FTD-MT-24-306-70

AD 727960

FOREIGN TECHNOLOGY DIVISION

COMBUSTION CHAMBERS OF GAS TURBINE ENGINES

by

Yu. M. Pchelkin

Approved for public release;
distribution unlimited.

273

UNCLASSIFIED
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY *(Corporate author)* Foreign Technology Division Air Force Systems Command U. S. Air Force	2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED 2b. GROUP

3. REPORT TITLE

Combustion Chambers of Gas Turbine Engines

4. DESCRIPTIVE NOTES *(Type of report and inclusive dates)*

Translation

5. AUTHOR(S) *(First name, middle initial, last name)*

Yu. M. Pchelkin

6. REPORT DATE 1967	7a. TOTAL NO. OF PAGES 259	7b. NO. OF REFS 17

8a. CONTRACT OR GRANT NO. b. PROJECT NO. 3066 c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) FTD-MT-24-306-70 9b. OTHER REPORT NO(S) *(Any other numbers that may be assigned this report)*

10. DISTRIBUTION STATEMENT

Approved for public release; distribution unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Foreign Technology Division Wright-Patterson AFB, Ohio

13. ABSTRACT

~~This~~ The report is a textbook for students studying power-engineering and machine building and it may also be useful to technical and engineering personnel and to advanced aeronautical engineering students. The book deals with gas-turbine fuels, fuel production, processing, properties, and characteristics, combustion, combustion theory and products, combustion-chamber design, construction, and operation. (AM7018513)

DD FORM 1 NOV 65 1473

UNCLASSIFIED
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Combustion Chamber						
Combustion Engineering						
Combustion Lines						
Combustion Mixture						

FTD-MT- 24-306-70

EDITED MACHINE TRANSLATION

COMBUSTION CHAMBERS OF GAS TURBINE ENGINES

By: Yu. M. Pchelkin

English pages: Cover to 259

Source: Kamery Sgoraniya Gazoturbinnykh Dvigateley.
Izd-vo "Mashinostroyeniye," Moscow, 1967,
pp. 1-207.

This document is a SYSTRAN machine aided translation, post-edited for technical accuracy by: Francis T. Russell.

Approved for public release; distribution unlimited.

UR/0000-67-000-000

PREPARED BY:

TRANSLATION DIVISION
FOREIGN TECHNOLOGY DIVISION
WP-AFB, OHIO.

A

Ю. М. ПЧЕЛКИН

КАМЕРЫ СГОРАНИЯ ГАЗОТУРБИННЫХ ДВИГАТЕЛЕЙ

Допущено Министерством высшего и среднего
специального образования СССР
в качестве учебного пособия
для студентов энергетических,
политехнических и машиностроительных вузов

Издательство „МАШИНОСТРОЕНИЕ“
Москва 1967

TABLE OF CONTENTS

U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

FOLLOWING ARE THE CORRESPONDING RUSSIAN AND ENGLISH DESIGNATIONS OF THE TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}
rot	curl
lg	log

This is a textbook for the course in "Combustion Chambers of Gas Turbine Engines" for power, polytechnical, and machine construction institutions of higher learning.

The first part of the textbook considers the fuels for gas turbine engines, the methods of producing and processing them and their basic characteristics and properties. The conditions necessary for fuel combustion, and the composition of the initial and end product are determined. The second part is devoted to the fundamentals of the theory of combustion. In the third part, the elements of the working process, the principles of design, and the calculation of combustion chambers are explained and the construction of chambers and their basic components are considered. Comparative data are given on the basic parameters and characteristics of combustion chambers, as the basis for stationary and traction gas turbine engines. The procedure is given for calculation of a combustion chamber and reheat combustion chambers are discussed.

The sections of the book in which materials pertaining to liquid fuel and general calculations of combustion are examined, and also the bases of the theory of combustion and elements of analysis of the working process of combustion chambers, can be useful also for students in senior courses at aviation institutes. The book will be of definite interest to technical-engineering workers.

Reviewers: the faculty of Moscow
Aviation Institute and Candidate
of Techincal Sciences A. S. Shteynberg.

Editor Doctor of Technical Sciences
Prof. D. N. Vyrubov.

INTRODUCTION

The combustion chamber is one of the main components of a gas turbine engine (GTD). These engines are being used ever more widely in technology, and in the immediate future, doubtlessly, they will occupy one of the leading places not only in aviation, but also in power engineering, in railroad and automobile transport, in ships and other special purpose installations.

In creating a GTD, one of the critical problems is the calculation and designing of the combustion chamber. The function of the combustion chamber is to bring heat to the working substance in a gas turbine power plant (GTU). Most frequently the working substance is air, to which heat is applied by means of combustion of a specific quantity of fuel in it as an oxidizer.

A substantial advantage in a GTD is the possibility of utilizing in it diverse types of fuel, both gas and liquid, as well as solid.

The organization of the working process and the characteristics of the combustion chambers differ substantially from other fuel burning devices utilized in engineering.

Today, the creation of reliable engineering methods for the analytical calculation of combustion chambers represents one of the most important assignments in the course of the further development of gas turbine engines.

Reliability, starting properties, long service life, and economy of operation of the combustion chamber, to a considerable extent, determine the characteristics of GTD as a whole.

1. FUEL

Fuel in engineering is the name given to substances which are capable in a process of complex, basically chemical reactions, defined by total combustion, and liberate a considerable quantity of thermal energy which can be further usefully used by man.

The following requirements are imposed on the substances utilized in industry as fuel:

1. Their reserves in nature must be sufficient to permit economically rational means of their production.

2. The possibility of utilization of considerable portion of the energy which is liberated during the combustion of a unit of mass or volume of the given substance.

3. The products of combustion must permit using them in the capacity of a working substance in power and engineering installations.

4. The substances and the products of their combustion must be harmless to the process which uses heat of a combustion, and to the apparatus and, absolutely, to the maintenance personnel, as well as the surrounding animal and vegetable world.

5. The possibility of utilization of a sufficiently cheap oxidizer, most of all air, for the oxidation of these substances.

6. These substances must have stable basic characteristis and properties under a considerable change in the parameters of the environment.

Those requirements are satisfied to the greatest degree by substances of organic origin which possess as their base such elements as carbon and hydrogen.

The utilization of fuel in its natural state is frequently impossible, since in so doing, it does not manage to guarantee a number of the additional requirements advanced by the user. Furthermore, it is sometimes inexpedient to burn the fuel in a natual state, since in so doing, individual components of the natural fuel are burned which are of great value for other technological processes. For example, the individual components of petroleum are used in the chemical industry for the production of alcohol, fatty acid, synthetic rubber, etc. This has caused the appearance in the capacity of technical fuels of a number of products of the processing of natural fuel, which have received the name of artificial fuel.

The basis of the general classification is the division of fuel according to its state of aggregation and the method of production (Table 1).

Table 1. Classification of fuel.

State of aggregation	Fuel	
	Natural	Artificial
Gas	Natural and petroleum industry gases	Generator, coke, blast-furnace, illuminating, water, underground gasification of coals and other gases.
Liquid	Petroleum	Gasoline, kerosene, solar oil, mazut, and other products of oil refining; alcohol, coal tar, and others.
Solid	Firewood, peat, brown coal and hard coal, anthracite, bituminous shales.	Wood coke, coal coke, and semicoke, carbon dust, and others.

The value of fuel in industry and, specifically in power engineering, is enormous.

It is known, for example, that the heat of fuel constitutes the basic portion of world production of all energy. Thus, in 1952, the portion of energy obtained from fuel formed 97.6% of the total amount. This required the utilization of approximately 3500 billion kilograms of theoretical fuel (possessing a heat of combustion of 7000 kcal/kg or 29,300 kJ/kg). In the overall energy resources in 1952, the fractions of various types of fuel were as follows: coal - 41.4%; petroleum - 26.5%; natural gas - 9.3%; brown coal, peat, shale, a water power, and others - 22.8%.

A considerable quantity of fuel is used by traction devices, moreover about 80% of this fuel consists of liquid (petroleum) fuel.

The demands of industry in the utilization of thermal energy increases from year to year.

The high rates of growth in the production and consumption of various types of fuel call for both expansion of old bases, and wide exploration and exploitation of a great number of new deposits. Apart from this, work is being done to investigate means of more rational and economical utilization of fuel in all branches of industry. A decrease in heat losses and a reduction in the norms of fuel expenditure are the most important national-economic problems.

CHAPTER I

GENERAL CHARACTERISTICS OF FUEL

The most important characteristics fuel from the point of view of behavior and results obtained in the process of its combustion are: the composition of the fuel, the heat of combustion, and the character of change in the composition of the fuel in the process of heating.

The chemical composition of fuel - this is the determining characteristic upon which in many respects all the remaining characteristics depend.

§ 1. The Composition of Fuel

The basic combustible elements of fuel are carbon and hydrogen. However, all the fuel used in practice contains the elements mentioned not in a free state, but in the form of various compounds such as carbon with hydrogen, and with oxygen, nitrogen, sulfur, and a number of other elements.

An intermixture of all possible compounds of these elements, as a rule, with an admixture of certain incombustible substances (moisture, a mineral part - the ash) called ballast, is commercial fuel.

Chemical analysis should show, which compounds and in which quantity make up a fuel. However in practice as yet there is no possibility to make a complete and accurate chemical analysis for any fuel.

Commercial fuels according to the degree of possible accuracy of their chemical analysis can be divided into three groups.

The first group basically comprises gas fuel. The gas components (CH_4, CO_2, CO, C_2H_4, N_2, and others) yield easily to separate determination.

The second group is formed of various types of liquid fuel of petroleum origin, consisting of hydrocarbons of different type and different molecular weight.

The third group of fuels comprise substances of unknown chemical nature, to which in first place belong the mined coals.

For the first group of fuels, chemical analysis permits accurately establishing, in volume percents, the content of individual elements of a gas mixture:

Hydrogen	H_2
Oxygen	O_2
Nitrogen	N_2
Carbon monoxide	CO
Carbon dioxide	CO_2
Methane	CH_4
Heavy hydrocarbons (sum)	C_nH_m and others
Total	100%

The heavy hydrocarbon content C_nH_m usually is given as a sum, since the quantity of them in gas fuel rarely exceeds 1-2%. The moisture content is determined separately and designated as W g/m^3.

For fuels of the second and third groups it is very difficult or practically impossible to make an accurate chemical analysis, therefore it is customary to give the information about them for all calculations in the form of an elementary and technical analysis. For these fuels they determine in percents according to weight, the content of: C, H, O, N, S, mineral compounds (ash) *A* and moisture *W*. In this way, the obtained data characterize the fuel formally as an intermixture of individual elements, and, naturally, they are not able to reflect a number of the characteristics, depending on the properties of the specific compounds of these substances. Nevertheless, from these data it is possible to conduct the entire complex of calculations necessary for utilization of the fuel in one or another fuel-burning installation.

The results of elementary analysis are recorded in Table 2.

Table 2. Composition of fuel of the second and third groups.

Designation of the mass	Components						
	C	H	O	N	S	A	W
O	Organic mass						
Г	Combustible mass						
C	Dry mass						
P	Working mass						

The quantity of moisture of solid fuel is considered equal to the loss of mass during its drying to 105°C in a stream of inert gas. The ash content is determined according to mass after complete combustion of a fuel charge.

The results of elementary analysis thus can serve only for determination of the quantity of the substances taking part in the combustion and forming the products of combustion. For the analysis of the course of the process of combustion a number of supplementary data are used, specifically, the complex of the results of the technical analysis, which serves for the determination of fuels of the third group. Thus, for hard coal, technical analysis additionally gives the yield of volatile substances, the yield and the quality of coke, and furthermore, its heat of combustion.

§ 2. Heat of Combustion

Heat of combustion of a fuel is the name given to the quantity of heat in kJ (kcal), which is liberated in complete combustion of 1 kg (or 1 m^3 - for gases) of a fuel under prescribed conditions (see § 8, Chapter III).

The complete fuel combustion is characterized by higher heat of combustion Q^p_B with the reduction of water vapors in the products of combustion to a liquid state at 0°C. However, in practice, they use the quantity of lowest heat of fuel combustion Q^p_H, being determined under condition of conversion into products of combustion of the moisture of the fuel and of products of combustion of hydrogen in the form of steam. If the moisture of the fuel and hydrogen content in it, respectively W^p and H^p are given in percent by weight, then

$$Q^р_в - Q^р_н = 6(W^р + 9H^р) \text{ kcal/kg or } 25{,}1\,(W^р + 9H^р) \text{ kJ/kg.} \qquad (1)$$

The heat of combustion of a fuel is determined by two methods:

1) by direct measurement during combustion of the fuel in special instruments - calorimeters;

2) by computation according to the heat of combustion of individual combustible elements of the fuel.

In the second case, it is necessary to know the content of the individual combustible elements in the fuel and the thermal effects with their complete oxidation.

For a gas fuel on the basis of data of chemical analysis and thermal effects during combustion of individual elements, the expression for determining the lowest heat of combustion takes the form

$$Q_{\text{н}}^{\text{р}} = 4{,}19 \left(\frac{68\,220\,CO + 57\,810\,H_2 + 192\,400\,CH_4}{100 \cdot 22{,}4} + \right.$$

$$\left. + \frac{320\,400\,C_2H_6 + 123\,700\,H_2S}{100 \cdot 22{,}4} \right) \text{kJ/kg}. \qquad (2)$$

where CO, H_2, CH_4, etc. - content of components in % by volume.

For the second and third groups of fuels, having only the results of elementary analysis and thermal effects of the corresponding chemically pure elements, naturally, it is not possible to calculate the heat of combustion of the commercial fuel. However, much experimental material allows the investigators to propose empirical equations, making it possible to calculate with greater or less accuracy the heat of combustion of various fuels. The most widely used is D. I Mendeleyev's formula:

$$Q_{\text{в}}^{\text{р}} = 4{,}19\,[81 \cdot C^{\text{р}} + 300 \cdot H^{\text{р}} - 26\,(O^{\text{р}} - S^{\text{р}})]\ \text{kJ/kg}; \qquad (3)$$

$$Q_{\text{н}}^{\text{р}} = 4{,}19\,[81 \cdot C^{\text{р}} + 246 \cdot H^{\text{р}} - 26\,(O^{\text{р}} - S^{\text{р}}) - 6W^{\text{р}}]\ \text{kJ/kg}, \qquad (4)$$

where $C^{\text{р}}, H^{\text{р}}, O^{\text{р}}, S^{\text{р}}, W^{\text{р}}$ — respectively are the content of components in percent by weight for the working mass of the fuel. Values Q_{B}^{P} and Q_{H}^{P} in kcal/kg will be 4.19 times less than these values in kJ/kg.

§ 3. The Relationship of Fuel to Heating

The following basic characteristic - the relationship of the fuel to heating - likewise is intimately connected with its chemical composition, as is the heat of combustion. However, it is expedient to examine this in the sections devoted to the individual types of fuel. We will note here only the fact that during heating, some substances, even with a change in the state of aggregation, do not change the arrangement of molecules - these are the so called heat-resistant fuels. However, there are heat-resistant fuels in which at specified temperatures old molecules are destroyed and new ones are formed which are resistant at the given temperatures of the molecule. As an example of a heat-resistant fuel we can use the fuels which are close in composition to pure carbon. Hydrogen, and carbon monoxide are heat-resistant. Less resistant during heating are the fuels obtained from petroleum, and the gas fuels.

CHAPTER II

TYPES OF INDUSTRIAL FUEL

§ 4. Gas Fuel

The combustion of a gas fuel in Gas Turbine Engine [GTD] (ГТД) chambers can be effected more simply and more qualitatively than a liquid one and much more so than a solid fuel. Natural gas and a number of industrial gases are cheap, and such advantages of gas as the absence of ash, ease of transportation, and ease of mixing with a gas oxidizer, simplicity of control, and simplicity of servicing the apparatus, assure great promises for its use.

Natural gas may be widely used in the different types of GTD, used in power-stations, in industrial installations and installations operating main gas lines, etc.

Natural combustible gases are basically an intermixture of various hydrocarbons - mostly of the methane series and insignificant quantities of CO_2, N_2, CO, and others.

In the gases of various deposits, the methane content reaches 90% and more (such gases are called dry). Their heat of combustion comprises 7000-9000 kcal/m^3 (29,300-37,800 kJ/m^3) under normal conditions. Gases, where higher hydrocarbons and also methane serve as the base, are called fat gases; they have Q_H^P on the order of 7000-15,000 kcal/m^3 or 29,300-62,600 kJ/m^3. The approximate composition in percents and other data of natural gases of a number of deposits are given in Table 3.

Table 3. Composition and basic indices of natural gases.

Indices	Deposit				
	Saratov	Dashava	Shebelinka	Buguruslan	Ishimbay
Composition in % by volume:					
Methane CH_4	94.0	98.0	93.8	76.7	44.5
Ethane C_2H_6	1.2	0.5	4.0	4.5	17.4
Propane C_3H_8	0.7	0.2	1.0	1.7	16.5
Butane C_4H_{10}	0.4	0.1	0.5	0.8	5.4
Pentane C_5H_{12}	0.2	--	0.2	0.6	2.5
Hydrogen sulfide H_2S	--	--	--	1.0	5.0
Carbon dioxide CO_2	0.2	0.1	0.1	0.2	0.3
Oxygen O_2	--	--	--	--	1.7
Nitrogen N_2 and others	3.3	1.1	0.4	14.5	6.7
Lowest heat of combustion $Q^{р}_{н}$ of dry gas:					
in kJ/m^3	35,700	35,500	38,100	33,600	53,200
in kcal/m^3	8,550	8,500	9,130	8,050	12,710

Along with the natural gases, artificial gas fuel is widely used. They are produced in special installations - gas generators, in which various types of fuel are gasified with the partial oxidation of their combustible mass, making the so-called generator gases.

Most often the basic fuel is coal. The generator gas produced in this way depending on the oxidizer (the blast) differs in composition and heat of combustion. The possible production and utilization of gases are shown in Table 4.

The air-gas is obtained with the supplying of air to a bed of fuel according to the diagram represented in Fig. 1. The fuel is charged into the shaft of a gas generator from above through special equipment. The quantity of air applied under the grate bars is insufficient for the complete combustion of all the fuel and it is calculated only for the oxidation of part of it. This part of the fuel forms a so-called oxidation zone. The gas products of complete oxidation, rising up the shaft get into the next layer of fuel (into the reduction zone) where, being reduced, they gasify the basic mass of solid fuel. Considering that carbon is the basis of the solid fuel, the total effect which determines the action of these two zones can be described by the following equation:

$$2C + O_2 + 3{,}76N_2 = 2CO + \\ + 3{,}76N_2.$$

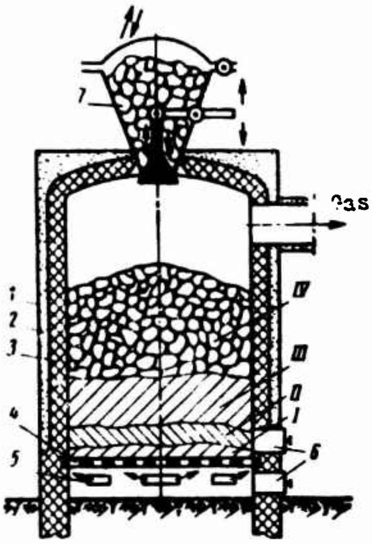


Fig. 1. Diagram of a gas generator: I - oxidation zone; II - reduction zone; III - dry distillation zone; IV - drying zone; 1 - shell; 2 - insulation; 3 - refractory lining; 4 - grate bars; 5 - openings for supply of oxidizer; 6 - openings for igniting the fuel and removal of slag; 7 - fuel for charging the shaft.

As a result, the gas contains 34.7% CO and 65.3% N_2 by volume. This is the theoretical composition of air-gas. In practice, it has insignificant quantities of various impurities: methane, hydrogen, oxygen, carbon dioxide, and others. The working heat of combustion of air-gas under normal conditions

$$Q^p_н \text{ to } 1000-1200 \text{ kcal/m}^3 \text{ or} \\ 4200-5000 \text{ kJ/m}^3.$$

Water gas is obtained in the supplying of water vapor to glowing carbon. The basic reaction is written thus:

$$C + H_2O_{nap} = CO + H_2 - \\ - 126\,300 \text{ kJ/kg}.$$

Here, in realization of the reaction a considerable quantity of heat is spent. Besides this, reactions of another type go on, also with expenditure of heat, for example:

$$C + 2H_2O_{nap} = CO_2 + 2H_2 - 84\,900 \text{ kJ/kg}.$$

Therefore, the layer of fuel must be heated, for example, by alternating the supply of steam and air, respectively interchanging the production of water-gas and air (blowoff)-gas, which is technological waste. In so doing, if it is necessary to have a regular supply of water gas to a consumer, then it is necessary to have two parallel operating generators and to change them over in turn.

Theoretically, water gas, as is evident from the basic equation, consists of 50% H_2 and 50% CO. The heat of the combustion of such gas is approximately 2.5 times higher than air-gas, and under normal conditions attains 2800 kcal/m^3 or 11,700 kJ/m^3. Practically, however, in the composition of water gas a number of supplementary components enter (CO_2, N_2, CH_4, and others), as a consequence of which is Q^p_H = 2400-2700 kcal/m^3 or 10,000-11,300 kJ/m^3.

A mixed gas is obtained, if the blowing is produced with a steam-air mixture. Such a process can be carried out continuously. The gas which is obtained in this case has average characteristics and composition, inasmuch as here there are products of both the water and air processes. In the production of combined gas many deficiencies in the production of air gas are removed: the high temperature of the shaft, which lowers the service life of the structure and magnifies the heat losses in the environment with the exiting gas. An approximate composition and the basic characteristics of generator gases are given in Table 4.

Table 4. Composition and basic indices of generator gases.

Indices	Air-gas from coke	Water gas		Blowoff gas		Mixed gas	
		from anthracite	from brown coal	from anthracite	from brown coal	from anthracite	from brown coal
Composition in % by volume:							
Hydrogen	0.9	48.0	50.0	2.3	11.0	13.5	14.0
Carbon monoxide	33.4	38.5	23.8	8.8	16.9	27.5	25.0
Methane	0.5	0.5	6.9	0.2	1.7	0.5	2.2
Heavy hydrocarbons	—	—	0.6	—	0.2	—	0.4
Hydrogen sulfide	0.4	0.4	0.2	0.1	—	0.2	1.2
Carbon dioxide	0.6	6.0	14.5	14.5	11.1	5.5	6.5
Nitrogen and others	64.2	6.4	3.8	73.9	58.9	52.6	50.5
Oxygen	—	0.2	0.2	0.2	0.2	0.2	0.2
Lowest heat of combustion							
in kcal/m^3	1100	2 465	2 700	350	970	1230	1410
in kJ/m^3	4600	10 300	11 300	1463	4050	5150	5900
Temperature of gas on outlet $t_{вых}$ in °C	800	675	270	700	335	750	330
Content of dust in gas in in g/m^3	10	—	—	10	45	10	45

The gas of underground gasification is obtained from the coal lying in a layer. The diagram of the process is this. Two holes are bored: one for forcing compressed air, the second for drawing off the gas. The two holes are connected by various methods, for example, a horizontal channel is burned through; for burning through is used condensed air and a special combustible. The permeability of the layer of coal ensures the passage of gases from the pressure hole to the gas withdrawing hole.

With underground gasification, the necessity for extracting the coal from the depths of the earth and its transportation are eliminated. In this instance, it is expedient to use the coal which cannot be developed in the usual manner. Some data on the underground gasification of gas are given in Table 5.

Table 5. Composition of gases from the underground gasification of coals.

Coal deposit	Composition in % by volume						Q_{n}^{p}	
	CO_2	CO	H_2	CH_4	H_2S	N_2+O_2	in kcal/m³	in kJ/m³
Moscow region	9—10	10.0	14—15	1.5—2.0	0.4—0.8	63—64	865	3620
Gorlovka.......	9—11	15—19	14—17	1.4—1.6	—	53—55	1000	4190

A whole series of gas fuels are obtained as byproducts, i.e., the technological waste of various processes: the thermal processing of petroleum (cracking and pyrolysis of petroleum products), coking and semicoking of coal, the blast-furnace process, and others. The approximate composition of the gases thus produced is given in Table 6.

Table 6. Composition of gases produced in various technological processes.

Gas	Composition in % by volume							$Q_{н}^{р}$ in kcal/m³ (kJ/m³)	$Q_{в}^{р}$ in kcal/m³ (kJ/m³)
	CO_2	CO	O_2	H_2	CH_4	C_nH_m	N_2		
Coke from coal........	3	6	1	56	22	2	10	3 780 (15 900)	4 280 (17 900)
Coke from shale.......	15	16	—	39	24	3	3	3 960 (16 600)	4 410 (18 500)
Semicoke from coal........	13	9	—	9	54	7	8	6 100 (25 300)	6 740 (28 200)
Petroleum.....	—	3	—	12	58	27	—	9 190 (38 400)	10 070 (42 100)
Blast furnace from coke....	10.5	28	—	2.7	0.3	—	58.5	940 (3 900)	960 (4 000)

§ 5. Liquid Fuel

The only natural source of liquid fuel is petroleum. Crude petroleum is not used as a fuel. Various products of its processing are used as commercial fuel, which permits using all the petroleum components more fully and expediently. Petroleum of various deposits frequently differ sharply from one another, however their basic component element is always carbon. Its content in petroleum reaches 85-87% by weight. The remaining part consists of hydrogen (up to 12-14%) and a very insignificant quantity (up to 1%) of oxygen, and also sulfur. True, a number of petroleums contain sulfur in large quantities (up to 3-4%). Such petroleums are called high-sulfur. All the mentioned elements, naturally, are not found in petroleum in a free state but in the form of a mixture of various organic compounds basically hydrocarbons of different molecular weights. Most frequently, the hydrocarbons belong to the methane series (C_nH_{2n+2}), or to the naphthenic series (C_nH_{2n}), or the aromatic (C_nH_{2n-6}). Accurate determination of the type and quantity of the individual compounds, which are contained in petroleum by chemical analysis is very complicated.

Usually petroleum and various grades of industrial liquid fuel obtained from it are characterized by its distillation into fractions which boil away at various temperatures.

The synthetic liquid fuels are basically products of petroleum refining. The simplest method is the distillation of petroleum, i.e., the evaporation of a number of its fractions during heating to a specified temperature with the subsequent condensation of the vapors. Thus, during heating to 200°C the fractions of various kinds of gasoline are separated, the density of which varies from 0.72 to 0.76. On the average, gasoline consists of 85% C and 15% H.

Its lower heat of a combustion $Q_H^p = 10{,}500$ kcal/kg or 43,900 kJ/kg. In modern gas turbine engines gasoline and even ligroin are hardly ever used as a fuel. Usually, GTD operate on heavier types of liquid fuel, for example, on kerosene, diesel fuel, solar oil, and mazut. Kerosene is a mixture of the petroleum fractions which boil away at a temperature of from 220-250 to 300-315°C. The density of kerosene varies from 0.79 to 0.87. Solar oil is driven off at temperatures from 280 to 360°C; its density is equal to 0.87-0.90.

The yield of light fractions from the petroleum of the majority of deposits most frequently comprises 25-30%. The remaining part - mazut, is a mixture of heavier hydrocarbons. The mazuts of various petroleums have approximately the following composition: $C^r = 86.0$ to 87%; $H^r = 12.0\text{-}13.0\%$; $O^r = 0.0\text{-}0.4\%$; $N^r = 0.0\text{-}0.5\%$; $S^r = 0.0\text{-}4.0\%$; $W^p = 0.0\text{-}10.0\%$. The heat of combustion of mazuts $Q_H^p = 9{,}500\text{-}10{,}200$ kcal/kg or 39,800-42,600 kJ/kg. Mazut is also used as a fuel and as a raw material for production of oils, light liquid fuel, and others. By using the cracking methods (splitting) of petroleum products, it is possible additionally to obtain considerable quantities (up to 40% of the weight of the mazut) cracking products - gasoline, kerosene, and others.

The cracking process consists in the thermal decomposition of heavy hydrocarbons, which takes place at temperatures of 300-700°C at atmospheric pressure or at increased pressure (to 50-100 [atm(abs.)] and more). Let us note that a certain quantity of liquid fuel is produced today by methods of liquefaction of gases and by the special processing of solid fuel (for example, by the method of hydrogenation of coals).

Liquid fuels are characterized by a number of physico-chemical properties.

1. *Viscosity* - the property of a fluid to resist the relative motion of particles causing the appearance of forces of internal friction between layers if the latter have various rates of motion. Viscosity is usually measured in degrees of conventional viscosity [°VU] (°ВУ) at a specific temperature *t*°C.

Conventional viscosity is the term applied to the ratio of the time of flow of 200 milliliters of a fluid (fuel) having a given temperature, through a calibrated opening to the time of flow of the same quantity of water at 20°C. At the given temperature the connection of VU units with kinematic viscosity expressed in Stokeses (cm^2/s) is the following:

$$v = \left(0{,}07319^{\circ}\mathrm{ВУ} - \frac{0{,}0631}{^{\circ}\mathrm{ВУ}}\right)\ cm^2/s.$$

At an increase in the temperature of a petroleum product, its viscosity is sharply reduced to determined boundaries whereupon it changes very slightly.

A number of typical dependences of viscosity on temperature is indicated in Fig. 2. Gasoline, kerosene, and diesel fuel have little viscosity (up to 2-5° VU) even at reduced temperatures (to 10-30°C). This permits using these fuels without preliminary preheating and obtaining good atomization in nozzles of various types. Heavier fuel (mazuts) even at temperatures of 10-20°C has considerable viscosity (5-20° VU and above) which impedes its pumping through pipelines, and rather fine atomization of such a fuel is generally impossible without first preheating it to 70-150°C and higher.

For a heavy liquid fuel, the change in viscosity depending on temperatures *t* can be determined from the following approximation expression:

$$^{\circ}\mathrm{ВУ}_t = {}^{\circ}\mathrm{ВУ}_{50}\left(\frac{50}{t}\right)^{n}.$$

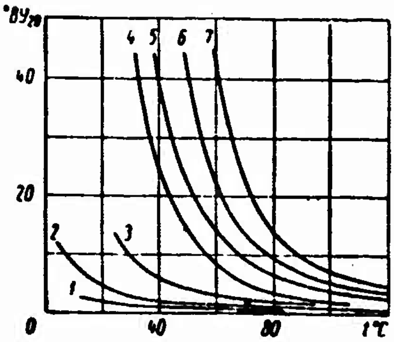


Fig. 2. Change in viscosity of a heavy liquid fuel depending on temperature: 1 - diesel fuel; 2 - solar oil; 3 - sulfurous mazut FS-5; 4 - naval mazut F-12; 5 - naval mazut F-20; 6 - fuel mazut 40; 7 - fuel mazut 80.

The values of the index of degree *n* at various values of conventional viscosity at 50°C ($°VU_{50}$) are provided below:

$°ВУ_{50}$	2	5	10	15	20
n	1,8	2,3	2,6	2,75	2,85

2. *Density of fuel* d_4^{20} is the ratio of the mass of the petroleum product in a given volume at a temperature of 20°C to a mass of water in the same volume at 4°C. In practice, the density of a liquid fuel is most frequently determined by an areometer. Its dependence on temperature can be found from the equation

$$d_4^t = d_4^{20} - a(t - 20).$$

The values of the temperature factor *a*, depending on d_4^{20} are provided below:

d_4^{20}	0,8	0,82	0,84	0,86	0,88	0,90	0,92	0,94	0,96	0,98	1.00	1,02
$a \cdot 10^4$	7,65	7,53	7,12	6,86	6,60	6.33	6,07	5.81	5.56	5,30	5,02	4.76

3. *Heat capacity of a liquid fuel* - this is the quantity of heat necessary to raise the temperature of 1 kg of fuel one degree. Heat capacity increases with a decrease in density, and also with

an increase in the temperature of the fuel. It is determined according to Cragoe's empirical equation:

$$c_t = \frac{4{,}19}{\sqrt{d_4^{15}}}(0{,}403 + 0{,}00081 \cdot t) \text{ kJ/(kg·degree).}$$

In calculations it is customary to take the value of heat capacity equal to 0.4-0.5 kcal/(kg·deg) or 1.65-2.10 kJ/(kg·deg) depending on the density of the fuel.

4. *Solidification point* - this is the temperature at which a fuel loses its mobility. This parameter, specifically, determines the possibility of pumping fuel through pipelines.

5. *Flash point* - this is the minimum temperature at which close to the surface of a fuel its vapor content under given conditions forms a combustible mixture capable of igniting from an outside flame.

6. *Ash content of fuel* - is the content of ash in a fuel. It rarely exceeds 0.10-0.15% by weight and it depends on the deposit and the method of petroleum refining, and also on the conditions of storage and transportation of the fuel. In clear petroleum products ash is practically absent. The remaining heavy fuels frequently have considerable ash content. In the combustion of a heavy liquid fuel (mazut) ash-forming substances make such compounds as oxides of various metals, sulfur, silicon, vanadium, and also sulphates and other compounds which can be deposited on elements of the blading of a turbine and cause intensive corrosion. Deposits are formed basically because of sodium compounds, vanadium anhydride, and other more complex compounds of vanadium and sodium, which are found in the flow of gases in molten form. The ash deposits reduce the aerodynamic qualities of the blades and change the flow area of channels, and as a result considerably reduce the power of a GTD. The corrosion of blades and other elements of a turbine is a chemical process which is sharply intensified with an increase in the temperature. The severest corrosion of GTD parts is caused by vanadium pentoxide V_2O_5,

and also sodium sulfate Na_2SO_4. The most dangerous is vanadium corrosion which is sharply intensified in the presence of sodium sulfate at a temperature of 650-700°C and higher.

At a temperature higher than 800°C, sodium sulfate is also able to dissolve the protective layer of the metal of the blades and cause corrosion.

Figure 3 shows the results of gas turbine engine tests showing the dependence of intensity of formation of deposits on the ash content in the fuel and the temperature of gases before a turbine. The plant worked on each fuel (hourly expenditure 25 kg/h) for 12 h. The initial heavy liquid fuel (curve 1) contained 1.8% sulfur, 0.05% ash (in which V_2O_5 - 0.03%, Na_2O - 0.005% and Fe_2O_3 - 0.001%). The results of tests on the same fuel with an addition of distillate containing ash in a quantity of 4.5 and 7% are represented by curves 2 and 3 respectively. It has been clarified that the intensity of formation of deposits increases with an increase in the pressure of gases and, naturally, with a decrease in excess air behind the combustion chamber. Figure 4 shows the dependence of formation of deposits in the blading of a turbine at various gas temperatures on the length of operation on a heavy liquid fuel containing 1.9% sulfur, 0.04% ash (V_2O_5 - 0.018%, Na_2O - 0.012%, and Fe_2O_3 - 0.002%). To detect the effect of individual ash components of a heavy liquid fuel on the character of deposits in the blading of a gas turbine the following experiment was conducted. A 500 horsepower gas turbine plant was operated on gas oil (containing practically no ash) with the artificial addition of various ash elements of a heavy liquid fuel. In so doing, the change in the turbine drag was determined (Fig. 5).

It has been established by experiments, that vanadium pentoxide, the melting point of which is 675°C, in molten state has exceptional ability to dissolve metal oxides from the surface of parts, causing the most severe corrosion. The corrosion of GTD parts is considerable even if vanadium content in the fuel is very small. Figure 6 gives the results of tests on a 750 kilowatt GTD, of 30 hours duration each

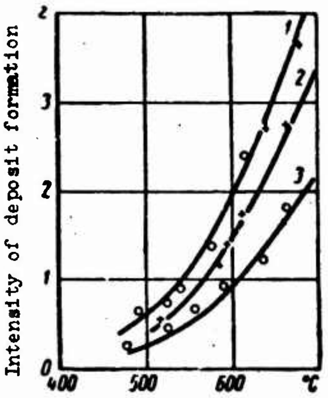


Fig. 3.

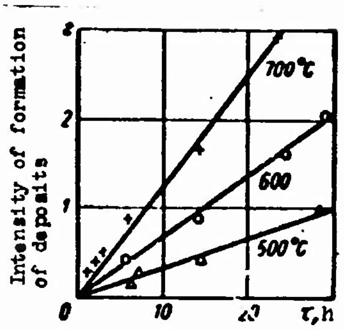


Fig. 4.

Fig. 3. The effect of ash content of a fuel and the temperature of a working medium on formation of deposits: 1 - heavy liquid fuel; 2 - mixture with a ratio of heavy fuel/distillate = 21/1; 3 - mixture with a ratio of heavy fuel/distillate = 11/1.

Fig. 4. Effect of test duration on formation of deposits.

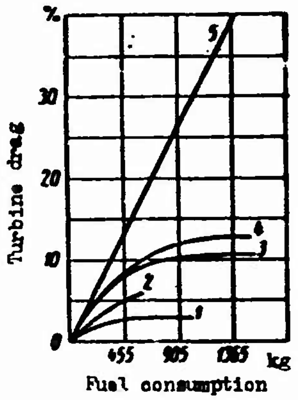


Fig. 5.

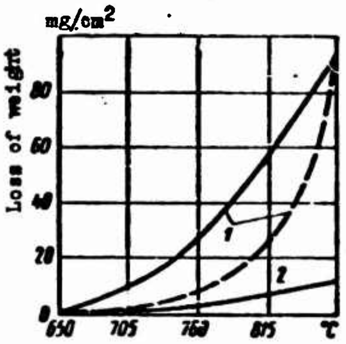


Fig. 6.

Fig. 5. Effect of component parts of ash on turbine drag: 1 - gas oil + Al (as Al_2O_3) 1800 ppm; 2 - gas oil + Mg (as MgO) 600 ppm; 3 - gas oil + Ca (as CaO) 860 ppm; 4 - gas oil + V (as V_2O_5) 430 ppm; 5 - gas oil + Na (as Na_2O) 280 ppm.

Fig. 6. Effect of temperature on corrosion of blades: 1 - alloy on an iron base; 2 - alloy on a nickel base; dotted curve - fuel contains only vanadium; solid curves - fuel contains vanadium, sodium, sulfur.

at a different temperature, running on fuel, containing two types of impurities: 1 - 0.025% vanadium and 2 - 0.C25% vanadium, 0.005% sodium, and 3.00% sulfur.

Corrosion [the loss of weight (mass) into mg/cm^2] is sharply magnified with an increase in temperature, especially in the presence of other components, where the corrosion of nickel alloys increases less than alloys on an iron base. Experiments indicated that, other conditions being equal, corrosion takes place most intensely in the first time period. This is evident from the following data:

Number of hours worked	10	50	100
Specific loss of weight (mass) in mg/(cm^2·h)	0.52	0.18	0.15

With the appearance of deposits on the parts, the intensity of corrosion increases sharply, as a result of which the turbine blades sometimes are destroyed after a few tens of hours. All this involves the utilization of a heavy liquid fuel in gas turbines in the case where the ash of the fuel contains a considerable quantity of injurious elements - basically vanadium, and also sodium. In this case also their absolute and relative quantity are important. Another undesirable element is sulfur. True, sulfur alone, without vanadium and sodium, does not cause specific hindrances when using the fuel even when its content reaches 3-5%.

For prevention of corrosion of the blading of gas turbines which operate on heavy liquid fuel containing injurious elements in the ash part, structural materials are used which are resistant to vanadium corrosion. It has been established, for example, that chrome-nickel alloys are more resistant than simple austenitic steel. The addition of a specific quantity of such elements as titanium, cobalt, silicon, and others to chrome-nickel alloys in a number of cases increases the resistance of steel to vanadium corrosion.

Limitation of corrosion and simultaneously the deposits of ash elements in the blading of a GTD, without considering means for reducing the temperature of the gases before the turbine to 600-700°C can be achieved by the following methods.

a) By introduction of special additives into the fuel, which, by changing the chemical and physical properties of the ash, increase its melting point. Kaolin, zinc oxide, magnesium oxide, aluminum oxide, dolomite, and others are used as additives. The powders of these substances, which possess particles in size up to 15 μm, in a quantity, as a rule, of from 50 to 200% by weight (mass) of the ash of the fuel are thoroughly intermixed with the fuel in special installations directly before its combustion or earlier, without permitting their precipitation. The cost of fuel in connection with the use of additives, as a rule, is not increased more than 2-3%. A more suitable method is the introduction into the fuel of additives which are soluble in it in a quantity of 1-3% by weight (mass) of ash.

b) By flushing the fuel. The solvents of salts of sodium, potassium, and others are able after their separation from the fuel to sharply reduce the content of injurious elements in it. Elimination of the flushing solution, most frequently by centrifuging methods can easily be provided for with a difference in specific weights of 3-5%. For flushing of fuels, aqueous solutions of magnesium sulphate, aluminum sulphate, calcium nitrate, and others are being used successfully.

c) By controlling the fuel combustion process. The essence of the method consists in changing the sizes of the fuel droplets being taken into the combustion chamber, limiting its combustion so that a certain residue would contain solid carbon capable of holding the ash, which in such form becomes relatively harmless. The process of the fuel combustion in this instance proceeds according to the following scheme: initially from a drop of fuel, fractions are evaporated and burned up which possess a low boiling point, then the heavy hydrocarbons are split and chemically carbon is partially liberated. The final part of the process is the combustion of carbon. The organometallic and inorganic compounds forming an ash have relatively low vapor pressure and therefore they are concentrated in the residual carbon nucleus. With incomplete combustion, these particles with great ash concentration are carried through the turbine and do not produce considerable deposits.

The most effective means for reducing corrosion and ash deposits in GTD is a combined utilization of the means mentioned. For example, frequently the method of flushing of the fuel with subsequent introduction of additives is used. In order to avoid the injurious action of the ash of a heavy liquid fuel, the content of vanadium and sodium in it should not exceed 0.0001-0.0005% and as a maximum must not be more than 0.001%. Sulfur content in the fuel usually does not exceed 2.0-3.0%, and content of mechanical impurities and water - respectively 0.1 and 0.5%.

Stationary and transport locomotive gas turbine engines basically use a heavy liquid fuel (mazut). To ensure qualitative and reliable action of the fuel supply apparatus and the engine as a whole, gas turbine fuels should have the following indices:

Solidification point in °C, not higher than..	10
Viscosity into °VU, not more than...........	6
Content in %, not to exceed:	
Sulfur...................................	3.5
Ash......................................	0.08
Vanadium.................................	0.001
Sodim....................................	0.001
Water....................................	0.5
Heat of a combustion in kcal/kg (kJ/kg)......	9,000-10,000 (37,650-41,900)

In stationary GTD the following types of commercial liquid fuel are used:

a) solar oil All-Union State Standard (GOST) 1666-51;

b) naval grade mazuts F-12 and F-20, GOST 10585-63 (a fuel of higher quality with a low solidification point);

c) naval mazut from the eastern sulphurous petroleum deposts, FS-5, GOST 10585-63;

d) petroleum fuel for locomotive GTD, GOST 10433-63;

e) gas-turbine fuel per Office of Technical Services, Petroleum Industry (UTU NP) [УТУ НП] 68-61.

In addition, for the purpose of utilization in Gas Turbine Power Plant [GTU] (ГТУ), the following fuels are being tested:

1) petroleum fuel (fuel mazut) low-sulfur, sulphurous, and high-sulfur grades: 20, 40, 60, 80, 100 GOST 10585-63;

2) ukta fuel (the residue of direct distillation of ukta of petroleum) possessing high viscosity;

3) heavy yareg, crude petroleum devoid of light gasoline fractions, and others.

For aircraft GTU today basically kerosene is used of grades T-1, TS-1, T-2 (GOST 10227-62), grade T-5 (GOST 9145-59), cracking-kerosene T-4, and others. The average composition of kerosene is: $C^p = 86\%$ and $H^p = 14\%$. The best fuels are T-1 and TS-1 (with a somewhat greater sulfur content). They contain ligroin-kerosene fractions of direct distillation of petroleum. Both fuels are stable, and retain their properties well during prolonged storage. The high density of kerosene T-1 makes it especially valuable since it provides an aircraft with greater range with the usual capacity of the tanks.

A number of the most important characteristics of the fuels for aircraft GTD are strictly regulated, which are determined according to the specifics of their operation. Thus, the solidification point must always be below -60°C. The solubility of water in fuels is usually less than 0.005% at t = 10-20°C, since at reduced temperatures the solubility of H_2O decreases, which leads to the separation of ice crystals clogging the filters. To eliminate this occurrence additives of isopropyl alcohol 0.5-1.0% by weight and other additives are introduced. In order to avoid the formation of vapor corks, the vapor pressure at specified temperatures is regulated. For all aircraft fuels and especially the fuels which are used in supersonic aircraft, the stability of properties at high temperatures is important.

In Table 7, a number of the basic characteristics of aircraft, stationary, and transport GTD fuels are given.

§ 6. Solid Fuel

The natural types of solid fuel are: fire-wood, peat, brown coals, coal, anthracite, and bituminous shales. It has been established that fossil solid fuel of all types is the product of decomposition and conversion of the residues of the ancient vegetable and, partially, the animal kingdom.

In gas turbine engines it is possible to use solid fuel of all types, however, the coals represent the basic interest. For the calculations connected with their combustion it is necessary to know the results of the so-called technical analysis, which indicates moisture content, the escape of volatile substances, the residue, and the quality of the nonvolatile solid mass (coke), the quantity of ash, sulfur, and the heat of combustion of the fuel. Moisture is determined, for example, by suspending the fuel prior to and after heating to 105°C in a current of inert gas. The yield of volatile components and coke are determined by the suspension; in so doing, a weighed sample of coal is roasted (at a temperature of 850 ± 25°C) in a closed crucible for 7 min. The results obtained are expressed in percents by weight referred to dry coal. The volatile substances are the gas products of decomposition of thermally unstable molecules of the combustible mass of the coal. The coke, remaining after the

Table 7. Basic characteristics of fuels utilized in GTD.

Fuel characteristic	Fuel					
	T-1 GOST 10227-62	TS-1 GOST 10227-62	Diesel GOST 4749-49	Naval mazut		Petroleum for locomotive GTD GOST 10433-63
				F-12 GOST 10585-63	FS-5 GOST 10585-63	
Density (d_4^{20})	0,80—0,85	min: 0,775	0,84	0,927	0,91	0,935
Kinematic viscosity in cSt... in cSt:						
at 0°C..................	4,0, min	2,5	—	—	—	—
at 20°C..................	1,5, min	1,25	—	—	—	—
Viscosity °VU$_{50}$(at 50°C)......	—	—	2,7	12,5	3,7	2,0
Temperature in °C:						
beginning of crystallization	—60, max	—60, max	—	—	—	—
ignition (in a closed crucible)...............	30	28	95	135	95	65
solidification..............	below —60	below —60	—20	—8	—3	5
Content in %:						
sulfur (total)...............	0,1, min	0,25	0,1	0,8	2,0	3,0
ash.........................	0,005, min	0,005	0,01, min	0,04	0,25	0,02
mechanical impurities.......		absent		—	—	0,04
water.......................	absent		0,06, min	0,1	0,1	absent
vanadium....................	—	—	—	0,0003	0,0005, min	0,0007, min
gummy substances............	—	—	—	—	—	25, min
Lower heat of combustion Q_H^p:						
in kcal/kg..................	10 250	10 250	10 300	10 060	9 950	9 475
in kJ/kg....................	43 000	43 000	43 100	42 000	41 650	39 650

heat treatment of the weighed sample of coal, according to external form is divided into the following forms:

a) non-caking (powdered);

b) conglomerate (one piece, but consisting of individual particles);

c) caking (a single piece) without individual grains.

The quantity of ash is determined by repeated heat treatment of a weighed sample of coal (1-2 grams) in an open crucible with subsequent weighing. The ash content is given in percent by weight, referred to dry coal. The total sulfur content is usually determined by combustion of a weighed sample in a mixture of magnesium oxide and soda with subsequent dissolution of the sulphates formed and weighing. The heat of combustion of the fuel is determined in a calorimeter.

In technical and elementary analyses of a fuel all the basic data are obtained which are necessary for the calculations connected with its combustion, and, in addition, certain information which is required for organizing an effective combustion process (Table 8).

To obtain reliable technical analysis data it is very important to select the fuel sample correctly. Usually, from 100 tons of fuel, for investigation they select from different batches equal 400 kg quantities. Further, the fuel is pulverized and mixed, and then they reduce this quantity by a special method of selection. Finally, 200-300 grams of coal dust is treated to analysis.

In order to most fully, effectively, and economically use solid combustible minerals, they are first processed.

Methods exist for the physical-mechanical processing of natural solid fuel, connected with the chemical conversion of the ignitable mass of the fuel. These include: drying, sorting, enrichment, briquetting, and crushing. In addition, there are physico-chemical

Table 8. Approximate composition of various solid fuels.

Fuel	Content in % by weight (maximum)							Natural moisture in %	Content of volatile substance in %	$Q_{н}^{р}$ in kJ/kg (in kcal/kg)
	$C^{г*}$	$H^{г}$	$O^{г}$	$N^{г}$	$S^{г}$	$A^{р}$	$W^{р}$			
Wood..............	45—50	5—6	40	0,6	—	1—2	45—55	50	80—87	7 120—9 220 (1 700—2 200)
Peat..............	55—60	5,5—6,0	33	2,5	0,3	3,0—9,0	25—45	75—85	70	10 480—12 600 (2 500—3 000)
Bituminous shales...	60—70	6—10	9—11	0,2—0,7	—	20—50	15—20	—	75—90	5 450—9 640 (1 300—2 300)
Brown coals.........	65—75	4,0—6,0	20	1,0—1,5	1,0—3,0	10—20	5,0—3,0	—	30—40	12 500—14 600 (3 000—3 500)
Coals..............	80—96	2,0—6,0	2,0—5,0	1,0—2,0	0,5—5,0	3—12	0.5—4.0	—	10—30	31 400—35 600 (7 500—8 500)

---

*Superscripts г and р respectively define the content of the given element in the so-called *combustible* or *working mass* of the fuel.

methods, connected with the change in the structure of a substance: gasification, liquifying, charcoal burning, semicoking and coking.

Physical-Mechanical Methods of Processing Solid Fuel

Basically, fuels such as fire-wood and peat undergo natural and artificial drying. Mineral coals are most frequently *sorted* into indivisual fraction (according to the size of the pieces). For *enrichment* of a fuel, the incombustible components are separated from the combustible mixture. Now this method is acquiring ever greater importance. Layers of gangue, pieces of soil, and extraneous impurities in the coal mass sharply increase its ash content. The mineral substances deposited in the coal mass in the process of its formation give a specific part of the ash. This part of the ash can be separated only after crushing the coal. Frequently there are present in coal mineral substances which entered into the composition of the vegetable mass, which made the coal formations. It is a very difficult, and sometimes impossible to separate the ash of these substances from the coal. After enrichment, as a result of the increase in the content of organic mass in the fuel, the heat of combustion of a unit of its weight is increased. The simplest means of enrichment is the sorting out of gangue.

Briquetting is usually used as a method of imparting a defined form and size to the fuel mass containing a great number of fines in natural form. In the first place they briquette wood fines, peat, brown coal, and also coal and anthracite. If the coal contains a sufficient quantity of bitumen and has a moisture content of 10-20%, then under high pressure (up to 1000 [atm(abs.)] it can be turned into a solid briquette without additional binding elements. Frequently in briquetting, binding elements are added, for example tar pitch, resin and others in quantity up to 5-7%. The process is conducted at raised temperatures and under a pressure of 100-300 [atm(abs.)]. Briquetting makes it possible to increase the heat of combustion of a unit of weight of fuel both because of the additives and also pre-drying, and furthermore, it improves the technological utilization of fuel which is uniform in shape and weight.

Crushing of fuel - this is the grinding to a dustlike state of coal fines and run-of-the mine fuel. Powered fuel finds greatest use in GTD. The latter is explained by the fact, that, as D. I. Mendeleyev pointed out, in this case, the conditions of combustion of solid fuel approximate the conditions of combustion of a gas fuel. With crushing, the surface of the fuel reacting with an oxidizer is increased, which facilitates intensification of the process of combustion. Therefore, the grinding of the coal must be as fine as possible. As a rule, in grinding a fuel is obtained with particle sizes from 0 to 300 μm, but particles with a size of from 30 to 70 μm predominate. Crushing is carried out in two stages: preliminary, after pre-drying; it is produced in roller or hammer crushers and produces pieces 10-30 mm, and final - in a ball drum mill, where they produce a powdered fuel. The fineness of grinding is determined by consecutive sifting of the dust through a series of sieves. The part of the sample remaining after sifting on a sieve is called residue *R*. This value in percent of the weight of the basic sample is grinding characteristic. Sieves are graded by numbers: 30, 50, 70, 80, 100. The number of a sieve indicates how many apertures there must be per 1 cm. A residue value, for example, R_{70} = 30 indicates that 30% of the sample does not pass through the apertures of a No. 70 sieve, i.e., these particles have a size greater than 86 μm.

Most frequently, for example, from the brown coals there is obtained dust R_{70} = 30-60%; from coal rich in volatile substances, R_{70} = 10-30%, from anthracites and lean coals R_{70} = 5-15%.

Physico-Chemical Methods of Processing Solid Fuel

The basis of all methods of physico-chemical processing of solid fuel is the process of distillation (dry distillation) which is carried out by heating the fuel without air access.

Charcoal burning - the process of dry distillation of wood, which is used to produce charcoal.

Coking of coals is based on the fact that coal in heating suffers substantial changes. Certain grades of coal undergo coking; it is performed according to the following scheme. After heating through and a little drying, a number of complex carbon compounds begin to break down. At a temperature of more than 200°C primary tars and gas are liberated. Passing through a bed of glowing coke, the tar vapors are further gasified. At a temperature of 400-500°C, the coal passes over into a plastic state which is caused by the melting of bitumens and the dissolution in them of the remaining solids. With a further increase in the temperature to 1000-1200°C, the plastic mass of coal again passes over into a solid state, and coke is obtained, which during the repeated heating could not be converted to a plastic state. Metallurgy is the basic user of coke.

Semicoking is used most frequently to produce liquid fuel from coals. Semicoking is performed at temperatures of 500-550°C; in so doing, semicoke, tar, gas, and tar water are produced, i.e., the so-called primary products, which do not undergo to the profound processes of thermal decomposition which accompany the coking process. The solid residue - semicoke, being a mechanically fragile product, has up to 10-20% volatile substances. Considerably more tar is obtained in semicoking than in coking (Table 9). The gas which is obtained in semicoking has a high heat of combustion (up to 8000-9000 kcal/m^3 or 33,500-37,700 kJ/m^3), but its yield is less than in coking.

Table 9. Approximate composition of the products of coking and semicoking of coals.

Process	Solid residue in %	Gas in m^3/t	Anhydrous tar in %
Coking at 1000-1100°C........	78—80	300—340	2,5—3,0
Semicoking at 450-550°C......	75—77	60—70	7,0—10,0

Solid fuels giving a large yield of primary tar and gas undergo semicoking, peat, shales, brown coals, and bituminous coal. Semicoke can be used as a power, industrial, and everyday fuel.

In comparison with other types, solid fuel is more difficult to use in GTD.

In principle this can be done by the following means:

1) burning directly, for example, coal in the combustion chambers of open cycle GTD;

2) burning solid fuel in GTD with an air boiler or in a closed cycle GTD;

3) preliminarily gasifying the solid fuel.

The last two methods are less widely used because of the difficulty in controlling the working process and also because of the bulkiness and response time of the installations.

Most of the research was carried out in the area of direct combustion of solid fuel in GTD combustion chambers in the form of powdered coal. Powder of bituminous coal, rich in volatile substances, and possessing rather high heat of combustion have been burned.

In direct combustion of solid fuel, it is necessary to cleanse the gas on outlet from the chamber from solid particle of ash and slag in order to avoid damage and wear of the blading of the gas turbine.

Tests have shown, that to ensure reliable operation of GTD operating on solid fuel, the gas before the turbine should not contain solid particles larger than 15-25 μm.

CHAPTER III

FUEL COMBUSTION

§ 7. Conditions Necessary for Fuel Combustion

Products of Combustion

To successfully conduct the process of combustion it is necessary to create specific conditions. In practice, the process of combustion is usually carried out in atmospheric air, the oxygen of which at a sufficiently high temperature combines with the carbon, hydrogen, and sulfur of the fuel, forming products of combustion (carbon dioxide, water vapor and SO_2). It is customary to assume that air contains by volume 21% O_2 and 79% N_2 (including argon and a number of other insignificant impurities) or by weight 23.2% O_2 and 76.8% N_2 respectively. It is natural that in the products of combustion nitrogen is present. If combustion takes place in an excess of air, then the products of combustion contain a defined quantity of oxygen.

In this way, the fuel and the oxidizer (air) comprise a mixture capable of entering into chemical reaction, while the temperature conditions, together with a number of other factors (the conditions of mixing, the tapping of the products of the reaction and heat, etc.) determine the possibility of ignition and the character of combustion of the mixture. The change in the composition of the mixture or, for example, the temperature conditions changes the rage of combustion, the composition of the products of combustion, and it can even lead to stopping the process.

Of the air components, only oxygen takes part in the process of combustion. It is natural that by changing the concentration of O_2 in the air it is possible to change the rate of a process. In pure oxygen greater rates of combustion are observed, than in air, and with a decrease in the content of oxygen in the air below 12-15%, combustion can entirely cease.

It is possible to theoretically determine the quantity of air necessary for complete oxidation of all combustible elements of a unit of weight (mass) of fuel in the following manner. If the fuel contains such combustible elements as C^p, H^p, S^p, then the reactions of complete oxidation of these elements with indication of the weight quantities of substances being consumed can be written thus:

$$\begin{array}{lll} C + O_2 = CO_2; & 2H_2 + O_2 = 2H_2O; & S + O_2 = SO_2; \\ 12 + 32 = 44; & 4 + 32 = 36; & 32 + 32 = 64, \end{array}$$

i.e., to oxidize 12 kg of C and to convert it into CO_2 requires 32 kg of O_2, and that means that for combustion of 0.01 kg of C (one weight percent) of oxygen, 0.01·32/12 kg will be required. Similarly for combustion of one percent of H_2 (by weight) it is necessary to have 0.01·32/4 kg of O_2. For sulfur, 0.01·32/32 kg of O_2 will be required. It is natural that if oxygen is contained in the fuel itself, then for combustion there is that much less oxygen of the air consumed which there is in the fuel itself. Therefore for combustion of 1 kg of fuel, the oxygen required will be:

$$L_{O_2} = \frac{0{,}01 \cdot 8}{3} C^p + 8 \cdot 0{,}01 H^p + 0{,}01 S^p - 0{,}01 O^p \text{ kg/kg}.$$

Since there is nitrogen in the air, as well as oxygen, then the weight quantity of air necessary for oxidation of 1 kg of fuel, the so-called stoichiometric quantity of air,

$$L_0 = \left(1 + \frac{76{,}8}{23{,}2}\right)\left(\frac{8 \cdot 0{,}01}{3} C^p + 8 \cdot 0{,}01 H^p + 0{,}01 S^p - 0{,}01 O^p\right) \text{ kg/kg}$$

or

$$L_0 = 0{,}1149\mathrm{C}^p + 0{,}3448\mathrm{H}^p + 0{,}0431\mathrm{S}^p - 0{,}0431\mathrm{O}^p \text{ kg/kg.} \qquad (5)$$

If this quantity of air must be expressed in volume units of V_0, then it must be divided by the weight (mass) of 1 m^3 air. At a temperature of 0°C and a pressure of 760 mm Hg, the density of air equals 1.293 kg/m^3. Then

$$V_0 = 0{,}265\left(\frac{\mathrm{C}^p}{3} + \mathrm{H}^p + \frac{\mathrm{S}^p}{8} - \frac{\mathrm{O}^p}{8}\right) \text{ m}^3\text{/kg.}$$

Usually for the combustion of 1 kg of fuel supply a greater quantity of air L is supplied than is theoretically required. A ratio of $L/L_0 = \alpha$ is called the excess air ratio. If complete oxidation of all combustible elements of the fuel takes place during combustion in the theoretically necessary quantity of air $\alpha = 1.0$ and heat losses are absent, then there will be obtained the maximum temperature of products of combustion possible in burning the given fuel. The temperature of combustion, generally speaking, can be changed due to a number of other conditions. For example, the temperature of combustion at constant pressure is distinguished from the temperature of a combustion with constant volume. However, the basic effect on temperature proves to be the conditions of heat emission, the ratio of oxidizer to fuel, and the degree of completeness of burning.

