІСТЕРСТВО ОСВІТИ І НАУКИ УКРАЇНИ
НАЦІОНАЛЬНИЙ АВІАЦІЙНИЙ УНІВЕРСИТЕТ
АЕРОКОСМІЧНИЙ ФАКУЛЬТЕТ
Кафедра авіаційних двигунів

ДОПУСТИТИ ДО ЗАХИСТУ
Завідувач кафедри
д-р техн.наук, проф..
____________ М.С. Кулик
"____" _________ 2020р.

ДИПЛОМНА РОБОТА

ВИПУСКНИКА ОСВІТНЬОГО СТУПЕНЯ
МАГІСТР
ЗА ОСВІТНЬО-ПРОФЕСІЙНОЮ ПРОГРАМОЮ
" ГАЗОТУРБІННІ УСТАНОВКИ І КОМПРЕСОРНІ СТАНЦІЇ "
(ПОЯСНЮВАЛЬНА ЗАПИСКА)

Тема: Методи підвищення паливної ефективності газотурбінних установок на базі конвертуваних авіаційних двигунів.

Виконав: ____ Фахар Мохаммад.

Керівник: канд. техн. наук, доцент___________________ Гвоздецький І.І.

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Нормоконтролер : __________________ /____________________ /

КИЇВ 2020

MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE

NATIONAL AVIATION UNIVERSITY

AEROSPACE FACULTY

AEROENGINES DEPARTMENT

PERMISSION FOR DEFENCE

Head of the Department
Doctor of Sciences (Engineering), prof.
___________M.S. Kulyk
"_____" ______ 2020

MASTER'S THESIS

ON THE EDUCATIONAL PROFESSIONAL PROGRAM "GAS TURBINE PLANTS AND COMPRESSOR STATIONS"

(EXPLANOTARY NOTE)

Theme : Methods of fuel efficiency increase for gas turbine plants at base of conversion aircraft engines

Performed by: ____________________________________ **Fakhar Mohammad**

Supervisor:
Ph.D. (Engineering), Assoc. Prof. _________________________ **Gvozdetskyi I.I.**

Advisers:

Labor precaution:

Ph.D., Assoc. Prof. _______________________________ **Kovalenko V.V.**

Environmental protection:
Ph.D. (Engineering), Assoc. Prof. _______________________ **Cherniyak L.M.**

Standard Inspector _________________________ /_______________________/

Kyiv 2020

NATIONAL AVIATION UNIVERSITY

Faculty: *Aerospace Faculty*

Department: *Aeroengines department*

Educational degree: *Master*

Specialty: *142 "Power Machinery"*

Educational Professional Program: *«Gas Turbine Plants and Compressor Stations»*

APPROVED BY

Head of the Department

__________M. Kulyk

" " ____2019

Graduation Work Assignment

Fakhar Mohammad

(surname , name and patronic of graduating student)

1. Theme: « **Methods of fuel efficiency increase for gas turbine plants at base of conversion aircraft engines** »

   approved by the Rector's order of September 26, 2019 № 2187/ст

2. The project to be performed from 15.10.2019 to 20.01.2020
3. Initial data for the project: Gas turbine plant should be calculated for standard ambient conditions T_{amb}=288 K, P_{amb}=101325 Pa;
4. The content of the explanatory note (the list of problems to be considered): Introduction; main part; Complex cycle calculations, calculation of heat exchangers; patents research; Special part; Labour precautions; Environmental protection; General conclusion .
5. The list of mandatory graphic materials: simplified scheme of gas turbine plant with heat exchangers, the drawing of heat exchanger. Microsoft office Power Point, AutoCAD, should be used to provide graphic support and presentation.

6. Time and Work Schedule

№	Stages of Graduation Project Completion	Stage Completion Dates	Remarks
1.	Literature review of materials concerning theme of diploma work.	15.10 - 30.10. 19	
2.	Patent review of the problem.	31.10-06.11.19	
3.	Mobile gas installation review	07.11-11.19	
4.	Analyzing complex cycle for gas turbine plant	12.11- 16.11.19	
5.	Analyzing fuel preheating	17.11 - 21.11.19	
6.	Analyzing the effect of regeneration and fuel preheating on thermal efficiency	22.11 - 30.11.19	
6.	Labor precaution	15.12 - 20.12. 19	
7.	Environmental protection	20.12 - 30.12.19	
8.	Arrangement of graphical part of diploma work	14.01 -15.01.20	
9.	Arrangement of the explanatory note	16. 01.20 - 20.01.20	

7. Advisers on individual sections

Section	Adviser	Date, Signature	
		Assignment Delivered	Assignment Accepted
Labor precaution	Ph.D., Associate Professor Kovalenko V.V.		
Environmental protection	Ph.D. (Engineering), Assoc. Prof. Cherniyak L.M.		

8. Assignment issue date: «_ _» __ __ 2019

Diploma work supervisor: ___________________ I.I. Gvozdetskyi

(supervisor signature)

Assignment is accepted for execution: ____________ _____ Fakhar Mohammad

(graduate student's signature) (date)

ABSTRACT

The explanatory note to the diploma work "Methods of fuel efficiency increase for gas turbine plants at base of conversion aircraft engines"

92 Pages, 18 Illustrations, 6 Tables, 41 Sources, 2 Appendixes-44pages

Object of research — Gas turbine plant with increased thermal efficiency based on converted aero engine.

Subject of investigation — methods to increase the thermal efficiency of mobile gas turbine plant.

Aim of diploma work — is to study the possibility of increasing the fuel efficiency of a converted aviation gas turbine engine when it is used as a drive for mobile electrical power plants.

Research method — the thermodynamic and gas dynamic calculation of the engine, analyzing the heat exchanger parameters and ways of its improving, heat exchanger calculation.

Scientific novelty — Methods of estimation of fuel efficiency improvement by implementing complex cycle and fuel preheating.

Practical value of diploma work results is extension of the possible spectrum of the mobile power-plant. It can be used for the educational purposes.

GAS TURBINE ENGINE, CONVERSION, POWER-PLANT, THERMODYNAMIC CALCULATION, GAS DYNAMIC CALCULATION, REGENERATOR, HEAT EXCHANGER, THERMAL EFFICIENCY, MOBILE POWER STATION.

CONTENTS

LIST OF ABBREVIATIONS AND SYMBOLS

GTP – Gas Turbines Plants

LNG – Liquefied Natural Gas

LDGT – Light-Duty Gas Turbine

GT – Gas Turbine

OPR – Overall Pressure Ratio

TET – Turbine Entry Temperature

HEPA – High Efficiency Particulate Air

HRSG – Heat Recovery Steam Generator

TBC – Thermal Barrier Coatings

FOD – Foreign Object Damage

EEC– Electronic Engine Control

PWPS – Pratt & Whitney Power Systems

GE – General Electric

GTU – Gas Turbine Unit

HP – High Pressure

PT – Power Turbines

GTE – Gas Turbine Engine

T_4 – Gas Temperature After Turbine

T_2 – Air Temperature After Compressor

K – Kelvin

η_{th} – Thermal Efficiency

π_c – Pressure Ratio

P_1 – Ambient Pressure

T_1 – Ambient Temperature

η_c – Compressor Isentropic Efficiency

η_t – Turbine Isentropic Efficiency

k – The Ratio of Specific Heats for Air

k_g – The Ratio of Specific Heats for Gas

σ_r – Effectiveness of Regeneration

INTRODUCTION

Gas turbine engines which has exhausted flight service life can be used for terrestrial objectives in the energetics. Turboprop (turboshaft) aircraft (helicopter) engine is the simplest to use as gas-turbine power plant, because in this engine, mechanical energy is transmitted on the outer shaft and can be used directly.

Such gas turbine power plant is indispensable for using in these cases, when:

– it is necessary to solve the problem of electrical and thermal energy, supply to a town or production plant - the modularity of the blocks allows to create any variants depending on the needs of its consumers;

– economic efficiency is important; modern gas turbines have efficiency up to 32 % in simple cycle, high efficiency of plant provides the possibility of producing cheaper electricity and heat generation and short payback period;

– it is industrial development of new areas of people's lives and the natural features of these territories have great importance; operation of the plant is ensured in the range of ambient temperatures from -50 to +50 °C when the adverse weather conditions like humidity, rain, snow effect (the level of stress and thermal condition of the aviation engine parts almost has no analogues among the products of mechanical engineering);

– it is necessary the automation of control plants; gas turbine stations based on aircraft engines is very maneuverable, and require a small time to start from cold condition to full load, can be automated and controlled remotely.

This way assumes application of heat regeneration using the exhaust gas heat to reduce fuel consumption which is necessary to achieve the required level of gas temperature entering the turbine unit.

It is known that preheating gasoline or other fuel for internal combustion engines can produce more efficient vaporization and higher combustion efficiency and engine performance than cold fuel.

In this work the regeneration and fuel preheating is investigated in order to recognize the increased thermal efficiency.

The result of this investigation were reported at several students conferences and are shown in scientific article [10]

The aim of this work is to study the possibility of increasing the fuel efficiency of a converted aviation gas turbine engine when it is used as a drive for mobile electrical power plants.

The result of this paper can be used to perform the conversion of this type of engine with high efficiency and low fuel consumption as a mobile power-plant which can be installed on a truck to generate electricity for different purposes in different environmental conditions with different type of fuels.

1. CONCEPT OF MOBILE GAS INSTALLATION BASED ON AVIATION ENGINE WITH REGENERATION AND PREHEATING OF FUEL

Conversion of aircraft engines that have exhausted their assigned flight service life time is an actual problem for both Ukraine and other countries with a developed aviation transport infrastructure [6]. At the same time, for converted aircraft engines, the demand for a significant increase in fuel efficiency as compared to that which these engines have when they are used on aircraft, where the minimum engine weight requirement is put in the first place, is put forward.

The essential to shrink the usage of fossil fuels have been increasingly part of our daily lives for both environmental and economic reasons. Decreasing the consumptions without compromising the engine performance can be a difficult task, to approach this goal one of the trustworthy way is to use regeneration, Regenerator is placed after the turbine and before the propulsion nozzle to heat the compressed air before entering the combustor, reducing the fuel consumption.

In this paper an analysis of the PT6T turboshaft engine with regeneration will be performed. In other words, observations will be made on how a regenerator affect the thermal efficiency of this engine beside the turboshaft engine (PT6 family) has been chosen to redesign as a mobile power-plant with regeneration. due to the exceptional compactness, low weight, flexibility, high efficiency and availability of the outer shaft (through which the mechanical energy is transmitted to electrical device) make this turboshaft engine as a good choice to use for terrestrial purposes.

The aim of this work is to study the possibility of increasing the fuel efficiency of a converted aviation gas turbine engine when it is used as a drive for mobile electrical power plants.

1.1 Mobile gas installation based on aviation engine

Gas turbine engines which has exhausted flight service life can be used for terrestrial objectives in the energetics. Turboprop (turboshaft) aircraft (helicopter) engine is the simplest to use as gas-turbine power plant, because in this engine, mechanical energy is transmitted on the outer shaft and can be used directly.

Such gas turbine power plant is indispensable for using in these cases, when:

– it is necessary to solve the problem of electrical and thermal energy, supply to a town or production plant - the modularity of the blocks allows to create any variants depending on the needs of its consumers;

– economic efficiency is important; modern gas turbines have efficiency up to 40% in simple cycle, high efficiency of plant provides the possibility of producing cheaper electricity and heat generation and short payback period;

– it is industrial development of new areas of people's lives and the natural features of these territories have great importance; operation of the plant is ensured in the range of ambient temperatures from -50 to +50 °C when the adverse weather conditions like humidity, rain, snow effect (the level of stress and thermal condition of the aviation engine parts almost has no analogues among the products of mechanical engineering);

– it is necessary the automation of control plants; gas turbine stations based on aircraft engines is very maneuverable, and require a small time to start from cold condition to full load, can be automated and controlled remotely.

1.1.1 Possible areas for the use of convertible aviation engines

The world demands more power infrastructure in distant areas and the ever-changing political and economic climate is often dictating how the electric power will be delivered. Our engineers can provide an array of generation and electrical distribution offerings to fit every requirement and configuration: Electrical Load, Accessible Fuel, Frequency, Voltage and Emissions Necessities.

The conversion of aero-engines into the industrial and marine gas turbine engines is an important business. The turbomachine which fits the energy of high temperature and high pressure gas flow into mechanical energy is commonly called a gas turbine engine. Separately from its use as an aircraft power plant, the gas turbine engine is also extensively used in industry and ships. Compared with conventional internal combustion engines and steam engines, the gas turbine has a much higher operating temperature. In its industrial application, the high temperature and high pressure gas expands and produces useful work through the turbine, which in turn energies the generator. The exhaust gas is still about 500 degrees C and it could go to the residual heat boiler, which in turn produces high pressure steam. The steam drives the steam turbine for electricity generation for the second time (the steam and the hot water can also be used directly). The waste gas from the boiler can be absorbed by the heat exchanger for the industry and daily uses. The heat energy of the fuel can be exploited in phases and the combined heat utilization degree will reach more than 80%.

Gas turbine design has advanced enormously in the last 20 years and progress continues. In the past higher efficiency was the primary aim of gas turbine development. Around 1990, typical complex-cycle efficiency was 50 percent. By 2010 the best installations were above 59 per cent.

Efficiency remains imperative to all manufacturers, but there has been a significant shift towards products that can operate more flexibility in load following and grid support roles. Whereas before combined-cycle plants were seen as baseload plants, now their role is equally likely to be for intermediate service and to provide auxiliary services.

An important part of this change is due to the increasing importance of mutable output renewable capacity such as wind and solar power to grids across the world. As the proportion grows, and turbine manufacturers are expecting these technologies to contribute 20 per cent to 30 per cent of grid capacity in Europe by 2020, so the combined-cycle plant, along with energy storage and hydropower, will be required to provide the key back-up for this intermittent capacity.

Efficiency and flexibility are not complementary. In order to gain the flexibility which is required for this type of service, some efficiency must be sacrificed. The main manufacturers are replying to this by offering complementary heavy gas turbine lines, one for ultimate efficiency and one that offers greater flexibility. Even so, high efficiency stays the equivalent of the space race within the gas turbine world, though one where gains are becoming progressively tough to find.

1.1.2 Aircraft gas turbine engines converting as possible way of improvement terrestrial installations

Aero-derivative gas turbines plants (GTP) are a widespread choice for energy generation thanks to their reliability, efficiency and flexibility.

Founded on advanced aircraft engine technologies and materials, they are meaningfully lighter, respond quicker and have a slighter footprint compared with their heavy industrial GTP. With up to 45 % efficiency compared to up to 35 % for heavier GTs, these turbines are frequently seen as a decent choice in smaller-scale energy generation. The turbines are moreover widespread due to their fuel flexibility — they permit a combination of natural gas and liquid fuel operation. As such, the global aero-derivative GT market is likely to grow at an annual growth rate of nearly 5 % between 2016 and 2020, according to a 2016 study by Technavio.

Asia, in certain, organizes many of these machines in power trains for Liquefied Natural Gas (LNG) plants. In the U.S., aero-derivatives are mostly being used in peaked operations, or to compensate for fluctuations in the grid produced by renewables or extreme weather conditions. Just around all the aero-derivatives on the market come from GE, Siemens and Pratt & Whitney Power Systems (PWPS).

The industrial gas turbine has developed in two orders: one is particularly designed and developed for industrial use mostly heavy-duty; another is the design on the basis of aero-technology or to convert the aero-engines; they are typically lighter in structure, and named light-duty gas turbine (LDGT). After the 1960s, the aero gas turbine engines have made quick progress, and the connected technology has touched a mature stage. The reliability and economy are significantly improved, the compression ratio is about 25, and TET reaches 1,400°C. That is one generation ahead of the ground heavy-duty gas turbines. Consequently, most of the industrial and marine light-duty gas turbines are adapted and developed from aero-engines. In this method, it can take the advantage of the advanced technology attained by the aviation industry with a lot of manpower, material, money and time. In the industrial countries, all successful aero-engines have their industrial and marine forms.

1.1.3 The main advantages of aero-derivative gas turbine engines

The gas turbine has the following advantages: light weight, small size, fast start, high reliability, no need for cooling water and flexible to numerous fuels such as diesel, heavy oil, petroleum, natural gas, tail gas, low heat value gas, etc. It is easy to mount or dismount, with a minor amount of construction work, and can be remotely auto-controlled. Consequently, the gas turbine engine is a more adaptable, advanced, energy saving power plant. The industrial countries in the world all develop it with excessive efforts. Therefore, it plays an significant role in the field of power and transportation.

Lighter in weight :

This engine has similar design with jet engine which has lighter weight because of limitation from lift force to make jet engine to fly. Aero-derivative gas turbine will eliminate the fan, fan case and the thrust nozzle, install it on ground, mount the load from compressor and generator on a separate shaft/same shaft with dissimilar configuration such as concentric shaft. These turbines are lighter in weight.

Clean burning:

Typically, these engines used natural gas. Due to natural gas is ease to use, low emission (NOx and SOx number), and low cost for maintenance operation.

Higher rotation speed:

Aero-derivative gas turbine can extent higher speed than conventional industrial gas turbine. It can be around 3.000 rpm for low speed rotor shaft and 10.000 rpm for high speed rotor shaft

Multiple applications:

It can be used in numerous project such as mobilized tank and carrier vessel for military dedications.

Less water consumption:

Since those engine using oil as cooling system (with natural convection through air and fan) it won't make water consumption as an important thing.

Possible Applications for Mobile Power Include:

- Emergency Power
- Power for Oil Rigs/Mining/Industrial Operations
- Pad or Section Power
- Electric Motor Fracking
- Enhanced Oil Recovery Operations
- Distributed Generation or Micro-Grids
- Construction Sites/Bridging Power
- Special Events
- Temporary Power During Maintenance of Main Systems
- Emergency Power for Natural Disasters
- Electric Power for Seasonal Peak Demands
- Grid Stability and Support

1.1.4 Quest for high efficiency

For the past decade an significant target for all the large gas turbine manufacturers has been to achieve 60 per cent efficiency in a combined-cycle plant under standard (ISO) conditions. Although several have now come close, that goal has yet to be achieved.

Efficiency is significant because it affects both operating costs and carbon emissions, both of which are important concerns for plant operators today. Achieving high efficiency requires optimization across the whole plant, including gas turbine, heat recovery steam generator (HRSG) and steam turbine. Between these three, nevertheless, gas turbine performance seems to be the key to pushing efficiency to 60 percent and beyond.

A gas turbine comprises three foremost components, a compressor, a combustion chamber and an expansion turbine integrated into a single unit. Thermodynamic efficiency of this unit depends on both the pressure and temperature drop of the working gas, air, from the turbine stage inlet to outlet. Increasing one or both can be used to escalation efficiency. For this, all three components play their part.

The optimization of compressor design can be used to deliver a higher pressure at the turbine inlet and this is being exploited in high efficiency designs. Thus General Electric has increased the gas turbine pressure ratio in its H system turbine to 23:1, aiming for 60 percent efficiency. The same pressure ratio is used by MHI in its J-series, nevertheless the higher turbine inlet temperature outcomes in efficiency in excess of 61 percent. Siemens SGT5-8000H has a pressure ratio of 18.2, slightly higher than its SGT5-4000F. Meanwhile, Alstom's turbines have operated since the mid-1990s with a pressure ratio of about 30:1.

The combustion chamber controls the temperature of the gas inflowing the turbine so changes here can affect overall efficiency. Although raising flame temperature is fairly straightforward compared to other changes, combustor designs have inclined to stabilize and all the leading companies now have proven designs which they utilize across their turbine ranges.

This leaves the turbine as the component where the greatest improvements in efficiency are likely to be made, however improvements here are becoming hard to win. Modern computer design has managed to the optimization of blade profiles to gain maximum efficiency so there is little that can be done there. Thus designers are forced to exploit the one area left to them, turbine gas inlet temperature. Since thermodynamic efficiency hinge on the overall temperature drop across the turbine, raising the inlet temperature should equate to an increase in efficiency.

The combustion gas temperature for GE's H-class gas turbine, one of the 60 per cent contenders, puts it in the 1500 °C class, roughly 100 °C higher than the company's F-class machines, which constitute the strategic large frame GE turbines in use today. Inlet temperature of the Siemens' new H-class machine, which is also hoping to attain above 60 percent efficiency, has not been released.

Nonetheless, both are likely to be topped by MHI. The latter's J-series turbine boasts an inlet temperature of 1600 °C, 100 °C above that of the same company's H-series machines and higher than any other product to date. This machine, which is being verified in Japan today, is being marketed as capable of 61 percent efficiency.

At these temperatures, the materials used to manufacture gas turbine components are being pressed to their limits. Blades, vanes and other hot gas path components are usually made from nickel-based alloys cast in single crystal form to surge strength and resistance to fracture and deformation. Even though not by MHI for their outsized machines. Single crystal blades do exhibit improved properties and MHI has used them successfully in their small units, but the single crystal form is expensive and has associated repair challenges so, says MHI, do not validate their use in large frames.

These alloys will start to melt at wherever between 1200 °C and 1400 °C. In order to avoid the consequent damage at the temperatures experienced within the gas turbine inlet, these components must be coated with thermal barrier coatings (TBCs) and then cooled internally thus that the actual metal does not approach its melting point. Current TBCs are based on ceramics that are normally made from zirconia doped with various other elements to make a heat resistant layer with very low thermal conductivity that can uphold a 300 °C temperature drop to the metal beneath.

The TBC slows the rate at which heat is transferred to the metal beneath. However, in order to prevent the temperature rising too high, the component must also be cooled internally, either with air or steam. Air cooling is the most common type in use. This entails abstemiously hot air which is extracted from the gas turbine and cycled to cool the hot path components.

Nevertheless, since this air is 'stolen' from the gas turbine it has the effect of dropping overall efficiency. Steam cooling, in which steam from the steam cycle of the combined-cycle plant is used to cool the components internally does not have the same efficiency cost. This is the answer adopted by MHI, and by GE for their high-efficiency G-class and H-class machines. Conversely, Siemens is using air cooling for their latest H-class turbine and Alstom's turbines are also air cooled.

Temperatures may be able to go upper still. MHI is taking part in a Japanese government project which has the goal of increasing the turbine inlet temperature to 1700 °C. The J-series turbine has already exploited some of the developments from this program but there are expected to be others yet to be used.

Overall efficiency of these large industrial gas turbines in simple cycle mode is between 38 per cent and 42 per cent. Higher efficiency can be achieved. The best available today appears to be GE's LMS100 with a simple cycle efficiency of 46 per cent. However, this is at the expense of a low gas outlet temperature which makes it less appropriate for complex-cycle applications.

Power plant flexibility involves a number of significant parameters. A flexible plant must be capable to start-up quickly, so time from start-up to full load will be essential. It must be capable to ramp both up and down to encounter dissimilar load demand levels with ease and speed. Efficiency must be preserved, not only at full load but at part load down to 50 percent or lower. Emissions must also be reserved low when operating at part load.

Furthermore, companies are looking at methods of 'parking' a complex-cycle plant overnight so that it ticks over at low load. This avoids the necessity of shutting the plant down, which has both speed and maintenance penalties, or continuous operation at say 50 per cent load, which will often not be the most economical option. One of the technological advances which may need to be sacrificed for flexibility is steam cooling of the inlet stage gas turbine components. Maintaining the cooling steam is vital during ramps but requires slower ramp speeds due to the complexity and overall integration of gas and steam cycles.

However, abandoning steam cooling is likely to result in a lower gas turbine inlet temperature and hence lower overall efficiency. This is part of the reason why some companies offer more than one product line, one using steam cooling for high efficiency and another with air cooling for greatest operational flexibility.

GE has six of its high-efficiency H-class combined-cycle machines operational, one at Baglan Bay in Wales, three in Japan at a Tokyo Electric Power plant and two in California. According to GE, these are all operating at baseload and though none has achieved the magic 60 per cent efficiency, GE believes it has shown the machine to be capable of reaching this efficiency.

As with GE, the company hopes the KA26 and KA24 products will appeal to utilities and power plant operators who expect to operate in both the power delivery and ancillary services markets.

At the same time the company has recently launched a new version of is 60Hz 7FA turbine which features 57.5 per cent efficiency in combined-cycle mode but fast start-up and ramp rates and a better turn-down capability.

Meanwhile, MHI has pinned its colours squarely on high efficiency. Its high efficiency J-series turbine builds on experience with the company's G-class machines which use steam cooling, though only in stationary parts such as combustion liners and blade rings. In the J-series, this is promised to lead to 61 per cent efficiency.

Toshishige Ai of MHI says the quest for high efficiency has particular value in Japan; the country has to import all its fossil fuel so efficiency is recognized as extremely important from both a cost and an emissions perspective. Nevertheless, MHI has recently acknowledged that flexibility is also important and has developed a fully air-cooled version of the G- series turbine. As for the J-series, six units have already been sold, with the first starting up in 2013.

Siemens has developed the SGT5-8000H (50 Hz), an air-cooled gas turbine with a generating capacity of 375 MW, which is expected to exceed 60 per cent efficiency in combined-cycle configuration.

In the meantime, Siemens is also paying attention to flexibility with its own fast-cycling concept, development of which began in 2002. This has led to the introduction of features for instance a once-through HRSG to eradicate the high-pressure drum, which in a more conventional HRSG takes a considerable time to reach the desired temperature. Adding a stack damper helps reduce the amount of heat loss during shutdown, allowing quicker hot start.

These methods are being applied to complex-cycle plants based on the company's present F-class turbines. Conferring to Siemens, two plants, one at Pont sur Sambre, France and the other at Irsching, Germany have established the capability of getting full load after overnight stoppage in 30 minutes, with efficiencies of over 58 percent and over 59 percent in that order.

Siemens has been looking wisely at grid support necessities too, and has developed methods to maintain power output in the event of a drop in frequency as well as remaining in stable operation through islanding. Ansaldo Energia, whose turbines are grounded on Siemens – the company was a licensee until 2004 – is following a similar path to other companies in regard to flexible operation.

While the biggest heavy duty gas turbines and combined-cycle plants often attract the greatest attention, there leftovers a market for smaller units. An important player here is Rolls-Royce, which has lately introduced a new version of its RB211 industrial gas turbine called the RB211-H63.

Conclusion

In this part the information about Possible areas for the use of convertible aviation engines, methods of improvement of their installations, advantages and drawbacks and recent developments by different companies in this sphere has been considered.

2. GAS TURBINE PLANT ON THE BASIS OF THE CONVERTED AERO ENGINE WITH REGENERATION

2.1 Power drive for the mobile power plant

As a power drive for the mobile power plant the turboshaft engine PT6T-3D (PT6 family), figure.2.1, figure.2.2, which consists of two power sections coupled to a combining gearbox with a novel clutch system enabling both twin and single engine operation, was chosen [2,3,4].

The engine consists of two sections that can be easily separated for maintenance: a gas generator supplies hot gas to a free power turbine. A two-shaft configuration consisting of a multi-stage compressor driven by a single-stage compressor turbine and an independent shaft coupling the power turbine to the output shaft. The starter has to accelerate only the gas generator, making the engine easy to start, particularly in cold weather. Air enters the gas-generator through an inlet screen into the low-pressure axial compressor. The air then flows into a single-stage centrifugal compressor, through a reverse flow combustor, and finally through a single-stage turbine that powers the compressors. Hot gas from the gas generator flows into the power turbine. The gas generator speed is around 36,000 rpm. For turboprop use, this powers a two-stage planetary output reduction gearbox, which rotates the power turbine at a speed of 1,900 to 2,200 rpm. The exhaust gas then escapes through two side-mounted ducts in the power turbine housing. The turbines are mounted inside the combustion chamber, reducing overall length.

