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THERMAL DIAGRAMS OF THERMO-ELECTRICAL DEVICES (SELECTED CHAPTERS)

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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*	"
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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

GREEK ALPHABET

Alpha	Α	α	α		Nu	Ν	ν	
Beta	Β	β			Xi	Ξ	ξ	
Gamma	Γ	γ			Omicron	Ο	ο	
Delta	Δ	δ			Pi	Π	π	
Epsilon	Ε	ε	ϵ		Rho	Ρ	ρ	ϱ
Zeta	Ζ	ζ			Sigma	Σ	σ	ς
Eta	Η	η			Tau	Τ	τ	
Theta	Θ	θ	ϑ		Upsilon	Τ	υ	
Iota	Ι	ι			Phi	Φ	φ	ϕ
Kappa	Κ	ϰ	κ	ϰ	Chi	Χ	χ	
Lambda	Λ	λ			Psi	Ψ	ψ	
Mu	Μ	μ			Omega	Ω	ω	

This book deals with the classification of thermal circuits of the thermoelectric, electro-generating and cooling-heating devices. Main attention is devoted to the description of new thermal circuits of similar thermoelectric devices.

It is intended for scientific and technical personnel, who deal with the problems of heat technology.

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PREFACE

Thermoelectric devices possess such valuable qualities as the noiselessness of work, reliability, the ability to work prolonged period of time without maintenance, compactness and independence. Specifically, these qualities explain the relatively extensive use of thermoelectric devices, and also determine the areas of their greatest expansion at present - power supply for the devices of space vehicles, radio of meteorological stations, navigational marine buoys and the various devices of ground and water transportation, cathode protection for pipelines and metallic constructions.

The wider introduction of thermoelectric devices and, in particular into steady-state power engineering, is inhibited by their comparatively low energy efficiency and the high cost of the installed kilowatt. Therefore, a large number of laboratories and scientific organizations in the Soviet Union and many foreign countries carry out intensive studies on the processes which take place in work of thermoelectric devices and develop new semiconducting thermoelectric materials. As a result, ever more effective and economical devices are created.

The primary directions of works on improving the efficiency of thermoelectric devices are aimed at seeking new materials possessing a high thermoelectric figure of merit coefficient $z = \frac{e^2\delta}{\lambda}$ 1/°K, improvement, and also on the development of the technology of commutation of thermoelectric modules and a more effective heat transfers on hot and cold surfaces of the thermoelectric batteries which ensure minimum losses of temperature, i.e., at seeking methods for an effective coupling of batteries with the hot and cold heat sources.

Insufficient attention is given to the developments of new, more perfect thermal circuits and to the analysis of their effect on energy effectiveness of thermoelectric devices.

In this work an attempt is made to examine the thermal circuits of thermoelectric devices from the standpoint of their thermodynamic perfection, to evaluate the possible effect of the selection of thermal circuit on the energy efficiency of batteries. From this viewpoint, it is interesting to examine also the thermoelectric batteries in which the principal amount of heat is supplied or removed not through the surfaces of hot and cold junctions but within the thermal elements. What we have in mind are thermal elements whose branches are made permeable for the cooled substance when the battery is operating under the conditions of a cooler or for a heat-transfer agent (coolant) while in the regime of an electric-power generator.

In the permeable thermal elements the internal transfer surface can be very developed, as a result of which the heat transfer between the gas (liquid) blown through them and the solid material occurs at small differences in temperatures, i.e., almost reversible. This system of heat exchange gives rise to the fundamental and essential features in the work of the thermal element. Specifically, this opens up the new possibilities of effect on the cooling coefficient of a thermoelectric battery or on its efficiency in the capacity as a generator.

CHAPTER I

BASIC ELEMENTS OF THERMAL CIRCUITS OF THERMAL ELECTRIC DEVICES

The thermal circuit of any thermoelectric device usually includes similar basic units independently of its designations. Thermoelectric battery under any operating conditions must have hot and cold heat sources, a system of heat supply from a hot source to hot junctions and its removal from cold junctions to the cold source, and also an external load for which the battery is operating in generating regime, or an external power supply when operating as a cooler or a heat pump.

In examining the existing structures of batteries, it should be noted that they are basically assembled from thermal elements of correct rectangular form and a lesser number of these structures - from thermal elements of circular form. Used most infrequently are the trapeziform, hexahedral and other forms which are selected during the designing of thermoelectric devices only from the consideration of the convenience of commutation and operational features, since neither shape nor geometric dimensions of thermal elements affect the efficiency of the battery.

The basic difficulties encountered in the manufacture of batteries are due to the need for the electric switching of a considerable number of elements p- and n-types. However, there is no need for assembling batteries of large capacity, realizing a parallel-series commutation of semiconductor materials. It is more simple and reliable to assemble batteries of any capacity from separate modules, which are several commuted thermoelectric pairs. Such thermoelectric modules of low capacity make it possible to construct a battery of any design and with any capacity with high reliability of the parallel-series commutation.

Analogously multistage thermal elements can also be commuted into low-capacity modules. The questions of technology on the manufacture of the p- and n-type junctions, their switching into thermoelectric modules and on the design formulation have been covered in many works [5, 16, 24], and there is no sense to dwell in detail on these questions.

Let us examine in more detail other common circuit elements of thermoelectric devices, which actively participate in the operation of a thermal circuit, giving it the characteristic features.

1. HEAT SOURCES

One of the principal units in the thermal circuits of thermoelectric generators is the source of thermal energy whose selection has an important significance when designing a generator, depending on the necessary duration of its continuous operation, designation, useful electric power, etc.

The basic heat sources in the generators manufactured at the present time and in those being designed are the combustion products of chemical fuels, radiant solar heat, radioisotopes, and nuclear reactors.

Chemical fuel is used basically as the heat source to supply power to radio equipment, for illumination, and for the cathode protection of metallic structures and pipelines which lie in almost inaccessible regions.

In the transport devices where the overall dimensions and weight of the electric power-supply system play a decisive role, the use of chemical fuels in the majority of cases is inexpedient, because to ensure a continuous operation of the generator a large amount of fuel is required.

When developing a heat source based on chemical fuel the primary attention is given to the designing of a burner device and to the selection of an optimum, from the standpoint of organization of the burning process, combustion chamber configuration. The burner device and combustion chamber must provide the necessary heat-flux densities at a given temperature of hot junctions and maximum combustion efficiency of fuel. When using high-temperature thermoelectric materials in the manufacture of branches of the thermal elements, the solution of this problem can cause considerable difficulties.

In principle, any type of chemical fuel can be used in the thermoelectric devices - liquid, solid and gaseous. The working substances of different thermal cycles (steam, hot water, depleted combustion products) can also be used as a heat source. In this case the generator can be situated almost any place in the thermal circuit of a particular heat-utilizing unit. In the case where the waste heat of the working components of thermal cycles or the exhaust gases of units operating on the combustion products of organic fuels are used to obtain electrical energy, such generators will contribute to an increase in energy of the main cycle.

Radiant solar heat is a promising and, in certain cases, cheap heat source for the generators of the outer-space and ground-based items which are located in the favorable with respect to weather areas of the Earth.

Favorable operating condition for the generators are determined by the number of sunshine hours and by total radiation which falls on the earth surface in a given region [3].

The utilization of radiant solar heat under conditions of space flights as a sole heat source for a generator is still rather complicated and expensive. This is explained by complexity and the high costs of the solar-radiation concentrators which create the heat-flux densities required for the normal operation of the generator or collectors which absorb the radiant energy, by the complexity of the tracking devices in the concentration systems which stay on the light as the space vehicle turns, by the need to use storage batteries when the space vehicle is in the shadow of the Earth or another planet. However, the present solar-power converters designed to operate onboard the space vehicle flying at certain distances from the Sun already have an advantage with respect to cost, weight per unit of power, and reliability over the systems with photoelectric elements, systems with a turbogenerator with nuclear-reactor type heat source, etc. Such results are obtained when comparing the various methods for the generation of electrical energy for the converters with the net power up to 100 kW [29].

An interesting direction of works on the creation of solar generators in space application, the successful development of which will make the position of such converters even stronger, is the development and creation of heat accumulators which ensure a normal operation of the generator while the space item is in the shadow. It is proposed that hydride and fluoride of lithium [32] be used basically as the heat-storing materials.

When constructing solar generators designed to be used on the ground the primary attention is devoted to the development of the solar-radiation concentrators with systems of diurnal tracking and annual declination. To date, a considerable number of concentrator structures have been designed and tested. Tests were made on devices having a system with an autonomous heating of each thermal element by a different concentrator and with paraboloidal concentrators of large diameter, operating for a group of thermocouples, different systems of solar-power receivers have been developed which convert the radiant energy received from the concentrators to thermal [1].

Radioisotope heat sources have been used quite extensively in the recent years in the creation of generators designed to be used for various purposes. The use of the heat released upon decay of radioisotopes to heat the hot junctions of a generator makes it possible to create a reliable, autonomous, comparatively long-working system of power supply to be used on space vehicles, and also on the ground and under water. Despite the fact that radioisotope sources cannot ensure large heat-flux densities, they are ideal heat sources for the generators of low output (tens and hundreds of watts) because they are small and simple to manufacture.

The radioisotope sources, unlike other heat sources suitable for use in thermoelectric devices, are characterized by the value of releasable heat output which decreases in time. Therefore, when selecting an isotope for the generator designed for a particular purpose, in addition to other conditions it is necessary to select the heat output of the isotopic fuel taking into account the value of the useful heat output required at the end of the operating period of this device. Sometimes the isotopic source is calculated from the average value of its heat output in the required period of the operating time of the generator.

The basic characteristics of radioisotopes which are suitable for use as the heat sources in generators are given in Table 1. Data on the specific heat output of the radioisotope fuel are taken from works [5, 16, 17, 29, 31, 34]. A certain discrepancy in these data is explained by the different specific gravity of the radioisotope fuels used.

The physico-chemical properties of the compounds which contain a radioactive isotope and are used as the radioisotope fuel must correspond to a number of specific requirements. The fuel should be easily prepared, should not react with the fuel-capsule

Table 1

Isotope	Fuel	Mode of decay	Halflife, in years	Heat output of fuel, in W/g	Heat output of pure isotope, in W/g
Po^{210}	Po	α	0.38	141.00	141,00
Cm^{242}	Cm_2O_3	α	0,45	99,5 — 120	120,00
Cl^{144}	ClO_2	$\beta - \gamma$	0,78	1,96 — 2,30	25,60
Pu^{238}	PuC	α	86,40	0,55 — 0,56	0,56
Pm^{147}	Pm_2O_3	$\beta - \gamma$	2,60	0.16 — 0,34	0.34
Cs^{137}	CsCl	$\beta - \gamma$	33.00	0,249 — 0,33	0,42
Sr^{90}	$SrTiO_3SrO$	$\beta - \gamma$	27,7 — 28,0	0,113 — 0.92	0,95
Co^{60}	CO	$\beta - \gamma$	5,30	0,30	17,40
Ru^{106}	Ru	$\beta - \gamma$	1.00	29,80	33,10
Cm^{244}	Cm_2O_3	α	17.60	2,30	2,80
Au^{198}	Au	—	1.00	29,50	—
Cr^{137}	—	$\beta - \gamma$	26,60	0.072	--
Tm^{170}	Tm_2O_3	β	0,35	1.03	15,60
Tl^{204}	Tl_2O_3	β	4.00	0.12	0,67
Th^{228}	ThO_2	α	1.90	141.00	170,00
U^{232}	UO_2	$\alpha - \gamma$	74,00	3,30	4,40
Am^{241}	Am	α	458,00	0,10	0,11

material, should be heat-resistant, and should have a high melting point. The concentration of the isotope in the fuel must ensure the required heat-flux densities [20]. In obtaining the fuel which corresponds these basic requirements, different inert compounds or metallic additions are introduced in it, which affects the specific gravity of the fuel. This explains the discrepancy in the data on the specific heat output of radioisotope fuels, published in the Soviet and foreign literature.

To date, a relatively large number of generators have been built with a radioisotope heat source which have a number of advantages in comparison with other heat sources, both in space as well as in the installations designed to operate on the ground and water with the power on the order of tens and hundreds of watts. However, their application is limited by biological hazards. The need for creating a biological protection leads to a significant increase in weight and raises the production cost of generators.

The nuclear sources of thermal energy are used during the developments of generators with high output. They have a long

Table 2

Reactor	Country	Heat output, in kW	Temperature in the active region, in °K	Fuel	Heat-transfer agent	Weight, in kg	Notation
SNAP-2(SDR)	USA	50	980	$ZrH+U^{235}$	NaK	113.5	Operated 1000 h
"Romashka"	USSR	40	2173	Uranium dicarbide +90% U^{235}	Without the heat-transfer agent	-	-
SNAP-10A	USA	15	-	$ZrH+U^{235}$	Without the heat-transfer agent	79-91	-
SNAP-8	USA	40	838	-	NaK	123	43 days in orbit 1300 km in 1965
SNAP-8	USA	500	1088	$ZrH+U^{235}$	NaK	228	Increased version SNAP-2
Martin-Marieta Corp.	USA	-	-	ZrH+U	Trisulfide phosphorus	-	Project
Martin-Marieta Corp.	USA	2500	-	UO_2+H_2O	Boiling water	-	-

autonomous operation of the thermoelectric device and large heat-flux densities at high temperatures.

There are two basic varieties of nuclear reactors which differ in the way heat is supplied to hot junctions of the generator: with an intermediate heat-transfer agent and with a direct heat supply from the fuel elements of the reactor or its outer shell. Table 2 gives the fundamental characteristics of some nuclear reactors designed for heating the hot junctions of generators.

With the aid of nuclear reactors, in which the heat is transferred by the heat-transfer agent from the active region to the hot junctions of the generator, it is possible to ensure isothermal heat supply over the entire surface of the hot junctions, in particular, if the substance which changes its state of aggregation in a given power cycle is used as the heat-transfer agent.

Furthermore, the intermediate heat-transfer agent also provides a partial radiation shielding for the semiconductor substance. The nuclear heat source with an intermediate heat-transfer agent makes possible the independent construction and final adjustment of the generator and the nuclear reactor. However, the heat-transfer agents used at the present do not make it possible to fully utilize the temperature capabilities of nuclear reactors and thermoelectric semiconductor materials.

In nuclear heat sources without the intermediate heat-transfer agent the thermal elements can be placed directly on the reflector surface, as this is done in the Soviet nuclear thermoelectric unit "Romashka" [17] (Fig. 1). Fuel elements in the form of plates from uranium dicarbide with 90% enrichment of U^{235} are arranged in graphitic cylindrical reactor container. The active region is surrounded by a reflector made from beryllium, which makes it possible to achieve 1270°K on the surface of the beryllium reflector. In this case the temperature of hot junctions of the thermal elements which are pressed against the beryllium reflector surface was 1223°K. Using the "Romashka" unit as an example, it is evident how important it is to improve the systems of contact heat exchange. With a more perfect method of heat transfer between the contiguous rigid surfaces it is possible to achieve a considerably lesser difference in temperatures between the heating surface and hot junctions, which would make it possible to make a more effective use of the thermoelectric materials whose

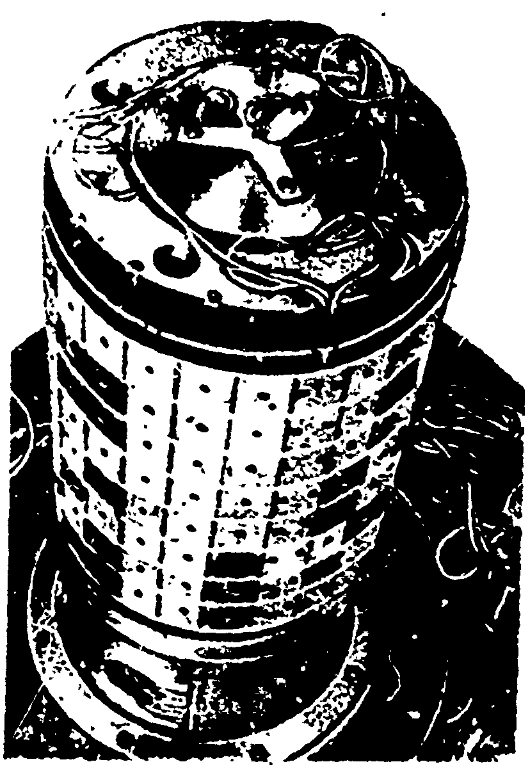


Figure 1. Nuclear reactor of thermoelectric unit "Romashka".

maximum efficiency is reached at the maximum operating temperatures.

The thermoelectric unit SNAP-10 [29] has a similar design which differs from the "Romashka" unit by lower temperature of hot junctions - 811°K.

In the units of similar designs the end faces of a nuclear reactor are not enclosed by thermal elements and require reliable insulation to avoid heat dissipation which can be considerable.

Compactness and relative simplicity distinguish the nuclear thermoelectric units in which thermal elements are combined with fuel elements of the reactor. The thermoelectric fuel elements can be used in a reactor of any type when the heat-transfer agent does not have the property of electric conductivity.

There are several design versions of fuel elements which are combined with thermal elements; however, they are all a modification of the basic version (Fig. 2). From the figure one can see that the thermal elements are placed on the capsule surface with a fissionable material. The fuel elements with this design can be of any shape (the most common elements are in the form of rods or plates).

When using radioisotope and nuclear heat sources the semiconductor thermoelectric materials operate under the effect of different froms of radiations which can cause changes in their thermoelectric properties. The principal effect on the material properties is exerted by γ-radiation of radioactive isotopes and nuclear reactors, and also by neutron radiation of reactors. Other forms of radiations that accompany fission reactions have small penetrating power and are usually absorbed by the structure elements without affecting the material properties. The questions connected with the effect of radiation on the properties of materials have still not been studied sufficiently. However, according

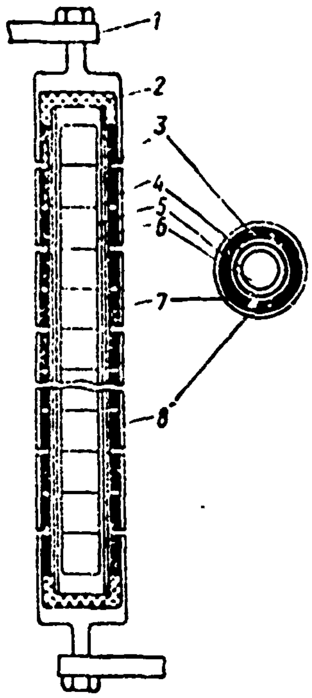


Figure 2. Thermal elements combined with fuel elements: 1 - current taps, 2 - electrical insulators, 3 - hot junctions of thermal elements, 4 - electrical insulation, 5 - fuel container, 6 - fuel, 7 - thermal elements, 8 - cold junctions.

to the literature there is a large number of semiconductor thermoelectric materials capable of working satisfactorily under the conditions of radioisotopic and nuclear-reaction heating of hot junctions of a generator.