In determining the theoretical temperature of combustion, it is considered that all the heat being liberated Q is imparted to the products of combustion. The final temperature $t_к$ of the products of combustion will depend, naturally, upon the quantity of heat imparted to them, their quantity and their heat capacity. If combustion proceeds at constant pressure, then the value of $t_к$ can be determined from the equation

$$Q = jc_p(t_к - t_н),$$

where j - quantity of products of combustion which are formed during combustion of 1 kg of fuel; c_p - heat capacity of the products of a combustion with p = const; $t_н$ - initial temperature of the system (of the combustible mixture).

If calculations are produced for 1 kg of fuel, then we obtain the lower heat of combustion of fuel $Q^р_н$. If $t_н$ = 0°C, then the equation takes the form $Q = jc_p t_к$.

Inasmuch as the products of combustion are a mixture of various gases, the heat capacities of which vary, then

$$Q = j_1 c_{p_1} t_к + j_2 c_{p_2} t_к + \cdots + j_n c_{p_n} t_к,^*$$

where $j_п$ and $c_{p_п}$ - respectively, quantity and heat capacity of a given component in the products of combustion.

Table 10 gives some values of the theoretical temperatures of combustion of a number of combustible substances.

Table 10. Theoretical temperatures of combustion of some combustible substances.

Combustible substance	Heat of a combustion				Theoretical temperature of combustion in °C
	in kJ/kg	in kcal/kg	in kJ/m³	in kcal/m³	
Coal..................	30 500	7 300	—	—	2010
Kerosene.............	43 900	10 500	—	—	1925
Carbon monoxide......	—	—	12 700	3035	2030
Hydrogen.............	—	—	10 750	2570	1970

The actual temperature of combustion of the fuel is usually lower than the theoretical since combustion, as a rule, proceeds with an excess of air ($\alpha > 1.0$) with some incompleteness of burning, and part of the heat is lost into the surroundings, and furthermore, the part of the heat is not liberated because of dissociation (see below).

---

*[Translator's Note: In several instances the character "n" appears to have been used in the original as a Cyrillic "п." The final typed copy will reflect this in many sections of the document].

The process of a combustion can begin not at 0°C, but at a considerably higher temperature. Thus, air after compression in a compressor can have on inlet to the combustion chamber of a GTD, a temperature several hundred degrees. In this case, the final temperature of combustion will be higher than the theoretical obtained under a condition of $t_H = 0°C$.

§ 8. Heat of Combustion with Various Conditions of the Process of Combustion

Heat of chemical reaction Q is the name given to that quantity of heat which is liberated (or absorbed) in any reaction between substances, calculated on 1 mole of the basic substance. If the fuel is burned in a vessel of constant volume, then we obtain the thermal effect of the chemical reaction Q_v or heat of combustion of fuel Q_v^p with constant volume. Combustion of fuel with $p =$ const makes it possible to determine the heat of combustion at constant pressure Q_p^p.

The heat of combustion of one and the same fuel, obtained with $p =$ const or with $V =$ const, is identical only when with equal initial and final temperatures the quantity of the products of combustion remains equal to the quantity of the mixture prior to combustion. Otherwise, with the change in the volume of the products of combustion by the value ΔV the external pressure performs work $p \cdot \Delta V$, which is converted into an equivalent quantity of heat. The difference in the value of Q_p^p and Q_v^p in most cases is small, therefore it is usually disregarded.

For example, during combustion of 1 m^3 of CO under usual atmospheric air conditions, the difference in the values of Q_p^p and Q_v^p amounts to approximately 0.4%, despite the fact that the reduction in volume of 1 m^3 of CO is equal to 0.5 m^3.

An expression for computing $Q_p - Q_v$ can be obtained by using the equation of the first law of thermodynamics, in reference to the chemical processes being considered. Since the change in external kinetic energy $d\frac{w^2}{2g}$ and external potential energy dh is usally small in comparison with the change in interior energy of a body dU, then, disregarding these changes, we determine

$$Q = \Delta U + AL.$$

The equation is written for 1 kg-mole of a gas.

Let us consider that if in the expression of evolved heat Q rates a plus sign, this corresponds to an exothermal reaction (it proceeds with evolution of heat), and a minus sign -- of an endothermic reaction (it proceeds with absorption of heat). Furthermore, the quantity $+L$ mean that the work is completed, and $+\Delta U$ indicates that inner energy is reduced. Finally, we will express the work being accomplished in the presence of a chemical reaction, in thermal units and designate as letter A. Then $\Delta U = Q + A$. When the work attains minimum (A_{min}), then the greatest quantity of heat Q_{max} will be evolved, which is called the thermal effect of reaction, i.e., where

$$V = \text{const} \quad A_{min} = 0 \text{ and } (Q_v)_{max} = \Delta U;$$

where

$$p = \text{const} \quad \Delta U = (Q_p)_{max} + p(V_2 - V_1)$$

or

$$(Q_r)_{max} = U_1 - U_2 + pV_1 - pV_2 \quad \text{or} \quad (Q_i)_{max} = \Delta I,$$

where I - enthalpy (heat content) of the gas.

Volume during chemical reactions changes as a result of changes in the number of molecules (moles) of a substance. Using, just as for ideal gases, the characteristic equation, for the process p = const we obtain

$$p(V_2 - V_1) = (n_2 - n_1)RT = \Delta nRT,$$

where n_1 and n_2 - the number of moles of gaseous substances prior to after the reaction.

Considering the universal gas constant $R = 1.985 \approx 2$ kcal/mole or 8.38 kJ/mole, after conversion we obtain

$$(Q_v)_{max} - (Q_p)_{max} = 2\Delta nT \text{ kcal}$$

or

$$(Q_v)_{max} - (Q_p)_{max} = 8{,}38\Delta nT \text{ kJ.}$$

In this way, if the reaction proceeding with p = const is accompanied by expansion (by an increase in the number of moles), then Δn - a positive quantity and $(Q_v)_{max} > (Q_p)_{max}$.

Consequantly, for the accomplishment of the action of expansion heat is spent, which decreases the amount of heat of the chemical reaction.

If in the reaction, the volume is reduced, then $(Q_p)_{max} > (Q_v)_{max}$. In this instance to the heat of the reaction there is added heat equivalent to the work of compression.

Example 1. Combustion of CO with V = const, t = 20°C gives $(Q_v)_{max}$ = 68,000 kcal/mole or 285,000 kJ/mole. We determine $(Q_p)_{max}$. With the reaction $CO + \frac{1}{2} O_2 = CO_2$ $\Delta n = n_2 - n_1 = 1 - 1.5 = -0.5$. Consequently, $(Q_p)_{max} = 68{,}000 + 2 \cdot 0.5 \cdot 293 = 68{,}293$ kcal/mole (288,100 kJ/mole).

In this way, for 1 kg of basic fuel, the difference comprises a value of $\frac{293}{28} \approx 10.5$ kcal/kg (43.7 kJ/kg).

Example 2. With the reaction $C + O_2 = CO_2$ and t = 15°C $(Q_p)_{max}$ = = 94,000 kcal/mole. We determine $(Q_v)_{max}$. Here $\Delta n = 0$, since the volume of solid carbon can be disregarded. Then:

$$(Q_p)_{max} = (Q_v)_{max} = 94\,000 \text{ kcal/mole } (394\,000 \text{ kJ/mole}).$$

In view of the fact that the difference $(Q_p)_{max} - (Q_v)_{max}$ for the majority of practically important cases in practice is not great, the difference in values of heat of combustion of fuels in practice is not usually considered.

A more substantial effect on the magnitude of heat of combustion of a fuel proves to be the temperature. The heat of combustion can be determined, burning a fuel at any temperature, observing only the condition of equality of the initial temperature of the mixture and the final temperature of the products of combustion. In general, for various temperatures, the heat of reaction will be different. However, practically, the heat of combustions of a fuel is considered not to depend on temperature.

When necessary, this difference can be taken into account in the following manner. It is assumed that the quantity of energy $Q_{\text{хим}}$, which system has prior to reaction in combined state at any temperature is the same. The heat obtained by a calorimeter from the products of combustion of a fuel with their reduction to the initial temperature of the experiment, for example to t_1, which will define the quantity of heat of combustion of the fuel is not equal (more accurately, is not equivalent) to the chemical energy which was included in the mixture prior to reaction, since the physical heat of the products of reaction equals the physical heat of the basic substances (fuel + oxidizer). These quantities of heat can be calculated, knowing the temperature of the experiment t_i, the heat capacity of the basic substances c_i, and the quantity of these substances m_i.

Suppose that in the reaction of combustion of fuel A in an oxidizer B there is obtained a product of reaction C and there is liberated heat of chemical reaction Q according to the equation

$$\alpha A + \beta B = \gamma C + Q,$$

where α, β, γ - the number of moles of substances A, B and C.

The molecular heat capacities of these substances we designate as c_A, c_B, c_C. Then the value $Q' = [\gamma c_C - (\alpha c_A + \beta c_B)] t_1 = \sum m_1 c t_1$ will be the difference between the heat of chemical reaction Q and the released chemical energy $Q_{\text{хим}}$, i.e.,

$$Q_{\text{хим}} = Q_{t_1} + \sum m_1 c t_1.$$

For temperature t_2

$$Q_{\text{кон}} = Q_{t_2} + \sum m_1 c t_2.$$

Then $Q_{t_2} - Q_{t_1} = \sum m_1 c (t_2 - t_1)$

or

$$Q_{t_2} - Q_{t_1} = (\alpha c_A + \beta c_B - \gamma c_C)(t_2 - t_1) = \sum mc (t_2 - t_1).$$

Let us note that the heat capacities of the substances change with the change in temperature, and this expression holds true only for infinitesimal range of temperatures, i.e., for

$$dQ = \sum mc\,dt \text{ or } \frac{dQ}{dt} = \sum mc.$$

In this way, in determining the heat of combustion of a fuel at constant pressure and constant volume

$$Q_p = Q_v + \Delta nRT,$$

where

$$\Delta n = \alpha + \beta - \gamma.$$

If in the reaction the volume does not change, then Q_p will equal Q_v.

Consequently, if a chemical reaction proceeds at constant temperature, the themal effect of chemical reaction depends only on the initial and final state of the mixture and does not depend on the path, along which the process proceeds (as follows from Hess's law).

2. FOUNDATIONS OF THE THEORY OF COMBUSTION

All occurrences of combustion with the use of a gas oxidizer and, particularly, the oxygen of the air, are basically divided into three characteristic groups:

1) combustion of gas fuels, giving a system of gas + a gas - the so-called homogeneous system;

2) combustion of liquid or solid fuels, giving a system of fluid or solid + a gas - the so-called heterogeneous system;

3) combustion of condensed systems where the fuel can be found in any state of aggregation (explosive substances).

The homogenous part of the system, which is present in a heterogeneous system is called the homogeneous phase of a heterogeneous system.

If a fuel and an oxidizer are previously thoroughly mixed to a uniform state, then this system is customarily called a uniform gas system.

A heterogeneous system will be the case when a fuel and an oxidizer are supplied to the place of ignition and combustion separately.

The basis of the processes of combustion is the chemical conversion of substances, therefore we will briefly examine the most important chemical situations underlying the theory of combustion. This can be done most simply and conveniently with an example of a homogeneous, chemically uniform system.

CHAPTER IV

STATICS AND DYNAMICS OF CHEMICAL REACTIONS

The statics of chemical conversions consider questions connected with the material and energetic side of combustion reactions; specifically, the quantities of substances necessary for the complete course of a process, their relationships, the composition of the products of reaction and its thermal effect are determined.

The chemical reaction of the combining of two substances, for example, *A* and *B*, with the formation of products of the reaction, for example *M* and *N*, taking into account the thermal effect of reaction *Q*, can be described by the equation

$$\alpha A + \beta B \rightleftarrows \mu M + \eta N \pm Q,$$

where α, β, μ, η - the number of moles of substances *A*, *B*, *M*, *N*, taking part in the reaction.

The symbol $\rightleftarrows$ indicates reversibility of a reaction. A plus sign before *Q* indicates that with the combining of substances *A* and *B* heat is liberated (an exothermal reaction); and a minus sign means that heat is consumed (an endothermic reaction).

§ 9. Chemical Equilibrium

From the law of actual mass it is known that in a reacting mixture at given temperature, pressure, or volume, and with the course of both a direct reaction between the basic substances, for example *A* and *B*, with the formation of substances *M* and *N*, with a reverse reaction between the products of direct reaction of *M* and *N* with the production of the basic substances *A* and *B*, after a sufficient time equilibrium necessarily approaches between all reacting substances *A*, *B*, *M* and *N*, which is characterized by a defined relationship between the concentrations of these substances, i.e., by the number of molecules of these substances located in unit of volume.

Thus, if the instantaneous concentrations of substances *A*, *B*, *M*, *N* are respectively designated as *a*, *b*, *m*, *n*, then at the moment of equilibrium

$$\frac{a^{\alpha}b^{\beta}}{m^{\mu}n^{\nu}}=K, \tag{6}$$

where *K* - equilibrium constant of the mentioned chemical reaction, a value constant for a given temperature.

With a change in temperature the value of *K* changes, i.e., the equilibrium of the system is shifted to one side or the other. Vant Hoff proposed the following relationship connecting *K* with absolute temperature *T*:

$$\frac{d\ln K}{dT}=\frac{Q}{RT^{2}}, \tag{7}$$

where *R* - universal gas constant; *Q* - thermal effect of the chemical reaction.

When $Q > 0$ with the increase of temperature the value of the equilibrium constant increases, and when $Q < 0$ - is reduced. Let us note that the chemical equilibrium of reversible reactions is a transient dynamic equilibrium. When a reaction outwardly ceases, in actuality both reactions go on - direct and reverse, but at identical rates. The moment when chemical equilibrium of the given reversible reaction begins, depends, as already mentioned, on temperature, pressure, and concentration of all substances found in the system. If temperature and pressure are constant, then the state of equilibrium basically depends on the concentrations of the components of the system.

system strives toward a state of equilibrium under the given conditions. However, in a number of cases, the rate of the reverse reaction is so small in comparison to the rate of the direct reaction that it can be disregarded. Then, conventionally, it is possible to assume a unidirectional flow of reaction, and reactions of this kind can be considered irreversible.

It is customary to classify reactions depending on the number of reacting molecules:

1. *Monomolecular reactions*, in which only one type of molecules suffers conversion, giving one or several new molecules, whereby the sum of the stoichiometric factors in the left part of the reaction equation is equal to one.

A diagram of such a reaction is $A \to M + N + \ldots$ or conventionally $A = M + \ldots$

For realization of such a reaction it is necessary that a molecule of substance A arrive at a reaction-capable state. As an example of a monomolecular reaction we can use the decomposition of acetone:

$$CH_3COCH_3 = CO + C_2H_6,$$

2. *Bimolecular reactions*, in which two uniform or different molecules reciprocally react, giving new molecules, whereby the stoichiometric factor in the reaction equation is equal to two (i.e., the sum of stoichiometric factors is equal to two). A diagram of the reaction is $A + B = M + N + \ldots$

As an example of bimolecular reactions we can use:

1) the combination of hydrogen with iodine

$$H_2 + J_2 = 2HJ;$$

2) the decomposition of hydrogen iodide

$$2HJ = H_2 + J_2.$$

To this value of K they assign a subscript c in order to emphasize its expression as concentration, unlike the expression of a constant as the partial gas pressures in a mixture. In the latter case, the value of equilibrium constants are assigned a subscript p. The value of K_p can be obtained thus. For each gas reacting in a system it is possible to write an equation of state

$$p_iV = n_iRT,$$

where p_i - partial pressure of gas in the mixture; V - total volume of the system; n_i - number of moles of the given gas in the system; R - universal gas constant; T - absolute temperature at which the reaction proceeds.

Designating $\frac{n_i}{V} = c_i$ — the molar concentrations of a given gas in the system, we obtain $c_i = \frac{p_i}{RT}$. Substituting appropriate values in the expression for K_c, we have

$$K_c = \frac{\frac{p_A^\alpha}{(RT)^\alpha}\cdot\frac{p_B^\beta}{(RT)^\beta}}{\frac{p_M^\mu}{(RT)^\mu}\cdot\frac{p_N^\eta}{(RT)^\eta}},$$

or in the general case for the reaction

$$\alpha A + \beta B + \gamma C + \cdots \rightleftarrows \mu M + \eta N + \cdots;$$

$$K_c = \frac{p_A^\alpha p_B^\beta p_C^\gamma}{p_M^\mu p_N^\eta}(RT)^{\mu+\eta-\alpha-\beta\cdots}$$

If we designate the first factor on the right as K_p, then

$$K_c = K_p(RT)^{\Delta i},$$

where Δi - the algebraic sum of the numbers of moles of the gas substances which take part in the reaction. The value

$$\frac{p_A^\alpha p_B^\beta}{p_M^\mu p_N^\eta} = K_p \qquad (8)$$

is the equilibrium constant of the chemical reaction expressed as partial pressures.

It is obvious that even if there is no necessity for considering the distinction in the value of R, the numerical values of K_c and K_p for one and the same conditions generally varies. Only during the course of a reaction without change in the number of moles, i.e., if $\Delta i = 0$, $K_c = K_p$. Let us note that the application of the mass action law and the expressions obtained on its base, strictly speaking, is limited to ideal gases; however the error in extending this law to real gases is small.

Let us note that the given relationships are valid even for heterogeneous systems. It is known that any liquid or solid has a defined vapor pressure at a given temperature, i.e., with T = const, the partial pressure of a saturated vapor of such a substance in the gas phase of a heterogeneous system is a constant value. In this way, a chemical reaction which takes place in the gas phase between a saturated vapor of a solid or liquid substance and gases, coming to equilibrium, will conform to all the laws of homogenous reactions. Consequently, by applying the mass action law to the homogenous phase a heterogeneous system it is possible not to take into account the partial pressure of the vapors of a solid or liquid substance, since it remains constant.

For example, for the reaction of the decomposition of calcium carbonate

$$CaCO_3 \rightleftarrows CaO + CO_2$$

according to the mass action law, the equilibrium constant for a given temperature

$$K'_p = \frac{p_{CaCO_3}}{p_{CaO}p_{CO_2}} = \text{const.}$$

Considering that in this case p_{CaCO_3} and p_{CaO} — the constant values, and substituting them in the numerical values of K_p, it is possible to write

$$K_p = \frac{1}{p_{CO_2}} = \text{const.}$$

Similarly

$$K_c = \frac{1}{c_{CO_2}} = \text{const.}$$

where c_{CO_2} — the equilibrium concentration of carbon dioxide.

Dissociation of Products of Combustion

The value of *K* determines the composition of an equilibrium mixture at a given temperature. Knowing the composition of a mixture, for the given conditions it is possible to calculate the process of dissociation, i.e., the process of decomposition of the products of a direct reaction to the initial elements. Usually dissociation is characterized by the degree of dissociation α, under which is understood the fraction of the moles of the substance which was decomposed to basic components.

We shall examine this process and its dependence on certain parameters in the example of the reaction $2CO + O_2 \rightleftarrows 2CO_2$ in more detail. Let us assume that the number of moles of CO_2 prior to dissociation was equivalent to two, as this results from the equation of chemical reaction. After dissociation, at a specified temperature in a state of equilibrium, the mixture of gases will have, besides CO_2, also CO and O_2. If in this case, the degree of dissociation is equal to α, then this means that from each mole of CO_2 (being used as the unit) fraction α dissociates, and from two moles, 2 α moles dissociate, giving respectively 2 α moles of CO and α moles of O_2. The undecomposed quantity of CO_2 will be equal to $2\cdot(1 - \alpha)$. Consequently, at equlibrium (after dissociation) the mixture will consist of:

$CO_2 = 2\cdot(1-\alpha)$ moles
$CO = 2\alpha$ moles
$O_2 = \alpha$ moles

Total . . . $2+\alpha$ moles

If we designate the total volume at the given temperature *T* as *V*, then the partial pressures of individual gases in the mixture will be

$$p_{CO_2} = \frac{2(1-\alpha)}{V} RT;\quad p_{CO} = \frac{2\alpha}{V} RT;$$

$$p_{O_2} = \frac{\alpha}{V} RT.$$

The equilibrium constant will equal

$$K_p = \frac{\alpha^3}{V(1-\alpha)^2} RT.$$

Using the general equation of state $pV = (2 + \alpha) RT$, it is possible to obtain

$$K_p = \frac{\alpha^3 p}{(2+\alpha)(1-\alpha)^2}. \qquad (9)$$

Equation (9) indicates that with T = const, with an increase in the pressure, the degree of dissociation of carbon dioxide is reduced. Knowing α for the given temperature and pressure, from equation (9) it is possible to calculate K_p and α at the same temperature for any other pressure.

Raising the temperature intensifies the decomposition of CO_2. At atmospheric pressure the dissociation of CO_2 becomes apparent at temperatures of 1500-2000°C and beyond that it sharply increases. If at 1500°C α is equal to approximately 0.8%, then at 2000°C it already equals 4%, at 2500°C - 19%, and at 3500°C - 40%. If the partial pressure of carbon dioxide in the products of combustion is small (let us say 0.1-0.2 [atm(abs.)]), then the degree of dissociation in this instances increases sharply, comprising at 1500°C already approximately 1.7%, and at 2000° - about 8%.

A similar correlation can be obtained also for the dissociation of water vapor to H_2 and O_2. Thus, at atmospheric pressure and a temperature of about 1500-1800°C, the degree of dissociation of $H_2O_{пар}$ comprises approximately 0.5%. Let us note that the reaction of formation of H_2O just as the reaction forming CO_2, is accompanied by a decrease in the number of moles. Therefore with the formation of H_2O, the effect of pressure on the degree of dissociation at invariable temperature, and consequently, also K_p is the same.

It is possible to show that the case of equilibrium of the reactions proceeding with an invariable number of moles (Δn = 0), with T = const does not depend on the change in pressure (or volume).

Effect of Dissociation on Temperature of Combustion

In the combustion chambers of Gas Turbine Engine [GTD] (ГТД), the process of dissociation is a negative occurrence, since it leads to the incomplete evolution of heat with the combustion of the fuel. In order to eliminate the incomplete evolution of heat, the conditions are controlled, under which the process of a combustion is carried out, specifically, by limiting the temperature. In formulating the equations of heat balance for determination of the temperature of combustion under actual conditions, it is necessary to take account of the effect of the process of dissociation. In general form, such an equation can be written in the following manner:

$$Q_{см} + Q = Q_{п.с} + Q_{ок} + Q_{д},$$

where $Q_{см}$ - enthalpy of the original combustible mixture (fuel and oxidizer); Q - thermal effect; $Q_{п.с}$ - enthalpy of the products of combustion; $Q_{ок}$ - heat lost to the environment; $Q_{д}$ - heat spent on the process of dissociation.

If $Q_{п.с} = \sum nct_г$ (where $\sum nc$ — the sum of the products of the number of moles of the individual products of a combustion times their molecular heat capacities), then the temperature at the end of the process of combustion can be computed using the equation

$$t_г = \frac{Q_{см} + Q - Q_{ок} - Q_{д}}{\sum nc}. \qquad (10)$$

In order to solve this equation, having two unknowns α and $t_г$, it is necessary to have a second equation which links these quantities. Such an equation is the dependence of the equilibrium constant upon α for T = const. Solving together equations (9) and (10), it is possible given various values of α from equation (10) to determine values of $T_г$ and to construct a graph of dependence of $t_г$ on α. Further, given values of temperatures, again from the equilibrium equation determine α and construct a second curve of dependence of $t_г$ on α. The point of intersection of both curves gives the values of the sought values of $t_г$ and α.

§ 10. Rate of Chemical Reaction

The rate of a homogeneous reaction is determined by the quantity of the substance reacting in a unit of volume for a unit of time. It is obvious that it is equal to the change in the concentration of one of the components of reacting system in a unit of time.

In any reaction some substances take part, whereby their concentrations change in the course of time. The concentrations of the original substances are reducted, and those of the products of reaction - increase. However, between them stoichiometric conformity is always carried out. Taking this into account, the rate of a chemical reaction $\alpha A + \beta B = \mu M + \eta N$ can be expressed either as an end product, as $+\frac{dm}{d\tau}$ or $+\frac{dn}{d\tau}$, or as the rate of consumption of one of the original substances $-\frac{da}{d\tau}$ or $-\frac{db}{d\tau}$. Here a, b, m, n - instantaneous concentrations of reacting substances, and τ - time.

Let di_A moles of substance A react for time $d\tau$. According to the reaction equation, for this there will be expended time $di_B = \frac{\beta}{\alpha} di_A$ moles of substance B. Referring the last expression to a unit of volume and time, we obtain

$$\frac{di_B}{V}\cdot\frac{1}{d\tau} = \frac{\beta}{\alpha}\cdot\frac{di_A}{V}\cdot\frac{1}{d\tau}$$

or $\frac{db}{d\tau} = \frac{\beta}{\alpha}\cdot\frac{da}{d\tau}$. Similarly, it is possible to obtain

$$\frac{dm}{d\tau} = -\frac{\mu}{\alpha}\cdot\frac{da}{d\tau} = \frac{\mu}{\eta}\cdot\frac{dn}{d\tau}$$

or

$$-\frac{1}{\alpha}\cdot\frac{da}{d\tau} = -\frac{1}{\beta}\cdot\frac{db}{d\tau} = \frac{1}{\mu}\cdot\frac{dm}{d\tau} = \frac{1}{\eta}\cdot\frac{dn}{d\tau}.$$

Subsequently the rate of the chemical reaction w will be determined by the value $\frac{dm}{d\tau}$.

§ 11. Classification

When any chemical reaction has begun, then at any moment of time two reactions are going on - direct and reverse, and the system strives toward a state of equilibrium under the given conditions. However, in a number of cases, the rate of the reverse reaction is so small in comparison to the rate of the direct reaction that it can be disregarded. Then, conventionally, it is possible to assume a unidirectional flow of reaction, and reactions of this kind can be considered irreversible.

It is customary to classify reactions depending on the number of reacting molecules:

1. *Monomolecular reactions,* in which only one type of molecules suffers conversion, giving one or several new molecules, whereby the sum of the stoichiometric factors in the left part of the reaction equation is equal to one.

A diagram of such a reaction is $A \to M + N + \ldots$ or conventionally $A = M + \ldots$

For realization of such a reaction it is necessary that a molecule of substance A arrive at a reaction-capable state. As an example of a monomolecular reaction we can use the decomposition of acetone:

$$CH_3COCH_3 = CO + C_2H_6,$$

2. *Bimolecular reactions,* in which two uniform or different molecules reciprocally react, giving new molecules, whereby the stoichiometric factor in the reaction equation is equal to two (i.e., the sum of stoichiometric factors is equal to two). A diagram of the reaction is $A + B = M + N + \ldots$

As an example of bimolecular reactions we can use:

1) the combination of hydrogen with iodine

$$H_2 + J_2 = 2HJ;$$

2) the decomposition of hydrogen iodide

$$2HJ = H_2 + J_2;$$

3) the decomposition of chlorine monoxide

$$2Cl_2O = 2Cl_2 + O_2 \text{ etc.}$$

3. *Trimolecular reactions,* in which any three molecules react, giving new ones, and the stoichiometric factor of the reaction equals three. A diagram of the reaction is: $A + B + C = M + N + \ldots$ An example of a trimolecular reaction will be the combustion of carbon monoxide [sic] $2NO + O_2 = 2NO_2$.

The last two types of reactions can be carried out only with the simultaneous collision of the appropriate number of reacting molecules. It is possible to speak formally about reactions in which more than three molecules take part. However, the simultaneous collision four or more molecules, in accordance with kinetic concepts, is unlikely. It is customary to consider, therefore, that in this instance the reaction proceeds consecutively through simple bi- or trimolecular reactions.

A reaction, the mechanism of which includes more than one elementary (mono-, bi- or trimolecular) reaction, is customarily called complex.

All unidirectional reactions, furthermore, are customarily distinguished according to their order, i.e., according to the total number of molecules entering into combination. It is natural that for elementary reactions their order characterizes the mechanism of the reaction.

In this way, a reaction which proceeds according to the diagram $A = M + N + \ldots$, will be a monomolecular reaction and, furthermore, this is a reaction of the first order; in the case of $A + B = M + N + \ldots$ we have a bimolecular reaction, or a reaction of the second order, etc.

The accepted classifications permits simplifying the form of writing of the reaction equations and, correspondingly, the calculation equations. For example, proceeding from the mass action law, the expressions for the rates of chemical reactions will be written thus:

1) for a monomolecular reaction

$$\frac{dm}{d\tau} = k \cdot a \frac{1}{\text{s} \cdot \text{cm}^3}$$

or

$$\frac{dn}{d\tau} = k' \cdot a \frac{1}{\text{s} \cdot \text{cm}^3}.$$

The dimension of the rate constant of a chemical reaction k $\frac{1}{\text{s}}$. Numerically k and k', naturally, can be different;

2) for a bimolecular reaction

$$\frac{dm}{d\tau} = k \cdot a \cdot b \frac{1}{\text{s} \cdot \text{cm}^3}.$$

Dimension of k is cm^3/s;

3) for a trimolecular reaction

$$\frac{dm}{d\tau} = k \cdot a \cdot b \cdot c \frac{1}{\text{s} \cdot \text{cm}^3}.$$

Dimension of k is cm^6/s.

The concentrations of components a, b, c are taken as molar and designate the number of moles of a substance in a unit of volume (1/cm^3). The value of the chemical reaction rate is determined by molecular or molar concentrations. However, in certain cases it is more suitable, for example, to use weight or relative concentrations.

§ 12. Change in Rate of a Chemical Reaction According to Time

From kinetic equations it is possible to determine the rate of reaction and the composition of the reacting medium at any moment of time, since the changes in the concentrations of all substances are connected with one another. To obtain dependence $w = f(\tau)$

it is convenient to introduce into the kinetic equation the concentration of one of the substances, and to replace the rest by the values of their initial values and their value of selected variable concentration.

For a monomolecular reaction $A = M + \ldots$, taking in as the basic concentration of the product of the reaction m, which at the initial moment of time is equal to zero, we can write for any time that

$$\frac{dm}{d\tau} = k\cdot a = k(a_0 - m),$$

where a_0 - initial concentration of decomposing molecules of substance A; a - concentration of a substance at the present instant of time.

Integrating the equation

$$\frac{dm}{a_0 - m} = k\cdot d\tau,$$

we obtain

$$\ln(a_0 - m) = -k\tau + \ln C.$$

Since with $\tau = 0$, $m = 0$, then

$$\ln C = \ln a_0 \text{ and } C = a_0$$

and then

$$\ln\frac{a_0 - m}{a_0} = -k\tau,$$

whence

$$m = a_0(1 - e^{-k\tau}).$$

The character of function $m = f(\tau)$ is shown in Fig. 7 (curve 1). Similarly, it is possible to obtain expressions for a bimolecular (curve 2) and a trimolecular (curve 3) reaction.

Figure 8 gives the character of dependence $\frac{dm}{d\tau} = \phi(\tau)$ for these reactions. The dependences show that the yield of the products of

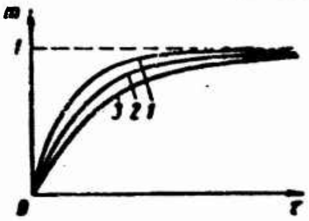


Fig. 7.

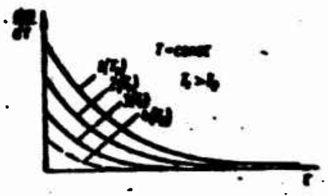


Fig. 8.

Fig. 7. Dependence of concentration of products of reaction on time.

Fig. 8. Dependence of the rate of a chemical reaction on time: 1 and 4 - monomolecular reactions; 2 - bimolecular reactions; 3 - trimolecular reactions.

a reaction in time is slowed down and decreases with the increase in the order of the reaction. The continuous drop in the rate of a reaction is explained by the reduction in concentration of the basic substances.

§ 13. Dependence of the Rate of a Chemical Reaction on Temperature

Using the dependence of the equilibrium constants of a chemical reaction on temperature, we have

$$\frac{d}{dT}\ln K=\frac{Q}{RT^2},$$

where

$$K=\frac{\overrightarrow{k}}{\overleftarrow{k}}.$$

We consider that Q - this is the difference between the two values E_2 and E_1, each of which is connected with the reaction rates constant of a reverse $\overleftarrow{k}$ and the rate constant of a direct reaction $\overrightarrow{k}$ in the same manner as for the equilibrium constant K with Q. Arrhenius, analysing experimental data, showed that for the rate constants of chemical reactions this relationship is valid

$$\frac{d}{dT}\ln k=\frac{E}{RT^2} \text{ or } \ln k=-\frac{E}{RT}+\text{const};$$

otherwise

$$k=\text{const}\cdot e^{-\frac{E}{RT}} \text{ or } k=k_0\cdot e^{-\frac{E}{RT}}.$$

As is known, the total number of collisions of molecules in a gas increases insignificantly with the increase in temperature. According to Arrhenius' law the conditions of equilibrium are determined by the frequency of the collisions only of the special (active) molecules capable of reacting, and not on the overall number of collisions of the considerably greater number of normal inactive molecules which at the given moment and under the given conditions are not able to enter into the reaction.

The active molecules differ from the normal ones in high energy level. Let us designate the thermal effect of the formations of active molecules $A_{\text{а}}$ from any molecule of substance A as E. The process of the change in the energy of a molecule without chemical conversion, proceeding according to the diagram $A_{\text{а}} = A_{\text{п}} + E$, is called activation. Energy E, absorbed by normal molecule $A_{\text{п}}$ with the formation of active molecule $A_{\text{а}}$, is called the heat (energy) of activation. According to the mass action law, the equilibrium constant of the process of activation

$$K_{\text{акт}} = \frac{a_{\text{а}}}{a_{\text{п}}} \text{ and } \frac{d}{dT} \ln K_{\text{акт}} = \frac{E}{RT^2},$$

where $a_{\text{а}}$ - concentration of active molecules; $a_{\text{п}}$ - concentration of passive molecules.

In this way

$$K_{\text{акт}} = \text{const} \cdot e^{-\frac{E}{RT}}.$$

At low temperatures, the quantity and concentration of active molecules are very small, therefore the concentration of normal molecules $a_{\text{п}}$ practically is very close to the general concentration a of the molecules of substance A. However, at infinitely high temperature, all molecules will be active and in the last equation const = 1. Since $a_{\text{а}} = K_{\text{акт}} a_{\text{п}}$, then $a_{\text{а}} \approx K_{\text{акт}} a = a \cdot e^{-\frac{E}{RT}}$, where the quantity $e^{-\frac{E}{RT}}$ indicates what portion of the total number of molecules is capable of reacting (active).

In this way, k_0 characterizes the total number of collisions of molecules, and quantity k - the number of active collisions. The majority of reactions obey Arrhenius' law. This permits calculating the values of k_0 and E, determining k and T by experimental means.

Taking the logarithm of the expression for k, we obtain

$$\ln k = \ln k_0 - \frac{E}{R} \cdot \frac{1}{T}.$$

This dependence in coordinates ln k and $\frac{1}{T}$ gives a straight line (Fig. 9) with an angular coefficient $\frac{E}{R}$. A segment on the axis ln k, intercepting the straight line is equal to ln k_0.

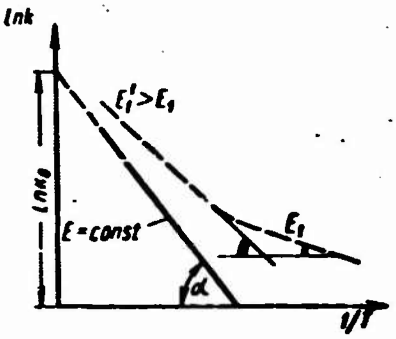


Fig. 9. Dependence of the rate constant of a chemical reaction on temperature.

If the experimental data are averaged not by a straight line, but by a broken line (dashed line in Fig. 9), then this means that for the given reaction there are a number of temperature ranges, in which quantities k_0 and E vary (E_1 and E'_1). Usually this is explained by the difference in the mechanism of the reaction or by the change in the state aggregation of the reacting substances at various temperatures.

In this way, each collision of molecules leads to a chemical reaction. A reaction proceeds only in the case when there is a collision of the active molecules which possess energy greater than E. In the majority of cases, the energy of the activation is greater than the mean total energy of the colliding molecules, and such collisions do not lead to a chemical reaction. The diagram of the process can be explained by the graph shown in Fig. 10. Suppose that the molecules of the basic substances in initial state I have a

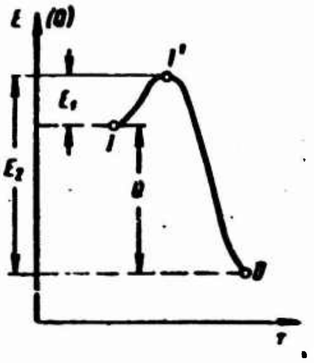


Fig. 10. Energy of activation of chemical reactions.

higher energy level than the products of reaction (state II). However, the change of the system from condition I to condition II requires a preliminary expenditure of energy, as a result of which the molecules of the basic substance are activated, passing over into reaction-capable state I'. The energy of activation, in particular, is expended on the destruction of old chemical bonds and their redistribution. Further, the system reacting returns energy E_1 and liberates heat $Q = E_2 - E_1$.

If quantity $E_1 > E_2$, then the reaction will be endothermic, similar to the process of the reverse change of the examined system from condition II to condition I. For this change, the energy of activation is equal to E_2. The amount of energy of activation is one of the chief factors which determine the rate of a chemical reaction, where the less the energy of activation, the faster the reaction proceeds.

§ 14. Dependence of Rate of a Chemical Reaction Upon Pressure

At an invariable temperature and with a given composition of a gas mixture, the equation of state gives the connection between partial pressures and molar concentrations of reacting substances:

$$p_A = aRT,\ p_B = bRT, \ldots, p_M = mRT, \ldots$$

If the total number of moles in unit of a volume is equal to z, then the full pressure in the gas system $p = zRT$. From these equations $RT = \frac{p}{z} = \frac{p_A}{a} = \frac{p_B}{b} = \cdots$, or $p_A = \frac{a}{z} p = \bar{a}p$; $p_B = \frac{b}{z} p = \bar{b}p$, etc. The ratios $\frac{a}{z} = \bar{a}$; $\frac{b}{z} = \bar{b}, \ldots$ are relative volume concentrations of substances A, B, ... In this way, the molar concentrations of the substances are equal to

$$a=\frac{p_A}{RT}=\bar{a}\frac{p}{RT};\quad b=\frac{p_B}{RT}=\bar{b}\frac{p}{RT};\ \ldots$$

Using the obtained expressions, it is possible to write the following equations for determining the rates of chemical reactions:

for monomolecular

$$\frac{dm}{d\tau}=k\cdot a=k\frac{p_A}{RT}=k\cdot\bar{a}\frac{p}{RT}=k\frac{\bar{a}}{RT}p\frac{1}{s\cdot cm^3};$$

for bimolecular

$$\frac{dm}{d\tau}=k\cdot a\cdot b=k\frac{p_A}{RT}\cdot\frac{p_B}{RT}=k\frac{\bar{a}\cdot\bar{b}}{(RT)^2}p^2\frac{1}{s\cdot cm^3};$$

for trimolecular

$$\frac{dm}{d\tau}=k\cdot a\cdot b\cdot c=k\frac{p_A}{RT}\cdot\frac{p_B}{RT}\cdot\frac{p_C}{RT}=k\frac{\bar{a}\cdot\bar{b}\cdot\bar{c}}{(RT)^3}p^3\frac{1}{s\cdot cm^3}.$$

Consequently, for reactions of the i-th order ($i \leq 3$)

$$\frac{dm}{d\tau}\sim p^i.$$

In this way, according to the type of dependence of the rate of reaction on pressure, it is possible to judge the order of a reaction. However, in practice, the order of a reaction does not always correspond to the value resulting from the type of chemical equation; sometimes experiment gives even a fractional value of the index of degree of dependence $\frac{dm}{d\tau}$ on p. This is explained by the fact that the course of the process proceeds through a number of intermediate compounds.

Turning to the general case, we will examine the dependence of rate of a reaction upon pressure, using relative concentrations of the reacting substances.

For a monomolecular reaction (the reaction of the first order)

$$\frac{d\left(\frac{m}{z}\right)}{d\tau}=\frac{d\bar{m}}{d\tau}=k\frac{a}{z}=k\cdot\bar{a}$$

or

$$\frac{d\bar{m}}{d\tau} = k \cdot \bar{a} \ 1/s$$

for a bimolecular reaction (a reaction of the second order)

$$\frac{d\bar{m}}{d\tau} = k \cdot \bar{a} \cdot b \frac{z}{z} \text{ and since } \quad z = \frac{p}{RT},$$

$$\text{then } \frac{d\bar{m}}{d\tau} = k \frac{\bar{a}\bar{b}}{RT} p \ 1/s$$

for a trimolecular reaction (a reaction of the third order)

$$\frac{d\bar{m}}{d\tau} = k \frac{\bar{a} \cdot \bar{b} \cdot \bar{c}}{(RT)^2} p^2 \ 1/s.$$

Consequently, in this instance, the general equation has the form

$$\frac{d\bar{m}}{d\tau} \sim p^{(\nu-1)}. \qquad (11)$$

It is necessary to turn attention to the fact that for monomolecular reactions the reaction rate of chemical reaction *k*, regardless of the units of measurement of concentrations always has the dimension 1/s.

§ 15. Dependence of the Rate of Chemical Reaction on the Composition of the Mixture

Since the total number of molecules of reacting substances at a specified temperature and with invariable pressure is constant, then the relative concentrations of all substances are connected with one another. For a bimolecular reaction, if the initial gas mixture contains two different substances, for example, a fuel *A* and an oxidizer (oxygen) *B*, $\bar{a} + \bar{b} = 1$.

Into the following moment of time, part of the initial substances will react, forming products of the reaction, for example, *M* and *N*. For this moment of time, $\bar{a} + \bar{b} + \bar{m} + \bar{n} = 1$. It is obvious that a portion of each substance will be changed in connection with the change in content of any substance, reacting or simply found in the mixture (inert).

We will determine how the initial rate of a chemical reaction changes in connection with the change in the composition of a component of the mixture. The value of the rate of reaction in general

$$\frac{dm}{d\tau} = k\left(\frac{p}{RT}\right)^2 \bar{a}\cdot\bar{b}\,\frac{1}{s\cdot cm^3};$$

since $\bar{a} + \bar{b} = 1$, then $\bar{b} = 1 - \bar{a}$ and

$$\frac{dm}{d\tau} = k\left(\frac{p}{RT}\right)^2 \bar{a}(1 - \bar{a}).$$

Consequently, the dependence of rate of a reaction on the composition of the mixture is determined by the product of $\bar{a}(1 - \bar{a})$. The character of the change $\frac{dm}{d\tau} \sim \bar{a}(1 - \bar{a})$ with a change of $\bar{a}$ from 0 to 1 is shown in Fig. 11 (curve 1). The rate of a chemical reaction reaches a maximum when $\bar{a} = \bar{b} = 0.5$.

If in the mixture, for example in the composition of an oxidizer, besides oxygen, there is still a third, inert gas (for example, nitrogen in air), then in principle the same conclusions can be made, but the value of the rate of the chemical reaction will be different. Let us consider each unit of substance B (air) consisting of two parts: the part of the oxidizer (oxygen) proper l, and the part of the inert admixture i (for example, nitrogen). Since $l + i = 1$, naturally, $\bar{a} + \bar{b} \times (l + i) = 1$.

The rate of the chemical reaction in this instance will be determined by the multiplier $\bar{a}\cdot l(1 - \bar{a})$ in the expression

$$w = \frac{dm}{d\tau} = k\left(\frac{p}{RT}\right)^2 \bar{a}\cdot l\cdot\bar{b} =$$
$$= k\left(\frac{p}{RT}\right)^2 \bar{a}\cdot l\,(1 - \bar{a}).$$

We will also have a maximum value of rate $\left(\text{when } \frac{dw}{da} = 0\right)$ where $\bar{a} = l\cdot\bar{b} = 0.5$ (curve 2 in Fig. 11).

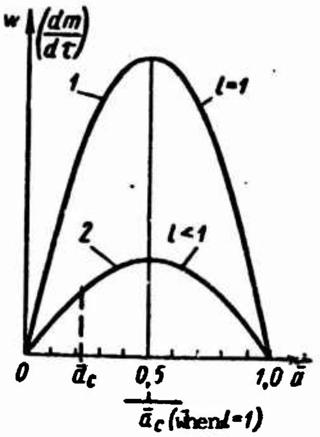


Fig. 11. Dependence of the rate of reaction on the composition of a mixture.

Consequently, the inert admixture, changing the quantitative composition of the initial stoichiometric mixture, does not change the relative content of the actually reacting substance which is necessary for attaining the greatest value of the rate of the chemical reaction.

In the first case, the stoichiometric composition which corresponds to the equation of the chemical reaction was determined by the value $\bar{a} = 0.5$. In the presence of an inert admixture, in order to find the stoichiometric value of concentration $\bar{a}_c$, it is necessary to jointly solve two equations: $\bar{a} = l\bar{b}$ and $\bar{a} + \bar{b} = 1$, which give $\bar{a}_c = \frac{1}{1+\frac{1}{l}}$. In this way, the less portion l, the less $\bar{a}_c$ will be and the greater $\bar{b}_c$. These general conclusions are valid also for other reactions, where in certain cases, admixtures can change the order of a reaction.

§ 16. The Course of a Chemical Reaction Under Adiabatic Conditions

At a constant temperature the rate of a chemical reaction is basically determined by the concentration of reacting substances, which drops in the course of time. Therefore, the maximum rate will be at the beginning of the process when the concentration of substances is greater (Fig. 12, curve 1). Combustion reactions are characterized by great evolution of heat which with a rapid course of the process and insignificant heat removal leads to a progressive growth of temperature. The effect of temperature on the rate of a

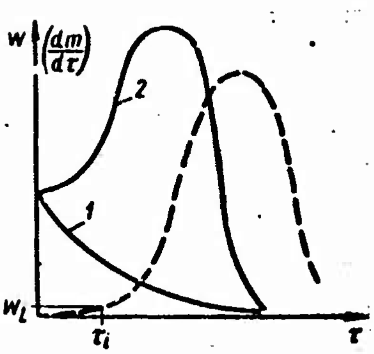


Fig. 12. Dependence of the rate of reaction on time.

reaction is considerably greater than the effect of the concentrations of the reacting substances. This also determines another picture of the change in the rate of reactions under adiabatic conditions (curve 2 in Fig. 12). In general, for example, for a trimolecular reaction, the rate

$$w=\frac{dm}{d\tau}=k_0\cdot e^{-\frac{E}{RT}}a\cdot b\cdot c.$$

Knowing the basic composition of a mixture, it is possible to find the connection between the concentrations of all the substances. Let, for example, $b = b'\cdot a$ and $c = c'\cdot a$. For simple reactions and a stoichiometric mixture $b' = c' = 1$; then the expression of rate of reaction will be formally written thus:

$$w=\frac{dm}{d\tau}=k_0\cdot e^{-\frac{E}{RT}}\cdot a^i, \tag{12}$$

where i - an exponent, numerically equal to the order of the reaction. In each unit of volume, under adiabatic conditions, let a reaction originate with an initial concentration a_0 (mole/cm^3) and an initial temperature T_0, attaining at some moment of time, values of a and T respectively, and at the end of the reaction $a_K = 0$ and temperature T_K. If the thermal effect of reaction for each mole of reacting substance is equal to q, and the heat capacity of the products of combustion with the given volume and in the given temperature range c_v, then the instant value of temperature $T=T_0+\frac{q}{c_v}(a_0-a)$ and final $T_K=T_0+\frac{a_0q}{c_v}$, or $\frac{c_v}{q}=\frac{a_0}{T_K-T_0}$. From the previous expression $a_0-a=(T-T_0)\frac{c_v}{q}$.

Then

$$a_0-a=a_0\frac{T-T_0}{T_K-T_0},$$

whence

$$a = a_0 \frac{T_{\text{к}} - T}{T_{\text{к}} - T_0}.$$

In this way

$$w = \frac{dm}{d\tau} = k_0 \cdot e^{-\frac{E}{RT}} a^l =$$
$$= k_0 \cdot e^{-\frac{E}{RT}} \left(a_0 \frac{T_{\text{к}} - T}{T_{\text{к}} - T_0}\right)^l.$$

Hence it is apparent that with an increase in T, despite the decrease in the expression in brackets, the value of w at first strongly increases because of the growth of $e^{-\frac{E}{RT}}$ until with the decrease in the concentration, the difference $T_{\text{н}} - T$ assumes very small values, and then is reduced to zero. Practically, the greatest value of $w_{\max}$ is reached usually after this, as 80-90% of the initial product is expended.

If the initial temperature is low, then the process will proceed just as shown in Fig. 12 by the dotted line.

Here on the time segment τ_i (the period of retardation of the reaction or the period of induction) the reaction practically proceeds unnoticed to a value of w_i, whereupon it begins to increase rapidly. The value of τ_i depend on a whole series of circumstances which will be examined below.

CHAPTER V

CHAIN REACTIONS

§ 17. Fundamentals of Kinetics of Chain Reactions

The characteristics of the actual course of many reactions cannot always be explained in terms of the propositions of the theory of the thermal activation of molecules. A number of reactions in general do not require any preliminary preheating and can proceed at low temperatures, are frequently isothermal, and nevertheless at considerable rates. There are processes, which can go on only actively with a specific light effect (photochemical reactions) or with introduction into the mixture of trace amounts of extraneous (frequently inert) admixtures. Experimental research has shown that the actual mechanism of the reaction is different.

It was explained, that frequently, reactions proceed faster and more easily as a result of the formation of intermediate active products possessing rather little activation energy, and not as the result of direct collisions between the molecules of the initial substances which possess high activation energy. As a rule, active intermediate products prove to be free radicals or atoms which react with the initial substances, giving an end product, and simultaneously one additional or several new active substances. The latter ensure the further course on the process called a chain reaction.

The simplest illustration of a photochemical chain reaction we can take is the combination of chlorine with hydrogen. The bimolecular reaction $H_2 + Cl_2 = 2HCl$ has great activation energy, about 40,000 cal/mole or 167,700 J/mole. This determines the fact that under usual conditions practically it cannot proceed in accordance with the given equation. The first cycle of the actual reacting is as follows: absorbing a specified number of light quanta $h\nu$, the chlorine molecule is excited ($Cl_2 + h\nu = Cl_2^*$) and it becomes capable of decomposing into atoms of $Cl_2^* = Cl + Cl$. The chlorine atoms, very active intermediate products, cause a further chain of conversions which proceed according to the diagram: $Cl + H_2 = HCl + H$. The activation energy of this reaction in all is about 6000 cal/mole or 21,110 J/mole. Here, besides the product of the reaction, there is obtained yet another and new active center in the form of a hydrogen atom, which produces the reaction

$$H + Cl_2 = HCl + Cl.$$

At this the first cycle of the given chain reaction finishes, after which follows the second, etc., which proceeds according to the diagram

$$\left.\begin{aligned} Cl + H_2 &= HCl + H \\ H + Cl_2 &= HCl + Cl \end{aligned}\right\} \quad \text{2nd cycle}$$

. . . .

Since the concentration of active centers does not change, then the given reaction with nonbranching chains proceeds at a constant rate (Fig. 13a) until a break occurs in a chain as a result of the combination $H + H = H_2$, $Cl + Cl = Cl_2$ or $H + Cl = HCl$ or until interaction with the wall. With sufficient elevation in the temperature, dissociation of the chlorine molecule is possible because of its collision with an *rapid* molecule $\bar{M}^*$, possessing increased energy according to the diagram

$$Cl_2 + M^* = Cl + Cl + M.$$

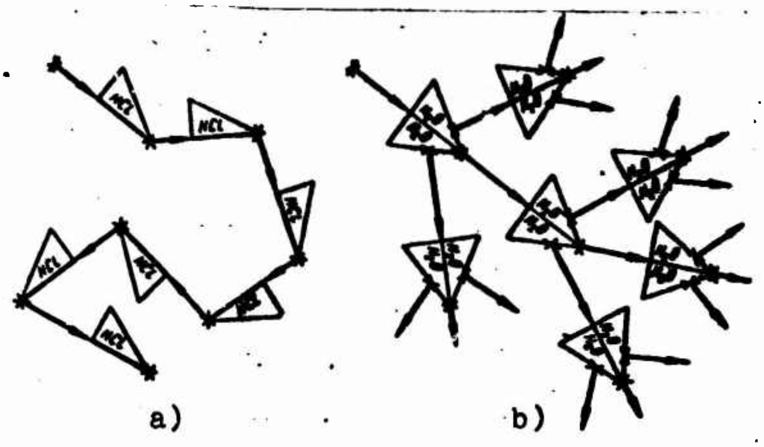


Fig. 13. Diagram of chain reactions: a) nonbranching; b) branching.

Consequently, the transmission of energy can also produce a chain reaction (energy chains) as the formation and the reacting of atoms or radicals (material chains).

Chain reactions with branching chains proceed conserably faster with a constantly increasing rate. Let us examine the reaction $2H_2 + O_2 = 2H_2O$. This reaction does not obey the kinetic equation of the type $w = k \cdot a_{H_2}^2 \cdot a_{O_2}$, since it has a chain reaction mechanism. The first event is usually thermal dissociation of the hydrogen molecule upon meeting with any *rapid* molecule:

$$H_2 + M^* = H + H + M$$

Further chain of conversion of the hydrogen atom is as follows:

$$\left.\begin{matrix} H + O_2 = OH + O \\ O + H_2 = OH + H \\ OH + H_2 = H_2O + H \\ OH + H_2 = H_2O + H \end{matrix}\right\} \text{1st cycle}$$

$$\text{Summary: } H + O_2 + 3H_2 = 2H_2O + 3H$$

Then a recurrence of the first cycle follows. In the beginning. 3 times as many centers are obtained (since after the first cycle we obtain two moles of products of reaction and three new active centers), the second time - 3 times greater number of centers, etc. (Fig. 13b).

It is natural that under some conditions the reaction can be slowed as a result of a break in the chains in connection with the destruction of part of the active centers, for example, according to the diagram $H + H + M = H_2 + M^*$. Energy, which is released with the combination of hydrogen atoms, in this case is absorbed by the M molecule.

The course of branching chain reactions is determined by the following basic processes:

1) by the generation of chains, i.e., by the formation of active centers and by the beginning of chain conversions;

2) by the branching of the chains; in this case one active center, reacting, forms a product of reaction and, in addition, two or more new active centers;

3) by the breaking of chains, which leads to the destruction of active centers (deactivation).

In the beginning of any reaction a certain time τ_i is spent on generating and accumulating active centers. In so doing, if the rate of branching of the chains is greater than the rate of their breaking, then the rate of the chemical reaction will increase in proportion to the increase in the number of active centers. If the breaking of chains exceed the rate of their branching, then a reaction will not be developed, its rate will approach a minimum steady value, i.e., the reaction will practically cease.

The time τ_i necessary for initiation and development of chains, bears the designation of delay or induction period of a reaction, when the reaction proceeds practically unnoticeably, and only after this time does the rate of reaction begin to increase to very great values.

Each active product gives rise to a chain of recurrent reactions, the number of which (the so-called length of the chain) depends on the surrounding conditions and, in particular, is determined by the factors of branching and breaking of chains.

§ 18. Rate of Chain Reactions

On the basis of the theory of chain reactions, first developed by N. N. Semenov, active centers can appear both because of thermal activation, and also in the course of the reaction itself.

Designating the average time of the existence of a particle from its origin to entry into a reaction as $\Delta\tau$, and the time to its possible destruction as a result of mutual collisions as $\Delta\tau_1$, it is possible to calculate the average time "life" of an active particle

$$\Delta\tau_{cp} = \frac{1}{\frac{1}{\Delta\tau} + \frac{1}{\Delta\tau_1}}.$$

If the instant concentration of active particles of the initial substance A is equal to a_a, then in a unit of time $a_a/\Delta\tau$ active particles will react in each unit of volume, i.e., the rate of a chain reaction $w \sim \frac{a_a}{\Delta\tau}$. If, as a result of each individual reaction on the average σ new active particles are created, then in a unit of time only as the result of a reacting there appears $w\sigma \sim \frac{a_a}{\Delta\tau}\sigma$ active particles.

Apart from this, because of the thermal motion of particles in a unit of time there appear a'_{a_Σ} active centers which take part in the development of a chain reaction.

Since the average life of particles equals $\Delta\tau_{cp}$, then in a unit of time $a_a/\Delta\tau_{cp}$ active particles will perish.

The general change in the concentration of active particles in the course of time will be

$$\frac{da_a}{d\tau} = a'_{a_\Sigma} + w\sigma - \frac{a_a}{\Delta\tau_{cp}}.$$

If the reaction is practically not proceeding, then $w \to 0$.

$$\frac{da_a}{d\tau} \to 0 \text{ and } \Delta\tau \to \infty.$$

In this case

$$a'_{a_\Sigma} = \frac{a_a}{\Delta\tau_1}.$$

Designating for this case a_a as a_{a_T}, we obtain

$$a_{a_T} = a'_{a_\Sigma}\cdot\Delta\tau_1 \approx a\cdot e^{-\frac{E}{RT}},$$

i.e., as one would expect, we obtain a number (a concentration) of active centers, forming stationary only because of thermal activation.

In this way, the rate of a chain chemical reaction is determined, in the first place, not by the overall number of active centers a'_{a_Σ}, obtained because of thermal activity, but only by those fractions a'_a of substance which actually reacted. In the second place, it is obvious that the overall number of individual reactions which proceed in a unit of time will exceed the number of reactions determined by the initial active centers a'_a, since each initial reaction causes a chain of secondary reactions which accelerate the reacting. If we designate the number of these secondary reactions in a chain (the so-called length of the chain) by the letter r, then the rate of reaction $w \sim a'_a \cdot r$.

In the course of a reaction each active center, having reacted, can create σ new centers. The value of r is determined only by those centers which determined the flow of secondary reactions. The remaining part of them could not react, specifically, because of breaking of chains. In this way, the factor of breaking of chains is unequivocally associated with the length of chain $r = \frac{1}{\beta}$ (where β - the characteristic of probability of chain break). Consequently, for reactions with nonbranching chains

$$w \sim a'_a \cdot r = \frac{a'_a}{\beta}.$$

In a process with branching chains, instead of a reacting active center, at least two, and generally σ new ones appear, whereby the appearance of each supplementary active center lessens the effect of the factor of chain break. Consequently, the length of the chain of reaction with branching chains will be characterized by the value $r = \frac{1}{\beta - \delta}$ (where δ – a factor which determines the probability of branching of a circuit). The quantities β and δ are associated with the values respectively of the second and third members of the right part of the equation $da_a/d\tau$, but not equal to them, and represent some fraction of them.

Factor δ takes into account the effect of not all newly appearing active centers, but only those which actually react. Values δ, β and r are functions of a number quantites, first of all, T and p. Therefore, under specific conditions it is possible to have the equation $(\beta - \delta) = 0$ and $r \to \infty$. This means that any initial chain is reproduced in one of its branches. Where $\delta > \beta$ and $\delta - \beta > 0$ because of the excess of the number of branchings over the breaks in the chains an intensive reaction of explosive type takes place. Values β and δ depend on temperature exponentially, similar to the dependence of a'_a. Dependences of the rate of a chain reaction on time are as follows.

The change in respect to time of the concentration of active centers which take part in a reaction

$$\frac{da_a}{d\tau} = a'_a + K_p \cdot a_a - K_{\text{об}} \cdot a_a, \qquad (13)$$

where products $K_p \cdot a_a$ and $K_{\text{об}} \cdot a_a$ - respectively, changes in the concentration of active centers in time as a result of branching and breaking of chains.