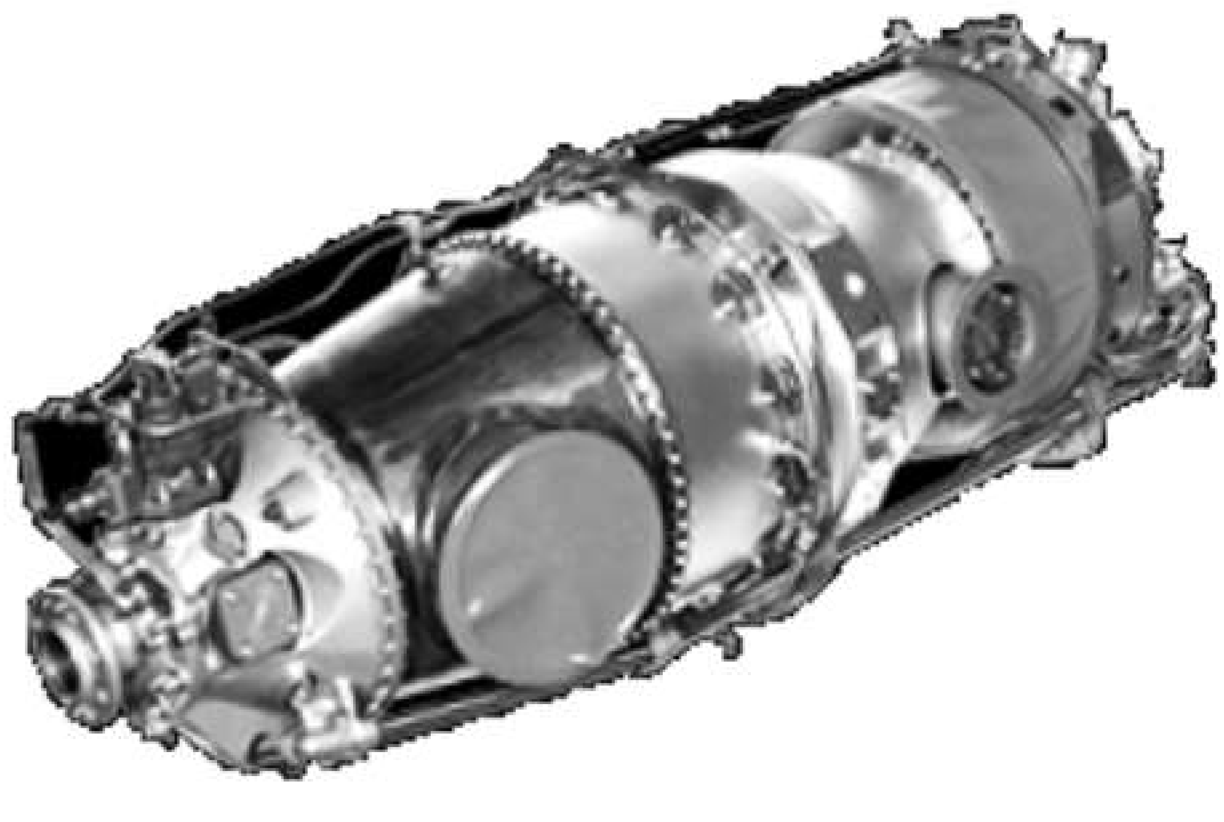

Figure.2.1. Turboshaft engine PT6

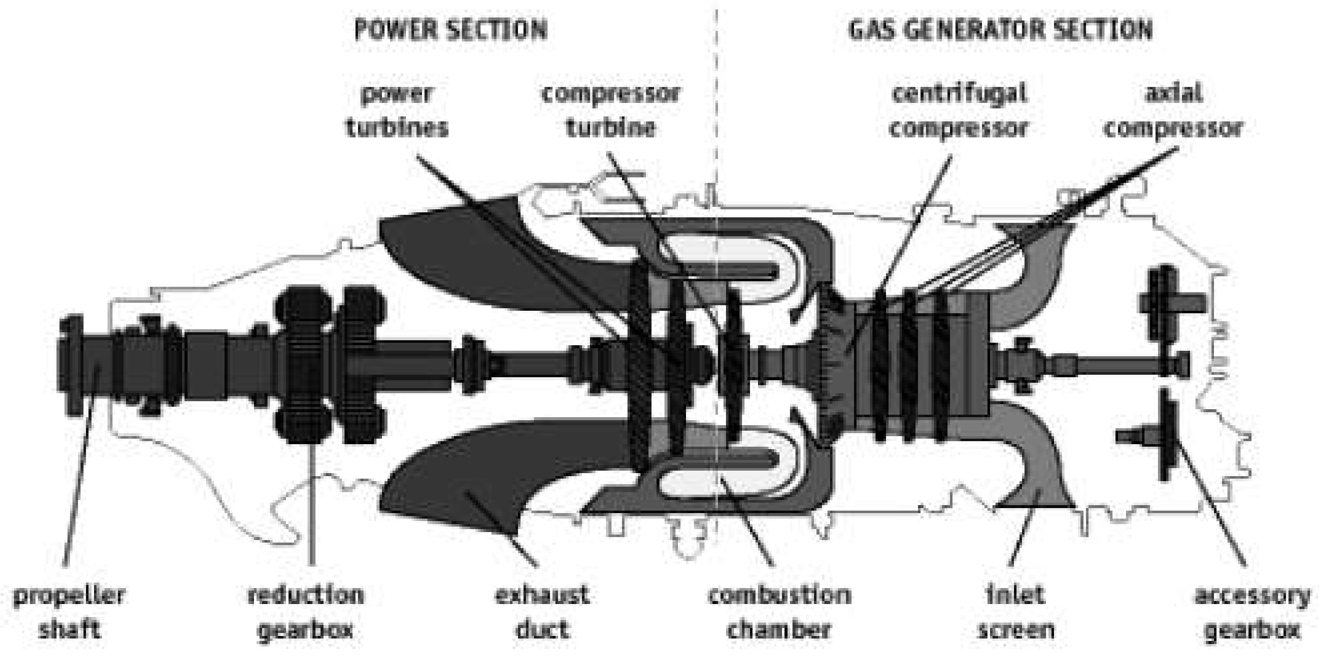

Figure.2.2. PT6A cross section

Smaller gas turbines are rated with a higher operating speed – mainly because of thermo-dynamical reasons. This is done in order to obtain better efficiencies. Due to the material-related maximum circumferential speed of the blades, the nominal speed of gas turbines with higher outputs is limited. Apart from these common models, which are often called 'Industrial Gas Turbines'. These aero-derivative gas turbines come – as the name implies – from the aircraft construction industry, i.e. they have been derived from an aircraft engine and stand out for their more compact design compared to industrial gas turbines which are designed for conventional applications in power stations.

2.2 Characteristics of gas turbine plant on the basis of the converted one power section of PT6T-3D turboshaft engine (PT6A)

Thermodynamic and gas dynamic calculation

The turboshaft engine PT6T-3D which consists of two PT6A power sections is the engine for which studies and calculations of the main characteristics were performed. Algorithms and calculation program are given in Appendix 1.

The engine main components are a compressor which consists of 3-stage axial flow and 1-stage centrifugal compressor, combustors which has Annular reverse-flow with 14 Simplex burners and turbine that has 1-stage gas generator power turbine with 2-stage free power turbine (independent 'free' power turbine with shrouded blades Forward facing output for fast hot section refurbishment). By performing thermodynamic and gas dynamic calculations we have recalculated the performance details of this engine as a gas turbine plant which are the followings:

Initial data:

- Maximum power output is around 670921 kW;
- Overall pressure ratio is 6.3;
- Air mass flow is approximately 2.5 kg/s;
- Gas temperature at turbine entrance 1363 K;

The results of thermodynamic calculation of the basic cycle are given in the Table 2.1 and Table 2.2

Table 2.1

Working body parameters in main gas dynamic station of cycle

	Pressure, Pa	Temperature, K	Specific volume, m^3/kg
Compressor entrance	101300	288	0.815
Compressor exit	638190	486.6	0.218
Turbine entrance	638190	1363	0.612
Turbine exit	101300	806.64	2.285

Table 2.2

Thermal parameters in processes

	Process 1-2	Process 2-3	Process 3-4	Process 4-1
Change in internal energy	$1.425 \cdot 10^5$	$6.288 \cdot 10^5$	$-3.992 \cdot 10^5$	$-3.721 \cdot 10^5$
Change in enthalpy	$1.995 \cdot 10^5$	$8.803 \cdot 10^5$	$-5.589 \cdot 10^5$	$-5.210 \cdot 10^5$
Change in entropy	0	$1.355 \cdot 10^5$	0	$-1.035 \cdot 10^5$
Flow work	$1.995 \cdot 10^5$	0	$5.569 \cdot 10^5$	0

2.3 Characteristics of gas turbine plant with regenerator

By comparing the temperature of the exhaust gas leaving the turbine with the temperature of the air leaving the compressor we have the possibility to use regeneration in this case. Schematic diagram of cycle with heat regeneration and its T-S diagram are shown in Figure 2.3 and Figure 2.4. Increase the efficiency of gas turbine plant can be carried out in various ways [1,7] and in particular due to the thermal perfection of the GTU scheme by introducing regeneration of exhaust gas heat.

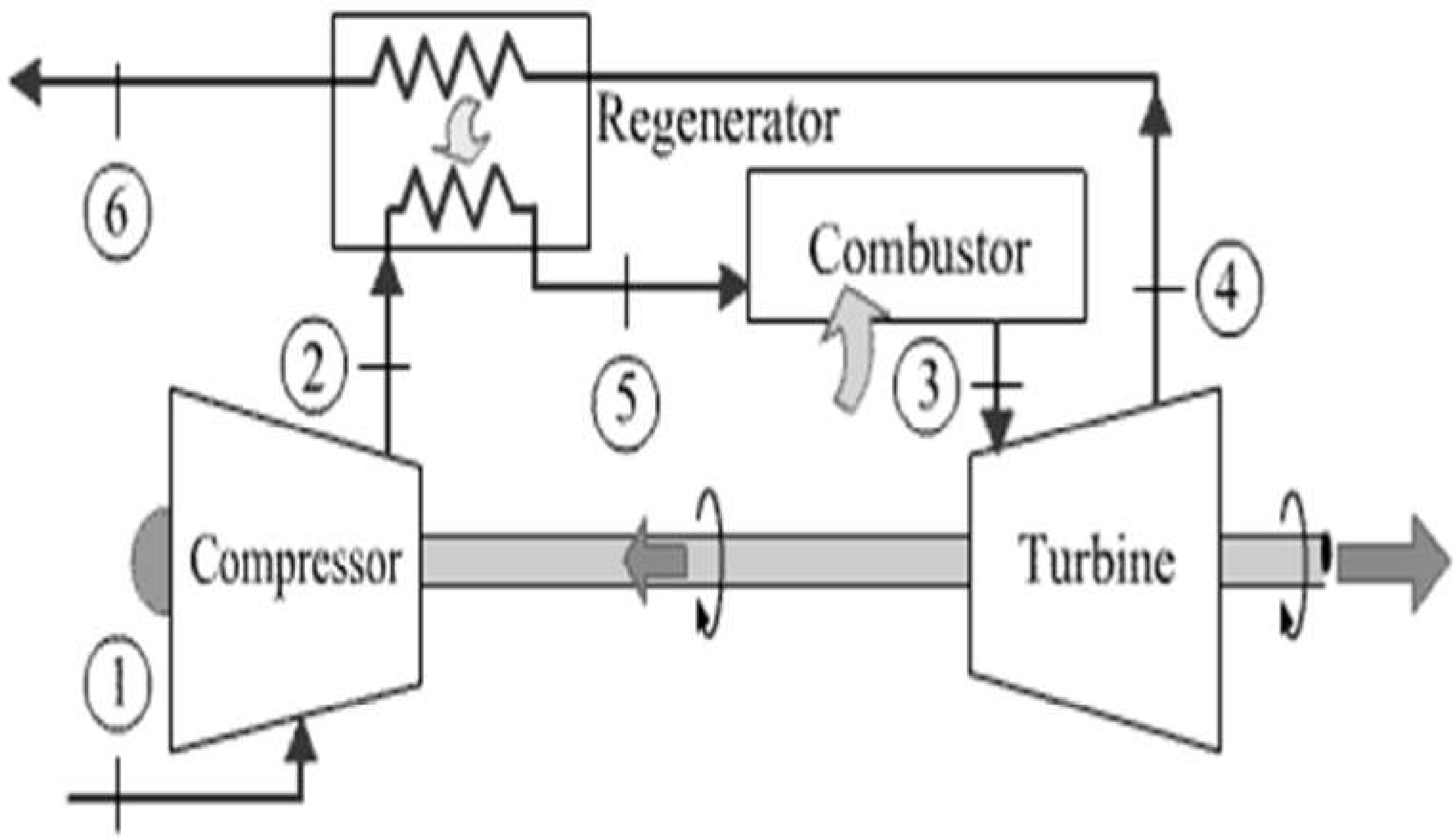

Figure 2.3. Schematic scheme of the regenerative Brayton cycle

The analysis of the cycle parameters (table.2.1) shows the possibility of regenerative using of exhaust gas heat to reduce fuel consumption. In a simple plant the gases leaving the turbine have a temperature of 533°C, the temperature of air leaving compressor is 213°C. The research of the regenerative cycle was carried out in two directions:

– evaluation of the efficiency of heat regeneration applying under the numerical values of the parameters of the existing plant;

–determination of the efficiency of the gas turbine plant operation at different values of the effectiveness of regeneration.

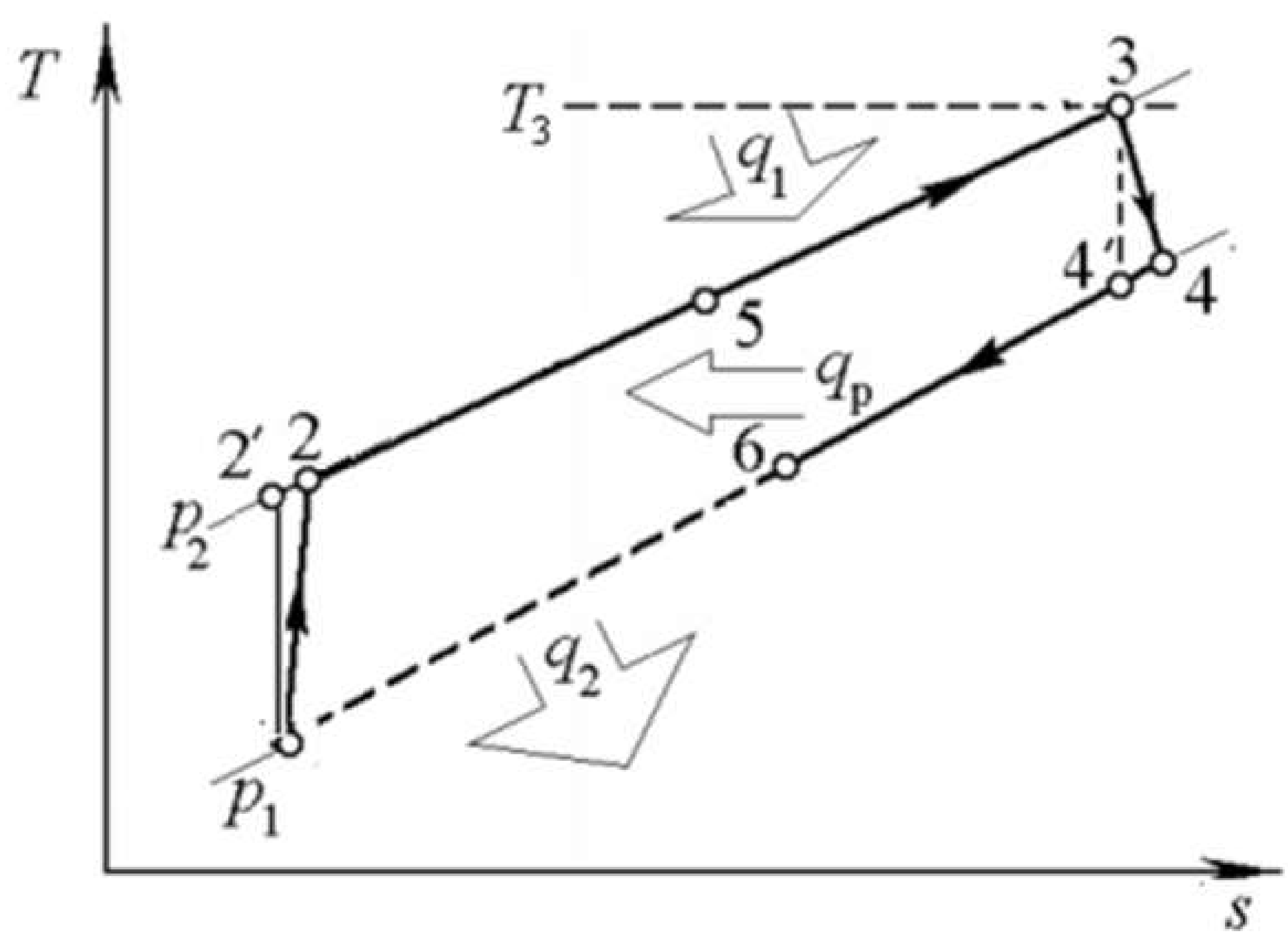

Figure 2.4. T-S diagram of the cycle with regeneration

It should be noted that the difference between T_4 and T_2 reflects only thermodynamically possible relative fuel economy due to regeneration and do not take into account the actual working fluid parameters change during operation. Calculations were carried out for air temperatures of 10; 20 and 30°C and temperatures of combustion products before the turbine of 1100, 1200 and 1300°C. Variation of T_2 with change of ambient temperature and variation of T_4 for different values of temperature before turbine were analyzed. In Figure 2.5 it is shown the change of temperature difference (T_4 - T_2) versus the pressure ratio.

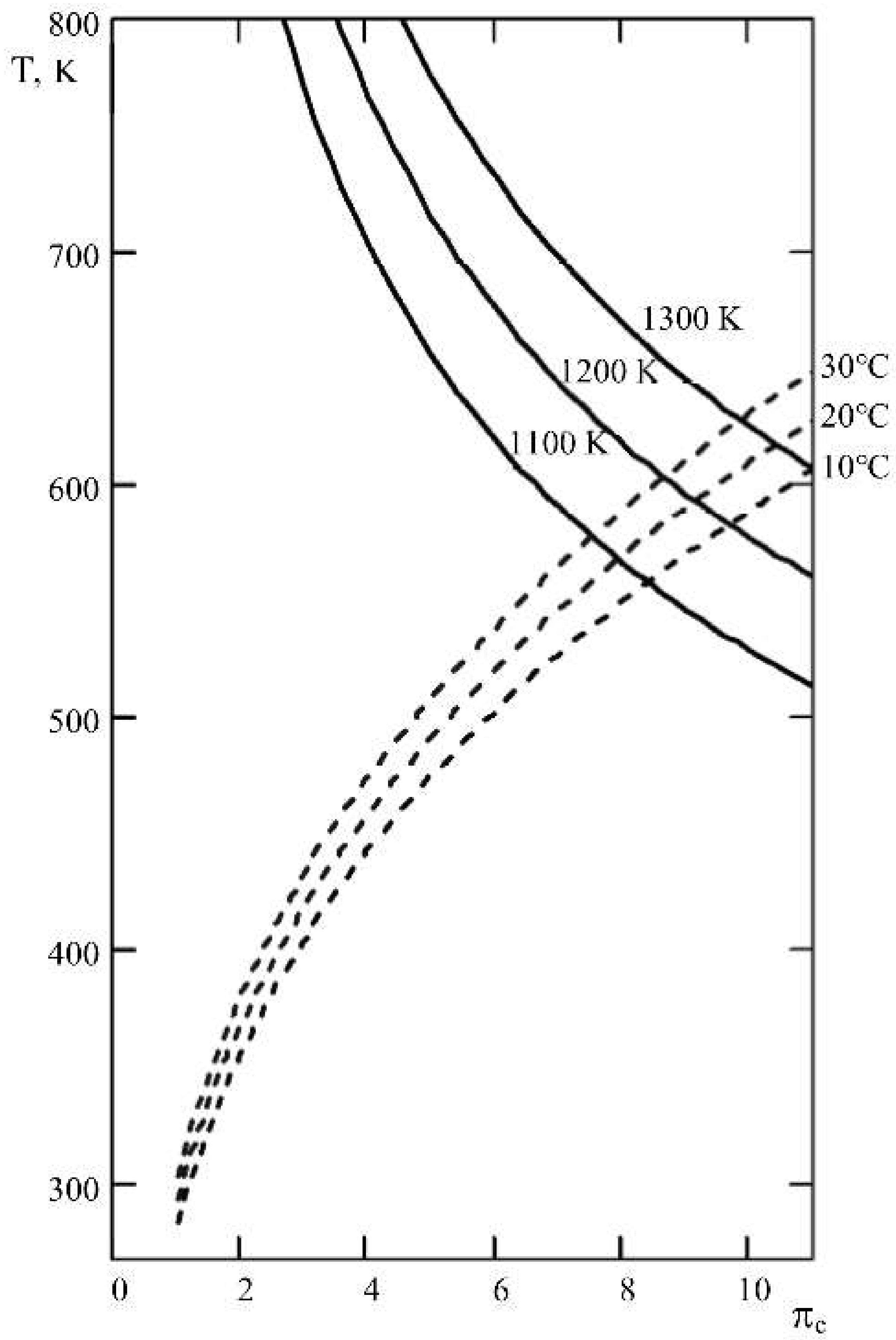

Figure 2.5 Change of temperature difference versus the pressure ratio

The influence of the final temperature of the compression process on η_t makes the task of finding the optimal value of effectiveness of regeneration urgent question.

The cycle was modeled using the thermodynamic analysis for the simple gas turbine. In the studies of the effectiveness and parameters of GTU the following initial data were taken: temperature and pressure of the working fluid at the inlet to the compressor T_1 =288 K and p_1 = 0,10131 MPa, pressure ratio – π_c=6.3, T_3 = 1363 K , η_c = 0,86 – compressor isentropic efficiency, η_t = 0,9 –turbine isentropic efficiency, k – the ratio of specific heats for air, k_g– the ratio of specific heats for gas, σ_r = 0,85– effectiveness of regeneration.

The air which is leaving the compressor is heated by exhaust gases in a counter-flow heat exchanger which is called recuperator and are usually constructed as shell-and-tube type heat exchangers using very small diameter tubes, with the high pressure air flowing inside the tubes and low pressure exhaust gas in multiple passes outside the tubes.

The thermal efficiency increases up to 0.419 due to regeneration since less fuel is used for the same work output.

Table 2.3

Cycle parameters

	Cycle work, J/kg	Heat addition, J/kg	Thermal efficiency
With Regeneration	191893.54	486016.16	0.419
Without Regeneration	191893.54	430881.47	0.32

By drawing the dependence of thermal efficiency and pressure ratio we can see the differences in thermal efficiency η_{th} for the Brayton cycle with and without regenerator and as it is evident in figure2.6 we realize that in lower pressure ratios (approximately from 2 to 6) the differences of thermal efficiency is considerable.

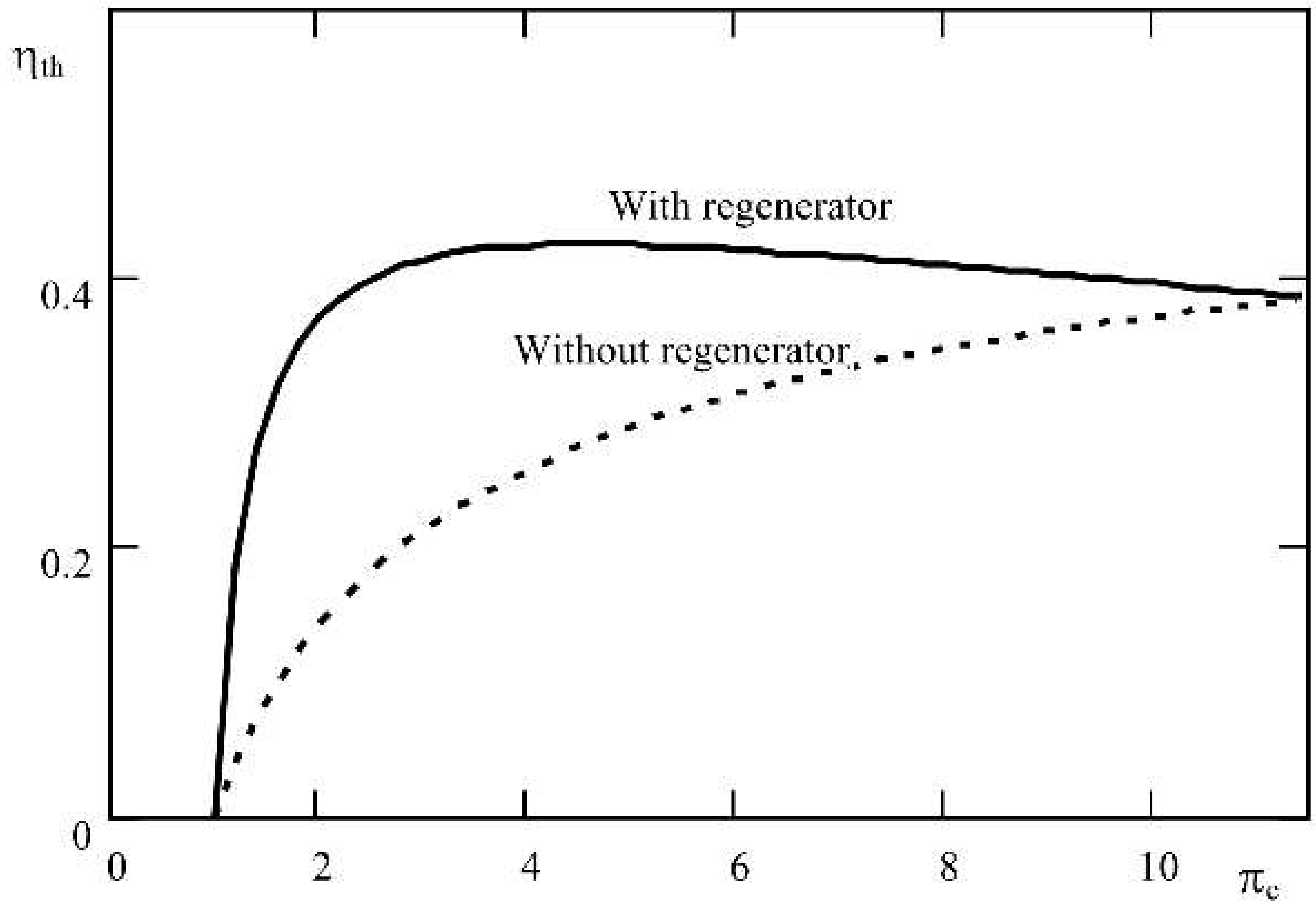

Figure 2.6 Thermal Efficiency vs. Pressure Ratio dependence without regenerator and with regenerator

2.4 Analysis of a regenerative Brayton cycle

For given ambient conditions T_1 and p_1, compressor pressure ratio π_c and turbine inlet temperature T_3, the following procedure can be followed to estimate engine performance [6].

$$p_2 = p_1 \pi_c ,$$

where π_c – pressure ratio , p_1 and p_2 –the compressor inlet and outlet pressure.

Because of the processes irreversibility, entropy generation takes place in processes 1-2 and 3-4, it takes into account by introducing the efficiency of the compressor η_c and turbine η_t.

The isentropic efficiencies of the compressor and turbine are

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1},$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}}.$$

Inlet compressor temperature T_2 and inlet turbine temperature T_4:

$$T_2 = T_1\left[1+\frac{1}{\eta_c}\left(\pi_c^{\frac{k-1}{k}}-1\right)\right];$$

$$T_4 = T_3\left[1-\eta_t\left(1-\pi_c^{\frac{k_g-1}{k_g}}\right)\right].$$

The temperature at the end of regeneration T_5 is defined by the effectiveness of the regeneratorσ_r:

$$T_5 = T_2 + \sigma_r\left(T_4 - T_2\right).$$

The compressor work:

$$w_c = c_p T_1\left(\frac{\pi_c^{\frac{k-1}{k}}-1}{\eta_c}\right),$$

where k –specific heat ratio for air, c_p – isobaric specific heat of air.

The turbine work:

$$w_t = c_{pg} T_3 \eta_t\left(1-\frac{1}{\pi_c^{\frac{k_g-1}{k_g}}}\right),$$

where k_g– specific heat ratio for gas (product of burning), c_{pg} – isobaric specific heat of gas.