The ambient medium is usually used as the cold source in the thermal circuits of thermoelectric devices. Depending on the purpose of the thermoelectric device water, air or outer space can be considered as the ambient medium. Heat from cold junctions of a generator or from hot junctions of a cooler to the cold source can be removed either directly or with the aid of an intermediate heat-transfer agent. Cold source has a substantial effect on the design and energy characteristics of a thermoelectric device, since the temperature level and the working medium determine the degree of intensity of heat removal from the junctions, which, in turn, determines the design and the overall dimensions of the heat transfer system.

2. HEAT SUPPLY AND REMOVAL SYSTEMS

These systems play a considerable role in the designing and operation of thermoelectric devices and units. The selection of methods for supplying heat to hot junctions of generators and cold junctions of coolers, in many instances, determines the weight-size indices of devices and their energy efficiency. During the development of those or other thermoelectric devices, usually all the basic modes of heat transfer are used - thermal conductivity, natural or forced convection or radiation - depending on their purpose and the specificity of operating conditions. Sometimes the combined systems of heat transfer are used, for example, the heat is transferred simultaneously by convection and radiation.

The transfer of heat by thermal conductivity occurs with the heating of hot junctions of a generator by direct pressing against the exothermal surface and during the removel of heat from its cold junctions to the heat-absorbing surface contiguous with them. In cooling devices it is the opposite. The thermal conductivity of materials used for the branches of thermal elements in these cases is of considerable significance during the designing of thermoelectric devices.

A direct transfer of heat from the source to the junctions of a thermoelectric device indicates a need for the creation of a heat transfer consisting of several layers of materials with different thermal conductivity - sealing, electroinsulating, commutating, etc. Therefore, the calculation of heat transfer is made by the dependences obtained for multilayer walls [28].

Nuclear thermoelectric systems such as "Romashka" and SNAP-10, in which thermal elements are arranged directly on the outer shell of a nuclear reactor, can serve as examples in the use of the thermal conductivity process for supplying heat from the source to the hot junctions. A similar principle is used to supply heat to the hot junctions in radioisotopic generators (Fig. 3).

Despite the relative simplicity of the heat-transfer process by thermal conductivity, this method has a number of significant deficiencies. In the case of a contact between the heat-transfer and heat-absorbing surfaces, it is not always possible to provide the necessary for a normal operation of the device heat-flux densities at a fixed temperature of the latter. Furthermore, it is not always possible to provide a convenient structural connection between the generator and the heat source. This connection can be the cause of the additional heat losses.

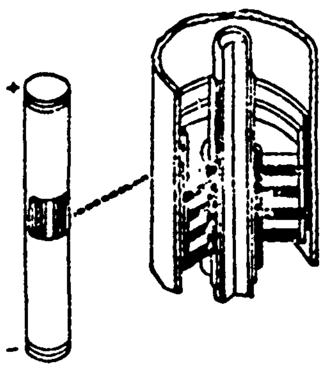


Figure 3. Radioisotope generator in which the heat is supplied to the hot junctions by thermal conductivity.

More efficient are the systems of convective supply and removal of heat in the thermoelectric devices which use gas, liquid or liquid-metal heat-transfer agents. The convective systems of heat supply make it possible to get rid of the rigid structural connection between the generator and the heat source. With a nuclear-reactor heating the convective heat-transfer system maintains a lower temperature of the reactor fuel elements with the same temperature of the hot junctions of the generator, accomplishing a considerable output of heat-flux densities from the reactor core. The rate of heat exchange during convective heating or cooling is determined by the velocity of the medium, its thermophysical properties, and the shape and dimension of the exothermal (heat-absorbing) surface.

During the supply or removal of heat from the junctions of thermoelectric devices under the conditions of natural (free) convection of the ambient medium the coefficient values prove to be very low.

For the calculation of heat-transfer coefficients α for plane walls and tubes the following equations can be recommended [16]:

$$\left.\begin{array}{l} Nu = 1{,}18(GrPr)^{1/8} \text{ for } 1\cdot10^{-3}<(GrPr)<5\cdot10^{2}, \\ Nu = 0{,}54(GrPr)^{1/4} \text{ for } 5\cdot10^{2}<(GrPr)>2\cdot10^{7}, \\ Nu = 0{,}135(GrPr)^{1/3} \text{ for } 2\cdot10^{7}<(GrPr)>2\cdot10^{13}, \end{array}\right\} \quad (1)$$

where $Nu = \frac{\alpha d}{\lambda}$, $Gr = \frac{g\beta l^3\Delta t}{\nu^2}$, $Pr = \frac{\mu c_p}{\lambda}$, β - volumetric expansion coefficient, ν - kinematic viscosity, l - characteristic dimension, Δt - temperature head, λ - coefficient of thermal conductivity, c_p - specific heat, α - heat-transfer coefficient, μ - coefficient of dynamic viscosity. In this case, values of α=3-6 W/m^2·deg.

With natural convection, to increase heat removal (heat supply) the ribbing of junctions is used to increase the exothermal (heat-absorbing) surface. In this case the coefficient of the transfer of heat from the ribbed surface to the ambient medium depends on the shape, size, and the relative position of ribs.

The natural cooling of the ribbed surfaces of junctions is used, basically in those instances when the questions of energy efficiency yield the primary consideration to the weight and size characteristics of thermoelectric devices.

The system of ribbing increases considerably the weight and the overall dimensions of the thermoelectric device and complicates its production. However, under the conditions of natural convection, it has little effect on the increase in energy efficiency of the device. Therefore, to increase the rate of heat exchange when working with gaseous media, fans are frequently used which create a forced flow of gas along the ribs. In many instances the expenditures of the portion of net power produced by the generator to drive the fan are economically justified. For example, in the generators for a cathode protection of gas lines where an additional consumption of the burned gas does not play a significant role, the installation of a fan which blows on the cold junctions, due to the increased heat transfer, makes it possible to obtain a considerably greater temperature gradient on the junctions and, therefore, also higher net power from the unit volume of the thermoelectric material.

The conditions of heat exchange with the blowing of the ribbed surfaces of various configurations with ribs having a different shape by gas flows have been studied sufficiently well as applied to their working conditions in the various thermal units and apparatuses. However, all these data have not been generalized, and there are no sufficiently clear recommendations for the calculation of the coefficients of the transfer of heat from the ribbed surfaces of the cold and hot junctions of thermoelectric devices.

In the SKB [Special design office] of semiconductor devices studies were carried out on heat emission from the radiators of thermoelectric devices. For radiators with a slotted clearance, data have been obtained on heat transfer which are described by the expression [8],

$$\mathrm{Nu} = 1{,}55\left(\mathrm{Pe}\,\frac{d}{c}\right)^{1/3} C, \qquad (2)$$

where $\mathrm{Pe}=vdc_p\gamma/\lambda$.

A study was made of the slits from 0.5 to 1.2 mm wide with hydraulic diameters d=1.6-2.2 mm. From the standpoint of optimum conditions of heat removal and aerodynamic resistance, clearances 1-1.2 mm wide proved to be the best. Such systems provide heat removal with a temperature gradient of 2-4°K with heat fluxes of 2.2.5 $\mathrm{W/cm^2}$ on the junctions of thermopiles.

Sometimes it is more expedient to use water for cooling of the ribbed junctions. A comparison of the efficiency of air and water heat removal from the hot junctions of a thermoelectric conditioner [26] showed that the use of water cooling can substantially increase the refrigerating capacity of the conditioner and its cooling coefficient with a considerable decrease in the height and number of ribs. Thermoelectric devices with water-cooled junctions are most recommended for use on boats, marine navigational buoys, deep-water equipment, etc.

Heat-transfer coefficients in such cooling systems are calculated from known empirical equations [18]. In particular, for the turbulent flow of liquids and gases in channels.

$$\mathrm{Nu} = 0{,}023\,\mathrm{Re}^{0{,}8}\,\mathrm{Pr}^{0{,}4}, \qquad (3)$$

where Re=Ud/ , d - channel diameter, U - rate of flow of the coolant.

To improve the conditions of heat removal from the cold junctions of a generator the method of heat absorption by the vaporizing liquid with its subsequent emission into the ambient medium through the condenser with developed exothermal surface is also proposed [31]. The freon-113 with the 320°K vaporization temperature was used as the liquid coolant. This design did not require any pumps for its pumping since condensate returned from the condenser to the cold junctions by the force of gravity.

The heat-supply systems with liquid-metal heat-transfer agents draw ones attention, in view of the fact that nuclear reactors can be used as a heat source for generators. Such heat-transfer agents, as compared with gas, make it possible to substantially decrease the expenditures of power for pumping, increase the heat-exchange process and decrease the pressure in the nuclear reactor contour. However, at present the temperature possibilities of semiconductor thermoelectric materials and nuclear reactors exceed those of the liquid-metal heat-transfer agents whose maximum operating temperatures do not exceed 1100-1200°K.

The heat-transfer coefficients during the flow of liquid metal can be calculated by the following dependences [7]:

for a turbulent flow in round ducts

$$\begin{aligned} Nu &= 5 + 0{,}0021\,Pe, \\ Nu &= 5 + 0{,}025\,Pe^{0{,}8}; \end{aligned} \qquad (4)$$

for a longitudinal flow around bundles of rods

$$Nu = 6 + 0{,}006\,Pe; \qquad (5)$$

for the conditions of natural convection

$$\begin{aligned} Nu &= 0{,}67\left(\frac{Pr^2 Gr}{1+Pr}\right)^{1/4} \text{ when } 10^2 < Gr < 10^9, \\ Nu &= 0{,}16\left(\frac{Pr^2 Gr}{1+Pr}\right)^{1/3} \text{ when } Gr < 10^9. \end{aligned} \qquad (6)$$

In these dependences it is recommended that the height and diameter be used as the determining dimension for vertical walls and the ducts, respectively. These dependences were experimentally confirmed not only for molten metals, but also for oil, water and air [7].

In many instances the systems of heat supply and removal within the thermal elements proved to be very efficient. In this case the branchs of thermal elements must be made permeable. Permeability from certain junctions to others can be accomplished by various means. These can also be the finely porous thermal elements prepared by the methods of powder metallurgy and assembled from individual tubes or rods with a relatively small diameter which are sintered with one another for better electrical contact. Thermal elements can also be constructed by the perforation technique by means of an electric spark, electrochemical or laser perforation of small-diameter holes and also by special pressing methods.

For the calculation of such systems it is necessary to know the value of the heat-transfer coefficients both within the thermal elements, between the semiconductor substance and the heat-transfer agent (coolant) blown through it, and on both surfaces - hot and cold.

The heat exchange within the capillary-porous bodies is characterized by high intensity because of the very developed heat-transfer surface. In view of the very small transfer area of channels through which the exothermal or heat-absorbing substance is blown, it does not obey the law of heat exchange as it flows through the ducts and channels.

In the finely porous thermal elements prepared by the methods of powder metallurgy, the internal transfer surface is usually not determined, since there are blind and uneven pores, pinching of pores within the material, etc. In such systems one usually uses the volumetric heat-transfer coefficients α^*, W/cm$^3\cdot$deg. In the equations of heat exchange the usual geometric values cannot be used as the determining dimension due to the nonreproducibility during the pressing of the dimensional characteristics of the capillaries. In these cases, the value which characterizes the hydraulic properties of this specimen is usually used.

Many works have been devoted to the study of heat exchange within the finely porous bodies blown by various substances; however, there is still no general dependence for the calculation of the heat-transfer coefficient because its value is determined by a large number of factors which are difficult to calculate, by the properties of the material, by the technology of pressing, by the size of the original powder, by the blowing velocity of a substance, etc.

When manufacturing the permeable thermal elements by assembling rods or by boring small-diameter holes, the heat-transfer surface within the thermal element can be determined accurately. When calculating perforated thermal elements the use is made of the heat-transfer coefficient α, W/cm^2·deg. Heat exchange within the material having a large number of capillaries per unit volume has not been studied sufficiently. At present there are no reliable formulas for the engineering calculations of perforated systems.

Temperature fields within the permeable walls which are blown by a substance will be examined below in more detail.

Heat exchange on the external surfaces of permeable thermal elements is complicated by the blowing in or out of the substance through the capillaries. For example, when creating a temperature gradient on the junctions of a generator by means of blowing of the coolant in the direction away from the cold junctions to the hot, the condition of heat exchange on the hot and cold side of the generator will be completely different. On the hot side the coolant which emerges from the capillaries creates a gas layer between the hot junctions and the heating flow, which considerably impedes the heat transfer from the heating flow. The greater the flow rate of the blowing substance, the thicker is this layer whose temperature is considerably lower than that of the heating flow and, hence, the lower is the value of the heat-transfer coefficient α. The heat-transfer coefficient α_1 from the permeable wall to the external flow from the exit side of the blown substance can be calculated by the formula

$$\frac{\alpha_1}{\alpha_H} = 1 - 0{,}19\left(\frac{M_2}{M_1}\right)^b \frac{\rho v_w}{\alpha_H/c_p}, \qquad (7)$$

which was confirmed experimentally for wide range of change in parameters [19], where $b=0.35$ when $0.2<M_2/M_1<1$, $b=0.7$ when $1<M_2/M_1<8$; α_H - heat-transfer coefficient on the impermeable wall; M_2 and M_1 - molecular weight of the main flow and the blown substance, respectively; ρv_w - specific consumption of the blown substance.

On the cold side of the generator where the coolant enters the capillaries, there occurs an increase in the heat emission rate due to the narrowing of the viscous layer near the wall.

In the area where the substance enters the permeable wall it is possible to examine two versions of thermal conditions. In the case when substance departs through surface into the permeable wall, it is possible to assume that there is no heat loss into the ambient medium from the cold side of wall. The heat which left the wall surface returns inside the wall with the blown substance. In this case, there is no sense to consider the heat-transfer coefficient α as the value which characterizes the amount of heat removed from the surface.

However, if part of the flow of the substance is blown through the permeable wall, while the other part washes its surface from the outside, then there is a permanent loss of heat from the cold side of wall. It is specifically this part of the heat flux that is most interesting and important when examining the heat balances of thermoelectric devices with permeable thermal elements. When carrying out heat calculations for such devices it is always necessary to know the amount of heat that they lose permanently into the ambient medium. The heat-transfer coefficient α from the permeable wall to the coolant flow washing it, which characterizes the permanent heat loss from a given section of the wall, can be determined from the following relationships [14]:

$$\mathrm{Nu} = 1{,}68 \cdot 10^{-3}\,\mathrm{Re}_{\text{вх}}^{1{,}74}\left(\mathrm{Re}_w \frac{l}{d_{\text{экв}}}\right)^{-1{,}24}\left(\frac{l}{d_{\text{экв}}}\Pi\right)^{2{,}34}$$

$$\text{for } \frac{l}{d_{\text{экв}}}\Pi < 2, \tag{8}$$

$$\mathrm{Nu} = 0{,}017\,\mathrm{Re}_{\text{вх}}^{1{,}74}\left(\mathrm{Re}_w \frac{l}{d_{\text{экв}}}\right)^{-1{,}24} \text{ for } \frac{l}{d_{\text{экв}}}\Pi > 2,$$

where l - length of the channel. In these relationships the equivalent diameter $d_{\text{экв}}$ of the channel formed by the permeable thermal elements and impermeable walls of the structure can be used as the determining dimension; $\mathrm{Re}_{\text{вх}}=\rho U_{\text{вх}} d_{\text{экв}}/\mu$ - Reynolds number based on the speed of the flow ahead of the permeable thermal elements; $\mathrm{Re}_w=\rho U_w d_{\text{экв}}/\mu$ - Reynolds number based on the velocity at which the substance is blown through the thermal elements; $\Pi=P_{\text{пр}}/P$ - ratio of the perimeter portion occupied by the permeable walls $P_{\text{пр}}$ to the full perimeter of the channel. This correction takes into account the fact that during the structural formulation of a thermoelectric device the coolant (cooled substance, heat-transfer agent) will flow through the channel whose one or several walls are permeable thermal elements. The dependences given above were obtained experimentally as a result of the study of heat emission from the finely porous walls while the air was pumped out through them [14].

The systems of heat removal by radiation are used when thermoelectric devices operate under space conditions. Such devices must have a cooler-emitter to remove the heat from the junctions. In this case the heat to radiating surfaces from the junctions is removed either by thermal conductivity, or an intermediate heat-transfer agent. The amount of heat removed by radiation into space can be determined by the Stefan-Boltzmann equation:

$$Q = F\varepsilon\sigma T^4, \qquad (9)$$

where ε - emissivity factor of natural radiation of the body, $\sigma = 5.68 \cdot 10^{-8}$ W/$m^2 \cdot deg^4$ is constant.

From this equation one can see the dependence of the area F of the heat exchanger-emitter on the temperature T of its surface at a fixed flow rate Q of the heat being removed.

As an example of this system can serve the thermionic converter, the ground version of the space unit "Romashka", in which the heat removal was realized by radiation from the ribbed surface of the cold junctions into the medium simulating the conditions of space (Fig. 4).