Expressions K_p and $K_{\text{об}}$ are determined respectively by values of δ and β and by the average life of the active centers:

$$K_p \sim \frac{\delta}{\Delta\tau} \text{ and } K_{\text{об}} \sim \frac{\beta}{\Delta\tau},$$

where the difference

$$K_p - K_{\text{об}} = \varphi \sim \frac{\delta - \beta}{\Delta\tau}.$$

In this way,

$$\frac{da_a}{d\tau} = a'_a + \varphi a_a.$$

As a result of change in external conditions values of δ and β change, and also their relationship, and this means, also the value of ϕ which can be both greater than zero and less than zero. The value a'_a, characteristic of the rate of the process of thermal formation of active centers, can be considered to depend on time. Then for ϕ = const, we obtain

$$\frac{da_a}{a'_a + \varphi a_a} = d\tau \text{ and } \frac{1}{\varphi}\ln\left(a'_a - \varphi a_a\right) = \tau + \ln C.$$

Since with $\tau = 0$ $a_a \to 0$, then $\ln C = \frac{1}{\phi}\ln a'_a$.

Consequently,

$$\ln\frac{a'_a + \varphi a_a}{a'_a} = \varphi\tau,$$

whence

$$a_a = \frac{a'_a}{\varphi}\left(e^{\varphi\tau} - 1\right). \qquad (14)$$

Since the rate of formation of the products of reaction (the rate of reaction w) is proportional to the concentration of active centers at the given instant, then, knowing how many products of reaction each active center gives, it is possible to determine the value $w \sim l \cdot a_a$ (where l - a factor, which indicates how many molecules of the product of reaction are obtained in the reacting of one active center). For example, for the reaction of the formation of H_2O, the factor $l = 2$.

On the other hand, the rate of reaction $w \sim \frac{a_a}{\Delta\tau}$, and if we consider that $K_p = \frac{\delta}{\Delta\tau}$ and $K_{об} = \frac{\beta}{\Delta\tau}$, then

$$w = \frac{a'_a}{\delta - \beta}\left[e^{\frac{\delta-\beta}{\Delta\tau}\tau} - 1\right]. \qquad (15)$$

For reactions with nonbranching chains ($\delta = 0$)

$$w = \frac{a_a'}{\beta}\left[1 - e^{-\frac{\beta}{\Delta\tau}\tau}\right].$$

With the increase of β, the quantity in brackets tends toward unity, and the value of the rate approximates $\frac{a_a}{\beta} = a_a' \cdot \tau$. In this way, reactions with nonbranching chains only formally where $\tau \to \infty$ and $\beta \to 0$ can have infinitely high rates. Actually, the rate of such reactions is limited. Reactions with branching chains ($\delta \neq 0$) can have three characteristic cases of flow:

1) $\phi < 0$ or $(\delta < \beta)$ - conditions analogous to a case when $\delta = 0$. Here $a_a \to \left(\frac{a_a \Delta\tau}{\beta - \delta}\right)$ and $w \to \frac{a_a}{\beta - \delta}$, i.e., rates cannot achieve significant values.

2) $\phi = 0$ or $\beta = \delta$. The rate of reaction here is determined by the process of thermal activation and by time, i.e., $\frac{da_a}{d\tau} = a_a'$ and the value of $w \sim a_a'\tau$.

3) $\phi > 0$ where $\delta > \beta$. This is the case of intensive flow of a reaction with unlimited growth in the rate in time (to the exhaustion of the initial mixture), i.e., the typical course of a chain reaction with branching chains. It is characteristic that very high rates of reaction can be obtained even at small values of τ, in particular, with the process where T = const, not only with small values of the difference $(\delta - \beta)$. Even with small values of a_a', the value of w, exponentially growing in time, becomes noticeable after a specified τ_i - the period of induction. In the course of τ_i, the value of rate w (Fig. 14) begins to increase sharply, attaining very high values. Figure 14 shows all the considered cases with φ values invariable in time.

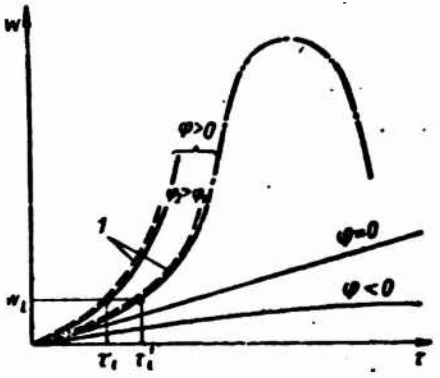


Fig. 14. Dependence of rate of a chain reaction on time.

For the case of $\phi > 0$, two curves are shown, moreover $\phi_2 > \phi_1$. Since the value of ϕ can also increase strongly in time, the curves of change of w will proceed more steeply as indicated by the broken line 1 in Fig. 14.

The value of rate w_i corresponding to τ_i, is determined practically by the available accuracy of measurement of the rates of reaction. Into the period of induction chains are generated, on the basis of the branching of which active products are accumulated, sufficient for ensuring the explosive character of the further course of the reaction.

CHAPTER VI

IGNITION OF A COMBUSTIBLE MIXTURE

§ 19. Self Ignition. Thermal Explosion

In the process of combustion, the thermal conditions in the reacting medium are determined both by the process itself and by the effects of external conditions. It is practically impossible to ensure uniformity in the course of a process in all parts of a volume since the mixture will be ignited initially in some part of the system, the heat from various parts is transferred to other parts and the environment differently. This determines the heterogeneity of state and the difference in the characteristics of the course of a process in various parts of a reacting system. In a volume there will be variations in temperatures, rates of reaction, concentration of substances, and so forth. If the volume is limited by walls, then about a cold wall the course of the reaction will be slowed and naturally, the concentration of the products of combustion here will be less than in the depth of the volume. This leads to the initiation of processes of heat-and mass-transfer. These processes (thermal conductivity and diffusion) taking place during a chemical reaction, create great complexity in the calculation of the characteristics of such occurrences. The essence of the process of self-ignition is more easily analyzed by following N. N. Semenov's method on the assumption that under defined conditions a mixture reacts equally in all parts of a volume. Let us assume that a vessel which possesses a volume of V cm^3, has been filled with a combustible gas mixture which possesses a temperature T. The temperature in the course of the process can change but it will be simultaneous in all parts of the volume. The temperature of the walls of the container T_s is preserved unchanged and will always equal temperature of environment T_o. Such a scheme determines that differences in temperatures will exist only at the boundary of the gas and the wall, and as a result of this, a difference in the concentrations of the reacting system will be lacking.

If the experiment is conducted at low temperature $T_o = T_s$, then $T \sim T_o$ since the reaction practically does not proceed. With an increase in the temperature $T_o = T_s$ temperature T is also raised. It is obvious that at a certain value of its rate the reaction becomes noticeable and with a further increase T will also increase. The heat being liberated in the course of the reaction increases the temperature of the mixture, determining the increasing difference $T - T_s$. If the removal of heat from the mixture to the wall is less than the evolution of heat as a result of the chemical reaction, then the temperature of the mixture progressively increases. This will sharply accelerate the reaction and will practically lead to its instantaneous completion — to a thermal explosion.

If in the combustion of each mole of the mixture there is liberated Q kcal (kJ) of heat, then the rate of the evolution of heat

$$q_н = QwV, \qquad (16)$$

where $w = k_0 \cdot a^i \cdot e^{-\frac{E}{RT}}$ mole/(cm^3·s) – the rate of the chemical reaction.

Part of this heat goes to heating the gas, and another part is carried off through the wall into the environment. The rate of heat emission from the gas to the wall can be determined thus:

$$q_o = \alpha s (T - T_s), \qquad (17)$$

where s – wall surface;
α – heat transfer coefficient.

Depending on the concrete value of the temperatures, coefficient α, surface s, concentration a, the pressure in the mixture (determining w), etc, the relationship between quantities q_n and q_0 can be different, as Fig. 15 graphically shows. For a given mixture with invariable pressure, a defined volume V, and α = const, by changing only T_s, it is possible to obtain a number of characteristic cases of the arrangement of curves q_n and q_0 given in Fig. 16. With the combination of curves q_n and q_0(I) the mixture is heated up to temperature T_1, since from $T = T_s$ and to $T = T_1$ quantity $q_n > q_0$. This circumstance changes where $T = T_1$, when $q_n = q_0$. Further, q_0 will be greater than q_n to point 2 where again $q_n = q_0$, and further q_n becomes greater than q_0. However, the system can be brought to this situation only by means of external preheating. Actually for this case, a steady equilibrium of the system will be observed only at point 1, where the reaction proceeds with a constant low rate and where $q_n = q_0$.

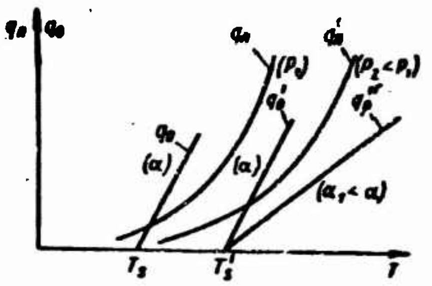


Fig. 1. Dependence of heat evolution and heat transfer on temperature during a chemical reaction.

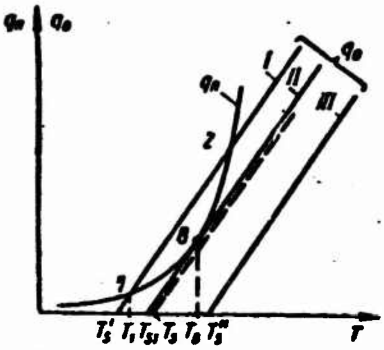


Fig. 16. Determination of the temperature of self-ignition of a mixture.

For the case q_n and q_0(III) everywhere $q_n > q_0$ up to considerable temperatures T and correspondingly w, in the system from the very beginning there will be a continuous increase in temperature, and the rate of reaction will correspondingly attain maximum values, and as a result an explosion will occur. The boundary of transition from the first case (the absence of an explosion) to the second will be the position of curves q_n and q_0(II). Here in the beginning, as just in the first case, the system rapidly reaches equilibrium at point B (T_B). But this equilibrium, in contrast to the state of the system which it had at T_1 will be unstable. An insignificant elevation in temperature T_s (broken straight line) is sufficient to generate an explosive process. It is clear that all the other higher values of T_s all the more so will lead to an explosion, and moreover in shorter time periods. In this way, the tangency of curves $q_п$ and q_0 is the critical case which determines the conditions of self-ignition of a system. Temperature T_B – this is the temperature of self-ignition which, as was shown, is not a physio-chemical constant of the given mixture, but characterizes the system as a whole. The value of T_B depends on the value of $V(s)$, the material of the wall, α, the conditions of the experiment p, a, and so forth.

Temperature T_B determines the course of the reaction for the period of induction $\tau_i = \infty$ which impedes its practical utilization in industry. Usually the temperature of self-ignition of a given mixture under given concrete conditions is considered to be that minimum temperature of wall T_s at which in the course of a specified segment of time an explosion occurs. That is to say, T_B or the temperature of the wall T_{s1} corresponding to this condition is not taken, but a somewhat higher temperature $T_s = T_{s1} + \Delta T_s$, at which $\tau_i \neq \infty$, and is equal to several seconds. The temperature of self-ignition for various mixtures can be compared only under the condition of their determination in similar instruments by identical methods.

N. N. Semenov pointed out that for combustible mixtures there can exist an area of characteristic parameter values in which under given conditions an explosion is possible. The boundaries of this area (explosive boundaries or self-ignition boundaries) can be found from analysis of curves $q_п$ and q_0(II) in Fig. 16. Taking into account that at point B $q_п = q_0$ and straight line q_0 is tangent to curve $q_п$, i.e., $\frac{\partial q_п}{\partial T} = \frac{\partial q_0}{\partial T}$, it is possible to write:

$$QVk_0a^le^{-\frac{E}{RT_B}} = \alpha s(T_B - T_s), \qquad (18)$$

$$\frac{E}{RT_B^2}QVk_0a^le^{-\frac{E}{RT_B}} = \alpha s. \qquad (19)$$

Dividing expression (18) by (19), we obtain

$$QVk_0a^ie^{-\frac{E}{RT_s}} = as\frac{RT_s^2}{E}.$$

since the difference between T_B and T_s is very small in comparison with T_s, and E is very great. If in the left part of equation (18) in the expression of degree it is considered that $T_B = T_s$, then allowing for the last relationship, it can be rewritten thus:

$$QVk_0a^ie^{-\frac{E}{RT_s}} = as\frac{RT_s^2}{E}.$$

After a logarithmic operation, we have

$$\ln\frac{a^i}{T_s^2} = -\frac{E}{R}\cdot\frac{1}{T_s} + \ln\frac{asR}{QVk_0E}. \qquad (20)$$

This expression gives the dependence $T_B = T_s = f(a)$. By changing concentration a through pressure $a^i \sim p^i$, it is possible to obtain dependence $T_B = F(p)$.

The form of these dependences is shown in Fig. 17.

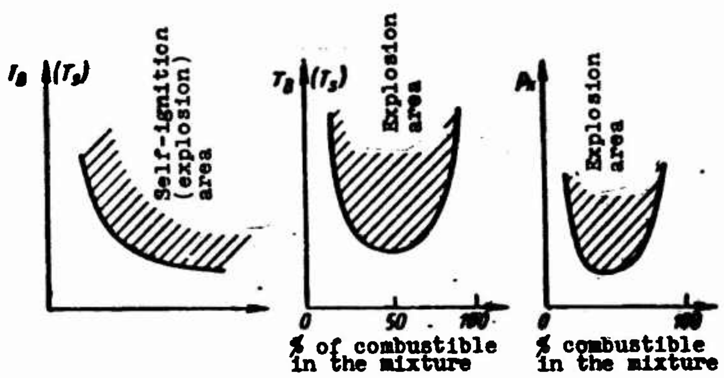


Fig. 17. Various dependences for the boundaries of the explosion area.

The Period of Ignition Delay

Depending on the reaction ability of the mixture, its state, and also the temperature of the wall and its features, a mixture will be heated up differently. Specifically, the rate of the increase in temperature of a mixture and the boundaries of its growth will change. The change in temperature in time will be determined by the difference $q_{\text{п}} - q_0$ and the heat capacity of the system $V \cdot c_v$, i.e., $\frac{dT}{d\tau} \sim \frac{q_{\text{п}} - q_0}{Vc_v}$ deg/s. If the conditions of heat evolution and heat transfer are characterized by curves $q_{\text{п}}$ and q_0(I) (see Fig. 16), then with a change in temperature from T_s To T_1 the positive value $\frac{dT}{d\tau}$ is always reduced to zero where $T = T_1$. The system cannot be heated up higher than this temperature (Fig. 18), and taking into account the consumption of the combustible, the temperature will even begin to drop (curve *I*). An analogous situation will be retained and with a certain growth of T_s until the condition of heat withdrawal q_0(II) is achieved. This maximum situation provides heating up of the system to temperature $T_{\text{В}}$. Any small increase in T_s will now lead to conditions of heat withdrawal q_0(III) when the temperature increases continuously up to very high values characteristic for combustion reactions. The certain retarding of the rate of growth of temperature around point $T_{\text{В}}$ (a point of inflection) will be increasingly less and less noticeable with a further increase of T_s as shown in Fig. 18. The point of inflection determines the end of the induction period of a thermal ignition τ_i and a change to an advanced chemical reaction. The locus of the points of inflection of the series of curves will determine dependence $\tau_i = f(T)$ (the dot-dash line in Fig. 18).

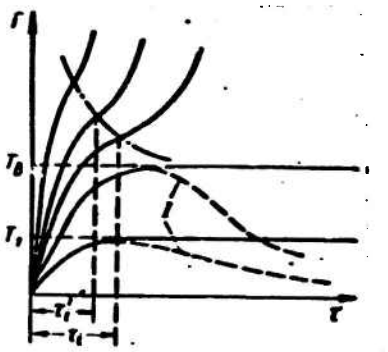


Fig. 18. Change in temperature of a mixture with time.

The temperature of self-ignition $T_{\text{В}}$ (T_s) depends also upon pressure, the concentration of the mixture, and so forth. Depending on the method of its determination, naturally, differences are possible in the value of this temperature.

Figure 19 shows the dependence of T_B on the composition of the mixture for various combustible gases with air. On the same graph the effect is shown on the pressure of pentane mixtures on the value of T_B. As the experiment showed, in individual cases, a stepwise flow of curves $T_B = f$ (concentration) can be observed. For other mixtures with a change in pressure, especially in the range of small values, there are also deflections of temperature from theoretical dependences.

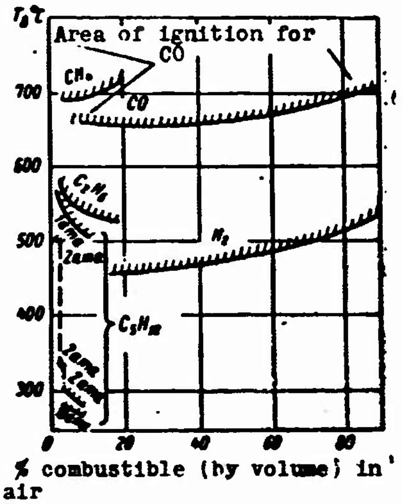


Fig. 19. Dependence of temperature of self-ignition of various gases in air on concentration of the combustible. ama = [atm(abs.)]

Thus, Fig. 20 shows a curve for methane which has an inflection. From the graph it is evident that at a temperature of about 650°C there is not one critical point, but three (*I*, *II*, *III*), in which the mixture passes over from an absence of explosion to ignition and back. The explanation of this fact can be found with the aid of the theory of chain reactions.

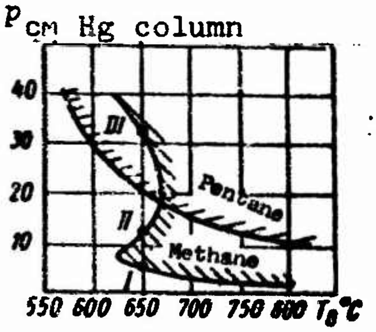


Fig. 20. Dependence of temperature of self-ignition of methane and propane on pressure.

§ 20. Self-Ignition of a Mixture in a Flow

To reveal the basic characteristics of self-ignition of a mixture moving in a flow, and the effect of the condition of its flow on self-ignition and combustion of the mixture, we will examine a simplified case.

Let us assume that a combustible mixture flows via any channel with a specified temperature of the walls, in parallel streams which possess identical rates. In this instance, each small volume of the gases can be considered isolated. The state of the mixture will be determined by the relationship of heat release and heat transfer. If the heat release as a result of the chemical reaction being developed exceeds the heat withdrawal, then in proportion to the motion of the mixture, its temperature will be increased up to the temperature of self-ignition. Intensification of the process can be effected both by the supply of heat from the heated wall and by reheating because of the mixing of the hot products of reaction and fresh mixtures. Although this admixture, somewhat reduces the concentration of the combustible, however the elevation in temperature ensures the growth of the rate of reacting. The chemical reaction here is determined by the heat release $q_1 = \omega VQ$, where V – the volume of the pipe (curve *I* in Fig. 21).

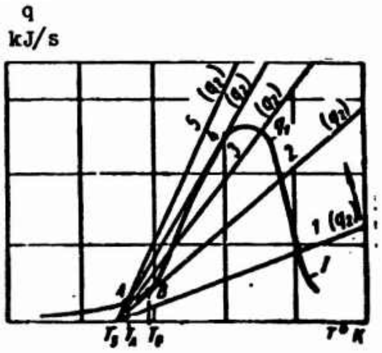


Fig. 21. Heat release and heat transfer in a system with self-ignition of a mixture in a flow.

Ignoring the losses of heat into the environment, it may be considered that the entire heat being released goes to reheating of the mixture. The products of the reaction continuously take away from the zone of reaction a quantity of heat

$$q_2 = Gc_p (T - T_0),$$

where G – expenditure of gas per second;

c_p – heat capacity at constant pressure;

T_0 – initial temperature;

T – temperature to which the mixture is heated.

At a wall temperature of $T_в$ and various expenditures (rates) of the mixture, the quantity of heat withdrawn q_2 will be represented as straight lines 1-5.

With the tapping of heat according to line 5 with a range of temperatures from $T_в$ to T_A, the evolution of heat is greater than the heat withdrawal ($q_1 > q_2$). If the process of heat withdrawal is determined by line 2, then preheating is continued to temperature T_B, but a further insignificant decrease in the rate (expenditure) will lead to withdrawal of heat according to line I. In this instance, the heat release exceeds the heat withdrawal, and the mixture ignites. It means that the maximum case of ignition is determined by line 2.

However, if combustion began with the tapping of heat, for example, on line 1, then it is possible to force the process up to rates producing a withdrawal of heat according to line 4 (maximum case). A steady action can be ensured with expenditures determining heat transfer proceeding over line 3.

In actual flows, the processes of combustion are complicated by the presence of nonuniformity of rates, temperatures, and concentrations of mixture over the section of the channel. In addition, if the rates of flow should exceed the rates of flame propagation in turbulent flows, measures are necessary to stabilize the flame front in a definite part of the volume of the fuel ignition device (the combustion chamber).

§ 21. Forced Ignition (Firing)

Forced ignition (firing) is the term applied to ignition of a combustible mixture which possesses a temperature considerably below T_B by intense heating of a small volume of it by an extraneous heat source. For heating, it is possible to introduce into the combustible mixture an outside flame, electrical spark, an incandescent solid, and so forth.

The physical side of the process of forced ignition is not fundamentally different from self-ignition. However, here the acceleration of the reaction initially goes in a small volume and only with time is it extended to the whole mixture. The calculation of this occurrence, naturally, is complex. The occurrence can be somewhat simplified if we consider that in the beginning of the process, the concentration of the combustible mixture everywhere is identical and is equal to initial c_0. These facts exclude the processes of diffusion of reacting substances from the discussion. The given condition can be used until an intense chemical reacting is initiated.

Firing by a Highly Heated Body

If we introduce a highly heated body (a plate) into a volume filled with a mixture with temperature T_0, then, depending on the value of T_{CT} and the reaction capability of the mixture, the effect will be different. In an inert mixture, the layers lying nearest to the plate are heated (curves 1, Fig. 22), but even at a certain distance l_1, the mixture, not capable of chemical reacting, will have a temperature close to T_0. If the incandescent body is introduced into a combustible mixture, and its temperature T_1' does not differ very greatly from T_0 at which the mixture practically does not react, then the field of temperatures nevertheless will change somewhat. In the layers directly adjacent to the plate, approximately at a distance of l_1, a more intensive chemical reaction will be initiated in the mixture which, in turn, will increase the temperature of these layers (curve 2). The heat being evolved will be characterized by the extent of the shaded areas between curves 1 and 2.

The sizes of these areas increase with the increase in the

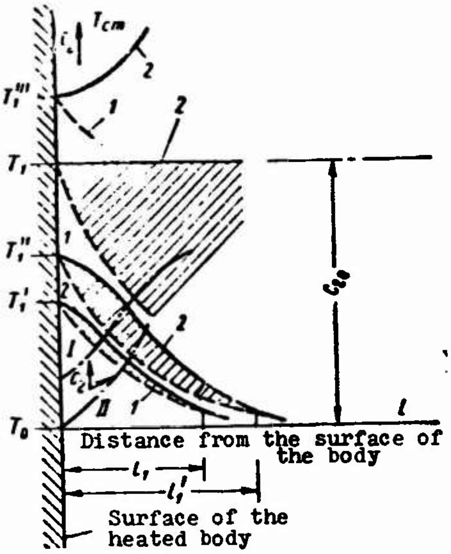


Fig. 22. Temperature fields and concentrations of a combustible mixture around the surface of a highly heated body.

temperature of the incandescent body T_{CT}. It is obvious that where $T_{\text{CT}} = T_1$ in the combustible mixture, at least, near the wall the temperature is constant and is equal to T_1.

For a combustible mixture, this value $T_{\text{CT}} = T_1$ will be critical, determining the condition of firing T_3. At that temperature $\left(\frac{dT}{dn}\right)_{cm} = 0$, ($n$ – normal to the wall). It is obvious that in this case an insignificant increase in T_{CT} will cause growth in the temperature of the combustible mixture and the higher T_{CT}, the greater it will be. In essence T_3 is analogous to the temperature of self-ignition T_{B}, moreover when $T_{\text{CT}} = T_3$, heat flow from the body to the mixture already beginning to react is absent, and the temperature of the mixture is determined only by the process, i.e., by the totality of occurrence of the heat transfer and the mass transfer in the presence of the chemical reaction being developed. An experiment shows that $T_3 > T_{\text{B}}$ and the less the size of the heated body the greater will be the difference $T_3 - T_{\text{B}}$ because of the sharp drop in temperature with distance from the body and the reduction of the concentration of the combustible c_{Γ} in the layers of gas near the wall (dot-dash curve I in Fig. 22). Sometimes cases are observed of reaction of the mixture near the wall, but without distribution of the reaction to the entire volume. It should be noted that if the body is made out of material which is a catalytic agent, then firing is impaired, and, other conditions being equal, the firing of the mixture will require a higher temperature of the wall. This is explained by the fact that the concentration of combustible at the wall in the latter case (dot-dash curve *II*) is sharply reduced in view of the adsorption of the mixture by the surface of decomposition and the intensive reacting of molecules at the wall.

Of the metals, the best catalytic agent is platinum.

Figure 23 gives comparative values of $T_з$ for bodies of identical size made out of various metals for some mixtures of natural gas with air. The dotted line shows the dependence of temperature $T_з$ for a body of smaller size.

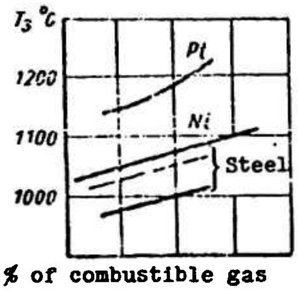


Fig. 23. The effect of the material of a body on the value of $T_з$.

Electric Spark Firing

In industry, for ignition of fuel electric spark firing is used most frequently because of its reliability and convenience in operation. In comparison with ignition by a highly heated body, the process of a firing with a spark is complex and also is not yet entirely clear.

It is known, however, that with an electric spark discharge, besides the increase in the temperature of the mixture, an important role is played by the ionization of the gas which leads to exciting the molecules. The electrical spark which jumps across the electrodes of a spark plug can, of course, be considered as a small incandescent (gas) body with a temperature of 6000-20,000°C. Experiment confirms that the higher the temperature and the bigger the electric spark channel, the more intense is the firing. This determines the paramount role of thermal processes. However, a number of the electrical characteristics which affect the reaction of the medium, and in connection with this, the growth of the reaction capability of the mixture, leading to improvement of firing, indicate the duality of the phenomenon of electric spark firing.

Thus, for instance, an experiment will show the dependence of firing upon current, but not on the square of current, as might be expected only in terms of thermal considerations. For each mixture according to a concentration, pressure, etc, there is a determined minimum spark power, below which firing does not proceed.

The character of the dependence of current necessary for firing on the composition of the mixture is shown in Fig. 24. An experiment will show the great effectiveness of the capacitive component of electric discharge in comparison with induction, which is confirmed by the thermal nature of the process.

From Fig. 24 it is evident that with the increase in the power of the spark, the boundaries of the components of the mixtures in which firing takes place can be enlarged up to a certain limit. The spark with which the effect of firing becomes practically independent of further increase in its power is called the saturation spark; just such a spark gives the most reliable firing, depending on other factors to a minimal degree.

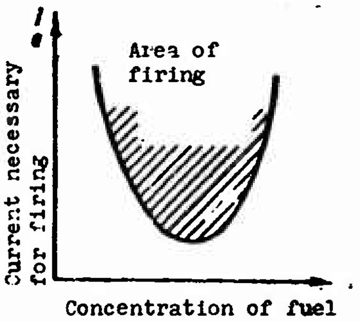


Fig. 24. Dependence of minimum power of a spark on the composition of a mixture.

CHAPTER VII

THE PROCESS OF FLAME PROPAGATION

The firing of a mixture in a rather small part of a system should ensure the further propagation of combustion over the entire volume. We will examine the course of the process of flame propagation (the combustion reaction), for example, in a horizontally arranged tube, closed at one end. Ignition is carried from the open end. After firing there is usually observed a rather gradual, uniform propagation of the flame along the tube with rates of up to several meters per second depending on the concentration of substances, pressure, temperature, and others. This type of flame propagation is called lamellar.

In long tubes, such a process proceeds only to a definite part of the length of the tube, approximately to a length $\left(\frac{1}{5}-\frac{1}{4}\right)l$, or somewhat greater, depending on length l and on the diameter of the tube, the concentration and type of fuel, the state of the mixture, and others. The form of the flame in this case recalls a meniscus the convex part directed towards the fresh mixture, but not symmetric relative to the axis. The diameter of the tube considerably effects the apparent rate of motion of the flame but only to a certain maximum value. With a decrease in the diameter to a certain minimum value, flame propagation in the tube becomes impossible.

After a uniform movement in the initial section of the tube, the flame moves with oscillations or with individual backward kicks simultaneously sharply increasing the average forward rate of motion to several hundred meters per second. This type of combustion is unstable. Individual powerful kicks cause blowout of the flame or even detonation flame propagation (an explosive wave). Detonation combustion is characterized by stable values of rates, but very great in magnitude, exceeding 1000-2000 m/s.

The propagation of flame by volume (lengthwise in the tube) is a shift of a defined area (zone) in which basically chemical conversions are carried out. It is natural that for this in some part of the zone or ahead of it the necessary preparatory processes must take place, for example, the preheating of the mixture. This layer, including the zone of preheating and the zone of chemical conversion, is called the flame front. In this way, the rate of combustion can be characterized by the linear movement of the flame front in a unit of time.

Laminar (or, as it is sometimed called, normal) propagation of the flame front, after ignition of the open end of the tube, proceeds at low rates under conditions of free expansion of the products of reaction (p = const). Fresh mixture is heated from the reacting layer in the flame front during combustion in tubes, and also in the case of combustion in a flow during laminar flow as a result of thermal conductivity. In laminar flow, the processes of exchange take place only because of the motion of molecules without an exchange of impulses and mass between neighboring streams. Laminar flows are carried out at low rates with Re < (2000-3000). The normal rate of flame propagation u_H, i.e., the rate measured on a normal to the surface of a front is a physico-chemical constant of the given mixture for determined conditions. This rate also, to a considerable extent, determines the rate of the propagation of the flame in a turbulent flow u_T taking place in gas turbine engine [GTD] (ГТД) combustion chambers.

A fundamentally distinct phenomenon will be the detonation propagation of the flame front with rates higher than the velocity of sound. In this instance, in the zone of combustion the pressure strongly increases because shock-adiabatic compression, and the development of the process is determined by the propagation of pressure jumps (shock waves).

§ 22. The Rate of Flame Propagation in a Laminar Flow

If in a medium at rest there are noted two consecutive infinitesimally close positions of the surface of the flame front, and if we designate as dn the amount of movement of the element of the flame front in respect to a normal to the surface in a given spot in the direction of fresh mixture, then in the interval of time $d\tau$, the value of $\frac{dn}{d\tau} = u_{н}$ will be called the normal rate of propagation of the flame front.

Hence it follows that the quantity of mixtures which burn down on equal areas of the flame front are identical. If the medium itself moves at rate w, then it is natural that the apparent rate of propagation of the flame front

$$v = u_{н} \pm w_{n}, \tag{21}$$

where w_n – projection of the velocity vector w in the direction of normal n.

The value of u_H naturally does not depend on whether the mixture moves or not, if along with the pure thermal conductivity in the process there is not molar transfer of heat such as takes place in a turbulent flow.

The Flame of a Gas Burner

The most suitable method for for determining value of u_H in the flame of a gas (Bunsen) burner was proposed by V. A. Michelson. The flame of the burner has two visible cone-shaped glowing surfaces: an internal *I* blue in color and an external *II* of yellowish hue (Fig. 25a). On the surface *I* the basic part of the chemical conversion is carried out, the depth of which is determined by temperature, by quantity of oxidizer, etc. In the external cone *II*, the fuel burns usually with the participation of oxygen from the surrounding atmosphere.

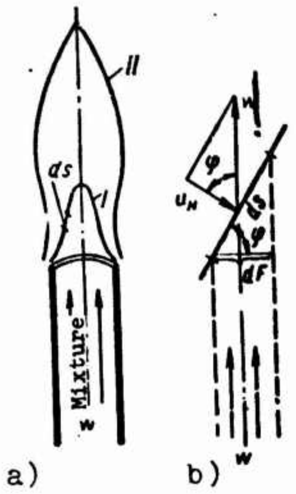


Fig. 25. Diagram of the flame of a gas burner.

During laminar flow, the flame surface is seemingly at rest ($v = 0$), being determined by the motion of the cold mixture at rate w and the counter motion of the flame front at a rate of $u_{н}$, i.e., $u_{н} = w_{п}$. As will be shown in § 29, a steady condition of the flame front requires the constant presence of a source of ignition of the fresh mixture. Fig. 25b shows an element of the flame surface ds, from which it is clear that $u_{н} = w \cos \phi$. From the second such triangle it follows that the element of the area of section of the burner $dF = ds \cos \phi$. Knowing $\cos \phi$ from the first expression, after integrating, we obtain

$$u_{н} = \frac{V}{S_{пл}} \tag{22}$$

where V in cm^3/s – volumetric expenditure of gas through the burner; $S_{пл}$ in cm^2 – flame surface, which is determined according to its photograph.

The fact that the flame does not have the form of a perfect cone, and is rounded at the base and top, shows that the value of $u_{н}$ is not identical everywhere. This occurs because of the heat withdrawal (a decrease in the temperature of the mixture) through the walls of the burner, which reduces $u_{н}$ and as a result of more intense preheating of the mixture at the top, which increases $u_{н}$.

Spherical Flame Propagation

Free spherical propagation of flames from a point source of ignition can be observed in an unrestricted volume of an undisturbed mixture. Under actual conditions, if the ignition of the mixture is effected in a closed volume, then pressure and temperature conditions (because of the effect of the wall) will be changed constantly, changing the character of the motion of the flame front. It is obvious that the component of apparent rate of propagation of the flame front caused by the expansion of the products of combustion, will also become variable, and so forth. We will explain the effect of the expansion of the products of combustion on the rate of motion of the flame. Let us consider the simplest diagram of a plane steady flow of a combustible mixture through a channel of constant section (Fig. 26a). The flame front (conditionally two-dimensional) is located in section *I*. We will select before it a small volume of fresh mixture 1, occupying a certain length of channel $l = I\text{-}II = u_{\text{н}} d\tau$. If there is no flow and the products of combustion 2 were also at rest, then the flame front $S_{\text{пл}}$ in $d\tau$ seconds would be at distance $u_{\text{н}} d\tau$ from the previous spot *I*, i.e., in section *II*. However, allowing for the expansion of the products of reaction and their rate of motion w_2, in reality the flame in $d\tau$ seconds will be in section *II'*, and section *I*, where the flame was at the moment of time τ will move itself into section *I'*, whereby $I\text{-}I' = w_2 d\tau$. The actual path of the flame (with apparent rate v) after time $d\tau$ will equal $v \cdot d\tau$. Since the mass of the selected volume prior to and after passage of flame along it did not change, then it is possible to write down that

$$u_{\text{н}} \varrho_1 = (v - w_2) \varrho_2$$

or

$$u_{\text{н}} = (v - w_2) \frac{\varrho_2}{\varrho_1}.$$

Turning to a case of a spherical flame, let us note that this type of combustion can be obtained also in a limited space if the walls which determine it can be distorted, preserving a constant

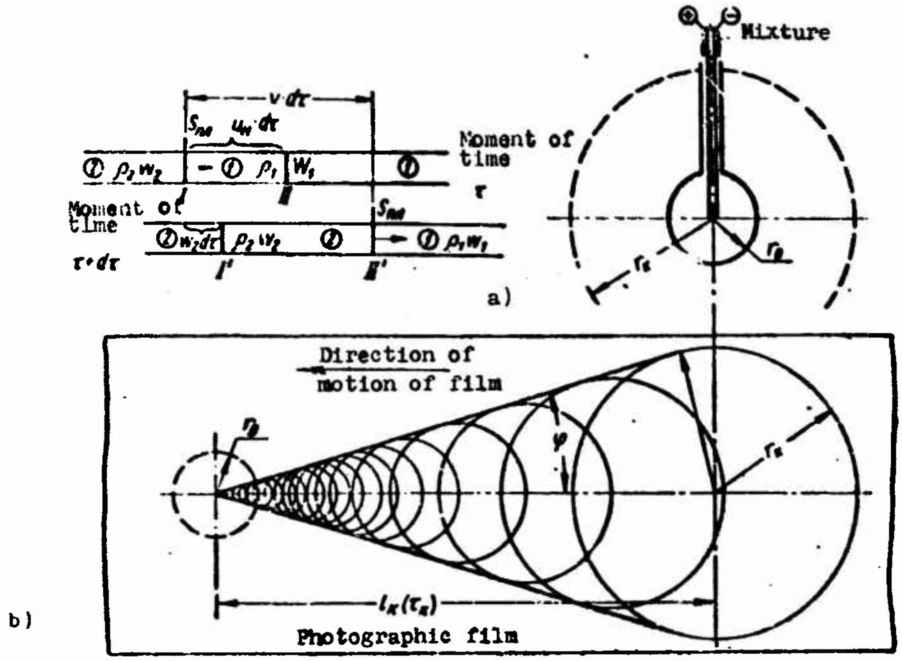


Fig. 26. Diagram of flame propagation.

pressure in the interior of the volume. This diagram can be carried out by blowing a soap bubble through the combustible mixture (Fig. 26b) and igniting the mixture by an electric spark in the center of the bubble. At the moment of ignition, the volume of a bubble with an initial radius r_0 is detached from the volume in which the combustible mixture is located. Combustion proceeds in the volume of the soap bubble (with p = const), naturally, distorting its walls until the flame front reaches these walls ($r = r_{\text{н}}$).

For this case, considering that $w_2 = 0$, we obtain $u_{\text{н}} = v\frac{\rho_2}{\rho_1}$. The value of $u_{\text{н}}$ is determined according to a photograph (on moving film) of the spherical flame, appearing within the bubble, which possesses initially at time $\tau_1 = 0$, $r = r_0$. At moment of time $\tau_2 = \tau_{\text{к}}$, the flame reaches the walls of the bubble and destroys it. At this moment, the radius of the soap bubble has a value of $r_{\text{к}}$.

Taking into account that prior to and after combustion, the mass of working body is invariable,

$$\frac{4}{3}\pi r_0^3\rho_1 = \frac{4}{3}\pi r_{\text{к}}^3\rho_2,$$

whence

$$\frac{Q_2}{Q_1}=\frac{r_0^3}{r_{\text{к}}^3}.$$

Furthermore, $v=\frac{r_{\text{к}}}{\tau_{\text{к}}}$, where $\tau_{\text{н}}$ – the total time of the course of the process of combustion. In this way,

$$u_{\text{н}}=\frac{r_0^3}{\tau_{\text{к}}r_{\text{к}}^2}. \qquad (23)$$

The given method of determining $u_{\text{н}}$ cannot be used if the mixture is able to diffuse through the layer of the bubble or react with it, or when the mixture, under the effects of moisture, changes its characteristics.

The Apparent Rate of Flame Propagation

The apparent rate v is always as a rule greater than $u_{\text{н}}$ and is not a physico-chemical constant of the given mixture. Its value depends, for example, on the arrangement and diameter of the tube filled with a combustible mixture. Figure 27a schematically shows the process of flame propagation in the initial period of time, beginning from the moment of igniting the mixture at the open end. In the beginning a spherical propagation of the flame is observed. As it approaches the walls, its motion is slowed, the stored volume of hot products of reaction, having less density than the fresh cold mixture, strives to occupy the upper layers, distorting the surface of the flame ($S_{\text{пл}}$ and $S'_{\text{пл}}$). Finally, in position I the form of the flame is established fully determined for the given conditions (the diameter of the tube $d_{\text{тр}}$, the area of its transverse section $S_{\text{тр}}$), and further, the flame front moves practically without being distorted, with an invariable rate v on a specified section of the tube comprising approximately (1/5-1/4) of its length. If the volume of the mixture which burned down in a unit of time $V_{\text{см}}$ in cm³/s, then $v=\frac{V_{\text{см}}}{S_{\text{тр}}}$. On the other hand $u_{\text{н}}=\frac{V_{\text{см}}}{S_{\text{пл}}}$, i.e., $u_{\text{н}}=v\frac{S_{\text{тр}}}{S_{\text{пл}}}$.

With an increase of $d_{\text{тр}}$ the ratio $\frac{S_{\text{пл}}}{S_{\text{тр}}}$ increases in connection with the distortions of the front and autoturbulence. Obviously, with an increase in d_{TP} the quantity of mixture reacting in a unit of time will increase, and in accordance with this rate v will increase (Fig. 27b).

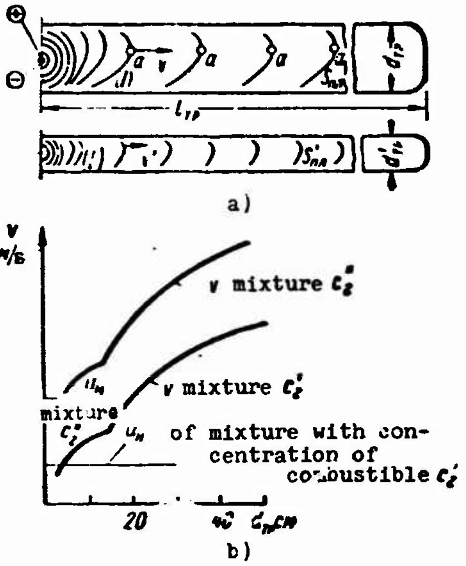


Fig. 27. The process of combustion of a mixture in a tube.

23. The Process of Normal Propagation of a Flame Front

The mechanism of normal distribution of a flame front is still not completely clear. However, this phenomenon can be explained in the following manner. In the first place, as is known, all combustion reactions are chain reactions and this means that a chemical conversion, because of the existence of active centers, proceeds through a series of intermediate compounds, diffusing from the zone of reaction into the fresh mixture and directly nascent in the mixture before reaction. Secondly, the combustion reactions are accompanied by the evolution of a great quantity of heat, during the transmission of which the conditions of the fresh mixtures are created which ensure an intensive course of the chemical reaction. It is logical to consider both these considerations together.

Today, the most widespread is the thermal theory of flame propagation, which is based on three basic equations: of thermal conductivity, of diffusion, and of chemical reaction, generally under conditions of a forced flow.

For simplicity let us examine a stationary two-dimensional flame front in a mixture which moves at rate $w = u_{\text{н}}$ towards the front (Fig. 28a). In the sketch, the width of the flame front δ is magnified; in reality it is equal to fractions of a millimeter. In the front, the fresh batch of mixture must be thoroughly heated (the zone of preheating $\delta_{\text{п}}$) and react chemically (zone δ_{x} - chemical, Fig. 28b). As a result of the strong dependence of rate of chemical reaction temperature, the reaction will basically proceed at a temperature close to $T_{\text{к}}$ (the end of the reaction). An intensive chemical reacting begins also at a high temperature $T_B \approx T_{\text{к}}$. From the initial temperature T_0 to T_B the fresh mixture is heated as a result of the molecular thermal conductivity of the zone of chemical reaction. The products of reaction themselves thus are heated insignificantly (from T_B to $T_{\text{к}}$). A stationary flow of the process is possible under a condition of equality of heat q_1 spent on heating the fresh mixture from T_0 to T_B to heat q_2 obtained from the zone of chemical reaction by thermal conductivity. Heat q_1 and q_2, relative to a unit of flame surface and divisible into a unit of time:

$$q_1 = c_p \rho_0 (T_B - T_0) u_{\text{н}}; \tag{24}$$

$$q_2 = \lambda \frac{dT}{dl}, \tag{25}$$

where c_p, ρ_0 and λ - respectively, heat capacity with p = const, density, and coefficient of thermal conductivity of the fresh mixture, considered constant in time; l - the length of the section being considered.

The part of the heat which goes to heating the reacting medium from temperature T_B to $T_{\text{к}}$ is usually disregarded, considering that the entire flow of heat from chemical reaction goes to heating the fresh mixtures.

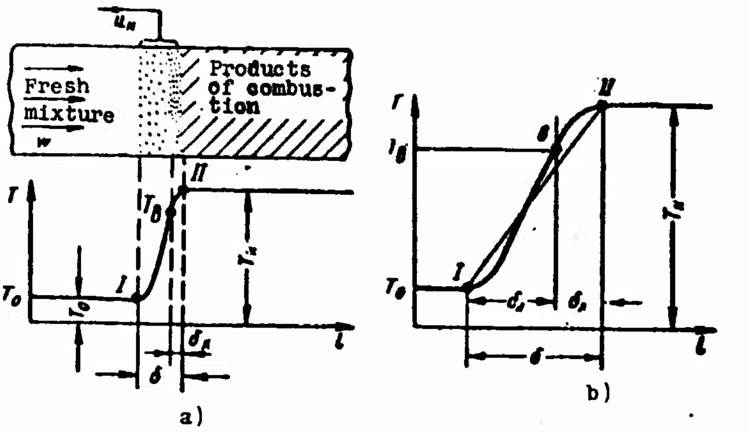


Fig. 28. Structure of a flame front during laminar combustion.

In this way, in expression (25) the derivative $\frac{dT}{dl}$ refers to the boundary of the zone of reaction. If, for simplicity, we take instead of the tangential line, the section, i.e., if $\frac{dT}{dl} \approx \frac{T_{\text{х}} - T_0}{\delta}$, then taking in expression (24) the value $T_{\text{в}} \approx T_{\text{х}}$, the normal rate $u_{\text{н}} = \frac{a}{\delta}$, where $a = \frac{\lambda}{c_p \rho_0}$ - the coefficient of thermal conductivity of the basic mixture. On the other hand, $u_{\text{н}} = \frac{\delta_{\text{х}}}{\tau_{\text{х}}}$, where $\tau_{\text{х}}$ - the time of the course of the chemical reaction.

The value of δ_x is less than δ, but connected with the latter by the character of the flow and the amount of thermal effect of the chemical reaction. Assuming that $\delta_{\text{х}} = \Phi(w)\,\delta$, where $\Phi(w)$ - a function, determined by the rate of chemical reaction, we obtain

$$u_{\text{н}} = \sqrt{\Phi(w)}\sqrt{\frac{a}{\tau_{\text{х}}}}. \qquad (26)$$

§ 24. Dependence of Normal Rate on the Parameters of the Combustible Mixture

Experimental data show that the normal rate of flame propagation u_H depends on the properties of the components of the combustible mixture, their relationship (concentration), on the pressure and temperature of the mixture. It is known that not stoichiometric mixtures (where $a = 1$) but mixtures with a certain insufficiency of air, when $a < 1$ (Fig. 29) have $(u_H)_{max}$. However, maximum heat release (and temperature) give stoichiometric mixtures. Increasing the temperature of the fresh mixture increases u_H (Fig. 30). Pressure acts unequally on u_H. For fuel-air mixtures, the increase in pressure, as a rule, reduces u_H (Fig. 31a). However, when $p < 1$ atm(abs.), the effect can be the reverse (Fig. 31b). If oxygen is used as the oxidizer in the mixture, then u_H for the majority of known mixtures practically does not depend on pressure (Fig. 31c). Impurities have considerable effect on u_H. Fig. 32 shows the change in the miximum value of u_H of gas-oxygen mixtures with the replacement of part of the O_2 by a corresponding part of nitrogen (dotted curve) or carbon dioxide (solid curve).

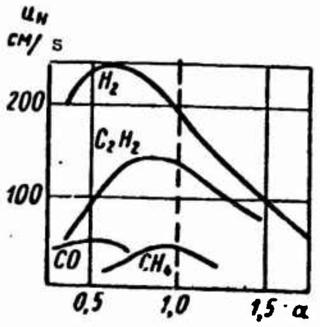


Fig. 29. Normal rate of flame propagation of gas-air mixtures, depending on the excess air ratio.

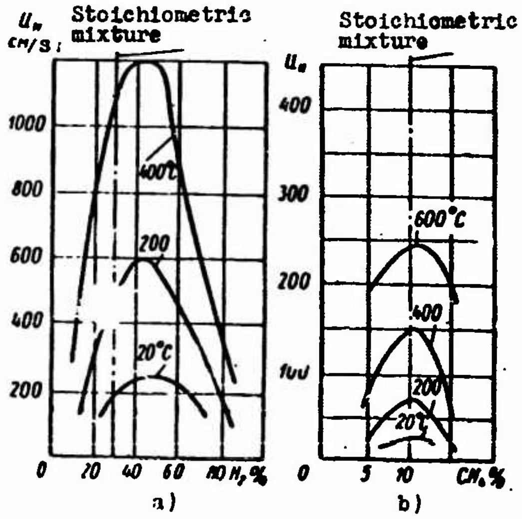


Fig. 30. Effect of the temperature of a gas-air mixture on the value of u_H for various combustible substances: a) hydrogen; b) methane.

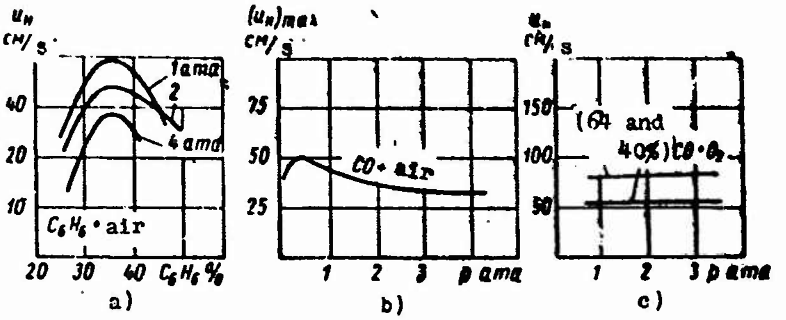


Fig. 31. Effect of pressure on value of u_H. [*ama* = atm(abs.)]

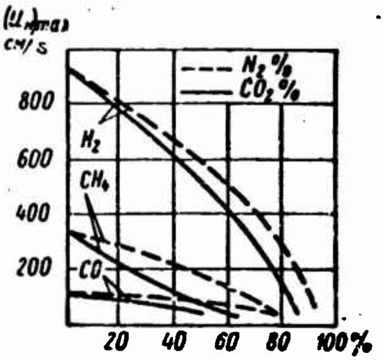


Fig. 32. Effect of impurities on value of u_H.

§ 25. Burnout Behind the Flame Front

The structure of the flame front has as yet been insufficiently studied. However, it can be considered that complete combustion of fuel never takes place in the flame itself. The zone of combustion always to one degree or another emerges beyond the point of the flame proper, forming behind the front a zone of burnout.

Figure 33 shows the burnout of a mixture of CO + air with change in initial pressure. When $p > 1$ atm(abs.) at a distance of fractions of a millimeter from the surface of the flame the mixture burns on the average to 80-90% and more. As an experiment shows, the greatest temperature is observed in the zone of burnout and not on the surface of the flame, the luminescence of which is determined basically by chemiluminescent radiation.

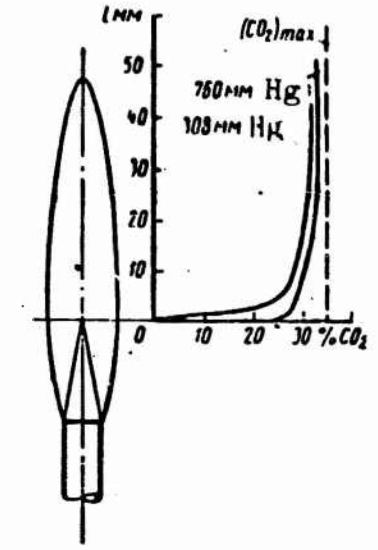


Fig. 33. Burnout behind the flame front.

§ 26. Flame Propagation in a Turbulent Flow

The process of flame propagation in a moving turbulent medium is of great practical interest. Unfortunately, this section of the theory of combustion is still insufficiently developed. The simplest example of such combustion is the flame of a gas burner with turbulent gas flow (Fig. 34a) in it. Here, just as during normal combustion, heat is transferred by thermal conductivity, which in principle permits relating turbulent combustion to a type of normal combustion. A substantial distinction consists in the fact that thermal conductivity in the turbulent medium is determined not only by molecular, but basically by molar mixing of elementary volumes. The same can also be said about diffusion. These processes here go on more intensively. Correspondingly turbulent combustion is characterized by a rate of flame propagation u_T considerably greater than normal u_H. Figure 34b shows the dependence of a ratio $\frac{u_T}{u_H}$ on the Re number (on the velocity of the gas in the burner) and the diameter of the burner. Figure 34c gives the character of dependence of u_T on the concentration of fuel in the mixture and the rate of flow w. As an experiment will show, during the combustion of mixture in a closed channel the rate of flame propagation in a turbulent flow u_T is defined by a series of values including the value of u_H of the given mixture, and also its by composition, pulsation rate of flow $\bar{v}$, by pressure, and temperature of the mixture, and others.

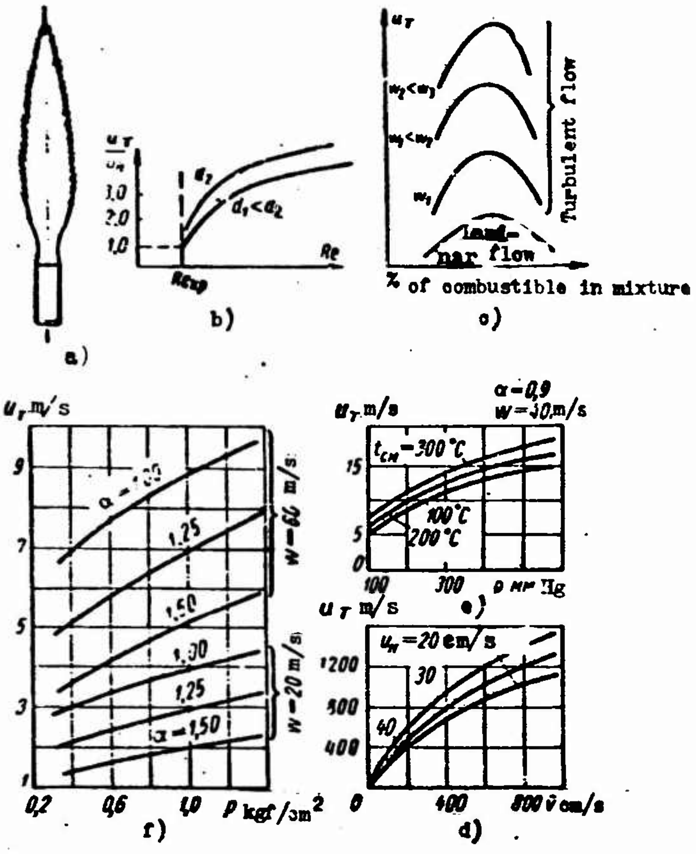


Fig. 34. Turbulent combustion.

Figure 34d gives the dependence of u_T on u_H and $\bar{v}$ for a gas-air mixture close to a stoichiometric composition. The combustible mixture here was city gas (a mixture of natural and coke gases). The experiment was performed at atmospheric pressure and a temperature of the mixture equal to 20°C. Figure 34e depicts the dependence of u_T on pressure at different temperatures of the mixture t_{CM} for a gasoline-air mixture. The effect of the composition of the mixture α on the value of u_T with a change in pressure and rate of flow w is shown in Fig. 34f.

The value of u_T is determined basically by the characteristics of the flow turbulence. As is known, in a turbulent flow the actual rate at each point continuously changes, ranging (fluctuating) around a specific mean value of w.

The amount of pulsation of the velocity is characterized by its root-mean-square deviation from the value of w after a certain interval of time.

The root-mean-square value of pulsation of the velocity of the flow depends on average velocity w. For flow in tubes they usually take $\bar{v} = k \cdot w$, where $k = 0.03$-0.10. The factor k depends on the conditions of flow, specifically, upon the roughness of the tube and so forth.

The most important characteristics of flow are: the path of mixing or the scale of turbulence l which characterizes the movement of the elementary volume of gas and the coefficient of turbulent exchange ε. Analogically to this, as the coefficient of molecular diffusion is proportional to the length of the path of free path of the molecules, so for turbulent diffusion, the value of ε is determined by the product $l \cdot |\bar{v}|$. In principle, two different cases of turbulent flame propagation are possible depending on the flow characteristics.

I. $l < \delta$ - the scale of flow turbulence is small. The value of l, in particular, is less than the width of the normal flame front δ. In this instance, the flame front is slightly distorted (Fig. 35a). Taking into account that $u_N \approx \sqrt{\frac{a}{\tau_x}}$ and considering the time of chemical reaction as invariable, it is possible to obtain the ratio of velocities

$$\frac{u_T}{u_N} \sim \sqrt{\frac{\varepsilon}{a}}$$

or

$$\frac{u_T}{u_N} \sim \sqrt{\frac{a + a_T}{a}} = \sqrt{1 + \frac{a_T}{a}},$$

where a - coefficient of thermal conductivity (molecular); a_T - pulsating component of the coefficient of turbulent exchange.

Since $a_T \gg a$, whereby $a_T \sim l \cdot \bar{v}$, then

$$u_T \sim u_N \sqrt{1 + \frac{l\bar{v}}{a}},$$

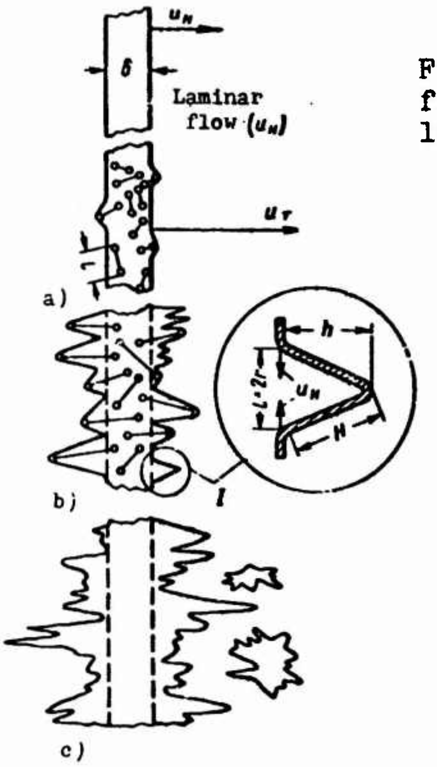


Fig. 35. Surface of flame front in laminar and turbulent flows.

i.e., with small-scale turbulence

$$u_т \sim \sqrt{\bar{v}} \sim \sqrt{\bar{w}} \sim \sqrt{\overline{Re}} \tag{27}$$

II. $l > \delta$ - the scale of turbulence is great. The flame front is distorted and jagged, and in individual cases ($l \gg \delta$ and $\bar{v} > u_н$) can even decompose into individual zones or throw independently burning volumes forward, (Fig. 35b and c).

Considering the projection of the flame front as an individual cone l (Fig. 35b), K. I. Shchelkin obtained that

$$\frac{u_т}{u_н} = \frac{s_{бок}}{s_{осн}} \sim \sqrt{1 + \left(\frac{\bar{v}}{u_н}\right)^2},$$

where $s_{бок}$ - lateral surface of the cone; $s_{осн}$ - area of the base of the cone.

If we designate $2r = l$, then the time during which the flame radially passes from the circumference to the center at a rate of u_H, $\tau \sim \frac{r}{u_н} = \frac{l}{2u_н}$. Because the pulsation rate $\bar{v}$, the flame in reality will describe a surface $S_{бок}$ and during time τ will move forward by an amount $h = \frac{l\bar{v}}{2u_н}$. The size of the forming cone

$$H = \frac{l}{2}\sqrt{1+\left(\frac{\bar{v}}{u_н}\right)^2},$$

and

$$\frac{S_{бок}}{S_{осн}} = \frac{\pi r H}{\pi r^2} = \sqrt{1+\left(\frac{\bar{v}}{u_н}\right)^2}. \qquad (28)$$

This means that with large-scale turbulence $u_т \sim \bar{v} \sim w \sim \mathrm{Re}$. Experimental research in general gives an expression of the type $u_т \sim \bar{v}^n$ and $u_т \sim w^n$, where $n < 1$. Furthermore, the value of u_T, naturally, depends on u_H, p, T (of the fresh mixture); however, especially with a great amount of turbulence, the determining factor is $w(\bar{v})$.

§ 27. Detonation Flame Propagation

During the combustion of a uniform combustible mixture in a tube, the apparent rate of motion of the flame in the course of time increases and sometimes can attain very high values which will not be changed any further. These maximum rates D, being measured in thousand of meters per second, define a special case - detonation combustion. Detonation propagation of a flame front is not possible in any mixture, but only in mixtures which possess large values of u_H. With initiation of detonation, the rate of the front changes abruptly.