The net work:

$$w_{net} = w_t - w_c = c_{pg} T_3 \eta_t\left(1-\frac{1}{\pi_c^{\frac{k_g-1}{k_g}}}\right) - c_p T_1\left(\frac{\pi_c^{\frac{k-1}{k}}-1}{\eta_c}\right)$$

Neglect the heat losses, which are usually small in the regenerator, it can be assumed that the heat given by the gas is equal to the heat received by the air. Heat input and heat output per cycle:

$$q_1 = c_p[(T_3 - T_2) - \sigma_r(T_4 - T_2)],$$

$$q_2 = c_{pg}(T_6 - T_1).$$

The cycle thermal efficiency

$$\eta_{th} = \frac{w_{net}}{q_1},$$

$$\eta_{th} = \frac{c_{pg}T_3\eta_t\left(1 - \dfrac{1}{\pi_c^{\frac{k_g-1}{k_g}}}\right) - c_pT_1\left(\dfrac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c}\right)}{c_p[(T_3 - T_2) - \sigma_r(T_4 - T_2)]}$$

$$\eta_{th} = \frac{c_{pg}T_3\eta_t\left(1 - \dfrac{1}{\pi_c^{\frac{k_g-1}{k_g}}}\right) - c_pT_1\left(\dfrac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c}\right)}{c_p\left[\left(T_3 - T_1\left[1 + \dfrac{1}{\eta_c}\left(\pi_c^{\frac{k-1}{k}} - 1\right)\right]\right) - \sigma_r\left(T_4 - T_1\left[1 + \dfrac{1}{\eta_c}\left(\pi_c^{\frac{k-1}{k}} - 1\right)\right]\right)\right]}$$

Figure 2.7 shows the effect of ambient temperature on thermal efficiency of gas turbine cycle with regeneration and without regeneration. Overall, in lower initial temperatures we have higher thermal efficiency in both cases. The character of the thermal efficiency change calculated at different values of the air temperature before the compressor is shown in Figure 2.7. The graphs show that the thermal efficiency of the cycle with regeneration is more than that of the cycle without regeneration for all temperatures. It is seen that the cycle thermal efficiency decreases with increase ambient temperature. This is due to the increase in compression work. The decrease of thermal efficiency for cycle with regeneration is more than for cycle without regeneration. The difference between thermal efficiency of gas turbine cycle with regeneration and thermal efficiency of cycle without regeneration decreases with air temperature increase.

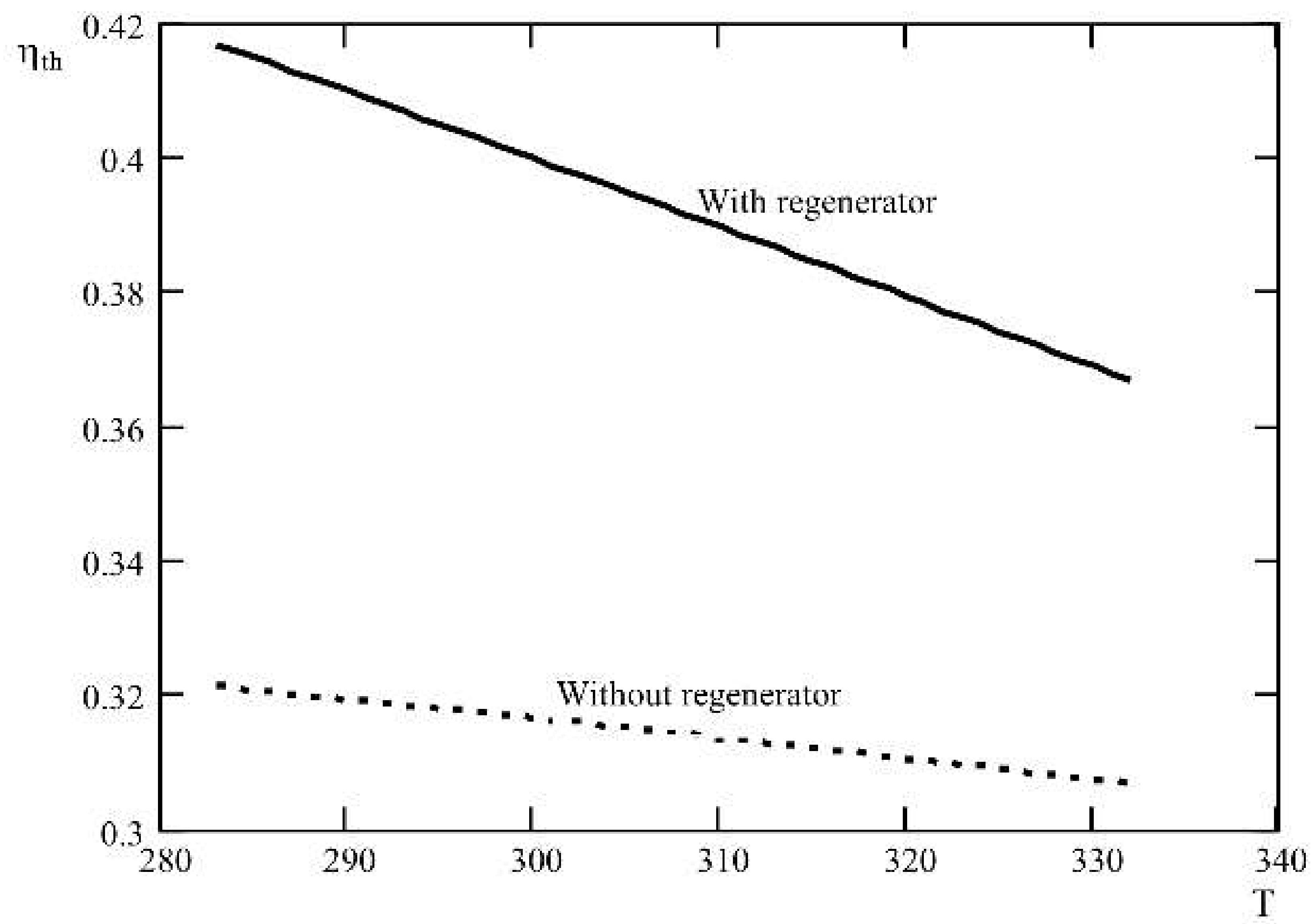

Figure 2.7 Ambient temperature on thermal efficiency dependence.

The effect of operation conditions (ambient temperature) and regenerative effectiveness on the thermal efficiency were analyzed. Results are given in table 1.2.4. Figure1.2.8 illustrates Increase of effectiveness of regeneration results in increase of the thermal efficiency of the gas turbine cycle. The thermal efficiency of gas turbine deteriorates significantly when the air ambient temperature rises.

Table 2.4

regenerative effectiveness on the thermal efficiency

T1	283 K			293 K			303 K		
σ_r	0,60	0,70	0,85	0,60	0,70	0,85	0,60	0,70	0,85
η_{th}	0.38	0.40	0.42	0.3	0.39	0.41	0.37	0.38	0.40

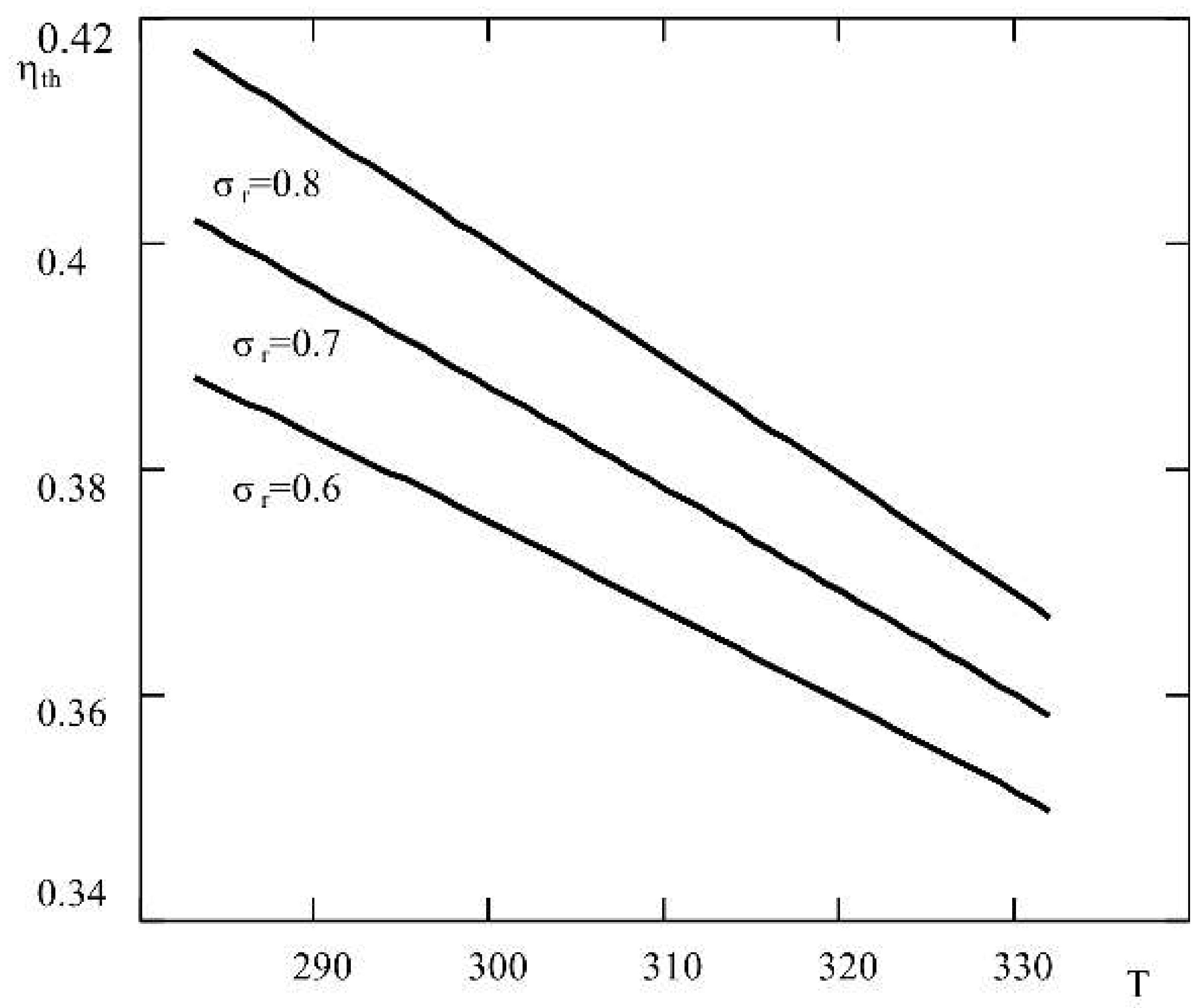

Figure 2.8. Variation of thermal efficiency with change of ambient temperature and regenerative effectiveness.

The relation between **supplied heat** versus compressor pressure ratios for regenerative gas turbine cycle at different values of effectiveness of regeneration depicted in Figure 2.9.

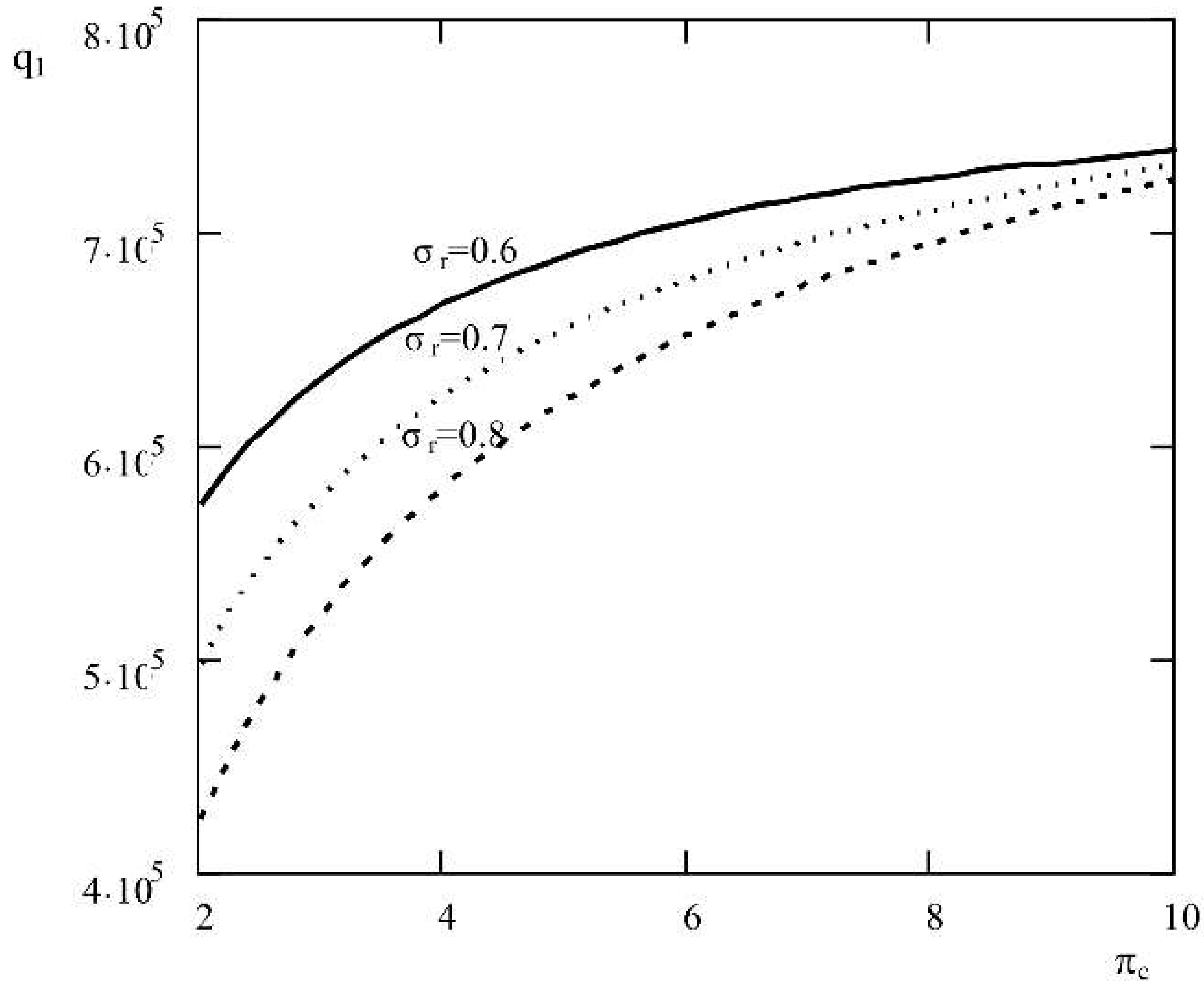

Figure 2.9. Dependence of the supplied heat versus pressure ratio

Figure 2.10 represents that high effective regenerator is much more suitable in lower pressure ratios due to the big differences in thermal efficiency, in our case (π_c=6.3) a high effective regenerator is not economically appropriate.

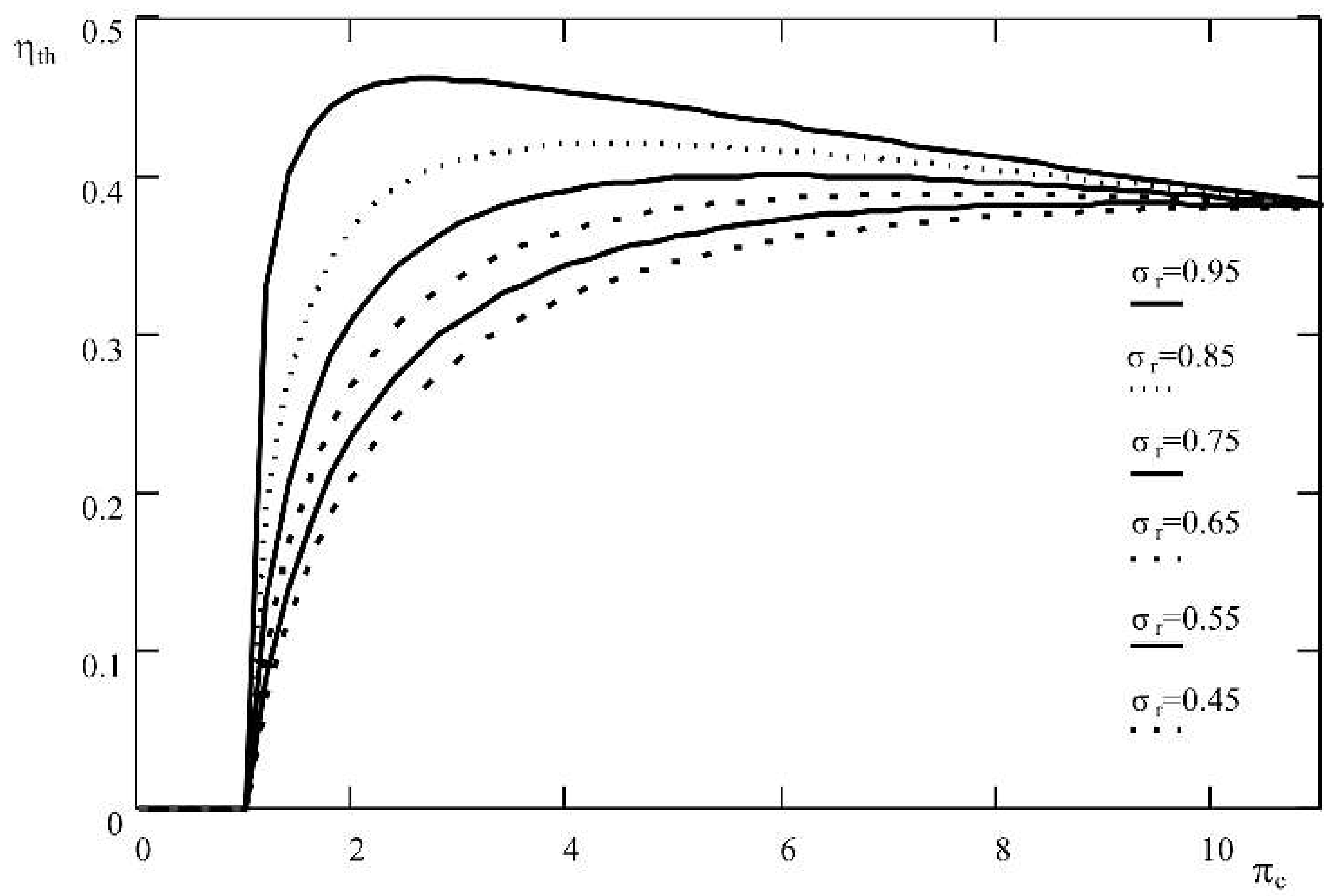

Figure 2.10 Dependence of the supplied heat-compression ratio

Analyzing calculation and the obtained results of the study of gas turbine plants on the basis of the converted engine, we can do the following summary:

- The numerical analysis showed that the turboshaft engine PT6T can be used for mobile power-plant, and to achieve high efficiency parameters it is necessary to add heat regeneration into the design of plant;
- Adding regeneration to basic gas turbine cycle results in the increase of thermal efficiency of the cycle;
- The pressure ratio and inlet air temperature exert a major influence on the thermal efficiency of a basic and regenerative cycles;
- The efficiency of the regenerative cycle increases with the increase of compression ratio to 4.5, then efficiency decreased with increased compression ratio, but in simple cycle the thermal efficiency always increases with increased compression ratio;
- The increase of ambient temperature causes to decrease thermal efficiency, but the increase of turbine inlet temperature increases thermal efficiency;
- The value of parameters should be in the range from 4.5 to 6 for the pressure ratio and from 1100 to 1370 K for the temperature of the gas before the turbine.

Conclusion

One possible way of creation of highly efficient gas turbine plant for mobile terrestrial power generator based on aircraft engines, that exhausted flight service lifetime on aircraft, is shown in this part.

This way assumes application of heat regeneration using the exhaust gas heat to reduce fuel consumption which is necessary to achieve the required level of gas temperature entering the turbine unit.

The calculations(Appendix1), carried out in the work, show that due to the use of the heat regenerator, the thermal efficiency value of a small-scale aviation engine with relatively low working process parameters can increase from 32% up to 42%.

3. PREHEATING OF FUEL

It is known that preheating gasoline or other fuel for internal combustion engines can produce more efficient vaporization and higher combustion efficiency and engine performance than cold fuel. However, preheating the fuel is often not practical because of other problems which may result, such as vapor lock.

3.1 Apparatus and operating method for an internal combustion engine

A method and apparatus for operating an electric ignition, internal combustion engine that substantially improves the fuel efficiency by utilizing heat normally discharged to the ambient to condition and prepare the fuel mixture prior to entry into the combustion chambers. The apparatus comprises a fuel vaporizer that transfers heat from the engine coolant system to the fuel mixture as it leaves a fuel introducing device such as a carburetor; a fuel mixture heater for heating the mixture above the vaporization temperature of the liquid fuel; and, a mixture homogenizer for thoroughly stirring the fuel mixture that is located in the fuel mixture flow path intermediate the vaporizer and heater. The homogenizer is operative to compress the fuel mixture under certain engine operating conditions and the heater forms the intake manifold for the engine and includes branch flow paths and associated conduits that communicate directly with each combustion chamber through a valve controlled port. The fuel mixture flow path from the homogenizer is constructed to minimize energy losses to the ambient. (yunick, 1985)

3.2 Fuel conditioning apparatus and method

A method and apparatus for operating an internal combustion engine that substantially improves the fuel efficiency by utilizing heat normally discharged to the ambient to condition and prepare the fuel mixture prior to entry into the combustion chambers. The apparatus comprises a fuel vaporizer, a fuel mixture heater and a mixture homogenizer located in a fuel mixture flow path intermediate the vaporizer and the heater. The fuel vaporizer includes structure defining an inner heat exchange chamber which receives air and entrained fuel discharged by a fuel introducing device such as a carburetor. The fuel mixture is heated and at least partially vaporized by engine waste heat derived from the engine cooling system or alternately the engine exhaust system. To facilitate the transfer of heat to the fuel mixture, a pair of heat exchange members are disposed in the chamber and include a supply tube defining a flow path for fluid carrying engine waste heat and a plurality of bristle-like heat exchange surfaces radiating outwardly from the supply tube. The bristle-like surfaces are located in heat exchange relation with the fuel mixture in the vaporizing chamber and transfer heat from the heat exchange fluid to the fuel mixture as the fuel mixture passes through the vaporizer. (yunick, 1987)

3.3 Fuel atomisation

Atomisation is the process whereby bulk liquid is transformed into a collection of drops. This transformation goes through the break-up of liquid jet into number of filaments, which in turn transform into droplets.

Disintegration of a liquid jet into a number of filaments, and then into small droplets, requires the surface tension forces of liquid to be overcome. It may happen on the three ways:

- By surface tension between moving liquid jet and steady air which destabilise the jet and causes its disintegration into filaments,
- By centrifugal forces of swirled liquid jet,
- Outer mechanical and electrostatic forces and by supersonic acoustic.

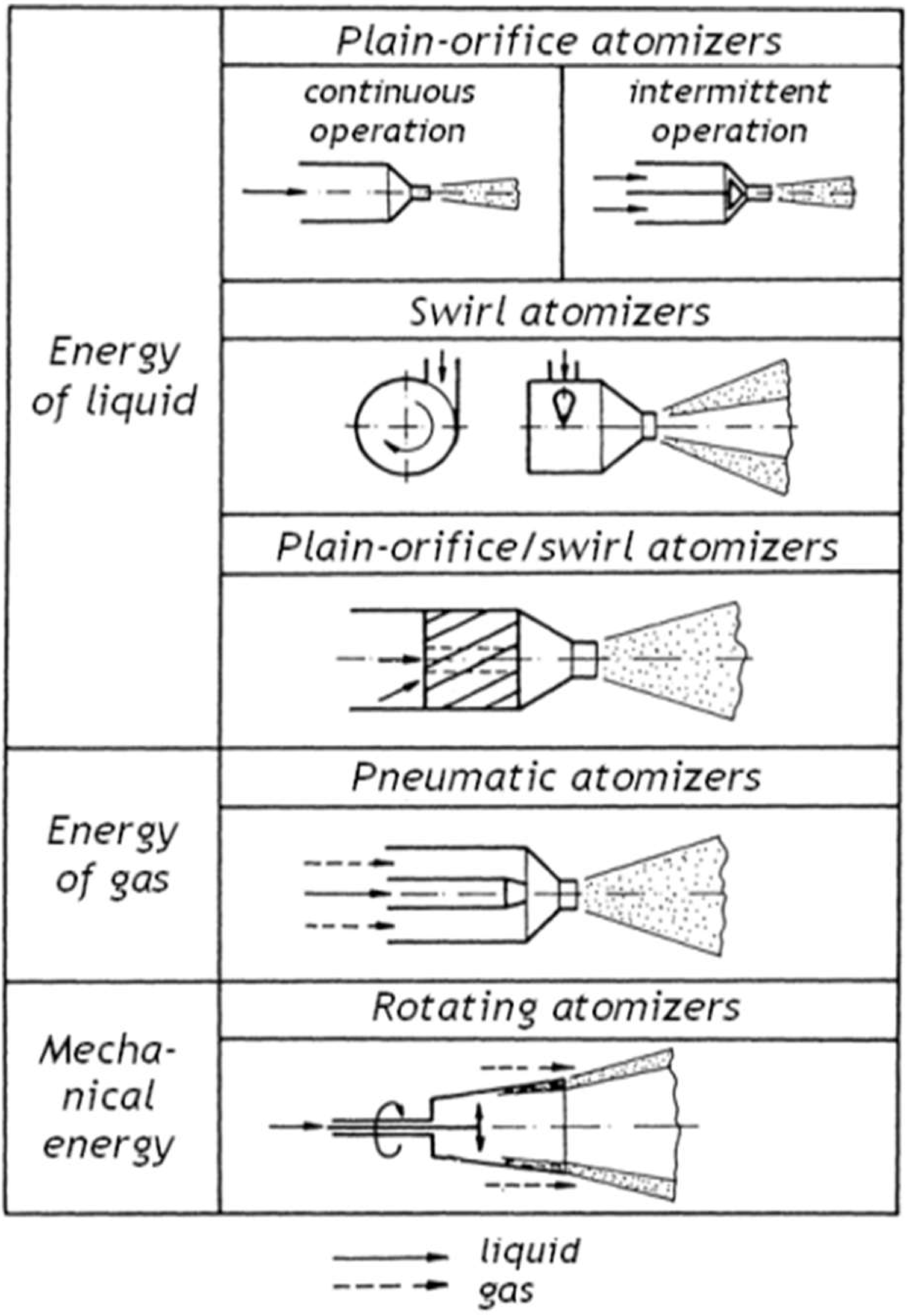

Figure 3.1 - Fuel atomisation with different energy

(Instytut Techniki Cieplnej i Mechaniki Płynów, n.d.)

3.4 Injectors

Atomization or the spray pattern provided by the injector for the engine has a large number of effects on how the engine can use the fuel you provide it. Liquid gasoline is not flammable; gasoline vapor is though.

A small droplet of fuel has a very large surface area compared to its liquid volume, this large ratio of area to volume allows the liquid to be converted to vapor faster by the engine heat than with large liquid droplets.

High performance engines normally rev faster than a production engine this means that the liquid fuel has to be converted to vapor more quickly as there is less time per engine revolution. (ASNU Injectors, n.d.)