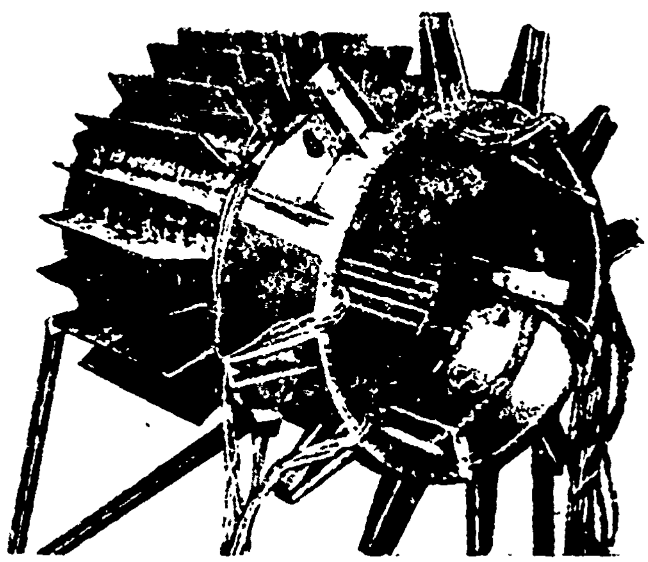


Figure 4. A thermionic converter of a nuclear-reactor system "Romashka".

3. POWER SOURCES

If for the generators, one of the basic elements is the source of thermal energy, then the operation of the thermoelectric coolers, conditioners, thermostats and heat pumps is affected considerably by the power supply. The selection of this source plays significant role because the direct current of large output with low voltage is necessary for the operation of thermoelectric devices. Storage cells, rectifiers and transformers of current, and direct-current generators suitable for the parameters of a given device are usually considered as power supplies.

The supplying of thermoelectric devices designed for domestic use and the industrial stationary devices in the presence of line electric power, in the majority of cases, is realized with the aid of rectifiers. However, rectifiers issue direct current with a certain percentage of pulsations. The presence of a variable component in the rectified current can be rather considerable for some circuits of rectifying devices. Current pulsations lead to a considerable deterioration of energy characteristics of batteries operating under conditions of coolers, conditioners or heat pumps. To exclude these, various types of filters are used, the most popular of which are the semiconductor diodes - germanium, silicon, selenic and cuprous oxide.

However, the use of filters for the smoothing of pulsations is sometimes difficult, since semiconductor thermopiles utilize high currents which reach hundreds of amperes. At the present time the industry produces germanium high-current diodes for currents in excess of 1000 A, which operate at relatively high efficiency.

Works carried out in the laboratory of semiconductors of the Odessa Technological Institute of the Food and Refrigeration Industry showed that the pulsations of current passing through thermopiles considerably affect their energy characteristics. For the single-phase full-wave rectification circuit most widely used in practice, the decrease in refrigeration and the cooling coefficient of units, as compared with the computed values (in calculations for a direct current without pulsations), can be 20-50% and higher depending on the operating temperature level of thermopiles [20].

A particularly strong deterioration in the energy indices occurs in the devices which operate at large differences in temperatures on the junctions of thermoelements. This situation forces one to strive for a maximum equalization of pulsations of the current that feeds the thermoelectric devices. If for any reasons it is impossible to eliminate or decrease the current pulsations, then the real constructions should be calculated by the formulas which consider their effect on the operation of the device.

To feed the thermoelectric devices which operate in the conditions of absence of line electric power, storage cells are used. The need for storage cells arises basically in the use of non-stationary devices. The operation of thermoelectric devices fed directly from a storage cell is limited in time by its capacity. In view of the fact that such devices utilize high current with low voltage, the storage cell is discharged rather rapidly. Therefore, to increase the operating time of a thermoelectric device, current converters, which are frequently used with the storage cell, in using from the storage cell a direct current of low power and relatively high voltage convert it to a current of high power and low voltage. The presence of a converter, naturally, leads to additional losses and the efficiency of the supply circuit decreases considerably.

Thermoelectric generators can often be used successfully as the power supply for the thermoelectric coolers, heaters and conditioners. The generator design can ensure the obtaining the necessary parameters at the output of a direct current for the feeding of a cooler or a heater.

CHAPTER II

CLASSIFICATION OF THERMAL CIRCUITS OF THERMOELECTRIC DEVICES

The joint arrangement of principal components examined in Chapter I makes it possible to create different circuits for the thermoelectric devices. The elements of thermal circuits during the operation are rigidly connected with one another and the creation of an efficient thermoelectric unit is impossible without the detailed analysis of their mutual effect.

The thermal circuits of thermoelectric devices are usually classed based on the difference of methods used to supply and remove heat on the junction surfaces of thermoelements. It is advisable to examine the generator circuits and base their comparison on a heat source as one of the principal units of the device. With regard to the circuits of the thermoelectric coolers, conditioners and heat pumps, their examination based on the difference in the heat supply and removal systems is justified.

1. GENERATOR CIRCUITS

Generator circuits with the heating of hot junctions by the combustion products are shown in Fig. 5. The heat to hot junctions is fed by convection from a gaseous flow of combustion products of an organic fuel. These circuits are differentiated by the way the temperatures gradient on the junctions and the removal of heat from the cold junctions of a generator are accomplished.

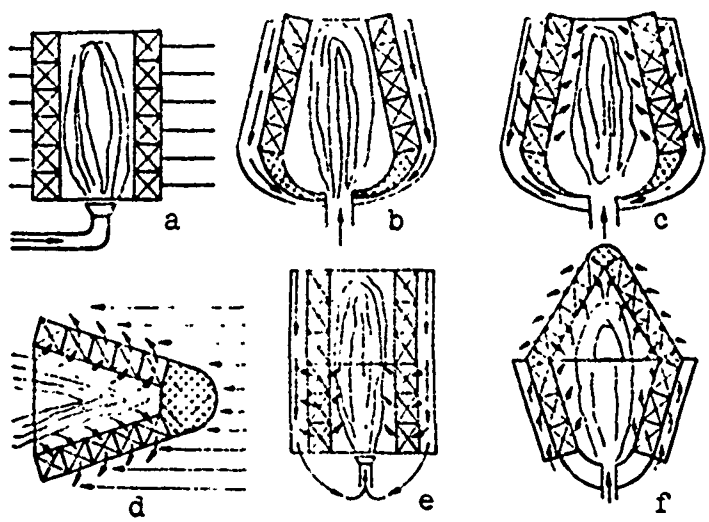


Figure 5. Generator circuits in which the heating of hot junctions is accomplished by the products of combustion.

The devices made according to the diagram shown on Fig. 5a, in which heat is removed by free convection from the ribbed surface of cold junctions, are usually used in difficult locations where it is impossible to continuously observe their operation. Specifically, they can be used for the cathode protection of gas and oil lines which are far removed from the populated areas. Under these conditions the energy efficiency of the supply source does not play a significant role, since very cheap fuel is available for the burner device of a generator; the reliability of a continuous operation of the device without maintenance becomes the predominant factor.

The generator of such a design will have very low efficiency. In a general case the efficiency of any generator where the heating of hot junctions is accomplished by the products of combustion is determined from the following expression: $\eta = N/BQ_p^H$, where N - useful electric power developed by the generator; B - consumption of fuel; Q_p^H - calorific value of burnt fuel.

It is known [6] that the useful electric power removed with the terminals of the generator depends on the properties of the used thermoelectric materials z and λ, the dimensions of δ and F, the battery, the temperature gradient ΔT on the junctions and on the ratio of the load impedance to the internal impedance m, i.e.

$$N = \frac{z\lambda m}{\delta (m+1)^2} \Delta T^2 F \, [W].$$

Hence it is evident that the temperature gradient on the junctions as a function of the developed electrical power squared is the determining factor for this material. The creation of the maximally possible temperature differential on the junctions of a battery is generally due to these two factors: the maximum permissible operating temperature of hot junctions, which is limited by the properties of the used semiconductor thermoelectric material, and the degree of the intensity of heat removal from the cold junctions. In this case it is assumed that the hot side of a thermopile can provide the heat-flux density necessary to maintain the maximum temperature of hot junctions with any degree of intensity of cooling of the cold junctions. The required heat-flux density in the combustion zone can be achieved by changing the flow rate of the burnt fuel and excess oxidant ratio, and also by producing optimum aerodynamic conditions for an intense heat exchange. It is natural that with cooling of cold junctions by means of free convection of the ambient medium it is not possible to remove large amounts of heat due to very low heat-transfer coefficients under these conditions, which means that it is not possible to achieve any large temperature differentials on the junctions. Therefore, in the designs such as shown in Fig. 5a, the removal of net power from the unit of thermopile surface is usually very small. Figure 6 shows one of the first Soviet generators, TGG-10, with a thermal circuit shown in Fig. 5a. This generator develops a net power of 10-12 W with a current strength of 1 A [4].

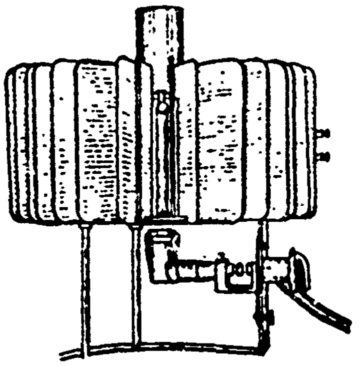


Figure 6. Generator TGG-10.

At present, the efficiency of devices which are powered by the combustion products is still rather low basically for two reasons. First, all the heat removed from cold junctions is lost into the ambient medium uselessly and, second, there is a considerable heat loss with the stack gases. It is possible to do something about the first heat loss, for example, the recovery of heat, (see Fig. 5b); however, in the case where the heat is lost with stack gases, the situation is more complex.

Combustion products leave the thermoelectric device at a temperature which exceeds that of hot junctions. This situation exists in all the designs made at present (see Fig. 5a and b), and also the systems which have not yet been examined, shown in Fig. 5c and e. The heat loss with stack gases in generators using organic fuel is the principal factor that determines their low efficiency.

In the case where the branches of thermal elements are made from high-temperature thermoelectric materials, the combustion products leaving the device have a high temperature and can be used in some other heat-utilizing device as a working medium. In this case the generator plays the role of a high-temperature extension of the main cycle, raising its energy efficiency. The total efficiency of this combined cycle is determined from the following relationship [27]:

$$\eta_k = \eta_b + \eta_r\psi(1 - \eta_b),$$

where η_0 - efficiency of the main cycle; η_T - efficiency of the high-temperature extension (efficiency of the generator); ψ - coefficient which is equal to the ratio of the heat supplied to the generator to the heat supplied directly to extended cycle.

When using a generator as the high-temperature extension above the cycle, which has the efficiency commensurate with its efficiency, the efficiency of the combined cycle can be increased almost two times. This will occur when a high-temperature generator is used together with a low-temperature generator.

A structural diagram of a generator designed to operate on organic fuel and constructed on the principle of heat recovery which is removed from the cold junctions (see Fig. 5b) is shown in Fig. 7. This generator, developed in the USA, can be made to produce different useful electric outputs [16]. The design of this generator provides for the utilization of any liquid organic fuel. Liquid fuel is located in a fuel tank under pressure. During the operation of the generator it is supplied through the line into a mixing chamber. With the evaporation of the fuel in the mixing chamber the air is drawn from the ambient medium through the inlet and the gas-air mixture is formed in the chamber which then is directed into the combustion chamber. Heat from the combustion products to the hot junctions of thermal elements is supplied with the aid of ribs which are on the surface of the combustion chamber. The ribbed surface of cold junctions is cooled by the air which circulates because of the pressure difference in the inlet and outlets, created by the ejecting effect of flow of the stack gases. A partial regeneration of heat removed from the cold junctions of the generator is realized by its transfer through the cooling surface and walls of the fuel container to the fuel.

Figures 5c and d shows the circuits which differ in principle from those manufactured at the present time by creating a temperature differential on the junctions.

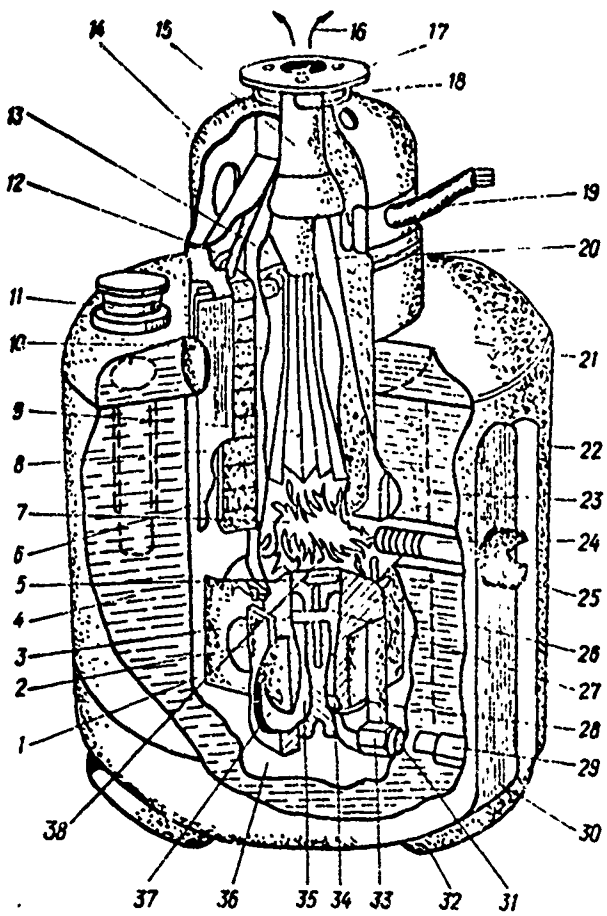


Figure 7. The design of a typical generator which operates on a liquid fuel: 1 - stiffening rib, 2 - ejector, 3 - radial fuel channels, 4 - liquid fuel, 5 - axial fuel valve, 6 - thermopiles, 7 - mixing chamber, 8 - cooling surface, 9 - combustion chamber, 10 - heat surface, 11 - fuel pump, 12 - air outlet, 13 - nozzle, 14 - jacket, 15 - exhaust pipe, 16 - exhaust, 17 - cover, 18 - flange, 19 - current taps, 20 - profiled cone,221 - fuel capacity, 22 - cowling of housing, 23 - cowling of the combustion chamber, 24 - igniter, 25 - ignition device, 26 - bottom of the mixing chamber, 27 - circular fuel pipe; 28 - air channel, 29 - jet, 30 - housing, 31 - fuel pipe, 32 - supports, 33 - control valve, 34 - fuel channel, 35 - ejection channels, 36 - air inlet, 37 - air cavity, 38 - bottom of the combustion chamber.

As was noted in Chapter I, the thermal elements of such generators should be permeable for the blowing of a substance in the direction from one junction to another. When the device operates according to the system shown in Fig. 5c, the coolant is blown through the permeable thermal elements which, by removing the heat from within the thermal elements, creates a temperature differential along their height and, upon emerging to the surfaces of the hot junctions is mixed with the heating flow. In contrast, in the device which operates according to the system shown in Fig. 5d, hot gas is blown through the thermal elements which gives off the heat within them and, after creating a temperature differential on the junctions, it exits onto the surfaces of the cold junctions and is mixed with the cooling flow.

The principle difference of these circuits from those constructed at present is in the fact that the principal amount of heat is supplied (removed) not through the surfaces of the hot (cold) junctions, but within the thermal elements through their very developed internal heat-exchange surface.

As was noted above, because of the heat exchange features in the finely porous or capillary systems the heat from the blown flow to the material of the thermal element branches (or vice versa) is transferred at an insignificant difference in temperatures of the heat-exchange media, i.e., almost thermodynamically reversible. This means that within the thermal elements the temperatures of the material and the blown substance will be equal or very close to one other along their entire height. Temperature profiles in the blown systems and their effect on the energy characteristics of thermoelectric devices will be examined in the following chapters.

Construction of permeable thermal elements from many of the thermoelectric semiconductor materials will not present any particular difficulties, as compared with the manufacture of the monolithic thermal elements from the same materials.

The technology of pressing of the finely porous permeable specimens has been mastered well for a whole series of materials both metallic and cermet and require only the finishing for many of the semiconductor thermoelectric alloys. From working experience it is evident that the creation of permeable branches for the thermal elements is a completely solvable problem. The most significant difficulty in producing of the finely porous batteries is the solution of the problem of parallel-series commutation of the thermal element branches with preservation of the permeability of the commuted surfaces.

In examining the expediency of using the permeable thermal elements from the standpoint of the insufficiently developed technology of their production at the present time one should consider the fact that permeability can be created not only by the methods of powder metallurgy, but also by other methods. For example, the permeable thermal elements can be obtained by means of electrochemical or electric-spark perforation of holes with a sufficiently small diameter in monolithic thermal elements prepared according to the known and well-developed procedures. The electrode used to make perforations can be made in the form of a comb which makes it possible to simultaneously pierce a certain number of holes. Such a method of creating a penetrability completely justifys itself in the work with comparatively soft thermoelectric materials. Figure 8 shows the thermoelectric modules which consist of five p - n-couples, whose branches are made from semiconductor thermoelectric materials based on Bi, Te and Se. In these modules, each square centimeter of the branch material has 15 holes 0.45-0.55 mm in diameter. The difference in the hole diameters is explained by the fact that the tungsten comb electrodes are not ideally even and parallel to one another.

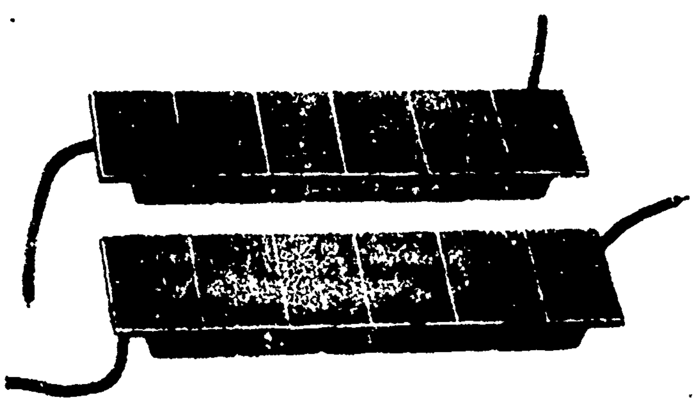


Figure 8. Thermoelectric modules with permeable thermal elements.