The rate of detonation flame propagation D, as an experiment will show, depends very slightly on the conditions under which the process is carried out, and on the parameters of the mixture, in particular, on temperature and pressure. The value of *D* changes noticeably with a change in the composition of the mixture, where similar to the change in the rate of sound, the value of *D* increases with the drop in the density of the mixture. Additions of a gas to the mixture, which reduces its density, usually increase *D*. Figure 36 shows the character of the change in the magnitude of *D* of a mixture of $2H_2 + O_2$ depending on the addition of various inert gases and hydrogen. The density of these gases in a g/*l* respectively is: H_2 - 0.0899; He - 0.178; N_2 - 1.25 and Ar - 1.78. The density (stoichiometric) of the mixture of $H_2 + O_2$ equals 0.537.

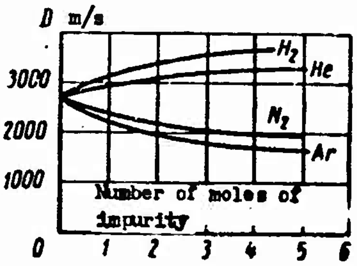


Fig. 36. Effect of impurities on the rate of detonation.

The length $L_Д$ on which a detonation front manages to be created from an ordinary flame depends on the pressure of the medium, the length of the tube, its diameter, initial temperature, the state of the surface of the duct, and others. The value of $L_Д$ increases with the growth in temperature, a decrease in *p*, and an increase in *l* and *d* of the tube. The formation of a detonation wave is very strongly affected by the roughness of the wall, reducing $L_Д$. If in a smooth duct $L_Д$ comprises usually 25-50 caliber, then by artificial roughening, this value can be reduced 5-10 times and more.

In a detonation wave, the parameters of the gas (temperature, pressure, density) are changed abruptly. It is possible to assume that upon detonation the mixture will be ignited by wave compression of the shock type. The formation of such a wave can be explained by the composition of waves (of sonic type) of the excited mixture, capable of chemically reacting with the evolution of heat. The source of these disturbances is the flame front, which moves with acceleration. This leads to the continuous generation of sound waves which overtake one another and encounter reflected waves, which under specific conditions leads to a shock wave. Continuous acceleration of the flame can appear as a result of expansion of the products of combustion, fluctuating processes, turbulization of the front due to the roughness of the walls of the duct, and so forth. Each of these factors magnifies the surface of the flame, the turbulilization of the media ahead of the front, i.e., there appears a change from u_H to u_T which, in turn, automatically accelerates the process. The theory of detonation combustion has not yet been developed sufficiently fully.

CHAPTER VIII

COMBUSTION OF FUELS

§ 28. Combustion of Gas Fuel

Into the working space of fuel burning devices, a gas fuel is supplied to special burners, on the outlet of which there is formed a cone of burning gas. Unlike the studied case of combustion when a combustible mixture (combustible + oxidizer) was intermixed beforehand in practice, as a rule, the combustible and an oxidizer are supplied separately to the burners. In so doing, depending on the character of their motion, one or another type of diffusion combustion takes place.

Laminar diffusion combustion

Let us imagine a case of efflux with low jet velocities of a combustible gas (not containing O_2) into a space filled with air. If we carry out ignition of the mixture, then combustion will continue because of the O_2 (ambient air entering into the stream of gas, i.e., combustion will proceed as a result of mutual molecular diffusion of O_2 and the gas. The mixture being formed can steadily and effectively burn down in that case when its concentration is close to stoichiometric ($\alpha = 1$). Therefore the surface of combustion determines the totality of those points in space where this condition is fulfilled. In circular burners the flame has a cone-shaped form (fig. 37a). The structure of the burning stream in section *A-A* is schematically shown in Fig. 37b. The stationary flow

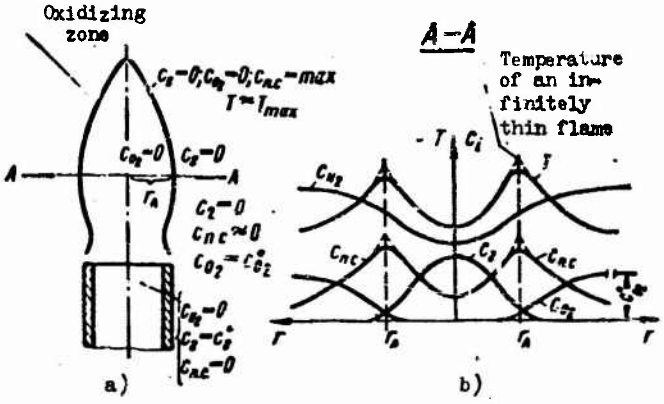


Fig. 37. Diagram of a burning jet of gas: a) concentration respectively; b) of gas; O_2 - of oxygen; N_2 - of nitrogen; n. c. - of products of a combustion; c^o - initial concentration of component.

of the process presupposes a continuous inflow of O_2 and gas to the flame front, the ignition of the mixture because of the heat of the chemical reaction which goes into the flame, and the diffusion of the forming products of combustion basically into the environment, but partially also inside the flame. Usually the rate of chemical reacting is immeasurably greater than the rate of diffusion, which determines the supply of the components necessary to the reaction. Therefore the process of combustion (and its rate) is basically determined by the rate of the physical process (the molecular diffusion of gases).

In individual cases, such a process of combustion can be described analytically. We will examine the combustion of a gas issuing from a central duct (Fig. 38) around which an oxidizer (oxygen) flows. For this case, using the simplifications which will be introduced below, the form of the surface of the flame as locus of points of space, possessing identical concentrations (where $\alpha = 1$) can be obtained from the equation of diffusion allowing for the consumption of the components of the mixture during chemical reaction:

$$\frac{\partial c}{\partial \tau} = D_c \Delta c \qquad (29)$$

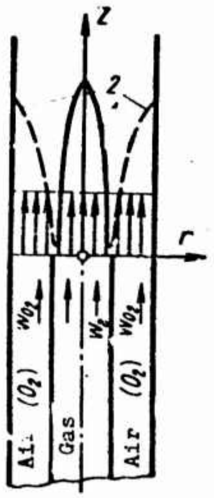


Fig. 38. Form of a flame during diffusion laminar combustion: 1 - when α > 1; 2 - when α < 1.

or

$$D_c\Delta c = \frac{\partial c}{\partial z}\cdot\frac{\partial z}{\partial \tau} = w\frac{\partial c}{\partial z},$$

where c - concentration; τ - time; D_c - gas diffusion, factor; Δ - Laplacian operator (the sum of the second derivatives by coordinates); w - velocity of the motion of a given component.

This equation can be formed both for the combustible gas and for the oxygen. For simplicity, let us take that the velocities and the diffusion factors of all gases are equal to one another. The consumption of each component during combustion is considered as a function ϕ_i. As a result, we obtain a system of two equations:

$$D_c\,\Delta c_z = w\,\frac{dc_z}{dz} + \varphi_z;$$

$$D_c\,\Delta c_{O_2} = w\,\frac{dc_{O_2}}{dz} + \varphi_{O_2}.$$

Since the gas and O_2 interreact, then naturally there is a connection between both values of ϕ_i of the type $\varphi_{O_2} = n\varphi_z$. The coefficient n represents the relationship of the corresponding stoichiometric coefficients in the equation of chemical reaction.

For example, for the reaction $2CO + O_2 = 2CO_2$ or $2H_2 + O_2 = 2H_2O$, the value of $n = 0.5$.

If from the first equation, having first multiplied it by n, we subtract the second, then, taking into account that $\varphi_{O_2} = n\varphi_z$, we obtain

$$D_c \Delta (nc_z - c_{O_2}) = w \frac{d}{dz} (nc_z - c_{O_2}). \tag{30}$$

In points where there is a stoichiometric mixture, there should be a flame, $nc_z - c_{O_2} = \bar{c} = 0$. In this case, the solution of the last equation gives $\bar{c} = f(z, r)$, i.e., the equation of the surface of an invariable concentration $\bar{c} = 0$ being taken as the flame front. A concrete solution in this simplest case is not simple despite very approximate assumptions, for example, about the instantaneous passage of the chemical reaction, as this was accepted in the derivation. It is obvious that the diffusion factors of the gases are not identical and that their values change with time and space in connection with the change of temperature etc.)

Turbulent Diffusion Combustion

This case of combustion is very important in practice. Let us examine this example. Air and gas are supplied separately, whereby the rate of air in comparison to the rate of gas is negligible, i.e., the jet of gas flows at high speed into motionless air. This is a case of a free flooded turbulent jet. A diagram of the jet is shown in Fig. 39. A jet expands because of the turbulent boundary layer, where a supplementary mass of air is drawn in from the environment. The initial section of the jet has a core where the velocity of the pure gas in identical and equal to the initial velocity in the opening of the burner. The concentration of the mixture as a result its of mixing in the boundary layer changes c_z c_z^0 $(c_{O_2} = 0)$ at the internal boundary to $c_z = 0$ $(c_{O_2} = c_{O_2}^0)$ at the external boundary of the layer. The field of velocities is similar to the field of concentrations. It is obvious that in each section along the length of the tongue of flame it is

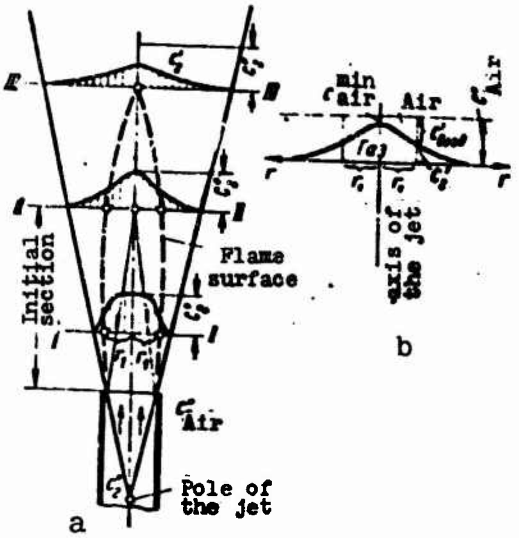


Fig. 39. Change in concentration of gas and air in a turbulent jet.

possible to distinguish the locus of points with r = const, where the mixture will be stoichiometric ($\alpha = 1$). Along the length of the jet we obtain correspondingly a surface which determines the position of the flame front. In any section, for example, a little higher than section II-II (Fig. 39b), it is possible to show together the change in the concentration of gas and air from $c_г$ on the axis $(c_{в.д})_{min}$ to $c_г = 0$ $(c_{в.д} = c^0_{в.д})$ at the external boundary. On a determined radius r_1, points will lie with concentrations respectively of gas and air $\dot{c}_г$ and $c_{в.д}$ corresponding to a stoichiometric mixture Similar points can be found in any section up to that distance from the end of the burner where at least on the axis it was $c_г = \dot{c}_г$. This extreme point defines the end of the tongue of flame. In Fig. 39 this point will be located on the axis in the section III-III. In this way the diagram of the development of a flame is analogous to the case of laminar flow; however, here the rate of the process is determined already by turbulent diffusion which considerably intensifies combustion.

As the experiment shows, with an increase in the velocity (volume expenditure) of the gas which issues from the burner, the boundary to laminar combustion advances there where the height of the flame is greatest. With further growth in velocity of flow vortex regions first at the apex of the flame, and then they extend even to its base. At first, turbulence is generated directly in the flame itself, even with a laminar flow of the jet of gas. This phenomenon is explained by the existence of a high gradient of velocities between the hot gases of the flame and the surrounding (motionless) air. With turbulent flow of a gas jet, the flame becomes completely turbulent; the height of the flame with the increase in velocity of flow is increased, but slowly.

The slight effect of the velocity of flow on the height of a diffusion turbulent tongue of flame is a characteristic feature of this type of combustion. The basic effect on the height of the flame here proves to be the type of fuel, whereby the greater quantity of theoretically necessary air that is required for complete combustion of a unit of the volume of this fuel, the greater the height of the flame. In the combustion of a gas in oxygen, the turbulent diffusion flame is considerably shorter than in air. The height of the flame can be decreased by the addition to the combustible gas which issues from the burner of a certain quantity of air (oxidizer), and also by increasing the intensity of the flow turbulence even with an invariable velocity of efflux. During turbulent diffusion combustion, a decisive factor is the process of turbulent mixing.

Experiments conducted on combustion chambers, showed that the length of the flame in them changes approximately according to linear law depending on the excess air ratio in the initial fresh mixture. Thus, for instance, with an increase in the ratio of the expenditure of fuel to the expenditure of air $\frac{G_T}{G_B}$ from 0.04 to 0.08, the comparative length of the flame in the combustion chamber is correspondingly magnified from 6 to 12 calibers.

With high velocities of gas flows, the possibility of steady combustion is determined by the rate of flame propagation. The latter observation touches upon a very important question - the stabilization of the flame front.

§ 29. Stabilization of the Flame Front

A stationary position of a flame front and a steady course of the process of combustion in a flow which moves with a velocity, exceeding the velocity of the flame propagation is possible only with continuous ignition of the mixture by any source. With cessation of ignition, the flame is torn off and taken away by the flow, and combustion in the given volume ceases. If an artificial source of ignition (a pilot burner, a constant electric arc discharge, etc.) is lacking then in order to ensure a stationary flame front in a flow, there should exist a zone where, one way or another, equality of the rates of motion of the mixture and the propagation of the flame front directed towards each other is obtained. This zone serves as the source of the firing from which the surfaces of the flame forming a stationary front will be propagated.

A concept regarding the mechanism of flame stabilization can be obtained most simply by considering the combustion of a pre-prepared uniform mixture, excluding the process of carburetion. Figure 40 shows a diagram of the initiation of a fixed source of ignition of a mixture issuing from a Bunsen burner in laminar fashion with velocity w. In contrast to the ideal case (see Fig. 38), the velocity of the mixture w at the wall of the burner, just as at the external edge of the boundary layer with escape into a motionless medium, will be equal to zero. It is obvious also that u_H in the tube will drop in proportion to its approach to the wall as a result of heat interchange. Velocity u_H will be reduced also in the boundary layer of the jet as a result of the decrease in the concentration of combustible in the mixture.

Combustion ceases not on the external boundary of the layer, but earlier, where the mixture becomes an incombustible.

In Fig. 40, where the curve of u_H is conventionally directed identically with the curve of w, it is evident that in section III curves w and u_H touch at points 1, constituting the circumference in the section of the jet. There will be an analogous position

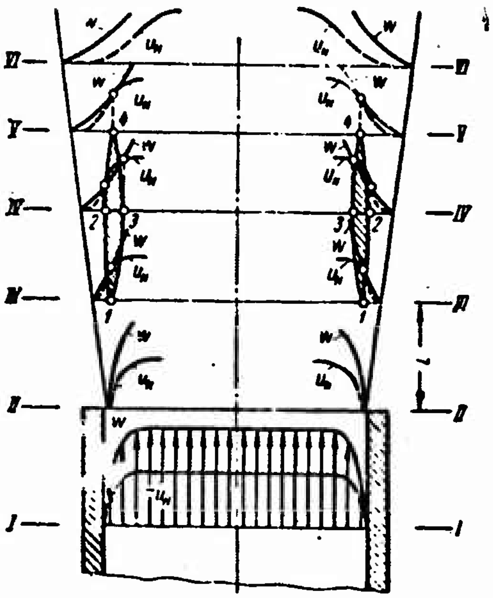


Fig. 40. Diagram of stabilization of a flame front.

in section V for points 4. In section IV the value of $w = u_{\text{H}}$ at two points 2 and 3, and between them $u_{\text{H}} > w$. This condition is preserved everywhere between sections III and V and it determines a specific area (the flashback zone, where $u_{\text{H}} > w$) limited by the surface 1-2-4-3-1 where $u_{\text{H}} = w$. Having carried out ignition at any point of the flashback zone, it is possible to obtain a stationary combustion of the mixture where it is obvious that the forward points of the flame surface and subsequently the points of ignition will be points 1, located a distance L from the mouth of the burner. Generally speaking, a stationary front can be obtained by igniting the mixture below section III since the flame removed by the flow wil proceed to points 1, where subsequently combustion will hold.

The mechanism of flame formation over the whole section of the tube in the ideal case of w = const can be explained by considering the motion of a spherical flame. Figure 41 shows the end of a burner and a characteristic section III, where the forward points 1 of the flashback zone are located. If the mixture does not move, then after ignition a flame from the points 1 will be extended spherically to the axis of the burner (the dashed arcs of circumferences); in so doing, the flame fronts will be joined at point A. Attention should be paid to the fact that as the front of combustion approaches

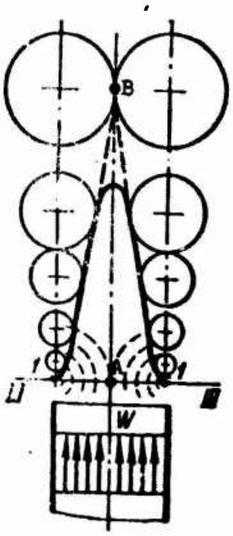


Fig. 41. Formation of the flame front of a gas burner.

point A the motion of the flame should be accelerated as a result of more intensive heating through the central volume and the corresponding growth of u_H. With motion of the mixture simultaneously with the expansion of the spheres, the latter are brought together by the flow and the flame front interlocks on the axis at point B. When w = const over the section of the burner and with constant u_H, the flame front should be a true fine-pointed cone. As a result of the growth of u_H at the apex of the flame (because of more intensive heating) and a drop in u_H at the base (because of the effect of the cold wall) in actuality the flame has curvature as is shown in the figure by the solid line. In the general case of $w \neq$ const, the form of the flame surface, naturally, is more complex and is determined by values of w and u_H in the volume of the mixture.

Figure 42 schematically depicts the mechanism of flame stabilization in a plane-parallel flow of a free turbulent jet. A combustible gas (fuel-air mixture) flows out into motionless air. The figure shows for a series of sections the character of the changein w, T, and u_T and the composition of the fuel-air mixture c, in the turbulent boundary layer. It is obvious that here also in a defined part of the boundary layer the formation of a flashback zone is possible.

It is known that $u_T = f(u_H, \bar{v}, l)$. Applying the theory of turbulent free jets, which determines the dependence of these values on

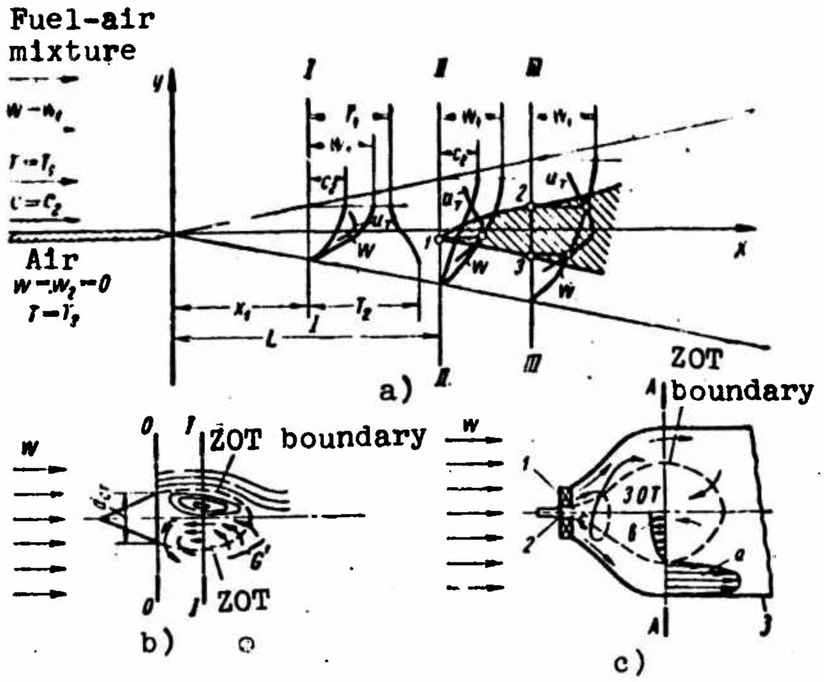


Fig. 42. Diagram of formation of the flashback zone: 1 - vane swirler; 2 - nozzle; 3 - flame tube of a combustion chamber.

coordinates, it is possible to rewrite this expression in the form

$$u_T = \varphi\left(\frac{y}{x};\ x\right). \qquad (31)$$

The temperature of the gas proves to have great significance for the position of the flame front, and particularly, for the value of *L*. With an increase in temperature *T*, *L* is reduced. It is obvious that a change in one or several parameters (*w*, *T*, the excess air ratio α, etc.) The position and form of the flashback zone and also distance *L* will be changed. Let us note that with a sharp change in the parameters of flow, disruption of the flame is possible.

In GTD combustion chambers, stabilization of the flame front is carried out by means of poorly streamlined bodies (plates, cones, etc.) Figure 42b gives a diagram of the flow flowing around a cone, behind which a zone appears with reduced pressure (outlined by the dotted line). The pressure drop deflects part *G'* of the basic flow into this zone, whereby the flow is directed counter to the basic flow (so-called reverse current zone - ZOT). The dimensions of this zone depend on the size of the stabilizer $d_{\text{ст}}$. The same effect can be obtained by swirling the flow with a vaned apparatus in the intake to the combustion chamber (Fig. 42c). The mechanism of flame stabilization behind poorly streamlined bodies is similar to that diagram examined earlier (Fig. 42a). In this case, from section 0-0 ot *I-I* (Fig. 42b) on the external surface of the reverse current zone a boundary layer is formed which can approximately be considered as a plane-parallel boundary layer of a free jet. Into GTD combustion chambers, as a rule, a ready combustible mixture is not introduced but individual currents of air and fuel which doubtlessly changes and complicates the mechanism of the process.

§ 30. Combustion of Liquid Fuel

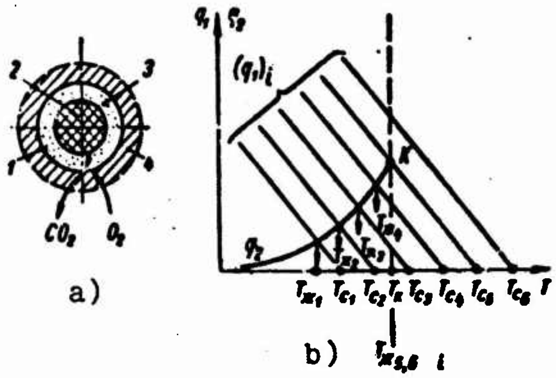
Into GTD combustion chambers and also into other fuel-burning devices, a liquid fuel is introduced into a current of air in finely divided state. Ignition of the liquid fuel is possible only after the formation of a combustible vapor-air mixture around its surface, and combustion is carried out only in the vapor phase. The mechanism of the process of combustion of a drop of fuel is as follows. In an oxidizing medium heated above the temperature of self-ignition, the drop begins to vaporize. As soon as there is formed about the surface of the drop a combustible vapor-air mixture, the latter self-ignites. Then combustion continues because of the subsequent continuous formation of a combustible mixture as a result of fuel evaporation. At the appearance of flame, vaporization, and that means also combustion, is intensified since the temperature is elevated.

The established process of combustion is determined by the equality of the rates of the processes of fuel evaporation and the combustion of the vapor-air mixture around the surface of a fluid. However, depending on which of these two processes is the more intense, the character of combustion will be changed. For example, if the rate of oxidation of fuel under the given conditions is very great in comparison with vaporization, then, naturally, the rate of the total process (combustion) will be determined only by the rate of the process of vaporization. This means that for intensification of combustion, in this case, it is useless to increase the rate of the first process, which is thus proceeding rapidly enough. In this case, the only measures which can help are those which accelerate the process of vaporization, which limits combustion. An opposite case is also possible. Furthermore, it must be taken into account that apart from these two most important processes, there is also significance, for example, in the processes of diffusion, which determine the mixing of the fuel vapors with the air and the removal of the products of reaction from the zone of combustion, and also the process of heat interchange. Consequently, the characteristics of the process of combustion of liquid fuel in a homogeneous vapor-air medium reflect the rules not only of the chemical process, but also the physical processes, including the processes determined by the interaction of the media which are in various states of aggregation (inhomogeneous processes).

We will examine the process of combustion of a drop of liquid fuel in motionless hot air (Fig. 43a). A cold drop with temperature $T_{ж}$, getting into an oxidizer medium which possesses a high temperature $T_c \gg T_{ж}$, begins to rapidly be heated thoroughly from the surface and to vaporize. Heat is transferred basically as a result of thermal conductivity evenly over the surface through the relatively thin layer of vapor, which diffuses radially from the surface of the drop. In the extreme case, when combustion does not limit the process and does not determine the thermal conditions of the system, the process of a vaporization acquires special significance. If the overall pressure does not exceed the critical, then the removal of vapors from the drop at a temperature lower than the boiling point proceeds very rapidly, and the partial pressure at the surface does not limit vaporization. In this way, it can be approximately considered that the rate of vaporization depends only on temperature. Thus, according to Arrhenius' exponential dependence

$$w = C \cdot e^{-\frac{L}{RT_{ж}}}, \tag{32}$$

where L - latent heat of vaporization; C - a constant which is determined by the kind of liquid.



a) b)

Fig. 43. Diagram of combustion of a drop of liquid fuel: 1 - combustion surface (zone); 2 - liquid; 3 - vapor layer; 4 - zone of afterburning and diffusion of O_2 and products of combustion.

In the first period of heating, the temperature of the drop increases to a certain value, whereupon the elevation in temperature ceases, and all the heat entering a unit of the surface of the drop is expended in vaporization. The quantity of a conducted heat

$$q_1 = \alpha(T_c - T_ж). \tag{33}$$

In this case equality $q_1 = q_2$ observed, where $q_2 = wL$ - heat being expended on the process of vaporization. Hence it is possible to determine the temperature of vaporization equilibrium $T_ж$. Figure 43b shows the character of the dependence of values of q_1 and q_2 on temperature. With an invariable value of the heat transfer coefficient α, the system of straight lines of q_1 will be equidistantly shifted depending only on the change in temperature T_c. The intersection of a given straight line q_1 and curve q_2 gives the value of $T_ж$. With an increase in T_c the difference $(T_c - T_ж)$ is increased despite the growth of $T_ж$ which is limited by the boiling point $T_к$. In this way an increase in T_c, as follows from equation (33), leads to an increase in the rate of vaporization.

The quantity of heat supplied to the surface of the drop which possesses at the given moment a radium r for a time $d\tau$,

$$d\dot{Q} = 4\pi r^2 \alpha (T_c - T_{\text{ж}})\, d\tau. \tag{34}$$

Because of this heat it will be possible to evaporate a quantity of liquid

$$dG = \frac{dQ}{L}; \tag{35}$$

$$dG = -4\pi r^2 \frac{1}{v_{\text{ж}}}\, dr, \tag{36}$$

where $v_{\text{ж}}$ - specific volume of liquid.

Then

$$\frac{4\pi r^2 \alpha (T_c - T_{\text{ж}})\, d\tau}{L} = -\frac{4\pi r^2}{v_{\text{ж}}}\, dr.$$

Hence it is possible to find either the value of $\frac{1}{v_{\text{ж}}} \cdot \frac{dr}{d\tau}$, characterizing the mass vaporization rate (combustion in this case), or the time of complete combustion of a drop:

$$\tau_0 = \frac{1}{v_{\text{ж}}} L \int_0^{r_0} \frac{dr}{\alpha (T_c - T_{\text{ж}})}\ \text{сек.} \tag{37}$$

If, for simplicity we take it, that $T_{\text{ж}} \approx T_{\text{к}} = \text{const}$, then

$$\tau_0 = \frac{\frac{1}{v_{\text{ж}}} L}{T_c - T_{\text{к}}} \int_0^{r_0} \frac{dr}{\alpha}. \tag{38}$$

The value of α is determined by the conditions of heat interchange during the motion of a drop in space. In the simplest case, in the absence of relative motion of the drop in the surrounding medium $\alpha = \frac{\lambda}{r}$, where λ - the coefficient of thermal conductivity of the gaseous medium. Then

$$\tau_0 = r_0^2 \frac{L}{2\lambda(T_c - T_к)\rho_ж}. \tag{39}$$

Under actual conditions, the diagram of the process, naturally, is complicated, however the effect of basic factors remains invariable; the time of a combustion is reduced considerably with the decrease in the radius of the drop and with the growth in the temperature of the medium. Furthermore, it also depends on a number of physical parameters.

In GTD the fuel is supplied to the combustion chamber by nozzles which atomize the liquid to the most minute drops with a diameter of from 20 to 200-300 μm. In view of the fact the fuel evaporates and its vapors are combined with air in the combustion chamber itself, the process of combustion obeys the laws of diffusion combustion.

§ 31. Combustion of Solid Fuel

The combustion of solid fuel is broken down into a number of stages: preheating, pre-drying, evolution of volatile substances and the formation of coke, the combustion of volatile substances and the partial gasification of coke, and the combustion of coke. The determining one is the combustion of the solid coke residue - carbon, since the portion of carbon, for example, in coal is usually more than 90% (by weight) and correspondingly for the portion of burned down carbon up to 90-95% (of the sum) of heat of combustion of all combustible elements is necessary. Furthermore, this the most prolonged stage in time; up to 90% of the whole time of the process is spent on it. At the same time, the presence of a bituminous part and the formation of volatile substances has great significance for the course of the process which is distinguished by the diversity of individual phenomena. A piece of coal introduced into highly heated air, after a certain time is enveloped by the shell of flame, although its temperature does not exceed 700-800°C. This flame given off from the coal and the ignited volatile substances, which, naturally, contribute to the further heating up and ignition of the heavier part of the mixture - the coke residue. If the piece is large, then flame may disappear after burning the mass of volatile substances. The yield of volatile substances completes the first stage (the period of preparation of the combustion of carbon) which proceeds with the expenditure of the solid supplied for pre-drying and initial heating of the fuel. This distinguishes the preparatory stage from the basic stage - the combustion of coke, which proceeds independently because of the great evolution of heat.

The change from the first preparatory stage to the basic one and the steady combustion of coke is not always possible, but only under specified conditions. To do this it is necessary that the temperature of the combustion zone be not lower than a certain value, which determines ignition and further combustion of carbon.

In this way, for solid fuels two characteristic temperatures exist and correspondingly, two processes of ignition - of the volatile substances and the coke residue, which are determined by the conditions of heat interchange and the reaction properties of the coke. It is obvious that the ignition temperature of coke determines the beginning of the steady combustion of solid fuel on the whole. The beginning of combustion of coke is accompanied by rapid utilization of O_2 from the adjacent gas layer and its substitution by products of combustion, which slows the process and even curtails it if provision is not made for the admission of O_2 from the following layers of air because of diffusion. Therefore, the elementary diagram of the combustion of carbon predetermines the equality of the rate of combustion expressed through the expenditure of oxygen, and the rate of admission of O_2 from the medium, equal to the amount of diffusion flow

$$q_{\text{д}} = -D_c \frac{dc}{dn},$$

where D_c - diffusion coefficient; n - a normal to the surface of a particle.

The intensity of diffusion depends on the character of motion of the medium, the concentration of O_2 at the surface, and the reaction capability of the coke. The intensity of the oxidizing reactions depends mainly on temperature. The consecutive stages in the process of combustion of solid fuel determine the dependence of the overall rate of this complex process on the rate of the course of the individual stage. In this case, the overall rate will always be less than the rate of the course of the slowest stage. To intensify combustion on the whole, it is thus necessary to accelerate the limiting one, the process proceeding most slowly. Equality in the rates of chemical reacting and the diffusion flow cannot always be effected. Thus, at reduced temperatures, the kinetics of the chemical reactions can limit the process of combustion. This range of temperatures will determine the so-called kinetic range.

At high temperatures the reverse picture is possible - the chemical reaction proceeds practically instantaneously, and the supply of O_2 limits its course. In this instance, the so-called diffusion range appears in which the determining factor will be the process of diffusion. The rate of diffusion can be increased, for example, by increasing the relative rate of flow or by reducing the size of the particles.

Steady-state operation of combustion presupposes equality in the quantity of O_2 reaching a particle as a result of diffusion and being consumed in a chemical reaction, with a constant concentration of O_2 on the surface of the particle c_p. This concentration is the greater, the greater the rate of diffusion, and in the limit strives toward a concentration of O_2 in the flow of c_0. The case when chemical reacting proceeds only on the surface of a particle, excluding the internal porous surface, is the simplest to analyze. The rate of a stabilized process of combustion of carbon can be determined according to the rate of the chemical reaction. If we consider that on the surface of a particle, reactions of the first order proceed, then the rate

$$w_s = k \cdot c_p \qquad (40)$$

must be equal to the specific flow of O_2 being delivered by diffusion to the surface of a reactive particle. This flow, allowing for convection, is determined from the equation

$$w_{\text{д}} = \alpha_{\text{д}} (c_O - c_F). \tag{41}$$

By analogy with the process of heat interchange, a factor $\alpha_{\text{д}}$ is introduced here - a constant rate of diffusion. It represents the ratio of diffusion flow reaching a unit of surface to the difference in concentration of O_2 (in the volume and on the surface). Since $w_c = w_{\text{д}}$, then

$$w_c = \frac{c_O}{\frac{1}{\alpha_{\text{д}}} + \frac{1}{k}}. \tag{42}$$

The value of $\frac{1}{\frac{1}{\alpha_{\text{д}}} + \frac{1}{k}} = k_s$ is called the apparent constant of rate of combustion of carbon. The reciprocal $\frac{1}{k_s} = \frac{1}{\alpha_{\text{д}}} + \frac{1}{k}$ determines the overall resistance of the system in the process of combustion because of the supply of O_2 (physical resistance) and also because of chemical reaction (chemical resistance). It is obvious that in the diffusion range $\frac{1}{\alpha_{\text{д}}} \gg \frac{1}{k}$ and, disregarding the comparatively small value of $\frac{1}{k}$, it can be considered that the rate in this case $(w_c)_{\text{д.о}} = \alpha_{\text{д}} c_O$, in the kinetic range $\frac{1}{k} \gg \frac{1}{\alpha_{\text{д}}}$ and, considering only chemical resistance, $(w_c)_{\text{к.о}} = k c_O$. It is natural that in the intermediate range, where the rate of admission of O_2 and the rate of chemical reaction are commensurable in value, it is necessary to take into account both factors and to use equation (42). Practically, combustion of solid fuel is characterized not by the value of w_c, but by the specific surface rate of combustion K_s^c, determining the quantity of carbon which burns down per unit of surface of a particle in a unit of time. Obviously,

$$K_s^c = \frac{1}{\varphi} w_c. \tag{43}$$

where ϕ - expenditure of O_2 per unit of weight (mass) of spent carbon.

A number of ways are suggested for determining the value of $\alpha_д$. In particular, by analogy with the process of heat transfer, there is introduced a diffusion Nusselt number $\mathrm{Nu}_д = \frac{\alpha_д \cdot d}{D}$, then

$$K_s^c = \frac{1}{\varphi} \frac{c_0}{\frac{1}{k} + \frac{d}{D\,\mathrm{Nu}_д}}, \tag{44}$$

where D - diffusion coefficient.

Unfortunately, in most cases it does not seem possible to use the presented relationship for practical calculations in view of the lack of dependence of the values entering into it on the conditions of conducting the process.

It is also obvious that the assumption about the presence of only surface reacting and about the lack of reaction on the interior surface of a porous particle in general is wrong, especially, if at low temperatures c_F is close to the value of c_0. In this instance, it is necessary to correct the value of $\overline{K_s^c}$. Furthermore, the mechanism of chemical combination of C and O_2 has thus far been studied insufficiently. It has been clarified that it is carried out through a number of stages in the course of which intermediate compounds of the type C_xO_y, simultaneously giving oxides CO and CO_2.

3. COMBUSTION CHAMBERS OF GAS TURBINE ENGINES

CHAPTER IX

STRUCTURAL DESIGNS AND OPERATION OF COMBUSTION CHAMBERS

§ 32. Requirements Imposed on Combustion Chambers

The combustion chambers of gas turbine engines [GTD] (ГТД) should have:

1. A high coefficient of fuel combustion efficiency $\eta_{\text{г}} = Q_{\text{I}}/Q_{\text{II}}$, where Q_{I} – the quantity of heat given off in the working volume of a chamber for a unit of time and spent on heating the working substance; Q_{II} – the complete quantity of heat which theoretically could be given off with complete fuel combustion.

Modern combustion chambers have $\eta_{\text{г}}$ for the whole range of working loads usually no lower than 0.95-0.99, and at rated conditions $\eta_{\text{г}} = 0.98\text{-}0.99$.

2. Small stagnation pressure losses in the chamber

$$\sigma_{\text{в}} = \frac{\Delta p^*}{p^*_{\text{в}}} \cdot 100 = \frac{p^*_{\text{в}} - p^*_{\text{г}}}{p^*_{\text{в}}} \cdot 100\,\%,$$

where $p^*_{\text{в}}$ – total pressure of air on intake to the combustion chamber in [atm(abs.)]; $p^*_{\text{г}}$ – total pressure of gases on outlet from the chamber in [atm(abs.)].

The value of δ, as a rule, does not exceed 20-30%. Usually they do not strive to reduce this exponent maximally, but secure a defined profile of gas temperatures according to blade height, in order to equalize the stresses over its length by elevating the temperature of the gas at the periphery of the blade.

5. Rapid and reliable starting, and also steady operation of the chamber under various conditions and in various modes. This is especially important for aircraft GTD, where there is required reliability in starting and in operation at considerable flight speeds at high altitudes and with reduced air temperature and pressure on intake to the chamber. These chambers must frequently operate reliably on an impoverished mixture to $\alpha_{\Sigma} = 40-70$ and more instead of $\alpha_{\Sigma} = 2-8$.

6. A sufficient chamber service life and convenient and safe servicing of it. It is very important also that assembly and disassembly of the chamber be rather simple and not require much time.

These chief general requirements in each concrete case for a specific type of GTD are supplemented by a number of specific requirements. For example, for stationary GTD it is important to have comparative cheapness in cost of materials and in the manufacture of the chamber; a capability of its operation on several fuels, for example, on gas and liquid or on light and heavy liquid fuels, and so forth.

§ 33. Basic Principles of Organization of the Operation of a Combustion Chamber

The elements of the theory of combustion examined above and those generalizations which were made on the basis of experimental materials, permit reaching a number of important conclusions which determine the following basic principles of design of combustion chambers:

1. The division of the combustion chamber into a combustion zone and a mixing zone. The temperature of gases before the turbine

t_{Γ} = 750-950°C is limited by the strength and service life of the materials of the parts of its blading (mainly the blades). In individual cases t_{Γ} does not exceed 600-700°C. Not one fuel will burn properly in combustion chambers under such conditions. Therefore for the combustion of the fuel it is necessary to separate only the part of the air which goes through the chamber, ensuring conditions for the formation of a reactive mixture and a high temperature of the process. This part of the air (primary air G_{I}) is directed to a specially separated volume of the combustion zone in the flame tube, where the fuel is introduced. The remaining part of the air (secondary air G_{II}), by passing the combustion zone, is directed through special openings to the mixing zone of the flame tube (Fig. 44). Here this air is mixed with the products of combustion emerging from the combustion zone, whereby the assigned gas temperature before the turbine is ensured.

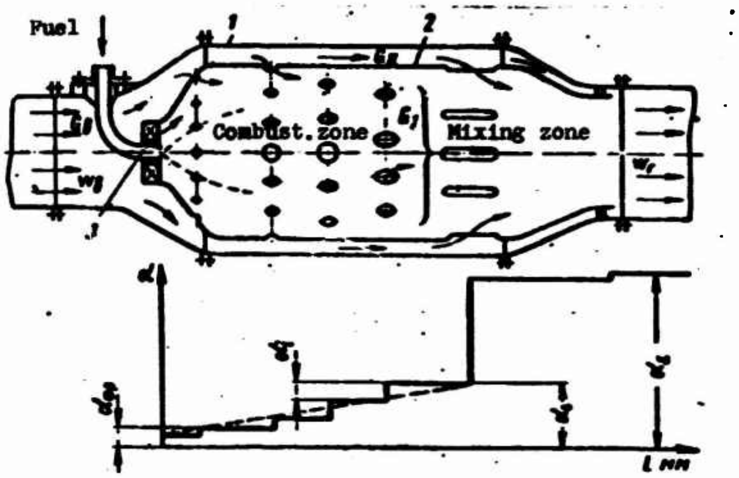


Fig. 44. Diagram of a combustion chamber and the distribution of primary air: 1 – casing; 2 – flame tube; 3 – nozzle.

2. Gradual (stepwise) supply of primary air along the length of the combustion zone. This is necessary because in the beginning for a fuel spray only a relatively small quantity of air is necessary (for combustion, it is rapidly vaporized into fine drops).

The value Δp^* takes into account three types of losses: a) loss to friction; b) turbulent losses (loss to shock in the diffuser, to the mixing of flows; losses depending on local drag, for example, in the front device because of local circulation flows, etc.); c) loss of pressure during heat supply. On the average for modern designs of GTD combustion chambers, the value of $\Delta p^*/p_{\text{в}}^* \cdot 100 = 1\text{-}3\%$; for aircraft GTD combustion chambers this value sometimes reaches 10%.

3. Small dimensions, i.e., the working volume should withstand high thermal loads relative to pressure $p_{\text{в}}$ on intake to the chamber. Thermal loads are characterized by combustion intensity of the working volume $H = Q_{\text{I}}/V_{\text{к}} p_{\text{в}}$, whereby

$$Q_{1} = G_{\text{т}} Q_{\text{н}}^{\text{р}} \eta_{\text{к}} \text{ kcal/h},$$

where $V_{\text{к}}$ – working volume of the chamber (flame tube) in m^3; G_{T} – hourly fuel consumption in kg/h.

In the combustion chambers of various classes of GTD the value of H fluctuates from $3 \cdot 10^6$ to $100 \cdot 10^6$ kcal/($m^3 \cdot h \cdot$[atm(abs.)]) and more. It is obvious that a decrease in the dimensions of the combustion chamber is also accompanied by a reduction in the weight (mass) of the construction, however in this case the pressure losses because of the increase in the rate of flow usually increase.

4. A prescribed field of gas temperature on outlet from the combustion chamber or a limited variation in the field of gas temperatures. This requirement ensures long service life and strength of the blades of the gas turbine. The variation of the temperature field is characterized by the exponent

$$\delta = \frac{T_{\max} - T_{\min}}{T_{\text{ср}}} \cdot 100\%.$$

where $T_{\max}$ and $T_{\min}$ – greatest and least temperature of the individual jets of gas in °K; T_{cp} – average temperature of gases on outlet.

Further, as the fuel is prepared, the following portions of primary air are required up to a theoretically necessary quantity, and in practice even exceeding the stoichiometric quantity. The excess air is necessary for facilitating the conditions of carburetion and for prevention of chemical incomplete burning of fuel, and also to decrease dissociation which appears at a high temperature of combustion. The optimum distribution of primary air along the length of the combustion zone can be finally established usually only during experimental final adjustment of the chamber on a test stand. Initially, during planning, usually a defined correlation of air distribution is assigned, frequently linear, as shown in the graph by the dotted line (Fig. 44). In this instance, the value of G_{I}, and that means also $\alpha_{\text{I}} = \alpha_{\Gamma} = G_{\text{I}}/L_0 G_{\text{T}}$, where G_{T} in kg/s – the fuel consumption depends on the type of the combustion chamber, the type of fuel, and the organization of the working process.

It is necessary that the average temperature of the gases in the combustion zone be equal to 1600-1900°C.

3. Turbulization of flow in the combustion zone. This is necessary for intensification of processes of mass- and heat-interchange, improvement of carburetion and an increase in the total rate of combustion because of the transition from laminar to turbulent combustion.

The flow is agitated by means of bladed swirlers (registers), poorly streamlined bodies, perforated plates, etc., which are installed in the forward, front part of the chamber (front device), and also by radial input of the airstream through openings in the walls of the flame tube.

4. Stabilization of the flame front in the combustion zone. The considerable boosts of the combustion chambers usually determine the average rate of flow, considerably greater than the rate of flame propagation u_{T}. Therefore, to retain the flame in the prescribed area of the combustion zone special measures are necessary. In the combustion chambers swirlers or the poorly streamlined bodies of the

front device serve as stabilizing agents. The zone of reverse currents installed behind these devices with reduced static pressure on the axis of the chamber, which is caused by the ejection of gas in a circular jet which issues from the register, and by centrifugal effect stabilizes the position of the flame, which ensures ignition of the entire fuel-air mixture. This is facilitated by the diffuser part of the front device. A diagram of the change in axial velocities in various sections along the length of the combustion zone of a chamber is given in Fig. 45a. Radial and tangential velocities will depend on the design of the front device and the parameters of flow, in particular, when using bladed registers – upon the blade angle to the axis of the chamber. This angle ϕ is usually taken equal to 45-65° (Fig. 45b). With further increase in it, the useful effect is not justified by the additional pressure losses.

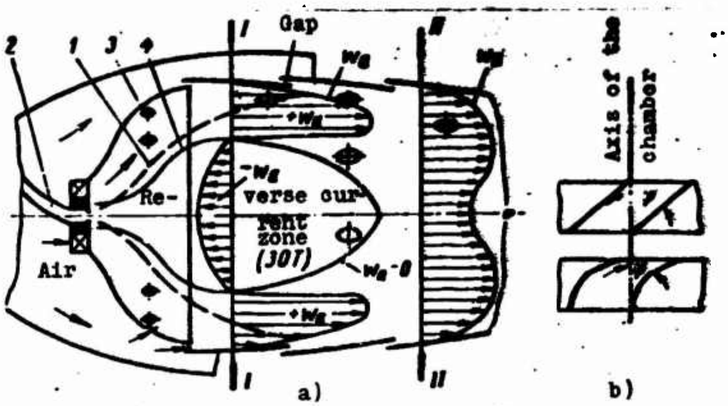
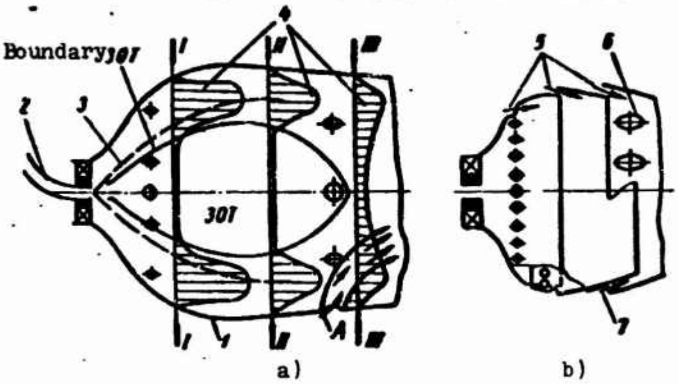


Fig. 45. Diagram of gas flow in the combustion zone: 1 – fuel cone; 2 – nozzle; 3 – openings; 4 – boundary of the reverse current zone.
[ЗОТ = ZOT = reverse current zone].

5. Optimum distribution of atomized fuel (liquid or solid) over the section of airflow without the fuel particles getting on the walls of the flame tube. It is expedient to supply a hollow conical flame of fuel in the area of flow adjacent to the outside of the zone of reverse currents, where the gradient of velocities will be maximum (Fig. 46a). This ensures the good mixing of the fuel with air which is necessary for realization of complete combustion.



a) b)

Fig. 46. Change in fuel concentration over the volume of the combustion zone: 1 – flame tube; 2 – injector; 3 – basic fuel cone; 4 – fields of local fuel concentrations; 5 – cooling air; 6 – opening for primary air; 7 – cooling air goes through the openings into the circular slot.

6. The organization of the cooling of the basic parts of the combustion chamber. As a rule, the air entering the chamber serves as a cooling agent. The flame tube with the front device and nozzle is heated most of all. For cooling the nozzles air is blown, passing between the housing of the nozzle and a tube located in the center of the swirler. This also prevents the coking of fuel as it leaves the injector nozzle. The front device and the flame tube are cooled by the secondary air which passes further into the flame tube past a series of fine openings or circular slots (Fig. 46b) located in several belts along the length of the flame tube. The circular slots which are shown in the upper part of the figure protect the walls more effectively, than the openings (by a continuous veil), however they let pass a considerable quantity of air, increasing α_{Γ}. This air lowers the temperature of the combustion zone without taking part in oxidation of the fuel, since near the walls there i hardly any fuel. To eliminate this deficiency, the entrance to a circular slot is guarded by a wall with a number of fine openings which apportion the air, but also ensure the formation of a circular veil moving along the wall at a slower rate, but wider. As an experiment showed, the thickness of the layer of air h determines the length of the effective cooling of the wall l. As a rule, $l \simeq (15\text{-}45)h$ and more and depends on the parameters of the air and other factors. Apart from interior

(film or jet) cooling, the flame tube is cooled outside by secondary air going to the mixing zone. In some designs of combustion chambers with small thermal loads the external flow of air proves to be sufficient for the cooling of the flame tubes of the combustion chambers. In modern highly stressed chambers, this flow is quite insufficient; moreover, it is even necessary to protect the housing of the chamber from thermal pressure from the side of the flame tube. Therefore, screens are frequently placed between the flame tube and the housing.

The chief principles mentioned of the design and organization of the working process of GTD combustion chambers for any purpose usually are supplemented by a number of considerations, depending on the specific characteristics of the equipment. In so doing, individual measures must be coordinated in order to satisfy the sometimes inconsistent general and special requirements imposed on the combustion chambers. Thus, for instance, for more effective cooling of the walls it is advisable to increase the rate of flow of the secondary air. However this is limited by the growth in pressure losses. In carrying out uniform distribution of the supply of primary air along the length of the combustion zone (streams *A* in Fig. 46a) one must not forget about the necessity to maintain a stable position for the reverse current zone to ensure ignition and combustion of the fuel. If, for example, powerful jets of primary air are introduced into the beginning of the combustion zone, then they decrease the volume of the reverse current zone or will even disrupt the steady course of the operating process.

§ 34. Classification of Configurations of Combustion Chambers

The type of construction and the operating conditions of combustion chambers depend on the type of fuel, the purpose of the GTD, its configuration, and also on the parameters of the cycle of the plant. The combustion chamber most frequently is built directly into the design of the engine, but in a number of cases it is located near it and linked with the compressor and the turbine by manifolds.

In the first case, the chamber is called built-in, in the second – remote. Remote chambers (Fig. 47a) are most frequently used in stationary gas turbine plants and rarely used on transport types (locomotive and vehicular gas turbine power plant [GTU] (ГТУ)). According to structural execution, chambers can be of the following types:

1) annular (Fig. 47b);

2) cannular (Fig. 47c);

3) sectional[1] (Fig. 47d);

4) individual (Fig. 47e).

The first three designs of designs are, as a rule, built-in chambers. Thus, the annular chamber, which is the most widely used in aircraft GTD as a result of its compactness and lightness, in the simplest GTD layout (compressor, chamber, and turbine), is located between the compressor and turbine, around the shaft of the turbocompressor (Fig. 47f). The working volume of the chamber is a continuous annular space between internal and external flame tubes. In a cannular chamber, just as in the annular, – have common internal and external casings. The flame tubes here are executed in the form of individual ducts; the flows of gas emerging from them are united in a circular gas collector directly before nozzle box of the turbine.

The number of flame tubes n (in Fig. 47, there are 6) can vary and it is selected depending on the arrangement and other requirements, usually equal to 6-12. The fuel burns in the flame tubes, connected to one another by ducts for the transfer of flame. The ducts, in the first place, during starting of the plant ensure ignition of the fuel in all chambers by the transfer of flames from

---

[1] In aircraft technology only the first three designs of chambers are used (especially the 2nd and 3rd) and they are referred to somewhat differently. Specifically, the sectional chambers are customarily called individual.

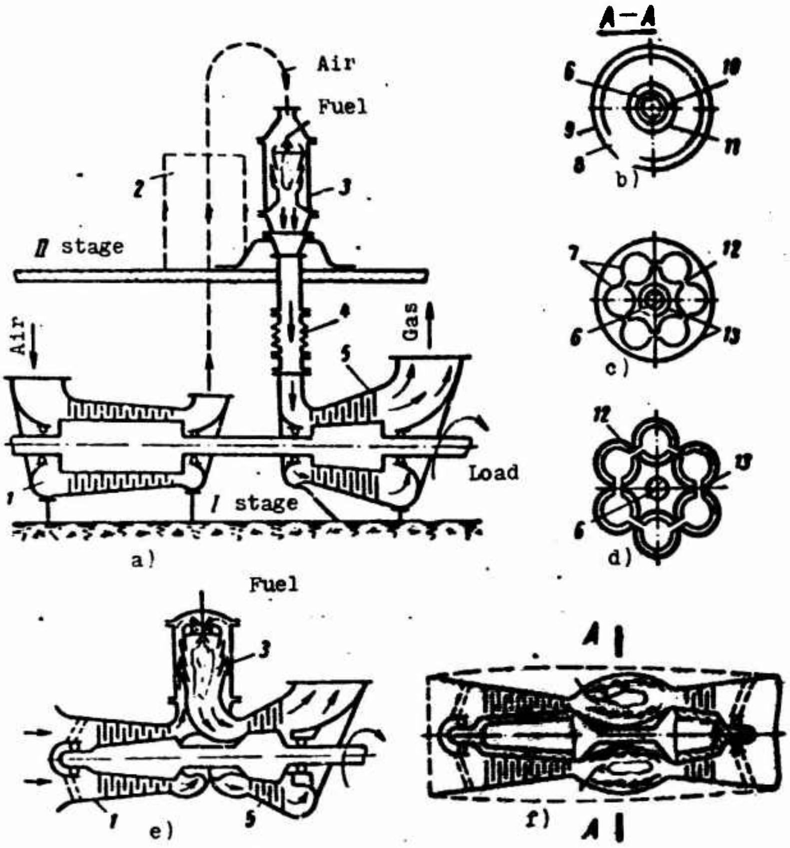


Fig. 47. Diagrams of GTD combustion chambers: 1 – high-pressure compressor; 2 – regenerator; 3 – combustion chamber; 4 – compensator; 5 – turbine; 6 – engine shaft; 7 – individual flame tubes; 8 – external flame tube; 9 – external casing of the chamber; 10 – interior casing; 11 – internal flame tube; 12 – duct for flame transfer; 13 – casing.

burning flame of the neighboring chamber equipped with an ignition device (ignition devices are installed in two-three flame tubes); in the second place, they ensure establishment of the operating process in an individual flame tube during accidental disruption of the flame and they contribute to the equalizing of pressure in the flame tubes. The flow of secondary air flows between the walls of the external and internal casing, washing the flame tubes, and further through the openings of the air wedge it also reaches inside of these tubes. From the flame tubes the gas is directed to the common annular gas collector and further to the turbine.

A sectional chamber has a unit of independent chambers, the number of which n = 6-12, and sometimes more. In this instance, the flame tubes and the housing each have their own chamber. In principle, all chambers of a unit are completely identical, they operate independently, but are also connected by ducts for the transfer of flame. Ignition devices are also installed only in some chambers in the unit. In sectional and cannular chambers, the number of injectors usually corresponds to the number of flame tubes. In annular chambers, the number of injectors is selected in such a manner that the fuel sprays wholly fill the annular working space of the flame tube. In such chambers sometimes one rotating or circular nozzle is used. To avoid burnout of the flame tubes and incomplete burning of the fuel, the drops should not fall on the walls of the flame tube. Individual chambers most frequently are made remote, but are sometimes built-in according to the diagram in Fig. 47e. This permits shortening somewhat the distance between the compressor and the turbine and decreasing the number of injectors. The latter is very important for small GTD where the consumption of air and fuel is small and the use of several chambers undesirable, since the nozzles are made in small sizes they frequently are obstructed and disrupt the operation of the chamber. Most frequently in GTD, the sectional combustion chambers are used, since they have a number of substantial advantages. These advantages include, first of all, cheapness and simplicity of experimental adjustment of the individual chamber on a test stand and, secondly, ease and convenience in servicing and installation (without dismantling the engine). It is known that adjusting operations after the calculation and designing of a chamber usually require considerable investments of time and resources. An annular chamber in final adjustment must be tested to natural value, which requires big investments and complete expenditure of air $G_{B\Sigma}$. For sectional chambers less expenditure of air $G_{B\Sigma}/n$ is necessary, since all n chambers are identical and only one of them is finally adjusted. In practice, in order to simplify the final adjustment in any configuration of the combustion chamber, at least part of these operations must be conducted with models. For annular chamber, research is most frequently conducted on a section — the part of the annular working volume limited by two radial sections.

However, model experiments do not give sufficiently reliable results. A developed design of a sectional combustion chamber can be used in the designing of a new gas turbine engine; only the number of chambers is changed in order to ensure the new assigned air consumption. Besides this, in sectional and cannular chambers in the limited volumes of their flame tubes it is easier to achieve the realization of more improved mixing of the flows of fuel, air and gas, and the realization of the operating process on the whole. However, an annular chamber has a number of its own very important advantages. It is lighter and more compact than the others. On the outlet from such a chamber, as a rule, more even fields of velocities and gas temperatures are obtained, the engine length is reduced somewhat, and friction flows are reduced. However when a large number of regular centrifugal injectors are used for annular chambers, their sizes prove insignificant even with great overall expenditures of fuel through the GTD. Furthermore, in order to disassemble an annular chamber, as a rule, it is necessary to completely disassemble the engine. Such disadvantages, although to a lesser degree, apply to the cannular chamber.

The remote (individual) combustion chamber is convenient because its disassembling, just like a sectional chamber, does not require even the partial dismantling of the engine. The number of injectors here can be selected as most expedient. Such chambers do not require that the gases on outlet have very even fields of temperature and velocity, since they are equalized in the duct which connects the chamber with the turbine. However these chambers have large overall dimensions and weight of ducts and gas collectors, and also additional pressure losses. Furthermore, the final adjustment of the chambers requires complete expenditures of air and large investments.

§ 35. The Operation of Combustion Chambers

The Scheme of the Process

The operation of a GTD combustion chamber is very complex and the presentation regarding it can be made simplest by considering its individual elements and the effect of various factors on them. The aerodynamics of gas and air flows, the character of the supply of fuel and its mixing with air, the ignition and stabilization of the flame, the conditions of mass- and heat-interchange, the regularity of heat release along the length of the chamber – these are the basic elements of the operation. Air entering the combustion chamber, as a rule, has considerable velocities up to 100 m/s and more, especially with a direct-flow diagram of the plant and with air intake directly from the compressor. In this instance, before the front device of the flame tube there is placed a diffuser in which the rate of the flow is reduced in stationary engines, usually to 30-60 m/s, and in aircraft – to 50-80 m/s. Further, through the front device and a number of side openings in the flame tube primary air G_{I} is fed into the combustion zone, a component part of the total expenditure of air $G_{в\Sigma}$. The quantity of primary air most frequently does not exceed 20-50% of $G_{в\Sigma}$. This ensures in the combustion zone the most advantageous concentrations of the mixture with an excess air factor $\alpha_{г} = \frac{G_{\mathrm{I}}}{G_{т}L_{0}} = 1.1\text{-}2.0$, where $G_{т}$ – fuel consumption.

Part of the primary air in a quantity of $G_{фр}$ enters through the front device directly into the beginning of the combustion zone. Usually, the value $\alpha_{фр} = \frac{G_{фр}}{G_{т}L_{0}} = 0.2\text{-}0.5$. With swirling of the air by blade swirlers, the air flow here obtains considerable axial, radial, and tangential component velocities. This air moves in a spiral between the reverse current zone (ZOT) and the flame tube. To avoid a break of the air flow coming out of the swirler from the walls of the diffuser of the front device, the angles of the opening of the diffuser θ and the flow must be consistent with one another. Usually good flow is obtained with a swirler blade angle $\varphi > \frac{\theta}{2}$.

The temperature over the section of the combustion zone [ZG] (ЗГ) increases from $T_{в}$ (the temperature of the entering air) at the walls of the flame tube in a layer diminishing in thickness toward the end of the ZG to a temperature $T_{г}$ (the process of combustion) at the boundary of ZOT. Within the reverse current zone, filled basically with the products of combustion, the temperature barely changes and has considerable magnitude. The position of the front and the whole flame in the ZG is schematically indicated in Fig. 48a. The combustion zone begins in that part of the volume where the rate of motion of the mixture is equal to the rate of the flame propagation. The increase in the width of the combustion zone along the length and the powerful effect of the lateral streams of primary air on its character and position are characteristic. With a change in the depth of penetration of streams, the combustion zone can be disrupted or severely distorted. Behind the stream individual sections of combustion can be generated which can sometimes increase the total surface of the flame and intensify the process because of the sharp agitation of the flow. Usually, in the combustion zone of the chambers, the intensity of turbulence is ten times more than in rough tubes.