"Internal combustion engines" by v. Ganesan

Higher fuel-injection pressures increase the degree of atomization. The fineness of atomization reduces ignition delay, due to higher surface volume ratio. Smaller droplet size will have low depth of penetration due to less momentum of the droplet and less velocity relative to air from where it has to find oxygen after vaporization. Because of this air utilization factor will be reduced due to fuel spray path being shorter. Also with smaller droplets, the aggregate area of inflammation will increase after ignition, resulting in higher pressure rise during the second stage of combustion. Thus, lower injection pressure, giving larger droplet size may give lower pressure rise during the second stage of combustion and probably smoother running. Hence, an optimum group mean diameter of the droplet size should be attempted as a compromise. Also the fuel delivery law i.e., change in the quantity of fuel supplied with the crank angle travel will affect the rates of pressure rise during second stage of combustion though ignition delay remains unaffected by the same. (Ganesan, 2003)

3.5 MathCAD calculation of thermal efficiency with fuel and regenerator in different initial temperature

$p_1 := 101300$ $k := 1.4$ $c_p := 1004.5$ $\eta_t := 0.9$

$r_p := 6.3$ $k_g := 1.33$ $c_{pg} := 1120$ $\eta_c := 0.86$

$T_3 := 1363$ $i := 0..49$

$T_3 = 1.363\times 10^3$ $\sigma_r := 0.8$

$T_3 = 1.363\times 10^3$

$T_1 :=$ READPRN("graph.txt")

$$T_{2_i} := T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1 \right) \right]$$

$T_1 =$

	0
0	283
1	284
2	285
3	286
4	287
5	288

$T_2 =$

	0
0	510.694
1	512.499
2	514.303
3	516.108
4	517.912

$$T_{4_i} := \frac{T_3 \cdot T_{1_i}}{T_{2_i}}$$

$T_4 =$

	0
0	755.304
1	755.304
2	755.304
3	755.304
4	755.304
5	755.304
6	755.304
7	755.304
8	755.304
9	755.304
10	755.304
11	755.304
12	755.304
13	755.304
14	755.304
15	755.304

$$\eta_{thermalregplusPod_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left(1 - \frac{1}{r_p^{\frac{k_g - 1}{k_g}}}\right) - c_p \cdot T_{1_i} \cdot \left(\frac{r_p^{\frac{k-1}{k}} - 1}{\eta_c}\right)}{c_p \cdot \left[\left[T_3 - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right] - \sigma_r \cdot \left[T_{4_i} - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right]\right] - 8177}$$

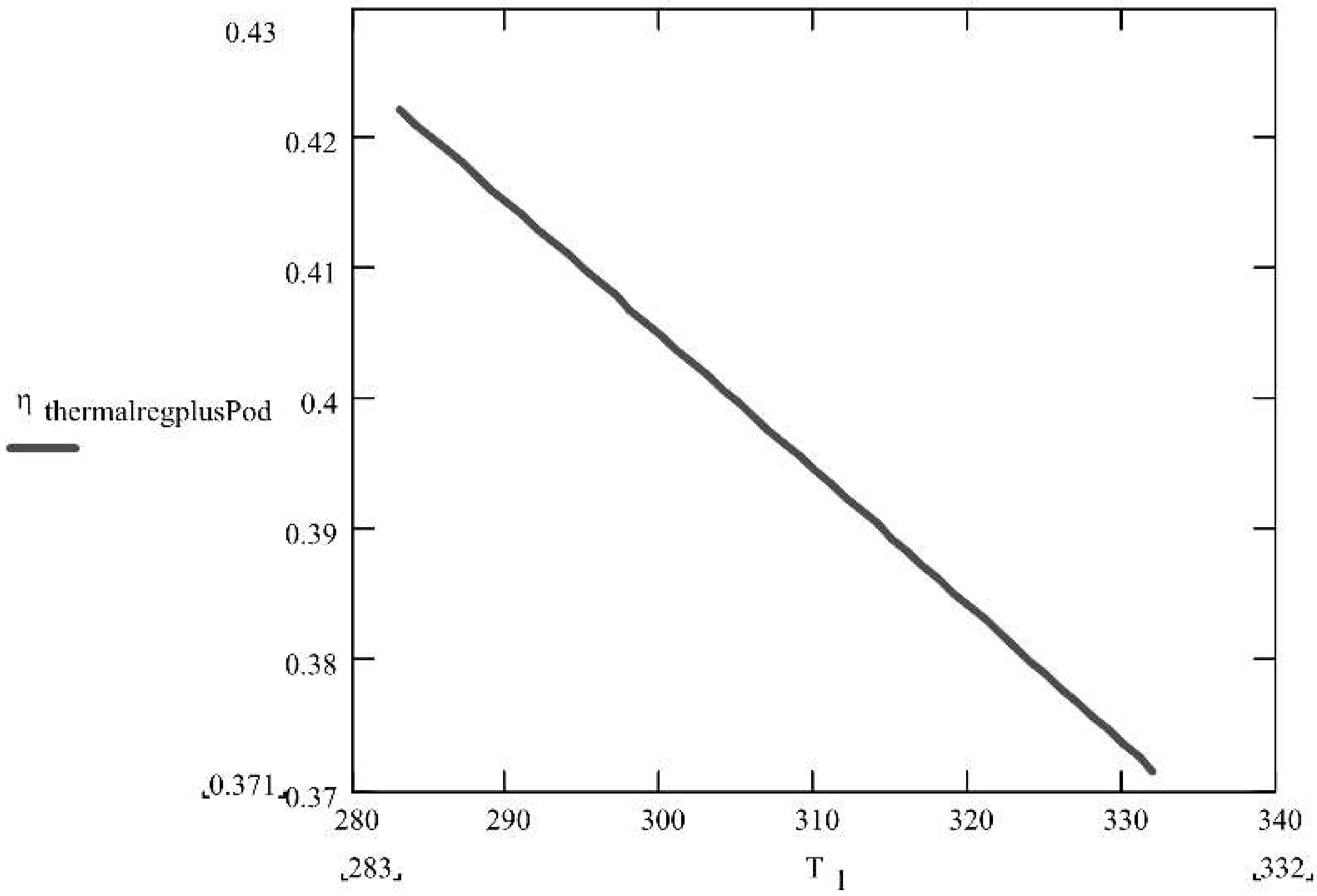

Figure 3.2 thermal efficiency with fuel and regenerator in different initial temperature

Conclusion

In conclusion, preheated fuel has been an area of study for decades, with advances made in the analytical methods used to study the behaviors of the fuels through injection and combustion. The challenges in implementing a preheating solution have been too great thus far, however the research is still continuing.

4. HEAT EXCHANGER CONCEPT

4.1 Concept of heat exchanger

A **heat exchanger** is a system used to transfer heat between two or more fluids. Heat exchangers are used in both cooling and heating processes. [8] the fluids may be separated by a solid wall to prevent mixing or they may be in direct contact. [9]

They are widely used in space heating, refrigeration, air conditioning, power stations, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. Another example is the heat sink, which is a passive heat exchanger that transfers the heat generated by an electronic or a mechanical device to a fluid medium, often air or a liquid coolant.[10]

4.2 flow arrangement

There are three primary classifications of heat exchangers according to their flow arrangement. In parallel-flow heat exchangers, the two fluids enter the exchanger at the same end, and travel in parallel to one another to the other side. In counter-flow heat exchangers the fluids enter the exchanger from opposite ends. The counter current design is the most efficient, in that it can transfer the most heat from the heat (transfer) medium per unit mass due to the fact that the average temperature difference along any unit length is higher. See countercurrent exchange. In a cross-flow heat exchanger, the fluids travel roughly perpendicular to one another through the exchanger.

For efficiency, heat exchangers are designed to maximize the surface area of the wall between the two fluids, while minimizing resistance to fluid flow through the exchanger. The exchanger's performance can also be affected by the addition of fins or corrugations in one or both directions, which increase surface area and may channel fluid flow or induce turbulence.

The driving temperature across the heat transfer surface varies with position, but an appropriate mean temperature can be defined. In most simple systems this is the "log mean temperature difference" (lmtd). Sometimes direct knowledge of the lmtd is not available and the ntu method is used.

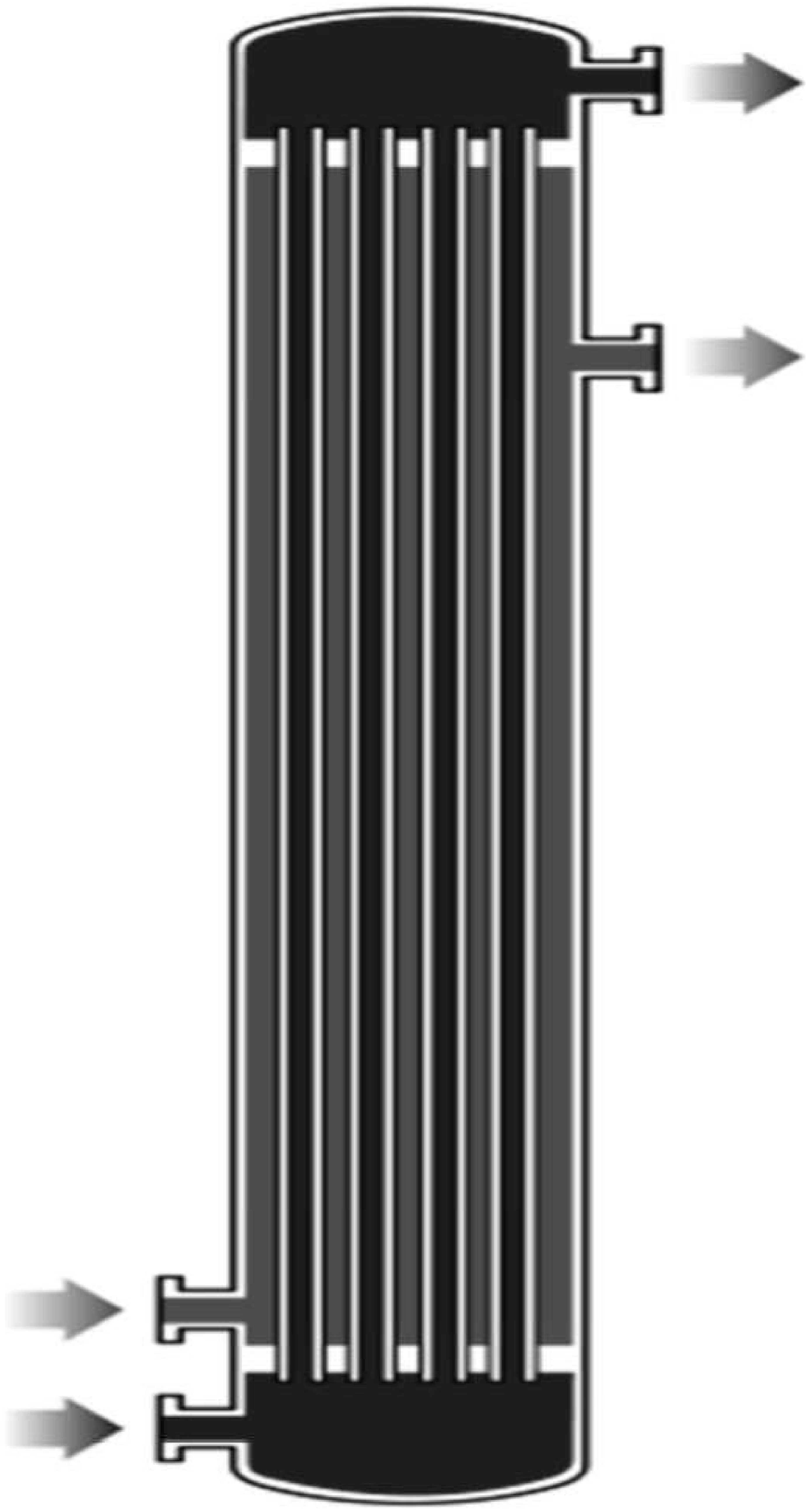

Figure 4.1. Shell and tube heat exchanger, single pass (1–1 parallel flow)

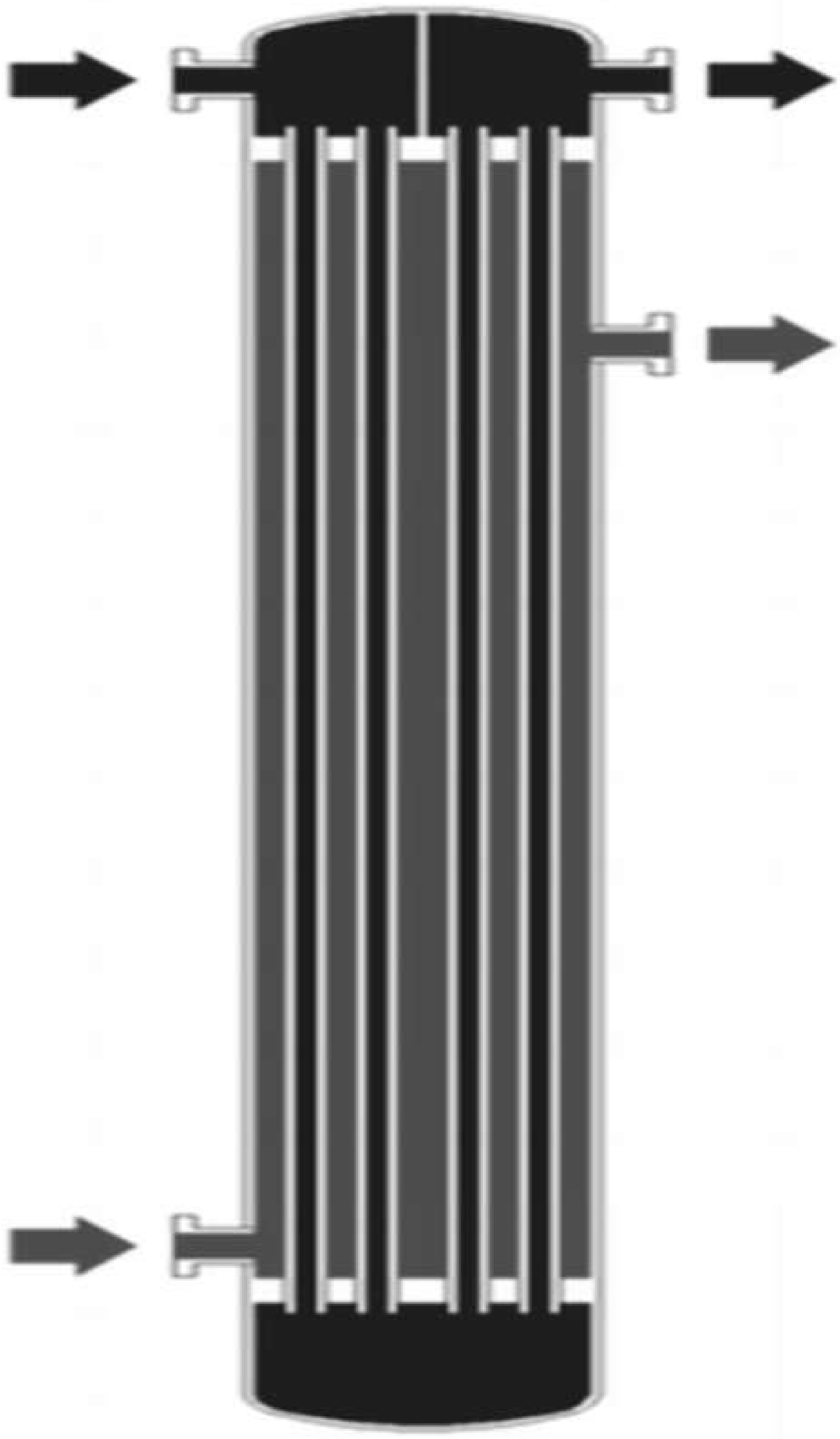

Figure 4.2. Shell and tube heat exchanger, 2-pass tube side (1–2 crossflow)

4.3 shell and tube heat exchanger

Shell and tube heat exchangers consist of a series of tubes which contain fluid that must be either heated or cooled. A second fluid runs over the tubes that are being heated or cooled so that it can either provide the heat or absorb the heat required. A set of tubes is called the tube bundle and can be made up of several types of tubes: plain, longitudinally finned, etc. Shell and tube heat exchangers are typically used for high-pressure applications (with pressures greater than 30 bar and temperatures greater than 260 °c).[11] this is because the shell and tube heat exchangers are robust due to their shape.

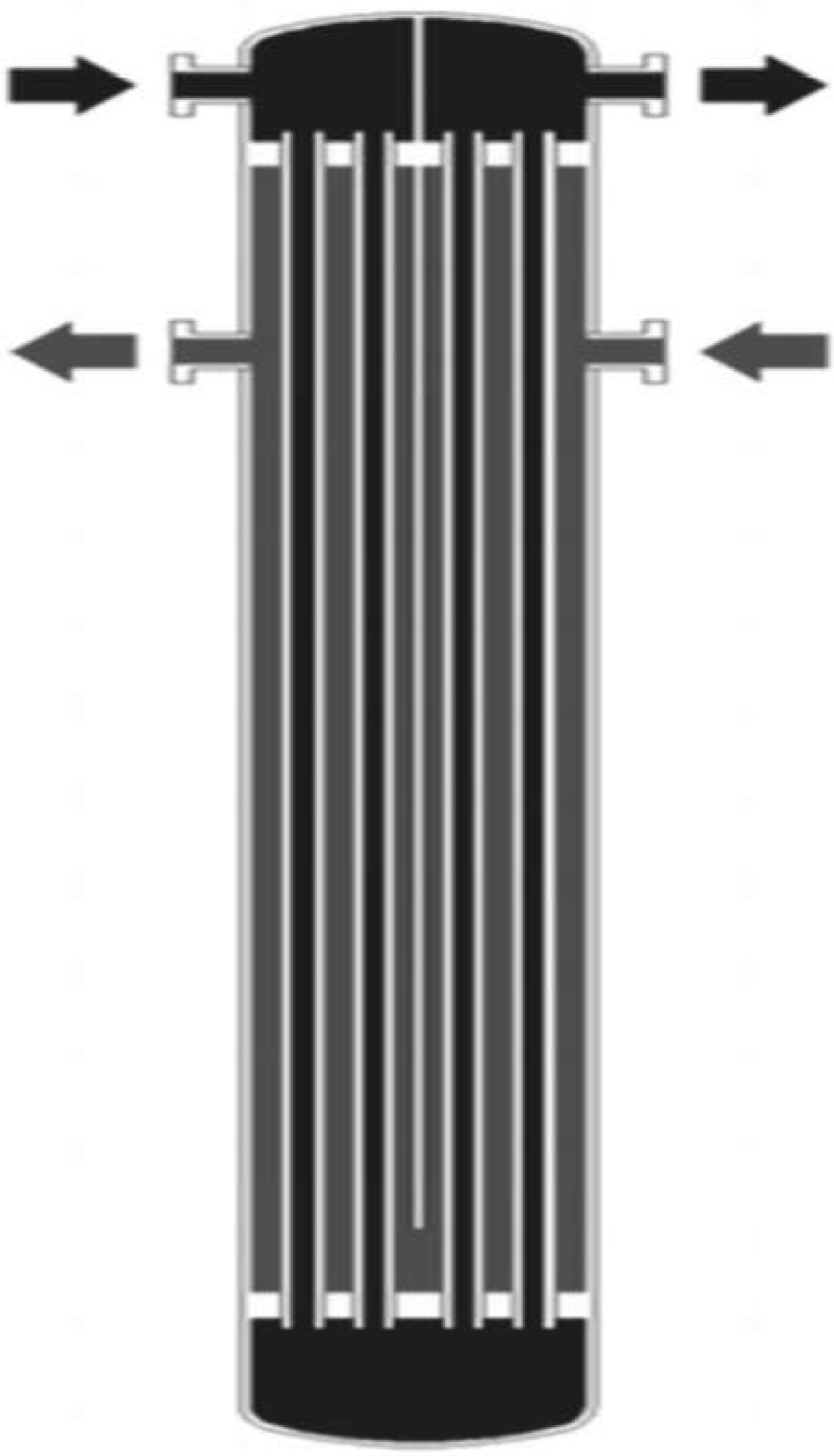

Figure 4.3shell and tube heat exchanger, 2-pass shell side, 2-pass tube side (2-2 countercurrent)

Several thermal design features must be considered when designing the tubes in the shell and tube heat exchangers: there can be many variations on the shell and tube design. Typically, the ends of each tube are connected to plenums (sometimes called water boxes) through holes in tube sheets. The tubes may be straight or bent in the shape of a u, called u-tubes.

- Tube diameter: using a small tube diameter makes the heat exchanger both economical and compact. However, it is more likely for the heat exchanger to foul up faster and the small size makes mechanical cleaning of the fouling difficult. To prevail over the fouling and cleaning problems, larger tube diameters can be used. Thus to determine the tube diameter, the available space, cost and fouling nature of the fluids must be considered.

- Tube thickness: the thickness of the wall of the tubes is usually determined to ensure:
  - There is enough room for corrosion
  - That flow-induced vibration has resistance
  - Axial strength
  - Availability of spare parts
  - Hoop strength (to withstand internal tube pressure)
  - Buckling strength (to withstand overpressure in the shell)
- Tube length: heat exchangers are usually cheaper when they have a smaller shell diameter and a long tube length. Thus, typically there is an aim to make the heat exchanger as long as physically possible whilst not exceeding production capabilities. However, there are many limitations for this, including space available at the installation site and the need to ensure tubes are available in lengths that are twice the required length (so they can be withdrawn and replaced). Also, long, thin tubes are difficult to take out and replace.
- Tube pitch: when designing the tubes, it is practical to ensure that the tube pitch (i.e., the centre-centre distance of adjoining tubes) is not less than 1.25 times the tubes' outside diameter. A larger tube pitch leads to a larger overall shell diameter, which leads to a more expensive heat exchanger.
- Tube corrugation: this type of tubes, mainly used for the inner tubes, increases the turbulence of the fluids and the effect is very important in the heat transfer giving a better performance.
- Tube layout: refers to how tubes are positioned within the shell. There are four main types of tube layout, which are, triangular (30°), rotated triangular (60°), square (90°) and rotated square (45°). The triangular patterns are employed to give greater heat transfer as they force the fluid to flow in a more turbulent fashion around the piping.

Square patterns are employed where high fouling is experienced and cleaning is more regular.

- Baffle design: baffles are used in shell and tube heat exchangers to direct fluid across the tube bundle. They run perpendicularly to the shell and hold the bundle, preventing the tubes from sagging over a long length. They can also prevent the tubes from vibrating. The most common type of baffle is the segmental baffle. The semicircular segmental baffles are oriented at 180 degrees to the adjacent baffles forcing the fluid to flow upward and downwards between the tube bundle. Baffle spacing is of large thermodynamic concern when designing shell and tube heat exchangers. Baffles must be spaced with consideration for the conversion of pressure drop and heat transfer. For thermo economic optimization it is suggested that the baffles be spaced no closer than 20% of the shell's inner diameter. Having baffles spaced too closely causes a greater pressure drop because of flow redirection. Consequently, having the baffles spaced too far apart means that there may be cooler spots in the corners between baffles. It is also important to ensure the baffles are spaced close enough that the tubes do not sag. The other main type of baffle is the disc and doughnut baffle, which consists of two concentric baffles. An outer, wider baffle looks like a doughnut, whilst the inner baffle is shaped like a disk. This type of baffle forces the fluid to pass around each side of the disk then through the doughnut baffle generating a different type of fluid flow.

Fixed tube liquid-cooled heat exchangers especially suitable for marine and harsh applications can be assembled with brass shells, copper tubes, brass baffles, and forged brass integral end hubs.[citation needed] (see: copper in heat exchangers).

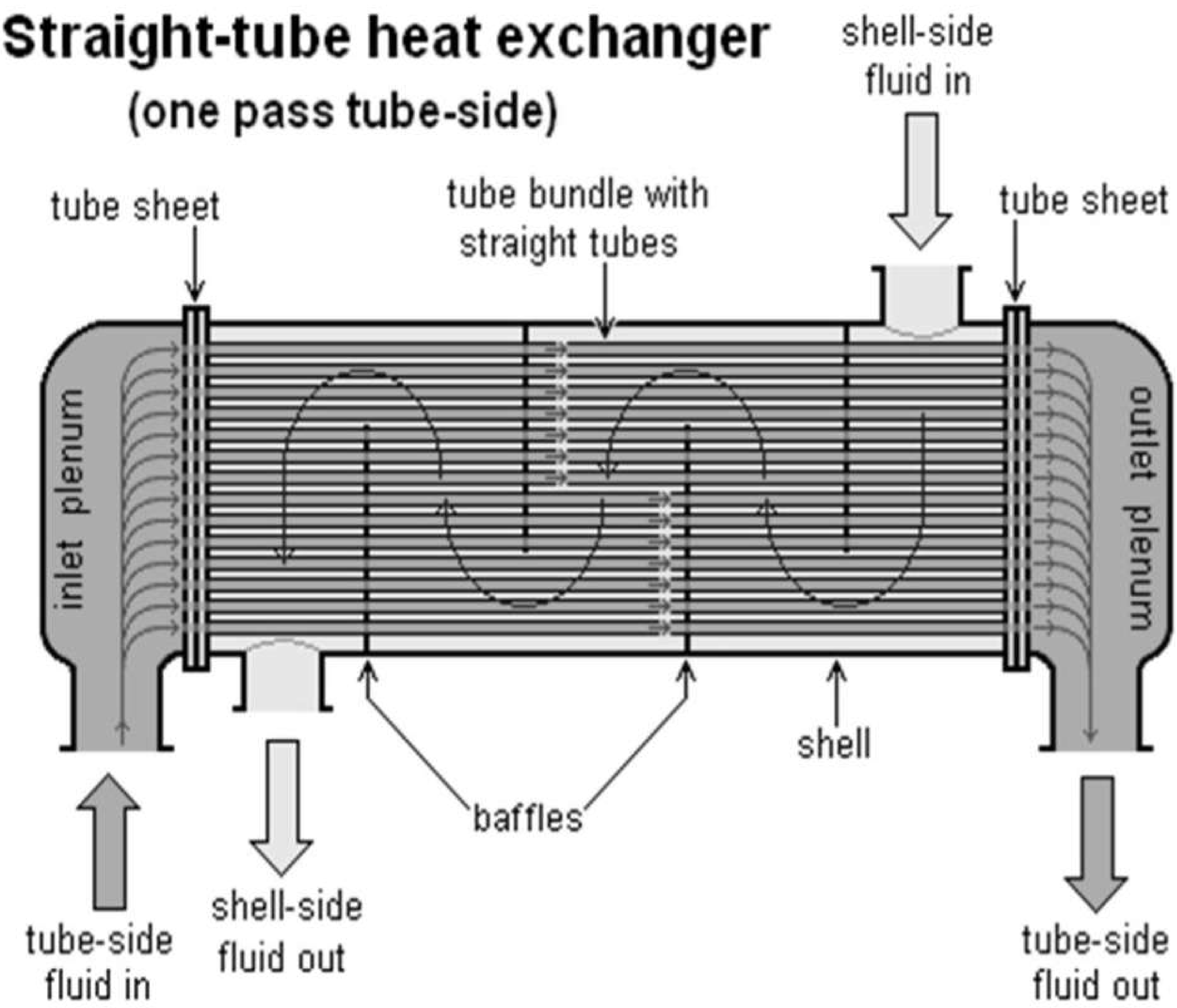

Figure 4.4 a shell and tube heat exchanger

4.4 Plate heat exchangers

Another type of heat exchanger is the plate heat exchanger. These exchangers are composed of many thin, slightly separated plates that have very large surface areas and small fluid flow passages for heat transfer. Advances in gasket and brazing technology have made the plate-type heat exchanger increasingly practical. In hvac applications, large heat exchangers of this type are called plate-and-frame; when used in open loops, these heat exchangers are normally of the gasket type to allow periodic disassembly, cleaning, and inspection. There are many types of permanently bonded plate heat exchangers, such as dip-brazed, vacuum-brazed, and welded plate varieties, and they are often specified for closed-loop applications such as refrigeration. Plate heat exchangers also differ in the types of plates that are used, and in the configurations of those plates. Some plates may be stamped with "chevron", dimpled, or other patterns, where others may have machined fins and/or grooves.

When compared to shell and tube exchangers, the stacked-plate arrangement typically has lower volume and cost. Another difference between the two is that plate exchangers typically serve low to medium pressure fluids, compared to medium and high pressures of shell and tube. A third and important difference is that plate exchangers employ more countercurrent flow rather than cross current flow, which allows lower approach temperature differences, high temperature changes, and increased efficiencies.