For solid high-melting materials (for example silicides) the electric-spark treatment is a long process; thus, this method is unsuitable for obtaining a large number of holes. Therefore, thermal elements made from refractory thermoelectric materials can be prepared using the method of slip casting or by pressing the original powder in the molds. Experimental specimens of perforated thermoelectric modules have already been obtained which are made from high-temperature materials which were prepared in dies by the methods of powder metallurgy. Thermal elements can also be obtained by sintering the previously pressed rods or small-diameter tubes or by other means.

The system shown in Fig. 5c makes it possible to achieve complete recovery of heat removed from the cold junctions of the battery just as the system in Fig. 5b; however, it has a whole number of advantages as compared with the latter.

In the operating of a generator with the second system the temperatures of the heating and cooling flows, which means also the temperatures of junctions, will sharply vary along the length of the generator. Furthermore, during the convective heat supply to the hot junctions and convective cooling of the cold junctions there are considerable parasitic temperature differentials both on the hot side of the thermopile (between the heating flow and hot junctions) as well as its cold side (between the cold junctions and the cooling flow). With their decrease, the ribbing of junctions is used which considerably complicates and weights the generator design. This impedes the possibility of a more or less full utilization of the maximum temperature head - between the initial temperature of the heating gas and the temperature of the coolant at the entrance to the device.

The system shown in Fig. 5c makes it possible to a considerable degree to eliminate the deficiencies from the system shown in Fig. 5b.

First, when using permeable thermal elements, such operating conditions under which the relatively high velocities of the blown substance provide an equality or almost full equality of the temperature of cold junctions and the temperature of the coolant ahead of them are possible. Consequently, it is possible o almost completely exclude the parasitic temperature differential on the cold side by ensuring a constant temperature of cold junctions of the battery along its entire length and close to the temperature of the cold source. This is possible, for example, with a sectional supply of the coolant when the length of sections is selected under the condition that the coolant does not get heated to any degree in the cooling channel prior to its entrance into the capillaries on the side of cold junctions.

Second, an oxidizer or a fuel can be used in such a generator as the coolant which is blown through the permeable thermal elements. The supply of the oxidizer (fuel), distributed along the length, makes it possible to maintain the temperature of hot junctions also almost constant along the length of the generator, i.e., as shown by calculations, at a sufficiently considerable distance from the combustion chamber (from the first thermal elements). This is achieved due to the gradual burnup of fuel in the oxidizer blown through the thermal elements or, on the contrary, due to the burning of fuel, blown through the thermal elements, in the flow of the oxidizer supplied in excess ahead of the first thermal elements.

Despite the difficult conditions of heat supply to the hot junctions (see Chapter I), this system ensures the obtaining of a maximum temperature differential on the junctions by most fully utilizing the available temperature head between the hot and cold heat sources.

Thus, under ideal conditions, this system can ensure a temperature constancy of the heating and cooling flows over the entire length of the battery with the preservation of a maximum temperature differential on the junctions. In this case the consumption of the coolant necessary for the creation of an identical temperature differential on the junctions in the system shown in Fig. 5c is incommensurably less than that in the operation according to that shown in Fig. 5b even with the presence of ribbing of the cold junctions of batteries in the latter system.

The comparison of the energy indices of the devices which operate according to the given schemes when they are used as a high-temperature superstructure of certain thermal cycle (the flow rate and the parameters of the entering hot gas flow are given) shows that the system in Fig. 5c provides a considerably greater yield of useful electric power with other conditions being equal. A more detailed analysis of the operation of the generator constructed in accordance with the scheme shown in Fig. 5c will be presented in the next chapter.

If in the system in Fig. 5c the temperature differential on the junctions of batteries is created due to the blowing of a coolant in the direction from the cold junctions to the hot (against the heat flux), then in the system shown in Fig. 5d the temperature differential on the junctions is created due to the blowing of the heat-transfer agent in the opposite direction. In the devices built in accordance with this thermal system the high-temperature products of combustion are blown through the permeable thermal elements in the direction from the hot junctions to the cold and, by releasing the heat within the material of thermal elements, create conditions for a normal operation of the generator with a sufficiently intense heat removal from the surface of its cold junctions. Due to the intense heat exchange on the hot side upon the entrance of the heating gas into capillaries, these systems permit one to make the temperature of hot junctions as close as possible to the temperature of the hot source. If the combustion products before the first thermal elements of a generator have an identical temperature, then the temperature of hot junctions can be maintained constant along the length of the generator when the heating gas to thermal elements is supplied either sectionally (as in the case of cooling) or from a closed volume perpendicular to the hot junctions. Such a system makes it possible to maximally utilize the temperature possibilities of the heat-transfer agent. The generator constructed according to this scheme is convenient for use as the utilizer of heat of the stack gases of any heat-utilizing device. In particular, the use of such a device as the second stage of the generator constructed according to the system in Fig. 5c ensures a considerable increase the efficiency of the thermal cycle of such a device (see Fig. 5f) [27].

Unlike all the systems examined above the system in Fig. 5d makes it possible to reduce to a minimum the heat loss with the stack gases. Actually, due to an intense heat exchange within the thermal elements, the actual conditions can be created to provide an exit for the heat-transfer agent on the side of the cold junctions at a temperature close to that of the cold junctions. Therefore, during a sufficiently intense external cooling of the cold junctions the heat-transfer agent will leave the device with a temperature close to that of the ambient medium.

Figures 5e and f show the combined thermal generator circuits. The diagram shown in Fig. 5e represents two generators built according to the systems in Fig. 5b and c, combined into one independent unit. The first high-temperature stage of such a generator makes it possible to obtain a maximally possible, under given conditions, temperature differential on the junctions due to the blowing-in of fuel through the permeable thermal elements and its gradual burning in the oxidizer supplied in excess at the entrance. The second low-temperature stage of this device uses the heat of the combustion products leaving the first stage, raising the total efficiency of the unit.

The system in Fig. 5f is most effective for the generators which use organic fuel. It provides for the operation of both stages of the generator with the maximally possible temperature differentials on the junctions and, at the same time, reduce the heat losses from the side of the cold junctions and losses with the stack gases to a minimum.

A more detailed analysis of the operation (of this and others described above) of the systems with permeable thermal elements and also of the methods for their calculation will be presented below.

The generator circuits with the nuclear-reactor and radioisotpic heating of hot junctions are shown in Fig. 9.

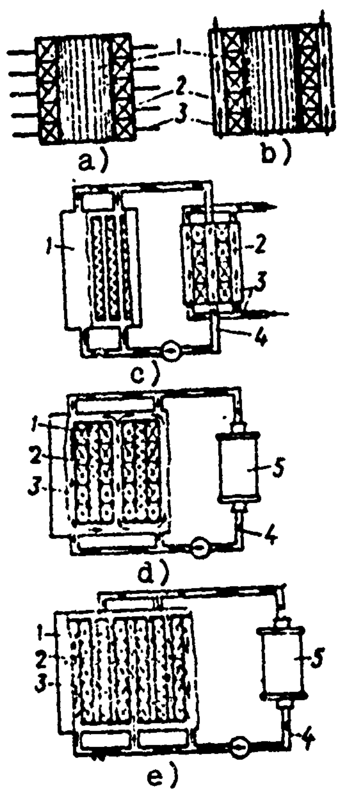


Figure 9. Generator circuits with the nuclear-reactor and radioisotopic heating: 1 - heating surface, 2 - thermal elements, 3 - cooling system for cold junctions, 4 - coolant circuit, 5 - heat exchanger.

Systems shown on Figs. 9a and b differ from the generator circuits heated by the combustion products, examined above, (see Fig. 5a and b) only by the way the heat is supplied to the hot junctions. In this case, hot junctions are heated by their direct thermal contact with the exothermal surface. In these systems the outer shell of a nuclear reactor or a capsule with radioisotope fuel can be examined as the exothermal surface. The systems which are intended to make use of the heat which is released during the fission reaction of radioactive substances for heating the hot junctions of thermal elements have the same deficiencies as the analogous systems in which the hot junctions are heated by the products combustions, which are described above.

The system shown in Fig. 5b is most advisable when using a device under the conditions where cold water is available to create a large temperature differential on the junctions (for an intense heat removal from the cold junctions of thermal elements). Such a system has been realized in a relatively large number of generators designed for marine purposes. The marine power system designed on the basis of the nuclear reactor SNAP-10 intended for space use can serve as an example. This system provides for the obtaining of electric power of approximately 350 W from 100 thermal elements.

In this device the thermal elements were placed between the outer surface of the reactor and the external housing of the unit. Heat discharge from the cold junctions of batteries was realized by means of thermal conductivity through the walls of the external housing into the marine medium [33]. At the same time, the power system for space use designed on the basis of the same reactor (Fig. 9a), in which the heat is removed by radiation from the ribbed cold junctions, produces a useful electric power of only 250 W.

The system in Fig. 9c is also used rather extensively in the development and manufacture of thermoelectric devices designed for both the space and marine use. The thermoelectric system SNAP-10A (Fig. 9c) has been used in the near-earth orbit at an altitude of 1300 km for 43 days in 1965 which, in the beginning (before the accident), was generating a net power of more than 500 W. The diagram of the operating cycle of this unit is shown in Fig. 10. The heat from the reactor core to the hot junctions of thermal elements was transferred by the liquid-metal heat-transfer agent NaK-78. This generator is made of 2880 thermal elements which consist of germanium-silicon alloys. The heat from cold junctions in this circuit is removed also by the intermediate heat-transfer agent which is cooled in the heat exchange-emitter. The surface of the heat exchanger-emitter was 5.82 m^2 at a mean temperature of 597°K [31, 34]. Based on this nuclear system thermionic converters for marine use were developed with useful electric power of 500-2000 W. Sodium-potassium eutectic was used both for supplying the heat from the reactor to the hot junctions of the generator and for removing the heat from the cold junctions to the heat exchanger cooled by the sea water [33]. This system passed successfully the prolonged tests.

A significant advantage of the system shown on Fig. 9c is that the intermediate heat-transfer agent serves as an additional radiation shielding for the semiconductor thermoelectric materials. This makes it possible to somewhat expand the number of materials

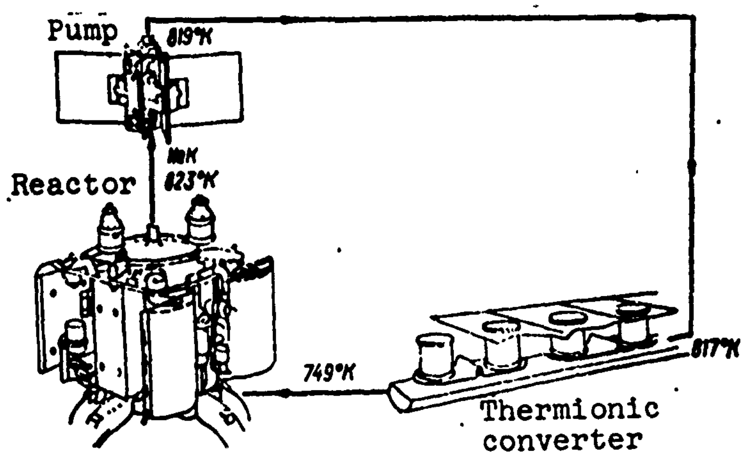


Figure 10. Diagram of the operating cycle of the SNAP-10A unit.

which are capable of working with a nuclear heat source. However, this system complicates the use of high-temperature materials, since at present there is still no heat-transfer agent of sufficiently high temperature that would satisfy the specific operating conditions of nuclear-reactor units.

Based on this system (see Fig. 9c) a deep-water device of the system "Neptune" was designed with an electric output of 15 kW, shown in Fig. 11. The best technological developments achieved for the main units in the systems of space application SNAP-10A, SNAP-8 and SNAP-2 [33] are combined in this device. The heat output of the nuclear system is 400 kW with the temperature in the active zone of 938°K. The nuclear reactor above which are the motors driving the control rods, which are separated from its housing by special shielding, is placed in the lower part of the device. A generator consisting of 12,000 lead-telluride elements is in the central section of the power device. Heat to the hot junctions of thermal elements is supplied by liquid NaK-78 which is heated in the reactor core from 700 to 938°K, while the heat from the cold junctions to the radiator is removed by boiling water during its natural circulation. From the surface of the radiator the heat is removed by the surrounding sea water.

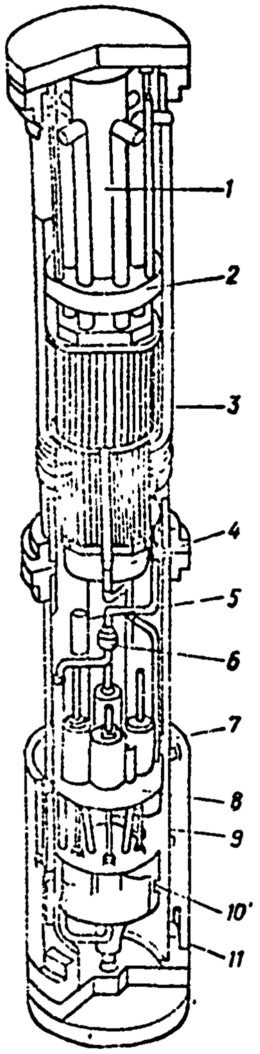


Figure 11. Thermoelectric device of the "Neptune" system: 1 - instrument compartment, 2 - heat shield, 3 - thermoelectric generator, 4 - water shielding, 5 - suspended expansion bend, 6 - pump, 7 - control rod drive, 8 - protection from γ-radiation, 9 - load-bearing shell, 10 - reactor, 11 - biological protection.

More compact generators can be built with nuclear-reactor heating by placing the thermal elements inside the active zone directly on the fuel elements.

The systems of such converters, shown in Fig. 9d, unlike those in Fig. 9c, make it possible to avoid the needless heat dissipation by the heat-transfer agent in the lines which couple the nuclear reactor with the generator. Furthermore, the indicated systems make it possible to utilize the high-temperature thermoelectric materials. However, in designing thermoelectric devices according to these systems, special attention should be give to the radiation stability of thermoelectric materials, since under these conditions they will operate without radiation shielding.

The system in Fig. 9e differs from that in Fig. 9d by the way the temperature differential is created on the junctions of the generator and the way heat is supplied to the hot junctions. In the schematic in Fig. 9d the heat-transfer agent removes the heat from the surface of the cold junctions and leaves the bounds of the reactor core at a temperature considerably lower than that of cold junctions. Heat to the hot junctions of thermal elements is supplied by thermal conductivity from the jacket of the reactor fuel elements.

In the system shown in Fig. 9e provision is made for the use of permeable thermal elements in the generator structure the temperature differential on the junctions of which is created by blowing of the heat-transfer agent in the direction from the cold junctions to the hot. By absorbing heat within the thermal elements the liquid or gaseous coolant exits into the clearance between the hot junctions and the fuel element jacket, at a temperature somewhat lower (or equal to) than that of hot junctions. In the clearance the heat-transfer agent is heated by coming in contact with the fuel element jacket which has higher temperature and leaves the bounds of the reactor at a temperature equal or higher than that of hot junctions. The heat to the hot junctions, in this case, is supplied from the fule element jacket by convection and by thermal conductivity throught the liquid-metal heat-transfer agent or by convection and emission when gaseous coolant is used.

From the comparison of these two systems it is evident that in the system on 9e, with other conditions being equal, the heat consumption from the hot source will be greater than in the system in Fig. 9d, due to the fact that the heat-transfer agent is heated to a higher temperature. However, this still does not mean that the efficiency of the device constructed according to the system shown in Fig. 9e will be necessarily lower than that in the device corresponding to the diagram shown on Fig. 9d. This should be examined in connection with the conditions of heat discharge from the surface of the heat exchanger-coolant into the ambient medium, which is transferred there with the heated heat-transfer agent and must be removed before it is fed again into the nuclear reactor.

When the heat from the radiator surface is removed by natural or forced convection, then the system in Fig. 9e has no advantages over that shown in Fig. 9d, since with an increase in the temperature head between the radiator surface and the cooling medium the amount of heat which must be removed is also increased. The linear dependence of the amount of heat removed in the exchanger on the difference in temperatures between the exothermal surface and the cooling medium (in accordance with Newton's law) $Q=\alpha F(T-t)$ makes it impossible to decrease the surface of the heat exchanger or increase the cooling depth of the heat-transfer agent.

With the removal of heat by radiation the amount of heat removed from the radiator surface depends on the temperature of its surface in the fourth degree (see (9)). Under these conditions the system shown in Fig. 9e has considerable advantages. Supplying the heat-transfer agent to the heat exchanger-radiator at a temperature which exceeds that of hot junctions of the generator makes it possible to reduce its surface or, with the same surface, to cool the heat-transfer agent to a lower temperature, which in turn, makes it possible to create a greater temperature differential on the generator junctions.

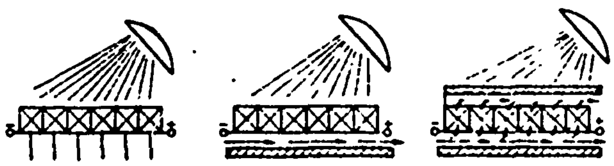
The study on the parameters of coolers-radiators, g... work [24], showed that for heat-exchange fluids the dependence of the surface of the cooler-radiator on the amount of removed heat and the temperatures of the heat-transfer agent at the entrance to the heat exchanger $T_{вх}$ and at the exit from it $T_{вых}$ has the form

$$F=\frac{Q}{\varepsilon\sigma}\cdot\frac{1}{3T_{вых}^{4}}\cdot\frac{\left(\frac{T_{вх}}{T_{вых}}\right)^{3}-1}{1-\frac{T_{вых}}{T_{вх}}}.$$

Consequently, the system shown in Fig. 9e permits one to improve the weight-size characteristics of power unit with the same electric output by decreasing the size of the cooler-radiator by supplying into it a heat-transfer agent at a higher temperature than that in the system shown in Fig. 9d, and, vice versa, with the identical surfaces of the heat exchanger-radiator, in the device constructed according to the scheme shown in Fig. 9e it is possible to obtain a higher useful electric output (greater temperature differential on the junctions) than in that shown in Fig. 9d, because of a deeper cooling of the heat-transfer agent.

The comparison of energy effeciency of these thermal circuits requires more detailed and specific calculations.