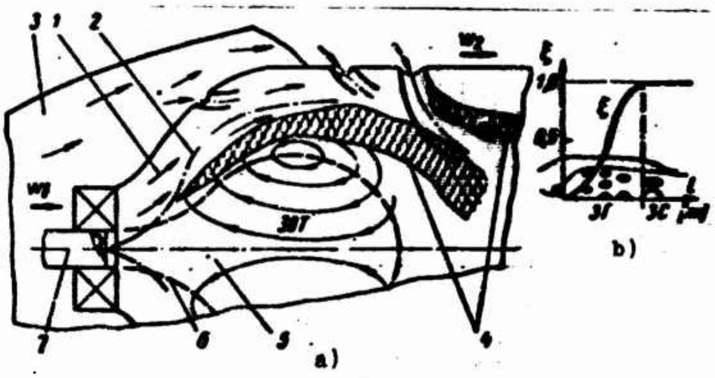


Fig. 48. Diagram of organization of operation in the combustion zone: 1 — air flow; 2 — basic fuel flow; 3 — diffuser; 4 — flame; 5 — products of combustion; 6 — boundary of the reverse current zone (ZOT); 7 — injector.

Changes in the processes of mass- and heat-interchange at various modes can have an effect on the position of the flame front. In the developed designs of chambers, these changes do not lead to a pronounced reduction in the quality of the process and the disruption of the flame over the whole range of working loads. The conditions of flame stabilization in the initial section of the flame tube are determined by the front devices. The quantities of heat, which are liberated along the length of the combustion zone are characterized by a coefficient of heat release ξ, the change in which along the length of the flame tube is shown in Fig. 48b. The character of the curve of heat release depends on many factors and, in particular, upon the distribution and means of supplying air along the length of the combustion zone. A considerable heat release proceeds at a certain distance from the injector nozzle, where the considerable quantities of fuel vapors are formed and correspondingly is fed into the primary air. The input of air in the form of lateral jets agitates the combustible mixture and develops the flame surface. However incorrect direction and excessive intensity of the jets can disrupt the reverse current zone. The process is also especially severely impaired and the flame even stripped with poor mixtures, i.e., with considerable excess air ratios α_r in the combustion zone.

The choice of the best distribution of primary air along the length of the combustion zone and the profile of the supply ducts is one of the most important elements in the planning and the experimental adjusting of the combustion chamber. Upon its solution depends the basic working characteristics of the chamber: the combustion efficiency of the fuel and the character of its change under various loads, cut-off characteristics, the heat density of the chamber, and others. An increase in the depth of penetration of the lateral jets frequently increases the combustion efficiency factor of the fuel, but, as a rule, considerably impairs the cut-off characteristics, narrowing the range of steady operation of the chamber.

The considerable thermal loads on the volume of the chamber require the development of a number of measures and, in particular, the protection of the walls of the flame tube against overheating. The walls are insulated from the flame zone by the flow of cold air; in so doing, it is necessary to ensure the absence of dead zones and local recirculating currents about the walls. These dead zones reduce the working volume and they cause local overheatings, warping, and burnout of the wall or formation of carbon on it. Most frequently, reverse flows appear either directly on outlet from the blade swirler, or at the junction of the diffuser of the front device with the first shell of the flame tube as a result of the disparity between the total pressure loss in the chamber is proportional to the square of the rate of flow

$$\frac{p_{\text{пот}\Sigma}}{p_{\text{вх}}} = \frac{p_{\text{пот}\Sigma}}{p^{*}_{\text{вх}}},$$

where ψ – dimensionless coefficient of the flow friction of the chamber; $\frac{\rho w^2}{2}$ – impact pressure on inlet to the chamber.

With cold blasting, coefficient ψ in a self-similar area has an invariable value. If we determine this coefficient in the combustion chamber during combustion of fuel $\psi_{г}$, then its value does not remain constant but increases in proportion to the decrease in the total excess air factor α_{Σ}. Therefore, the resistance of combustion chambers can be compared only during identical modes, specifically, with identical α_{Σ}.

Research indicates that $\psi_{г}$, determined in the chamber during combustion with $\alpha_{\Sigma} \approx 5$, can be greater than ψ obtained during its cold blasting by 25-35%, and when $\alpha_{\Sigma} \approx 3$, by 45-55%.

Dependence of $\psi_{г}$ on ψ for a cylindrical channel can be obtained in the following form

$$p^{*}_{1} - p^{*}_{2} = \rho_1 w_1 (w_2 - w_1) + \psi \frac{\rho_1 w_1^2}{2}.$$

Taking into account that

$$\psi_{г} = \frac{p^{*}_{1} - p^{*}_{2}}{\frac{\rho_1 w_1^2}{2}},$$

we obtain

$$\psi_{г} = \psi + 2\left(\frac{w_2}{w_1} - 1\right)$$

or

$$\psi_{г} = \psi + 2\left(\frac{T_2}{T_1} - 1\right),$$

the angle of opening of the cone and the flow of air. Sometimes this can be eliminated by increasing the blade angle of the register. Generally speaking, increasing the angle also increases the volume of the ZOT, which as a rule, is useful. However, in so doing, the losses of pressure can be increased and sometimes the flow on outlet from the chamber will be impaired, where swirling flow and even reverse flows will be preserved. Such zones can be more correctly removed by a change in the profile of the walls of the front device or by feeding air through additional openings. Other factors affect the size of the ZOT, apart from the swirling of the flow. For example, during the flow of air in the chamber without a supply of fuel and combustion, the ZOT has larger volume than during combustion, since during combustion the temperature of the gases increases, the density is reduced and the expansion of the gases of the basic flow which is moving between the flame tube and the reverse current zone reduces its diameter and length. In principle, as research shows, the structure of the flow in this case is not changed, which, naturally, cannot be said about the pressure losses in the chamber.

Pressure Losses

The pressure losses in the combustion chamber determine the reduction in the total pressure along its length, which, just like temperature, characterize the proper work capacity of the gas. With cold blasting of a cylindrical chamber when temperatures and rates of flow are identical on intake and outlet $\frac{p^*_{\text{вых}}}{p^*_{\text{вх}}} \ll \frac{p_{\text{вых}}}{p_{\text{вх}}}$, losses can be estimated both in respect to total and static pressures. With the combustion of fuel $T_{\text{вых}} \gg T_{\text{вх}}$ and $\Delta p^* = \phi\frac{\rho w^2}{2}$, i.e., the loss of total pressure even for a cylindrical chamber is considerably more than the static pressure losses. Taking into account that usually combustion chambers are made cylindrical, the estimation of resistance must always be made only in respect to total pressure losses.

In combustion chambers the gas practically always flows in a self-similar area, i.e., at such large Re figures, when the character of its motion does not depend on the Re number. In this instance, since

$$\frac{w_z}{w_0} = \frac{T_z}{T_0}.$$

In actual combustion chambers

$$\psi_z = \psi + A\left(\frac{T_z}{T_0} - 1\right),$$

where A — an experimental factor obtained as a result of research on similar designs.

During cold blasting of a chamber the value of ψ reflects purely hydraulic losses and may be connected with the value considered earlier (Chapter IX)

$$\sigma = \frac{\Delta p^*}{p_0^*}$$

in the following manner

$$\psi = \frac{p_0^* - p_k^*}{\frac{\rho_0 w_0^2}{2}} = \frac{\sigma}{\frac{\rho_0 w_0^2}{2 p_0^*}}.$$

Taking into account that

$$\rho_0 = \frac{p_0}{gRT_0}$$

and

$$p_0^* = p_0\left(1 + \frac{k-1}{2} M_0^2\right)^{\frac{k}{k-1}},$$

we obtain

$$\sigma = \frac{\psi k M_0^2}{2\left(1 + \frac{k-1}{2} M_0^2\right)^{\frac{k}{k-1}}}.$$

where $M_в$ — Mach number on inlet.

The value of σ takes into account the friction loss of the working medium around the walls of the ducts, the losses connected with circulation flows, contraction and expansion of flows, the mixing of the gas streams, local resistance, and loss on the intake diffuser of the chamber. Losses in the diffuser are sometimes very considerable, comprising the basic portion of the total losses of a chamber. These losses increase sharply with an increase in the angle of opening of the diffuser. The total head loss in the diffuser on impact

$$\Delta \mathring{p}_{д} = k\frac{\varrho_в w^2_{вх}}{2}\left(1 - \frac{F_{вх}}{F_{вых}}\right),$$

where k — coefficient of softening the impact.

The change in coefficient k depending on the angle of opening of an equivalent (circular) diffuser θ is shown in Fig. 49. At considerable rates of air intake to the diffuser (about 100 m/s) and large angles of θ (more than 15-25°) $\Delta \mathring{p}_{д}$ can comprise the main portion not only of σ, but also of $\sigma_г$. Frequently the problem is complicated also by the fact that it is necessary to make a diffuser which is not circular and symmetrical, which often requires special experimental final adjustment of it. With a decrease in θ, $\Delta \mathring{p}_{д}$ drops, but commensurable with the pressure loss due to friction in the diffuser becomes

$$\Delta \mathring{p}_{д.т} = \xi_{тр}\frac{l}{d_э}\cdot\frac{\varrho w^2}{2},$$

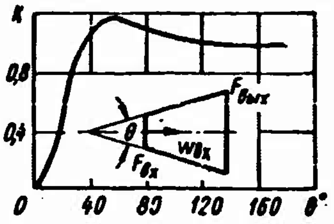


Fig. 49. Dependence of the coefficient of softening the impact on the angle of opening of the diffuser.

where l — length of the diffuser; $d_э$ — equivalent diameter; $\xi_{тр}$ — coefficient of friction, which is a Re function.

When $5 \cdot 10^3 < \mathrm{Re} < 100 \cdot 10^3$

$$\xi_{тр} = \frac{0{,}3164}{\mathrm{Re}^{0.25}}.$$

Frequently, by using experimental data, they compute the sum of losses in the diffuser according to the expression

$$\Delta p^{*}_{д\Sigma} = \psi_д \frac{\rho w^2}{2},$$

where $\psi_д$ — coefficient of flow friction of the diffuser.

Pressure Losses in Mixing Air Streams

The total pressure losses in the mixing of the lateral streams which flow into the flame tube can be estimated on the strength of the diagram presented in Fig. 50a. We will examine a cylindrical element of a combustion chamber with an infinitely thin flame tube wall. This condition excludes its effect on the direction of the flowing stream, the axial component of the rate of which w_2 is equal to the rate of the air in the circular gap, and the radial component is equal to the rate in the openings of the flame tube w_r. The radial velocity determines the expenditure of air G_r flowing into the flame tube which depends on the drop in static pressures Δp in the circular gap and the flame tube. If in section *I-I* prior to the admission of the lateral streams the expenditure of air in the flame tube was equal to $G'_{ж.тр}$ at a rate of motion of air $w'_{ж.тр}$ and a pressure of $p'_{ж.тр}$, then after mixing with the lateral streams in section *II-II*, its velocity becomes equal to $w''_{ж.тр}$, pressure $p''_{ж.тр}$ and expenditure increases to $G''_{ж.тр}$. If we designate $\frac{G_r}{G'_{ж.тр}} = n$ and $\frac{w_2}{w'_{ж.тр}} = m$, then $G''_{ж.тр} = G'_{ж.тр}(n+1)$ and $w''_{ж.тр} = w'_{ж.тр}(n+1)$.

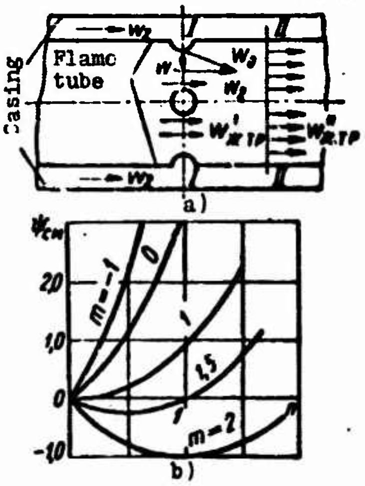


Fig. 50. Diagram of mixing of lateral streams with basic flow in a flame tube.

From the equation of the quantity of motion

$$F\left(p'_{ж.тр}-p''_{ж.тр}\right)=\frac{G''_{ж.тр}}{g}w''_{ж.тр}-\left(\frac{G'_{ж.тр}w'_{ж.тр}+G_r w_2}{g}\right)$$

it is possible to determine the static pressure losses of a flow in the flame tube after influx and mixing lateral streams with it:

$$\Delta p_{см_{I-II}}=p'_{ж.тр}-p''_{ж.тр}=\frac{\varrho\left(w'_{ж.тр}\right)^2}{2}\left[2n\left(2+n-m\right)\right],$$

since $F=\frac{G'_{ж.тр}v}{w'_{ж.тр}}$, where v – specific volume of the gas in the flame tube which can be considered identical for sections *I-I* and *II-II*.

Total head losses in the mixing of the streams can be obtained from Bernoulli's equation for the streaming of a current of constant mass inside the flame tube between sections *I-I* and *II-II*:

$$p'_{ж.тр}+\frac{\varrho\left(w'_{ж.тр}\right)^2}{2}=p''_{ж.тр}+$$
$$+\frac{\varrho\left(w''_{ж.тр}\right)^2}{2}+\Delta p_{см_{I-II}}.$$

Using the previous designation, after conversions we obtain

$$\Delta p^{*}_{см_{I-II}}=\frac{\varrho\left(w'_{ж.тр}\right)^2}{2}\times$$
$$\times\left(n^2+2n-2nm\right).$$

In combustion chambers the value of $\Delta\dot{p}_{\text{см}}$, as a rule, is positive, i.e., the pressure is reduced. However, with the mixing of the streams, the total head in the passing flow with the influx of lateral streams into it car also be increased because of the fact that, the flow is speeded up by the incoming streams. This is easy to indicate, having examined the character of dependence obtained for $\Delta\dot{p}_{\text{см}}$ or the value proportional to it, $\dot{\psi}_{\text{см}} = \frac{\Delta\dot{p}_{\text{см}}}{\frac{\varrho (w'_{\text{ж.тр}})^2}{2}}$, called the total pressure loss factor in mixing. A change of $\dot{\psi}_{\text{см}}$ with a change of n and m is shown in Fig. 50b. The value of $\dot{\psi}_{\text{см}}$ at some positive values of m and small values of n can be negative.

With an increase in n, the value of $\dot{\psi}_{\text{см}}$ becomes positive because of the excess of losses which appear during the deflection and mixing of streams over the increase of total head, which appears with the acceleration of the passing flow by the lateral streams, which possess a high horizontal component of velocity. Let us note, that if the directions of the flows in the annular clearance and the flame tube are opposite, as takes place in the reverse-flow combustion chambers, then the total head losses at mixing sharply increase. The conditions of mixing are substantially changed with a change in the relationship of the velocities in the clearance and the flame tube and the change in the quantity of air which flows out in the form of streams from the clearance into the flame tube. Diameter d or number z of the openings for lateral streams and the drop in static pressures Δp do not affect $\Delta\dot{p}_{\text{см}}$ under the condition that

$$G_r = \frac{\pi d^2}{4} z\mu \sqrt{\frac{2g\Delta p}{\gamma}} = \text{const}.$$

However for the operating process d, z and Δp have very great significance since they define the depth h of penetration of the cold air streams into the mixture. Depending on the place of supplying the air, this depth must be different, just as the number of openings in the given section. The air which cools the wall of the flame tube must not be introduced deeply, in order to generate a continuous layer, washing the wall. The streams of primary air being introduced along the length of the combustion zone should not intersect the flow of gases in the flame tube. Considerably more intense and penetrating to the center of the chamber must be the streams which enter the mixing zone, which permits equalizing the temperature of the gases more rapidly and completely. The depth of penetration of the streams depends on the pressure drop head, of the inclination of the axis of the opening to the axis of the passing flow, the diameter of the opening, its form, the thickness of the wall, the cleanness and profile of the edges of the opening, and so forth. Air is introduced into the flame tube in the form of deflected streams which spread over the wall or perpendicular to the flow of gas in the flame tube. Figure 51a and b gives diagrams of these versions of the influx. For determination of the depth of penetration of the stream during its influx into the flame tube perpendicular through the round or close to the round openings it is possible to use the following relationship obtained on the basis of experimental data:

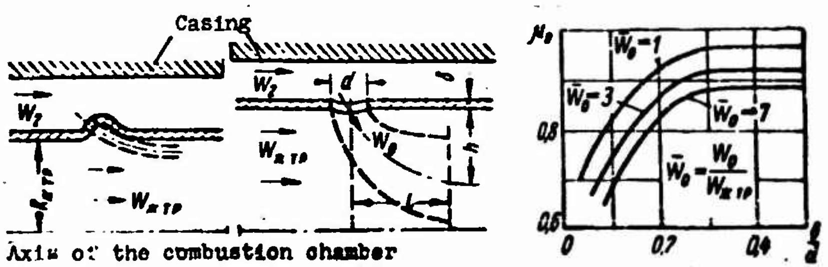
$$h = d_э\left[0{,}3 + 0{,}415\left(\frac{w_г}{w_{ж.тр}}\right)\right]\left(\frac{l}{d_э}\right)^{0.63},$$

where $d_э$ – equivalent diameter of an opening; l – length, to the depth of which the penetration of the stream is equal to h

$$w_г = \mu_э\sqrt{2g\Delta p\rho}.$$

Here the flow coefficient

$$\mu_э = f\left(\frac{w_г}{w_{ж.тр}};\ \frac{\delta}{d_э}\right).$$



a) b) c)

Fig. 51. Diagram of the influx of lateral streams along the length of the combustion zone.

The dependence of the flow coefficient is given in Fig. 51c for round openings with rounded edges with a radius of 0.1-0.2 mm.

Pressure losses to friction about the walls of the ducts are determined theoretically just as friction loss in the diffuser, but account is taken of the profile of the duct and the flow condition of the mixture. Frequently, the flow friction of the total pressure of elements of the chamber, for example, the front devices, are determined separately. Usually these resistances are expressed by a dependence of the type

$$\Delta p^{*}_{\phi p} = \psi_{\phi p} \frac{\varrho w^{2}}{2},$$

where $\psi_{\phi p}$ — the coefficient of flow friction of the front device determined experimentally.

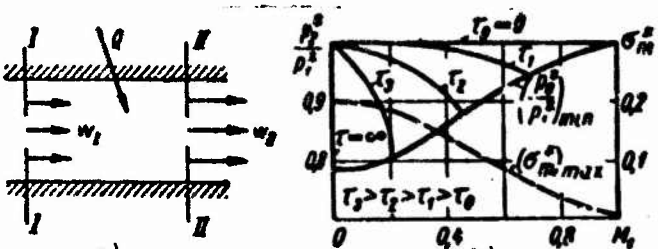
Research shows that the sum of purely hydraulic losses in cold and hot blasting is not changed substantially. It is natural that some difference will appear in connection with the change in the flow parameters, but the basic distinction is made by the pressure loss appearing during combustion, being determined by the supply of heat to the working medium.

Pressure Losses in the Gas Flow During Heating

From thermodynamics and gas dynamics it is known that when heat is supplied to a gas flow its total head drops. In so doing, the magnitude of the losses strongly depends on the form of the duct. In an expanded duct the losses will be less than in a constricted one. Figure 52a shows an element of a cylindrical duct from section *I-I* (all parameters of the gas are known) to section *II-II*. We will determine in a one-dimensional diagram of flow, the total head loss in the flow appearing because of the application of heat Q.

The equation of state $pv = RT$, after differentiation, gives

$$dT = \frac{p\,dv + v\,dp}{R}.$$



a) b)

Fig. 52. Diagram of gas flow through duct F = const during heating.

The equation of the first law of thermodynamics $dQ = c_p DT - Av dp$ with the substitution of the value of dT will have the form

$$dQ = \frac{1}{R}(c_p p\,dv + c_v v\,dp).$$

From the equation of continuity for F = const in the form $w/v = G/F = G_F$ = const we have $w = v \cdot G_F$. Substituting w in the equation of quantity of motion

$$-v\,dp = d\frac{w^2}{2g},$$

we obtain that

$$dv = -\frac{g}{G_F^2}dp,$$

whence a specific volume v or v_2

$$v = \frac{g}{G_F^2}(p_1 - p) + v_1.$$

Consequently,

$$dQ = \frac{c_v}{R}\left(\frac{g}{G_F^2}p_1 + v_1\right)dp - \frac{(c_p + c_v)}{R} \cdot \frac{g}{G_F^2}p\,dp. \tag{45}$$

Using Mach number $M = \sqrt{\frac{w^2}{kgRT}}$, the equation of expenditure and the equation of a state, it is possible, having converted the right-hand part of the equation, to rewrite equation (45) thus:

$$dQ = \frac{c_v T_1 (1 + k M_1^2)}{p_1 k M_1^2} dp - \frac{c_v T_1 (k+1)}{p_1^2 k M_1^2} p\,dp.$$

Considering for simplicity heat capacities c_p and c_v as constant, we designate multipliers with *dp* and *pdp* in the right-hand part respectively as C_1 and C_2. Integrating the expression obtained within limits from 0 to Q and from p_1 to p or p_2, we obtain

$$Q = C_1 (p - p_1) - C_2 \frac{p^2 - p_1^2}{2}.$$

Having designated $(p_1 - p)$ as Δp, we obtain the quadratic equation $\frac{C_2}{2} \Delta p^2 - (C_2 p_1 - C_1) \Delta p + Q = 0$, the solution of which gives

$$\Delta p = \left(p_1 - \frac{C_1}{C_2}\right) - \sqrt{\left(p_1 - \frac{C_1}{C_2}\right)^2 - \frac{2}{C_2} Q}.$$

Having substituted the values C_1 and C_2 after conversions we finally receive

$$\Delta p = \frac{p_1 k}{k+1} (1 - M_1^2) - p_1 \sqrt{\left[\frac{k}{k+1} (1 - M_1^2)\right]^2 - \frac{2k^2 M_1^2}{k+1} \cdot \frac{Q}{c_p T_1}} \qquad (46)$$

and the relative fall of static pressure

$$\sigma_r = \frac{\Delta p}{p_1} = \frac{k}{k+1} (1 - M_1^2) - \sqrt{\left[\frac{k}{k+1} (1 - M_1^2)\right]^2 - \frac{2k^2 M_1^2}{k+1} \cdot \frac{Q}{c_p T_1}}, \qquad (47)$$

where k — adiabatic exponent.

The value T_1 is determined according to the stagnation temperature from the expression

$$T_1 = \frac{T_1^*}{1 + \frac{k-1}{2} M_1^2}.$$

The value $\sigma_r = 0$ when $Q = 0$, and also at an intake velocity of $w_1 = 0$ ($M_1 = 0$). This means that in these cases the static pressures prior to and after preheating are equal, i.e., $p_1 = p_2$. Furthermore, from equation (47) it follows that with M_1 and T_1^* given for a unit of mass of air it is possible to conduct the fully determined maximum quantity of heat Q_{max}. This quanitty corresponds to the condition of equality to zero of a radicand where the flow in the duct attains the speed of sound ($M_2 = 1$). In this case

$$(\sigma_r)_{max} = \frac{k}{k+1}(1-M_1^2),$$

and the maximum relative preheating of the mixture

$$\left(\frac{Q}{c_pT_1^*}\right)_{max} = \frac{(1-M_1^2)^2}{2(k+1)M_1^2\left(1+\frac{k-1}{2}M_1^2\right)}. \qquad (48)$$

Hence it is apparent that when M_1 $Q = 0$ and $\sigma_r = 0$. Consequently, in a cylindrical tube the gas can move at the speed of sound only in the absence of preheating.

The growth of the quantity of conducted heat at given M_1 and T_1^* increases the temperature of the gas and correspondingly reduces its density. The rate of flow increases, but, in turn (with a constant mass flow rate of the gas), the static pressure drops along the length of the duct. It is obvious that with continuous increase in preheating the moment can arrive when (at a given p_1) the velocity of the gas in the duct attains the speed of sound, and the duct is not able to admit the given expenditure of gas. Considering the element of a duct from section *I-I* to section *II-II* where $Q < Q_{max}$, it is possible to find $\Delta p = (p_1 - p_2)$ and then p_2.

From the expression $p^* = p\left(1+\frac{k-1}{2}M^2\right)^{\frac{k}{k-1}}$ there is determined p_2^*, if the M_2 number is known. If the parameters of a flow after preheating are known and gas constant R = const, then

$$M_2^2 = \frac{w_2^2}{k_gRT_2} = \frac{w_1^2}{k_gRT_1}\left(\frac{w_2}{w_1}\right)^2\left(\frac{T_1}{T_2}\right).$$

When $R_1 = R_2$, from the equation of expenditure γw = const, we have

$$\frac{T_1}{T_2} = \frac{p_1}{p_2} \cdot \frac{w_1}{w_2} = \frac{\left(\frac{w_1}{w_2}\right)}{1-\sigma_T}.$$

From the equation of quantity of motion

$$p_1 - p_2 = \varrho_1 w_1 (w_2 - w_1);$$

$$\frac{w_2}{w_1} = \frac{\Delta p}{\varrho_1 w_1^2} + 1 = \frac{\Delta p p_1}{p_1 \varrho_1 w_1^2} + 1 = \frac{\sigma_T}{k M_1^2} + 1.$$

Consequently,

$$M_2^2 = M_1^2 \frac{\frac{\sigma_T}{k M_1^2} + 1}{1-\sigma_T} = \frac{\sigma_T + k M_1^2}{k(1-\sigma_T)}.$$

In this way,

$$\frac{\dot{p}_2}{\dot{p}_1} = \frac{p_2\left(1 + \frac{k-1}{2} M_2^2\right)^{\frac{k}{k-1}}}{p_1\left(1 + \frac{k-1}{2} M_1^2\right)^{\frac{k}{k-1}}} =$$

$$= (1-\sigma_T)\frac{\left[1 + \frac{k-1}{2} \cdot \frac{\sigma_T + k M_1^2}{k(1-\sigma_T)}\right]^{\frac{k}{k-1}}}{\left(1 + \frac{k-1}{2} M_1^2\right)^{\frac{k}{k-1}}}. \qquad (49)$$

The relative loss of total head only because of preheating of gases:

$$\dot{\sigma}_T = \frac{\dot{p}_1 - \dot{p}_2}{\dot{p}_1} = 1 - \frac{\dot{p}_2}{\dot{p}_1}$$

or finally

$$\sigma_r^* = 1 - (1 - \sigma_r) \times$$

$$\times \frac{\left[1 + \frac{k-1}{2} \frac{\sigma_r + kM_1^2}{k(1-\sigma_r)}\right]^{\frac{k}{k-1}}}{\left(1 + \frac{k-1}{2} M_1^2\right)^{\frac{k}{k-1}}}. \qquad (50)$$

The value $(\sigma_r^*)_{max}$ with $\sigma_r = (\sigma_r)_{max}$

$$(\sigma_r^*)_{max} = 1 - \frac{1 + kM_1^2}{k+1} \left[\frac{k+1}{2 + M_1^2(k-1)}\right]^{\frac{k}{k-1}}. \qquad (51)$$

The value T_2^* can be obtained from the expression $Q = \Delta I^*$:

$$T_2^* = T_1^* + \frac{Q}{c_{p_{cp}}},$$

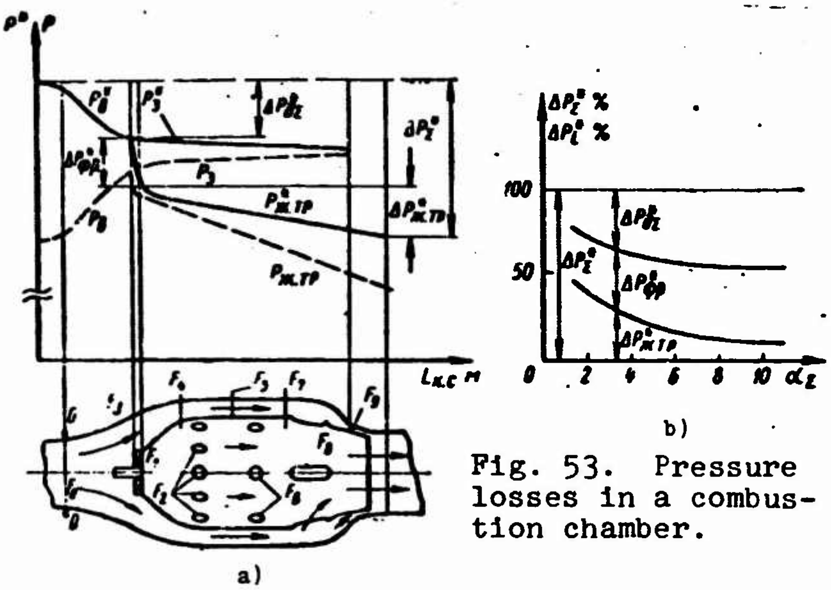
where $c_{p_{cp}}$ — average value of heat capacity in the temperature range from T_1^* to T_2^*.

The value of σ_r^*, consequently, depends on Mach number M_1 and relative preheating $\tau = \frac{Q}{c_p T_1^*}$, since $\sigma_r^* = f(M_1$ and $\sigma_r)$, and $\sigma_r = f_1$ $(M_1$ and $\tau)$.

With M_1 = const $\sigma_r^* = 0$, if $\sigma_r = 0$, i.e., with $\tau = \frac{Q}{c_p T_1^*} = 0$ (no preheating). The value σ_r^* will be maximum where $\tau_{max} = \left(\frac{Q}{c_p T_1^*}\right)_{max}$, which will also determine $(\sigma_r)_{max}$. With growth of M_1 to 1, the value of $(\sigma_r^*)_{max}$ drops to zero, as shown in Fig. 52b. This is normal since with the increase in M_1 there is a reduction in the maximum possible preheating, and with $M_1 = 1$ we obtain $\left(\frac{Q}{c_p T_1^*}\right) = 0$. With $M_1 \to 0$ and $\tau \to \infty$ the value of σ_r^* attains the greatest value somewhat exceeding 0.2.

§ 36. Components of Total Pressure Losses

The character of the change in the static and total pressure along the length of the combustion chamber is different for the secondary and primary air duct (in the annular clearance and inside the flame tube). Naturally, the total pressure of the working medium drops in any event, but in a different way, while the static pressure in the clearance, as a rule, is increased. This is explained to the fact that along the length of the annular duct the quantity of air is reduced, since it is supplied stepwise inside the flame tube, the cross-sectional area of the duct, as a rule, changes little or remains constant. The section of the duct is made constant in order to increase the drop in static pressures towards the end of the duct to ensure deep supply of air streams in the mixer. Static pressure along the length of the flame tube drops. The character of the change in pressure can be seen in Fig. 53a where the following basic components of total losses of total head Δp^*_{Σ} are shown: 1) losses on the intake diffuser $\Delta p^*_{\partial\Sigma}$; 2) losses in the front device and openings $\Delta p^*_{\text{фр}}$; 3) pressure losses inside the flame tube $\Delta p^*_{\text{ж.тр}}$. The latter include losses on mixing of the flows, friction losses and thermal losses. It is obvious that under various conditions the absolute value of the total losses of the total head in the chamber, the values and the relationship of the individual components will change. During combustion of fuel in the chamber and with $G_{\text{в}} = \text{const}$, the portion of $\Delta p^*_{\text{ж.тр}}$ will increase with the decrease in α_{Σ}, since the supply of heat to the flow is increased. The qualitative picture of these changes is shown in Fig. 53b. Results of experiments show that during cold blasting of the chambers the value of $\Delta p^*_{\text{ж.тр}} \approx 5-12\%$, $\Delta p^*_{\text{фр}} \approx 30-45\%$ and $\Delta p^*_{\partial\Sigma} \approx 35-50\%$ of the total losses. During fuel combustion with $\alpha_{\Sigma} = 3-4$, total losses of total head in the combustion chamber increase by approximately 30-50% in comparison with cold blasting, and the fractions of individual losses become commensurable, attaining approximately 25-35%. The decrease in total head in the combustion chambers determines the possibility of improving the course of operation as a result of reduction of the velocity of air and the formation of a zonal arrangement of a flow behind the front device which is necessary for ensuring carburetion and stabilization of the flame front, and so forth. The individual loss components cannot be completely removed, but their rational reduction without impairing the operating characteristics of the chamber is advisable. In the



a)

b)

Fig. 53. Pressure losses in a combustion chamber.

greatest degree this concerns losses in the diffuser since they constitute a substantial fraction of the overall losses, and in correct planning these losses can be reduced without impairment of the operating process.

The diagrams examined for determination of the individual types of losses of total head present the idea regarding the basic correlations of the change in these losses and the effect of various parameters on them. However, the actual picture of gas flow and the supply of heat at various local rates and temperatures, and nonuniform distribution of heat along the length and over the section of a chamber is more complex. Therefore, there still have to be developed methods for the analytical calculation of the total losses of total head of actual combustion chambers. Nevertheless, by using experimental data and assuming that hydraulic and thermodynamic losses do not depend on one another, it is possible with a sufficient degree of accuracy to estimate the pressure losses under actual conditions. In this case the losses connected with the supplying of heat can be determined according to the diagram examined above for a one-dimensional flow or from empirical dependence; the hydraulic losses — according to the total experimental dependence of the type $\Delta p^* = \psi \frac{\rho w^2}{2}$, which is obtained in cold blasting of chambers.

§ 37. Hydraulic Calculation of a Combustion Chamber

Hydraulic calculation serves for determination of losses of total head and the distribution of the gas-air flows via the ducts of the combustion chamber. Calculation can be conducted, if the following are given: the temperature of air $T_в$, pressure $p_в$ and the total expenditure of air $G_в$ on intake to the chamber, the excess air ratio α_Σ (or the expenditure of fuel G_T) and the temperature of gases on outlet from the chamber $T_г$, the heat of fuel combustion $Q_н^р$ and all the geometric dimensions of the combustion chamber and its elements, including the location, size, and number of openings for primary and secondary air. The flow area of the openings are laid out in the design of the chamber as shown in Fig. 53a. Calculation is performed by the method of successive approximations in the following order:

I. The characteristic values on intake to chamber are determined:

1) the specific volume of air in section *0-0*

$$v_в = \frac{RT_в}{p_в};$$

2) the cross-sectional flow area for air F_0;

3) the velocity of air $w_в = \frac{G_в v_в}{F_в}$;

4) the speed of sound a according to temperature $T_в$;

5) Mach number on intake $M_в = \frac{w_в}{a}$.

Using the gas-dynamic function tables for the velocity coefficient $\lambda = \frac{w_в}{a_{кр}}$, it is determined that $\tau(\lambda) = \frac{T_в}{T_в^*}$; $\pi(\lambda) = \frac{p_в}{p_в^*}$; $\varrho(\lambda) = \frac{\varrho_в}{\varrho_в^*}$ and further $T_в^* = \frac{T_в}{\tau(\lambda)}$; $p_в^* = \frac{p_в}{\pi(\lambda)}$; $\varrho_в^* = \frac{\varrho_в}{\varrho(\lambda)}$.

II. The areas of all flow areas, F_i, are determined.

III. Approximate calculations are made of the relative expenditures of air passing through each section of F_i: $G_i = F_i \mu_i \sqrt{\frac{2g\Delta p}{v_в}}$, considering that the pressure drop Δp is identical everywhere.

The sum of the expenditures must, naturally, be equal to the given expenditure $\sum G_i = G_1 + G_2 + \ldots + G_i = G_в$. Relative expenditures by sections: $\bar{G}_i = \frac{G_i}{G_в} = \frac{F_i \mu_i}{\sum F_i \mu_i}$, where μ_i – the coefficients of expenditure of the corresponding sections taken from handbooks for the given type of opening, slot, and so forth.

IV. The losses of total head are determined according to secondary air duct, i.e., the friction loss and impact on the intake diffuser and the friction losses in the annular clearance between the flame tube and casing. Additionally, using gas-dynamic functions, all parameters for the duct are determined: p_i, w_i, T_i, ρ_i.

V. Static pressure is determined along the length of the flame tube. As a first approximation, the pressure drop on intake to the flame tube can be represented in the form of $\Delta p_{ж.тр} = \frac{\rho w_в^2}{2}$. Having determined friction losses in the flame tube $\Delta p_{тр}$ and thermal losses $\Delta p_{тепл}$ according to empirical equations total and static pressure are computed for a control section and at the end of the flame tube. On outlet from the chamber $p_г = p_в - \Delta p_{ж.тр} - \Delta p_{тр} - \Delta p_{тепл}$. In the capacity of a control section, a section is selected where all the primary air is already introduced inside the flame tube, i.e., the end of the combustion zone.

VI. The actual drops in the static pressures in each section are determined:

$$\Delta p_i = p_i - p_{ж.тр}$$

where $p_{ж.тр}$ – static pressure inside the flame tube in the given section; p_i – static pressure for the secondary air duct in the given section.

VII. The actual expenditures of air are computed (in the second approximation):

$$G_i' = F_i \mu_i \sqrt{\frac{2g\Delta p_i}{v_{\text{в}}}};$$

for simplicity, $v_{\text{в}}$ can be considered invariable. Total expenditure $\sum G_i' = G_{\text{в}}'$. Frequently, the second approximation does not give correspondence between $G_{\text{в}}'$ and the given expenditure of air $G_{\text{в}}$. If this difference is more than approximately 5%, then expenditure is determined according to the third approximation, where it is convenient to proceed thus:

1) to determine the relative expenditures of air $\bar{G}_i = \frac{G_i'}{G_{\text{в}}'}$;

2) to calculate the expenditures of air in each section

$$G_i'' = \bar{G}_i' \cdot G_{\text{в}},$$

where $G_{\text{в}}$ — the actual total expenditure of air in the chamber. Further calculation must be repeated, beginning with III.

§ 38. Determination of the Composition of Gas On Outlet From the Combustion Chamber

The question has already been considered above as to how to determine the quantity of the individual products of the oxidation reaction of the combustible elements of the fuel and their totals in the combustion of a unit of weight (mass) of fuel with the theoretically necessary (stoichiometric) quantity of air L_0 when the excess air ratio $\alpha = 1$. The products of combustion obtained in this case are customarily called pure products of combustion.

In actual combustion chambers, as a rule, $\alpha_\Sigma \neq 1$, but is considerably more than one. In this way, always besides the pure products of combustion, in the gases there is present excess air in the quantity $(\alpha_\Sigma - 1)$. In this way it is possible to determine the composition of the gases after the combustion chamber, knowing the value of α_Σ. The value of α_Σ can be determined from the heat balance equation which links the quantity of heat entering the combustion chamber and leaving it. The heat balance equation of the combustion chamber relative to 1 kg of fuel at a temperature reading of 0°C, has the form

$$i_T^*\Big|_0^{t_T} + \alpha_\Sigma L_0 i_в^*\Big|_0^{t_в} + Q = (1 + \alpha_\Sigma L_0)\, i_г^*\Big|_0^{t_г}. \qquad (52)$$

The first member of the left part of the equation expresses the heat content of 1 kg of fuel at its stagnation temperature t_T^* before the nozzle, the second member – the heat content of air which is necessary for 1 kg of fuel at the stagnation temperature of air on intake to the chamber $t_в^*$, the third member – the heat which went to heating the gases. The right-hand part of the equation – this is the heat content of gases emerging from the combustion chamber with temperature $t_г^*$.

The value

$$Q = Q_{в0}^p - Q_{H_2O} - Q_{окр} - Q_{нед}.$$

Here $Q_{в0}^p$ – higher heat of combustion of fuel at 0°; Q_{H_2O} – heat of vaporization; $Q_{окр}$ – heat leaked by the combustion chamber into the environment; $Q_{нед}$ – the heat leaked as a result of the chemical and mechanical incomplete burning of fuel.

Knowing the higher heat of combustion of the fuel $Q_{в t_0}^p$, determined at the standard temperature of calorimetric measurement t_0, the value of $Q_{в0}^p$, using Hess' law, can be defined as the sum: $Q_{в t_0}^p$ plus the difference of the heat content of the products of reaction and the initial mixture at corresponding temperatures. Since $Q_{в t_0}^p = Q_{н t_0}^p + Q_{H_2O}$ or, for simplicity, dropping the subscript t_0 for Q, at standard temperature we have $Q_в^p = Q_н^p + Q_{H_2O}$. and, considering the heat of vaporization at 0°C equal to the standard temperature of calorimetric measurement t_0, it is possible to write (see § 8):

$$Q^p_{\text{вп}} = Q^p_{\text{н}} + Q_{H_2O} + \Big[(1 + \alpha_\Sigma L_0)\, i^*_{\text{г}}\Big|_0^{t_0} -$$

$$- \alpha_\Sigma L_0 i^*_{\text{в}}\Big|_0^{t_0} - i^*_T\Big|_0^{t_0}\Big]. \tag{53}$$

The relationship of the heat content of the products of combustion and of air and their change depending on temperature is shown in Fig. 54. Substituting the obtained expression $Q^p_{\text{вп}}$ in the heat balance equation and designating $Q^p_{\text{н}} - Q_{\text{окр}} - Q_{\text{нед}} = Q^p_{\text{н}}\eta_{\text{г}}$, where η_{Γ} – the combustion efficiency of the fuel, we obtain

$$i^*_T\Big|_0^{t_T} + \alpha_\Sigma L_0 i^*_{\text{в}}\Big|_0^{t_{\text{в}}} + \eta_{\text{г}} Q^p_{\text{н}} +$$

$$+ (1 + \alpha_\Sigma L_0)\, i^*_{\text{г}}\Big|_0^{t_0} - \alpha_\Sigma L_0 i^*_{\text{в}}\Big|_0^{t_0} - i^*_T\Big|_0^{t_0} =$$

$$= (1 + \alpha_\Sigma L_0)\, i^*_{\text{г}}\Big|_0^{t_{\text{г}}};$$

let us note, that

$$i^*_{\text{в}}\Big|_0^{t_{\text{в}}} = \left(c_{p_{\text{в}}} t^*_{\text{в}} - c_{p_{\text{в}}} 0\right) = c_{p_{\text{в}}} t^*_{\text{в}},$$

where $c_{p_{\text{в}}}$ – the average weight heat capacity of air in the range of temperatures from 0° to $t^*_{\text{в}}$C; furthermore, the difference

$$i^*_T\Big|_0^{t_T} - i^*_T\Big|_0^{t_0} = i^*_T\Big|_{t_0}^{t_T}$$

at a fuel temperature of t^*_T greater than t^*_0, which is usually equal to 15°C. Then after conversion the heat balance equation takes the form

$$\eta_{\text{г}} Q^p_{\text{н}} = (1 + \alpha_\Sigma L_0)\, i^*_{\text{г}}\Big|_{t_0}^{t_{\text{г}}} - \alpha_\Sigma L_0 i^*_{\text{в}}\Big|_{t_0}^{t_{\text{в}}} - i^*_T\Big|_{t_0}^{t_T}. \tag{54}$$

In this way

$$\alpha_\Sigma = \frac{\eta_{\text{г}} Q^p_{\text{н}} - i^*_{\text{г}}\Big|_{t_0}^{t_{\text{г}}} + i^*_T\Big|_{t_0}^{t_T}}{L_0\left(i^*_{\text{г}}\Big|_{t_0}^{t_{\text{г}}} - i^*_{\text{в}}\Big|_{t_0}^{t_{\text{в}}}\right)}$$

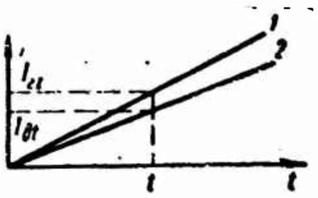


Fig. 54. Graph of the heat content of air and pure products of combustion: 1 – products of combustion; 2 – air.

and

$$\eta_г = \frac{(1+\alpha_\Sigma L_0) i_г^{*}\Big|_{t_0}^{t_г} - \alpha_\Sigma L_0 i_в^{*}\Big|_{t_0}^{t_в} - i_т^{*}\Big|_{t_0}^{t_т}}{Q_н^р}$$

or

$$\alpha_\Sigma = \frac{\eta_г Q_н^р - \left(c_{p_г} t_г^{*} - c_{p_г} t_0^{*}\right) + \left(c_{p_т} t_т^{*} - c_{p_т} t_0^{*}\right)}{L_0\left[\left(c_{p_г} t_г^{*} - c_{p_г} t_0^{*}\right) - \left(c_{p_в} t_в^{*} - c_{p_в} t_0^{*}\right)\right]} \qquad (55)$$

and

$$\eta_г = \frac{(1+\alpha_\Sigma L_0)\left(c_{p_г} t_г^{*} - c_{p_г} t_0^{*}\right)}{Q_н^р} - \frac{\alpha_\Sigma L_0\left(c_{p_в} t_в^{*} - c_{p_в} t_0^{*}\right) - \left(c_{p_т} t_т^{*} - c_{p_т} t_0^{*}\right)}{Q_н^р}. \qquad (56)$$

The expression obtained for the combustion efficiency of fuel $\eta_г$ is used in treating the results of tests and in the creation of characteristics of the combustion chambers. For computing α_Σ according to expression (55) it is necessary to know the composition of the gases for which is found its average weight heat capacity on outlet from the chamber $c_{p_г}$. In order to determine α_Σ without resorting to the method of consecutive approximations, we will present the products of a combustion of 1 kg of fuel with the given α_Σ in the form of a sum of the pure products of (complete) combustion $G_{ч.п}$ in the amount of $(1 + L_0)$ kg per each kilogram of fuel and excess air. The quantity of this air per each kilogram of burned fuel amounts to

$$(1 + \alpha_\Sigma L_0) - (1 + L_0) = (\alpha_\Sigma - 1) L_0.$$

Equation (54) then can be rewritten thus:

$$\eta_г Q_н^р = (1 + L_0) i_{ч.п}\Big|_{t_0}^{t_г} +$$
$$+ (\alpha_\Sigma - 1) L_0 i_в\Big|_{t_0}^{t_г} - \alpha_\Sigma L_0 i_в\Big|_{t_0}^{t_0} - i_T\Big|_{t_0}^{t_T}$$

or

$$\eta_г Q_н^р - (1 + L_0) i_{ч.п}\Big|_{t_0}^{t_г} +$$
$$+ L_0 i_в\Big|_{t_0}^{t_г} + i_T\Big|_{t_0}^{t_T} = \alpha_\Sigma L_0 i_в\Big|_{t_0}^{t_0}$$

whence finally,

$$\alpha_\Sigma = \frac{\eta_г Q_н^р - (1 + L_0)\left(c_{p_{ч.п}} t_г - c_{p_{ч.п}} t_0\right)}{L_0\left(c_{p_в} t_г - c_{p_в} t_0\right)} +$$
$$+ \frac{L_0\left(c_{p_в} t_г - c_{p_в} t_0\right) + \left(c_{p_T} t_T - c_{p_T} t_0\right)}{L_0\left(c_{p_в} t_г - c_{p_в} t_0\right)}. \qquad (57)$$

The average weight heat capacity of pure products of combustion at constant pressure $c_{p_{ч.п}}$ can be determined according to the known individual component of the pure products of combustion. The average mean molar heat capacity of pure products of combustion

$$(\mu c_p)_{ч.п} = (\mu c_p)_{RO_2} r_{RO_2} + (\mu c_p)_{N_2} r_{N_2} + (\mu c_p)_{H_2O} r_{H_2O},$$

where r_{RO_2} – volume fraction of the sum (CO_2 + SO_2); r_{N_2} and r_{H_2O} – volume fractions of N_2 and H_2O respectively; μc_{p_i} – molar heat capacities of the corresponding substances when p = const.

The molecular weight of the pure products of combustion

$$\mu_{ч.п} = \mu_{RO_2} r_{RO_2} + \mu_{N_2} r_{N_2} + \mu_{H_2O} r_{H_2O}.$$

The average weight heat capacity of the pure products of combustion

$$c_{p_{\text{ч.п}}} = \frac{(\mu c_p)_{\text{ч.п}}}{\mu_{\text{ч.п}}}.$$

Having calculated α_Σ, it is possible to determine the value of the relative expenditure of fuel

$$g_{\text{T}} = \frac{G_{\text{T}}}{G_{\text{B}}},$$

where G_{T} and G_{B} – expenditure of fuel and air entering the combustion chamber respectively, measured in kg/h or kg/s.

Since $\alpha_\Sigma = \frac{G_{\text{B}}}{G_{\text{T}} L_0}$, then

$$g_{\text{T}} = \frac{1}{L_0 \alpha_\Sigma}. \tag{58}$$

Further, the value is determined for the average weight heat capacity of the gas mixtures on outlet from the combustion chamber $c_{p_\text{г}}$:

$$c_{p_\text{г}} = \frac{(1 + L_0) c_{p_{\text{ч.п}}} + (\alpha_\Sigma - 1) L_0 c_{p_\text{в}}}{1 + \alpha_\Sigma L_0} \tag{59}$$

or

$$c_{p_\text{г}} = \frac{(1 + L_0) c_{p_{\text{ч.п}}} + \alpha_\Sigma L_0 c_{p_\text{в}} - L_0 c_{p_\text{в}} + c_{p_\text{в}} - c_{p_\text{в}}}{1 + \alpha_\Sigma L_0}$$

and consequently,

$$c_{p_\text{г}} = c_{p_\text{в}} + \frac{1 + L_0}{1 + \alpha_\Sigma L_0} (c_{p_{\text{ч.п}}} - c_{p_\text{в}}). \tag{60}$$

§ 39. Temperature Condition of the Flame Tube Wall

The flame tube absorbs heat from the flame of burning fuel and, in turn gives it off to the air which is cooling it and partially to the casing of the chamber. From the casing a certain portion of this heat leaks into the surrounding medium, while the basic part is transferred to the same cooling air which goes between the casing and the flame tube. The thermal action of the flame on the flame tube in various sections, other conditions being equal, will differ depending on the character of the flow of air along the interior surface of the wall of the tube, the presence of a protective air layer, the form of the flame surface, and others. In this way, in general the change in the temperature of the wall of the flame tube over its length is rather complex, and as yet a reliable method for its analytical calculation has not been created. However, the order of magnitude of the temperature of the flame tube wall can be determined analytically by the method of consecutive approximations, proceeding from the simplest diagram of the phenomenon and assuming that during a steady process the losses of heat into the environment from the casing of the chamber are small and can be neglected. In such calculation, for the characteristic temperature of the flame tube the temperature of the wall is taken in a calculated section and it is considered that it is invariable through the thickness of the wall; furthermore, the protective action of films or streams inside the flame tube usually is not taken into account. These prerequisites give a necessarily overstated calculated temperature of a wall. Figure 55 shows a diagram of the process of heat interchange

$$Q_{\text{ф}} = Q_{\text{ж.тр}} = Q_{\text{в}} \qquad (61)$$

where Q_{Φ} — heat given off by the flame to the wall of the flame tube for a unit of time; $Q_{\text{ж.тр}}$ — heat given off by the wall of the flame tube to the air; Q_{B} — heat obtained by the cooling air going between the flame tube and the casing.

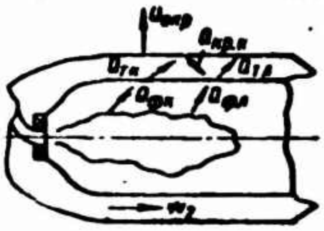


Fig. 55. Calculation of the temperature of the wall of a flame tube.

In turn,

$$Q_{\text{ф}} = Q_{\text{ф.л}} + Q_{\text{ф.к}},$$

i.e., it is made up from the heat given off by the flame to the wall by radiation and convection.

The value $Q_{\text{ж.тр}} = Q_{\text{т.к}} + Q_{\text{т.л}}$, i.e., is equal to the sum of the heat given off by the wall to the cooling air by convection, and the heat given off by the wall by radiation of the interior surface of the casing. However, $Q_{\text{т.л}} = Q_{\text{кр.к}}$, i.e., are equal to the heat given off by the casing to the cooling air as a result of a convection. This means the heat obtained by the cooling air

$$Q_{\text{в}} = Q_{\text{т.к}} + Q_{\text{т.л}} = Q_{\text{т.к}} + Q_{\text{кр.к}} = Q_{\text{ф}}.$$

The heat given off by the flame by means of radiation, is computed according to the equation

$$Q_{\text{ф.л}} = \varepsilon'_{\text{ст}}\varepsilon_{\text{г}}\sigma_0 F_{\text{ж.тр}}\left[\left(\frac{T_{\text{г.ф}}}{100}\right)^4 - \left(\frac{T_{\text{ж.тр}}}{100}\right)^4\right], \qquad (62)$$

where σ_0 – emission constant of an ideal black body; $F_{\text{ж.тр}}$ – interior surface of the flame tube; ε_g – conventional degree of blackness of the flames approximately equal to 0.07-0.10; $\varepsilon'_{\text{ст}}$ – emissivity factor of the enclosure;

$$\varepsilon'_{\text{ст}} = 0{,}5(1 + \varepsilon_{\text{ст}}),$$

where $\varepsilon_{\text{ст}}$ – the degree of blackness of the surface of the given material. For oxidized steel $\varepsilon_{\text{ст}} = 0{,}8-1{,}0$.

The calculated temperature of the gas in the flame $T_{\text{г.ф}}$ in first approximation can be considered equal to the theoretical temperature of the gas in the combustion zone with an excess air ratio α_{Γ}. The unknown temperature of the wall of the flame tube $T_{\text{ж.тр}}$ [°K] which must first be given enters into the expression $Q_{\text{ф.л}}$. Usually it is equal to

700-850°C depending on the material of the flame tube and the operating conditions of the chamber. Experiments show that $Q_{\text{ф.л}}$ comprises the basic part of $Q_{\text{ф}}$. Nevertheless, in a number of cases it is impossible to ignore

$$Q_{\text{ф.к}} = \alpha_{\text{ф.тр}} F_{\text{ж.тр}} \Delta t_{\text{ср.ф}}, \tag{63}$$

although the computing of this value in general is complex. At the beginning it is necessary to determine the coefficient of heat transfer from the gas to the wall $\alpha_{\text{ф.тр}}$ and the average difference of temperatures $\Delta t_{\text{ср.ф}} = (T'_{\text{г.ф}} - T_{\text{ж.тр}})$, where $T'_{\text{г.ф}}$ — the average characteristic temperature of the gas-air flow. It can be considered that $T'_{\text{г.ф}} = T_{\text{в}} + C\,(T_{\text{г.ф}} - T_{\text{в}})$ where $T_{\text{в}} < T'_{\text{г.ф}} < T_{\text{г.ф}}$. Coefficient C is determined experimentally. Usually C = 0.5-0.8.

For calculation of $\alpha_{\text{ф.тр}}$ it is possible to use the equation

$$\alpha_{\text{ф.тр}} = 74{,}3 c_{p_{\text{г.ст}}} (g\mu_{\text{г.ст}})^{0.18} \frac{G_1^{0.82}}{d_{\text{ж.тр.э}}^{1.82}} \left(\frac{T'_{\text{г.ф}}}{T_{\text{ж.тр}}}\right)^{0.35}, \tag{64}$$

where heat capacity and molecular weight of the gas are taken at the temperature of the wall

$$Q_{\text{т.к}} = \alpha_{\text{т.в}} F_{\text{ж.тр}} \Delta t_{\text{срТ}}, \tag{65}$$

where $\alpha_{\text{т.в}}$ — coefficient of heat transfer from the flame tube to the cooling air, and the average effective difference of temperatures

$$\Delta t_{\text{срТ}} = \frac{t_{\text{ж.тр}} - t_{\text{в}}}{2{,}3 \lg \dfrac{t_{\text{ж.тр}} - t_{\text{в}}}{t_{\text{ж.тр}} - t'_{\text{в}}}}.$$

The temperature to which the cooling air will be heated $t'_{\text{в}}$, can be found by the method of consecutive approximations from the equation $Q_{\text{ф}} = Q_{\text{в}}$, i.e., $Q_{\text{ф.л}} + Q_{\text{ф.к}} = (\alpha_{\Sigma} - \alpha_{\text{г}}) L_0 G_T c_{p_{\text{в}}} (t'_{\text{в}} - t_{\text{в}})$. The average heat capacity of the air in the $t_{\text{в}}$ and $t'_{\text{в}}$ temperature range must be given. The temperature of the exterior surface of the flame tube is determined from the equation

$$t'_{\text{ж. тр}} = t_{\text{ж. тр}} - \frac{Q_{\phi}\delta}{\lambda F_{\text{ж. тр}}}\ ^{\circ}\mathrm{C}.$$

The value $\alpha_{\text{т. в}}$ is found from the expression

$$\alpha_{\text{т. в}} = \mathrm{Nu}\frac{\lambda}{d_{\text{ж. тр. э}}},$$

where λ – coefficient of thermal conductivity of air when $t = t_{\text{ж. тр}}$; δ – thickness of the wall of the flame tube; $d_{\text{ж. тр. э}}$ – equivalent diameter of the flame tube.

The Nu number can be calculated using the equation $\mathrm{Nu} = 0.018\,\mathrm{Re}^{0.8}$, where $\mathrm{Re} = \frac{w_2 d_{\text{ж. тр. э}}}{\nu}$.

In this way, for computation of $\alpha_{\text{т. в}}$ it is necessary to have given the value of the velocity of cooling air w_2.

The quantity of heat being transferred from the flame tube to the wall of the casing by radiation,

$$Q_{\text{т. л}} = \sigma_0 \varepsilon_{\text{п}} F'_{\text{ж. тр}}\left[\left(\frac{T'_{\text{ж. тр}}}{100}\right)^4 - \left(\frac{T_{\text{кр}}}{100}\right)^4\right], \tag{66}$$

where $F'_{\text{ж. тр}}$ – exterior surface of the flame tube;

$$F'_{\text{ж. тр}} = \pi(d_{\text{ж. тр}} + 2\delta)\,l_{\text{ж. тр}}.$$

Usually δ = 1.5-3 mm. The emissivity factor

$$\varepsilon_{\text{п}} = \frac{1}{\frac{1}{\varepsilon_{\text{ж. тр}}} + \frac{F'_{\text{ж. тр}}}{F_{\text{кр}}}\left(\frac{1}{\varepsilon_{\text{кр}}} - 1\right)},$$

where $\varepsilon_{\text{ж. тр}}$ and $\varepsilon_{\text{кр}}$ – the degree of blackness of the exterior surface of the flame tube and interior surface of the casing $F_{\text{кр}}$; $\varepsilon_{\text{п}} = 0{,}67-0{,}71$.

In the expression $Q_{\text{т.л}}$ there enters the temperature of the interior surface of the casing $T_{\text{кр}}$, which is given, considering that $T_{\text{кр}} = T_{\text{в}} + c'(T'_{\text{ж.тр}} - T_{\text{в}})$, where the coefficient $c' = 0.05\text{-}0.20$.

Substituting in equations (62)-(65) the accepted values, we check to see whether the equality $Q_{\text{ф}} = Q_{\text{ж.тр}}$ takes place. If equality is not obtained, it is necessary to change the accepted values of $t_{\text{ж.тр}}$ or w_2 and to repeat the calculation.

The very simple diagram examined in a number of specific cases should be augmented by special calculations which reflect the effect of film, jet or other cooling changing the temperature condition of the wall of the flame tube, or must be specifically defined, allowing for experimental research on heat exchange in chambers of a given design.

CHAPTER X

DESIGN, CALCULATION AND OPERATING CHARACTERISTICS OF COMBUSTION CHAMBERS

§ 40. Design of Combustion Chambers

Aircraft gas turbine engines [GTD] (ГТД), as a rule, have an annular, cannular, or sectional combustion chamber. Typical diagrams of the arrangement of combustion chambers on aircraft GTD with centrifugal and axial-flow compressors are presented in Fig. 56. Figure 57 shows an exterior view of an aircraft GTD with a centrifugal compressor, having a sectional combustion chamber, consisting of nine individual can-type chambers. The centrifugal compressor specified the slanted arrangement of the combustion chambers. The individual elements of the chamber can be seen in Fig. 58. The intake diffuser of chamber 1 is connected by a collar with the duct of the compressor 3 and the casing of the chamber 2. In two of the nine chambers starters have been installed, consisting of an electric igniter and starting injector. The starting device is shown separately in Fig. 59. The flame tube 4 is secured in casing 2 (Fig. 58).

The forward attachment of the flame tube, the diagram of which is given in Fig. 60, is carried out by three tubes attached to the casing, which enter the sleeves welded to the flame tube. The radial clearance between the face of the tube and the external surface of the flame tube permits it to expand freely (indicated by the dotted line) during heating without disturbing the centering in the casing. Two such tubes 5 of the chamber shown in Fig. 58 are used as interconnecting tubes for the transfer of the flames.