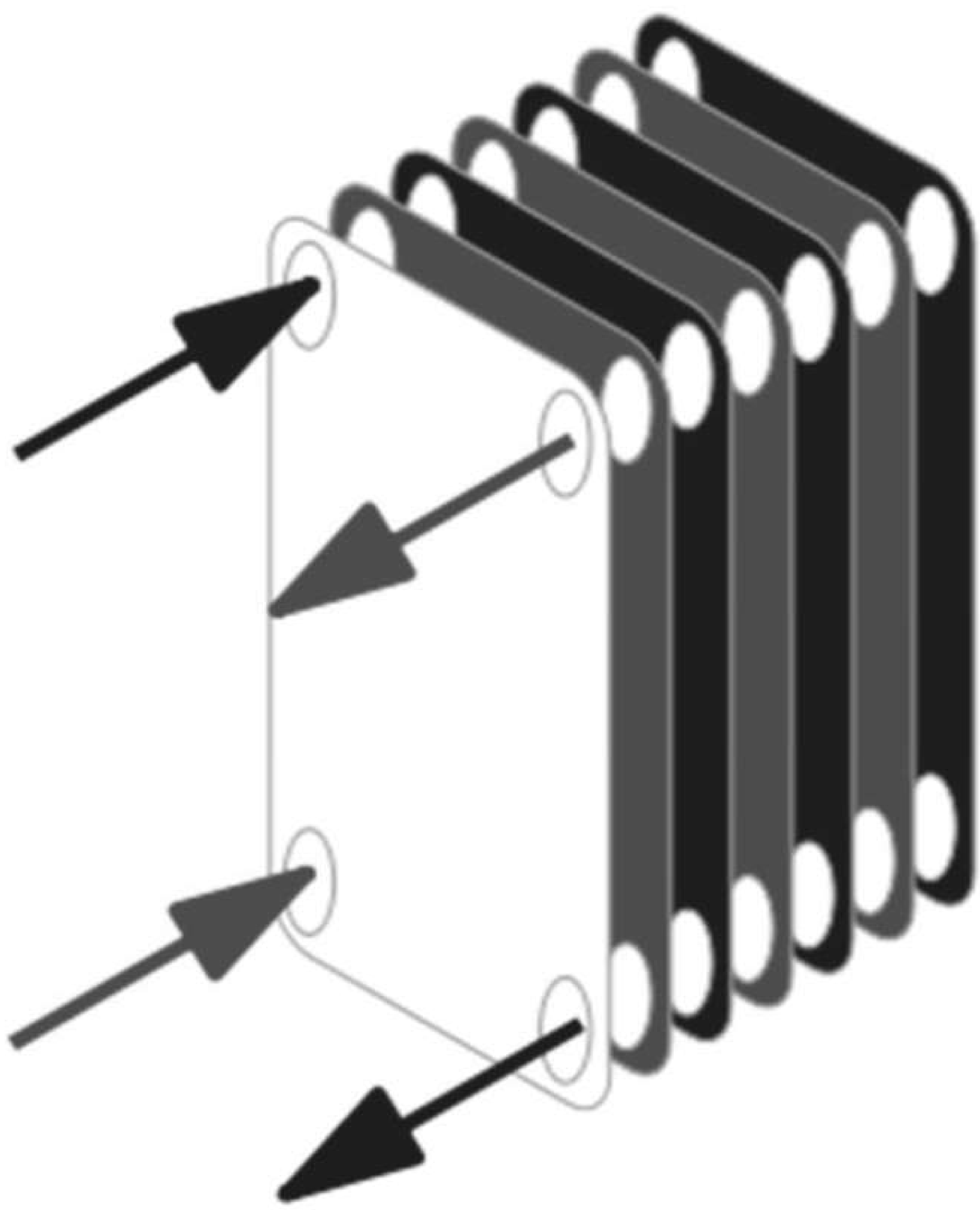

Figure 4.5. Conceptual diagram of a plate and frame heat exchanger.

4.5 Plate and shell heat exchanger

A third type of heat exchanger is a plate and shell heat exchanger, which combines plate heat exchanger with shell and tube heat exchanger technologies. The heart of the heat exchanger contains a fully welded circular plate pack made by pressing and cutting round plates and welding them together. Nozzles carry flow in and out of the plate pack (the 'plate side' flow path). The fully welded plate pack is assembled into an outer shell that creates a second flow path (the 'shell side'). Plate and shell technology offers high heat transfer, high pressure, high operating temperature, uling and close approach temperature. In particular, it does completely without gaskets, which provides security against leakage at high pressures and temperatures.

Heat exchangers are widely used in industry both for cooling and heating large scale industrial processes. The type and size of heat exchanger used can be tailored to suit a process depending on the type of fluid, its phase, temperature, density, viscosity, pressures, chemical composition and various other thermodynamic properties.

In many industrial processes there is waste of energy or a heat stream that is being exhausted, heat exchangers can be used to recover this heat and put it to use by heating a different stream in the process. This practice saves a lot of money in industry, as the heat supplied to other streams from the heat exchangers would otherwise come from an external source that is more expensive and more harmful to the environment.

4.6 Heat exchanger design

An assembled of two heat exchangers that transfer heat from exhaust gas flow to a compressed air after the compressor section and fuel before the combustion chamber include a first heat exchange section and a second heat exchange section located adjacent the first heat exchange section. The first heat exchange section is located within a first housing that at least partially encloses a first fluid volume. The second heat exchange section is located within a second housing that at least partially encloses a second fluid volume. A first plurality of heat exchange tubes traverses the first heat exchange section, and a second plurality of heat exchange tubes traverses the second heat exchange section.

An exhaust gas flow path of the heat exchanger includes the first fluid volume and the interiors of the second plurality of heat exchange tubes. A coolant flow path of the heat exchanger includes the second fluid volume and the interiors of the first plurality of heat exchange tubes.

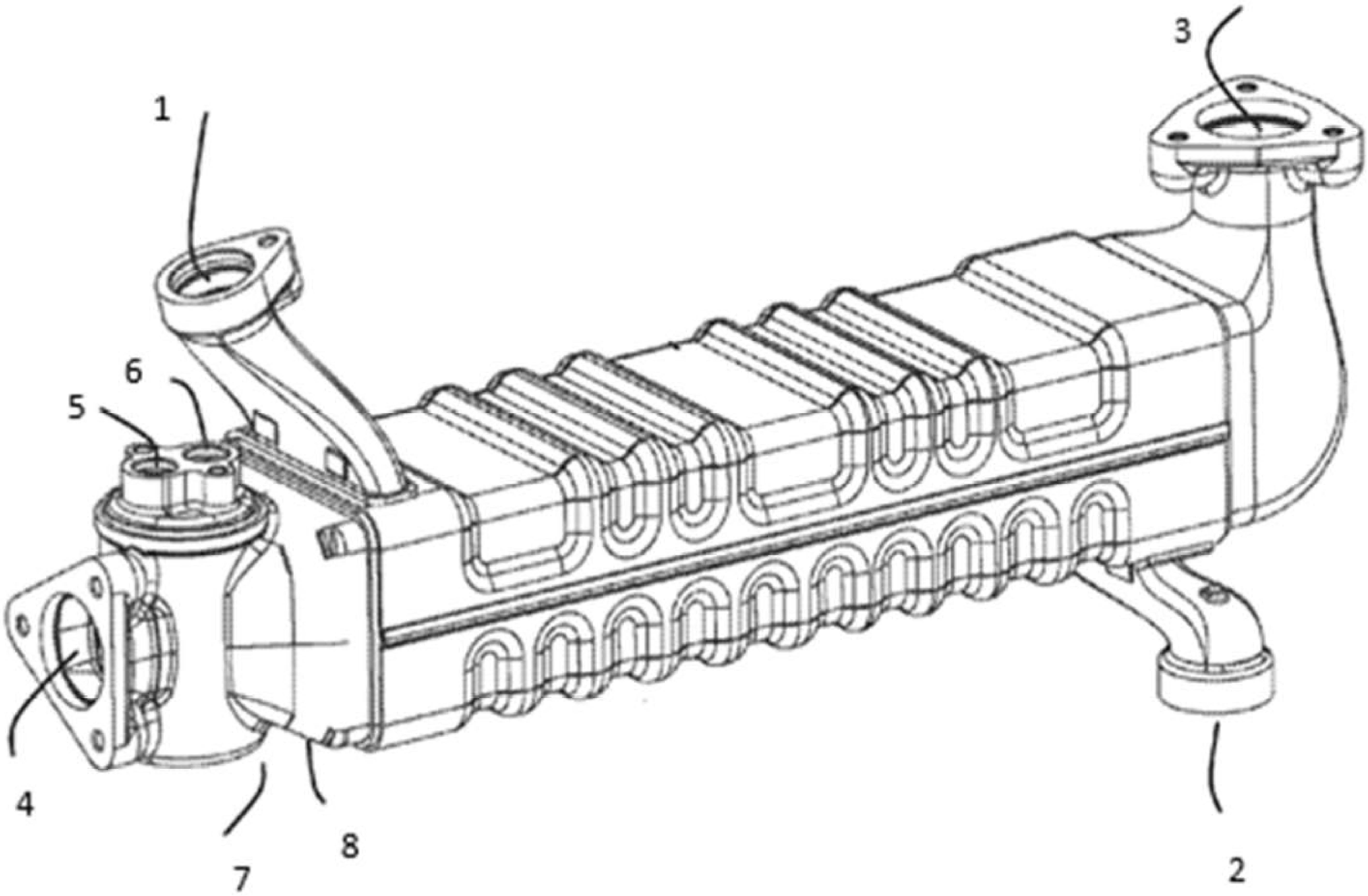

Figure 4.6 schematic diagram of heat exchanger

1.the inlet port of compressed air 2. The outlet port compressed air 3. The inlet port of exhaust gas 4. The outlet port exhaust gas 5,6. The inlet port of fuel 7,8. The outlet port of fuel.

4.6.1 initial data for calculating the counter flow heat exchanger first section exhaust gas-compressed air

$T_4 = 806\ K$ exhaust gas stream inlet temperature

$T_2 = 516\ K$ compressed air stream inlet temperature

$G_2 = 2.5\ Kg/s$ compressed air mass flow

$G_4 = 2.55\ Kg/s$ exhaust gas mass flow

$U = 250\ W/m^2\ K$ overall heat transfer coefficient (gas-gas)

$C_{p.gas} = 1120\ J/kg\ K$ exhaust gas specific heat

$C_{p.air} = 1004\ J/kg\ K$ compressed air specific heat

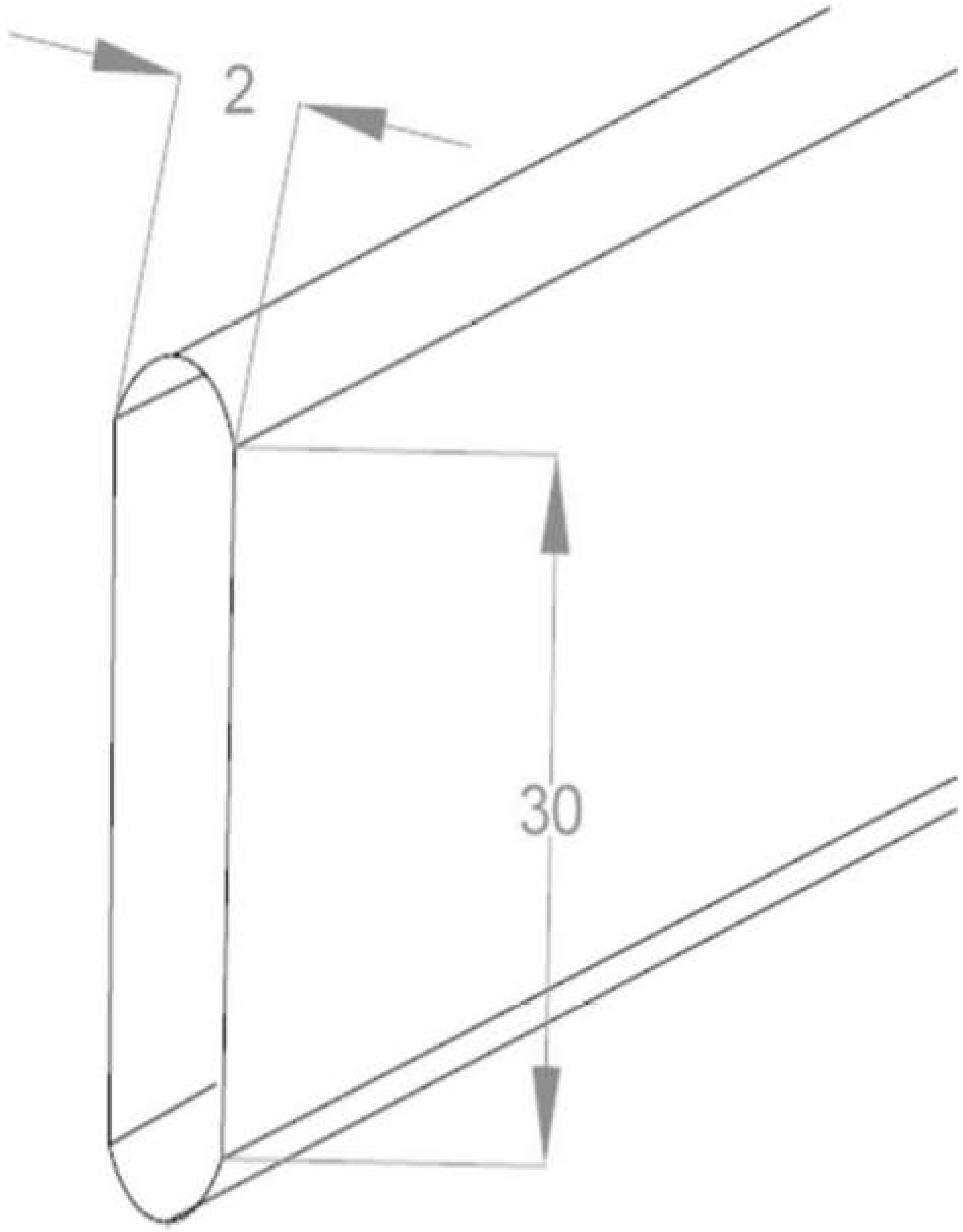

Figure 4.7 perspective view of heat exchanger element

Length of this part is 150 cm

Surface area:

$A = (2.\pi * r^2) + (2 * \pi * l * r) + l * h * 2 = 950 + 9000 = 9950cm^2 = 0.995\ m^2$

Considering that the heat exchanger includes 24 section then:

$A = 1.0909 * 24 = 23.88\ m^2$

$$mCp_a = m_4 * Cp_4 = 2856$$

$$mCp_b = m_2 * Cp_2 = 2510$$

$$C_{min} = min(mCp_4, mCp_2) = 2510$$

$$C_{max} = max(mCp_4, mCp_2) = 2856$$

$$Q_{max} = C_{min} * (T_4 - T_2) = 727900$$

$$Cr = \frac{C_{min}}{C_{max}} = 0.879$$

$$NTU = U * \frac{A}{C_{min}} = 250 * \frac{26}{2510} = 2.378$$

If (cr =1)

$$\Sigma = \frac{NTU}{1 + NTU}$$

Else

Heat transfer efficiency

$$\Sigma = \frac{1 - exp\big(-NTU * (1 - Cr)\big)}{1 - Cr * exp\big(-NTU * (1 - Cr)\big)} = 74\%$$

Total heat transfer

$Q = \sigma * C_{min} * (T_4 - T_2) = 534\ kW$

Compressed air stream outlet temperature

$$T_{2-2} = \frac{Q}{mCp_2} + T_2 = 455\ ℃$$

Exhaust gas stream outlet temperature

$$T_{4-2} = T_4 - \frac{Q}{mCp_4} = 346\ ℃$$

4.6.1 initial data for calculating the counter flow heat exchanger second section exhaust gas-liquid fuel

$T_{4-2} = 620\ K$ exhaust gas stream inlet temperature to second section

$T_{fuel-inlet} = 273\ K$ fuel inlet temperature

$G_{fuel} = 0.0818\ Kg/s$ fuel mass flow

$G_{4-2} = 2.55\ Kg/s$ exhaust gas mass flow

$U = 70\ W/m^2\ K$ overall heat transfer coefficient (gas low pressure-liquid)

$C_{p.gas} = 1120\ J/kg\ K$ exhaust gas specific heat

$C_{p.fuel} = 2600\ J/kg\ K$ fuel specific heat

Surface area:

$A = (2.\pi * r^2) + (2 * \pi * l * r) == 0.05\ m^2$

Considering that the heat exchanger includes 32 tube then:

$A = 0.05 * 32 = 1.6\ m^2$

$$mCp_a = m_4 * Cp_4 = 2856$$

$$mCp_b = m_2 * Cp_2 = 212.68$$

$$C_{min} = min(mCp_4, mCp_2) = 212.68$$

$$C_{max} = max(mCp_4, mCp_2) = 2856$$

$$Q_{max} = C_{min} * (T_4 - T_2) = 73587$$

$$Cr = \frac{C_{min}}{C_{max}} = 0.074$$

$$NTU = U * \frac{A}{C_{min}} = 250 * \frac{26}{2510} = 0.527$$

If (cr =1)

$$\Sigma = \frac{NTU}{1 + NTU}$$

Else

Heat transfer efficiency

$$\Sigma = \frac{1 - exp(-\mathrm{NTU} * (1 - \mathrm{Cr}))}{1 - \mathrm{Cr} * \exp(-\mathrm{NTU} * (1 - \mathrm{Cr}))} = 41\%$$

Total heat transfer

$$\mathrm{Q} = \sigma * \mathrm{C}_{\mathrm{min}} * (\mathrm{T}_4 - \mathrm{T}_2) = 30\ \mathrm{kW}$$

Fuel outlet temperature

$$\mathrm{T}_{\mathrm{fuel-outlet}} = \frac{\mathrm{Q}}{\mathrm{mCp}_2} + \mathrm{T}_2 = 335\ °\mathrm{C}$$

Exhaust gas stream outlet temperature

$$\mathrm{T}_{4-3} = \mathrm{T}_4 - \frac{\mathrm{Q}}{\mathrm{mCp}_4} = 140\ °\mathrm{C}$$

4.7 Patents review

4.7.1 overview of patent no us9777680b2

Review of patent no us9777680b2

Class: f02m26/28 layout, e.g. Schematics with liquid-cooled heat exchangers

Inventors: david janke, viswanath setty, jonathan wattelet, thomas grotophorst.

The essence of invention

According to an embodiment of the invention, a heat exchanger to transfer heat from an exhaust gas flow to a liquid coolant includes a first heat exchange section and a second heat exchange section located adjacent to the first heat exchange section. The first heat exchange section is located within a first housing that at least partially encloses a first fluid volume. The second heat exchange section is located within a second housing that at least partially encloses a second fluid volume. A first plurality of heat exchange tubes traverses the first heat exchange section, and a second plurality of heat exchange tubes traverses the second heat exchange section. An exhaust gas flow path of the heat exchanger includes the first fluid volume and the interiors of the second plurality of heat exchange tubes. A coolant flow path of the heat exchanger includes the second fluid volume and the interiors of the first plurality of heat exchange tubes.

In some embodiments, the first fluid volume is arranged upstream of the interiors of the second of the second plurality of heat exchange tubes along the exhaust gas flow path. In some embodiments the second fluid volume and the interiors of the first plurality of heat exchange tubes are arranged fluidly in parallel along the coolant flow path. In some embodiments, a header plate separates the first fluid volume from the second fluid volume.

In some embodiments the second plurality of heat exchange tubes are flattened tubes defining a tube major dimension and a tube minor dimension smaller than the tube major dimension. In some embodiments the second plurality of heat exchange tubes are spaced apart from one another in the tube minor dimension and individual ones of the first plurality of heat exchange tubes are aligned with spaces between adjacent ones of the second plurality of heat exchange tubes.

In some embodiments, the heat exchanger includes a coolant inlet manifold in fluid communication with inlet ends of at least some of the first plurality of heat exchange tubes, and a coolant outlet manifold in fluid communication with outlet ends of at least some of the first plurality of heat exchange tubes. At least one of the coolant inlet manifold and the coolant outlet manifold are located within a wall section of the first housing. In some embodiments the coolant inlet manifold, the coolant outlet manifold, and the first plurality of heat exchange tubes are part of a replaceable cartridge.

According to another embodiment of the invention, a method of recirculating a flow of exhaust gas includes receiving the flow of exhaust gas, at an exhaust gas inlet temperature above a threshold temperature, into an exhaust inlet of a heat exchanger and directing the flow of exhaust gas through an inlet diffuser of the heat exchanger from the exhaust inlet to open ends of a plurality of exhaust tubes. A liquid coolant is directed through a plurality of coolant tubes arranged within the inlet diffuser in order to maintain the tube walls of the plurality of coolant tubes at a wall temperature substantially below the threshold temperature. The temperature of the flow of exhaust gas is reduced to an intermediate exhaust temperature below the threshold temperature prior to the exhaust gas reaching the open ends of the plurality of exhaust tubes by transferring heat through the tube walls to the liquid coolant flowing through the plurality of coolant tubes. The flow of exhaust gas is directed through the plurality of exhaust tubes, and a liquid coolant is directed over outside surfaces of the plurality of exhaust tubes. The temperature of the flow of exhaust gas is reduced from the intermediate exhaust temperature to a desired exhaust gas outlet temperature by transferring heat to the liquid coolant flowing over the outside surfaces of the plurality of exhaust tubes.

In some embodiments a flow of liquid coolant is separated into a first portion and a second portion. The first portion is directed to flow through the plurality of coolant tubes, and the second portion is directed to flow over the plurality of exhaust tubes.

Emission concerns associated with the operation of internal combustion engines (e.g., diesel and other types of engines) have resulted in an increased emphasis on the use of exhaust gas heat exchangers. These heat exchangers are often used as part of an exhaust gas recirculation (EGR) system, in which a portion of an engine's exhaust is returned to the combustion chambers. Such a system displaces some of the oxygen that would ordinarily be inducted into the engine as part of the fresh combustion air charge with the inert gases of the recirculated exhaust gas. The presence of the inert exhaust gas typically serves to lower the combustion temperature, thereby reducing the rate of NOx formation.

In order to achieve the foregoing, it is desirable for the temperature of the recirculated exhaust to be lowered prior to the exhaust being delivered into the intake manifold of the engine. In the usual case, engine coolant is used to cool the exhaust gas within the exhaust gas heat exchanger (typically referred to as an “EGR cooler”) in order to achieve the desired reduction in temperature. The use of engine coolant provides certain advantages in that appropriate structure for subsequently rejecting heat from the engine coolant to the ambient air is already available for use in most applications requiring an EGR system.

Due in large part to the elevated temperatures of the exhaust gas that they encounter, EGR coolers are known to be prone to thermal cycle failure. The desire for increased fuel economy continues to drive the engine operating temperatures upward, further exacerbating the problem. Above a certain temperature, the material properties of the metals used to produce the heat exchanger rapidly degrade, and the operational lifetime of the heat exchanger is substantially reduced. In order to combat this problem, it often becomes necessary either for the heat exchanger to be produced of more expensive alloys that can withstand these higher temperatures, or to increase the size and weight of the heat exchanger using the current materials, neither of which is desirable. Thus, there is still room for improvement.

4.7.2 overview of patent no us7607301 b2

Review of patent no us 7607301 b2

Class: f02m31/093 air intake passage surrounding the exhaust gas passage; exhaust gas passage surrounding the air intake passage.

Inventors: masaki harada, haruhiko watanabe

The essence of patent

The present invention is made in view of the above disadvantages. Thus, it is an objective of the present invention to address at least one of the above disadvantages.

To achieve the objective of the present invention, there is provided an exhaust gas heat exchanger for a system where a compressing means for compressing intake gas is located upstream of an internal combustion engine, and a part of exhaust gas discharged from the internal combustion engine flows into the intake gas at a merge part located upstream of the compressing means, the exhaust gas heat exchanger including a heat exchanging member. The heat exchanging member is provided adjacently to the merge part, wherein the heat exchanging member exchanges heat between the intake gas and the part of the exhaust gas such that the part of the exhaust gas is cooled by the intake gas.

To achieve the objective of the present invention, there is also provided an exhaust gas recirculation system, which includes a compressing means, a cooling heat exchanger, an internal combustion engine, and an exhaust gas heat exchanger. The compressing means compresses intake gas. The cooling heat exchanger cools the intake gas, which is compressed by the compressing means. The internal combustion engine, to which the intake gas cooled by the cooling heat exchanger is introduced, wherein a part of exhaust gas discharged from the internal combustion engine flows into the intake gas at a merge part located upstream of the compressing means. The exhaust gas heat exchanger is provided adjacently to the merge part, wherein the exhaust gas heat exchanger exchanges heat between the intake gas and the part of the exhaust gas, which is discharged from the internal combustion engine, such that the part of the exhaust gas is cooled by the intake gas.

To achieve the objective of the present invention, there is also provided an exhaust gas heat exchanging method. In the method, a part of exhaust gas discharged from an internal combustion engine is introduced into an exhaust gas heat exchanger provided adjacently to an intake pipe, through which intake gas flows to the internal combustion engine. Heat is exchanged between the intake gas and the part of the exhaust gas by use of the exhaust gas heat exchanger such that the part of the exhaust gas is cooled by the intake gas. The part of the exhaust gas, which is cooled by the exchanging of the heat, is introduced into the intake gas at a merge part in the intake pipe.

Conclusion

Heat exchangers are widely used in industry both for cooling and heating large scale industrial processes. The type and size of heat exchanger used can be tailored to suit a process depending on the type of fluid, its phase, temperature, density, viscosity, pressures, chemical composition and various other thermodynamic properties.

An exhaust gas heat exchanger comprising 2 section can give us compactness and somehow it will be easier to carry out the necessary maintenance it should also be noted that the amount of pressure loosing of compressed air in first section of heat exchanger is around 4 to 5%.

5. LABOUR PRECAUTION

The meaning "labour protection" is determined in article1 of the Ukrainian Law "About Labour Protection". Labour protection is the system of legal, social-economical, organization-technical, sanitary-hygienic and health-prophylactic methods and method that are directed on saving life, health and workability of a human during the work.

The general law of Ukraine that determines the main aspects of labour protection is the Constitution of Ukraine, Labour Code of Ukraine, the Ukrainian law about the labour protection and other normative acts, which regulate the mutual relations between different subjects of the right in sphere of labour precaution.

Ukrainian Law About Labour Precaution defines a regulation for realization of constitutional right of the citizens on protection of their life and health during labour activity, regulates in participation of appropriate State organs the relationships between enterprise proprietor, establishment and organizations or its representative organ (further - proprietor) and worker in safety questions, working hygiene and occupational environment and installs single organization order of labour precaution in Ukraine.

Labour precaution is based on following foundation-stone of State policy:

- priority of life and health of the workers in relation to results of their occupational activity at enterprise, total responsibility of the proprietor for creation of safe and non-hazard working conditions;
- social protection of the workers, full compensation of harm to persons, which suffered from accidents on production and occupational diseases;
- establishment of single standards in labour precaution for all of enterprises, irrespective of property forms and appearances of their activity;
- use of economic management methods in labour precaution, taking of policy of preferential taxing, that contributes to creation of safe and harmless conditions of work, State participation in financing of arrangements in labour precaution;

- teaching of the population, their professional preparation and upgrading in labour precaution questions.

The object of preventive labor is the shop, and the subject is an engineer.

In diploma work, in accordance with the assignment, an analysis of possible ways to improve the design of the free turbine assembly of the projected gas turbine plant in order to increase its efficiency and reduce the cost of it manufacturing was made.

5.1 Dangerous and harmful production factors that arise during operation and maintenance of the designed GTP

The working process of mobile gas turbine plant mainly is outdoor and due to the less limit of work place of GTP labor precaution plays an important rule to carry the work out in safe conditions.

Work of the mobile GTP is as follows:

– operate the GTP in reserve, in the stage of start-up and stop;

– check all auxiliary technological equipment of GTP

– checking of all existing electrical equipment that are used;

– elimination of small faults.

Checking and operation of equipment and mechanisms are made by circumvention by developed and approved routes.

In the operation and maintenance of gas turbines these dangerous and harmful production factors can act on the workers:

A) Noise (MPL 75 dB).

The consequences: functional disorders from a number of systems and human organs, namely: central nervous system, cardiovascular system (changes in blood pressure, pain in the heart, etc.), organs of hearing (hearing loss, the occurrence of hearing loss and deafness, etc.), digestive disorders (loss of appetite, occasional heartburn and nausea, etc.), the vestibular apparatus (recurrent dizziness, feeling of instability, etc.), agencies of view (deterioration of stability of clear vision), the muscular system (change of muscle strength, decreased muscular work and so forth.).

B) A mixture of hydrocarbons (hazard class 4, MPC 300 mg /).

The consequences: disease (caused by industrial and professional); poisoning, including acute, i.e. for one shift; increase injuries; reduced productivity.

C) Oil (hazard class 3, MPC 5 mg /).

The consequences: disease (caused by industrial and professional); poisoning, including acute, i.e. for one shift; increase injuries; reduced productivity.

D) The lack of natural light in the workplace, lack of artificial lighting. (SN 150-300 lux);

The consequences: loss of productivity, not only manually, but also mental.

E) Dangerous electric current values in a network.

The consequences: defeat worker by electrocuted.

5.2 Organizational and structural and technical arrangements to reduce the impact of dangerous and harmful factors

Based on the list of identified dangerous and harmful factors is necessary to develop specific measures that exclude the manifestation of industrial dangers and harmful factors or restrict them within acceptable norms.

Technical and organizational measures to reduce the maximum level of dangerous and harmful factors can be achieved during the development of this project through the implementation of the following below listed technical and organizational measures.

A) Activities of Protection Against noise.

To reduce the impact of excessive noise we must use ordinary cotton wool. It is made as swabs and is inserted into the ear. Also section of the compressor station, where there is the gas turbine, the special noise absorbing material is put. To meet the norms of noise we can use such document [34].