Generator circuits in which hot junctions are heated by the radiant solar energy, shown in Fig. 12, are different in the way the temperature differential is created on thermojunctions. In all these systems the heat to the hot junctions of thermal elements is supplied in the form of radiant energy by solar radiation. Thus, in devices designed for ground use, concentrators of solar radiation are usually used with such generators due to insufficient densities of thermal solar radiation per unit of earth surface.



a) b) c)

Figure 12. Generator circuits with solar heating.

The heat removed from the surface of cold junctions of the generators constructed according to the diagrams shown in Figs. 12a and b is lost completely. It cannot be regenerated due to the specificity of the thermal energy soruce. The use of a heated coolant in any other heat-utilizing devices is also inexpedient since it has low temperature - lower than the temperature of cold junctions of a generator, which, of course, has a negative effect on the efficiency of helium devices. However, for these devices which have a "gratuitous" source of thermal energy, efficiency does not play such an important role as in generators with other types of heat sources for heating of hot junctions.

The most important question which determines the desirability of obtaining the electrical energy with the aid of solar generators is their cost. The development of the technological processes for the manufacture of solar thermoelectric devices and the decrease in cost of the concentrators of radiant solar energy and the semiconductor batteries, in many respects connected with this, will contribute ever more to their wide application in the areas with a large number of sunny days.

A large number of generators with helium heating have been constructed and tested under varied conditions, which were designed according to the diagrams in Figs. 13a and b and calculated for producing an useful electric output of up to several tens and even hundreds of watts [3].

Solar generators which use concentrators of radiant solar energy have a characteristic nonuniform heat-flow distribution (and also temperatures) on the surface of hot junctions of semiconductor thermocouples. In this case the thermal conditions of operation of thermal elements in a battery can be varied, which has a negative effect on its total efficiency. The nonuniformity of heating of the hot junctions in many structures reaches considerably high values - hundreds of degrees. However, as shown in work [15], for some thermoelectric materials the nonuniformity of heating can not only not lower the operational efficiency of a generator, but even increase it somewhat due to a more efficient operation of certain thermal elements at temperatures different from the calculated mean. This situation is determined completely by the temperature dependence of the figure of merit z of the thermoelectric materials being used. If the value z increases with an increase in the temperature of hot junctions, then the efficiency of conversion also increases in comparison with the operating efficiency of a device in the case of an uniform heating due to the fact that the temperature of hot junctions of a certain number of thermal elements exceeds the mean calculated temperature for a generator. When using materials which have an inverse dependence of the figure of merit on temperature, the nonuniformity of the heat flux on the part of hot junctions causes a decrease in the efficiency of a generator.

The tendency to provide identical temperature conditions for thermal elements in batteries led to the creation of solar generators with thermal elements of unequal height and generators in which each thermal element is heated by its own concentrator.

The diagram shown in Fig. 12c, when realized, unlike diagrams 12a and b, will permit one to obtain a coolant with a considerably higher temperature, which will ensure the possibility of its further effective utilization in other devices or in other stages of the generator. As was noted above, in structures with permeable thermal elements the coolant emerges onto the surface of hot junctions at a temperature close to that of junctions. Thus, the structure of a generator constructed according to this diagram will make it possible by means of solar heat to simultaneously obtain the useful electrical energy and a hot heat-transfer agent for its further utilization.

2. COOLER, CONDITIONER AND HEAT PUMP CIRCUITS

In the thermoelectric heating and cooling devices the difference in temperatures on the hot and cold junctions of thermal elements is created because of the Peltier effects, which occur under the effect of the flowing electric current. In these devices the direct-current power supply is the thermal circuit unit which is similar to the thermal energy source in generators.

If the thermal energy sources determine the diversity of thermal generator circuits, then it is expedient to classify the thermoelectric heating and cooling circuits only from the standpoint of diversity in the heat supply and removal systems.

Specific requirements are imposed on the dc power supplies (see Chapter I) in order to ensure a normal operation of a device. In the circuit analysis of coolants, heaters and conditioners the investigation of the effect of the type and properties of the power supply on other circuit elements is of no interest if it satifies all the necessary requirements. Consequently, the circuits of such devices should be distinguished, in the first place, by the way the heat is supplied and, removed from junctions -

radiation, convection (natural and forced) and heat exchange within permeable thermal elements.

The possible circuits for coolants, heaters and conditioners are shown in Fig. 13.

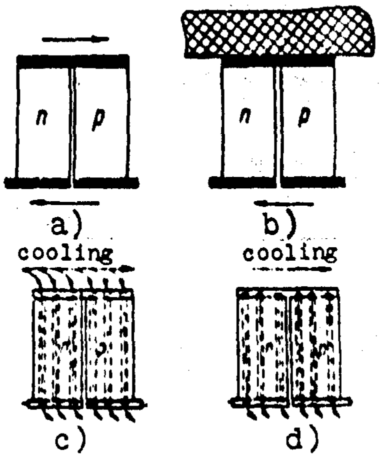


Figure 13. Cooler, heater and conditioner circuits.

The heat supply and removal systems by radiation are not used in practice due to the low operating temperatures of cold junctions or hot junctions of these devices. The practical implementation of heat removal from hot junctions by radiation can be realized in thermoelectric coolers designed for space application.

In the existing structures of conditioners and coolers, most common is the removal of heat from the ribbed hot junctions and supply of heat to the ribbed cold junctions by convection from a gaseous or liquid flow. In view of this fact the batteries are classified according to the nature of heat removal to the batteries of a type air - air, liquid - liquid and combined, liquid-gas. The batteries of all these types operate according to the scheme, shown in Fig. 13a. To intensify the heat-transfer process on the hot and cold junctions of such devices the ribbing of various forms is used.

In certain cases, in the cooling devices the heat removal from hot junctions or heat supply to cold junctions is realized not by means of convection on the ribbed surfaces, but by contact heat exchange with the cooled or heat-absorbing surface. Such coolers are subdivided into the types, mass - mass and mass - air.

The thermal element circuit in which the heat exchange is realized on one junction by convection with the ambient medium while on another there is contact heat exchange with the housing mass is shown in Fig. 13b.

The schematic diagram of a cooler shown in Fig. 14 corresponds to the schematic shown in Fig. 13a [21]. In such a cooler, intended for use in an automobile, the heat to the cold junctions is supplied by natural convection of air contained in the cooled space. The heat is removed by the air flow which washes the hot junctions of thermal elements. This design ensures the maintenance of a temperature differential between the ambient medium and within the cooled space up to 298°K.

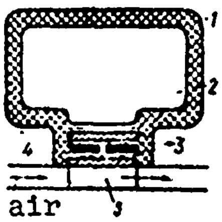


Figure 14. Cooler circuit for an automobile: 1 - thermal insulation, 2 - cooling chamber, 3 - contact lubrication, 4 - heat-transfer plate, 5 - cooling air.

The thermopile module, developed at the SKB of the Institute of Semiconductors of the Academy of Sciences of the USSR, of the thermoelectric chamber PTK-1 designed for testing the various equipment is schematically shown in Fig. 15 [25]. This module consists of two stages of thermal elements and is capable of developing a refrigerating capacity of 11 W using the electric power of 453 W. Heat to the cold junctions of thermal elements is supplied from the cooled space through the ribbed surface, from hot junctions the heat is removed to a massive plate by contact-transfer of heat. To increase the heat removal, holes are drilled in the plate through which the cooling water circulates. Thus, in this case, a combined heat transfer is realized in accordance with the circuits shown on Fig. 13a and b.

Figure 13c shows circuits of permeable thermal elements in which the cooled substance is blown through the capillaries (pores) in the direction from hot junctions to the cold, transferring the

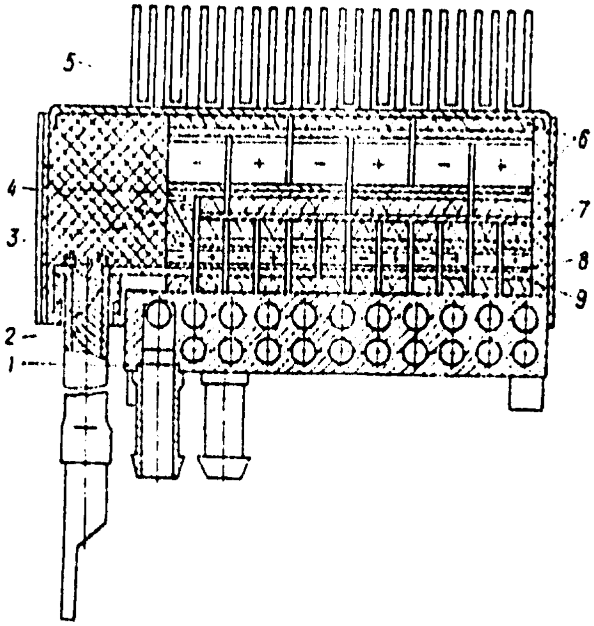


Figure 15. Diagram of a battery of the PTK-1 chamber: 1 - copper plate, 2 - current input to the first stage, 3 - copper sheet, 4 - elastic copper angles, 5 - radiator, 6 - heat transfers, 7 - branches of thermal elements, 8 - lead damping plate, 9 - copper commutation plate.

heat to the material of the branches of thermal elements along their entire height.

In contrast to the monolithic thermal elements, in the permeable thermal elements the heat from the cooled substance is removed not on the surface of cold junctions, but in the space of the thermal element. Heat from the surface of hot junctions can be removed in two ways. When the cooled substance circulates in a closed loop, the device based on the system, shown in Fig. 13c is advisable. In this case the cooled substance is supplied to the thermal elements through the channels made in the commutation plates of hot junctions, and some other substance serves as the heat-transfer agent to remove the heat from the hot junctions.

Operation of permeable thermal element is possible also in an open circuit shown in Fig. 13d, when heat from hot junctions is removed by the flow of a substance, a certain portion of which is cooled by being sucked through the permeable thermal elements.

The permeable thermal elements which cool the flows of a substance will be examined more thoroughly in Chapter V.

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CHAPTER III

PECULIARITIES OF TEMPERATURE FIELDS IN PERMEABLE THERMAL ELEMENTS

Let us consider in detail the fundamental differences when the heat is supplied to the junctions and when it is removed in monolithic and permeable thermal elements.

When using permeable thermal elements the bulk of heat is supplied or removed to the material of the branches of thermal elements not through the surfaces of hot or cold junctions, but within the branches of thermal elements. As was already mentioned, the branches of thermal elements are made porous or perforated, which gives them a very developed internal surface. With the blowing-in of a heat-transfer agent or a coolant through such permeable thermal elements in a direction from certain junctions to others, the process of heat exchange between the blown substance and the material of branches proceeds very intensely. The enormous intensity of heat exchange under such conditions is achieved because of the extremely developed heat-exchange surface.

For the finely porous pressed materials the volumetric heat-transfer coefficients from the wall material to the gas blown through capillaries reach the order of 10^5 W/m^3 × deg. In perforated walls with capillaries of up to 1 mm in diameter the coefficients heat transfer to the blown gases can be 100 $W/m^2 \cdot deg$ and higher. In this case the large heat-exchange area makes it possible to transfer very large amounts of heat from the exothermal surface to the heat-absorbing medium, which indeed determines the small difference in temperatures between the heat-absorbing and exothermal media, at which the heat exchange within the permeable thermal elements occurs.

During heat exchange the difference in temperatures between the exothermal surface and the heat-absorbing medium determines the thermodynamic loss of the process. The smaller the difference the less is the loss due to the irreversibility of the heat-exchange process. Consequently, in the permeable thermal elements where the bulk of heat is supplied (removed) within the thermal elements with a small difference in temperatures, this loss due to the irreversibility of heat exchange can be minimum.

When supplying (removing) heat in monolithic thermal elements through the junctions surfaces, where the intensity of heat exchange is not so considerable and the transfer surface is immeasurably smaller, the loss due to the irreversibility of heat exchange is high due to large temperature heads between the heat transferring media. The ribbing of junctions permits one to reduce this loss considerably. However, this is achieved with considerable complication and an increase in cost of the structure.

The presence of internal heat exchange in permeable thermal elements imparts into their operation fundamental and significant features. Specifically, a new possibility presents itself to affect the energy characteristics of thermoelectric devices - net power (temperature differential on junctions) and the efficiency of the generator, or the refrigerating capacity and the cooling coefficient of coolers and conditioners.

As is known [28], temperature profile in the material of monolithic thermal element is described by the following expression:

$$T = T_1 + \frac{T_2 - T_1}{\delta} y + \frac{Q_v \delta}{2\lambda} y - \frac{Q_v}{2\lambda} y^2. \tag{10}$$

where Q_v - internal heat release in the material of branches (Joule heat), δ - height of thermal element, λ - thermal conductivity of the material of branches (average for the p - n-junction); T_2 and T_1 - temperatures of the hot and cold junctions respectively.

From this expression it is evident that the temperature profile bears a nonlinear nature. The nonlinearity of the temperature profile along the height of thermal element is determined by the value of the internal heat release. The Joule heat released within the branches as the electric current flows through them increases the return flow of heat to the cold junctions and the temperature gradient as it moves from the hot junctions to the cold. In this case, there is less heat entering the thermal element from the hot side than leaving it from the cold side. The difference between the heats is equal to the Joule heat which is released in the material of branches as the electric current flows.

From expression (10) it is evident that the form of the temperature profile in a monolithic thermal element depends on the temperature differential on surfaces, the height of the thermal element, thermal conductivity of the material from which it is made, and on the value of Joule's heat release. However, in the case of a permeable thermal element the form of the temperature profile in the material of branches is affected by rather a larger number of values. First of all it is affected by the direction of blowing: towards the heat flux (from the cold junctions to the hot) or in the direction which coincides with the direction of the heat flux (from hot junctions to the cold). Furthermore, the form of the profile is changed considerably by the structure of the permeable material (finely porous or perforated) and the heat-transfer coefficients within the branches.

In obtaining expressions to describe temperature profiles in the permeable thermal elements, we will examine the thermal element circuit shown in Fig. 16.

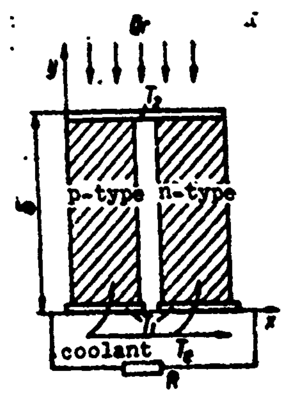

Figure 16. A diagram of permeable thermal element.

1. METHODS FOR THE CALCULATION OF TEMPERATURE FIELDS

First let us examine a case where a substance (gas, liquid) is blown through the thermal elements in the direction from the cold junctions to the hot, i.e., in the direction opposite to the heat flux in the material of branches. We will use the following principal designations: T_2 and t_2 are respective temperatures of the material and the blown substance on the hot side when $y=\delta$; T_1 and t_1 are respective temperatures of the material and the blown substance on the cold side when $y=0$. The specific mass velocity of the blown substance ρv_w pertains to the entire surface of the finely porous thermal element and to the cross-section area of the capillaries in the perforated thermal elements.

When studying the temperature fields in permeable thermal elements it is necessary to consider not only the processes of thermal conductivity but also the processes of convective heat exchange of the mass of the material of branches and the substance blown through the surfaces of pores or capillaries. This means that the temperature profiles in the material of branches and in the blown substance will be strictly connected with one another within the limits of height of the thermal element.

Let us examine a one-dimensional problem for obtaining temperature profiles in the material of branches and in the blown substance (see Fig. 16).

With blowing through a permeable wall the coolant is heated and removes the heat from the material of the wall within it. The coolant is heated due to the rise in the amount of heat passing through the element dy of the wall by means of thermal conductivity $\lambda(F-F_{nop})\frac{dT}{dy}-\left[-\lambda(F-F_{nop})\left(\frac{dT}{dy}+\frac{d^2T}{dy^2}dy\right)\right]$ and also due to the release of internal heat in this elementary volume (with the passage of electric current - Joule heat), i.e.

$$\rho v_w c_p F_{nop} dt = \lambda(F-F_{nop})\frac{d^2T}{dy^2}dy + q_v(F-F_{nop})dy, \tag{11}$$

where F_{nop} - cross-section area of capillaries.

Hence

$$\frac{dt}{dy} = \frac{\lambda}{\rho v_w c_p \Pi}\cdot\frac{d^2T}{dy^2} + \frac{q_v}{\rho v_w c_p \Pi}. \tag{12}$$

or

$$\rho v_w c_p F_{nop} dt = \alpha\pi d z_1 (T-t)dy,$$

where

$$T = \frac{\rho v_w c_p d}{4\alpha}\cdot\frac{dt}{dy} + t. \tag{13}$$

From equations (12) and (13) it is easy to obtain a differential equation for the change in the coolant temperature along the height of the thermal element:

$$\frac{d^3t}{dy^3} + \frac{4\alpha}{\rho w_w c_p d} \cdot \frac{d^2t}{dy^2} - \frac{4\alpha \Pi}{\lambda d} \cdot \frac{dt}{dy} + \frac{q_v}{\lambda} \cdot \frac{4\alpha}{\rho w_w c_p d} = 0. \qquad (14)$$

The general solution of these equations has the form

$$t = C_1 + C_2 \exp\left(B - \frac{A}{2}\right)y + C_3 \exp\left[-\left(B + \frac{A}{2}\right)y\right] + ACy, \qquad (15)$$

where C_1, C_2 and C_3 - integration constant, α - heat-transfer coefficient within thermal elements (in capillaries), α_v - volumetric heat-transfer coefficient within the finely porous branches of thermal elements, d - diameter of capillaries, $\Pi = \frac{F_{nop}}{F - F_{nop}}$ - ratio of the area of the transfer cross section of capillaries to the area of a monolithic material of the thermal element, q_v - internal heat release referred to the unit volume of material (in the case of Joule heat release $q_v = \frac{z\lambda\Delta T^2}{\delta^2 (m+1)^2}$). Constant A, B and C have a somewhat different form depending on the type of the material of the thermal elements: porous or perforated.