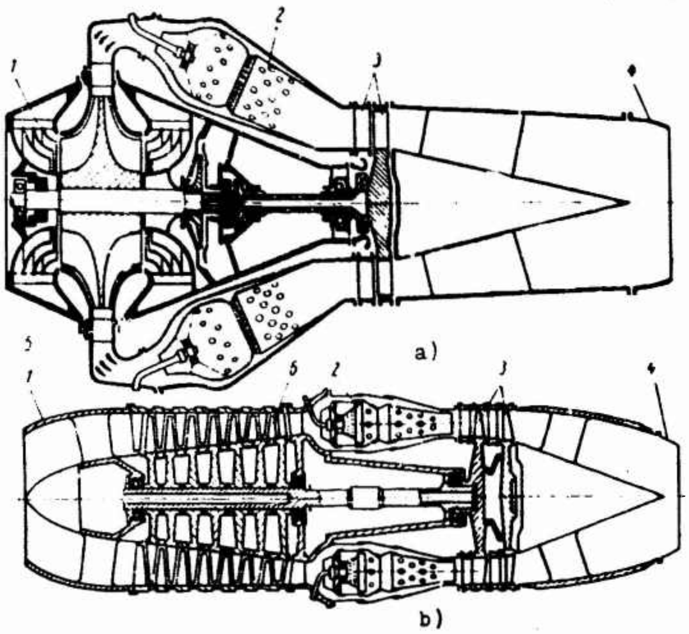


Fig. 56. Diagram of arrangement of combustion chambers on a GTD: a) with centrifugal compressor; b) with axial-flow compressor; 1 – intake; 2 – combustion chamber; 3 – turbine; 4 – outlet nozzle; 5 – centrifugal compressor; 6 – axial-flow compressor.

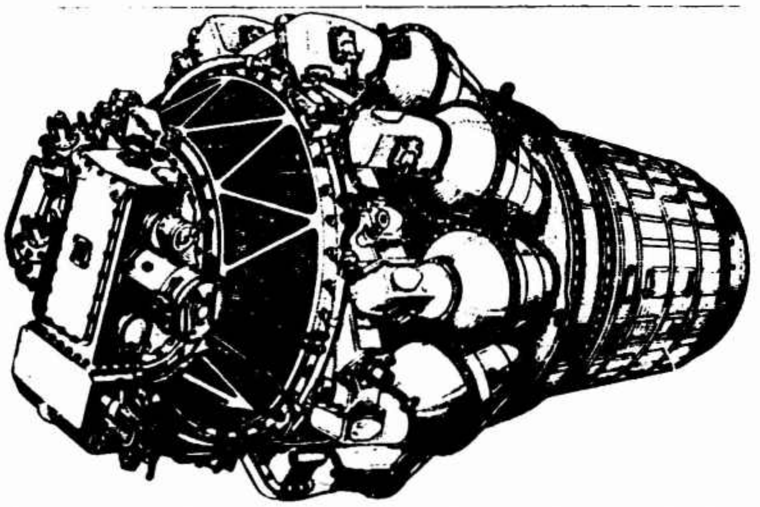


Fig. 57. Exterior view of an aircraft GTD with centrifugal compressor.

The forward part of the flame tube is attached to the casing and can expand in an axial direction by shifting in the sliding back support. The tail part of the flame tube on its circumference has eight stellite built-up collars 6, which slip on the annulus of the casing. The flame tube is made of EI435 sheet steel in a thickness of 1.2-2.5 mm. The individual parts of the flame tube are seam welded.

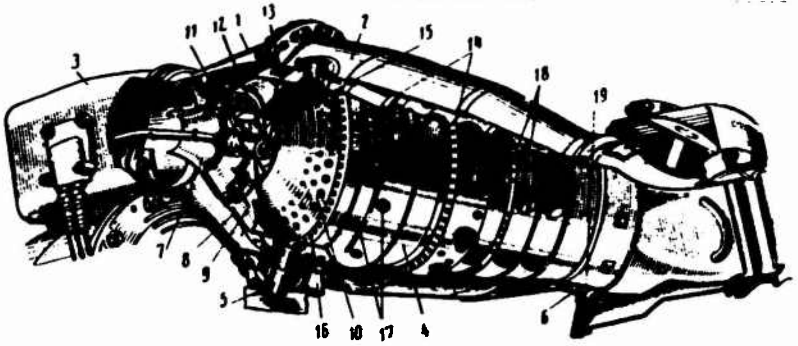


Fig. 58. Aircraft GTD combustion chamber.

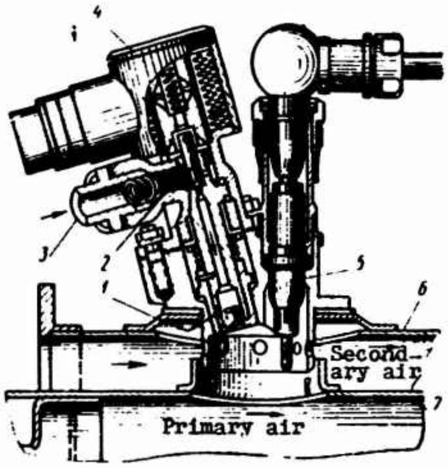


Fig. 59. Starter: 1 - injector; 2 - filter; 3 - pipe for fuel feed; 4 - electromagnetic valve; 5 - igniter; 6 - casing; 7 - flame tube.

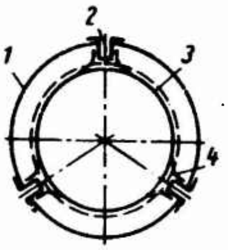


Fig. 60. Diagram of front attachment of flame tube: 1 - casing; 2 - tube; 3 - flame tube; 4 - sleeves.

The front device has an intake cone 7 for measuring out the air which goes to the front, and a hemispherical face shell 8 in the interior which there is located a perforated partition 9 and further, a conical shell 10 with several rows of openings.

In the center of the conical shell there is located a duplex, dual-nozzle centrifugal injector 11, which is set in the interior ring of the blade swirler 12, the ten blades of which are set at an angle of 50° to the axis of the chamber. In the place of the weld of the hemispherical shell with the conical cuts 13 are made along the axis of the chamber, which permits the hotter conical shell to freely expand.

The following, cylindrical section of the flame tube has a special stamping 14 for rigidity with openings of small diameter through which air enters, which cools the next part of the flame tube. Cylindrical shell 15 is cooled by air entering through openings 16 in the conical shell. Along the length of the cylindrical shell lateral openings 17 are made for supply of primary air. A part of the primary air enters through the following similar shell. Behind it goes a conical shell with openings of large diameter 18 – these are the mixing ports, through which cooling air enters. The casing of each of the nine chambers is mounted with negative allowance in the ring of the common gas collector 19 on intake to the turbine.

In case of unsuccessful starting, in all combustion chambers provision is made for the overflow of fuel from the lower points of the gas collector and the shell through a system of drainage valves. The heat density of the chamber comprises about 40·16 kcal/(m^3·h × × atm(abs.)). Pressure loss $\frac{\Delta p^*}{p_e^*} \approx 4\%$. The combustion efficiency of the fuel attains a value of $\eta_г \approx 0{,}97-0{,}99$.

A longitudinal section of an annular combustion chamber of an aircraft GTD with an axial compressor is shown in Fig. 61. Exterior 9 and interior 4 flame tubes are made from a stamping of sheet steel type EI652 with a thickness of 1.6 mm and welded to front part 12 which has twelve heads 2 with a blade swirler 15 and centrifugal injector 1 in the center. To the series of heads there are welded sleeves 14, into which pins 13 enter, fastening the front part of the flame tube.

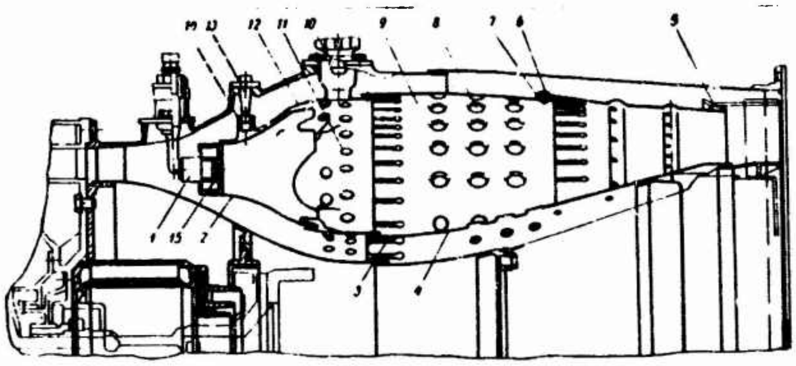


Fig. 61. Annular combustion chamber.

The tail part of the chamber slides by its collars 5 in the ring of the gas collector. The shells of the flame tube are connected through adapter plates 7 by rivets 6 and provide admission of an annular film of cooling air inside the flame tube. To reduce rigidity and to avoid splitting and warping, the ends of the shells have longitudinal cuts 3. The openings 11 and 8 for the feed of primary and cooling air are flanged or are compressed by special applied pistons for their protection against splitting. The chamber is provided with two igniters 10.

Figure 62 gives a general view of the flame tube of an annular combustion chamber of another aircraft GTD, where the mixer is made in the form of slot nozzles 1 on the interior flame tube and nozzles 4 on the external flame tube. The chamber has ten heads 3 with blade swirlers 2.

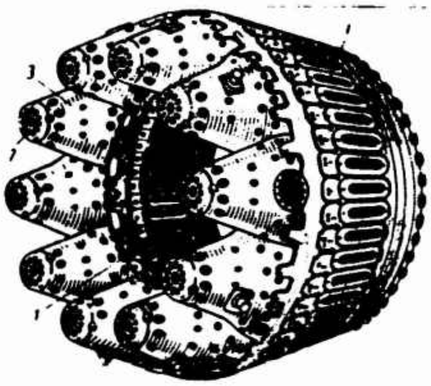


Fig. 62. Exterior view of a flame tube of an annular combustion chamber.

Figure 63 gives a diagram of the arrangement of a reverse-flow combustion chamber of an aircraft GTD with a centrifugal compressor. The centrifugal injector 1 is installed in the center of the front device 2, consisting of a plate with openings. Each of the ten chambers has an integral cylindrical flame tube 4 with rows of stampings 3 of the "grater" type and openings 5.

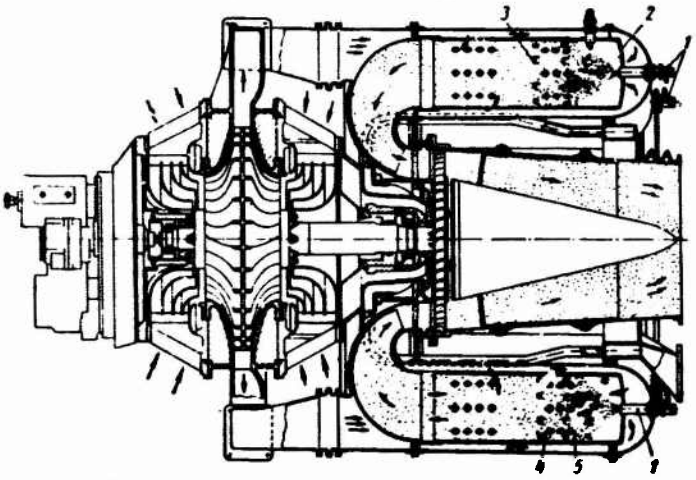


Fig. 63. Aircraft GTD with reverse-flow sectional combustion chambers.

Figure 64 shows the center part of the engine of the first gas-turbine locomotive from the Kolomna diesel locomotive plant with 6000 h.p. in two sections, with a sectional combustion chamber 1, having a front device 2 and two concentric blade swirlers. The blade angle of the swirler is equal to 45°. The number of individual combustion chambers n = 6, the basic fuel – mazut. On three of the six chambers for firing the mixture there is installed an igniter 3 similar to that presented in Fig. 59. The forward part of the telescoping flame tube, made out of sheet steel Kh23N18 with a thickness of 2 mm and consisting of a number of welded shells, is fixed in housing by six radial plates 4 welded onto the front device. The plates are compressed between flanges and have radial clearances which provide a capability of thermal expansion. The tail part of the flame tube slides on slits in an annulus of the housing.

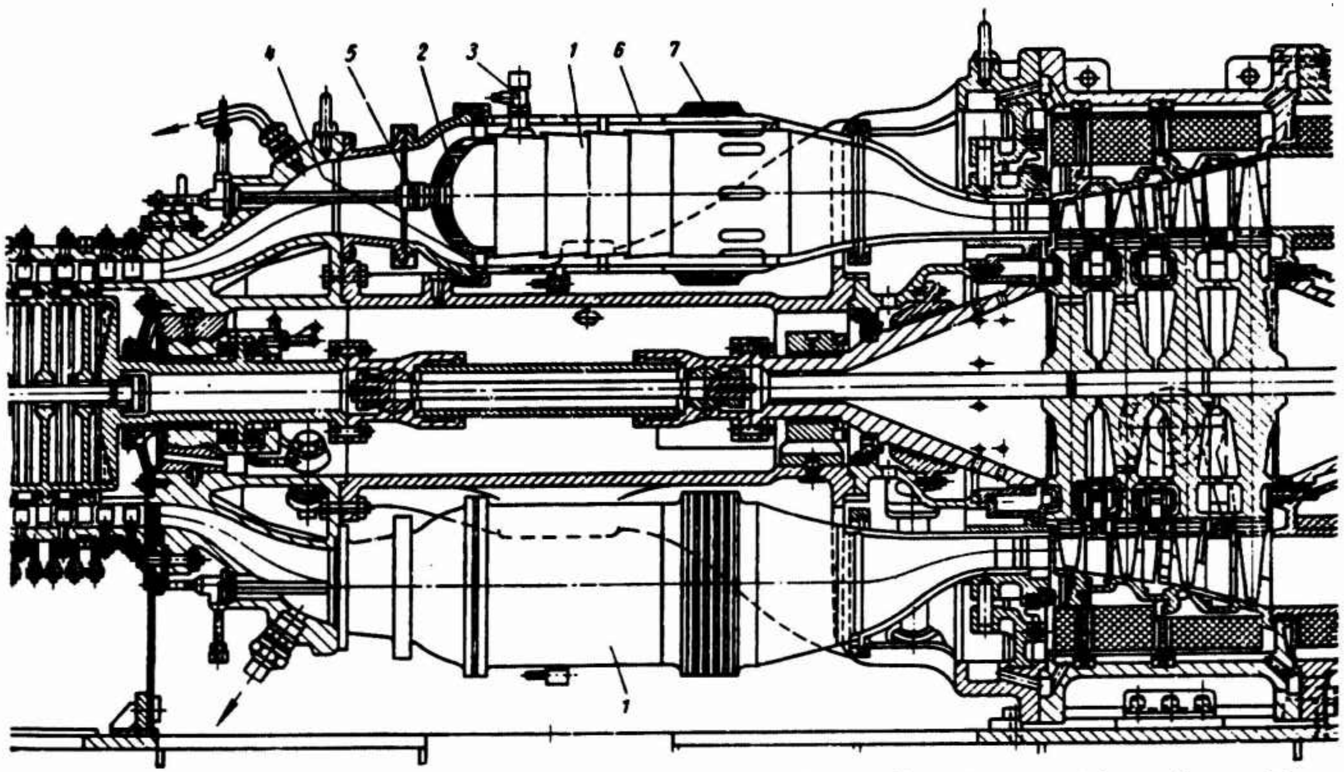


Fig. 64. Center part of a gas-turbine engine of a locomotive from the Kolomna plant.

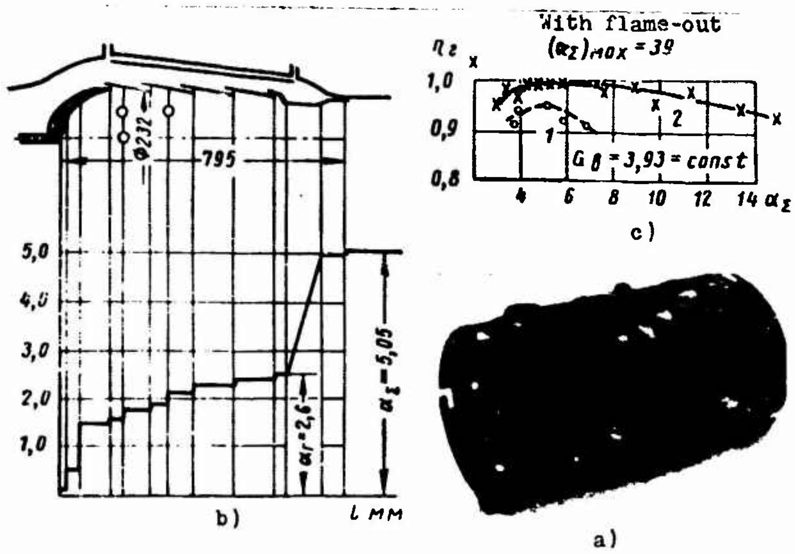
Along the length of the flame tube openings have been provided for the feed of primary air and for eight mixer slots. Injector 5 is centrifugal, dual-nozzle, duplex. On housing 6, made out of St. 3 steel with a thickness of 3 mm, there is a bellows 7 to receive thermal expansion.

Between the casing and the flame tube a screen is located made out of 1Kh18N9T steel 1.5 mm thick. The temperature of gases before the turbine is equal to 727°C with an excess air ratio $\alpha_{\Sigma} = 5.05$. Air pressure behind the compressor is equal to 6 atm(abs.), and the air temperature is 240°C. The air consumption through one chamber $G_{B} = 3.93$ kg/s. The average velocity of air in the chamber, calculated from the parameters of the air on outlet from the compressor is equal to 17 m/s; the velocity of the cooling air $w_2 \approx 50$ m/s. Conditional velocity of flow in the combustion zone $w_{Г} = 10$ m/s. The calculated combustion intensity of the working volume of the chamber is $12 \cdot 10^6$ kcal/(m^3·h·atm(abs.)). The excess air ratio in the combustion zone $\alpha_{Г} = 2.6$.

In each chamber there stands one duplex dual-nozzle centrifugal injector with a fuel cone angle $\beta_{T} = 90°$ and the maximum fuel consumption $G_{T} = 205$ kg/h at a fuel pressure equal to 56 kgf/cm^2.

The combustion chamber was tested on diesel fuel and mazut (F-12, FS-5 and a sulphurous mazut per Petroleum Industry Departmental Technical Specifications [VTU NP] (ВТУ НП) 427-55) with various expenditures and parameters of air from rated conditions to idling of the engine.

Figure 65b for rated condition gives a diagram of the distribution of air along the length of the flame tube; Fig. 65c gives a graph of the change in the combustion efficiency of the fuel $\eta_{Г}$ with a change of α_{Σ} ($G_{в} = 3.93 = \text{const}$). On partial loads even on diesel fuel this model of a chamber operated insufficiently well. Formation of carbon and partial warping of the shells of the flame tube were observed (Fig. 65a).



c)

b)

a)

Fig. 65. Results of research on the GTD combustion chamber of the Kolomna gas turbine locomotive: 1 – mazut; 2 – diesel fuel.

Another model of this GTD chamber is shown in Fig. 66a. Here the front device and the igniter are changed. The blade angle of the swirler is equal to 65°. The cone of fuel of the main jet is ignited by creeping discharge electric igniters 1 installed in three of the six chambers. Here in the capacity of a working fuel, mazut preheated to 120°C is used, and solar oil serves as the primary fuel. To reduce heat losses into the environment, between the housing and the flame tube a screen 2 is located made of sheet steel with a thickness of 1.2 mm. Tests of this model of a chamber were also conducted with diesel fuel (curve 2) and mazut (curve 1) and they gave very good results (Fig. 66b and c).

The combustion efficiency of the fuel η_{Γ} of this chamber does not drop below 0.92-0.93 even under partial loads. The temperature of the wall of the flame tube does not exceed 600°C; the degree of discontinuity of the temperature of the gases on outlet from the chamber δ = 8-12% (instead of 26-32% with the first model). The total head losses in the chamber did not exceed 2%. Starting the chamber which has the igniter is performed reliably in 5-8 s; the entire chamber unit starts in 20-40 s.

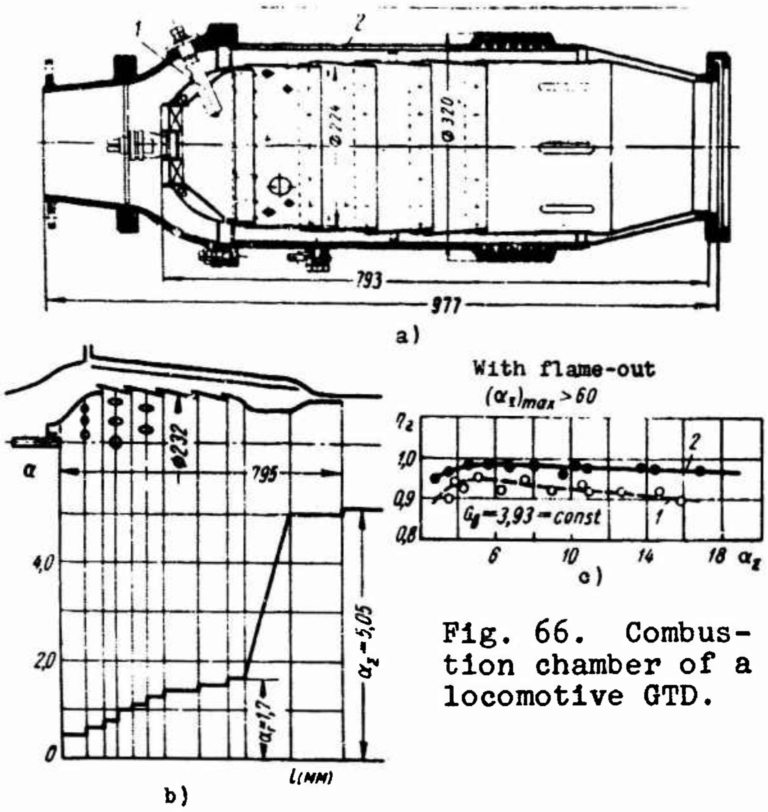


Fig. 66. Combustion chamber of a locomotive GTD.

One of the subsequent models of a GTD for a locomotive from the Kolomna diesel locomotive plant is developed with a counter-current combustion chamber (a reverse-flow chamber) according to the diagram presented in Fig. 67. The chamber is also sectional, using the basic parts of the direct-flow GTD chamber of the first model. With such an arrangement an access to the chamber is simplified and the length of the engine rotor is shortened.

A locomotive GTD, built in Czechoslovakia has one remote reverse-flow combustion chamber (Fig. 68) with a telescoping flame tube 1. Four centrifugal single-stage injectors are attached in tube 2 located in the center of blade swirler 3. The basic fuel – mazut, and the starting fuel – diesel fuel. Igniter 4 has an electric plug and booster injector.

An interesting design of an annular combustion chamber 1 is the marine GTD of 276 h.p. built by the French firm Turbomeca. A longitudinal section of the center part of the GTD is shown in Fig. 69a. From the centrifugal compressor 2 air with a pressure of

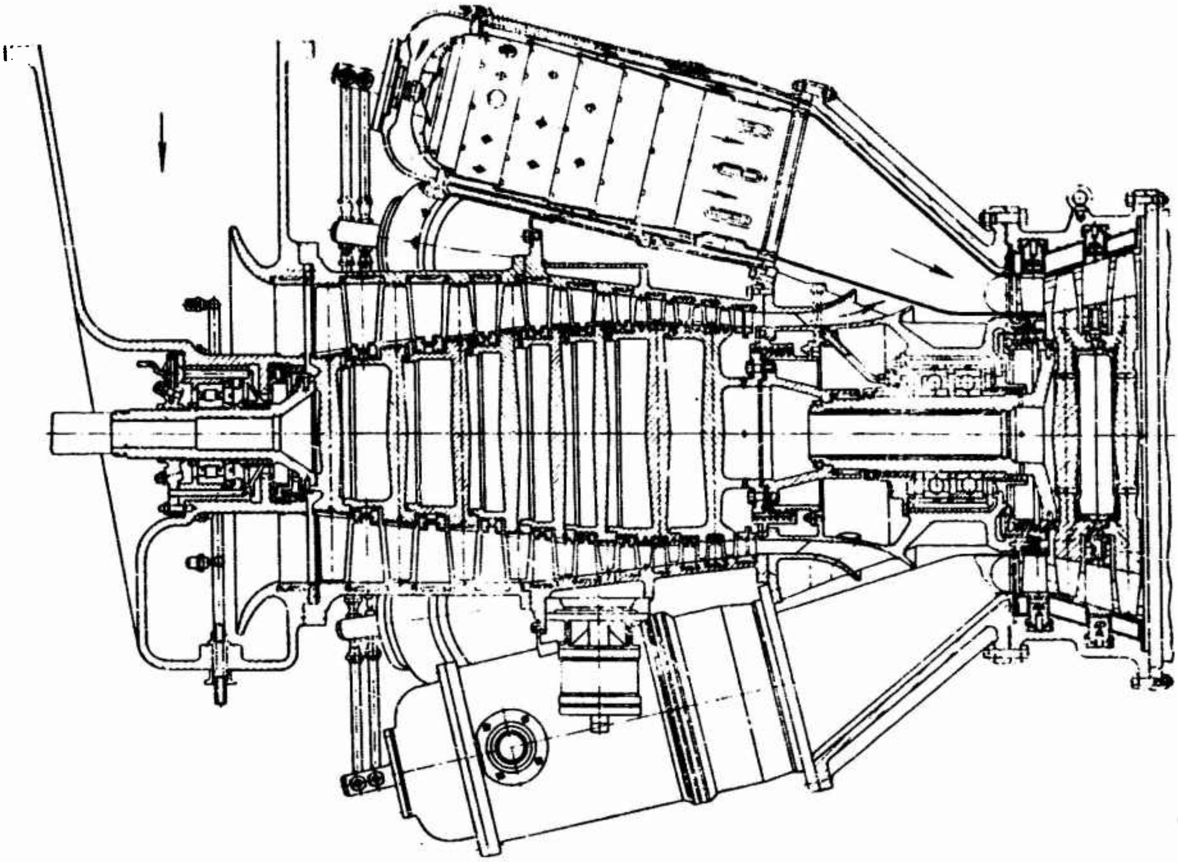


Fig. 67. Sectional reverse-flow combustion chamber of a locomotive GTD.

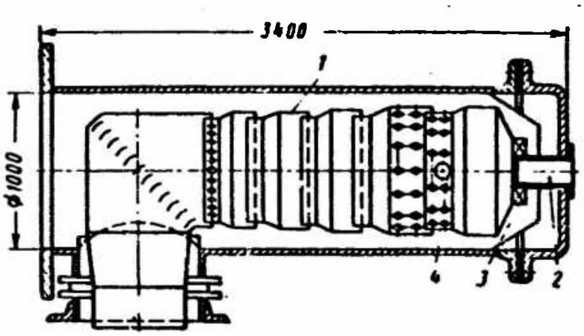


Fig. 68. Remote reverse-flow combustion chamber.

3.7 atm(abs.) enters annular shell 3 and then through a series of openings and slots inside the toroidal flame tube 4. The temperature of gases on outlet from the combustion chamber equals 800°C. Gas oil serves as the fuel. For starting, an electric igniter 5 is used, which ignites the circular fan of fuel 6 which is delivered via tube 8 and is sprayed through the openings of rotor 7 which turns at 35,000 r/min.

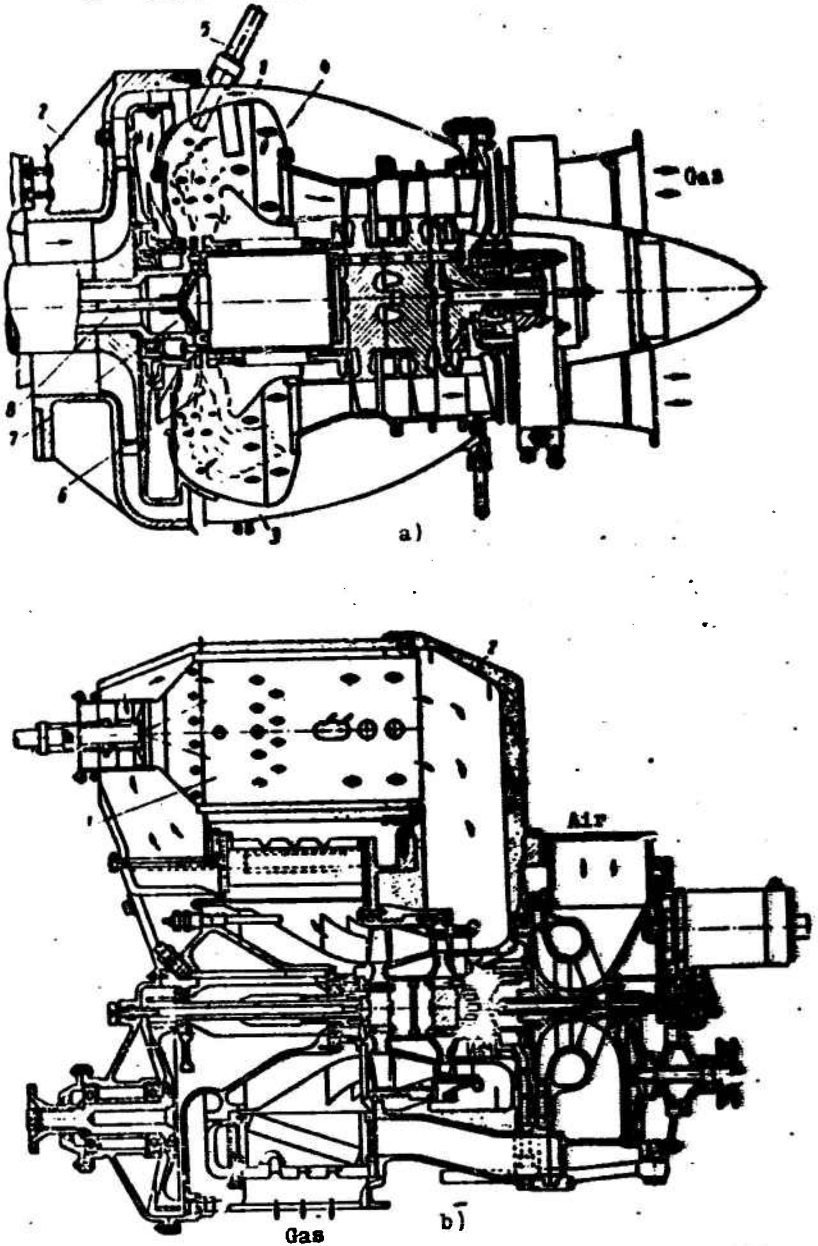


Fig. 69. Diagrams of GTD: a) marine with annular combustion chamber; b) of a seagoing launch with individual combustion chamber.

Fuel is supplied at low pressure and is apportioned by a special valve. Starting time of the GTD does not exceed 20 s. Fuel consumption amounts to 0.471 kg/(horsepower hour).

Figure 69b provides a longitudinal section of a GTD of a seagoing launch of N_e = 200 horsepower with an individual combustion chamber 1 the housing of which has thermal insulation 2 to reduce heat losses into the environment.

Figure 70 shows versions of the remote combustion chamber of the GT-25-700 of the Leningrad Metal Plant for operation on liquid fuel and gas. The flame tube – telescoping with openings in the shells for the supply of primary air. The shells of the flame tube are fastened on longitudinal lathes 1, rigidly connected in the upper part of the housing. Along the length of the gas collector supplementary sliding hangers 2 are provided. The mixer is made in the form of slit nozzles 3, which are protected against flame action by the deflectors 4, under which the cooling air enters.

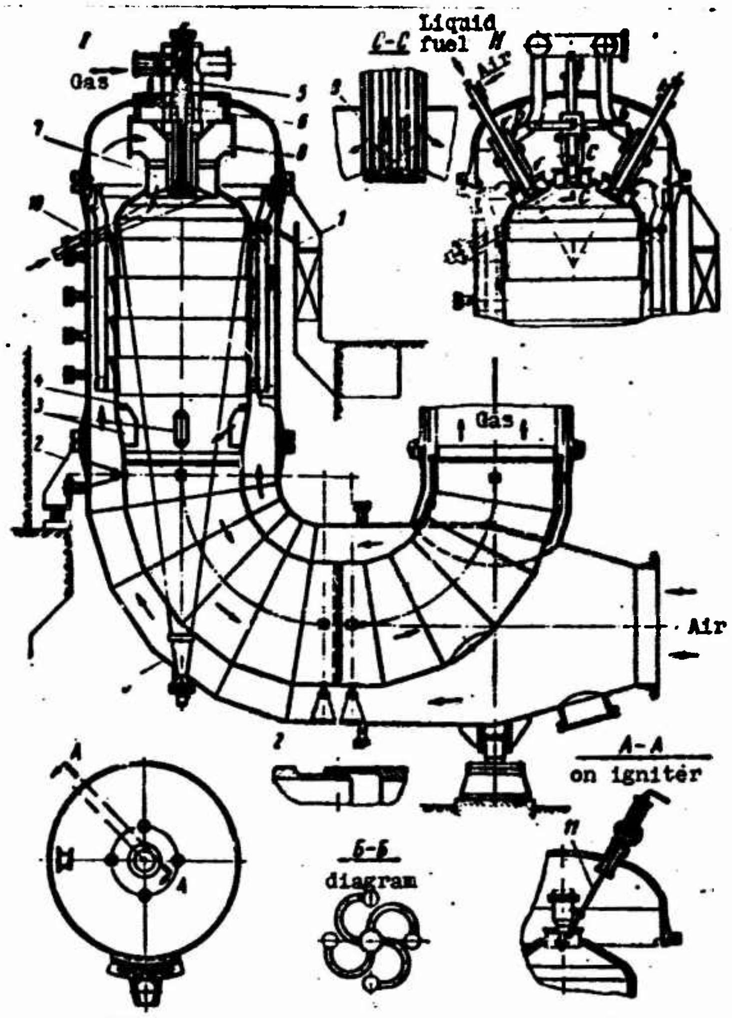


Fig. 70. Remote reverse-flow combustion chamber GT-25-700.

The chamber of version I to operate on gas has in the center of the basic gas burner an electric plug 5 and the liquid-fuel starting injector. Around it is located annulus 6 with the ribs of the pilot gas burner to which a gas-air mixture of constant composition is fed under all conditions, which ensures steady combustion of the basic gas cone. The following annulus 7 serves for supplying gas to the main burner representing alternating interblade channels of the swirler for air and gas. Air from the annular cavity of the chamber enters those channels through a special elbow-shaped inlet 8, and the gas – through ports 9 (version II, cross section *C-C*). In the end of the chamber at the outlet and along its length there are installed inspection ports 10 for visual observation of the operation in the combustion zone. Version II of the chamber has been developed to operate on gas and liquid fuel. In the center of each five gas burners a liquid fuel injector is installed. The electrodes of igniter 11 (section *A-A*) pass between the swirler vanes.

In the GTD combustion chambers solid fuel can also be used, particularly, coal, peat, and others.

At the Baumann Moscow Higher Technical School [MVTU] (MBTY) a device has been developed for burning coal dust in a direct-flow combustion chamber, a diagram of which is shown in Fig. 71. In the chamber Moscow brown coal dust and Donets gas coal dust are burned, giving a residue $R_{70} \approx 20-30\%$ on a No. 70 sieve. The length of the flame tube 5, depending on the fuel and the conditions of the experiment, can be changed from 795 to 1963 mm. The diameter of the flame tube $d_{ж.тр} = 232$ мм. Fuel consumption is changed from 7 to 50 kg/h. Air pressure attains 1.5 atm. Coal dust in the primary air flow part in a ratio (by weight) of approximately 1:1 is delivered via pipe *I* to injector 3 shown in Fig. 72. This air pressure at 0.2-0.6 atm(abs.) exceeds the air pressure in the chamber. The dust-laden air flowing around the cone enters the combustion zone of the chamber. Additionally, part of the primary air on pipe *II* (see Fig. 71) and the center annulus between walls 6 and 7 at first enters the lower part of the flame tube, and then, having cooled it, rises the blade swirler 4 of the front device. Secondary air via pipe *III* and the external annulus goes to adjustable nozzle 8 of mixer 9.

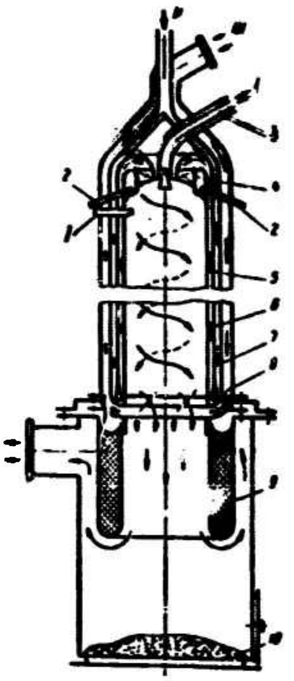


Fig. 71. A coal dust combustion chamber.

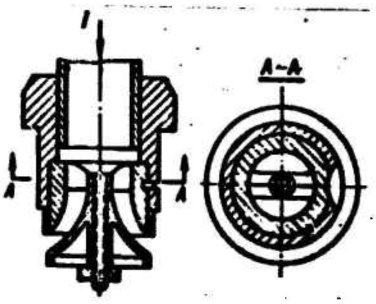


Fig. 72. A coal dust injector.

The chamber operates with "dry slag," i.e., the temperature rate is maintained at such a level that the slag will not melt on outlet from the working volume. A basic portion of the fraction of solid slag is borne by the gas flow from the chamber into the purifying system and a certain part sinks into slag-collector 10. The chamber is started by an electric igniter 1, which ignites the cone of liquid fuel of the starting injectors 2. The system for purifying the gas from slag has two steps. The coarse cleaning step is an inertial louvered device (Fig. 73). Trapped slag 3 with a small quantity of gas is directed into a slag-separating cyclone of the type shown in 7 (Fig. 76), where the slag is deposited and the gases return to the cleaning current in this step. Further, the gases proceed into the fine cleaning step – the turbocyclone, a diagram of which is shown in Fig. 74. Initially, the rotor 1 of this equipment, together with

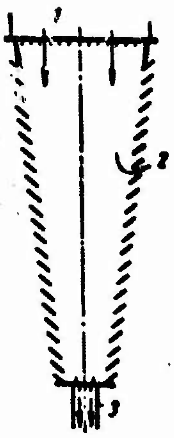


Fig. 73. Diagram of a louvered gas purifier: 1 – dust-laden flow; 2 – purified flow; 3 – trapped dust.

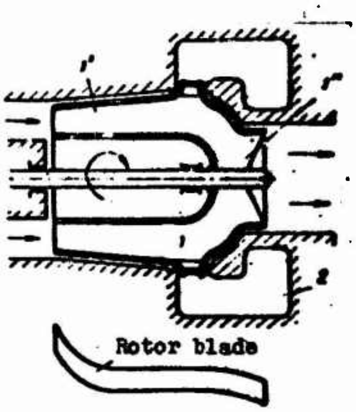


Fig. 74. Diagram of a turbocyclone gas purifier.

compressor blades 1' was put into rotation by the gas flow passing through turbine blades 1". Later a drive was made from a special electric motor. Under the action of centrifugal forces, the solid particles of slag are transported to the periphery and fall, together with a small quantity of gas, into the slot and then into the circular dust-collector 2, whence the gas is drawn off through an additional cyclone, and is introduced into the purified flow. With an average rotational speed of the bladed turbocyclone equal to 50 m/s, the overall coefficient of cleaning reaches 85%, whereby all particles larger than 20 μm are trapped. As the experiment shows, particles smaller than 20 μm do not cause significant damage to parts of the flow-through part of the turbine and in practice do not reduce the life and reliability of the operation of the engine. Figure 75 shows the dependence of the combustion efficiency coefficient $\eta_г$ on the excess air ratio $\alpha_г$ in the combustion zone for Dorets gas coal dust (curve 1) with a length of flame tube $l_{ж.тр} \doteq 796$ мм and for Moscow coal (curve 2) with $l_{ж.тр} = 1963$ мм.

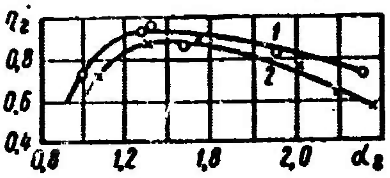


Fig. 75. Dependence $\eta_z = f(\alpha_z)$ for the coal dust combustion chamber.

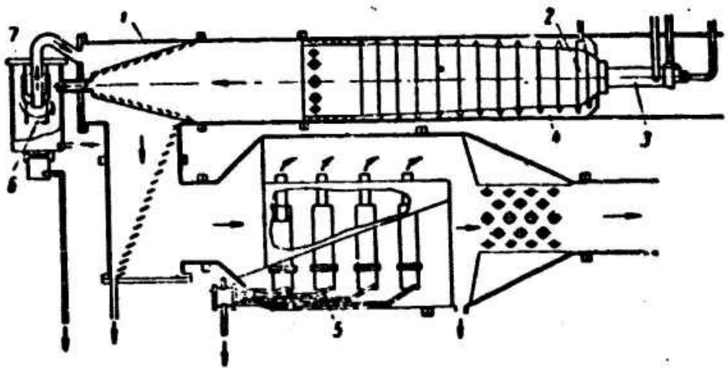
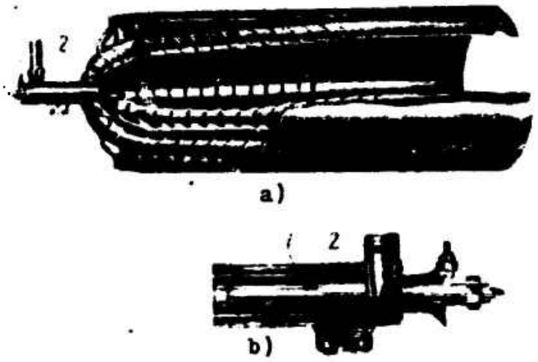


Fig. 76. Diagram of the combustion chamber and the purifying system of the Alco gas turbine locomotive.

The American firm Alco-Allis-Chalmers built a gas-turbine locomotive operating on coal dust with a direct-flow combustion chamber with "dry" slag removal. A diagram of this combustion chamber and of the two-stage purifying system is shown in Fig. 76. Chamber 2 has a flame tube of the telescoping type with row of circular shells fastened by four longitudinal saw-tooth plates 4. In the center of the front device there is composite injector 3, the construction of which and its position in the chamber are shown in Fig. 77.



a)

b)

Fig. 77. Coal dust combustion chamber and injector.

The central tube 1 (Fig. 77b) ends in the liquid fuel (solar oil) starting injector, which simultaneously also serves as the pilot injector, which constantly ensures the steady combustion of the solid fuel. The dust-air mixtures enter the cavity around the starting injector, swirling as a result of the tangential feed 2 of the flow (Fig. 77a).

The first step in purifying is made in the form of a louvered ash-remover 1 (Fig. 76). The trapped slag with a small quantity of gas tangentially enters cyclone 7 and descending in a spiral around central tube 6 sinks to the bottom of the cyclone, while the gas via central tube 6 returns to the basic gas flow.

The second purifying step 5 is the battery of cyclones 7.

The gas turbine locomotive using solid fuel underwent a rather long period of experimental operation, in the course of which tests indicated sufficiently reliable operation of the combustion chamber and the purifying system.

Figure 78 shows a diagram of a cyclone coal dust combustion chamber of the I. I. Polzunov Central Boiler and Turbine Institute [TsKTI] (ЦКТИ) with liquid slag removal. The dust-laden air is fed to the injector via channel 2, in the center of which the liquid fuel starting injector 1 is located. The basic mass of air for combustion is fed in via tangential channel 7. The cyclone chamber is made from sheet steel with rows of welded metal radial pins, extending both inside and outside the chamber. The interior part of the chamber is coated with a fire-resistant chromite mass. The pins on the outside are cooled by the air which enters through branch connection 3, and goes between the interior pin-studded and external (solid) housing, and further – to mixing chamber 4. The liquid slag flows off through tap 6 into slag bunkers 5, which have a water seal and locking mechanism which ensures continuity of operation of the slag removal system.

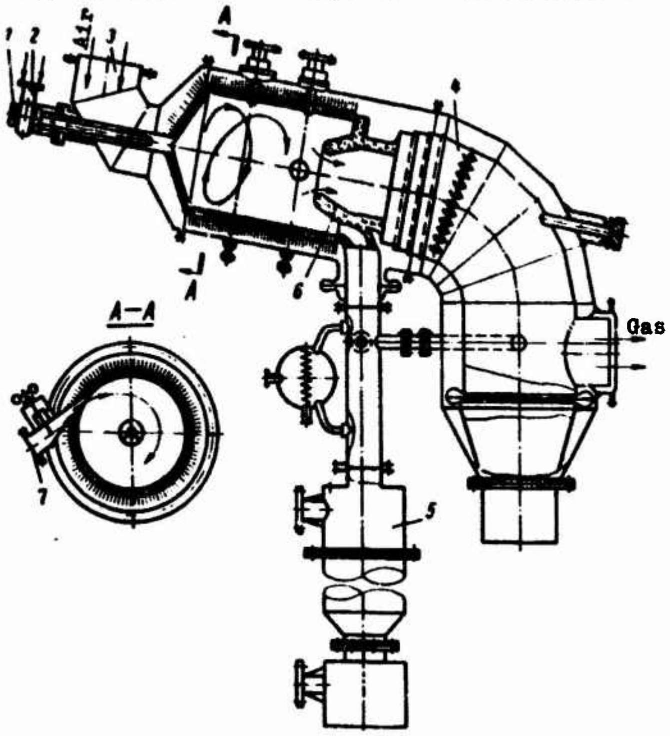


Fig. 78. Diagram of a cyclone coal dust combustion chamber.

It must be noted that the possibility of using solid fuel is one of the most important advantages of gas turbine engines over internal combustion piston engines.

§ 41. Basic Parameters of Combustion Chambers

The change in the basic parameters of the working medium along the length of the combustion chamber is shown schematically in Fig. 79. In the central part of the chamber a decrease in velocity of flow w_{cp} is characteristic. This makes it possible to increase the time of stay of the mixture in the combustion zone

$$w_{cp} = \frac{G_{в} v_{в}}{F_{cp}},$$

where $G_{в}$ – total weight expenditure of air in the chamber; F_{cp} – area of cross section of the housing according to the internal diameter of the chamber $D_{к}$, less the area of section of the walls of the flame tube; $v_{в}$ – specific volume of air on intake to the combustion chamber.

Velocity w_{cp} is an arbitrary value, since the specific weight is taken from the parameters of air on intake, but from the value of F_{cp} the area of cross section of the reverse current zone is not deducted, where there is no direct flow along the axis. However this is a convenient average characteristic of the design of the chamber, which for a given class of GTD fluctuates within rather narrow limits. In exactly the same manner it is possible to conditionally compute the average velocity of flow in the interior of the flame tube $w_{\text{г}}$, determining it according to the entire interior area of cross section of the flame tube, the expenditure of primary G_{I}, and the specific weight of air on intake to the chamber.

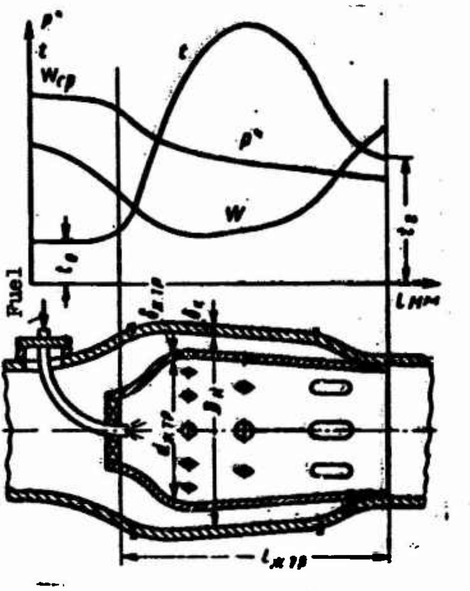


Fig. 79. Change in basic parameters along the length of a combustion chamber.

In planning a new design of a combustion chamber the values of conditional velocities w_{cp} and $w_{\text{г}}$ are taken within limits characteristic for chambers of this class of GTD. This permits calculating the initial interior diameter of the flame tube and the casing.

In Table 11, according to the data of a number of completed designs of combustion chambers of various classes of engines, tentative values of the quantities mentioned are given. Furthermore, in the table, values are shown for combustion intensity H, the velocities of cooling air w_2 in the clearance between the flame tube

Table 11. Basic parameters of GTD combustion chambers.

Gas turbine engines	Fuel	Velocities in m/s				$\alpha_{фр}$	$\alpha_г$	$H \cdot 10^{-6}$ in kcal/(m^3 × h × atm(abs.)) $\left(\frac{kJ}{m^3 \cdot h \cdot atm(abs.)}\right)$	$\left(\frac{l}{d}\right)_{ж.тр}$	α_Σ
		$w_г$	w_t	$w_з$	$w_{ср}$					
Aircraft	Kerosene	70—100—(120)	17—27	60—100	20—45	0,3—0,5	1,2—1,7—(2,2)	30—70 (125—290)	2,0—3,0*	(3)—3,5— 4,5—(5)
Stationary and transport	Heavy liquid and gas	60—80—(100)	7—17	50—80	15—25	0,3—0,5	1,5—2,2	5—30 (21—125)	2,5—3,5	(4)—4,5— 5,5—(6)
Stationary and transport (direct flow) with "dry" slag	Coal dust	55-80-(100)	4—8	50—70	10—20	0,5—0,8	1,4—2,2	3—15 (13—63)	3—4	4,5—6,0
with "liquid" slag	—	55—80—(100)	4—8	60—80	10—20	0,2—0,5	1,1—1,4	3—15 (13—63)	3—4	4,5—6,0

---

*For cannular chambers and circular combustion chambers up to 4.

and the casing (or shield), the excess air ratios $\alpha_{фр}$ for the expenditure of air going through the swirler, $\alpha_г$ in the combustion zone and α_Σ of the total, the relationship of length to diameter of the flame tube $\frac{l}{d_{ж.тр}}$, and also velocity $w_г$ in the openings of the flame tube.

§ 42. System of Calculation of Combustion Chambers

In planning a combustion chamber for a given gas turbine engine, the following values must be known:

1) total expenditure of air in the engine $G_{в\Sigma}$;

2) temperature of air on intake to the combustion chamber $t_в$;

3) pressure of air on intake to the chamber $p_в$;

4) temperature of the gas on outlet from the chamber $t_{\text{г}}$;

5) kind of fuel;

6) class (purpose) of the engine.

Knowing the kind of fuel, it is possible to determine from handbooks its composition and lower working heat of combustion at standard temperature. If such data are lacking, the heat of combustion Q^p. just as the stoichiometric quantity of air L_0 kg/kg can be calculated from expressions (4) and (5).

For a given class of engine a type of combustion chamber is selected. A sectional combustion chamber is accepted, then further calculation is conducted on one of the n chambers (sections) on air expended through it $G_{\text{в}} = \frac{G_{\text{в}\Sigma}}{n}$. The number of chambers n should be selected optimally both from the point of view of the arrangement of the whole engine, its dimensions on midsection, and from the point of view of operating conditions and the characteristics of the chamber itself. Each chamber should be spaced from the neighboring one by a distance which ensures reliable operation of the branch connections for the transfer of flames. Division of the general flow of air into n parts should not lead to the necessity of using injectors of excessively small dimensions. The total surface of ducts and the number of individual working volumes should not increase too much pressure losses and the nonuniformity of gas temperature after the chamber.

The chamber is calculated according to the following scheme.

1. For determination of the composition, heat capacity of the working gases, and the expenditure of fuel through the combustion chambers, a total excess air ratio α_{Σ} is computed according to expression (57), preliminarily assigned by the value of the coefficient of combustion efficiency of the fuel $\eta_{\text{г}}$. The value of $\eta_{\text{г}}$, as a rule, must be taken within limits of $\eta_{\text{г}} = 0.97$-0.99. Knowing

α_Σ, g_T is determined according to expression (58), and c_{p_Γ} according to expression (60) and, finally, the weight (mass) expenditure of fuel $G_T = g_T \cdot G_в$.

2. For determination of the internal diameter of the flame tube, values are assigned of average conditional velocity and excess air ratio in the combustion zone: w_Γ and α_Γ (for example, in Table 11). Then

$$G_I = \frac{\alpha_г}{\alpha_\Sigma} G_в,$$

whence the area of cross section of the flame tube for the midsection

$$F_{ж.тр} = \frac{G_I v_г}{w_г}$$

and the internal diameter of the flame tube

$$d_{ж.тр} = \sqrt{\frac{4F_{ж.тр}}{\pi}}.$$

3. For determination of the length of the flame tube values are assigned for the combustion intensity of the chamber working volume H. Then the working volume

$$V_к = \frac{G_T Q_н^р \eta_г}{H p_к} м^3.$$

Knowing the volume and area of the flame tube, they determine its length $l_{ж.тр} = \frac{V_к}{F_{ж.тр}}$ and they check the ratio $\left(\frac{l}{d}\right)_{ж.тр}$, which must be selected as characteristic for the given class of engines. If the ratio $\left(\frac{l}{d}\right)_{ж.тр}$ exceeds the limits characteristic for this class of engines, then it is necessary to reexamine the choice of values of α_Γ, w_Γ and H.

4. For the determination of the internal diameter of the casing, thickness of the walls of the flame tube $\delta_{ж.тр}$ and velocity w_2 of secondary air in the clearance between the flame tube and the casing

(shield) are assigned. The value of $\delta_{\text{ж.тр}}$ in chambers of aircraft and transport GTD is usually equal to 1.5-3 mm; in stationary GTD for remote chambers attains 4-5 mm.

The area of section of the annular clearance,

$$F_2 = \frac{G_{II} v_2}{w_2},$$

where $G_{II} = G_в - G_I$, G_I — the part of the primary air introduced inside the flame tube up to cross section on the midsection of the flame tube, where its diameter is equal to $d_{\text{ж.тр}}$. This part of the air can be found, having been assigned by the distribution of primary air along the length of the combustion zone, for example, as shown in Fig. 44. This simultaneously gives the law of change in the coefficient of heat elimination along the length of the combustion zone (Fig. 48b). First it is necessary from the overall value of $l_{\text{ж.тр}}$ to separate the length of the combustion zone and the length of the mixing zone. As a rule, the length of the combustion zone comprises 55-65% of $l_{\text{ж.тр}}$.

Further, knowing F_2, they determine the internal diameter of the casing

$$D_к = \sqrt{\frac{4}{\pi} F_2 + (d_{\text{ж.тр}} + 2\delta_{\text{ж.тр}})^2}.$$

This stage of the calculation is completed by determining from the cross sections of the flame tube the number and diameters of the openings for the feed of primary, cooling, and secondary air allowing for the remarks mentioned above (see Fig. 46b, 51a and b). Let us remember that the air entering through the openings inside the flame tube executes various functions, and this determines the place and the method of supplying it.

Air for internal cooling of the flame tube is supplied along the interior surface of the wall to a depth of several mm, either by a continuous circular film, or through a great number of small openings (with diameters of 2-5 mm with pitch of 15-30 mm on circumference). These openings frequently are located on special stampings (ridges) on the flame tube. The primary air which takes part in the oxidation of the fuel is supplied in this part of the flame tube deeper (to 0.5-0.7 of the radius of the flame tube), distributed in bands. The diameter of the openings here are greater $(d_{\text{отв}} = 12 \div 30 \text{ мм})$. The number of openings in a band depending on the diameter of the flame tube, can be taken as $n_{\text{отв}} = (4) - 6 -$ 12 - 16 - (20), respectively, with $d_{\text{ж.тр}} = (150) - 200 -$ 300 - 400 - (450) mm. Air is fed into the mixing zone for cooling the gases. The depth of penetration of the air jets h in the mixer is greater still (up to 0.95-1.0 of the radius of the flame tube).

As indicated in § 35, the expenditure of air through the openings of the flame tube is determined by the expression

$$G_{\text{отв}} = \mu \frac{\pi d^2_{\text{отв}}}{4v_{\text{в}}} n_{\text{отв}} w_{\text{г}},$$

where $w_{\text{г}}$ – velocity of air in the openings.

The depth of penetration of the jets is determined according to the empirical equation $h = d_{\text{отв}} \left[0{,}3 + 0{,}415\left(\frac{w_{\text{г}}}{w_{\text{ж.тр}}}\right)\right]\cdot\left(\frac{l}{d_{\text{отв}}}\right)^{0{,}85}$. After calculation of the openings a design is selected for the front device of the flame tube.

5. Knowing the geometry of all the ducts of the combustion chamber, temperature rate of the walls of the flame tube are calculated according to the diagram given in § 39. This calculation determines the correctness of the choice of value of w_2. If the temperature of the wall, allowing for the supplementary effect of the internal film cooling is unsatisfactory, the value of velocity w_2 accepted in 4 must be changed.

6. Knowing the system of the design and the basic operating parameters of the combustion chamber, determine the distribution of the airflows and the total head loss by individual channels and by the chamber as a whole (according to the diagram shown in § 37).

This finishes the calculation of the chamber. Further on the elements and parts of the design are worked out: the fastening of the flame tube, the system of connection of its individual shells, the design of the injector, the fuel feed system, and others. After planning, the stage of experimental research and modification of the chamber usually follows.

§ 43. Experimental Research and Characteristics of Combustion Chambers

The experimental research on a newly created combustion chamber should determine whether the actual characteristics of its operation correspond to those necessary and which were obtained in the calculations and planning. If the actual characteristics are worse than calculated, then modification actions are performed, in the process of which, by changing the individual elements of the design, the distribution of air along the length of the chamber, the twisting of the flow in the bladed swirler, the velocity of flows via ducts and in openings and others, the necessary indices of operation are attained. An experimental installation should have a source of compressed air and system of equipment for measuring the parameters which determine the characteristics of the combustion chamber, of which the most important is the value of the coefficient of fuel combustion efficiency η_{Γ}.

It is necessary to determine the following values:

1) consumption of air G_{B} in kg/s;

2) temperature of air on intake to the combustion chamber T^{*}_{B} in °K;

3) the total air pressure on intake to the chamber p^{*}_{B} in kgf/cm^{2};

4) fuel consumption G_{T} in kg/h;

5) the temperature of the fuel before the injector $T^*_\text{Т}$ in °K;

6) the total pressure of gases on outlet from the chamber $p^*_\text{г}$ in kgf/cm^2;

7) the temperature of gases on outlet from the chamber $T^*_\text{г}$ in °K.

For measurement of air consumption it is necessary to use standard flow measuring devices, for example, nozzles, diaphragms, and so on, and to install them on a straight section of duct. $G_\text{в}$ is computed according to the known expression

$$G_\text{в} = \alpha_p \varepsilon F_p \sqrt{\frac{2g\Delta p_p}{v_\text{в}}} \text{ kg/s},$$

where α_p – the flow coefficient; ε – the coefficient of jet compression; F_p – the cross section area of the flow-measuring device in m^2; Δp_p – the difference in static pressures in front of and after the flow-measuring device in kgf/m^2; v_p – the specific volume of air before the flow-measuring device in m^3/kg.

For computing v_p, the static pressure and air temperature are measured. The air temperature is usually measured at a couple (two to three) points on cross section, most often by thermocouples (for example, chromel-copel), and later is averaged. At an even field of temperatures and velocities on inlet to the flow-metering device its average arithmetical value can be taken.

The static pressure (excess) is measured in the annular header, which is connected by drillings with the air manifold. For computing absolute pressure $p_\text{в}$, the barometric pressure is measured. The total head of air on intake to the combustion chamber is determined by the total head pipes. Fuel consumption is determined by weighing.

Determination of the parameters of the gas on outlet from the chamber $(p^*_\text{г}, p_\text{г}, T^*_\text{г})$ presents considerable difficulties in view of the fact that the field of temperatures and velocities of the flow are usually not uniform here. To obtain accurate values of the average values of parameters, a great number of measurements are performed and the readings averaged, whereby calculation is made of the expenditure of gas passing through that part of the section, where these parameters were measured. Most often, the temperature of the gas is determined as an average weight $(T_{\text{г}}^*)_{\text{ср}}$, considering static pressure for the section of the measuring section as constant. For determination of this temperature, the cross section of the area of the measuring section are broken down into a sufficiently large number of equal sections and on each of them (usually at four points in respect to two mutually perpendicular diameters) the temperature and total pressure of the gas are measured.