B) Activities of Protection Against hydrocarbons mixtures and oil vapor.

When working with toxic substances and special liquids there is required use of personal respiratory protection and skin.

Increased fumes indoors of auxiliary systems installation for pumping of gas eliminated by organizing forced-air ventilation.

The requirements for ventilation systems:

– The amount of fresh air must match the number to be removed; the difference between them should be minimal.

– Supply and exhaust systems should be properly placed. Fresh air must be supplied to those parts of the premises where the minimum amount of harmful emissions, and to remove, where there is maximum emissions.

– The ventilation system must not cause hypothermia and overheating of staff.

– The ventilation system must not create noise in the workplace exceeding the maximum permissible limits.

– The ventilation system must be electrical, fire, and explosion-proof, simple in design, performance and reliability in operation.

Remote control is defined in the block and is located zone of defeat at the likely destruction of the engine and supercharger, which makes the opportunity to protect personnel from a number of harmful and dangerous factors of destruction and supercharger drive parts in their destruction. In addition, turbine and pumping compartments is separated by concrete wall and locked in the protective cover, which also acts as a thermal screen and noise stopping design elements.

C) For protection from the electric current impact.

To eliminate electric shock during maintenance, equipment of auxiliary systems (electric motors, electric pumps) must be grounded.

According to the requirements of the Rules for Electrical Installation resistance of protective grounding at any time of the year should not exceed 4 ohms -Voltage up to 1000 V.

For the work with electrical equipment that generates electromagnetic fields more than 1500 V / m, working overalls is issued and the time spent in a room with such equipment is monitored. All methods must corresponds to [35].

D) In order to ensure the required level of illumination of premises.

In the dark and with little natural light, artificial lighting facilities of auxiliary systems and external mercury lamps are used. To highlight the hard reaching places a portable lighting device is used. Also, to meet the norms of illumination we can use such document [36].

5.3 Fire and explosive safety

Intensive growth in production, transportation and storage of natural gas is significantly aggravated the problem of fire and explosion precautions on objects of gas industry. The fire and explosive safety regulates with the help of such normative document as [37]. Leakage of large amounts of natural gas in the crash leads inevitably to the formation of explosive and fire hazardous gas-air mixture.

Especially danger is the accumulation of electrostatic charges in gas jet leak through the holes, non-tightness, and ruptures.

The main hazardous factor of fire at its occurrence is an intense heat release in the combustion zone in the form of high temperature and powerful thermal radiation.

Even more dangerous are the following fires at compressor stations and other gas facilities in which it is possible accumulation large amounts of fuel gas because of its leakages, the rapid filling premises space by gas with the formation of explosive mixtures. In the majority of such cases fire can begin with an explosion or lead to a powerful explosion at an early stage of its development.

The main source of fire danger on GCU is a gas turbine engine, which uses natural gas as fuel. Also one of the main reasons of emergency situations appearance leading to fire and explosion on CS is damage of gas pipes unity, breaks of drive and supercharger elements, damage of oil pipes. All these factors cause the natural gas exhaust in station apartment or hot oil under high pressure leakage. At presence of inflammation source (hot surfaces of combustion chambers and exhaust collectors, electric flashes) the fire occurs. At absence of inflammation source in the danger explosive gas-air or oil-air mixture is formed. The explosion of such mixture leads to serious consequences.

Supercharger is also a source explosion hazard, as its working fluid is natural gas(the main component is methane near 95%. Possible gas losses in case tightness of joints or other reasons, creating an explosive mixture of air with the gas concentration at the latest by 15%.

As in compressor shop there are the superchargers, so it belongs to category A according fire and explosion hazard.

For fighting fires can use different extinguishers, hand and arranged in wheelchairs: carbon dioxide – ОУ-2,3,5, ВВ-25; powder – ОП-2, ОП-2Б, ОПС-2, ОПУ-2(5)-02, ОПА-100. In addition to hand there is the automatic fire extinguishing system.

Automatic fire extinguishing developed based on the analysis of possible fire situations and provides fire protection and engine compartment blower through early detection point and fire suppression following it through feeder fire extinguishing substance (as in work units, and in finding it in reserve or repairs). Filing carried during activation fire sensors remotely - from starting a fire signal device located in a compartment or carrier-automatic and manual start-up pen compartment fire.

Fire extinguishing system includes the aggregate of, pipelines or sprinkler of discharge nozzles. Of aggregate includes two batteries БАГЕ - 4 – 1, БАГЕ - 2 - 1 universal signaling pressure of each compartment and the protected electroarc gauges, located in an isolated compartment with fireproof walls and ceilings with fire below 0,75.

During work of system automatically alarm fire in compartment comes from sensors to signal fire starting device that gives impetus to undermine squib head section electrostart-up main charge batteries and automatic control system for an emergency stop unit. After head-gates, Opened rope head electrostart-up, fire extinguisher substance (CO2) comes from cylinders in the pipeline and through a check valve in the irrigation of nozzles.

When using the unit impulse at undermining squib batteries engine compartment seems delayed 15 ... 20 seconds. This is because the engine compartment hood is equipped with mechanical ventilation and to prevent the release of substances in the exhaust fire extinguisher substance mine due to ejection, it is necessary, first of all, turn off fans.

When you turn on the remote system by pressing compartment in automation or fire the starting signal device seems to pulse in automated management for emergency stop driving motor and undermining squib head section electrostart backup battery. At admission fire extinguisher matter pipeline works warning assembly issuing a control signal to the signal fire starting device. For electric pressure gauge pressure remote control is performed in each of the cylinders.

In the Compressor shop is prohibited:

– to lay temporary electric networks;

– to use the case of machines, pipelines and metal structures of buildings as the grounding of electric welding and welding devices;

– to dry the clothes in the devices of central heating, hot surfaces units and gas piping;

– block up the passages and exits from the premises, and approaches to fire extinguishers and exterior fixed ladders;

– work in hazardous locations in the shoes with steel shoes and steel nails;

– use open flame for thawing pipes, constipation devices and other equipment;

– to carry out electric welding work in violation of regulations in force, and instructions;

– carry out any work associated with the replacement and repair of armatures for oil wires and dismantling regulation details (except replacement of manometers) when running the unit.

For quick elimination of emergency and clear interaction is essential that all staff know their specific responsibilities and actions in the event of fire. To do this regularly primary trainer classes on fire suppression, the model foci for the occurrence of which must be specified in the instructions to eliminate fires in shops, buildings and other areas of the station.

In case of fire production staff must:

– immediately block access of gas or oil to the fire;

– call the fire department;

– to take measures to extinguish the fire extinguishing tools that are available;

– inform the leadership of the compressor plant;

– turn off the ventilation system.

In accordance with applicable laws the operator shall be issued such funds of individual protection:

– cotton suit

– tarpaulin boots

– insulating gloves - duty;

– insulating mask - duty;

– headphones earplugs - duty or liners like "Ear Plugs".

Conclusion

According to this part we can consider that the labor precaution is important enough which can have great impact on the operation of GTP directly and indirectly.

In this part we have considered:

– chosen labor precaution subject – driver and object –Mobile gas turbine plant;

– listed harmful and dangerous factors that influence on working personnel during compressor shop work and given some recommendations for reduction of harmful impact on human health;

– described fire and explosion prevention rules and given recommendations how to avoid dangerous situations during work at compressor shop.

6. ENVIRONMENTAL PROTECTION

6.1 International requirements for emission of harmful substances into the environment

Convention on transboundary air pollution over long distances (LRTAP) was signed in Geneva in 1979. To fulfill obligations regarding strategies and policies in accordance with the protocols that were signed in the framework of the Convention by the member states is needed to provide data about emissions of the Executive Body of the Convention. This includes protocols:

– Helsinki Protocol on Sulfur (1985);

– Sophia Protocol on NOx (1988);

– Montreal Protocol on substances destroying the ozone layer (1989);

– The Geneva Protocol on VOC (volatile organic compounds) (1991);

– Oslo protocol on sulfur (1994).

United Nations Framework Convention on Climate Change.

The Convention was adopted in June 1992 at the World Summit in Rio de Janeiro and entered into force in March 1998. The main purpose of the Convention is "stabilization of greenhouse gas concentrations in the atmosphere at a level that prevents dangerous anthropogenic interference with the climate system."

6.2 The main documents of Ukraine concerning the emission of harmful substances

The main documents in decision of questions protection and preservation of the purity of the air are the law of Ukraine on protection of the atmosphere and the adoption of the Cabinet of Ministers, which determine responsibility departments, enterprises and organizations for the development of standards for maximum permissible concentrations of harmful emissions into the atmosphere.

Maximum single and daily maximum permissible concentration (MPC) have the main importance, which are not perceptible to the human reflex systems (some values of main pollutants are mentioned in table 3.1).

Table 6.1

MPC of some components in the atmosphere

Compound	Chemical formula	MPC on workplace	$MPC_{m.s.}$ mg /m^3	$MPC_{c.c.}$ mg /m^3
Nitrogen Dioxide	NO_2	9	0,085	0,04
Nitric Oxide	NO	30	0,6	0,06
Carbon Monoxide	CO	20	5	1
Sulfur Dioxide	SO_2	---	0,5	0,05
Hydrogen Sulfide	H_2S	---	0,008	0,008
Soot	C	3,5	0,15	0,05
Non-toxic dust	----	---	0,5	0,15
3,4-Benzopiren	$C_{20}H_{12}$	0,00015	---	0,000001
Ozone	O_3	---	0,16	0,03

6.3 The composition of the combustion products

The process of burning fuel, which is the basis GTE workflow, is accompanied by the emission of exhaust gases into the atmosphere. These gases are a mixture of combustion products with excess air.

There are complete and incomplete combustion. In the process of complete combustion of all the chemical energy of fuel turns into heat, there is no chemical ($q_3 = 0$) and mechanical ($q_4 = 0$) incomplete combustion. In combustion products, besides the nitrogen and excess oxygen, there is combustible oxides elements of higher orders (CO_2, H_2O, SO_2). The composition of the products is called theoretical combustion.

In the process of incomplete combustion ($q_3 > 0$ та $q_4 > 0$)to the theoretical composition the products of incomplete combustion is added in the form of: CO, H_2, CH_4 (if natural gas is burned); C та $C_x H_y$, and also CO (if solid and liquid fuels is burned).

Depending on the ratio of fuel and oxidant (usually - clean air or air that is ballasted by combustion products) distinguish stoichiometric and non-stoichiometric combustion.

The basis of burning - oxidation reaction of combustible components of fuel which is resulted in the original substance (fuel and oxidant) become components (combustion products) with other physical and chemical properties. A characteristic feature is the burning process that occurs quickly, and is accompanied by intense heat release, the sharp rise in temperature and the formation of hot combustion products with different degrees of luminosity.

The process of burning gaseous fuels can be divided into two stages: the first - form a combustible mixture (a mixture of fuel and air); second - its heating, ignition and combustion. A more complex process of burning liquid fuels. The initial stage is heat, spraying and evaporation of fuel. Droplets and a pair of fuel droplets mix with the air and fuel mixture evaporates, ignites and burns. The composition of combustion products and quantitative ratio of individual components they depend on the fuel properties and composition, as well as the completion of the combustion reaction. You can verify this on the example of burning natural gas, which, as you know, primarily consists of the sum of hydrocarbons C_nH_m. Besides hydrocarbons C_nH_m, in natural gas coming from wells, contain some amount of moisture H_2O and hydrogen sulfide H_2S. Natural gas before supplying it to the consumer is cleaned, removing moisture and hydrogen sulfide.

Generalized equation of combustion of hydrocarbon with the formation of the theoretical composition of the combustion products is:

$$C_nH_m + \alpha(n+\frac{m}{4})O_2 + \alpha(n+\frac{m}{4})(\frac{100}{\psi_{O_2}} - N_2) =$$

$$= nCO_2 + \frac{m}{2}H_2O + (\alpha-1)(n+\frac{m}{4})O_2 + \alpha(n+\frac{m}{4})(\frac{100}{\psi_{O_2}} - 1)N_2,$$

where α - the excess air ratio; n, m - the number of atoms in a molecule of a particular hydrocarbon; ψO_2 - the volume fraction of oxygen in the oxidizers, %. As a result of combustion of hydrocarbons $C_n H_m$ one molecule of the fuel components is formed n molecules of carbon dioxide CO_2 i m /2 molecules of water vapor H_2O. These components belong to the so-called greenhouse gases, increasing the concentration of which determines the change in conditions of thermal equilibrium on the surface of the Earth. In addition, in areas with large concentrations of gross emissions of water vapor can be negative consequences associated with the local change of the microclimate.

Besides the complete combustion products CO_2 and H_2O, nitrogen N_2 and excess oxygen O_2 in smoke gases may contain incomplete combustion products as carbon monoxide CO, that in everyday life is called carbon monoxide, hydrogen H_2 and unburned hydrocarbons C_xH_y ozone of the toxic products of incomplete combustion is formaldehyde CHOH. The latter is characterized by the toxicity that hundreds of times greater than the toxicity CO.

Among the many reactions that occur in the combustion zone, the so-called intermediate reactions that result can form soot C and aromatic hydrocarbons are very toxic, the main representative of which is benzopyrene $C_{20}H_{12}$.

Sooty particles get into space in the form of aerosols and their with increasing concentration negative effect on respiratory organs.

Among intermediate reactions in the combustion zone dissociation reactions also occur. For example, the reaction of oxygen dissociation:

$$O_2 \rightarrow 2O,$$

which is resulted in one molecule formed of two atoms of oxygen.

Oxygen atoms O are characterized by high chemical activity and, faced with a generally neutral nitrogen molecule N_2 at high temperatures promote the reaction of the so-called thermal oxide NO:

$$N_2 + O \rightarrow NO + N .$$

In the result of this reaction active nitrogen atom N is formed. It reacts with oxygen to formation of additional nitric oxide and reactive atomic oxygen:

$$O_2 + N \rightarrow NO + O.$$

In Combustion products the corresponding amount of nitrogen oxidesis formed, the concentration of which depends on many factors and mainly on the temperature level in the combustion zone. Therefore, these oxides are called thermal. The theory of their formation was first developed by J. B. Zeldovich.

In the process of burning liquid and solid fuels, like from natural gas, components products of complete combustion CO_2 and H_2O are formed, that are greenhouse gases and sulphur oxides SO_2 and partially SO_3. Sulfur oxides are also greenhouse gases, but the greatest environmental hazard is the high toxicity and ability to form acid rain.

The nature of the formation of sulfur oxides when burning liquid and solid fuels are different.

During the combustion of liquid fuels as a source of formation SO_2 there are compounds, containing organic sulfur, such as methyl mercaptan – CH_3S .

In addition, the macromolecules of liquid and solid fuels include thiophene C_4H_4S, which is also a source of formation SO_2.

If you know the elementary composition of fuel, then the chemical equation:

$S + O_2 = SO$

can be seen, that during the combustion of 1 kg fuel will produce 2S p on 100kg of sulfur dioxide (where S p – sulfur content of the fuel mass work, %).

The composition of liquid and solid fuel includes nitrogen fuel N p in the form of compounds type prole C_4H_5N and pyridine C_5H_5N. These connections can be an additional source of formation so-called fuel nitrogen oxides NO and NO_2.

During the combustion of liquid and solid fuels compared with natural gas, additionally formed not only oxides of sulfur and fuel nitrogen oxides, but the problem arises with the formation of ash. This is especially true for the incineration of high-ash solid fuel, the ash content of which Ap exceed 30 %. The combustion of solid fuels leads to two environmental problems:

– emissions of ash particles in aerosol form into the atmosphere;

– the accumulation of large amounts of solid waste in the form of ash-slag dumps near the source of combustion of fossil fuels.

The presence of ash- slag heaps causes thermal pollution of the lithosphere and atmosphere, distort the landscape and groundwater contamination. Introducing appropriate to the level quantity q4 unburned fuel prolongs oxidation and other reactions inside the dump volume to form a "bouquet" of toxic gases that pollute the atmosphere and affect the climate.

6.4 Climate formation factors

In a broader sense, the amount of energy received from the sun, and the quantity that is lost to space determine the equilibrium temperature and climate of Earth. This energy spread around the globe through winds, ocean currents, and other mechanisms that affect the climates of different regions.

Factors that form the climate is called climatology factors or "external mechanisms." They include such processes as and oceans reflect radiation, the formation of the mountains and the movement of continents and changes in the concentration of greenhouse gases. There is also a large variety of feedback responses to climate change, which can both increase and reduce the initial impact. Some parts of the climate system, such as oceans and ice caps, respond slowly to climate change, while others respond more quickly. There are also key factors of thresholds, above which the possible of occurrence of rapid change.

Mechanisms can also be internal or external. Internal factors are natural processes that occur within the climate system (e.g. thermohaline circulation). External factors can be natural (e.g., changes in solar radiation) and anthropogenic (e.g. greenhouse gases).

No matter the primitive climatology factors are internal or external, the response of the climate system might be fast (for example, sudden cooling using ambient air in the volcanic ash reflecting sunlight), slow (e.g. thermal expansion of ocean water due to warming) or a combination (e.g., sudden loss of the ability of the surface of the Arctic ocean to reflect light that is the loss of albedo due to melting sea ice due to gradual warming of water). Thus, the climate system can react, but a full response factors can be generated in the course of centuries, or even longer.

A special place is occupied by the activities of human society, which also affects climate, but much less than other factors. Influence of economic activities of mankind on the climate is particularly increased in the second half of the twentieth century. This influence is expressed in a contaminated atmosphere, and as a consequence, a General increase in air temperature and climate warming. This will lead to the melting of glaciers and rising sea levels. The main pollutants of the atmosphere are thermal power generation, industry and road transport. Emissions of thermal power plants account for 20-30% of the total air pollution. They emit the products of fuel combustion (oil, coal, natural gas) - SO2, SO3, NO2, CO2 and other. The emissions of the industrial enterprises (metallurgical, oil refining, chemical and others) constitute 15-25% of the total air pollution.

6.5 Recommendations to reduce impacts

Feature of the products of combustion is that they are a mixture of several simultaneously present toxic components. Therefore, along with MPC additionally impose the norms of maximum permissible emissions (MPE), expressed in units of mass of pollutant per unit of fuel consumption.

For stationary and transport power limitation of emissions of harmful substances from the combustion of fossil fuels are limited by such substances:

– for CO (ГОСТ 21204-97) 300 mg/m3;

– for NOx 150 mg/m3 with a volume of oxygen concentration 15% in dry sample of combustion products is reduced to standard atmospheric conditions (reduced temperature 273 K, pressure 0,1013MPa, ГОСТ 28775-90).

Methods to reduce emissions [30]:

– as fuel is used for a drive of supercharger a pumping natural gas is used.It reduces the emission of harmful combustion products;

– on the places of GTE the tank for collecting sludge fuel is provided (gas condensate), oils, technical fluids and pallets to exclude flow of fluids when replacing units and assemblies. This prevents contamination of the soil, with technical exploitation and repair of gas turbine engines, as well as during routine maintenance on the unit;

– technical staff strictly follows the rules according to maintenance of environmental actions to protect nature from industrial factors;

– improving the efficiency of processes of mixture formation, hence the right choice α (α- the total excess air ratio in the combustion chamber) and the distribution throughout the volume of the combustion chamber;

– decrease the delay time of ignition and combustion;

– improving the quality of feed gas using high-pressure and low-pressure nozzles;

–the optimal organization of the working process in the combustion chamber and raise the temperature in the zone of the chemical reaction;

– increasing the volume of the primary zone and the residence time therein, the use of preheating gas;

– reducing the flow of air for film cooling of the walls of the flame tubes;

– using the principle of micro torch of combustion gas in the combustion chamber;

– adding to the combustion zone of water or steam, which leads to additional oxidation of CO in carbon dioxide;

– the right choice the length of the fault according to the correct calculations, and followed by fine-tuning it;

– the use of pure gases in the combustion process and carrying out additional measures for this;

– use of two and three-stage combustion gases in the combustion chamber for more complete combustion;

– the use of chemical additives and mixtures to ensure the flow of more environmentally friendly reactions.

Methods to reduce emissions of oxide and nitrogen dioxide

One of the most acute problems of our time is the protection of atmospheric air from emissions of nitrogen oxide NO_x and therefore investigated a range of methods and ways to reduce their emissions further and will be considered.

The reduction in the formation of nitrogen oxides mainly is to reduce or eliminate "thermal" NO, which is achieved by changing the maximum combustion temperature and oxygen content in the zone of maximum temperatures. Often the combustion gas is possible to completely eliminate the "thermal" NO, bringing the concentration of nitrogen oxides to the level of "fast" - or to 100-120 mg/m^3 for relatively cold air and up to 150-200 mg/m^3 in hot air, and the design of the nozzles has a crucial role on the yield, since it is the process of mixing the gas with air. Therefore, almost all the techniques are effective if they affect the conditions of mixture formation and combustion in the base of the torch.

Conclusion

In this chapter the products influence of combustion of fossil fuels on the climate, methods to reduce the toxic and hazardous substances, which directly influence the formation and climate change were considered. As can be seen from the analysis of environmental characteristics, nature of formation of the main pollutants CO and NOx are very complex and despite the knowledge of many methods of reducing of their concentration in the emissions from the gas turbine still not perfectly studied and has many outstanding issues. Although now mainly gas turbines meet the modern standards, but to improve the ecological situation of this little (emissions only one GPU in an average year is approximately: NOx =120t/y; CO =169t/y), therefore it is necessary to introduce more stringent standards for emissions so it will encourage the production of low-emission engines.

GENERAL CONCLUSIONS

The task for the master thesis has been solved by estimating the methods and possibilities to improve the effectiveness of the gas turbine plant based on converted aircraft engine.

1. It should be noted that in addition to electricity, electricity generating plant produces heat. According to experts at transporting heat long distances the loss is up to 40...50 %. Therefore, the location of power plants close to the consumer is important; considerably smaller losses of energy will be when high power capacity plants are replaced by small, placed close to the consumer. This makes gas turbines of small and average power perspective.
2. This work proposes to use a exhausted flight resource engine as a mobile power station with complex cycle and fuel preheating in remote areas and areas recovering from natural disasters. The use of mobile plants is the best solution for the electricity supply for people living in areas prone to natural disasters. It is very important to make a mobile power station more portable and easy to use in emergency situations.

3. The calculations, carried out in the work, show that due to the use of the heat regenerator, the thermal efficiency value of a small-scale aviation engine with relatively low working process parameters can increase from 32 % up to 42 %.

4. As for preheating of fuel it can give us around one more percent of thermal efficiency and the better start up engine and also in a case of liquid fuel a better atomization can be achieved.

5. According to labor precaution part we can consider that the labor precaution is important enough which can have great impact on the operation of GTP directly and indirectly.

6. In the chapter the products influence of combustion of fossil fuels on the climate, methods to reduce the toxic and hazardous substances, which directly influence the formation and climate change were considered. As can be seen from the analysis of

environmental characteristics, nature of formation of the main pollutants CO and NOx are very complex and despite the knowledge of many methods of reducing of their concentration in the emissions from the gas turbine still not perfectly studied and has many outstanding issues. Although now mainly gas turbines meet the modern standards, but to improve the ecological situation of this little (emissions only one GPU in an average year is approximately: NOx =120 t/y; CO =169 t/y), therefore it is necessary to introduce more stringent standards for emissions so it will encourage the production of low-emission engines.

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Inventors: Harada, Masaki (Kariya, JP), Watanabe, Haruhiko (Chiryu, JP)

Appendix 1

Thermodynamic calculation and gas dynamic calculations

The aim of thermodynamic calculation is to determine the parameters of the working fluid flow in specific sections of the installation and power density, specific fuel consumption, the main gas turbine efficiency. For a given power and found specific air flow capacity is determined to install. The results of thermodynamic calculations are used in the following gas-dynamic calculation to determine the geometric parameters of gas turbines and in general.

The necessary data for thermodynamic calculation are: gas temperature at the outlet of the combustion chamber Tg * degree increase in air pressure in the compressor π_C^*, shaft horsepower turbine power N_e.

Are established as settlement conditions - pressure and temperature of air at the inlet of gas turbines. If they are not addressed specifically, the calculation is performed to a standard atmospheric conditions for a city in Iran called Isfahan:

$p_i = 101300$ Pa; $T_i = 288$ K.

Choice constructive scheme and main parameters for the installation process T_g^* and π_C^* based on an analysis of parameters performed attitudes and tendencies of their development.