From the equations given above it is easy to find the expressions for the distribution of temperatures of the material and the blown substance within the limits of height of the thermal element, after determining the integration constants from the conditions of invariability of temperatures of the material and the blown substance on its surfaces (when y=δ and y=0). In this case we will obtain the following expressions:

$$\begin{aligned} T = T_1 + ACy + (T_1 - t_1 - C)\frac{2B + A}{2B - A} \times \\ \times \left\{\exp\left[\left(B - \frac{A}{2}\right)y\right] - 1\right\} + \\ + \left(\left(\frac{B}{A} + \frac{1}{2}\right)^2 \left\{\exp\left[\left(B - \frac{A}{2}\right)y\right] - 1\right\} + \\ + \left(\frac{B}{A} - \frac{1}{2}\right)^2 \left\{1 - \exp\left[-\left(B + \frac{A}{2}\right)y\right]\right\}\right)\frac{2AC_3}{2B - A}, \end{aligned} \qquad (16)$$

$$t = T_1 - C + ACy + (T_1 - t_1 - C)\frac{2A}{2B-A}\Big\{\exp\Big[\Big(B - \frac{A}{2}\Big)y\Big] - \Big(\frac{B}{A} + \frac{1}{2}\Big)\Big\} + \Big(\Big(\frac{B}{A} + \frac{1}{2}\Big)\exp\Big[\Big(B - \frac{A}{2}\Big)y\Big] + \Big(\frac{B}{A} - \frac{1}{2}\Big)\exp\Big[-\Big(B + \frac{A}{2}\Big)y\Big] - 2\frac{B}{A}\Big)\frac{2AC_2}{2B-A}, \quad (17)$$

where

$$C_2 = \frac{(T_2 - T_1 - AC\delta)\left(\frac{B}{A} - \frac{1}{2}\right) + (T_1 - t_1 - C)\left(\frac{B}{A} + \frac{1}{2}\right)\left\{1 - \exp\left(B - \frac{A}{2}\right)\delta\right\}}{\left(\frac{B}{A} + \frac{1}{2}\right)^2 \exp\left[\left(B - \frac{A}{2}\right)\delta\right] - \left(\frac{B}{A} - \frac{1}{2}\right)^2 \exp\left[-\left(B + \frac{A}{2}\right)\delta\right] - 2\frac{B}{A}}. \quad (18)$$

The heat-transfer coefficient, which characterizes heat exchange per unit volume of the material of branches of the thermal element (α_V), is usually used in practice for the finely porous thermal elements when it is not possible to determine the heat-exchange surface. In the case of the perforated thermal elements, where the surface of internal heat exchange can be determined relatively simply, the heat-transfer coefficient, which characterizes the transfer of heat through the unit surface of capillaries is used. When calculating temperature fields by this dependences, it is possible to use these heat-transfer coefficients, which is taken into account by the varied form of the A, B and C constants.

The differential equations which describe the temperature profiles in the permeable thermal elements, blown in the direction which coincides with the direction of the heat flux (from the hot junctions to cold), and which were obtained by analogy with the expressions given above for the opposite direction of blowing [10] have the form

$$
\begin{gathered}
T = t - \frac{\rho v_{\Sigma} c_p F_{\text{пор}}}{\alpha \pi d z_1} \cdot \frac{dt}{dy}, \\
\frac{d^3 t}{dy^3} - A\frac{d^2 t}{dy^2} - \frac{\alpha \pi d z_1}{\lambda (F - F_{\text{пор}})} \cdot \frac{dt}{dy} - \frac{q_v A}{\lambda} = 0.
\end{gathered} \tag{19}
$$

The general solution of this system can be presented in this form:

$$
\begin{gathered}
T = C_1 - \left(\frac{B}{A} - \frac{1}{2}\right) C_2 \exp\left[\left(B + \frac{A}{2}\right) y\right] + \\
+ \left(\frac{B}{A} + \frac{1}{2}\right) C_3 \exp\left[-\left(B - \frac{A}{2}\right) y\right] - ACy + C, \\
t = C_1 + C_2 \exp\left[\left(B + \frac{A}{2}\right) y\right] + \\
+ C_3 \exp\left[-\left(B - \frac{A}{2}\right) y\right] - ACy.
\end{gathered} \tag{20}
$$

Integration constants C_1, C_2 and C_3 are determined with the boundary conditions of temperature constancy of the blown substance and the material of branches of the thermal elements on the surfaces of junctions (when y=δ and y=0). In this case the expressions which describe the temperature profiles with the given direction of blowing take the form

$$
\begin{gathered}
T = T_1 - ACy + \frac{2B - A}{2B + A}(T_2 - t_2 - C)\frac{\exp\left[\left(B + \frac{A}{2}\right) y\right] - 1}{\exp\left(B + \frac{A}{2}\right)\delta} + \\
+ \left(\left(\frac{B}{A} - \frac{1}{2}\right)^2 \exp\left[-\left(B - \frac{A}{2}\right)\delta\right] \times \right. \\
\times \left\{1 - \exp\left[\left(B + \frac{A}{2}\right) y\right]\right\} - \left(\frac{B}{A} + \frac{1}{2}\right)^2 \exp\left[\left(B + \frac{A}{2}\right)\delta\right] \times \\
\left. \times \left\{1 - \exp\left[-\left(B - \frac{A}{2}\right) y\right]\right\}\right) \frac{2AC_3}{(2B + A)\exp\left(B + \frac{A}{2}\right)\delta},
\end{gathered} \tag{21}
$$

$$
\begin{gathered}
t = T_1 - C - ACy - (T_2 - t_2 - C)\frac{\left(\frac{B}{A} - \frac{1}{2}\right) + \exp\left(B + \frac{A}{2}\right) y}{\left(\frac{B}{A} + \frac{1}{2}\right)\exp\left(B + \frac{A}{2}\right)\delta} + \\
+ \left(\left(\frac{B}{A} - \frac{1}{2}\right)\exp\left[-\left(B - \frac{A}{2}\right)\delta\right]\left\{\left(\frac{B}{A} - \frac{1}{2}\right) + \right.\right. \\
\left. + \exp\left(B + \frac{A}{2}\right) y\right\} + \left(\frac{B}{A} + \frac{1}{2}\right)\exp\left(B + \frac{A}{2}\right)\delta \times \\
\left. \times \left\{\exp\left[-\left(B - \frac{A}{2}\right) y\right] - \left(\frac{B}{A} + \frac{1}{2}\right)\right\}\right) \frac{2AC_3}{(2B + A)\exp\left(B + \frac{A}{2}\right)\delta}.
\end{gathered} \tag{22}
$$

where

$$C_2 = \frac{\left(\frac{B}{A}+\frac{1}{2}\right)(t_2 - T_1 + C + AC\delta)\exp\left[\left(B+\frac{A}{2}\right)\delta\right] + (T_2 - t_2 - C)\left[\frac{B}{A}-\frac{1}{2}+\exp\left(B+\frac{A}{2}\right)\delta\right]}{\left(\frac{B}{A}-\frac{1}{2}\right)^2\exp\left[-\left(B-\frac{A}{2}\right)\delta\right] - \left(\frac{B}{A}+\frac{1}{2}\right)^2\exp\left[\left(B+\frac{A}{2}\right)\delta\right] + 2\frac{B}{A}\exp A\delta}. \tag{23}$$

The equations obtained for the description of temperature profiles in the material of branches of the thermal elements and in the heat-transfer agent (coolant) blown through them, both in the case of blowing from the cold junctions to the hot (equations (16) and (17)) and in the case of the opposite direction of blowing (equations (21) and (22)), have a rather cumbersome form. They can be simplified somewhat with the calculation of the devices operating in certain ranges of consumption of the blown substance, with certain geometric dimensions of thermal elements and in certain other cases when terms of a second order of smallnesses appear which can be disregarded without lowering the accuracy of the calculation. However, it is necessary to note that with the available diversity of the high-resolution computers, a numerical solution of these equations is no problem without any simplification (algebraic).

As was already mentioned, in the expressions given above for the distributions of temperatures in the material of branches and in the substance blown through them with both directions of blowing the values of the A, B and C coefficients have a varied form depending on whether the thermal element is finely porous or perforated.

These coefficients take the following form:

for the finely porous thermal elements

$$A = \frac{\alpha_v}{\rho v_w c_p},\ B = \sqrt{\left(\frac{A}{2}\right)^2 + \frac{\alpha_v}{\lambda}},\ C = \frac{z\lambda\Delta T^2}{\delta^2 (m+1)^2 \alpha_v};$$

for the perforated thermal elements

$$A = \frac{4\alpha}{\rho v_w c_p d},\ B = \sqrt{\left(\frac{A}{2}\right)^2 + \frac{4\alpha\Pi}{\lambda\alpha}},\ C = \frac{z\lambda\Delta T^2 d}{\delta^2 (m+1)^2 4\alpha\Pi}.$$

2. ANALYSIS OF TEMPERATURE FIELDS

Let us examine in specific examples the nature of change in temperature profiles in the blown thermal elements with different directions of blowing depending on the operating conditions. Let us assume we have a thermal element 1 cm in height which has 25 capillaries on each square centimeter of its surface with the capillaries having the diameter of 0.1 cm. Figures 17a and b shows temperature profiles in the material of branches (solid line) and in the substance blown through the capillaries (broken line) for the thermal element described above.

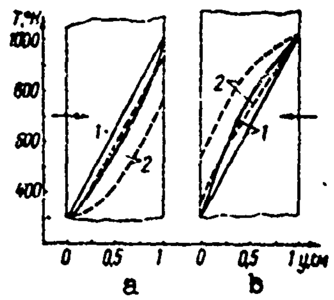

Figure 17. Temperature profiles in permeable thermal elements.

Figure 17a corresponds to the case when the direction of blowing is from the cold junctions to the hot, i.e. when the blown substance is heated in the capillaries. The temperature profiles shown in the figure were calculated by formulas (16) and (17). Figure 17b shows the plotted temperature profiles for the opposite direction of blowing of the substance, i.e. when the blown substance was cooled within the thermal element. These curves were calculated by formulas (21) and (22). In Fig. 17 curves 1 correspond to the low specific flow rate of the blown substance of 10^{-5} kg/cm^2·s and curves 2 - to the flow rate of 10^{-4} kg/cm^2•s. Temperatures on the hot and cold sides are taken to be 1000 and 300°K respectively, while temperatures of the surface through which the substance is injected and of the blown substances, for simplicity, are assumed to be equal to one other.

Figure 17 shows a considerable effect of the flow rate of the blown substance on the curvature of profiles. If at low mass velocity of blowing the temperature profiles in the material of branches of the thermal elements are close to linear (analogous to monolithic thermal elements), then with an increase in the flow rate of the blown substance they curve sharply. As was already mentioned, the curving of a profile causes temperature gradients on hot and cold surfaces, which means also the heat that enters the thermal element on the hot side and leaves it on the cold. The difference in these heats is explained by the heating or cooling of the blown substance within the capillaries.

As seen from Figs. 17a and b, with the equality of temperatures of the blown substance at the entrance to the thermal element and the corresponding surface of the thermal element as the blown substance moves through capillaries, this equality is not preserved and the difference between the temperatures of the substance and the material increases. The irreversibility of heat exchange occurs due to the small heat-transfer coefficient in the capillaries. The calculations by the expressions given above and also the experimental data [28] show that in the finely porous systems this irreversibility is considerably smaller and, under certain conditions, it vanishes. However, in capillary thermal elements the irreversibility of heat exchange is substantially less than in the monolithic elements even with an intense ribbing of junctions.

The diameter of the capillaries and their number per unit surface also have a considerable effect on the curvature of temperature profiles. With a relatively small number of capillaries the internal heat exchange will have no effect on temperature profiles in the thermal element material between holes. This situation was observed during the blowing of perforated plates with air [10] in a direction from the cold side to the hot.

Figure 18 shows the temperature profiles determined in the material of a perforated plate with 22 holes per 1 cm^2 with 0.5 mm in diameter at various mass flow rates of the blown air ρv_w. The broken lines show the results of the calculations made by theoretical dependence (13). It is possible to see that the agreement between the theoretical and experimental data is relatively good.

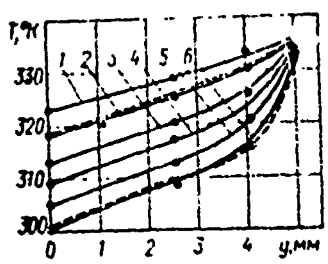

Figure 18

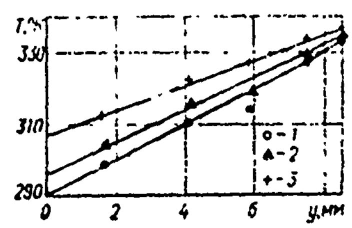

Figure 19

Figure 18. Theoretical and experimental temperature profiles in a perforated wall: 1 - ρv_w=1.52·10^{-5} kg/cm^2·s, 2 - ρv_w=3.16·10^{-5} kg/cm^2·s, 3 - ρv_w=0.181·10^{-3} kg/cm^2·s, 4 - ρv_w=0.375·10^{-3} kg/cm^2·s, 5 - ρv_w=0.574·10^{-3} kg/cm^2·s, 6 - ρv_w=0.685·10^{-3} kg/cm^2·s.

Figure 19. Temperature profiles in a perforated wall with a small number of holes: 1 - ρv_w=1.25·10^{-3} kg/cm^2·s, 2 - ρv_w=0.725·10^{-3} kg/cm^2·s, 3 - ρv_w=0.326·10^{-3} kg/cm^2·s.

Figure 19 shows the results of blowing of a perforated plate with 10 holes for every 1 cm^2 with 0.8 mm in diameter. As one would expect, with a small number of capillaries the [illegible] of temperature profiles in the wall material is not observed even at high mass velocities of the blown air. The temperature through the wall thickness, in this case, changes according to a linear law which is characteristic for monolithic walls.

Thus, the peculiarities of temperature fields in the permeable thermal elements make it possible, to a certain degree, to affect the heat fluxes on the thermal element surface, which means also its energy characteristics. The qualitative and quantitative analysis of this effect will be given below in the examination of specific systems of thermoelectric devices with permeable thermal elements.

The peculiarities of temperature fields in permeable thermal elements considered above can have a significant effect also on the method of averaging of the parameters of the materials of branches of the thermal elements operating in this temperature range [9].

In monolithic thermal elements where the temperature profile is close to linear, the mean temperature is near the center of the thermal element branch. In the case of blown thermal elements, a considerable portion of height of the thermal element can operate at a weakly changing temperature and the mean temperature will be not at the middle, but nearer to the hot or cold side depending on the direction of blowing.

Taking into account the fact that the properties of many thermoelectric materials change rather sharply with a change in temperatures, one should average the parameters of permeable thermal elements both with regard to temperature and height of the branches.

CHAPTER IV

CALCULATION OF THERMOELECTRIC GENERATORS WITH PERMEABLE THERMAL ELEMENTS

Generators with monolithic thermal elements, their calculation methods, the analysis of operation and the design features of devices designed for various purposes are discussed in considerable detail in the literature and, it seems inadvisable to present the known data at this time. At present, the information concerning the calculation methods, design features, etc. for the generators with permeable thermal elements is available only in periodicals. Therefore, let us consider in detail the composition of the calculation methods and analyze the operation of the generators with permeable thermal elements, and also examine their thermal circuits.

As was noted in the previous chapters, it is possible to examine two directions of blowing of the permeable thermal elements by liquid or gaseous substance - from cold junctions to hot and vice versa. The direction of blowing of a substance, in the direction of the heat flux or vice versa, substantially affects the temperature profile in the thermal element material (see Chapter III), which means also the heat fluxes on the surface of junctions, which, in turn, affects the generator operating conditions and its energy characteristics. Under these conditions, the calculation methods for such generators will also be completely different.

1. CALCULATION OF A GENERATOR BLOWN BY A COOLANT IN THE DIRECTION FROM THE COLD JUNCTIONS TO THE HOT

With the indicated direction of blowing of a substance the temperature differential on thermoelectric junctions with any possible heat source is created basically due to the heat extraction by the cooled substance within the permeable thermal elements and partially (in certain cases) on the surface of cold junctions.

Virtually any gas or liquid can be used as the coolant. When the generator operates with the heating of hot junctions by the heat of the combustion products of organic fuels to create a temperature differential on the junctions of permeable thermal elements one can use a fuel or an oxidizer, which enter into a reaction with one another as they enter onto the surface of the hot junctions and heating them.

Let us examine a generator with internal cooling of thermal elements and the heating of hot junctions by the combustion products of an organic fuel which is schematically shown in Fig. 20.

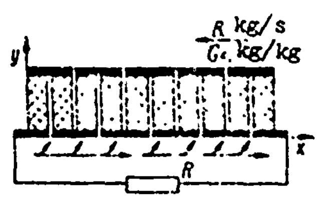

Figure 20. Generator circuit with internal cooling and heating of hot junctions by the products of combustion.

Liquid or gaseous fuel is supplied to the device with a deficiency or an excess of the oxidizer. Before the battery there is a mixture of the unburned fuel and the products of combustion in an operational mode with a deficiency in the oxidizer. Unburned fuel is burned up in the oxidizer which is blown through the permeable thermal elements in the direction from the cold junctions to the hot.

In such a system the cooling of cold junctions of the thermal elements can be realized in two ways depending on the required temperature differential on the junctions for obtaining the necessary net power.

First, a coolant can be supplied against the heating flow and thereby realize a joint cooling of cold junctions. In this case, part of the coolant flow is sucked through the permeable thermal elements and removes the heat within the capillaries. The other part of the coolant flow, in removing the heat from the surface of cold junctions, is subsequently utilized as the primary air for the oxidation of fuel thereby returning the heat removed from the cold junctions to the combustion zone.

Second, the temperature differential on thermojunctions can be created only due to the blowing of the coolant in the direction from the cold junctions to the hot. In this case, to realize the conditions where a battery can operate with a large temperature differential on the junctions, the coolant can be supplied in different sections along the length (it is sectionalized).

In both thermal circuits described there is virtually no heat loss from the side of the cold junctions of a generator. The entire heat returns with the coolant (in this case with the oxidizer) to the combustion zone.

In calculating such a generator it is assumed that its geometric dimensions, the temperature differential ΔT on the junctions, the type of fuel (calorific value and composition), the flow rate of primary air $G_B = \beta L_0$, kg/kg (β - deficiency or excess air coefficient), the initial consumption of the coolant ahead of the battery G_0, kg/s, the initial temperature of the coolant before the battery t_0^H, °K, the specific flow rate of the coolant through the permeable thermal elements ρv_w, kg/cm^2·s are known.