The readings of the thermocouples of each section are averaged, usually arithmetically, just as the readings of the total head pipes.

For each section the value of impact head is determined

$$q_i = p_i^* - p_i = \frac{w_i^2}{2g v_i} \text{ kg/m}^2$$

The weighted-mean temperature of gases on outlet from the combustion chamber

$$(T_{\text{г}}^*)_{\text{ср}} = \frac{\int_F (T_{\text{г}}^*)_i \, dG_i}{\int_F dG_i} \text{ °K}$$

where $(T_{\text{г}}^*)_i$ — stagnation temperature of the gases, average for the given area of section dF_i; $dG_i = \frac{w_i dF_i}{v_i}$ — the weight expenditure of gas through section dF_i; w_i - the velocity of the gas flowing through a given area in m/s.

Considering that impact pressure $q_i = \frac{w_i^2}{2g v_i}$, the value of area $dF_i = \pi d(r^2)$ and the specific volume of gas $v_i = \frac{RT_i}{p_i}$ (where p = const),

$$(T_z^*)_{cp} = \frac{\int_0^{r^2} (T_z^*)_i \sqrt{\frac{q_i}{(T_z^*)_i}}\, d(r^2)}{\int_0^{r^2} \sqrt{\frac{q_i}{(T_z^*)_i}}\, d(r^2)}.$$

The value of $(T_z^*)_{cp}$ is determined from the graph of dependences $\left[(T_z^*)_i \sqrt{\frac{q_i}{(T_z^*)_i}}\right]$ and $\left[\sqrt{\frac{q_i}{(T_z^*)_i}}\right]$ on r^2. The value of $(T_z^*)_{cp}$ is obtained as partial from the division of areas under the curves.

The dependence $\eta_г = f(\alpha_\Sigma)$ when G_B = const and G_T = var, as presented in Fig. 80a (curve 1) serves as the general characteristic of the chamber reflecting combustion efficiency of the fuel. Greatest combustion efficiency $(\eta_г)_{max}$ with $(\alpha_\Sigma)_{opt}$ occurs between the blowout limits of the flame as a result of considerable impoverishment (α_{max}) or enrichment by fuel (α_{min}) of the combustibel mixture. A number of factors have an effect on the value $(\eta_г)_{max}$, character and position of the curve $\eta_г = f(\alpha_\Sigma)$. The determining ones are the distribution and the means of supplying primary air G_I along the length of the combustion zone.

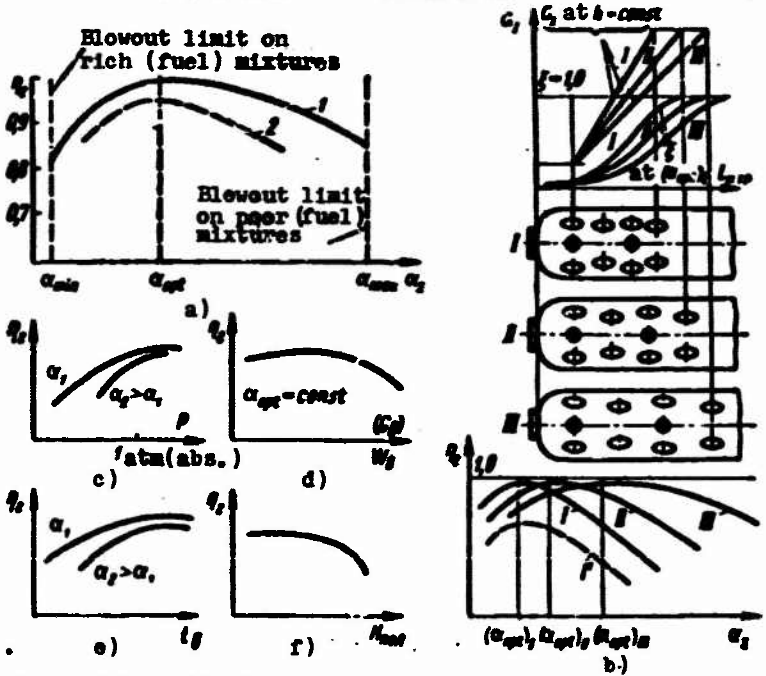


Fig. 80. Fuel combustion efficiency in a combusticn chamber.

Figure 80b shows how, for a given chamber (flame tube) with an invariable quantity of G_{I}, changing the length of the section of primary air supply, it is possible to change characteristic $\eta_{\text{г}} = f(\alpha_{\Sigma})$. If with an invariable $(\eta_{\text{г}})_{\max}$ air G_{I} is fed into a short section (version I), then the process of heat evolution proceeds more intensely, and the curve of the coefficient of heat evolution ξ rises sharply. If the air is fed into a stretched section (version II), then the ξ curve is flat, but in turn curve $\eta_{\text{г}} = f(\alpha_{\Sigma})$ is flatter, which permits on partial loads (departing from $\alpha_{\text{орт}}$) to have rather high values of $\eta_{\text{г}}$. From this example it is possible to make such a conclusion. For example, having version II, it is possible to operate well with $\alpha_{\Sigma} < (\alpha_{\text{opt}})_{\text{II}}$, if we change the air distribution, approximating version I, i.e., to effect an earlier supply of primary air. Otherwise, the value of $\eta_{\text{г}}$ will be low because the air is supplied too late, and on outlet from the chamber the flame will appear, diverging beyond the limit of the flame tube.

The absolute value of $\eta_{\text{г}}$ depends even more upon the character of the input of the lateral jets. Thus, for example, if in the chamber of version I the depth h of the supply of the jets to the flame tube is considerably decreased because of the decrease in the diameter of the openings with a corresponding increase in their number, then the curve $\eta_{\text{г}} = f(\alpha_{\Sigma})$ will go lower (Fig. 80b, dashed curve *I'*). It is natural that with an increase in h there is a certain value of $h_{\text{орт}}$ above which the effect will even be negative. Let us remember that an increase of h can also be attained by an increase in velocity w_2 in the clearance between the flame tube and the casing, however, in this case, pressure losses in the chamber increase.

The parameters of the working medium have a considerable effect on the combustion efficiency of the fuel, especially the temperature and velocity, and also the air pressure on intake to the chamber. With a decrease in air temperature $t_{\text{в}}$ the value of $\eta_{\text{г}}$ is also reduced (Fig. 80a, dashed curve 2), especially at a temperature of $t_{\text{в}} < (100-150)°\text{C}$. In this case, $\eta_{\text{г}}$ drops the more, the greater is α_{Σ}. Approximately of the same character, but in respect to the absolute value, the effect of air pressure $p_{\text{в}}$ is considerably less. Pressure begins to considerably affect combustion efficiency when $p_{\text{в}} < 1$ atm(abs.) as shown in Fig. 80c. With a decrease in $t_{\text{в}}$ and $p_{\text{в}}$ the blowout limits are narrowed. An increase in the velocity of air $w_{\text{в}}$ on intake, as a rule, reduces $\eta_{\text{г}}$ (Fig. 80d), since the time of the stay of the fuel in the combustion zone is reduced the aerodynamic conditions are changed, and flame stabilization is impaired. However with a considerable reduction of $w_{\text{в}}$ below calculated in the range far from optimum operating conditions there is also observed a certain reduction of $\eta_{\text{г}}$. Figure 80e and f shows the character of the dependence of $\eta_{\text{г}}$ on $t_{\text{Б}}$ and flight altitude $H_{\text{пол}}$ (the effect of $t_{\text{в}}$ and $p_{\text{в}}$). The latter dependence is substantial for aircraft GTD.

For the combustion chamber of any GTD, the characteristic of disruption of the operating process which determines the area where steady fuel combustion is possible is very important. This characteristic also basically depends on composition of the mixture α_{Σ}, velocity $w_{\text{в}}$, temperature $t_{\text{в}}$, and air pressure $p_{\text{в}}$ on intake to the chamber (Fig. 81a). The starting characteristic which determines the range of values of air velocities of air at which can be performed the reliable starting of the combustion chamber is also important. The starting range, naturally is narrower than the range of steady combustion (Fig. 81b).

Very important are the dependences of pressure losses

$$\frac{\Delta p^*}{p^*_{\text{к}}}\cdot 100=\frac{p^*_{\text{к}}-p^*_{\text{г}}}{p^*_{\text{к}}}\cdot 100\%$$

on the regimes and other factors in the combustion chamber. A basic effect on these losses proves to be velocity $w_{\text{в}}$, and also the value of α_{Σ} determining the thermal component of the pressure losses (Fig. 81c and d).

During tests of combustion chambers, it is important to know the field of temperatures of the gas on outlet (Fig. 81e). The temperature graph is usually constructed along a radius or diameter of the outlet section of the chamber $d_{\text{к}}$. From this field it is

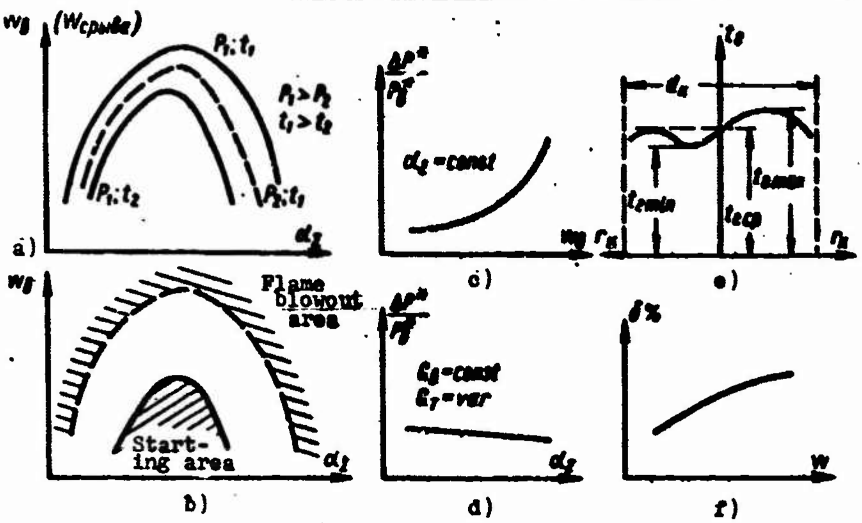


Fig. 81. Characteristics of combustion chambers.

possible to determine the degree of nonuniformity of the gas temperature

$$\delta = \frac{T^*_{max} - T^*_{min}}{(T^*_г)_{ср}} \cdot 100\%,$$

which, as a rule, changes considerably, for example, with a change in the air velocity $w_в$ (Fig. 81f).

An Example of the Determination of Basic Parameters and Dimensions of a Combustion Chamber.

To determine the basic dimensions of a combustion chamber of a locomotive gas turbine engine.

Basic data:

1) air consumption $G_{вΣ} = 22.6$ kg/s;

2) air pressure on intake to the combustion chamber $p_в$ = 6.0 kgf/cm^2;

3) air temperature on intake to the combustion chamber $t_в$ = 236°C;

4) gas temperature before the turbine $t_Г$ = 730°C;

5) fuel – naval mazut (low-sulfur) M12 corresponding to GOST 1626-44.

Temperature of preheating of fuel before the injector t_T = 120°C.

Elementary composition of fuel in respect to working mass:

$C^p = 84{,}7\%$, $H^p = 10{,}36\%$; $S^p = 0{,}97\%$; $O^p = (O^p + N^p) = 0{,}77\%$; $A^p = 0{,}20\%$; $W^p = 3{,}00\%$.

Calculation Scheme.

1. Let us select the type of combustion chamber. Taking into account the purpose of the GTD, for the sake of simplicity of experimental modification of the combustion chamber and its operation, we shall select a combustion chamber layout of the sectional type. Let us assign a value of the coefficient of fuel combustion efficiency $\eta_Г$ = 0.985.

For convenience in the arrangement of the GTD and in obtaining optimum dimensions (for the midsection) of the combustion chamber and the rest of the assemblies of the engine, we shall take the number of chambers n = 6. Then the air consumption through each chamber will be G_B = 3.93 kg/s.

2. We shall determine the lower heat of combustion of fuel Q_H^p and the stoichiometric quantity of air L_0:

$$Q_H^p = 81 \cdot C^p + 246 \cdot H^p - 26\,(O^p - S^p) - 6 \cdot W^p = 9400 \text{ kcal/kg};$$

$$L_0 = 0{,}1149 \cdot C^p + 0{,}3448 \cdot H^p + 0{,}0431 \cdot S^p - 0{,}0431 \cdot O^p = 13{,}3\,\frac{\text{kg of air}}{\text{kg of fuel}}.$$

3. We shall determine the quantity, composition, and the heat capacity of the products of combustion:

a) the quantity of "pure" products of combustion (where α = 1.0):

$$V_{RO_2} = 0{,}0186\,(C^p + 0{,}37 \cdot S^p) = 1{,}572\ \text{m}^3/\text{kg};$$
$$v_{RO_2} = 0{,}506\ \text{m}^3/\text{kg};$$
$$V_{N_2} = 0{,}615 \cdot L_0 + 0{,}008 \cdot N^p = 8{,}186\ \text{m}^3/\text{kg};$$
$$v_{N_2} = 0{,}790\ \text{m}^3/\text{kg};$$
$$V_{H_2O} = 0{,}0124\,(9 \cdot H^p + W^p) + (V_{H_2O})_{возд} = 1{,}272\ \text{m}^3/\text{kg};$$
$$v_{H_2O} = 1{,}246\ \text{m}^3/\text{kg};$$
$$(V_{H_2O})_{возд} = 0{,}001 \cdot L_0 \cdot d_0 = 0{,}001 \cdot 13{,}30 \cdot 6 = 0{,}0798\ \text{m}^3/\text{kg};$$

b) the weight composition of "pure" products of combustion

$$G_{ч.г} = \sum \frac{1}{v_i} V_i = 14{,}3\ \frac{\text{kg of gases}}{\text{kg of fuel}};$$

c) the average weight heat capacity of "pure" products of combustion

$$(c_p)_{ч.г} = \sum (c_p)_i r_i$$

where r_i — weight portion of each element. The average weight heat capacity of air is taken from tables.

4. We shall determine the value of overall excess air ratio from expression(57): α_Σ = 5.15.

5. We shall determine total fuel consumption

$$G_{т\Sigma} = \frac{3600 G_{в\Sigma}}{\alpha_\Sigma L_0} = 1238\ \text{kg/h}.$$

Fuel consumption in one combustion chamber $G_т = \frac{G_{т\Sigma}}{n} = 206{,}0$ kg/h.

6. We shall determine the basic dimensions of the combustion chamber. The area of cross section of the flame tube $F_{ж.тр} = \frac{G_1 v_в}{w_в}\ \text{м}^2$. From Table 11 we take the value $\alpha_г$ = 2.0; $w_г$ = 10 m/s and H = $12 \cdot 10^6$ kcal/($\text{m}^3 \cdot$h$\cdot$atm(abs.)). Then the consumption of primary air $G_1 = G_в \frac{\alpha_г}{\alpha_\Sigma} =$ = 1.56 kg/s. Specific volume of air (according to its parameters on

intake to the combustion chamber) $v_0 = \frac{RT_0}{p_0} = 0{,}25$ м³/кг. Consequently, $F_{ж.тр} = \frac{1{,}56 \cdot 0{,}25}{10} = 0{,}0391$ м² and the internal diameter of the flame tube $d_{ж.тр} = 0{,}224$ м.

In accordance with the accepted value of combustion intensity H, the complete length of the flame tube $l_{ж.тр} = 0{,}805$ м. In this case the overall length of the casing of a combustion chamber with branch connections will be approximately equal to 910-930 mm. The determined value $\left(\frac{l}{d}\right)_{ж.тр}$ should correspond to the values shown in Table 11. The value $\left(\frac{l}{d}\right)_{ж.тр} = 3{,}5$.

Given the thickness of the wall of the flame tube and velocity w_2, it is possible to determine the **diameter** of the casing of the combustion chamber.

CHAPTER XI

FUEL SUPPLY SYSTEM

§ 44. Liquid Fuel Supply Systems

The fuel supply system must ensure keeping of a certain quantity of fuel and supplying it with defined parameters into the combustion chamber in quantities which correspond to the operating condition of the engine. A very simple schematic diagram of a fuel supply system of an aircraft gas turbine engine system [GTD] (ГТД) is represented in Fig. 82. The fuel tank 1, shown schematically in the form of one volume, which ensures the necessary engine operating time for a calculated load, as a rule, is divided into several tanks located in various parts of the aircraft. By switching the valves, it is possible to take fuel boosted by pump 2 from any tank.

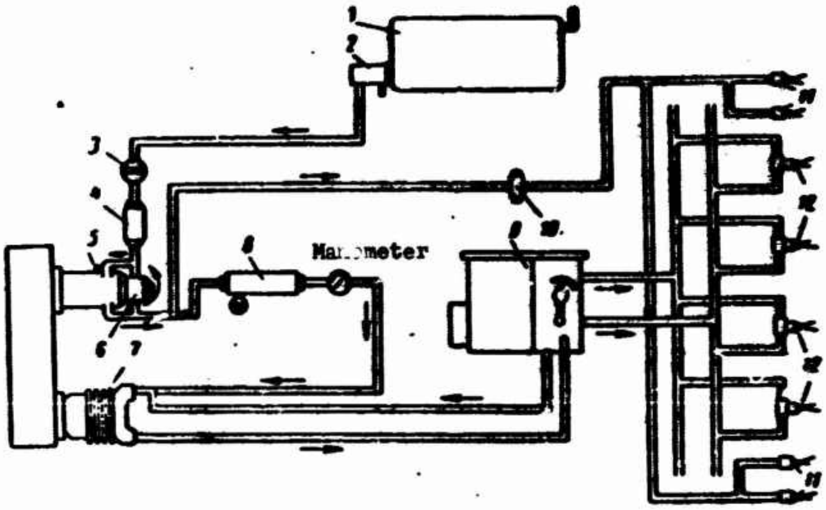


Fig. 82. Diagram of an engine fuel system.

On the intake to the pump there is usually installed a preliminary fuel filter. Further along on the pipe, the fuel by-passing valve 3 of the fire emergency cutoff goes through the first-stage fuel filter 4 and enters the low pressure pump 5. Depending on the resistance of subsequent parts of the system and of the quantity of fuel consumption, part of it returns through bypass valve 6 to suction pump 5. The remaining part during the starting of the engine with valve 10 open enters starting injectors 11, on the intake of which there usually are the secondary fuel filters. After starting the chambers, valve 10 is closed, and the fuel from pump 5 goes to the basic secondary fuel filter 8 and further on to the high pressure pump 7. From pump 7, passing through distributing valve 9, the fuel reaches the corresponding stages of the operating two-channel centrifugal injectors 12.

Depending on the engine load established by the pilot or by the system of automation, part of the fuel from valve 9 passes to pump 7. The high pressure pump [NVD] (НВД) is usually actuated from the shaft of the engine, and the low pressure pump [NND] (ННД) has a separate electrical drive from on-board batteries; after starting, the NND is actuated from an electric generator. As NVD usually gear or plunger pumps are used, and as NND – gear, rotational, rotary, and others. The filters used are mesh, slit, felt, mesh-paper, ribbed and so forth.

Stationary and transport GTD operating on heavy liquid fuel of the mazut type have in the fuel-supply system a number of auxiliary elements. Figure 83 shows one of the versions of the diagram of the fuel-supply system of a factory test stand for testing combustion chambers operating on mazut. On the stand provision is made for the simultaneous installation of two mazut combustion chambers and one auxiliary chamber which operates on kerosene. This combustion chamber can be used for preheating the air which goes into the mazut chambers. The fuel-supply system includes mazut tanks 1 and 2, and a tank 3 for solar oil and a tank 4 for kerosene. Fuel is supplied to these tanks from reservoir tanks by corresponding pumps 7-9. Kerosene is used in the preheating chamber 10 for the starting system and as

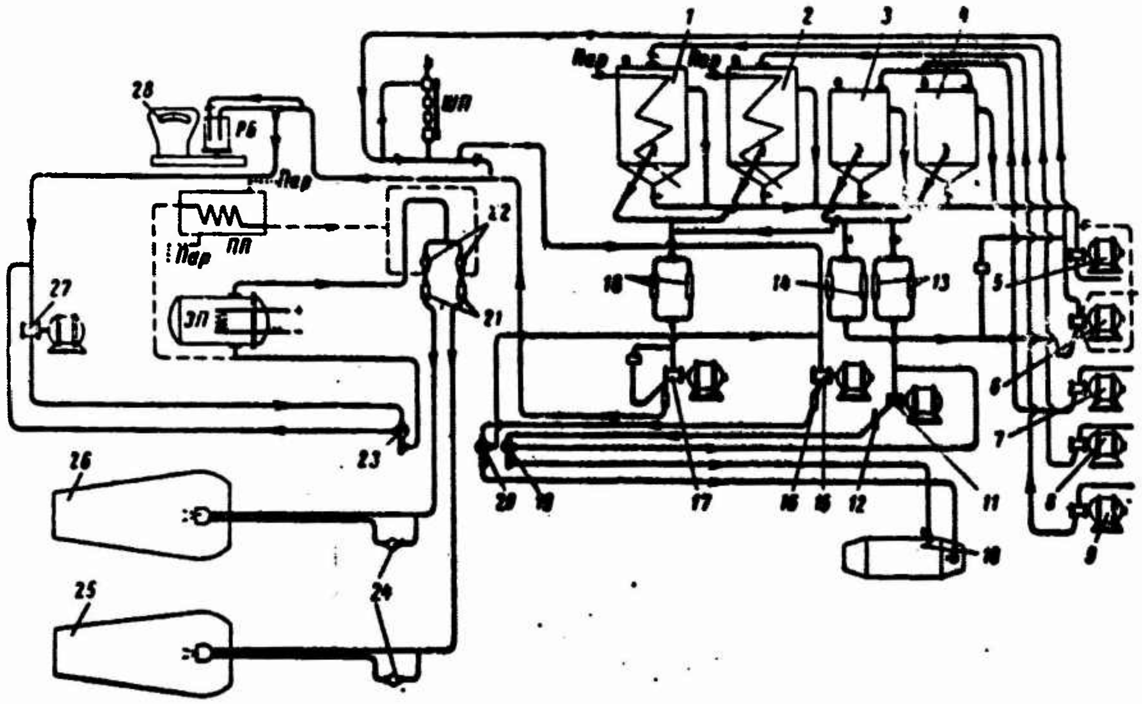


Fig. 83. Diagram of a fuel system of a test stand for combustion chambers. [Пар = Steam].

the basic fuel. The starting system of the preheating chamber has an igniter unit with an electric igniter and starting injector. Kerosene is supplied by pump 11 and on the way from the tank to the injector it passes through the primary filter 13 and the secondary filter 12.

Regulating valve 19, located before the injector, directs part of the fuel back for suction to the gear pump 11, ensuring a fuel pressure of 2-3 kgf/cm^2. Kerosene is supplied to the working injector, by-passing the primary filter [FGO] (ФГО) 14, and is supplied with the low pressure pump (NND) 6 and NVD 15, after which is located the secondary filter [FTO] (ФТО) 16. Part of the fuel for the NVD passes through valve 20 back to the suction pump.

For measurement of the consumption of kerosene, a flow-gauge [ShP] (ШП) is used. In basic chambers 25 and 26 the mazut is ignited which is first heated in tanks (1 or 2) to 30-60°C by means of a coil through which steam is passed. This ensures easy pumping through of the fuel by pump 17 through FGO 18 and the part of the manifolds up to NVD 27. Depending on the load, part of the fuel through valve 23 is directed back for suction by NVD 27. The remaining fuel before admission in the injector is additionally heated to 100-150°C or in an electric preheater [EP] (ЭП), or in a steam preheater [PP] (ПП).

Valves 22 shut off the supply of fuel in the case when only one chamber is operating. Further the fuel passes through FTO 21, after which it enters the first stage of the injector and with an increase in the load (and correspondingly in the fuel pressure) through valve 24 – the second stage of the duplex nozzle. For measuring the consumption of mazut service tank [RB] (РБ) is used, mounted on scales 28, from which during measurement the fuel is supplied to NVD 27.

Before cutting off the fuel supply and shutting down the combustion chambers, the system must be freed of mazut to avoid its congealing in the system. To do this, before a chamber is shut down it is switched to a supply of solar oil (tank 3). After a certain time, after exhausting the mazut remaining in the system, and having filled it with solar oil, the fuel supply can be cut off. Subsequent starting of the basic chambers is made on solar oil. After a successful start, the system is switched over to operate from one of the tanks with mazut. The overflow of fuel from tanks and the discharge of residue are performed, using pump 5.

The engine of the gas turbine locomotive produced by the Kolomna Diesel Locomotive Plant has the fuel-supply system, the schematic diagram of which is shown in Fig. 84. The capacity of the heavy fuel mazut tank is equal to 9000 kg; the tank for light fuel (solar oil) has a capacity of about 200 kg. Before the injectors, the mazut is heated to 120°C. Preheating is carried out by steam which enters from a special boiler, at first in the main tank up to 30-40°C, and then – in the primary preheater 5 up to 60-80°C, and finally, in the main preheater 9 – up to 120°C. The primary and secondary filters are duplicated in order to provide for their cleaning without stopping the GTD.

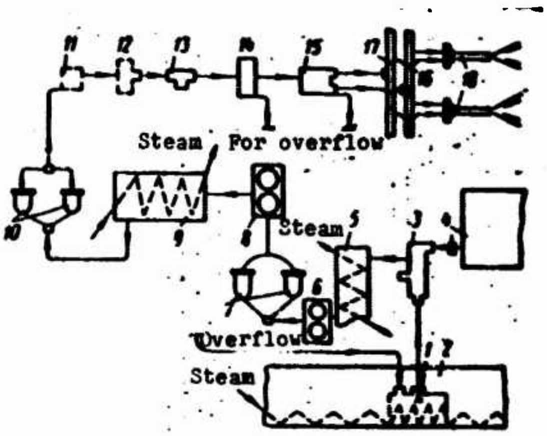


Fig. 84. Diagram of the fuel-supply system of a locomotive GTD: 1 – fuel-sampling device; 2 – tank for mazut; 3 – switch; 4 – tank of light fuel; 5 and 9 – preheaters; 6 – low pressure pump; 7 – primary filter; 8 – high pressure pump; 10 – secondary filter; 11 – temperature regulator; 12 – pressure regulator; 13 – control system valve; 14 – cutoff valve; 15 – regulating valve; 16 and 17 – collectors of the two injector stages; 18 – working nozzles.

The gear type pumps have bypass valves which do not permit an excessive increase in pressure in the system.

Elements of the Liquid Fuel-Supply Systems

The main fuel tanks of stationary gas turbine power plant [GTU] (ГТУ) are designed usually for 20-25 hours of operation. When using mazuts the tanks are fitted with heating devices with coils, which are heated either by steam or by gas or electric heaters. A diagram of a device for measuring the level (consumption) of viscous (heavy) liquid fuel is shown in Fig. 85. The fuel level in the tank is recorded by a column of water being displaced from tank 3 into a glass tube with a scale 2. The tank is filled with fuel through branch connection 1; the fuel is heated by a steam coil 4. With a change in the fuel level to a value of ΔH_T, the water level in the measuring tube will change by a value of ΔH_B, whereby $\Delta H_T \gamma_T = \Delta H_B \gamma_B$. Consequently, the quantity of fuel consumed

$$G_T = \Delta H_T \gamma_T F = \Delta H_B \gamma_B F,$$

where F – the fuel surface area in the tank; γ_T and γ_B – the specific weights of fuel and water respectively.

The primary filters frequently are reticular, have meshes of 0.1-0.5 mm. The secondary filters should hold particles larger than

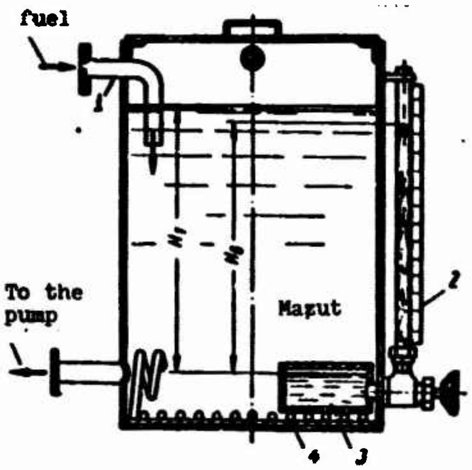


Fig. 85. A heavy liquid fuel service tank.

0.07 mm. These filters are also mesh, but more frequently are lamellar or with a filling of fibrous or porous materials. A design of a lamellar filter is schematically represented in Fig. 86a. By turning flywheel 1, the slots between plates 3, divided by cross pieces 4 are cleaned by cleaners 2. A design of a mesh filter is represented in Fig. 86b. Here in housing 6 with cover 5, a cylinder 7 with grid 8 is installed.

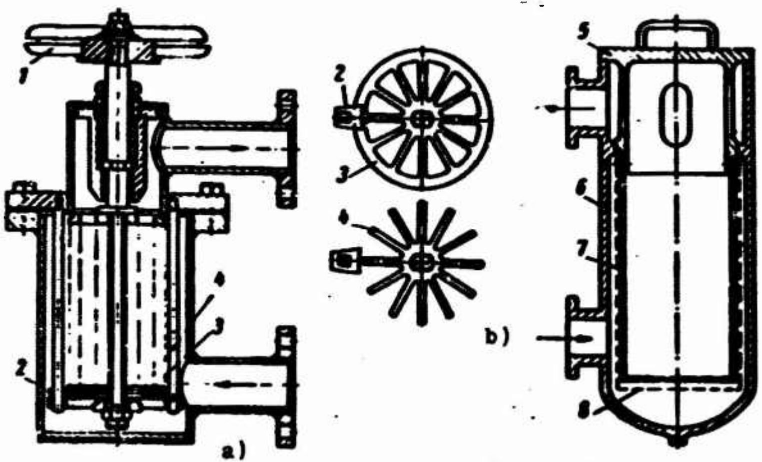


Fig. 86. Diagrams of fuel filters.

A diagram of a gear pump is given in Fig. 87. Driving gear 1 is identical with the driven one set in rotation either from the shaft of the GTD, or from an electric motor. Fuel moves from the suction cavity into the cavity located along the walls of the housing in the notches between the teeth of both gears. If the number of

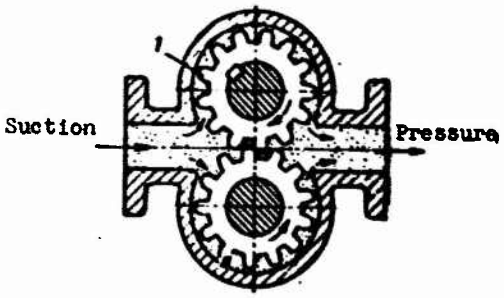


Fig. 87. Diagram of a gear pump.

teeth for every gear is z, and the volume being displaced by one gear tooth entering the notch of another is equal to V, then the theoretical pump capacity

$$Q_{т.н} = 2zVn \ м^3/\mathrm{min},$$

where n – the number of revolutions of the pump shaft. Volume V is equal to the product of the section of a tooth times its width. Because of the escape of fuel into the clearances between the teeth of the gears and between the teeth and the housing, its actual capacity $Q_д = Q_{т.н}\eta$. The value of volume efficiency η for big pumps attains a value of 0.8-0.9 and for low capacity pumps $\eta = 0{,}65 - 0{,}75$. The power consumed by the pump can be determined from the expression

$$N = 1{,}635 \frac{Q_д p}{\eta_н} \ \mathrm{kW}$$

where $Q_д$ – the actual volume feed of the pump in m^3/min; p – the pressure of the fuel after the pump in at; $\eta_н$ – the efficiency of the gear pump; usually $\eta_н = 0{,}55 - 0{,}72$.

A diagram of a very simple steam preheater of fuel is indicated in Fig. 88. For preheating the fuel it is also possible to use the heat of the gases coming out of the turbine.

Construction and Designing of Injectors

In GTD, most frequently centrifugal injectors are used. A very simple diagram of such a nozzle is represented in Fig. 89. From the

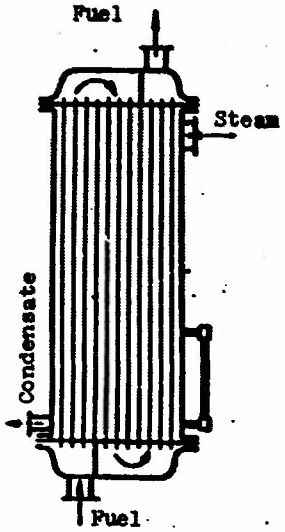


Fig. 88. Diagram of a steam fuel preheater.

annular duct the fuel via a series of tangential openings enters the swirl chamber 1 at velocity w_0. From this chamber the fuel through nozzle 2 with radius r_c flows out into the combustion chamber in the form of a conical film with an angle at the vertex β_T and it breaks up into minute drops. A scheme for designing a very simple single-channel injector for an ideal incompressible fluid was proposed by G. N. Abramovich.

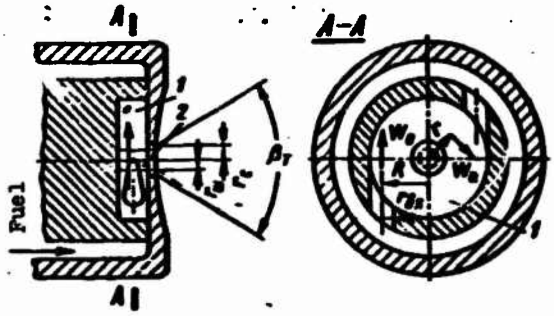


Fig. 89. Diagram of a single-stage centrifugal injector.

Let the value of R define the distance from the axis of the nozzle to the axis of the tangential ducts supplying fuel to the swirl chamber. The velocity of the fuel flowing through these ducts,

$$w_0 = \frac{G_T v_T}{n\pi r_{вх}^2}, \tag{67}$$

where G_T – the flow of fuel per second; v_T – the specific volume of fuel; n and $r_{\text{вх}}$ – number and radius of the tangential inlet channels.

This velocity w_0 is also the peripheral velocity $(w_u)_{0r}$ of the jet stream of fuel moving at a radius R from the axis in the swirl chamber. The moment of the quantity of motion of a particle of a fluid relative to the axis of the nozzle must remain unchanged and equal to initial value

$$w_0 R = w_u r, \tag{68}$$

where r – the distance from the axis of the nozzle to the particle which is moving inside the interior of the swirl chamber and the nozzle; w_u — the tangential component of velocity of this particle.

Let us note that the flow of the fluid will be vortex-free, since rotation of the particles around its axes is absent. From expression (68) it follows that with a decrease in the current radius r of value w_u should grow unlimitedly. However, in actuality the growth of w_u is limited. The law of conservation of energy in the form of a Bernoulli equation

$$p_T + \frac{1}{2g v_T}(w_u^2 + w_a^2) = p^*_{T0} \tag{69}$$

where p^*_{T0} — the total excess fuel pressure before the inlet to the injector over the pressure of the medium where it is injected; p_T — the local static pressure of the fuel in the nozzle; w_a — the axial component of velocity of the flow in the nozzle.

The effect of gravity can be disregarded since the fuel pressure is relatively great. From expressions (68) and (69) it follows that when $r \to 0$ the value of $w_u \to +\infty$, and $p_T \to -\infty$, which is physically impossibel, since tensile forces cannot appear in a liquid. In practice, in proportion to its approach to the axis of the nozzle and with a decrease in r, the value of w_u increases until the simultane simultaneously decreasing value of p_T becomes equal to the pressure of the medium where the fuel is being injected. Further, the value of p_T cannot be changed, since closer to the axis of the nozzle there is no fuel; here through the nozzle of the injector enters a gas (air) forming an axial air swirl with radius $r_{ш}$. In this way, at radius $r_{ш}$ the fuel pressure $p_T = p_{шT} = 0$. Consequently, in the nozzle the fuel moves along an annular section and

$$F = \pi (r_c^2 - r_{ш}^2) = \sigma \pi r_c^2,$$

where $\sigma = 1 - \left(\frac{r_{ш}}{r_c}\right)^2$ – the coefficient of filling the transverse section of the nozzle.

For the flow characteristic on the nozzle of the injector we find the distribution of a pressure over its section, using conditions which determine the motion of a swirling flow. We will examine an element of a ring of liquid with interior radius r and external $(r + dr)$. The length of the arc of the circular element $dl = r \cdot d\theta$, where θ – the central angle between two radii r. We take the width of a circular element as equal to one. The centrifugal force should be balanced by the difference of pressures on the internal and external surface of the ring, i.e.

$$1 \cdot dl \cdot dp_T = \frac{w_u^2}{r}\, dm, \qquad (70)$$

where the mass of the element $dm = \frac{1}{g v_T}\, dr \cdot dl \cdot 1$.

Since according to the law of conservation of moment, the quantity of motion, specifically,

$$w_u = \frac{(w_u)_{ш} r_{ш}}{r},$$

then equation (70) can be rewritten thus:

$$dp_T = \frac{1}{g v_T} (r_{ш})^2 (w_u)_{ш}^2 \cdot \frac{dr}{r^3}.$$

Integrating this expression, we obtain

$$p_T = -\frac{1}{2g v_T}(r_w)^2 (w_u)^2_w \frac{1}{r^2} + \text{const}.$$

We determine the constant of integration from the condition that on the boundary of the air cable ($r = r_w$) excess pressure $(p_T)_w = 0$, i.e.

$$\text{const} = \frac{1}{2g v_T}(w_u)^2_w.$$

Consequently, the general expression for distribution of pressure in the cross-section of the nozzle will be

$$p_T = \frac{1}{2g v_T}\left[(w_u)^2_w - w_u^2\right]. \qquad (71)$$

The substitution of expression (71) in equation (69) gives an expression for the axial component of flow velocity for the section of the nozzle:

$$w_a = \sqrt{2g v_T \dot{p}_{T_0} - (w_u)^2_w}. \qquad (72)$$

Hence it follows that $w_a = \text{const}$ over the section of the injector nozzle. Knowing the axial rate of flow, the per second volume flow rate of fuel through the injector can be calculated from the expression $Q_T = \pi r_c^2 \sigma w_a$. The value $(w_u)_w = \frac{w_0 R}{r_w}$; from the flow rate equation we determine

$$w_0 = \frac{Q_T}{\pi n r_{ax}^2}.$$

Consequently,

$$Q_T = \pi r_c^2 \sigma \sqrt{2g v_T \dot{p}_{T_0} - \frac{Q_T^2 R^2}{n^2 \pi^2 r_{ax}^4 r_w^2}};$$

hence

$$2g v_T \dot{p}_{T_0} = \frac{Q_T^2}{\pi^2 r_c^4}\left[\frac{1}{\sigma^2} + \frac{R^2 r_c^2}{n^2 r_{ax}^4}\left(\frac{r_c^2}{r_w^2}\right)\right].$$

From an expression for σ, we determine the ratio $\frac{r_c^2}{r_w^2} = \frac{1}{1-\sigma}$. Let us designate $\frac{R^2 r_c^2}{n^2 r_{ex}^4} = A^2$. The value of *A* is called the geometric characteristic of a centrifugal injector. After substitution we obtain

$$Q_T = \frac{\pi r_c^2}{\sqrt{\frac{A^2}{1-\sigma}+\frac{1}{\sigma^2}}}\sqrt{2g v_T \dot{p}_{T_0}} = \pi r_c^2 \mu \sqrt{2g v_T \dot{p}_{T_0}}. \qquad (73)$$

The weight flow

$$G_T = \pi r_c^2 \mu \sqrt{\frac{2g \dot{p}_{T_0}}{v_T}}. \qquad (74)$$

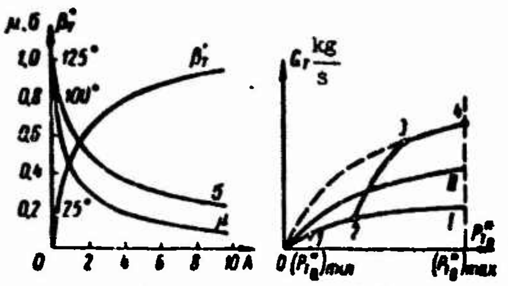
The value $\frac{1}{\sqrt{\frac{A^2}{1-\sigma}+\frac{1}{\sigma^2}}} = \mu$ is called the injector flow coefficient. With a change of σ the value of μ passes through a specific maximum value. With small σ, the value of μ is small, since the section of the nozzle is small. The value of μ is reduced with very large values of σ, since in this case r_w is small and a considerable part of the energy is spent on the achievement of high tangential velocities which lowers w_a. A steady flow process is obtained when μ_{max} (at maximum flow rate). The value of σ at which $\mu = \mu_{max}$ can be obtained by differentiating the expression for μ with respect to σ and assuming $\frac{d\mu}{d\sigma} = 0$. In so doing

$$A = \frac{\sqrt{2}(1-\sigma)}{\sigma \cdot \sqrt{\sigma}}.$$

After substitution of this value of *A* in the expression for μ, we obtain

$$\mu_{max} = \sqrt{\frac{\sigma^3}{2-\sigma}}.$$

With a change of *A* from zero to infinity, coefficient μ is changed from one to zero. Figure 90a shows a graph of dependence of μ and σ



a) b)

Fig. 90. Characteristics of a centrifugal injector.

on the geometric characteristic *A*. On the same graph there is indicated the dependence of the angle of expansion of the fuel cone β_T on the value of *A*.

On outlet from the injector nozzle, the pressure on the section of the stream (if we do not take into account the forces of surface tension) is unchanged and equal to the pressure in the medium where the fuel is injected.

The excess pressure in the injector nozzle is converted on outlet into impact pressure because of the fact that the axial component of the velocity of the stream is increased, moreover unevenly. At the boundary of the air swirl it will have least value, but on the circumference of the stream – it is greatest. However during the flow of an incompressible fluid the growth of axial velocity with unchanged consumption should be accompanied by a decrease in the area of cross-section of flow. In this case the flow coefficient is not changed. The external radius of the air swirl on outlet from the nozzle is greater than in the interior of the swirl chamber. The values of the axial components of velocity on outlet from the nozzle can be obtained from equation (69) under a condition of equality to zero of the static pressure of the fuel in the stream $(p_T = 0)$. Then

$$w_a^2 + w_u^2 = 2g v_T p_{T_0}^*. \tag{75}$$

Analogically to expression (68) $w_u = \frac{R w_0}{r}$, and since

$$w_a = \frac{Q_T}{n\pi r_{\text{вх}}^2} = \frac{\pi r_c^2 \mu \sqrt{2g v_T p_{T_0}^*}}{n\pi r_{\text{вх}}^2},$$

$$w_u = \frac{Rr_c}{nr_{вх}^2} \cdot \frac{r_c\mu}{r} \sqrt{2gv_T \dot{p}_{T_0}} = A\frac{r_c\mu}{r}\sqrt{2gv_T \dot{p}_{T_0}}.$$

Substituting w_u in equation (75), we obtain

$$w_a = \sqrt{1 - \frac{A^2\mu^2 r_c^2}{r^2}}\sqrt{2gv_T \dot{p}_{T_0}}. \qquad (76)$$

Consequently, with the increase in r, w_a increases.

The angle of expansion of the fuel cone emerging from the injector nozzle (β_T). is determined by the ratio $\frac{w_u}{w_a}$. This ratio changes radially, and at the boundary of the air swirl the angle $(\dot{\beta}_T)_m > (\dot{\beta}_T)_{пер}$ of the angle on the periphery of the stream. It is customary to determine the average angle $\dot{\beta}_T$ according to the values of velocities on the average radius of the useful cross-section of the stream:

$$r_{cp} = \frac{r_c + r_m}{2} = \frac{r_c}{2}(1+\varepsilon),$$

where

$$\varepsilon = \frac{r_m}{r_c}.$$

Then

$$\operatorname{tg}\left(\frac{\beta_T}{2}\right) = \left(\frac{w_u}{w_a}\right)_{cp}. \qquad (77)$$

Using the general expressions w_u and w_a, it is possible to obtain

$$(w_u)_{cp} = \frac{2\mu A}{1+\varepsilon}\sqrt{2gv_T \dot{p}_{T_0}}$$

and

$$(w_a)_{cp} = \sqrt{1 + \frac{4\mu^2 A^2}{(1+\varepsilon)^2}}\sqrt{2gv_T \dot{p}_{T_0}}$$

and consequently,

$$\mathrm{tg}\left(\frac{\beta_T}{2}\right)=\frac{2\mu A}{\sqrt{(1+s)^2-4\mu^2A^2}}. \qquad (78)$$

The given method of calculation, based on the analysis of a simplified model of an ideal liquid, can be used with a sufficient approximation for practical calculations. Computation of the friction of the liquid on the walls complicates the dependences.

A simple centrifugal single-stage single-nozzle injector has hydraulic characteristic of type I, shown in Fig. 90b, where $G_T \approx \sqrt{p^{*}_{T_0}}$. Since the value of $p^{*}_{T_0}$ is limited by the capabilities of a given pump and rarely exceeds 60-90 atm(abs.), then the range in change in the flow rate of fuel in this instance is small. It should be borne in mind that with low pressures (for kerosene below 3-5 atm(abs.) and solar oil 7-10 atm(abs.)) the quality of the fuel atomization becomes unsatisfactory. Consequently, the actual range of ratios of consumption

$$\frac{(G_T)_{\max}}{(G_T)_{\min}}=\sqrt{\frac{(p^{*}_{T_0})_{\max}}{(p^{*}_{T_0})_{\min}}}$$

is narrowed even more – to 3-5. In gas turbine engines, especially vehicular, as a rule, it is necessary to ensure considerably greater consumption ratios (about 10-15 and more). For this, systems either are required with a large number of, in proportion to the need, parallel connected simple sprayers, or more complex designs, for example duplex (one or two-nozzle) injectors (Fig. 91a, the characteristics of the action of stages I and II are given in Fig. 90b), injectors with supply control because of the partial removal (bypass) of fuel under incomplete loads (Fig. 91b), etc. In duplex injectors (Fig. 91a), the fuel under small loads is supplied only through the first stage (1), and under big loads both stages (1 and 2) operate. In so doing the total characteristic takes the form 1-2-3-4 (Fig. 90b). When using centrifugal injectors with controlled fuel removal (Fig. 91b), the range of change in the fuel consumption is also expanded because of the decrease in $(p^{*}_{T_0})_{\min}$, and furthermore, the

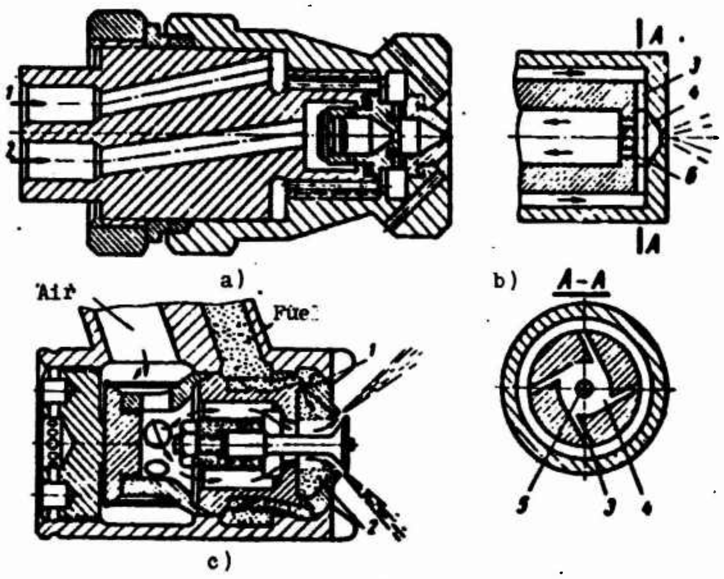


Fig. 91. Diagrams of injectors.

quality of fuel atomization under small loads is improved. In such injectors, through tangential ducts 3 to the swirl chamber 4 under all loads a quantity of fuel is supplied, corresponding to the consumption of fuel under full load. Depending on the engine mode, a greater or smaller part of the fuel, through channels 6 and a special valve is drawn off back to the suction pump.

Under all loads, even when a small quantity of fuel is supplied through nozzle 5, the quality of atomization remains good, since an identical quantity of fuel reaches the swirl chamber and the energy of rotation of flow is not changed. This permits reducing the value of $(G_T)_{\min}$.

The withdrawal of fuel can be carried out by different methods. There are designs of injectors in which it is possible to completely shut off the supply of fuel to the combustion chamber without stopping the pumping of fuel through the injector. This is necessary during operation on a heavy liquid fuel when after switching off the combustion chamber, the fuel residues can congeal in the manifold and hinder the subsequent starting of the engine.

The composite pneumocentrifugal injector should be noted, which is shown in Fig. 91c, the operating principle of which is clear from the figure. The use of compressed air available to the GTD, permits even with an insignificant expenditure of it, not exceeding fuel consumption in weight, to sharply improve the action of the injector.

An Example of the Determination of Basic Parameters and Dimensions of Centrifugal Injectors.

To calculate a single-stage centrifugal injector with a fuel cone angle $\beta_T = 60^\circ$.

Basic data

Fuel consumption $G_T = 40{,}5 \cdot 10^{-3}$ kg/s with a pressure drop $\Delta p_T = 36 \times 10^4$ kgf/m^2.

Fuel – kerosene; $\frac{1}{v_T} = 830$ kg/m^3.

Calculation scheme

1. Let us select the diagram of an injector shown in Fig. 89.

2. From the graphs (Fig. 90a) for angle $\beta_T = 60^\circ$ we find the value of the geometric characteristic $A = 0.90$ and the flow coefficient $\mu = 0.461$.

3. From the expression for weight flow of fuel (74) we determine the diameter of the nozzle:

$$d_c = \sqrt{\frac{4G_T}{\pi\mu\sqrt{\frac{2g\Delta p_T}{v_T}}}} = 1{,}21 \cdot 10^{-3} \text{ м}.$$

We take in round numbers $d_c = 1{,}2$ мм.

4. Let us select the elements of the injector in accordance with the value of A:

a) we take the number of intake tangential ducts $n = 3$ (usually $n = 2$-4);

b) the arm of twist (radius R, determining the position of the intake ducts) can be taken as $R = 2d_c$. Then $R = 2 \cdot 1.2 = 2.4$ мм.

5. We determine the diameter of the intake tangential ducts:

$$d_{вх} = 2\sqrt{\frac{Rr_c}{nA}} = 1.47 \text{ мм};$$

we take in rounded figures $d_{вх} = 1.5$ мм.

6. Let us select a number of supplementary dimensions:

a) the diameter of the swirl chamber $D_к = 2(R + r_{вх}) = 6.30$ мм;

b) the height of the cylindrical swirl chamber h is made somewhat greater than $d_{вх}$; we take $h = 1.8$ mm;

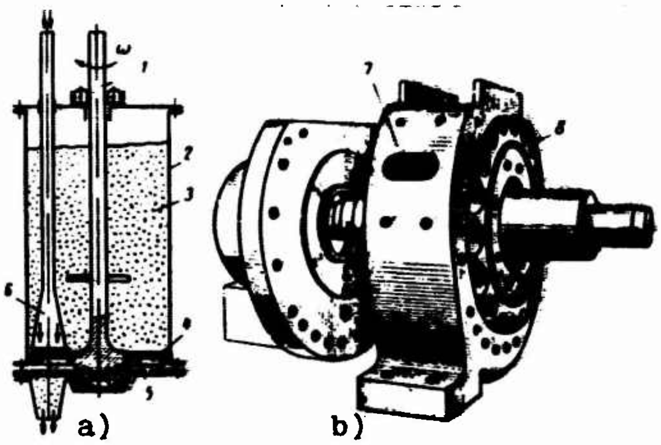
c) the length of the nozzle l_c (the thickness of the front wall of the nozzle) is usually taken as equal to approximately $0.50 d_c$; we take $l_c = 0.6$ мм;

d) the length of the intake tangential ducts $l_к$ is usually taken as equal to approximately $2d_{вх}$; we take $l_к = 3$ mm.

Hydraulic calculation must be determined more accurately, allowing for rounding off of a number of the calculated dimensions.

§ 45. Solid Fuel Supply Systems

As a result of experimental research on the combustion of solid fuel in GTU, were found more rational types of elements of fuel supply devices and of the fuel supply system as a whole were found. To supply fuel to the injector of the coal dust combustion chamber shown in Fig. 71 a mesh type dust feeder (Fig. 92a) was used. Its rotor 1 is driven by an electric motor and in the lower part it has a ring with spokes 4 to which a metal grid 5 is fastened. Coal dust 3 is placed over the grid in bin 2. Over the output opening a blowoff duct is installed with cone 6, via which condensed air is supplied for transportation of the coal. A layer of dust caught by the rotation grid is blown off by compressed air and the aeromixture enters the injector. The fuel supply is regulated by changing number of revolutions of the rotor.



a) b)

Fig. 92. Diagrams of dust feeders.

A diagram of a combustion chamber based on the design of the Baumann Moscow Higher Engineering School [MVTU] (МВТУ) for a gas turbine locomotive of $N = 6000$ h.p. in two sections working on Donets grade Г coal dust is shown in Fig. 93. The basic airflow II enters the chamber through an upper branch connection and is divided into two parts. One part goes upward to a bladed register 6, the other turns downward to the mixer. Along the length of the telescopic flame tube 1 there are openings to introduce primary air and circular slots for air which cools the shell. Ignition is carried out by igniter 2, which has an electric igniter and a liquid (starting) fuel injector.

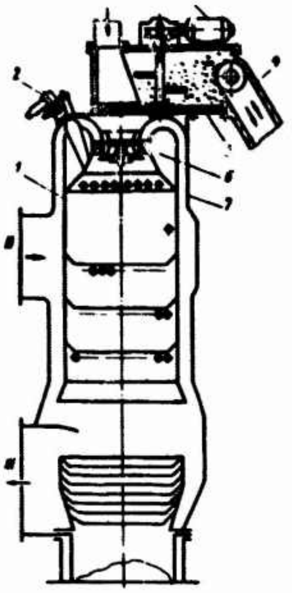
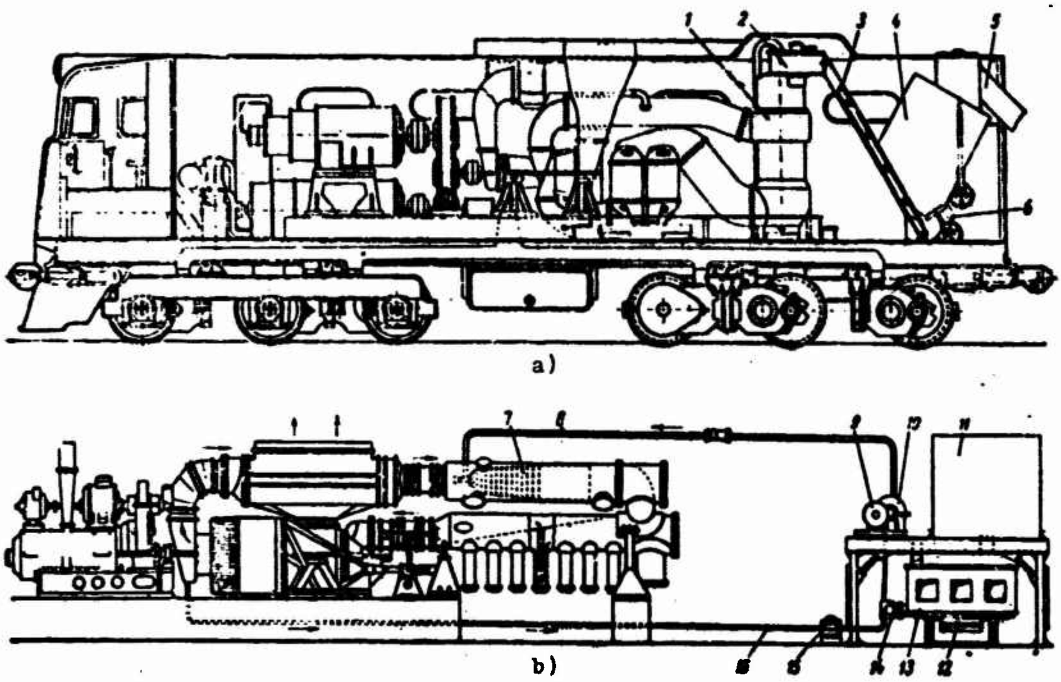


Fig. 93. Diagram of the combustion chamber of a gas turbine locomotive (MVTU design).

Coal dust from mesh dust feeder 5 by airflow I in a 1:1 proportion (by weight) enters the injector located in the center of the front device. The chamber operates with dry slag. Part of the heavy ash and slag sinks to the lower part of the chamber. The working gas with particles of ash on outlet form the chamber at first falls into a louvered type primary purifying stage and further on into the secondary purifying stage – a turbocyclone. Here, particles larger than 20 μm are retained and the purified gas enters the turbine. Dust feeder 5 is attached to the upper part of the chamber housing 7 and the pressure inside its bin is the same as that in the combustion chamber. The rotor of the dust feeder is set in motion by electric motor 3. The fuel is loaded into it by conveyer 4 from an intermediate bin which is also under pressure.

Figure 94a gives a longitudinal cross-section of the first section of the locomotive where the gas turbine plant is located. The combustion chamber 1 has dust feeder 2 which is loaded by conveyer 3 from intermediate bin 6 which connects through a valve device with dispensing bin 4. Into the dispensing bin with a capacity of 4 t, the fuel is fed by conveyer 5 through a second valve device from the main bin with a capacity of 15 t which is located on a trailer tender.



a)

b)

Fig. 94. Diagrams of the installation of solid fuel GTU on locomotives.

The fuel consumption during operation of the plant at rated load is equal to 1200 kg/h. When bin 6 is loaded, the upper valve of bin 4 is closed and the pressure in it is built up to the value of the air pressure in the chamber; in so doing, the lower valve is opened and the fuel is delivered to intermediate bin 6. When bin 4 is loaded the lower valve is closed and the upper one is opened, connecting bin 4 with the atmosphere.

Figure 94b shows a diagram of a GTD and fuel supply system of the gas turbine locomotive of the Alco Allis-Chalmers firm. Fuel consumed by this locomotive amounts to 1500 kg/h; the maximum size of coal dust particles is equal to approximately 150 μm. When idling the fuel consumption amounts to approximately 600 kg/h. Starting is carried out by means of an electric igniter and liquid fuel supplied by the starting injector, which during the transition to solid fuel is not shut off and operates as a pilot.

In the initial layout lump coal, which was supplied from the main bin by a stoker and by a worm conveyer, was directed towards the back section of the bin where it was ground up in a crusher to sizes of pieces less than 50 mm. Further on, the coal dropped onto two parallel belt conveyers, which shifted it through the drying section being warmed by escaping gases. After this, the coal is directed by worm conveyers to the intermediate bin from which a transverse worm conveyer loads it into a mill with a capacity of 2000 kg/h.

The mill was driven from an electric motor of about 50 h.p. From the mill the coal dust was supplied by worm conveyer to two pipes, through which it was transported by compressed air into the dispensing bin of the cellular (gear) dust feeder. Atmospheric pressure is maintained in the bins.

After tests, the fuel feed system on the locomotive was changed and made according to the diagram in Fig. 94b. Fuel is loaded into the main bin 11 with a capacity of 8 tons and crushed into pieces in size up to 5 mm. The bottom of the bin is warmed by gases coming out of the regenerator. From bin 11 the coal is shifted just as in the previous diagram into intermediate bin 13 with a capacity of 4.5 tons, which is mounted on a scale device 12, which is utilized for measurement of the fuel consumption.

The coal drops into mill 9 with the aid of feeder 14 in a stream of compressed air which enters from the main compressor on through manifold 16 and is additionally compressed in a special compressor 15. Further on, the coal dust under pressure drops into a special bin from which cellular dust feeder 10 directs the aeromixture via manifold 8 towards the two combustion chambers 7.

Cellular (drum) dust feeder (Fig. 92b) consists of a cogwheel 8, turned by an electric motor. With the rotation of the wheel, in the cells between the teeth and the housing the coal dust is transported from the bin to a slot 7, where dust is caught by the flow of compressed air. This simplifies the fuel supply system, but, as operational experience showed, does not guarantee prolonged and reliable operation of the device. After a certain operating time, the clearances between the rotor and stator are enlarged as a result of wear of the parts, and the losses disrupt the normal supply of fuel. Special sealing systems eliminate this defect, but severely complicate the design of the dust feeder.