Initial data:

Initial data for thermodynamic and gas dynamic calculations:

$\mathrm{N_e} = 671$ KW Power of the engine

$\pi_c^* = 6.3$ Pressure ratio

$T_g^* = 1363K$ Total gas temperature

$T_i = 288\ K$ Initial air temperature

$p_i = 101300$ Pa   Initial air pressure

Thermodynamic calculation

E-E

$$T_e^* = 288\,K$$

$$p_e^* = \sigma_e * p_i = 98261\,Pa$$

$\sigma_e = 0.95 \div 0.98$ - Coefficient of regeneration of full pressure in air intake

C-C

$p_C^* = \pi_C^* \cdot p_e^* = 6.3 \cdot 98261 = 619044.3 Pa$

R=287.2 J/kg*K; Gas constant for gas

k=1.4 ;Isentropic exponent for gas

Work of compressor $L_C = \frac{kRT_e^*}{k-1}\left(\pi_C^{*\frac{k-1}{k}} - 1\right) \cdot \frac{1}{\eta_C^*} =$

$$= \frac{1.4 * 287.2 * 288}{0.4} * (6.3^{0.285} - 1) * \frac{1}{0.87} = 2.295 \times 10^5\ J/Kg$$

Efficiency of compressor

$$\eta_C^* = \frac{\pi_C^{*\frac{k-1}{k}} - 1}{\pi_C^{*\frac{k-1}{k\eta_{ck}^*}} - 1} =$$

$$= \frac{6.3^{0.285} - 1}{6.3^{0.317} - 1} = 0.87$$

$\eta_{ck}^* = 0.9$

Efficiency of stage of compressor $\eta_{ck}^* = 0.89 \div 0.91$

$$T_C^* = T_e^* + \frac{L_C}{\frac{kR}{(k-1)}} = 288 + \frac{2.29505 \times 10^5}{\frac{1.4*287.2}{1.4-1}} = 516.3\ K$$

G-G

$$p_g^* = \sigma_{CS} \cdot p_C^* = 0.97 * 619044.3 = 600472.97 Pa$$

$$\sigma_{CS} = 0.97 \div 0.99$$

$$T_g^* = 1363K$$

$$C_{CS} = 848 + 0.208\left(T_g^* + 0.48 \cdot T_C\right) =$$

$$= 848 + 0.208(1363 + 0.48 * 516.3\ ) = 1610.82\ J/Kg.K$$

$$g_{\text{пал}} = \frac{C_{kc}\left(T_г^* - T_k^*\right)}{H_u \eta_г} = \frac{1610.824(1363-516.3)}{42.5*10^6*0.97} = 0.033$$ Relative fuel consumption

$H_u = 42.5 * 10^6 J/kg$ - lowest heat rejection ability of liquid fuel

$\eta_g = 0.97$ - Coefficient, taking into account nonfull fuel burning and heat losses through walls of combustion chamber

$$q_1 = C_{CS}\left(T_g^* - T_C^*\right) = 1610.824\ (1363 - 516.3) = 1.363\ \times\ 10^6\ J/Kg$$

TC-TC

$$L_T = \frac{L_C}{\eta_M \cdot (1 + g_{\text{пал}}) \cdot (1 - g_{cool} - g_{отб})} =$$

$$= \frac{2.29505\ \times\ 10^5}{0.99 * (1 + 0.033\ \ ) * (1 - 0.04 - 0.01)} = 236228.9\ J/Kg$$

$g_{отб} = 0,01$

$\eta_{м} = 0.99$

$g_{cool} = 0.04$

$$T_{gS}^{*} = T_{g}^{*} - \frac{L_{T}(k_{g}-1)}{k_{g}R_{g}} =$$

$$= 1363 - \frac{236228.9 * (1.33 - 1)}{1.33 * 288} = 1159.48\,K$$

$k_{g} = 1.33$ $\qquad R_{g} = 288\,J/kg \cdot K$

$$p_{gS}^{*} = p_{g}^{*}\left(1 - \frac{T_{g}^{*} - T_{gS}^{*}}{T_{g}^{*} \cdot \eta_{T}^{*}}\right)^{\frac{k_{g}}{k_{g}-1}} =$$

$$= 600472.971 * \left(1 - \frac{1363 - 1159.48}{1363 * 0.9}\right)^{\frac{1.33}{1.33-1}} = 2.89 * 10^{5}\,Pa$$

Efficiency of turbine $\eta_{T}^{*} = 0.9$

PT-PT

$$p_{T}^{*} = p_{i} \cdot (1.03 \div 1.05) = 104339Pa$$

$$\pi_{PT}^{*} = \frac{p_{gS}^{*} \cdot \sigma_{II}}{p_{T}^{*}} =$$

$$= 2.89042 * 10^{5} * \frac{0.98}{104339} = 2.71$$

$$L_{PT} = \frac{k_{g}}{kg-1} R_{g} T_{gS}\left(1 - \frac{1}{\pi_{PT}^{*\frac{k_{g}-1}{k_{g}}}}\right)\eta_{PT}^{*} =$$

$$= \frac{1.33}{1.33-1} * 288 * 1159.48 * \left(1 - \frac{1}{2.71^{\frac{0.33}{1.33}}}\right) 0.89 = 2.624 \times 10^5 \, J/Kg$$

$$\eta_{PT}^{*} = 0.89 \div 0.91$$

$$T_T^{*} = T_{gS}^{*} - \frac{L_{PT}}{\frac{k_g}{k_g - 1} \cdot R_g} =$$

$$= 1159.48 - \frac{2.62489 \times 10^5}{\frac{1.33}{0.33} * 288} = 933.33 \, K$$

$$C_T = \lambda_T \sqrt{\frac{2k_г}{k_г + 1} R_г T_T^{*}} = 0,5 \sqrt{\frac{2 * 1.33}{1.33 + 1} 288 * 933.33} = 276.97$$

$$\lambda_T = 0.5$$

$$T_T = T_T^{*} \left(1 - \frac{k_г - 1}{k_г + 1} * \lambda_T^2\right) = 933.33 * \left(1 - \frac{1.33 - 1}{1.33 + 1} * 0.5^2\right) = 900.28 \, K$$

$$P_T = P_T^{*} \left(1 - \frac{k_г - 1}{k_г + 1} * \lambda_T^2\right)^{\frac{k_г}{k_г - 1}} = 104339 * \left(1 - \frac{1.33 - 1}{1.33 + 1} * 0.5^2\right)^{\frac{1.33}{1.33 - 1}}$$
$$= 90229.25 \, Pa$$

Calculation of main parameters of gas turbine unit

$$N_{е.п} = \eta_{МРТ} L_{РТ} (1 + g_{пал}) =$$

$$= 0.99 * 2.62489 \times 10^5 * (1 + 0.033) = 268439.62 \, Wt$$

$\eta_{МРТ}$ = 0.99–0.995

$$\eta_e = \frac{L_{PT}}{q_1} = \frac{2.62489 \times 10^5}{1.3638846808 \times 10^6} = 0.1924$$

$C_e = 3600 g_{пал}/N_{е.п}$$=3600 * \frac{0.033}{268.43962563} = 0.442\ kg/Kwt/h$

$G_C = N_e/N_{е.п}$$=\frac{671}{268.43962563} = 2.5\ kg/s$

Components

Compressor: 3-stage axial + 1-stage centrifugal flow compressor (Reverse flow, radial inlet with screen for FOD (Foreign Object Damage) protection)

3-stage axial/1-stage centrifugal and if work is equal:

$L_C$$=2.11963 \times 10^5\ J/Kg$

Table 1

Work of compressor stages

№ stage	1 $\pi = 1.24$	2 $\pi = 1.31$	3 $\pi = 1.27$	4 centrifugal Stage $\pi = 3$	$\pi_\Sigma = 6{,}3$
Work of stage	23445.8 J/Kg	30240.9 J/Kg	30490.6 J/Kg	145327.7 J/Kg	$L_k$$=2.29505 \times 10^5\ J/Kg$

R=287.2 J/kg*K;

k=1.4

η^*_{PT} - it is stage efficiency, taken into limits $\eta^*_{PT} = (0.88 \div 0.91)$

1st stage:

$$L_C = \frac{kRT_e^*}{k-1}\left(\pi_C^{*\frac{k-1}{k}} - 1\right)\cdot\frac{1}{\eta_C^*} =$$

$$= \frac{1.4 * 287.2 * 288}{0.4} * (1.24^{0.285} - 1) * \frac{1}{0.9} = 20337.2\ J/Kg$$

Temperature of stagnated air flow after the 1st stage determination:

$$T^*_{1st} = T^*_i + \frac{L_{PT}}{\frac{k}{k-1}R} = 288 + \frac{20337.2}{\frac{1.4}{0.4} * 287.2} = 308.23\ k$$

Pressure of stagnated air flow after the 1st stage determination:

$$P^*_{1st} = \pi_{1stS} * P^*_i = 1.24 * 98261 = 121843.64\ Pa$$

2nd stage:

$$L_C = \frac{kRT^*_i}{k-1}\left(\pi_C^{*\frac{k-1}{k}} - 1\right) \cdot \frac{1}{\eta^*_C} =$$

$$= \frac{1.4 * 287.2 * 308.23}{0.4} * (1.31^{0.285} - 1) * \frac{1}{0.9} = 27539.5\ J/Kg$$

Temperature of stagnated air flow after the 1st stage determination:

$$T^*_{2nd} = T^*_i + \frac{L_{PT}}{\frac{k}{k-1}R} = 308.23 + \frac{27539.5}{\frac{1.4}{0.4} * 287.2} = 335.62\ k$$

Pressure of stagnated air flow after the 1st stage determination:

$$P^*_{2nd} = \pi_{2ndS} * P^*_{1st} = 1.31 * 121843.64 = 159615.16\ Pa$$

3rd stage:

$$L_C = \frac{kRT^*_e}{k-1}\left(\pi_C^{*\frac{k-1}{k}} - 1\right) \cdot \frac{1}{\eta^*_C} =$$

$$= \frac{1.4 * 287.2 * 335.62}{0.4} * (1.31^{0.285} - 1) * \frac{1}{0.9} = 29986.7/Kg$$

Temperature of stagnated air flow after the 1st stage determination:

$$T^*_{3rd} = T^*_{2nd} + \frac{L_{PT}}{\frac{k}{k-1}R} = 335.62 + \frac{29986.7}{\frac{1.4}{0.4} * 287.2} = 365.45\ k$$

Pressure of stagnated air flow after the 1st stage determination:

$$P_{3rd}^{*} = \pi_{3rdS} * P_{2nd}^{*} = 1.27 * 159615.1684 = 202711.26\ Pa$$

Calculation of gas-dynamic gas turbine plant

The aim is to determine gas-dynamic calculation diametric sizes in specific sections of flow of the installation, the number and frequency of rotor rotation, the number of stages of the compressor and turbine distribution of compression (expansion) between the stages and steps, refine your GTU.

As a result of input data used thermodynamic cycle calculation of the actual installation.

During the gas-dynamic calculation based on statistics made designs GTP selected axial velocity component in the air inlet compressor - CBA and angular velocity at the outer diameter of the impeller first stage compressor. These parameters largely determine the diametrical dimensions of the gas turbine, the number of degrees of compressor and turbine and axial dimensions and weight of the installation.

Determination diametrical size at the entrance to compressor

$C_{AB} = 120 m/s$ - entrance velocity

$$\lambda_{AB} = \frac{C_{AB}}{18.3 \cdot \sqrt{T_{B}^{*}}} = \frac{120}{18.3\sqrt{288}} = 0.386$$ - driven velocity

$$q(\lambda_{\text{ав}}) = 1{,}5774\lambda_{\text{ав}}(1 - 0{,}166\lambda_{\text{ав}}^{2})^{2,5} = 1.5774 * 0.386(1 - 0.166 * 0.386^{2})^{2.5} = 0.57$$

$$F_{Ce} = \frac{G_{C} * \sqrt{T_{H}}}{P_{B}^{*} * q(\lambda_{aB}) * m_{a}} = \frac{2.5 * \sqrt{288}}{98261 * 0.57 * 0.0403} = 0.0188\ m^{2}$$

$m_{a} = 0.0403$ - indicator of air

$\overline{d}_в = D_{в.вт}/D_{в.к}$ $\qquad \overline{d}_B = 0{,}5...0{,}6$

$$D_{в.к} = \sqrt{4F_в / \left[3{,}14\left(1 - \overline{d}_в^2\right)\right]} = \sqrt{\frac{4*0.0188}{3.14*(1-0.5^2)}} = 0.178$$

$$D_{B.BT} = \sqrt{D_{в.к}^2 - 4F_B/3.14} = \sqrt{0.178^2 - 4*0.0188/3.14} = 0.087\ m$$

$$h_B = 0.5(D_{B.K.} - D_{B.BT}) = 0.5*(0.178 - 0.087\ ) = 0.0455\ m$$

$$D_{B.CEP} = D_{B.BT} + h_B = 0.087 + 0.0455 = 0.1325\ m$$

Determination of diametrical sizes at the exit from axial compressor

$$C_{ak} = 110 \div 140 m/s$$

$\lambda_{ak} = \dfrac{C_{ak}}{18.3\sqrt{T_{3rd}^*}} = \dfrac{100}{18.3\sqrt{365.45}} = 0.285$- reduced velocity at the entrance

$q(\lambda_{ak}) = 1.5774\,\lambda_{ak}(1 - 0.166\,\lambda_{ak}^{\ 2})^{2.5} = 1.5774*0.285(1-0.166*0.285^2)^{2.5} = 0.434$- function of flow compactness

$$F_k = \frac{G_C \cdot \sqrt{T_{3rd}^*}}{m \cdot p_{3rd}^* \cdot q(\lambda_{ak})} =$$

$$= \frac{2.5*\sqrt{365.45}}{0.0403*220783.82\ *0.434} = 0.0123\ m^2$$

$$D_{KK} = D_{B.K.} = const = 0.178$$

- bushing diameter at the exit from compressor:

$$D_{К.ВТ} = \sqrt{D_{КК}^2 - 4F_К/3.14} = \sqrt{0.178^2 - 4*\frac{0.0123}{3.14}} = 0.1265\ m$$

- length of the blade at the last stage of high-pressure compressor stage.

$$h_С = 0.5\left(D_{КК} - D_{К.ВТ}\right) = 0.5*(0.178 - 0.1265) = 0.02575\ m$$

- mean diameter at the exit from HPC

$D_{B.CEP} = D_{B.BT} + h_K = 0.1265 + 0.02575 = 0.1522\, m$

Take into account last stage of compressor is centrifugal

Determination diametrical size at the entrance to Centrifugal compressor

$$\mathrm{k} = 1.4$$

$$\xi = 1$$

$$R = 287.2$$

$$\mu_s = 0.7$$

$$\pi_{CF} = 3$$

T^*_{3rd} =365.45 K

$$P^*_{3rd} = 202711.263868\, Pa$$

$$u_2 = \sqrt{\frac{L^*_{sk}}{\mu_s}} = \sqrt{\frac{\frac{kR}{k-1} * T^*_{3rdS} * (\pi_C^{*\frac{k-1}{k}} - 1)}{\mu_s}} = \sqrt{\frac{\frac{1.4 * 287.2}{1.4-1} * 365.45 * (3^{\frac{1.4-1}{1.4}} - 1)}{0.7}}$$
$$= 439.89\, m/s$$

$$P^*_{cf} = \pi_{cf} * P^*_{3rd}$$

$$P^*_{cf} = 3 * 202711.263868 = 608133.79$$

$$c_{1cr} = \sqrt{\frac{2kRT^*_{3rd}}{(k+1)}} = \sqrt{\frac{2 * 1.4 * 287.2 * 365.45}{(1.4+1)}} = 349.93\, m/s$$

$$assume\ c_{1u} = 330\, m/s$$

$$c_1 = \sqrt{c_{1a}^2 + c_{1u}^2}$$

$$c_1 = \sqrt{100^2 + 330^2} = 344.8\ m/s$$

$$c_{1cr} > c_1$$

$$\lambda_1 = \frac{c_1}{c_{1cr}} = \frac{344.8}{349.93} = 0.985$$

$$\lambda_{1a} = \frac{c_{1a}}{c_{1cr}} = \frac{100}{349.93} = 0.2857$$

Pressure loss

$$\sigma_{ip} = \frac{1}{1 + \xi_{\mathrm{ip}}\lambda_1^2 \left[1 - \frac{k-1}{k+1}\lambda_1^2\right]^{\frac{1}{k-1}}}$$

$$\sigma_{ip} = \frac{1}{1 + 0.05 * 0.985^2 \left[1 - \frac{1.4-1}{1.4+1}0.985^2\right]^{\frac{1}{1.4-1}}} = 0.96$$

$$\sigma_{vna} = \frac{1}{1 + 0.05 * 0.985^2 \left[1 - \frac{1.4-1}{1.4+1}0.985^2\right]^{\frac{1}{1.4-1}}} = 0.615$$

Change of mass flow rate

if $\sigma_{ip} = \sigma_{ex}$

$$\mu_{\text{вх}} = \sigma_{ex}^{\frac{k-1}{k}} \sqrt{\frac{1 - \left[1 - \frac{k-1}{k+1}\lambda_1^2\right]}{1 - \sigma_{ex}^{\frac{k-1}{k}} \left[1 - \frac{k-1}{k+1}\lambda_1^2\right]}}$$

$$\mu_{\text{ex}} = 0.96^{0.2857} \sqrt{\frac{1 - \left[1 - \frac{1.4-1}{1.4+1}0.985^2\right]}{1 - 0.96^{0.2857} \left[1 - \frac{1.4-1}{1.4+1}0.985^2\right]}} = 0.9599$$

$$m = \sqrt{\frac{k}{R}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} = \sqrt{\frac{1.4}{287.2}\left(\frac{2}{1.4+1}\right)^{\frac{1.4+1}{1.4-1}}} = 0.0404$$

$$q(\lambda_{1a}) = \lambda_{1a}\left(\frac{k+1}{2}\right)^{\frac{1}{k-1}}\left(1-\lambda_{1a}^2\frac{k-1}{k+1}\right)^{\frac{1}{k-1}}$$

$$q(\lambda_{1a}) = 0.2857\left(\frac{1.4+1}{2}\right)^{\frac{1}{1.4-1}}\left(1-0.2857^2\frac{1.4-1}{1.4+1}\right)^{\frac{1}{1.4-1}} = 0.435$$

$$F_1 = \frac{G_C * \sqrt{T_{3rdS}^*}}{\mu_{ex}\sigma_{ex} m P_{3rd}^* \, q(\lambda_{1a})}$$

$$F_1 = \frac{2.5 * \sqrt{365.45}}{0.9599 * 0.96 * 0.0404043 * 202711.263868 * 0.435} = 0.014\, m^2$$

$$\bar{d}_0 = 0.4 \ldots 0.55$$

$$D_1 = \sqrt{\frac{4F_1}{3.14(1-\bar{d}_0^2)}} = \sqrt{\frac{4 * 0.014}{3.14(1-0.4^2)}} = 0.145\, m$$

$$D_{1av} = \sqrt{\frac{2F_1(1+\bar{d}_0^2)}{3.14(1-\bar{d}_0^2)}} = \sqrt{\frac{2 * 0.014(1+0.4^2)}{3.14(1-0.4^2)}} = 0.11\, m$$

$$D_{sl} = \bar{d}_0 * D_1 = 0.4 * 0.145 = 0.058\, m$$

Determination of diametric sizes at compressor turbine entrance:

$$F_1 = \frac{G_T \cdot \sqrt{T_g^*}}{p_g^* \cdot \sigma_{к.з} \cdot \sigma_{ca} \cdot q(\lambda_Г) \cdot m_Г \cdot \sin\alpha} = \frac{2.46 * \sqrt{1363}}{600472.971 * 0.98 * 0.98 * 1 * 0.0396 * \sin 20} = 0.0116\, m^2$$

$$q(\lambda_Г) = 1$$

$$\sigma_{к.з} = 0.98;\ \sigma_{ca} = 0.98;\ m_Г = 0.0396;\ \alpha = 20^\circ$$

$$G_T = G_C(1+g_{пал.})(1-g_{cool}-g_B) = 2.5 * (1+0.033)(1-0.04-0.01) = 2.46\, kg/s$$

$$D_{T.CEP.1} = (1.05 \div 1.1)D_{B.K} = 1.1 * 0.178 = 0.196\, m$$

- the height of turbine vane height by means of formula:

$$h_{blade} = \frac{F_1}{3.14 \cdot D_{T.CEP.1}} = \frac{0.0116}{3.14 * 0.196} = 0.0188\, m$$

- working wheel diameter:

$$D_{T.K.1} = D_{T.CEP.1} + h_{blade} = 0.196 + 0.0188 = 0.2148\, m$$

- working wheel bushing diameter:

$$D_{T.BT.1} = D_{T.CEP.1} - h_{blade} = 0.196 - 0.0188 = 0.1772\, m$$

$$D_{sl} = \sqrt{D_{tk1}^2 - \frac{4}{\pi}F_1} = \sqrt{0.2148^2 - \frac{4}{3.14} * 0.0116} = 0.177\, m$$

$$h_b = \frac{0.2148 - 0.177}{2} = 0.0189$$

Determination of velocity at the middle diameter of turbine

$$u_{mk.cep.} = u_{в.к.} \cdot \frac{D_{T.CEP.}}{D_{B.K.}} = 335 * \frac{0.196}{0.178} = 368.8\, m/s$$

$$C_1 = \frac{L_{tk}}{u_{tmn} cos\alpha_1} = \frac{236228.9}{368.8 * cos15} = 663.12\, m/s$$

$$\lambda_1 = \frac{C_1}{C_{cr}} = \frac{663.12}{18.5 * \sqrt{1363}} = 0.97$$

$$q(\lambda_1) = 1$$

$$c_{1a} = c_1 sin\alpha_1 = 663.12 * sin15 = 171.62$$

Determination of diametric sizes at compressor turbine exit and Entrance of power turbine

Gas parameters:

$$T_{гс}^* = T_г^* - \frac{L_{TK}(k_г - 1)}{k_г R_г} =$$

$$= 1363 - \frac{236228.9 * (1.33 - 1)}{1.33 * 288} = 1159.48\,K$$

$$p_{гс}^* = p_г^* \left(1 - \frac{T_г^* - T_{гс}^*}{T_г^* \cdot \eta_{Tk}^*}\right)^{\frac{k_г}{k_г - 1}} =$$

$$= 600472.971 * \left(1 - \frac{1363 - 1159.48}{1363 * 0.9}\right)^{\frac{1.33}{1.33-1}} = 2.89 * 10^5\,Pa$$

$$F_{pt} = \frac{G_{PT}\sqrt{T_{гс}^*}}{m_g P_{гс}^* \sigma_\pi q(\lambda_{гс}) sin\alpha} = \frac{2.556 * \sqrt{1159.48}}{0.0309 * 2.89042 * 10^5 * 0.98 * 1 * \sin 20}$$

$$= 0.029\,m^2$$

$$\alpha = 20 \dots 25$$

$$G_{PT} = G_k(1 + g_{пал.})(1 - g_В) = 2.5 * (1 + 0.033)(1 - 0.01) = 2.556\,kg/s$$

$$m_g = 0.309$$

$$\sigma_\pi = 0.98 \dots 0.99$$

$$q(\lambda_{гс}) = 1$$

$$D_{T.CEP.2} = D_{T.CEP.1} = \quad 0.196\,m$$

- the height of turbine vane height by means of formula:

$$h_{blade2} = \frac{F_2}{3.14 \cdot D_{T.CEP.2}} = \frac{0.029}{3.14 * 0.196} = 0.0471\ m$$

$$D_{T.K.2} = D_{T.CEP.2} + h_{blade2} = 0.196 + 0.0471 = 0.2431\,m$$

$$D_{T.BT.2} = D_{T.CEP.2} - h_{blade2} = 0.196 - 0.0471 = 0.1489m$$

3-3

Exit of power turbine

$$F_3 = \frac{G_{PT} \cdot \sqrt{T_T^*}}{p_T^* \cdot q(\lambda_T) \cdot m_\Gamma} = 2.556 * \frac{\sqrt{933.33}}{104339*0.781*0.0309} = 0.031\ m^2$$

$$\lambda_T = 0.6$$

$$q(\lambda_T) = 1.527 \cdot \lambda_T (1 - 0.143\lambda_T^{\ 2})^{3.02} = 1.527 \cdot 0.6(1 - 0.143 \cdot 0.6^2)^{3.02} = 0.781$$

$$D_{T.BT.3} = D_{T.BT.2} = 0.1489\ m$$

$$D_{T.K.3} = \sqrt{D_{T.BT.3}^2 + \frac{4F_3}{3.14}} = \sqrt{0.1489^2 + 4 * \frac{0.031}{3.14}} = 0.2483\ m$$

determine the height of turbine vane height by means of formula:

$$h_{\text{blade}} = 0.5(D_{T.KK} - D_{T.BT.}) = 0.5 * (0.2483 - 0.1489) = 0.0497\ m$$

$$N_C = G_C L_C = 2.5 * 2295005 = 1102036.5 Wt$$

$$N_T = G_T L_T = 2.46 * 2362228 = 1150120.88\ Wt$$

$$N_{PT} = G_{PT} L_{PT} = 2.556 * 262489 = 670921.88\ Wt$$

$$n_C = \frac{60u_{\text{вк}}}{3,14D_{\text{вк}}} = \frac{60*335}{3.14*0.178} = 35962.21\ rot/min$$

$$n_{PT} = \frac{60u_{\text{с.т.сер}}}{3,14D_{\text{тс.сер}}} = \frac{60*280}{3.14*0.196} = 27297.54\ rot/min$$

Appendix 2

The calculation regarding the regenerator

The calculation and graphs was carried out by MathCAD program.

1.the Brayton cycle with and without regenerator:

$p_1 := 10130($

$r_p := 6.3$

$k := 1.4$

$k_g := 1.33$

$\sigma_r := 0.85$

$T_3 := 1363$

$T_1 := 28\varepsilon$

$\eta_c := 0.8\epsilon$

$\eta_t := 0.\varsigma$

$p_2 := p_1 \cdot r_p$

$c_p := 1004.\!$

$p_2 = 6.382\times 10^5$

$c_{pg} := 112($

$$T_2 := T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1 \right) \right]$$

$$t_4 := T_3 \cdot \left[1 - \left(1 - r_p^{\frac{k_g - 1}{k_g}}\right) \cdot \eta_t\right]$$

$T_2 = 519.717$

$t_4 = 2.073 \times 10^3$

$$T_4 := \frac{T_3 \cdot T_1}{T_2}$$

$T_4 = 755.304$

$$T_5 := T_2 + \sigma_r \cdot (T_4 - T_2)$$

$T_5 = 719.966$

$$w_c := c_p \cdot T_1 \cdot \left(\frac{r_p^{\frac{k-1}{k}}}{\eta_c}\right)$$

$w_c = 5.692 \times 10^5$

$$w_t := c_{pg} \cdot T_3 \cdot \eta_t \cdot \left(1 - \frac{1}{r_p^{\frac{k_g - 1}{k_g}}}\right)$$

$w_t = 5.037 \times 10^5$

$$w_{net} := w_c - w_t$$

$w_{net} = 6.546 \times 10^4$

$$q_1 := c_p \cdot \left[(T_3 - T_2) - \sigma_r \cdot (T_4 - T_2)\right]$$

$q_1 = 6.459 \times 10^5$

$$q_2 := c_{pg} \cdot (T_5 - T_1)$$

$q_2 = 4.838 \times 10^5$

$$\eta_{thermalreg} := \frac{c_{pg}\cdot T_3\cdot \eta_t\cdot\left(1-\frac{1}{r_p^{\frac{k_g-1}{k_g}}}\right)-c_p\cdot T_1\cdot\left(\frac{r_p^{\frac{k-1}{k}}-1}{\eta_c}\right)}{c_p\cdot\left[\left[T_3-T_1\cdot\left[1+\frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}}-1\right)\right]\right]-\sigma_r\cdot\left[T_4-T_1\cdot\left[1+\frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}}-1\right)\right]\right]\right]}$$

$\eta_{thermalreg} = 0.419$

$$\eta_{thermalwiot} := \frac{c_{pg}\cdot T_3\cdot \eta_t\cdot\left(1-\frac{1}{r_p^{\frac{k_g-1}{k_g}}}\right)-c_p\cdot T_1\cdot\left(\frac{r_p^{\frac{k-1}{k}}-1}{\eta_c}\right)}{c_p\cdot\left[\left[T_3-T_1\cdot\left[1+\frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}}-1\right)\right]\right]\right]}$$

$\eta_{thermalwiot} = 0.32$

2.Dependence of pressure ratio and temperature

$p_1 := 101300$

$k := 1.4$

$k_g := 1.33$

$\sigma_r := 0.85$

$T_3 := 1100$

$\eta_c := 0.86$

$\eta_t := 0.9$

$T_1 := 283$

$r_p :=$ READPRN("graph4.txt")

$c_p := 1004.5$

$i := 5..110$

$r_p =$

	0
0	0
1	0.2

$c_{pg} := 1120$

$$T_{2_i} := T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]$$

$T_{2_i} =$

283
300.596
316.206
330.294
343.175
355.071
366.145
376.521
386.296
395.549
404.341
412.723
420.739
428.425
435.812
442.926

$$T_{4_i} := \frac{T_3 \cdot T_1}{T_{2_i}}$$

$T_{4_i} =$

$1.1 \cdot 10^3$
$1.036 \cdot 10^3$
984.486
942.493
907.117

$T_{12} := 293$

$T_3 := 1200$

$$T_{22_i} := T_{12} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]$$

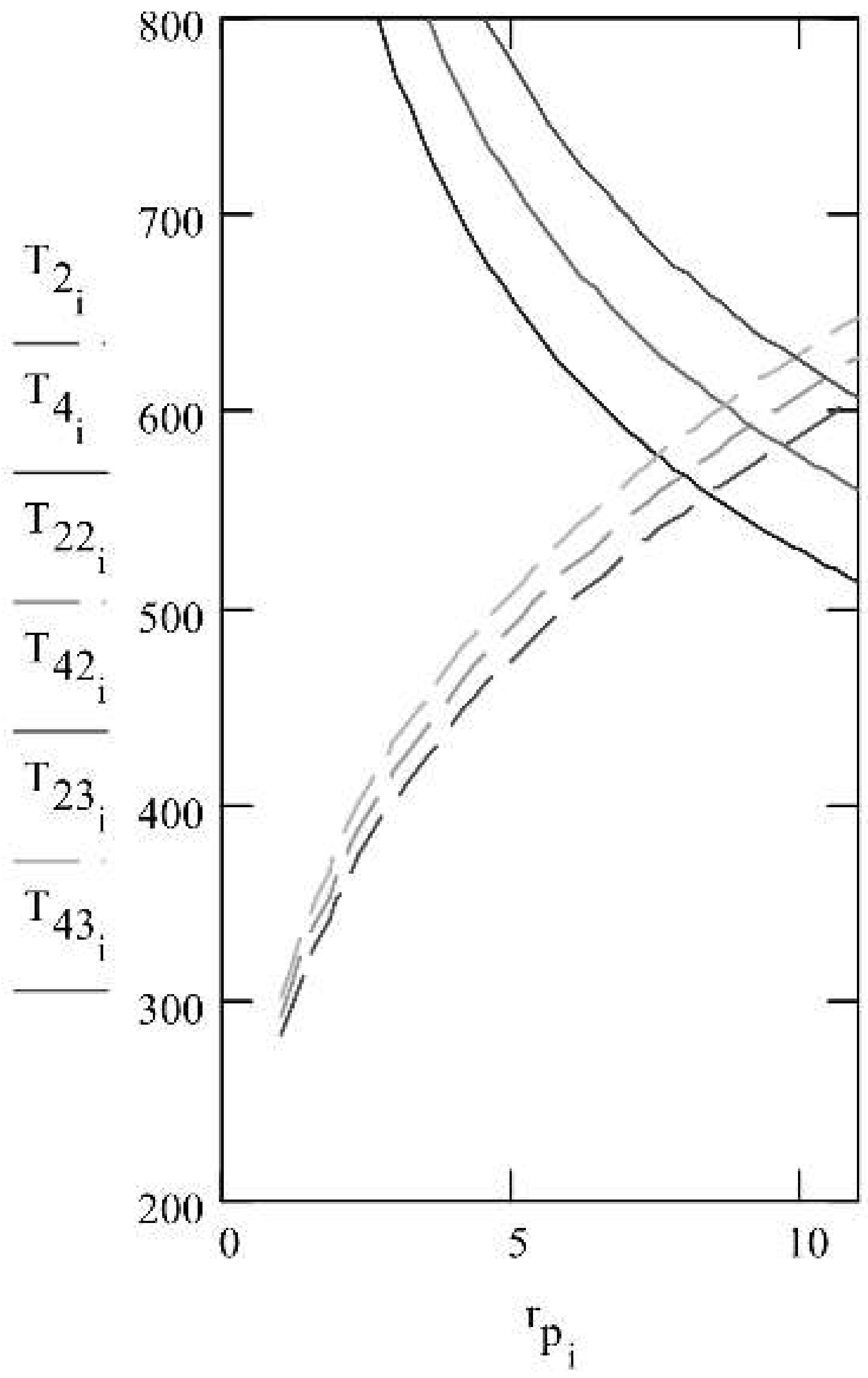
$$T_{42_i} := \frac{T_3 \cdot T_{12}}{T_{22_i}}$$