With the known type of fuel and the deficiency (excess) air coefficient the temperature of the heating mixture before the battery (when $x=l$) can be determined according to the procedure [22].

By using this methodology it is possible to obtain the initial temperature of the combustion products of any organic fuel of known elementary composition as a function of the deficiency or excess coefficient of oxidizer β at the entrance to the device. Consequently, with the aforementioned known initial data, when calculating a generator the initial temperature of the heating flow before the first thermal elements (when $x=l$) is easily determined.

The fundamental equations of heat balances of a generator with permeable thermal elements in accordance with Fig. 22 will have the following form.

The heat of the heating gas due to convective heat transfer is transferred to the hot junctions of thermal elements, which can be described by the following equation which takes into account the process of mixing of the heating flow and the coolant being blown-in:

$$B(G_{\text{в}}+1)c_p dt_3 + \rho v_w k_1 b(l-x)c_p dt_3 - \rho v_w k_1 b c_{\text{в}} dx(t_3 - t_2) + qbdx = \alpha_1 bdx(t_3 - T_2), \quad (24)$$

where t_3 - current temperature of the heating flow; b - perimeter of the generator (along the surface of hot junctions); k_1 - ratio of the cross-section area of the capillaries to the overall area of the thermal element.

The heat-transfer coefficient α_1 from the flow of the heating gas to the hot junctions of thermopiles with the presence of the blowing-in of the substance through the pores or capillaries can be computed, for example, by formula (7). An additional specific quantity of heat q (referred to the area of hot junctions of thermal elements) is released as a result of the afterburning of fuel in the oxidizer blown in through the permeable thermal elements when the generator operates under conditions of $\beta<1$, i.e., when the first thermal elements receive the heating gas which contains the products of combustion and the unburned fuel:

$$q = \frac{BQ_p^{н}(1-\beta)}{\sum b'}. \tag{25}$$

Depending on the deficiency coefficient in the primary air and the flow rate of the secondary air blown through thermal elements, supplementary heat release will occur at different distances from the battery (from the cross-section of fuel injection). This distance can be determined from the expression

$$l - x = \frac{1-\beta}{\rho v_{в} k_1 b} L_0 B. \tag{26}$$

The heat transmitted to the hot junctions of thermal elements by convection from the heating flow is expended on the heat removed from their surface as a result of thermal conductivity of the branch material and for the Peltier heat which is absorbed on the surface of hot junctions:

$$\alpha_1 b\,dx(t_{г} - T_{г}) = \lambda k b\,dx \left(\frac{dT}{dy}\right)_{y=\delta} + \frac{kz\lambda}{\delta(m+1)} \Delta T T_{г} b\,dx, \tag{27}$$

where

$$K = \frac{F - F_{пор}}{F}.$$

The difference of the heats, of that which entered onto the surface of hot junctions and of that which left the thermal elements on the surface cold junctions, is expended for the useful power developed by the generator and for the preheating of the coolant from the temperature on the boundary of cold junctions t_1 to the temperature on the boundary of hot junctions t_2:

$$\frac{z\lambda k}{\delta(m+1)}\Delta T T_2 b dx + \lambda k b dx\left(\frac{dT}{dy}\right)_{y=\delta} - \frac{z\lambda k}{\delta(m+1)}\Delta T T_1 b dx -$$
$$- \lambda k b dx\left(\frac{dT}{dy}\right)_{y=0} =$$
$$= \frac{z\lambda m k}{\delta(m+1)^2}\Delta T^2 b dx + \rho v_w k_1 c_p b dx (t_2 - t_1). \tag{28}$$

The Peltier's heat which is released on their [junctions] surface and the heat which arrived there as a result of thermal conductivity of the material of branches of the thermal elements, with the exception of a certain quantity of heat is expended from the side of cold junctions of the battery for heating of the coolant which washes their surface. The latter returns inside thermal elements and then to the combustion zone with the blown-in coolant, i.e.

$$G_0 c_p dt_0 - \rho v_w k_1 c_p b x dt_0 - \rho v_w k_1 c_p b dx (t_1 - t_0) =$$
$$= \alpha_0 b dx (T_1 - t_0) = \lambda k b dx\left(\frac{dT}{dy}\right)_{y=0} +$$
$$+ \frac{z\lambda k}{\delta(m+1)}\Delta T T_1 b dx - \rho v_w k_1 c_p b dx (t_1 - t_0). \tag{29}$$

After rewriting these balance equations in a more convenient form for the solution we will obtain a system of equations which make it possible to determine the temperature distributions of the junctions, coolant and heat-transfer agent along the length of the battery, and also all its other characteristics:

$$\frac{B(G_0+1)c_p}{\alpha_2 b}\cdot\frac{dt_2}{dx} + \frac{\rho v_w k_1 c_p}{\alpha_1}(l-x)\frac{dt_1}{dx} -$$
$$- \frac{\rho v_w c_p k_1}{\alpha_1}(t_2 - t_1) + \frac{q}{\alpha_1} = t_2 - T_2, \tag{30}$$

$$\frac{\alpha_1}{k\lambda}(t_2 - T_2) = \frac{z\Delta T}{\delta(m+1)}T_2 + \left(\frac{dT}{dy}\right)_{y=\delta}, \tag{31}$$

$$\frac{z\Delta T^2}{\delta(m+1)^2} + \left(\frac{dT}{dy}\right)_{y=\delta} - \left(\frac{dT}{dy}\right)_{y=0} = \frac{\rho v_w c_p \Pi}{\lambda}(t_2 - t_1), \tag{32}$$

$$\frac{G c_p}{\alpha_0 b} \cdot \frac{dt_0}{dx} - \frac{\rho v_w c_p k_1}{\alpha_0} x \frac{dt_0}{dx} - \frac{\rho v_w c_p k_1}{\alpha_0}(t_1 - t_0) = T_1 - t_0, \qquad (33)$$

$$\frac{\alpha_0}{\lambda k}(T_1 - t_0) = \left(\frac{dT}{dy}\right)_{y=0} + \frac{z\Delta T}{\delta(m+1)} T_1 - \frac{\rho v_w c_p \Pi}{\lambda}(t_1 - t_0). \qquad (34)$$

where z and λ - respectively, the figure of merit of the material of branches of the thermal elements and the coefficient of thermal conductivities of the material, which are assumed to be average for the p - n-pair. Furthermore, the value of z takes into account also the imperfection of commutation of the thermal elements. The heat-transfer coefficient α_0 from the cold junctions to the coolant flow characterizes the permanent loss of heat with a portion of the coolant which washes the surface of junctions. The value of this coefficient can be determined from criterial equation (8) obtained for the case where a substance is sucked through a finely porous surface.

In solving the system of equations (30)-(34) one more equation is necessary - for the temperature of the coolant at the exit from the capillaries (when y=δ) - obtained from the expression which describes the temperature distribution in the coolant flowing within the permeable thermal elements (14):

$$t_2 = T_1(1 + k_2) - t_1 k_2 - C(1 + k_2) + AC\delta + k_3(\Delta T - AC\delta), \qquad (35)$$

where

$$k_2 = \frac{\left(\frac{B}{A} + \frac{1}{2}\right)\exp\left(B - \frac{A}{2}\right)\delta + \left(\frac{B}{A} - \frac{1}{2}\right)\exp\left[-\left(B + \frac{A}{2}\right)\delta\right] - 2\frac{B}{A}}{\left(\frac{B}{A} + \frac{1}{2}\right)^2\exp\left(B - \frac{A}{2}\right)\delta - \left(\frac{B}{A} - \frac{1}{2}\right)^2\exp\left[-\left(B + \frac{A}{2}\right)\delta\right] - 2\frac{B}{A}},$$

$$k_3 = \frac{\left[1 - k_2\left(\frac{B}{A} + \frac{1}{2}\right)\right]\exp\left(B - \frac{A}{2}\right)\delta + \left(\frac{B}{A} + \frac{1}{2}\right)(k_2 - 1)}{\frac{B}{A} - \frac{1}{2}}.$$

Thus, by solving the system of equations (30) - (35) we obtain expressions for the temperature distributions along the length of batteries

for the cold junctions

$$T_1 = \frac{t_0\left(\frac{\alpha_0}{\lambda k} + \frac{\rho v_w c_p \Pi}{\lambda}\right) + \left(\frac{dT}{dy}\right)_{y=0} + \frac{\rho v_w c_p \Pi}{\lambda} C}{\frac{\alpha_0}{\lambda k} + \frac{\rho v_w c_p \Pi}{\lambda} - \frac{z\Delta T}{\delta(m+1)}}, \qquad (36)$$

for the coolant which washes the surface of cold junctions,

$$t_0 = \left[t_n^H + \frac{\left(\frac{dT}{dy}\right)_{y=0}\left(1 + \frac{\alpha_0}{\rho v_w c_p k_1}\right) - C\left[1 - \frac{z\Delta T}{\delta(m+1)}\right]}{\frac{z\Delta T}{\delta(m+1)}\left(1 + \frac{\alpha_0}{\rho v_w c_p k_1}\right)} - \frac{C\left(\frac{\rho v_w c_p \Pi}{\lambda} + \frac{\alpha_0}{\lambda k} - \frac{z\Delta T}{\delta(m+1)}\right)}{\frac{z\Delta T}{\delta(m+1)}\left(1 + \frac{\alpha_0}{\rho v_w c_p k_1}\right)}\right] \times$$

$$\times \left(1 - \frac{\rho v_w k_1 b}{G_0} x\right) - \frac{\left(\frac{dT}{dy}\right)_{y=0}\left(1 + \frac{\alpha_0}{\rho v_w c_p k_1}\right) - C_1\left[1 - \frac{z\Delta T}{\delta(m+1)}\right]}{\frac{z\Delta T}{\delta(m+1)}\left(1 + \frac{\alpha_0}{\rho v_w c_p k_1}\right)}, \qquad (37)$$

for the heating gas

$$t_3 = T_2\left[1 + \frac{z\Delta T}{\delta(m+1)} \cdot \frac{k\lambda}{\alpha_1}\right] + \frac{k\lambda}{\alpha_1}\left(\frac{dT}{dy}\right)_{y=\delta}, \qquad (38)$$

for the coolant at the entrance to the capillaries (when y=0)

$$t_1 = T_1 - C, \quad \text{and} \qquad (39)$$

for the coolant as it leaves the capillaries

$$t_2 = T_1 + k_7. \qquad (40)$$

When calculating by these equations one should remember that the instantaneous values of temperatures t_0, T_1 and t_2 participate in the dependences (37)-(40). When determining, for example, the temperature of the heating gas in any section it is necessary to substitute into expression (38) the values of temperatures t_2, t_0, T_2 that correspond to this section.

The order of calculation of a generator with permeable thermal elements is the following. First, from equation (37) we determine the change in the temperature of the coolant flow along the length. After this, from equation (36) we calculate the temperatures of cold junctions of the thermal elements in the corresponding sections. And then using equations (38)-(40) we determine changes in the coolant temperatures on the surfaces of hot and cold junctions, and also the temperature of the heating flow.

The following conventional designations are used in the dependences above

$$k_4 = \frac{\left(\frac{B}{A}+\frac{1}{2}\right)\exp\left(B-\frac{A}{2}\right)\delta + \left(\frac{B}{A}-\frac{1}{2}\right)\exp\left[-\left(B+\frac{A}{2}\right)\delta\right]}{\left(\frac{B}{A}+\frac{1}{2}\right)^2\exp\left(B-\frac{A}{2}\right)\delta - \left(\frac{B}{A}-\frac{1}{2}\right)^2\exp\left[-\left(B+\frac{A}{2}\right)\delta\right]-2\frac{B}{A}},$$

$$k_5 = \left(B+\frac{A}{2}\right)\left\{\exp\left(B-\frac{A}{2}\right)\delta - \left(\frac{B}{A}+\frac{1}{2}\right)\left[\exp\left(B-\frac{A}{2}\right)\delta-1\right]k_4\right\},$$

$$k_6 = \frac{2\frac{B}{A}\left(\frac{B}{A}+\frac{1}{2}\right)}{\left(\frac{B}{A}+\frac{1}{2}\right)^2\exp\left(B-\frac{A}{2}\right)\delta - \left(\frac{B}{A}-\frac{1}{2}\right)^2\exp\left[-\left(B+\frac{A}{2}\right)\delta\right]-2\frac{B}{A}},$$

$$k_7 = AC\delta - C + k_6(\Delta T - AC\delta).$$

The temperature gradients on the surfaces of hot and cold junctions of thermal elements taking into account the adopted designations will have the form

$$\left(\frac{dT}{dy}\right)_{y=\delta} = AC + \frac{\rho v_w c_p \Pi}{\lambda}(\Delta T - AC\delta) k_6, \qquad (41)$$

$$\left(\frac{dT}{dy}\right)_{y=0} = AC + 2\left(\frac{B}{A} - \frac{1}{2}\right)(\Delta T - AC\delta) k_6. \qquad (42)$$

From these expressions it follows that the temperature gradients on the hot and cold sides of the battery do not depend on the distance along battery and are constant along its entire length. The flow rate of the substance blown through them has the main effect on the value of the temperature gradients.

The expression for the distribution of temperature of the heating flow along the length of the battery can also be derived from equation (30). If expression (38) is valid for the case where a generator operates with the initial excess oxidant ratio (and also for determining the necessary initial and final temperatures of the heating flow), then from equation (30) it is possible to obtain a temperature profile of the heating gas, which takes into account the supplementary heat release during the afterburning of the unburned fuel in the operating mode when β<1. In this case we will obtain the following expression:

$$\begin{aligned} t_3 = &\left\{t_3^{H} - t_2 + \frac{\lambda}{\rho v_w c_p \Pi}\left[\frac{z\Delta T T_2}{\delta(m+1)} + \right.\right. \\ &\left.\left. + \left(\frac{dT}{dy}\right)_{y=\delta} - \frac{q}{k\lambda}\right]\right\}\left[1 + \frac{\rho v_w k_1 b(l-x)}{B(G_s+1)}\right]^{-1} + \\ &+ t_2 - \frac{\lambda}{\rho v_w c_p \Pi}\left[\frac{z\Delta T T_2}{\delta(m+1)} + \left(\frac{dT}{dy}\right)_{y=0} - \frac{q}{k\lambda}\right]. \end{aligned} \qquad (43)$$

Thus, using the equations given above we can determine the distributions of all temperatures (junctions, coolant and heat-transfer agent) along the length of the battery. The consumption of fuel, which ensures the obtained temperature distributions, can be easily determined with a joint solution of equations (38) and (43). Furthermore, the consumption of fuel can be determined from the equation of total heat balance of the generator being considered:

$$BQ_p^n = N + [B(1 + \beta L_0) + \rho v_w k_1 F] c_p (t_3^n - t_0^n) + \alpha_n F (T_1 - t_0). \qquad (44)$$

The system of equations (37)-(44) makes it possible to make a full thermal calculation of a generator whose hot junctions are heated by the combustion products of an organic fuel and the temperature differential on the junctions is created due to the blowing of a coolant in the direction from cold junctions to the hot through the permeable thermal elements.

The efficiency of such a generator in the most general case is calculated as the ratio of the obtained useful electrical power to the quantity of heat released during the combustion of fuel, i.e.,

$$\eta = \frac{N}{BQ_p^n}. \qquad (45)$$

With such a determination of efficiency of the device all possible heat losses are considered, including the heat loss with stack gases. If we take into account that the heat of stack gases after the generator can be utilized in some other heat-utilizing device or in the second stage of the generator with lower-temperature thermal elements, for example, as in thermal circuits shown in Figs. 5e and f, then the efficiency of the very process of conversion of heat to electrical energy in the device can be estimated by defining the efficiency of the thermionic converter as the ratio of useful electrical power to the heat which passed over the surface of hot junctions:

$$\eta_h = \frac{N}{\Pi_2 + \lambda k F \left(\frac{dT}{dy}\right)_{y=\delta}}. \qquad (46)$$

where Π_2 is the heat absorbed on hot junctions as a result of the Peltier effect.

Given below are some results of a theoretical analysis of operation of the considered generator, obtained with the aid of this method for its calculation. To analyze the effect of various factors on the energy characteristics of a generator its variance calculations were carried out on the computer "Promin'".

During variance calculations the following values were assigned which remained constant: average for the p - n-pair characteristic of the material of branches of the thermal elements $z=0.5\cdot 10^{-3}$ 1°K and $\lambda=0.01$ W/cm•°K; diameter of the capillaries $d=0.1$ cm and their number per unit surface of the battery $z_1=25/cm^2$, which determined the values of quantities $k_1=\frac{F_{пор}}{F}=0.196$ and $k=\frac{F-F_{пор}}{F}=0.804$; physical characteristics of the air coolant $\lambda_0=2.7\cdot 10^{-4}$ W/cm•°K, $c_{p0}=10^3$ W•s/kg•deg, $Pr_0=0.71$, $\mu_0=1.8\cdot 10^{-1}$ kg/cm•s; initial temperature of the air collant $t_0^H=300$°K; perimeter of the battery b=10 cm, and also the geometric dimensions of the channel for the passage of the coolant. It was assumed that the generator operated at maximum power, i.e., when m=1.

Natural gas with the calorific value $Q_p^H=0.485\cdot 10^8$ W•s/kg•deg and in the amount theoretically necessary for a complete combustion of air $L_0=16.76$ kg/kg was used as the fuel. The dependence of the initial temperature t_3^H of the combustion products on the coefficient of excess or deficiency of the oxidizer, as noted above, can be easily obtained under these conditions.

In accordance with the given methodology, when calculating the generator, the geometric dimensions of the battery and the temperature differential on thermojunctions (i.e., useful electrical power) are assigned with the known initial consumption and temperature of the coolant. In the process of calculation the distributions of temperatures of the coolant t_0, hot and cold junctions T_2 and T_1, and also the temperature distribution of the combustion products t_3 necessary for ensuring these conditions with the known heat-transfer coefficient α_1 from the hot side are determined along the length of the battery.