§ 46. Gas Fuel Supply Systems

In gas turbine plants both artificial and natural gas fuel can be used. The artificial fuel is obtained in the gasification of liquid or solid fuel in gas generating plants or as the technological product of another production.

Figure 95 shows a diagram of a GTU operating with a solid fuel gas generator, giving a mixed gas. A steam-air mixture is supplied to blowing device 11; the coal enters the shaft through the loading device 1. Air for blowing is supplied from the main compressor of the GTU after additional 10-15% compression in blower 8. The quantity of this air comprises 5-10% of the total consumption of air.

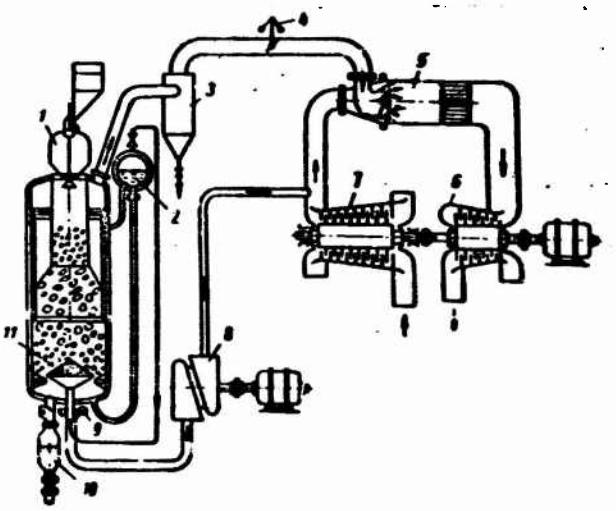


Fig. 95. Diagram of a GTU with generation of combustible gas under pressure: 1 — coal loading bin; 2 — steam collector; 3 — gas scrubbing; 4 — gas flow rate regulator; 5 — combustion chamber; 6 — turbine; 7 — compressor; 8 — blower; 9 — generator grid drive; 10 — slag removal bin.

Additional compression is necessary to overcome the resistance of the gas generator, the gas scrubbing system, and the pipes for ensuring mixing with air in the combustion chamber. The system for producing steam simultaneously serves for cooling the housing of the gas generator.

From the generator, the gas enters the gas scrubbing system 3 where the dust particles are removed, and then it is supplied to the combustion chamber. If the physical heat of the gas is not lost in the systems of gas scrubbing and pipes, the layout will be rather effective and the efficiency of the gasification process will attain 95-96%. It should be noted, however, that the power supply system in this case will be rather inertial and starting and stopping the GTD is complicated.

Ensuring normal operation of a gas generator under pressure presents certain difficulties, particularly, fuelling, slag elimination, and others. Furthermore, the gas contains a certain quantity of tar, which settles on the walls of the manifolds and the elements of the control system, which impairs their action.

The fuel supply system of a GTU is simplified considerably, if the system is fed by gas from a gas line system. But this, naturally, is possible only for stationary plants. Figure 96 shows a diagram of such a GTU fuel supply.

Gas from the system enters the gas compressor 4 through filter 1 through the gas-flow regulator 3. Automatic valve 2 shuts off the plant in the event of a dangerous drop in the gas pressure in the system. By compression in the compressor (two-stage with intermediate cooling) there is ensured an excess of gas pressure over the air pressure in the combustion chamber of approximately 0.2-0.5 atm(abs.) and more, depending on the system of carburetion.

During operation on partial loads, part of the gas will pass back through gate 5 to the intake of the compressor. Before entering the combustion chamber the combustible gas can be heated by spent gases in regenerator 6. The control system unit 8 determines the flow rate of the gas and provides for emergency shutdown of the GTD. During prolonged shutdowns the piping system is blown through valve 7.

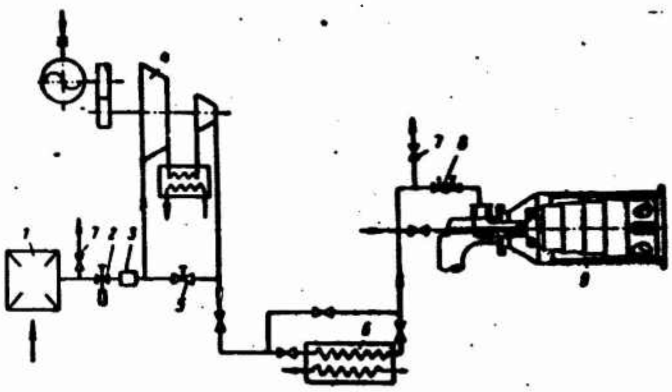


Fig. 96. Diagram of a GTU fuel system, operating on gas from a gas main.

§ 47. Materials of Combustion Chamber Parts

A number of parts and assemblies of the basic combustion chambers and of the combustion chambers for the intermediate preheating of gases operate under severe conditions. This applies particularly to the flame tubes, the elements of the front device, gas collectors, shields, the parts of the injector, and so forth. The gases washing them have a high temperature and contain, besides oxygen, some substances acting aggressively on the materials, such as steam, carbon dioxide, sulfur dioxide, and sometimes the particularly injurious compounds of vanadium, sodium, and others. The action of the gases causes gas corrosion and in certain cases destroys the parts of the flow-through part of the GTU in very short periods of time. To reduce corrosion and increase the service life of parts, apart from the special requirements imposed on fuels and the number of special measures for purifying it, it is necessary to also impose special requirements for the materials from which the parts themselves are manufactured. The materials from which the basic parts of the chamber are made, should be oxidation-resistant, i.e., resistant to corrosion in a gaseous environment at high temperatures. Heat resistance for these parts is frequently not so important. However, it is necessary, for example, that the metal of the flame tube be minimally sensitive to local stress concentrations and have a sufficiently high fatigue limit at high temperatures and be able to operate for a long time under conditions of high temperature. The individual parts of the injector, for example the nozzles, the elements of the swirl chamber, should resist abrasion and wear. Apart from this, it is very important that the materials of all parts be rather easily mechanically machined, stamped, etc. A rather important role is played by the cost of the material. In each specific case it is very important to select the metal which meets the requirements mentioned. Detailed data on the properties of materials are indicated in appropriate handbooks. Here we shall limit ourselves only to pointing out some of the grades of steel and approximate recommendations for their use.

The material of the flame tube must usually operate at wall temperatures of 600 to 900°C. In stationary and transport GTD, frequently the flame tubes are manufactured from chrome-nickel steel EYalT (1Kh18N9T), EI417, and low-alloy steel EI415.

For high-temperature aircraft GTD, steel of the type EI435, EI402, EI417 (Kh23N18), EI602, EI652, EI696, EI437, EI437B is used. Nickel alloys EI437, EI437B, and others contain up to 70-80% nickel; they are scarcer and their cost is great.

The shields of the combustion chambers, and also the reducers and gas collectors most often are manufactured from steel of the type EYalT (1Kh18N9T), EI402, and others. The casing of the combustion chambers of stationary GTD is often made from material of the type of regular carbon steel 10. For more overloaded engines and for the housing steel of type EYalT is used. The pairs of injectors (reducers, atomizers, nozzles, etc.) are manufactured from steel of types ShKh15, Kh12, 38KhA, and so forth. Fuel lines are often made of steel EYalT (1Kh18N9T).

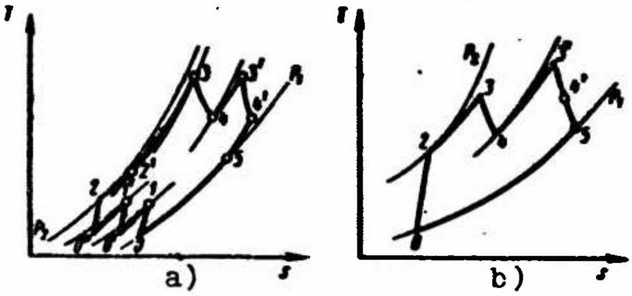
Individual elements of the casings (diffusers, reducers, and others) of the combustion chambers of aircraft GTD are manufactured from various aluminum alloys.

CHAPTER XII

GAS REHEAT COMBUSTION CHAMBERS

One of the most important means for significantly raising the effectiveness of a gas turbine engine cycle is the reheating of the gas in the process of its expansion. In a stationary gas turbine power plant [GTU] (ГТУ) it is more expedient to combine reheating with the intercooling of air in the process of compression, and sometimes by heat recovery of exhaust gases.

Such a working cycle in a *Ts*-diagram is shown in Fig. 97a, and a layout of the plant is represented in Fig. 98. Line 0-1, 0'-1', and 0"-2 shows the air compression in the low, medium and high pressure compressors [KND, KSD, KVD] (НHД, НСД, НВД). Sections 1-0' and 1'-0" represent the processes of cooling of air in intermediate coolers [PKh] (ПХ) between the compressors.



a) b)

Fig. 97. Cycles of GTD with gas reheating.

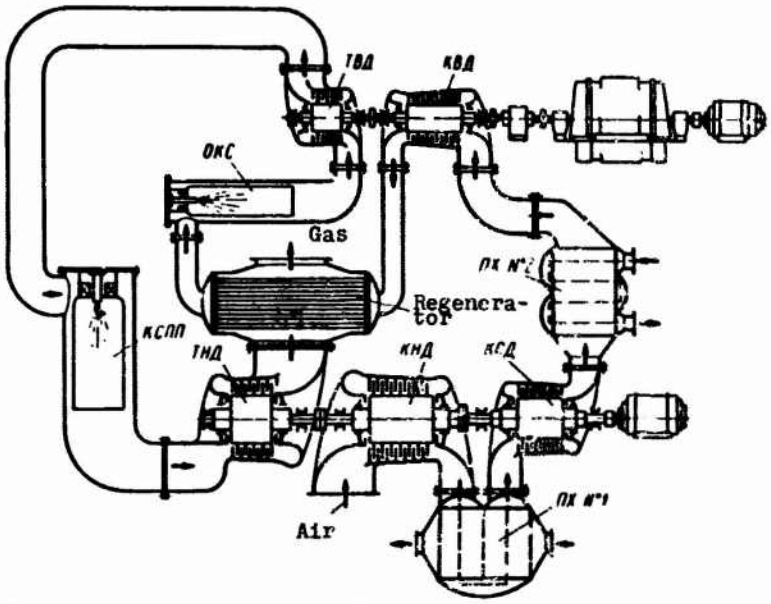


Fig. 98. Diagram of a compound gas turbine plant.
DESIGNATIONS:
ОКС = OKS = main combustion chamber;
ТВД = TVD = high pressure turbine;
КСПП = KSPP = reheat combustion chamber;
ТНД = TND = low pressure turbine.

Section 2-2' represents the reheating of air in the regenerator, and 2'-3 – the supply of heat to the air in the main combustion chamber. Line 3-4 corresponds to the expansion of gases in the high pressure turbine [TVD]; on a section 4-3' heat is additionally supplied to gases in the reheat chamber; 3'-4' represents expansion in the low pressure turbine [TND], and line 4'-5 – the output of heat by the gas in the regenerator, and 5-0 – the arbitrary line of heat elimination corresponding to the elimination of the working medium in an actual cycle.

In some aircraft gas turbine engines [GTD] (ГТД) gas reheat is also used, whereby either the auxiliary combustion chamber is installed between turbine stages (Fig. 99), or the fuel burns additionally after the turbine in the afterburner located in front of the jet nozzle (Fig. 100). A working cycle of such GTD is represented in the Ts-diagram in Fig. 97b. Here after compression (line 0-2) and preheating of the air in the main combustion chamber

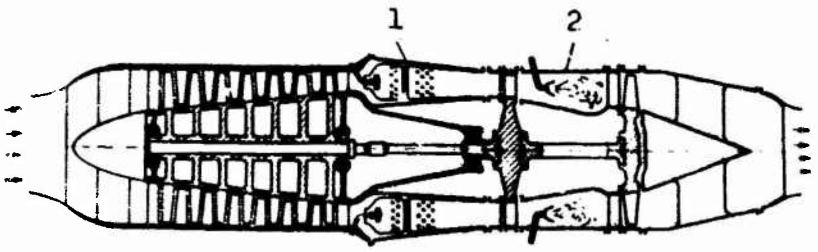


Fig. 99. Diagram of an aircraft GTD with a two-stage combustion: 1 – main chamber; 2 – reheat chamber.

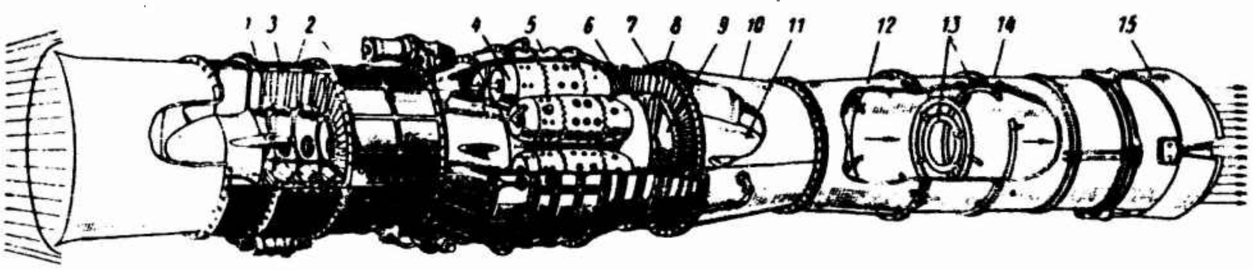


Fig. 100. Diagram of an aircraft gas turbine engine with afterburner: 1 – compressor rotor blade; 2 – compressor housing; 3 – stator blade; 4 – flame tube of the combustion chamber; 5 – combustion chamber housing; 6 – turbine nozzle box blade; 7 – turbine rotor blade; 8 – turbine housing; 9 – turbine wheel; 10 – external cone; 11 – internal cone; 12 – afterburner fuel injection nozzle; 13 – flame holder (exploded view); 14 – afterburner housing; 15 – adjustable nozzle.

(line 2-3) expansion follows in the high pressure turbine (line 3-4). Further, the gas is heated in the auxiliary combustion chamber (reheating) (line 4-3'). The gas can expand partially in the low pressure turbine (line 3'-4'), and partially in the jet nozzle (line 4'-5) or entirely in the jet nozzle.

In aviation, the most widely used is a version, with an afterburner. In stationary GTD the reheat chambers operate continuously as a result of which the effectiveness of cycle is increased. In aircraft engines, the afterburner operates for a short time, usually not more than 2-3 min and only under specified conditions where increased thrust is required: takeoff to reduce the length of the takeoff run of the aircraft or during climb, when it is necessary to gain altitude rapidly. In this way it is possible to increase the thrust of the engine at high flight speeds approximately twice and on takeoff – by 30-50%. The use of an afterburner increases the specific fuel consumption, especially during operation of the engine during the start.

§ 48. Afterburners

A schematic diagram of an afterburner of an aircraft GTD is provided in Fig. 101. On the intake of the chamber a diffuser is installed, in which the velocity of the gases emerging from the turbine is reduced to 100-160 m/s, and it is possible to carry out further combustion of the fuel. These velocities considerably exceed the velocities on intake to the main chamber.

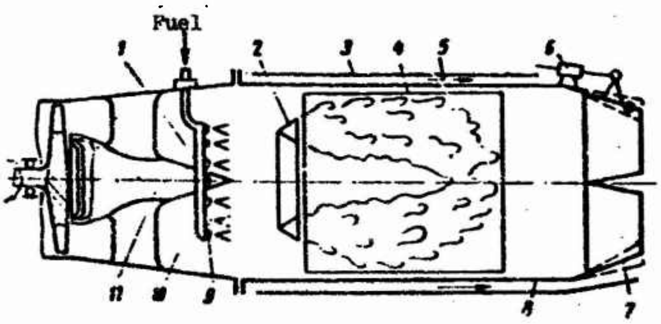


Fig. 101. Diagram of an afterburner: 1 – annular fuel manifold; 2 – flame holder; 3 – exterior shell; 4 – antivibration shield; 5 – cooling air; 6 – nozzle control mechanism; 7 – adjustable nozzle; 8 – afterburner housing; 9 – injector; 10 – diffuser; 11 – cone.

Despite a certain decrease in oxygen content in the gases as a result of the combustion of the fuel in the main chamber, the process of additional combustion proceeds here under rather favorable conditions since the temperature of the gas flow is higher. Furthermore, a certain number of active products of combustion in the mixture contributes to the acceleration of the chemical reactions of combustion.

It should be noted that in afterburners there are no rigid requirements for the form of the temperature field on outlet, since the gases from the chamber are not directed into the flow-through part of the turbine. This makes it possible at higher average temperatures of gas emerging from the chamber to limit the position of the combustion zone less rigidly.

In the afterburner there are installed flame holders, most frequently, annular ones, consisting of poorly streamlined bodies: a cone (an angular profile), a plate, and so forth. Fuel is supplied to the gas flow up to the flame holders by several centrifugal or jet injectors. In this way, the fuel evaporates sooner and the vapors are intermixed with the gas which ensures effective combustion in the combustion zone after the flame holder. To improve the mixing, the fuel is frequently supplied counter to the gas flow.

Inside the housing after the flame holders, as a rule, an antivibration shield is installed. Usually this is a corrugated duct (longitudinal corrugations) with a large number of fine openings. The function of this shield – to exclude vibrating combustion of fuel, which is accompanied by an unpleasant noise and by vibration which destroys the structure. The shell of the chamber is shielded from the flame cone and is coated both on the outside and on the inside with insulating materials. Between the housing and the shell usually flows the cooling air flow from the compressor of the engine, but more frequently appearing because of the impact pressure or the ejection effect of the gas jet. The length of the afterburner is equal to 1.5-2 m and more, and depends on the velocity and temperature of the gas on intake, the diagram of the working process, the conditions of arrangement of the engine installation and so forth. The gases, the temperature of which can attain 1800°C and more, emerge through the adjustable nozzle. If on switching on the afterburner, the cross-section of the nozzle is not increased, then pressure after the turbine will be built up and the difference in pressures before and after the turbine is decreased. To keep the number of revolutions of the turbine constant, it is necessary to increase the temperature of the gases before it, but this is undesirable and sometimes impossible.

The shape of the flowthrough part of the afterburner is of great significance. In order to decrease pressure losses, it is expedient to make it conical, expanding the chamber. However this increases the overall dimensions of the chamber, which is undesirable, since the diameter of the chamber becomes greater than the diameter of the turbine and, consequently, it will increase the cross-section of the engine.

The shapes of the flame holders of afterburners are very diverse. To increase reliability of burning, a flame holder is used (Fig. 102a) engaging the so-called precombustion chamber (or forechamber) with the flame tube, inside which a pilot flame burns. Often along the length of chamber there are spaced several annular flame holders (the exploded view flame holder) (Fig. 102b), which reduces the blockage of the cross section and reduces the pressure losses and simultaneously increases the reliability of flame stabilization. Flame holder systems of the type shown in Fig. 102c are also used.

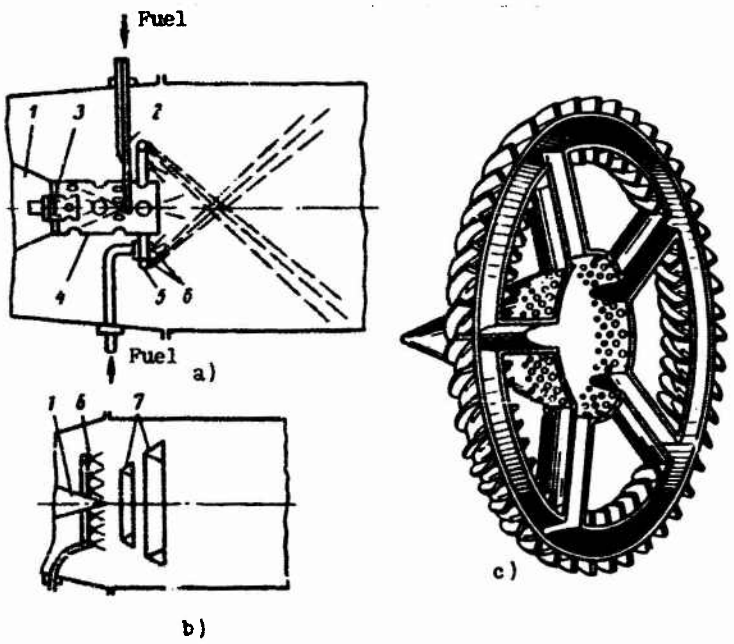


Fig. 102. Diagrams of flame holders of afterburners: 1 – bullet; 2 – booster nozzle; 3 – electrical igniter; 4 – precombustion chamber; 5 – fuel manifold; 6 – main injectors; 7 – flame holder.

The total resistance of all elements of the afterburner and, particularly the flame holders, should be minimum since it remains even when the supply of fuel to the chamber is cut off. Usually, the coefficient of hydraulic loss of afterburners

$$\xi_{\text{ф.к}} = \frac{\Delta p^*}{\frac{w^2}{2v}}$$

does not exceed 3-5, while in the main chambers this value reaches 8-12. Nevertheless a switched off afterburner reduces the thrust of an engine on takeoff by approximately 1.5-2.5%.

Along with certain favorable conditions of fuel combustion in the afterburner (particularly, the increase in gas temperature) the realization of such fuel combustion encounters specific difficulties.

With a high temperature of gases after the main chamber, the process of combustion in the afterburner can be impaired because of the small quantity of oxygen in the gases entering it. Deterioration of the combustion process occurs because of the lowering of pressure in the chamber, which drops below atmospheric as the altitude of flight increases.

With a decrease in the fuel pressure on intake to the injector it is difficult to obtain sufficiently good atomization of the small quantities of fuel which the individual nozzles supply. In afterburners the coefficients of combustion efficiency of the fuel, as a rule, are lower than in the main chambers and range from 0.85 to 0.97.

Starting of the afterburner under any conditions should be reliable. Therefore, despite the fact that the temperature of the gas on intake is frequently sufficient for self-ignition of the fuel, there is also used a system of an electrical firing. Self-ignition can be also used in principle, but only in the case when the time of stay of the combustible mixture in the flow before the flame holders is sufficient and exceeds the time of the induction period. In connection with this, fuel injection before the turbine is used. However, in case of accidental burning of fuel in the flowthrough part, it may put the turbine out of commission.

§ 49. Gas Reheat Combustion Chambers

Intermediate reheating of gases began to be used in GTU comparatively recently. In this case, the design of the gas reheat chambers usually repeated the designs of main chambers. Figure 103 shows a diagram of a reheat combustion chamber of a compound GTD with a power of 10,000 kW, made by the firm of Brown-Bovery for an electric power station operating on gas fuel. Experimental operation in the period from 1946 to 1951 showed sufficiently reliable operation of the plant. Gas fuel for firing and simultaneously for the pilot burner of the chamber is supplied via a central tube, inside which is located an electric igniter. Over the circular cavity surrounding this tube the basic flow of gas fuel is supplied. This gas is mixed with the primary gas which goes over a circular duct to a blade swirler emerging from openings in the walls of hollow blades (section *A-A*). The quantity of primary gas entering from the basic gas flow to the swirler is regulated by dampers which are located on ports of the external duct of the burner device. The telescopic flame tube consists of a number of finned shells and the mixer has a row of openings.

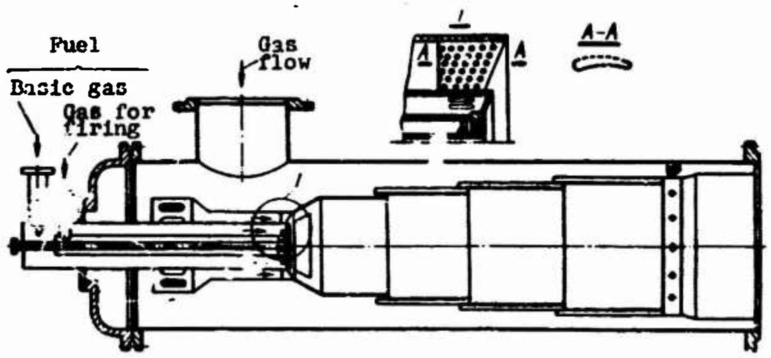


Fig. 103. Reheat combustion chamber for the natural gas GTD made by the firm of Brown-Bovery.

The temperature of gases before the low and high pressure turbines is identical and equals 600°C, while the pressure is equal approximately to 11 atm(abs.) and 4.6 atm(abs.) respectively. The temperature of the gases after the high pressure turbine on intake to the reheat chamber equals 400-440°C.

The combustion chambers, basic and intermediate preheating, of the marine gas turbine plant of the firm of Elliott, created in the period 1944-1951 were made according to the same diagram shown in Fig. 104. This original elbow chamber is used by the firm for a number of GTD designs. Air enters the chamber per arrow *A* tangentially, which provides rotary motion within the combustion zone. Primary gas 1 enters through slots 2 and ports inside the conical front device 3. The chamber operates on diesel fuel which is supplied to injector 4, and is ignited in starting by a retractable electric igniter. The section of the manifold between the chamber and turbine serves as a mixer. The wall of the flame tube consists of a flame sheet 5. The external housing of chamber 6 is solid and thick, is protected by a layer of refractory brick backing and special fire-resistant cotton.

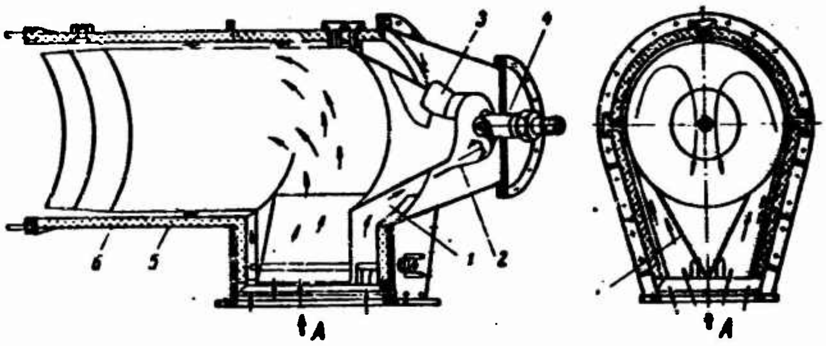


Fig. 104. Elbow combustion chamber.

In the gas turbine plant GT-12-3 of the Leningrad Metal Works with a complex cycle intended for operation on gas from the underground gasification of coal and on solar oil, both the basic chamber and the reheat combustion chamber are made according to the same plan. The reheat chamber has only slightly larger dimensions (Fig. 105).

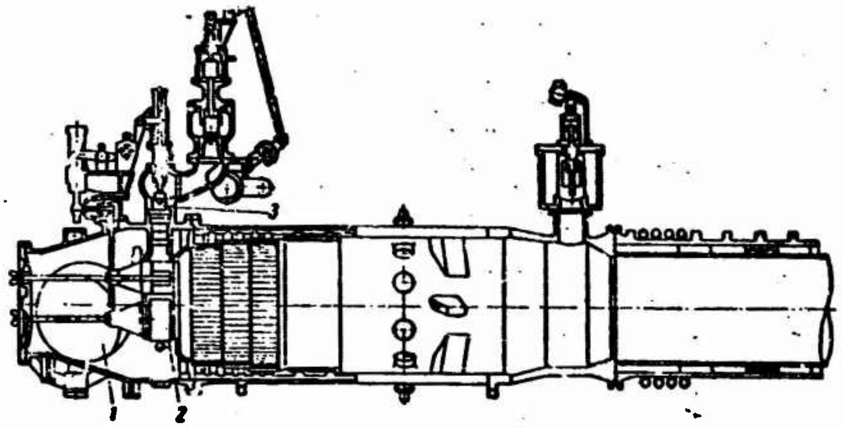


Fig. 105. Reheat combustion chamber of the GT-12-3 plant.

The flow of gas through the branch connection 1 i ed into the head part of the chamber. Primary gas enters through three gas burners 2, where its quantity is regulated by dampers. Fuel gas from special compressors under a pressure exceeding the pressure in the chamber by 0.4-0.8 atm(abs.), is supplied to burners 2 through upper valve 3. The temperature of the gas after the main chamber and the reheat chamber (the low pressure combustion chamber) is identical and equals 650°C. The temperature of the gases before the reheat chamber is 460°C. Secondary gas is fed into the nozzles of the mixer between the finned shells of a telescoping flame tube and the housing. The fuel is ignited by an electric spark. In the center of each gas burner there are injectors for supplying liquid fuel which enables the GTD to operate on solar oil.

The examined GTD made by domestic and foreign plants show that with a moderate temperature of gases before the turbines (up to 650-750°C), when on intake to the reheat chamber, the gases have a temperature up to 500°C, the chambers can be made according to the usual plans. In so doing, because of the more severe conditions for cooling the individual elements, it is necessary to make the shells of the flame tube finned and to increase the velocities of the cooling gas. However, elevation of the temperature of the working gas in prospective designs of gas turbines and the necessity of reducing pressure losses on the gas duct require the development of special designs of the gas reheat combustion chambers.

Today, the processes of compression and expansion tend to approach isothermal, and there is also a tendency to increase the temperature of the working body on intake to the gas turbine. This, in turn, increases the temperature of the gases on intake to the reheat chamber and the number of steps in preheating, i.e., the number of reheat chambers. Therefore domestic plants are installing 2-3 reheat chambers on the new gas turbine plants; in the future, layouts are possible with a large number of reheating steps. In this instance, the heating of the working body in each reheat chamber will be comparatively slight (to 100-200°C).

When developing special plans for the reheat chambers it is necessary to take into account the following characteristics of their operation:

1. Sufficiently high temperatures of gas on intake to the reheat chamber, which facilitates the process of fuel combustion and intensifies it.

2. Higher temperatures of the elements of the reheat chambers, which can reduce the service life of the construction.

3. Greater relative effect of the pressure losses in the gas reheat chambers on the parameters of the working body.

4. Considerably smaller quantities of additionally burned fuel than in main chambers, which complicates the designing of the injectors and the burners.

5. Smaller oxygen content in comparison with iar in the gases which enter the reheat chamber. This is especially important to take into account with high temperatures of the working body and a large number of reheat chambers.

These features determine the necessity for development of special plans for gas reheat chambers. In this case, in order for a process of expansion to approach isothermal, the problem may be posed of sequential combustion of fuel between the individual steps of the turbine, directly in the flowthrough part. In this direction appropriate developments are being conducted; however, the general solution to this problem has not yet been found, since it has not been possible to ensure the necessary service life and reliability of operation of the gas turbine blades.

Combustion of the fuel in the flowthrough part of the turbine must be carried out so that local combustion centers (flames) will not be formed with the temperatures exceeding the average permissible from the point of view of proper working order and durability of the material of the blades. Such microflame combustion should go on in the whole volume of interblade space of a given stage and simultaneously and in sequence on the stages of the flowthrough part of the turbine.

Such schemes have not yet been achieved in practice; however, a number of intermediate solutions are already known, particularly, the schemes of so-called built-in direct-flow annular reheat chambers. A diagram of such a chamber is given in Fig. 106a. The flowthrough part of the turbine in turn is divided by the reheat chamber into a high pressure part and a low (medium) pressure part. The reheat chamber includes elements of the afterburners, but has insignificant length as a result of the low gas velocities (in comparison with afterburners) in the combustion zone, although greater than in the main chambers (w_c = 70-120 m/s and more).

Just as in the afterburners, the housing is protected from emission by a shield. In the clearance between the housing and the shield the same gas passes through, which enters the reheat chamber if its temperature is 200-300° lower than the temperature permissible for the material of the wall of the shield. In this instance, the housing is coated with thermal insulation.

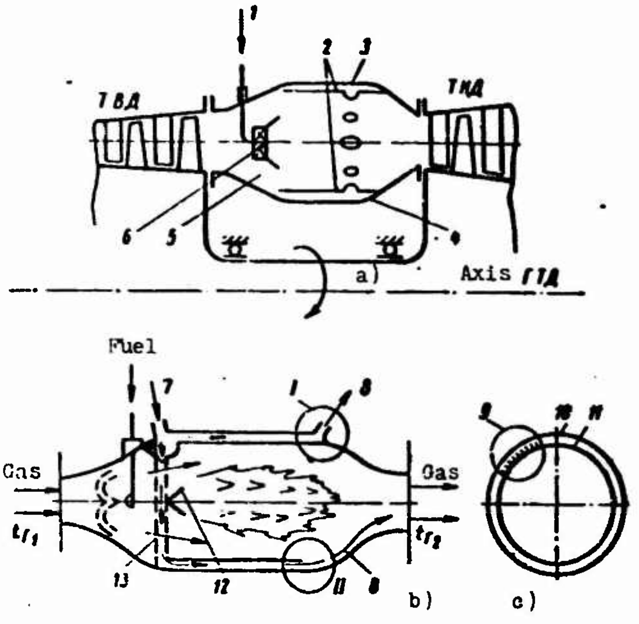


Fig. 106. Diagram of gas reheat combustion chambers: 1 – fuel; 2 – exterior interior shields; 3 – exterior housing; 4 – interior housing; 5 – gas reheat combustion chamber; 6 – injector; 7 – cooling air feed; 8 – cooling air tap; 9 – cooling fins; 10 – housing; 11 – shield; 12 – flame holder; 13 – branch connection for supplying cooling air to the interior shield.

For more reliable cooling or in case of high gas temperature on intake, the reheat chamber is cooled with air bled from the GTD compressor. The cooling scheme can be closed (the upper half of the diagram in Fig. 106b) or open with the emergence of the cooling air into the flow of gases (the lower half of the diagram in Fig. 106b).

The reheat combustion chamber can be not only circular, but also, for example, sectional. In this instance, the layout of the cooling air channel (Fig. 106c) is simplified, since the necessity of the feeding air to the interior screen (to the housing) is eliminated. For protection of the housing and more intense cooling of the shield, its external surface finned (Fig. 106c). The consumption of air of such cooling schemes is insignificant (1.0-2.0% of the gas consumption) and the pressure losses are moderate.

The special cooling system of the reheat combustion chamber can be used in a complex cycle GTU for the regenerative preheating of air (Fig. 107). Air from the low pressure compressor (KND) through an intermediate cooler (PKh) enters the high pressure compressor (KVD) and then goes into the main combustion chamber (OKS). From this chamber, the gases first go into the high pressure turbine (TVD) and after that into the gas reheat combustion chamber (KSPP No. 1) of

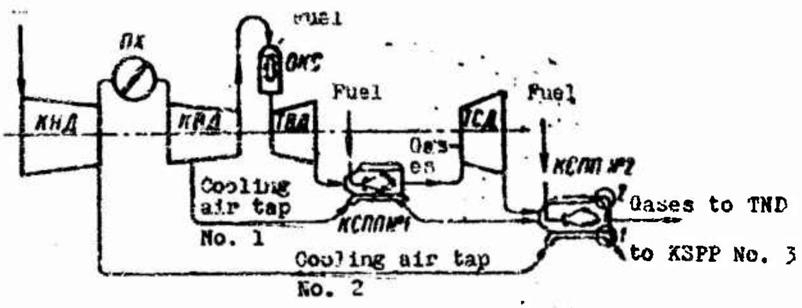


Fig. 107. Diagram of a GTD with the regenerative reheat combustion chamber. DESIGNATION: ТСД = TSD = medium pressure turbine.

regenerative type. This chamber is cooled by air specially bled from the corresponding KVD stage, which is further led into reheat chamber KSPP No. 2. Heated air is used as primary air and partially makes up for the lack of oxygen in the gas on intake to the reheat chamber KSPP No. 2. Such a layout has even greater significance for reheat chamber KSPP No. 3 where the oxygen in the gases is still less. If there is no chamber KSPP No. 3, then the cooling air of chamber KSPP No. 2 is mixed with the gases on outlet from it as indicated for KSPP No. 2 (Fig. 107).

The supply of smal quantities of fuel with insignificant preheating of gases in the reheat combustion chambers is carried out by the corresponding injection devices. One of the diagrams of the supply of liquid fuel is shown in Fig. 108. This device, which is an air injector combined with a flame holder, is located on the intake to the reheat combustion chamber.

The basic part of the gases flows around this device and a certain part of primary gas 1, swirled by vanes 3, as indicated by arrow A, catches the stream of fuel which issues from a central tube under excess pressure Δp = 0.2-0.4 atm(abs.) and is intermixed with it. Further, the intermixture of gas and fuel flowing around cone 2 and passing through bladed register 1 flows out in swirling flow II into the combustion zone of the chamber.

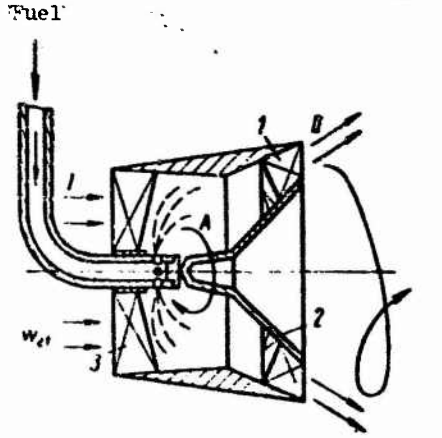


Fig. 108. Diagram of an air injector.

The flow rate of the fuel is regulated by a bypass. Good atomization and carburetion is determined by the high velocity of the gas flow and by its high temperature. Despite low flow rates of fuel, the openings in the tube supplying the fuel, can be large, since the pressure drop between the fuel and the gas is small. Injectors of this type can be used even in the main chambers.

The plan of the compound injector for supplying both liquid and gas fuel (Fig. 109) is made on this principle. Fuel here is supplied via tube 1 to the annular duct and flows out into the gas (air) flow according to the arrows with subscript G_T through a number of openings. Primary gas (air) moving in the direction of arrow G_1 and being swirled by vanes 2 catches the fuel and emerges into the combustion zone in the direction of arrow G_3. Such a diagram for an annular reheat combustion chamber (or main chamber) is shown in Fig. 110. Fuel manifold 1, just as central cone 2, are annular. Around the ring of the fuel manifold there are borings 3, through which the fuel exhausts into the flow of gas (air).

§ 50. Principles of Calculating Reheat Combustion Chambers

The general approach to planning and calculating reheat combustion chambers and the plan of the sequence of calculation remains the same as for the main combustion chambers. To calculate reheat chambers the following should be given:

1) total gas consumption of the GTD, $G_{g\Sigma}$;

2) temperature of gas on intake to the reheat chamber t_{g};

3) pressure of gas on intake to the chamber, p_{g};

4) temperature of gas on outlet from the chamber, t_{g_1};

5) fuel (its form and grade);

6) the class of the engine.

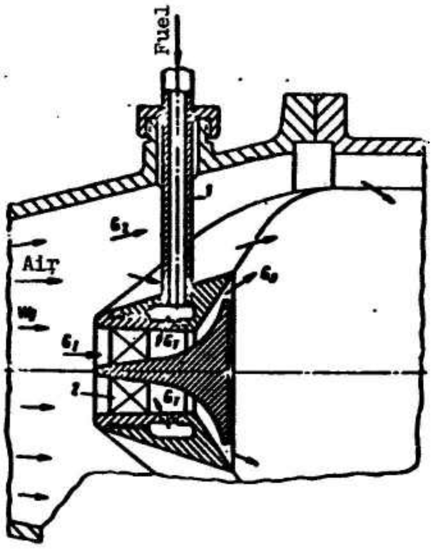


Fig. 109. Diagram of a compound injector.

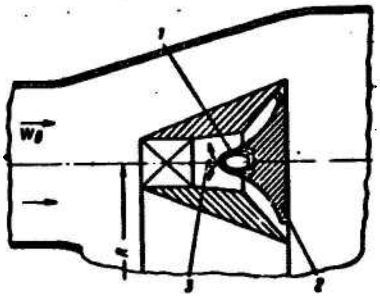


Fig. 110. Diagram of an annular injector: 1 – annular fuel manifold; 2 – flame holder; 3 – fuel.

Just as in calculating the main chamber, initially according to the design of the engine and to the requirements imposed on it, the layout and the number of chambers are selected. If a section design has been accepted, then design of one of n chambers of the block is calculated for consumption $\frac{G_{г\Sigma}}{n}$.

For simplicity, in the future we will consider a GTU with one individual main chamber and one individual reheat chamber. Then on the intake to the reheat chamber $G_г = G_{г\Sigma}$. In the absence of any additional taps of the working medium

$$G_г = G_в + G_T = G_T(1 + \alpha_\Sigma L_0), \qquad (79)$$

where $G_в$ — total air consumption; G_T — total fuel consumption in the main combustion chamber; L_0 — stoichiometric quantity of air; $\alpha_\Sigma = \frac{G_в}{G_T L_0}$ — overall excess air ratio in the main combustion chamber.

In the overall quantity of gases $G_г$, entering the reheat chamber there are pure products of combustion of the fuel which burned down in the main chamber in a quantity of $G_{ч.г}$, and a remainder of air not used $G_{ов}$.

The quantity of pure products of combustion comprises

$$G_{ч.г} = G_T(1 + L_0). \qquad (80)$$

The quantity of air remaining

$$\begin{aligned} G_{ов} = G_T(1 + \alpha_\Sigma L_0) - G_T(1 + L_0) = G_T L_0(\alpha_\Sigma - 1) = \\ = G_в - G_T L_0. \qquad (81) \end{aligned}$$

In the reheat chamber, to raise the temperature of the gases from $t_г$ to $t_{г1}$ it is necessary to burn a thus far unknown quantity of fuel G_{T_2} kg/s.

To burn 1 kg of fuel it is theoretically necessary to have L_0 kg of air. But the air does not enter the reheat chamber and the gas with a lesser content of oxygen. Let us assume that for the oxidation of each kilogram of fuel there will be required L_{01} kg of gas. The value of L_{01} kg is determined in the following manner. In the flow of gas $G_г$ for each kilogram of air remaining it is necessary to have pure products of combustion

$$\frac{G_{\Sigma в}}{G_в} = \frac{G_T(1+L_0)}{G_T L_0(\alpha_\Sigma - 1)} = \frac{1+L_0}{L_0(\alpha_\Sigma - 1)},$$

consequently,

$$L_{01} = L_0\left[1 + \frac{1+L_0}{L_0(\alpha_\Sigma - 1)}\right] = \frac{1+\alpha_\Sigma L_0}{\alpha_\Sigma - 1}. \qquad (82)$$

In this way, the overall excess air (gas) ratio in the reheat chamber

$$\alpha_{\Sigma 1} = \frac{G_г}{L_{01} G_{T1}} = \frac{G_T(1+\alpha_\Sigma L_0)}{G_{T1}(1+\alpha_\Sigma L_0)}(\alpha_\Sigma - 1) = \frac{G_T}{G_{T1}}(\alpha_\Sigma - 1). \qquad (83)$$

Since in this expression the values of G_{T1} and $\alpha_{\Sigma 1}$, then it is necessary to use the heat balance equation for defining first the value of $\alpha_{\Sigma 1}$, and then G_{T1}. The heat balance equation of the reheat chamber , relative to 1 kg of fuel supplied and 0°C:

$$i_T\Big|_0^{t_T} + \alpha_{\Sigma 1} L_{01} i_г\Big|_0^{t_г} + Q = (1+\alpha_{\Sigma 1} L_{01})\, i_{г1}\Big|_0^{t_{г1}}. \qquad (84)$$

The right-hand part of the equation expresses the heat content of the intermixture of gases on outlet from the chamber at a temperature of $t_{г1}$. The first member of the left part of the equation – this is the heat content of 1 kg of fuel at the temperature of fuel on intake to the chamber, the second member – the heat content of the gas entering the chamber in a quantity which is necessary for 1 kg of fuel.

The value of Q is equal to the part of the heat which was given off in the combustion of the fuel which goes to heating the working medium. It is defined just as during the analysis of the heat balance equation [expression (52)] for the main combustion chamber:

$$Q = Q_н^р - Q_{H_2O} - Q_{окр} - Q_{нед}, \qquad (85)$$

where Q_{H_2O} — the heat spent on evaporation of moisture in the products of combustion; $Q_{окр}$ — the heat returned to the environment: $Q_{нед}$ — the heat lost as a result of incomplete chemical and mechanical burning.

It is especially necessary to explain the value $Q^p_{в0}$. In the given expression this should be the higher heat of combustion of the fuel relative to 1 kg of fuel in the mixture at 0°C, of a mixture analogous in composition in the reheat chamber. Taking into account what was pointed out above (see § 38), it is permissible here to determine the value of $Q^p_{в0}$ during the combustion of fuel in pure air. Then

$$Q^p_{в0} = Q^p_{н t_0} + Q_{H_2O} + \Big[(1 + \alpha_{\Sigma_1} L_{01})\, i^{*}_{z1}\Big|_0^{t_0} - \\ - \alpha_{\Sigma_1} L_{01} i^{*}_{e}\Big|_0^{t_0} - i^{*}_{T}\Big|_0^{t_0}\Big]. \qquad (86)$$

Further, the index t_0 – the standard temperature at which the heat of combustion of fuel Q^p is determined, will be omitted. Substituting expression (86) in equation (85) and designating

$$Q^p_н - Q_{окр} - Q_{нед} = \eta_{z1} Q^p_н,$$

where η_{z1} — the coefficient of combustion efficiency of the fuel in the reheat chamber, it is possible to rewrite the heat balance equation in this way:

$$i^{*}_{T}\Big|_0^{t_T} + \alpha_{\Sigma 1} L_{01} i^{*}_{e}\Big|_0^{t_e} + Q^p_н \eta_{z1} + (1 + \alpha_{\Sigma_1} L_{01})\, i^{*}_{z1}\Big|_0^{t_0} - \\ - \alpha_{\Sigma 1} L_{01} i^{*}_{e}\Big|_0^{t_0} - i^{*}_{T}\Big|_0^{t_0} = (1 + \alpha_{\Sigma 1} L_{01})\, i^{*}_{z1}\Big|_0^{t_{z1}}.$$

Adding the corresponding members of the equation, we obtain

$$\eta_{z1} Q^p_н = (1 + \alpha_{\Sigma 1} L_{01})\, i^{*}_{z1}\Big|_{t_0}^{t_{z1}} - \alpha_{\Sigma 1} L_{01} i^{*}_{e}\Big|_{t_0}^{t_e} - i^{*}_{T}\Big|_{t_0}^{t_T}. \qquad (87)$$

From this equation it is not possible to directly obtain the value of $\alpha_{\Sigma 1}$ since the heat capacity of the intermixture of gases on outlet from the chamber $\overline{c_{p_{z1}}}$ is unknown. Therefore, by analogy with the solution of the heat balance equation for the main chamber we determine separately the quantity of pure products of combustion and air in the gases after the reheat chamber. It is known that on intake to the chamber, for each $(1+\alpha_\Sigma L_0)$ kg of gas there was $(\alpha_\Sigma - 1) L_0$ kg of air. This means in $(\alpha_{\Sigma 1}\cdot L_{01})$ kg of gas, necessary in the chamber for 1 kg of the fuel entering, the air will be

$$\frac{(\alpha_\Sigma - 1) L_0 \alpha_{\Sigma 1} L_{01}}{(1+\alpha_\Sigma L_0)} \quad \frac{\text{kg}}{\text{kg}}.$$

After combustion in the gas after the reheat chamber, for each burned kg of fuel there remains air

$$\left[\frac{(\alpha_\Sigma - 1) L_0 \alpha_{\Sigma 1} L_{01}}{(1+\alpha_\Sigma L_0)} - L_0\right] \frac{\text{kg}}{\text{kg}}. \qquad (88)$$

Analogically to this, of the pure products of combustion on intake to the reheat chamber for each $(1+\alpha_\Sigma L_0)$ kg of gas there was $(1+L_0)$

Then for $(\alpha_{\Sigma 1}\cdot L_{01})$ kg of gas of these products there will be

$$\frac{(1+L_0)\alpha_{\Sigma 1} L_{01}}{(1+\alpha_\Sigma L_0)} \quad \frac{\text{kg}}{\text{kg}}.$$

After combustion, for each kilogram of fuel there will be

$$\frac{(1+L_0)\alpha_{\Sigma 1} L_{01}}{(1+\alpha_\Sigma L_0)} + (1+L_0) \; \frac{\text{kg}}{\text{kg}}. \qquad (89)$$

of pure products of combustion. It is obvious that the sum of these components of the working medium on outlet from the chambers, calculated according to expressions (88) and (89), should give the total amount of gas which is necessary for 1 kg of fuel being burned. This quantity is equal to (1 + $\alpha_{\Sigma 1} L_{01}$) kg. Consequently,

$$\frac{(\alpha_\Sigma - 1) L_0 \alpha_{\Sigma 1} L_{01}}{(1+\alpha_\Sigma L_0)} - L_0 + \frac{(1+L_0)\alpha_{\Sigma 1} L_{01}}{(1+\alpha_\Sigma L_0)} +$$
$$+ (1+L_0) = 1 + \alpha_{\Sigma 1} L_{01}.$$

Then equation (87) can be rewritten thus:

$$\eta_{k1} Q^p_n = \frac{(1+L_0)\alpha_{\Sigma1}L_{01}}{1+\alpha_\Sigma L_0} I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} + (1+L_0) I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} +$$

$$+ \frac{(\alpha_\Sigma - 1)L_0\alpha_{\Sigma1}L_{01}}{1+\alpha_\Sigma L_0} I^{\circ}_{\text{в}}\Big|_{t_0}^{t_{z1}} - L_0 I^{\circ}_{\text{в}}\Big|_{t_0}^{t_{z1}} - \alpha_{\Sigma1}L_{01} I^{\circ}_{\text{в}}\Big|_{t_0}^{t_0} - I^{\circ}_T\Big|_{t_0}^{t_T}. \qquad (90)$$

In the composition of expressions (88) and (89) it was already considered in what quantity the air and pure products of combustion enter into the composition of $(\alpha_{\Sigma1} \cdot L_{01})$ units of the working medium on intake to the reheat chamber. Taking this into account, it is possible in equation (90) for the next to the last member of right hand part to be expressed by the sum of the quantities of air and pure products of combustion. Then equation (90) takes the form

$$\eta_{k1} Q^p_n = \frac{(1+L_0)\alpha_{\Sigma1}L_{01}}{(1+\alpha_\Sigma L_0)} I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} + (1+L_0) I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} +$$

$$+ \frac{(\alpha_\Sigma - 1)L_0\alpha_{\Sigma1}L_{01}}{(1+\alpha_\Sigma L_0)} I^{\circ}_{\text{в}}\Big|_{t_0}^{t_{z1}} - L_0 I^{\circ}_{\text{в}}\Big|_{t_0}^{t_{z1}} - \frac{(1+L_0)\alpha_{\Sigma1}L_{01}}{(1+\alpha_\Sigma L_0)} I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_0} -$$

$$- \frac{(\alpha_\Sigma - 1)L_0\alpha_{\Sigma1}L_{01}}{(1+\alpha_\Sigma L_0)} I^{\circ}_{\text{в}}\Big|_{t_0}^{t_1} - I^{\circ}_T\Big|_{t_0}^{t_T}$$

Summarizing, we obtain

$$\eta_{k1} Q^p_n = \frac{(1+L_0)\alpha_{\Sigma1}L_{01}}{(1+\alpha_\Sigma L_0)} I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} + (1+L_0) I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} +$$

$$+ \frac{(\alpha_\Sigma - 1)L_0\alpha_{\Sigma1}L_{01}}{(1+\alpha_\Sigma L_0)} I^{\circ}_{\text{в}}\Big|_{t_z}^{t_{z1}} - L_0 I^{\circ}_{\text{в}}\Big|_{t_0}^{t_{z1}} - I^{\circ}_T\Big|_{t_0}^{t_T}. \qquad (91)$$

After conversions

$$\eta_{k1} Q^p_n + L_0 I^{\circ}_{\text{в}}\Big|_{t_0}^{t_{z1}} + I^{\circ}_T\Big|_{t_0}^{t_T} - (1+L_0) I^{\circ}_{\text{ч.п}}\Big|_{t_0}^{t_{z1}} = \frac{\alpha_{\Sigma1}L_{01}}{1+\alpha_\Sigma L_0} \times$$

$$\times \left[(1+L_0) I^{\circ}_{\text{ч.п}}\Big|_{t_z}^{t_{z1}} + (\alpha_\Sigma - 1)L_0 I^{\circ}_{\text{в}}\Big|_{t_z}^{t_{z1}}\right].$$

whence

$$\alpha_{\Sigma 1} = \frac{1+\alpha_{\Sigma} L_{0}}{L_{01}} \times$$

$$\times \left[\frac{\eta_{z1} Q_{H}^{p} - (1+L_{0})\, i^{\bullet}_{\text{ч.п}} \Big|_{t_{0}}^{t_{z1}} + L_{0} J^{\bullet}_{\text{в}} \Big|_{t_{0}}^{t_{z1}} + i^{\bullet}_{T} \Big|_{t_{0}}^{t_{T}}}{(1 \cdot L_{0})\, i^{\bullet}_{\text{ч.п}} \Big|_{t_{z}}^{t_{z1}} + (\alpha_{\Sigma}-1) L_{0} J^{\bullet}_{\text{в}} \Big|_{t_{z}}^{t_{z1}}} \right], \qquad (92)$$

and in expanded form, using the average weight heat capacities, we obtain

$$\alpha_{\Sigma 1} = \frac{1+\alpha_{\Sigma} L_{0}}{L_{01}} \times$$

$$\times \left[\frac{\eta_{z1} Q_{H}^{p} - (1+L_{0}) \left(c_{p_{\text{ч.п}}} t^{\bullet}_{z1} - c_{p_{\text{ч.п}}} t^{\bullet}_{0}\right) + L_{0} \left(c_{p_{\text{в}}} t^{\bullet}_{z1} - c_{p_{\text{в}}} t^{\bullet}_{0}\right) + \left(c_{p_{T}} t^{\bullet}_{T} - c_{p_{T}} t^{\bullet}_{0}\right)}{(1+L_{0}) \left(c_{p_{\text{ч.п}}} t^{\bullet}_{z1} - c_{p_{\text{ч.п}}} t^{\bullet}_{z}\right) + (\alpha_{\Sigma}-1) L_{0} \left(c_{p_{\text{в}}} t_{z1} - c_{p_{\text{в}}} t_{z}\right)} \right]. \qquad (93)$$

Let us remember that the values $\bar{c}_{p_{\text{в}}}$ and $c_{p_{\text{ч.п}}}$ have already been determined in general form in Chapter XI.

Knowing the value of $\alpha_{\Sigma 1}$, the calculation of the reheat chamber is continued according to the system examined in § 43. However, in this case it will be necessary for number of values and parameters to be given. The average values of these quantities are given in Table 12.

Table 12. Average values of basic parameters of reheat combustion chambers.

Chambers	Fuel	Velocity in m/s			$\alpha_{\text{ф.р}}$	α_{z}	$h \cdot 10^{-6}$ in $\frac{\text{kcal}}{\text{m}^3 \cdot \text{h} \cdot \text{atm}}$ $\left(\frac{\text{kJ}}{\text{m}^3 \cdot \text{h} \cdot \text{atm}}\right)$, * atm = atm(abs.)	$\frac{l}{d}$	α_{Σ}
		w_{z}	w_{cp}	$w_{\text{в}}$					
Aircraft afterburner..	Kerosene	[25—60]	70—125	70—125	[0,2—0,5]	1,1—1,5	20—150 (167—418)	2—3	1,5—2,0
Reheat chambers of stationary and transport GTD	Heavy liquid and gas	20—60	50—100	60—110	0,3—0,7	1,5—2,5	6—25 (25,1—106)	2—3	1,4—4,0

Note. The values shown in the table in brackets for certain combustion chamber layouts are not characteristic. In view of relatively meager experience in the operation of reheat combustion chambers, the values of a number of characteristic values are as yet insufficiently reliable and are given as tentative.

In stationary GTD provision can be made for not single, but repeated supplementary preheating of the gas. In this instance, the calculation of the second and subsequent reheat chambers will not differ from the calculation of the first chamber. It is necessary only to pay special attention to the fact that for realization of complete fuel combustion it is necessary to provide a sufficient concentration of oxygen in the gas on intake to each chamber. In determining the composition of the working medium in any consecutively placed reheat chamber with repeated preheating of gases it is necessary to take into account the following.

After the combustion of fuel in the first reheat chamber, in the working gas after this chamber there remains air

$$G_{в1} = G_в - L_0 G_T - L_0 G_{T1} = \\ = G_в - L_0 (G_T + G_{T1});$$

corresponding to pure products of combustion after the chamber

$$G_{ч.г1} = G_T (1 + L_0) + \\ + G_{T1} (1 + L_0) = (1 + L_0) \times \\ \times (G_T + G_{T1}).$$

In sum, the quantity of gas (working medium):

$$G_{г1} = G_в + G_T + G_{T1} = \\ = G_T (1 + \alpha_\Sigma L_0) + G_{T1} = \\ = G_{в.1} + G_{ч.г1}.$$

In the calculation of the second reheat chamber we consider the first two combustion chambers (the main one and the first reheat chamber) as one fuel-burning device.

The total quantity of fuel which burned in it will be

$$G_T = G_T + G_{T1} \text{ kg/s}.$$

The total quantity of gas before the second reheat chamber is determined from the expression

$$G_{z1} = G_{z} = G_{e} + G_{T} \text{ kg/s}.$$

The total excess air ratio in the first fuel-burning device

$$\alpha_{\Sigma} = \frac{G_{e}}{L_{0} G_{T}}.$$

These data are taken as initial, and following the method of calculation of the first reheat chamber, the second reheat chamber is calculated, and so on.

BIBLIOGRAPHY

1. Abramovich G. N. Turbulentnyye svobodnyye strui zhidkostey i gazov (The turbulent free jets of liquids and gases). M., Gosenergoizdat, 1948.

2. Zel'dovich Ya. B. Teoriya goreniya i detonatsiya gazov (The theory of combustion and detonation of gases). Izd-vo AN SSSR, 1944.

3. Zuyev V. S. i Skubachevskiy L. S. Kamery sgoraniya vozdushnoreaktivnykh dvigateley (Jet engine combustion chambers). M., Oborongiz, 1958.

4. Idel'chik I. Ye. Gidravlicheskiye soprotivleniya (Flow friction). M., Gosenergoizdat, 1954.

5. Il'yashenko S. M. i Talantov A. V. Teoriya i raschet pryamotochnykh kamer sgoraniya (Theory and calculation of "straight-through" combustion chambers). M., Izd-vo "Mashinostroyeniye," 1964.

6. Knorre G. F. Topochnyye protsessy (Furnace processes). M., Gosenergoizdat, 1951.

7. Kuznetsov L. A. Kamery sgoraniya statsionarnykh gazoturbinnykh ustanovok (Combustion chambers of stationary gas turbine plants). L., Mashgiz, 1957.

8. Kurochkin N. N. Kamery goreniya gazoturbinnykh dvigateley (Combustion chambers of gas turbine engines). M., Gosenergoizdat, 1955.

9. Linchevskiy V. P. Toplivo i yego szhiganiye (Fuel and its combustion). M., Metallurgizdat, 1959.

10. Mikhaylov A. I., Gorbunov G. M. et al. Rabochiy protsess i raschet kamer sgoraniya gazoturbinnykh dvigateley (The operating process and calculation of combustion chambers of gas turbine engines). M., Oborongiz, 1959.

11. Mikheyev M. A. Osnovy teploperedachi (Fundamentals of heat transfer). M., Gosenergoizdat, 1949.

12. Paushkin Ya. M. Khimicheskiy sostav i svoystva reaktivnykh topliv (Chemical composition and properties of jet fuels). Izd-vo AN SSSR, 1958.

13. Paushkin Ya. M. Khimiya reaktivnykh topliv (Chemistry of jet fuels). Izd-vo AN SSSR, 1962.

14. Raushenbakh B. V., Belyy S. A. and others. Fizicheskiye osnovy rabochego protsessa v kamerakh sgoraniya vozdushno-reaktivnykh dvigateley (Physical fundamentals of the working process in combustion chambers of jet engines). M., Izd-vo "Mashinostroyeniye," 1964.

15. Storozhuk Ya. P. Kamery sgoraniya gazoturbinnykh ustanovok (Combustion chambers of gas turbine installations). In the collection "Novyye mashiny, oborudovaniye i sredstva avtomatizatsii." Seriya III. M., TsBTI, 1963.

16. Frank-Kamenetskiy D. A. Diffuziya i teploperedacha v khimicheskoy kinetike (Diffusion and heat transfer in chemical kinetics). Izd-vo AN SSSR, 1947.

17. Khitrin L. N. Fizika goreniya i vzryva (The physics of combustion and explosion). Izd-vo MGU, 1957.