$T_{13} := 303$

$T_3 := 1300$

$$T_{23_i} := T_{13} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]$$

$$T_{43_i} := \frac{T_3 \cdot T_{13}}{T_{23_i}}$$



3. Effect of ambient temperature on thermal efficiency for simple and regenerative cycle

$p_1 := 101300$

$r_p := 6.3$

$k := 1.4$

$k_g := 1.33$

$c_p := 1004.5$

$c_{pg} := 1120$

$\eta_t := 0.9$

$\eta_c := 0.86$

$T_3 := 1363$

$T_1 :=$ READPRN("graph.txt")

$\sigma_r := 0.8$

$T_3 = 1.363\times 10^3$

$i := 0 .. 49$

$T_1 =$

	0
0	283
1	284
2	285
3	286
4	287
5	288
6	289
7	290
8	291
9	292
10	293
11	294
12	295
13	296
14	297

$$T_{2_i} := T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1 \right) \right]$$

$$T_{4_i} := \frac{T_3 \cdot T_{1_i}}{T_{2_i}}$$

$T_2 =$

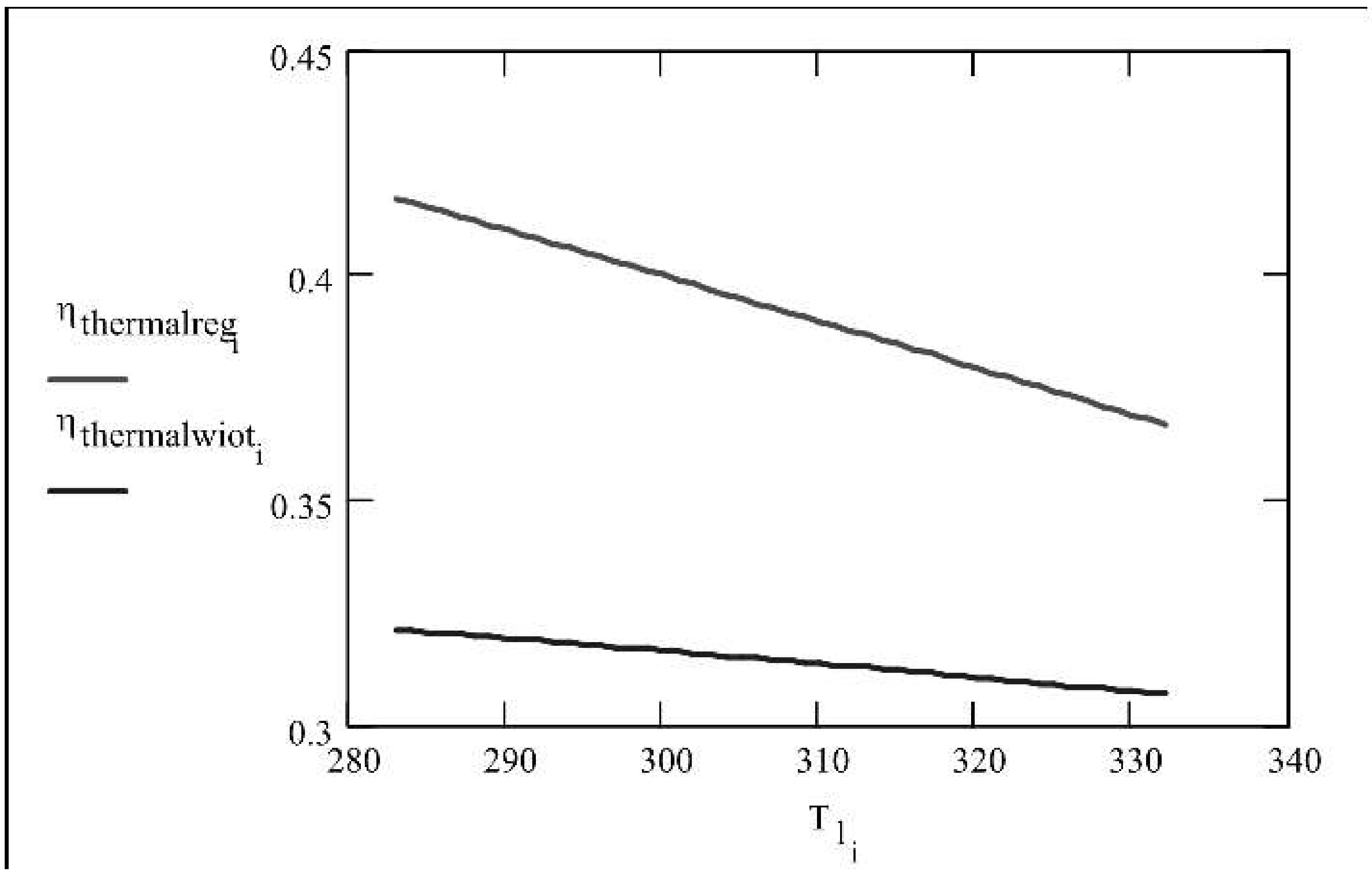
	0
0	510.694
1	512.499

$T_4 =$

	0
0	755.304
1	755.304
2	755.304
3	755.304
4	755.304
5	755.304

$$\eta_{thermalreg_i} := \frac{c_{pg}\cdot T_3\cdot \eta_t\cdot\left(1-\frac{1}{r_p^{\frac{k_g-1}{k_g}}}\right)-c_p\cdot T_{1_i}\cdot\left(\frac{r_p^{\frac{k-1}{k}}-1}{\eta_c}\right)}{c_p\cdot\left[\left[T_3-T_{1_i}\cdot\left[1+\frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}}-1\right)\right]\right]-\sigma_r\cdot\left[T_{4_i}-T_{1_i}\cdot\left[1+\frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}}-1\right)\right]\right]\right]}$$

$$\eta_{thermalwiot_i} := \frac{c_{pg}\cdot T_3\cdot \eta_t\cdot\left(1-\frac{1}{r_p^{\frac{k_g-1}{k_g}}}\right)-c_p\cdot T_{1_i}\cdot\left(\frac{r_p^{\frac{k-1}{k}}-1}{\eta_c}\right)}{c_p\cdot\left[\left[T_3-T_{1_i}\cdot\left[1+\frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}}-1\right)\right]\right]\right]}$$



4.Dependence of pressure ratio and thermal efficiency

$p_1 := 101300$

$k := 1.4$

$k_g := 1.33$

$\sigma_r := 0.85$

$T_3 := 1363$

$\eta_c := 0.86$

$\eta_t := 0.9$

$T_1 := 288$

$r_p :=$ READPRN("graph4.txt")

$c_p := 1004.5$

$i := 5 .. 110$

$r_p =$

	0
0	0
1	0.2

$c_{pg} :=$ 112(

$$T_{2_i} := T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right]$$

$$T_{4_i} := \frac{T_3 \cdot T_1}{T_{2_i}}$$

$T_{4_i} =$

$1.363 \cdot 10^3$
$1.283 \cdot 10^3$
$1.22 \cdot 10^3$
$1.168 \cdot 10^3$
$1.124 \cdot 10^3$

$$\eta_{thermalreg_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i} \right)^{\frac{k_g - 1}{k_g}}} \right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1}{\eta_c} \right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right] \right] - \sigma_r \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right] \right] \right]}$$

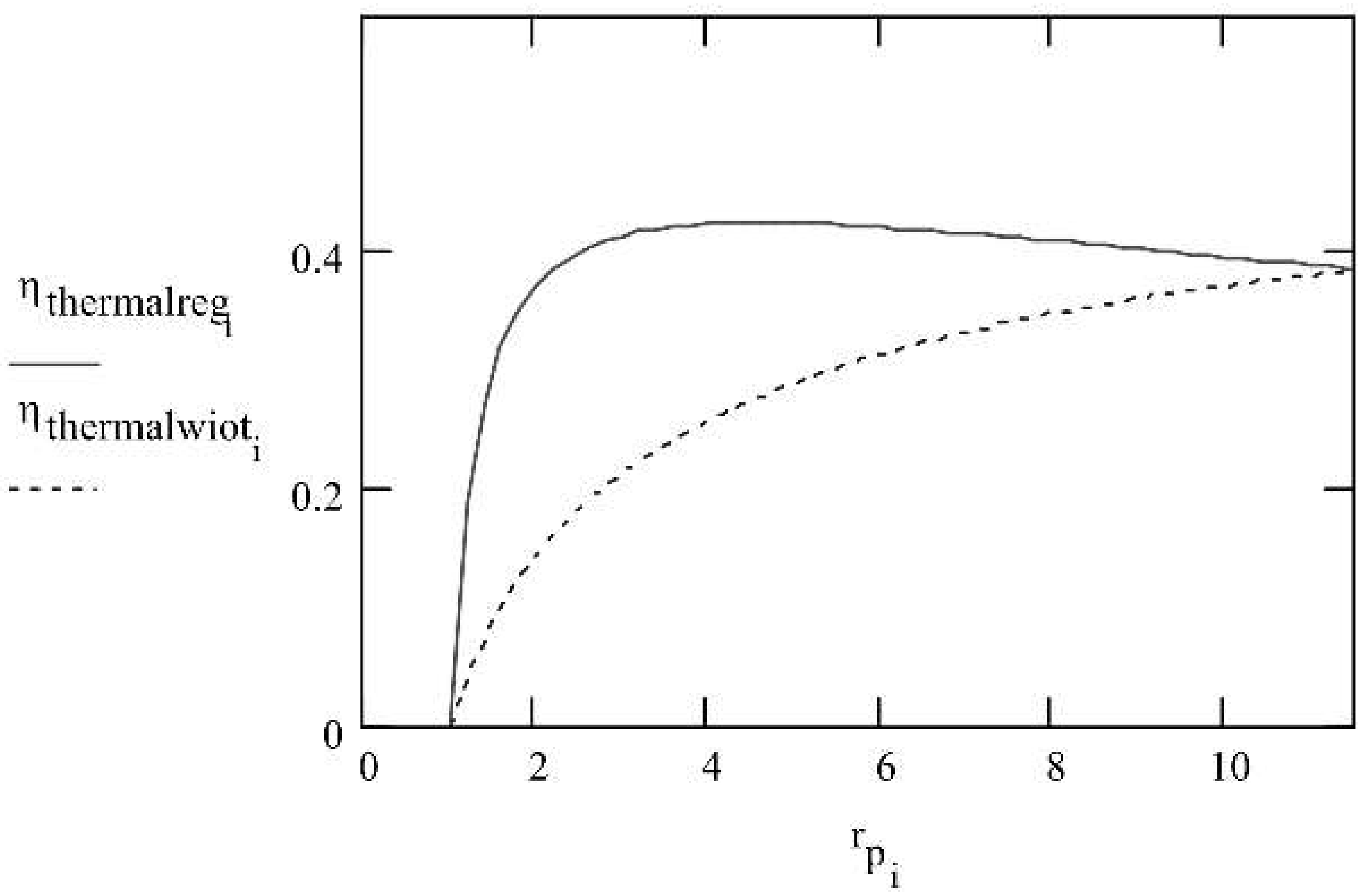
$\eta_{thermalreg_i} =$

0	
0.188	
0.273	
0.32	
0.35	

$$\eta_{thermalwiot_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i} \right)^{\frac{k_g - 1}{k_g}}} \right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1}{\eta_c} \right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right] \right] \right]}$$

$\eta_{thermalwiot_i}$

0
0.04
0.073
0.1
0.123
0.143
0.16
0.175
0.189
0.202
0.213
0.224
0.233
0.242
0.25
0.258



5. supplied heat versus compression ratios for regenerative gas turbine cycle at different values of effectiveness of regeneration

$\sigma_r := 0.45$

$p_1 := 101300$

$r_p := 6.3$

$k := 1.4$

$k_g := 1.33$

$r_p :=$ READPRN("graph4.txt")

$T_3 := 1363$

$T_1 := 293$

$\eta_c := 0.86$

$\eta_t := 0.9$

$p_2 := p_1 \cdot r_p$

$r_p =$

	0
0	0
1	0.2

$c_p := 1004.5$

$i := 5..110$

$c_{pg} := 1120$

$$T_{2_i} := T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right]$$

$T_{2_i} =$

	0
0	293
1	311.218
2	327.379

$$T_{4_i} := \frac{T_3 \cdot T_1}{T_{2_i}}$$

$T_{4_i} =$

	0
0	$1.363 \cdot 10^3$
1	$1.283 \cdot 10^3$

$$\eta_{thermalreg_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i} \right)^{\frac{k_g - 1}{k_g}}} \right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1}{\eta_c} \right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right] \right] - \sigma_r \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i} \right)^{\frac{k-1}{k}} - 1 \right] \right] \right] \right]}$$

$$\sigma_{r2} := 0.55$$

$$\eta_{thermalreg2_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i}\right)^{\frac{k_g - 1}{k_g}}}\right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1}{\eta_c}\right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right] - \sigma_{r2} \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right]\right]}$$

$$\sigma_{r3} := 0.65$$

$$\eta_{thermalreg3_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i}\right)^{\frac{k_g - 1}{k_g}}}\right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1}{\eta_c}\right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right] - \sigma_{r3} \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right]\right]}$$

$$\sigma_{r4} := 0.75$$

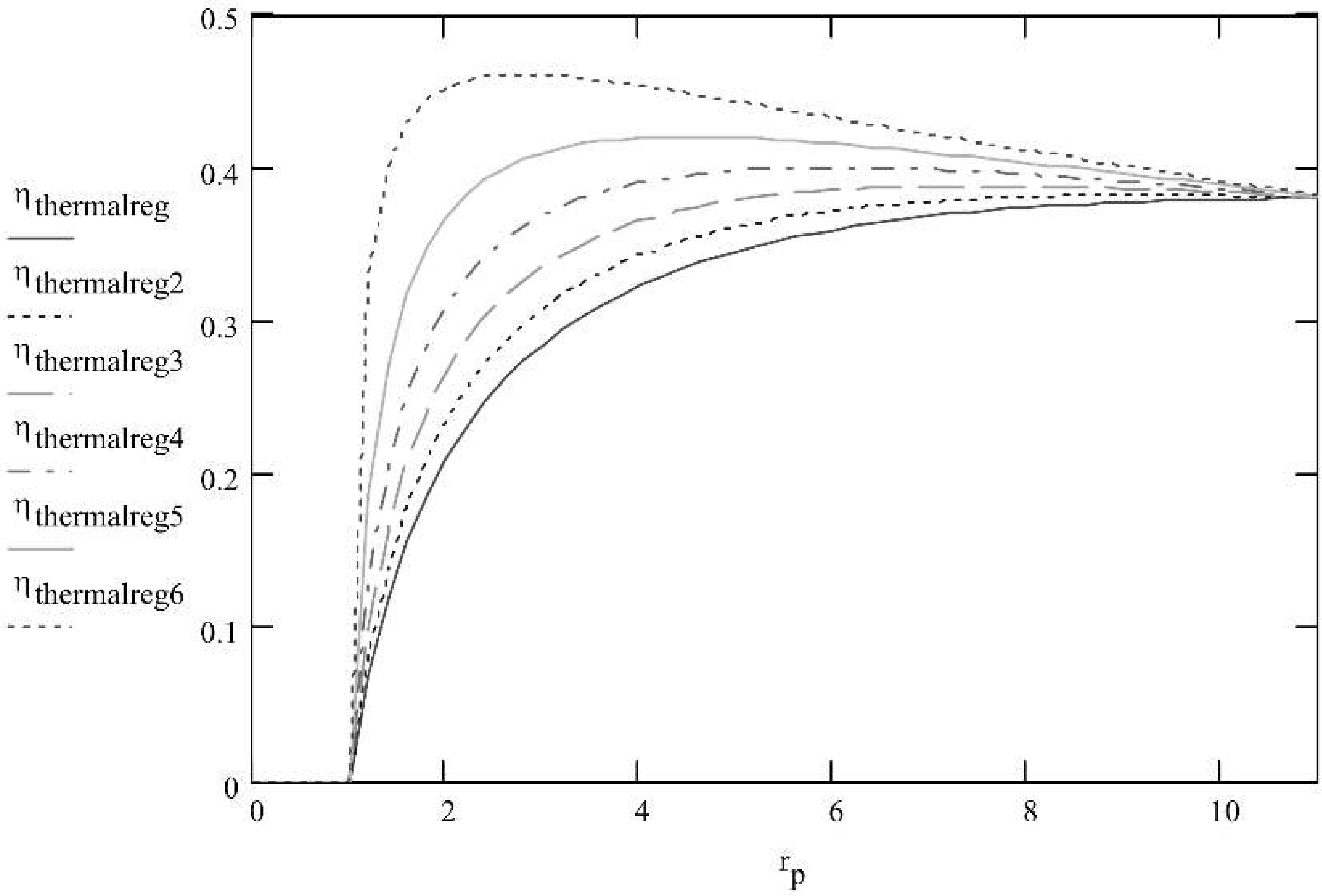
$$\eta_{thermalreg4_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i}\right)^{\frac{k_g - 1}{k_g}}}\right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1}{\eta_c}\right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right] - \sigma_{r4} \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right]\right]}$$

$$\sigma_{r5} := 0.85$$

$$\eta_{thermalreg5_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i}\right)^{\frac{k_g - 1}{k_g}}}\right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1}{\eta_c}\right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right] - \sigma_{r5} \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right]\right]}$$

$\sigma_{r6} := 0.95$

$$\eta_{thermalreg6_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left[1 - \frac{1}{\left(r_{p_i}\right)^{\frac{k_g - 1}{k_g}}}\right] - c_p \cdot T_1 \cdot \left[\frac{\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1}{\eta_c}\right]}{c_p \cdot \left[\left[T_3 - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right] - \sigma_{r6} \cdot \left[T_{4_i} - T_1 \cdot \left[1 + \frac{1}{\eta_c} \cdot \left[\left(r_{p_i}\right)^{\frac{k-1}{k}} - 1\right]\right]\right]\right]}$$



6. The dependence of the supplied heat on the compression ratio

$T_{11} := 283$

$k := 1.4$

$\sigma_r := 0.6$

$r_p := \text{READPRN}(\text{"graph3.txt"})$

$\eta_c := 0.86$

$T_3 := 1363$

$c_p := 1004.5$

$$T_2 := T_{11}\cdot\left[1 + \frac{1}{\eta_c}\cdot\left(r_p^{\frac{k-1}{k}} - 1\right)\right]$$

$r_p =$

	0
0	2
1	2.2
2	2.4
3	2.6
4	2.8
5	3
6	3.2
7	3.4

$$T_4 := \frac{T_3\cdot T_{11}}{T_2}$$

$T_2 =$

	0
0	355.071
1	366.145
2	376.521
3	386.296

$q_{11} := c_p\cdot\left[\left(T_3 - T_2\right) - \sigma_r\cdot\left(T_4 - T_2\right)\right]$

$T_4 =$

	0
0	$1.086\cdot10^3$
1	$1.053\cdot10^3$
2	$1.024\cdot10^3$
3	998.531
4	975.174
5	953.971

$q_{11} =$

	0
0	$5.717 \cdot 10^5$
1	$5.871 \cdot 10^5$
2	$6.004 \cdot 10^5$
3	$6.121 \cdot 10^5$
4	$6.225 \cdot 10^5$
5	$6.317 \cdot 10^5$

$T_{12} := 283$

$k := 1.4$

$\sigma_r := 0.7$

$r_{p2} :=$ READPRN ("graph3.txt")

$\eta_c := 0.86$

$T_3 := 1363$

$c_p := 1004.5$

$$T_2 := T_{12} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_{p2}^{\frac{k-1}{k}} - 1 \right) \right]$$

$r_{p2} =$

	0
0	2
1	2.2
2	2.4
3	2.6
4	2.8
5	3
6	3.2
7	3.4

$$T_4 := \frac{T_3 \cdot T_{12}}{T_2}$$

$$q_{12} := c_p \cdot \left[\left(T_3 - T_2 \right) - \sigma_r \cdot \left(T_4 - T_2 \right) \right]$$

$T_{13} := 283$

$k := 1.4$

$\sigma_r := 0.8$

$r_{p3} := \mathrm{READPRN}\ (\text{"graph3.txt"})$

$\eta_c := 0.86$

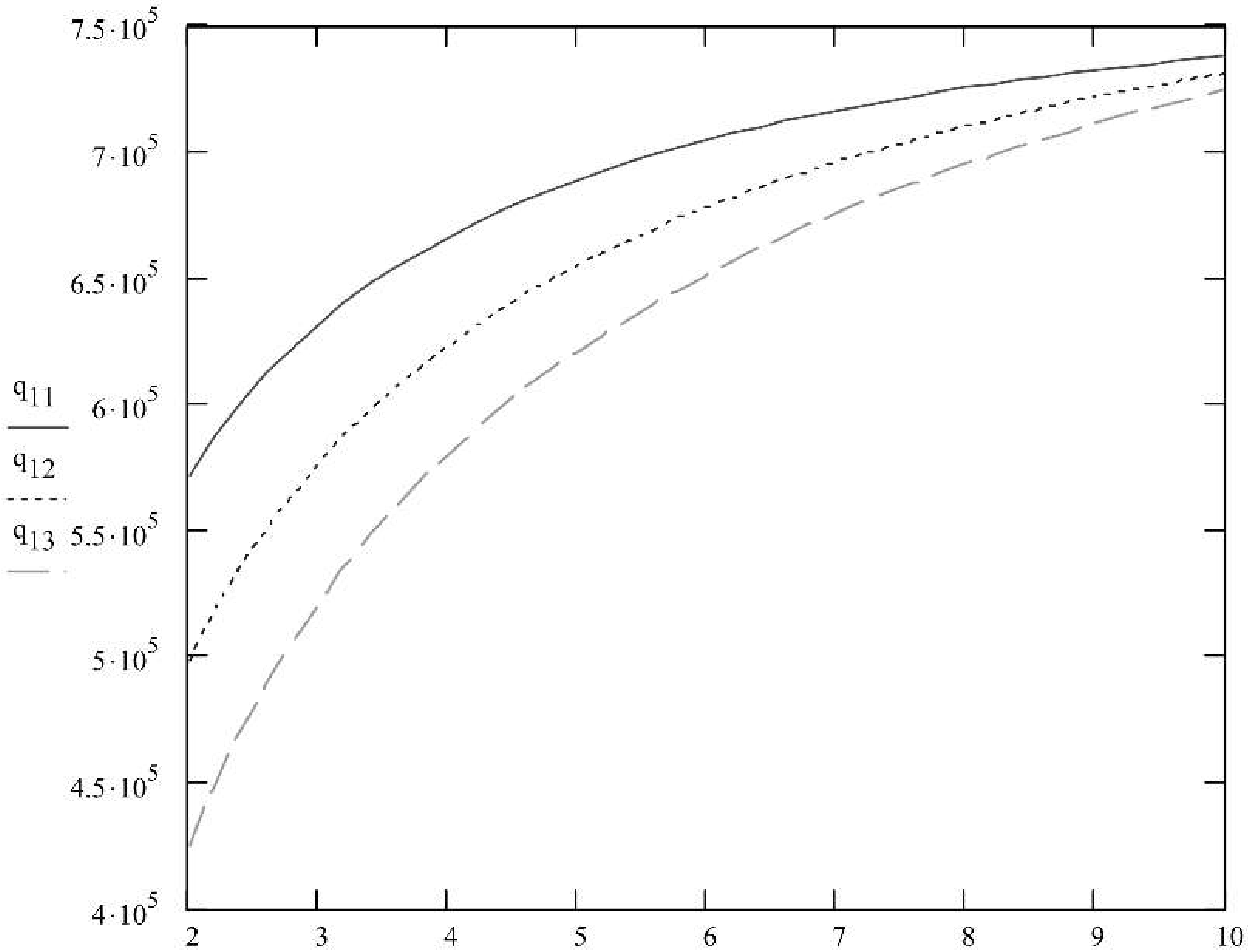
$T_3 := 1363$

$c_p := 1004.5$

$$T_2 := T_{13} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_{p3}^{\frac{k-1}{k}} - 1 \right) \right]$$

$$T_4 := \frac{T_3 \cdot T_{13}}{T_2}$$

$$q_{13} := c_p \cdot \left[\left(T_3 - T_2\right) - \sigma_r \cdot \left(T_4 - T_2\right) \right]$$



7. Variation of thermal efficiency with change ambient temperature and regenerative effectiveness

$p_1 := 101300$

$r_p := 6.3$

$k := 1.4$

$k_g := 1.33$

$c_p := 1004.5$

$c_{pg} := 1120$

$\eta_t := 0.9$

$\eta_c := 0.86$

$T_3 := 1363$

$T_1 :=$ READPRN ("graph.txt")

$\sigma_{r1} := 0.6$

$\sigma_{r2} := 0.7$

$T_3 = 1.363 \times 10^3$

$T_1 =$

	0
0	283

$\sigma_{r3} := 0.8$

$i := 0..49$

$$T_{2_i} := T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1 \right) \right]$$

$$T_{4_i} := \frac{T_3 \cdot T_{1_i}}{T_{2_i}}$$

$T_2 =$

	0
0	510.694
1	512.499
2	514.303
3	516.108
4	517.912

$T_4 =$

	0
0	755.304
1	755.304
2	755.304
3	755.304
4	755.304
5	755.304
6	755.304
7	755.304
8	755.304
9	755.304
10	755.304
11	755.304
12	755.304
13	755.304
14	755.304
15	755.304

$$\eta_{thermalreg1_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left(1 - \frac{1}{r_p^{\frac{k_g - 1}{k_g}}}\right) - c_p \cdot T_{1_i} \cdot \left(\frac{r_p^{\frac{k-1}{k}} - 1}{\eta_c}\right)}{c_p \cdot \left[\left[T_3 - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right] - \sigma_{r1} \cdot \left[T_{4_i} - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right]\right]}$$

$\eta_{thermalreg1} =$

	0
0	0.388
1	0.387
2	0.387
3	0.386
4	0.385
5	0.384
6	0.384
7	0.383
8	0.382
9	0.381
10	0.38
11	0.38
12	0.379
13	0.378
14	0.377

$$\eta_{thermalreg2_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left(1 - \frac{1}{r_p^{\frac{k_g - 1}{k_g}}}\right) - c_p \cdot T_{1_i} \cdot \left(\frac{r_p^{\frac{k-1}{k}} - 1}{\eta_c}\right)}{c_p \cdot \left[\left[T_3 - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right] - \sigma_{r2} \cdot \left[T_{4_i} - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right]\right]}$$

$\eta_{thermalreg2} =$

	0
0	0.402
1	0.401
2	0.4
3	0.399
4	0.398
5	0.398
6	0.397
7	0.396
8	0.395
9	0.394
10	0.393
11	0.392
12	0.391
13	0.391
14	0.39

$$\eta_{thermalreg3_i} := \frac{c_{pg} \cdot T_3 \cdot \eta_t \cdot \left(1 - \frac{1}{r_p^{\frac{k_g - 1}{k_g}}}\right) - c_p \cdot T_{1_i} \cdot \left(\frac{r_p^{\frac{k-1}{k}} - 1}{\eta_c}\right)}{c_p \cdot \left[\left[T_3 - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right] - \sigma_{r3} \cdot \left[T_{4_i} - T_{1_i} \cdot \left[1 + \frac{1}{\eta_c} \cdot \left(r_p^{\frac{k-1}{k}} - 1\right)\right]\right]\right]}$$

$\eta_{thermalreg3} =$

	0
0	0.417
1	0.416
2	0.415
3	0.414
4	0.413
5	0.412
6	0.411
7	0.41
8	0.409
9	0.408
10	0.407
11	0.406
12	0.405
13	0.404
14	0.403