The determination of fuel consumption with a joint solution of equations (38) and (43) or with the aid of the dependence of the initial temperature of fuel combustion on the excess oxidant ratio assumes that the distribution of temperature t_3 of the heating gas along the length of battery, calculated by formula (43), will be satisfied under the calculating conditions.

Under the described conditions the necessary initial temperature t_3^H of the fuel combustion products can be ensured in the case of two different operating modes of the device: during the combustion of fuel at the entrance to the generator with a deficiency in the primary oxidizer and during the combustion with an excess of the oxidizer. Due to the blowing-in of part of the oxidizer through the permeable thermal elements along the length of the battery into the combustion zone, the initial coefficient of the excess (deficiency) of the oxidizer varies along the length.

When the generator operates with an excess of the oxidizer supplied at the entrance, as they move toward the exhaust the combustion products are diluted by the blown-in coolant, thereby lowering their temperature due not only to the removal of heat to the hot junctions of thermal elements, but also due to the physical mixing with a colder coolant.

In the case when the necessary initial temperature of the combustion products is ensured by the combustion of fuel with a deficiency in the initial oxidizer, the oxidizer fed through the capillaries contributes to the liberation of a supplementary amount of heat along the length due to the afterburning of the unburnt fuel before the first thermal elements.

Figure 21 shows the consumption of fuel B as a function of the mass velocity pv_w of the oxidizer blown through the permeable thermal elements of the generator 30 cm long whose junctions maintain the temperature differential - $\Delta T = 500°K$, with the height of branches of the thermal elements $\delta = 1$ cm. For the same conditions, Fig. 22 shows a change in the coefficient of excess (deficiency) β of the oxidizer along the length l of the battery. As can be seen from these figures the operating mode of the battery during the combustion of fuel with a deficiency in the primary air requires less fuel for obtaining the same useful electrical power than the operating mode with the initial oxidizer excess.

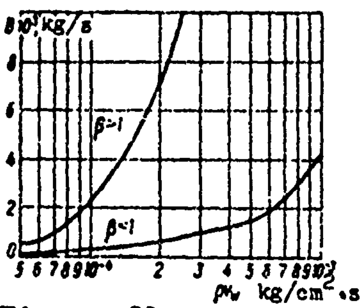

Figure 21

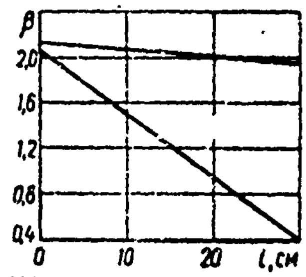

Figure 22

Figure 21. Consumption of fuel B as a function of the velocity pv_w at which the oxidizer is blown.

Figure 22. Change in the excess oxidant ratio β along the length l of the thermopile.

The situations noted above are easily explained by the fact that the operating mode requires a considerably greater consumption of the combustion products with the necessary initial temperature to maintain the necessary temperature conditions during the mixing-in of colder air along the length of the battery. This air blown through the permeable thermal elements is necessary to maintain the design temperature differential on the junctions.

When the generator operates with the combustion of fuel with a deficiency in the primary air ($\beta>1$) the decrease in temperature of the heat-transfer agent due to the mixing-in of a colder coolant (oxidizer) blown through the capillaries is compensated to a certain extent by supplementary heat liberation during the afterburning of the unburned fuel in the primary air. In this case the consumption of fuel, with all other conditions being equal, will be less in the operating mode when $\beta<1$. In this case the forward section of the battery serves as the burner device. With a decrease in the amount of the blown-in oxidizer this part increases and with consumption $\rho v_W=5\cdot10^{-4}$ kg/cm$^2\cdot$s it comprises two thirds of the length of the generator.

The efficiency of such a generator with an output of 75 W (constant temperature differential on junctions $\Delta T=500°K$) varies depending on the specific flow rate of the coolant blown through the permeable thermal elements according to the curves shown in Fig. 23. From this figure one can see that the efficiency in obtaining from such a generator the given useful electrical power increases in proportion to the decrease in the flow rate of the coolant blown through the thermal elements.

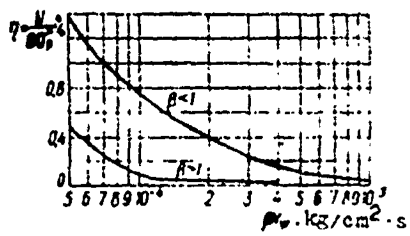

Figure 23. Efficiency η of thermopiles as a function of the flow rate ρv_W of the oxidizer blown through the capillaries.

Figure 24 shows a change in the temperatures t_0 of the coolant which washes the surface of the cold and hot junctions of thermal elements T_2 and the initial temperature t_3 of the combustion products which is necessary to ensure these, in section $x=l$, i.e., at the point where the combustion products enter the generator. From the figure one can see that with such a process of cooling the ratio of the substance blown through the capillaries to the entire amount of the coolant being supplied to the cold junctions is equal to 0.58), in the area of low flow rates of the blown-in substance the coolant is heated to considerable temperatures, which entails an increase in the operational temperature level of thermal elements and, consequently, also an increase in the initial temperature of the combustion products. However, in this case, the difference in temperatures between the hot junctions and the heating flow decreases. This situation can be easily explained by examining Fig. 25 which shows the ratio of the temperature gradients on the surfaces of hot and cold junctions of thermal elements as a function of the flow rate ρv_w of the coolant blown through them.

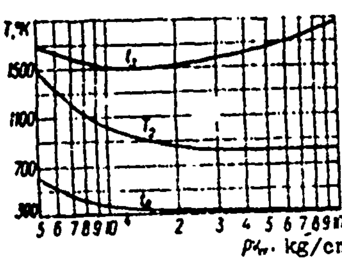

Figure 24

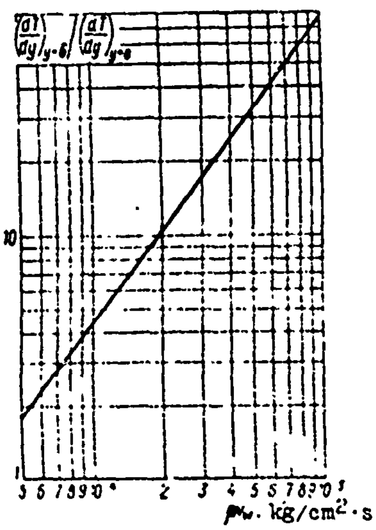

Figure 25

Figure 24. Change in temperatures of the heat-transfer agent, coolant and junctions depending on the oxidizer consumption ρv_w in section $x=l$.

Figure 25. The ratio of temperature gradients on the junctions $\left(\frac{dT}{dy}\right)_{y\to\delta}/\left(\frac{dT}{dy}\right)_{y=0}$ as a function of oxidizer consumption ρv_w.

At low velocities of blowing the ratio $\left(\frac{dT}{dy}\right)_{y=\delta}/\left(\frac{dT}{dy}\right)_{y=0}$ is small and increases sharply with an increase in the flow rate of the coolant blown through the capillaries. This fact, in turn, means that at low velocities of blowing considerably more heat is expended from the side of cold junctions for heating of the coolant flow than in the case of large ρv_w. In the limit, with an increase in ρv_w the amount of heat received as a result of thermal conductivity of the material of branches of the thermal elements on the surface of cold junctions and expended for heating of the coolant which washes the latter can approach zero. In this case the coolant flow will be heated basically due to ' Peltier's heat released on the surface of cold junctions, which is also small due to their low temperature. This heat causes a very intense cooling which is readily seen from Fig. 2?, where at high velocities of blowing-in of the coolant its temperature before the last thermal elements (counting from the entrance of the coolant flow) barely differs from the initial.

With an increased velocity of the blowing-in of the coolant the outflow of heat is sharply increased by thermal conductivity of the material of branches from the hot junctions of the thermal elements. This heat is expended basically for preheating of a large amount of coolant blown through the permeable thermal elements to a relatively high temperature. In this case, the initial temperature of combustion products also begins to increase with an increase in the velocity of the blown coolant.

Figure 26 shows a change in the coefficient of excess (deficiency) β of the oxidizer along the length l of the generator for specific oxidizer consumption, [illegible] through the capillaries and the temperature differential on junctions ΔT=800°K. This figure also confirms the advisability that the generator operate in the mode with the initial deficiency in the oxidizer. The dependences β shown in this figure ensure an identical operating temperature regime of the generator, but with considerably different fuel consumptions.

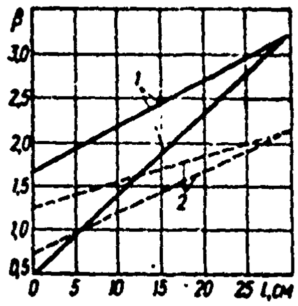

Figure 26. The effect of the flow rate of the blown-in oxidizer on the change in the excess oxidant ratio β.
1 - $\rho v_w = 10^{-3}$ kg/cm^2·s
2 - $\rho v_w = 10^{-4}$ kg/cm^2·s

The efficiency of the generator on the blowing-in velocity of the coolant, given in Fig. 23, was calculated from expression (45). As was already mentioned, this expression takes into account the decrease in the efficiency of the generator due to heat dissipation with stack gases.

The heat dissipation with stack gases is the determining factor both in generators with permeable thermal elements and in generators with monolithic thermal elements. When generators with permeable and monolithic thermal elements operate in an identical temperature interval and with an identical temperature differential on junctions, their efficiency, according to formula (45), will be identical under the condition of full regeneration of heat leaving the cold junctions in both devices.

An increase in the efficiency of the generator with permeable thermal elements, in comparison with the generator with impermeable thermal elements, can be achieved by creating a larger temperature differential on junctions (obtaining higher net power) as a result o. a more intense heat exchange with the blowing of the coolant through the capillaries. In this case the consumption of the coolant will be considerably less than in devices with monolithic thermal elements, even with an intense ribbing of the cold junctions of the latter.

The comparison of the operation of the generators with permeable and impermeable thermal elements, not allowing for heat dissipation with stack gases, causes definite difficulties due to the absence of precise boundaries into which both generators could be placed guaranteeing analogous conditions of their operation. Actually, the efficiency of generators with monolithic thermal elements is calculated from a known relationship [6]:

$$\eta = \frac{N}{\Pi_2 + \lambda F\left(\frac{dT}{dy}\right)_{y=h}} = \frac{N}{\Pi_1 + N + \lambda F\left(\frac{dT}{dy}\right)_{y=0}}, \qquad (47)$$

i.e., useful electrical power is referred to the amount of heat which entered the thermal elements from the side of hot junctions, or to the quantity of heat which reached the surface of cold junctions plus the net power. In the permeable thermal element these quantities of heat differ sharply from one another due to heat removal to the coolant inside the thermal elements. Therefore, this equality for the permeable thermal elements is not correct.

The values of temperature gradients on the surfaces of hot and cold junctions of the monolithic and permeable thermal elements are shown in Fig. 27 as a function of the temperature differential on junctions with the flow rate of the coolant through the permeable thermal element - 10^{-4} kg/cm^2·s. As seen from Fig. 27, the temperature gradients on the junctions of the permeable thermal element differ very sharply. The ratio of the temperature gradients on the hot and cold junctions for this permeable thermal element depends only on the flow rate of the coolant blown through the capillaries. An increase in the flow rate of the coolant blown through the capillaries causes a considerable increase in the difference of the values of the temperature gradients on the surfaces of hot and cold junctions of the permeable thermal element. Consequently, if the efficiency of the permeable thermal element is computed analogously with a monolithic thermal element (dashed curve) according to formula (47) then, when calculating using the amount of heat which

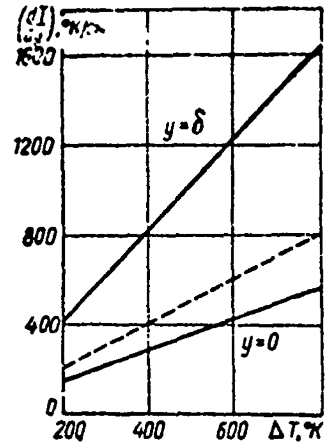

Figure 27. Values of temperature gradients on the junctions of monolithic and permeable thermal elements.

departed on the cold side, it will be considerably higher than when calculating using the amount of heat which entered the thermal element from the hot side. If one assumes that the heat removed by the coolant within the thermal element returns to the heating flow and can subsequently be used, the efficiency of the permeable thermal element which characterizes the perfection of conversion of the thermal energy to electrical can be computed by the formula (47). In this case the net power is referred to the amount of heat which departs from the side of cold junctions, plus the value of the removed power.

Figure 28 shows the efficiency of monolithic and permeable thermal elements as a function of the temperature differential on the junctions, calculated by formulas (47). The dependences shown in this figure have a purely theoretical interest. When comparing the operating efficiency generators with monolithic and permeable thermal elements, designed for some specific operating conditions, an individual approach is necessary taking into account all the peculiarities of the device being developed.

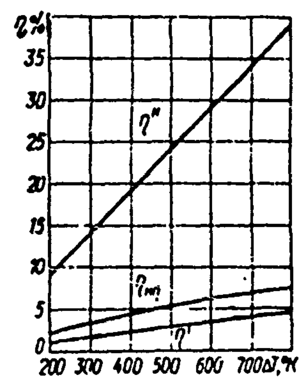

Figure 28. The effectiveness of energy conversion as a function of the temperature differential on thermojunctions.

2. CALCULATION OF A GENERATOR BLOWN BY A HEAT-TRANSFER AGENT IN THE DIRECTION FROM HOT JUNCTIONS TO THE COLD

The circuit of the thermal element which operates as a generator of electric power, blown by heat-transfer agent in the direction from hot junctions to the cold, is shown in Fig. 29 [13].

Hot heat-transfer agent with initial temperature t_3 is blown through the capillaries made in the branches of thermal elements of the generator. In this case the heat to the battery will be fed not only through the surfaces of hot junctions of the thermal elements, but also within the capillaries. Such a battery can be used, for example, for the utilization of heat of the combustion products produced in any heat-utilizing device. Naturally, under these conditions the flow rate of the hot heat-transfer agent and its initial temperature before the battery are assigned.

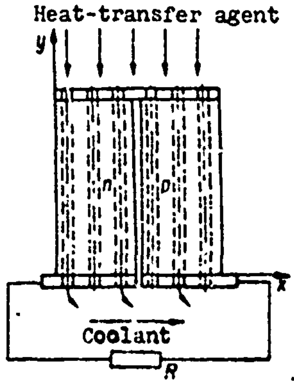

Figure 29. A diagram of a thermal element blown by the heat-transfer agent in the direction from hot junctions to the cold.

Let us assume that the battery is made from permeable thermal elements with identical geometrical characteristics and its dimensions are known. This means that with known averaged parameters of the materials of branches of the thermal element, its internal electrical resistance r is known. If we know the electrical load resistance R and the necessary net power which is quenched on it, then from relationship (6) it is possible to find the temperature differential which must be maintained on the junctions of this battery:

$$\Delta T = \sqrt{\frac{N\delta(m+1)^2}{z\lambda mblk}}. \quad (48)$$

This gradient is created due to cooling of the heat-transfer agent from the initial temperature t_3 before thermal elements to the temperature of the coolant flow t_0, i.e.,

$$\rho v_w c_{pr} k_1 bl(t_3 - t_0) = N + \Pi_1 + + \rho v_w c_{pr} k_1 bl(t_1 - t_0) + \lambda kbl\left(\frac{dT}{dy}\right)_{y=0}$$

or, after simplifying this expression, we will obtain

$$\rho v_w c_{pr} k_1 bl(t_3 - t_1) = N + \Pi_1 + \lambda kbl\left(\frac{dT}{dy}\right)_{y=0}, \quad (49)$$

where Π_1 - Peltier's heat released on the cold junctions of permeable thermal elements.

From the side of the hot junctions of permeable thermal elements the temperature of the hot heat-transfer agent decreases due to the absorption of the Peltier's heat on the surface of hot junctions also due to heat removal as a result of thermal conductivity of the material of branches of the thermal elements, i.e.,

$$\rho v_w c_{pr} k_1 bl(t_3 - t_2) = \frac{z\Delta T\lambda}{\delta(m+1)} T_2 kbl + \lambda kbl\left(\frac{dT}{dy}\right)_{y=\delta}. \quad (50)$$

From expressions (49), (50) it is not difficult to find the temperature of the heat-transfer agent at the entrance to the capillaries (when y=δ) by calculating the temperature gradient on the surface of hot junctions using equation (21):

$$\frac{\rho v_w c_{pr}}{\lambda} \cdot (t_3 - t_2) = \frac{z\Delta T^2}{\delta(m+1)} + \frac{z\Delta T}{\delta(m+1)} T_1 +$$
$$+ 2B \frac{\left(\frac{B}{A} - \frac{1}{2}\right)\exp A\delta}{M} \left\{(t_2 - T_1)\left(\frac{B}{A} - \frac{1}{2}\right) + \right.$$

$$+\Delta T+C\left[(1+A\delta)\left(\frac{B}{A}+\frac{1}{2}\right)-1\right]\Bigg\}+$$
$$+\left(B-\frac{A}{2}\right)(\Delta T+T_1-t_2-C)-AC.$$

From this expression we obtain

$$t_2=\frac{k_2}{k_3}T_1+\frac{k_4}{k_3}, \tag{51}$$

where

$$k_2=2B\frac{\left(\frac{B}{A}-\frac{1}{2}\right)^2\exp A\delta}{M}-\frac{z\Delta T}{\delta(m+1)}-\left(B-\frac{A}{2}\right),$$
$$k_3=\frac{\rho v_w c_{pr}\Pi}{\lambda}+2B\frac{\left(\frac{B}{A}-\frac{1}{2}\right)^2\exp A\delta}{M}-\left(B-\frac{A}{2}\right),$$
$$k_4=\frac{\rho v_w c_{pr}\Pi t_3}{\lambda}-\frac{z\Delta T^2}{\delta(m+1)}-$$
$$-2B\frac{\left(\frac{B}{A}-\frac{1}{2}\right)\exp A\delta}{M}\left\{\Delta T+C\left[(1+A\delta)\left(\frac{B}{A}+\frac{1}{2}\right)-1\right]\right\}-$$
$$-\left(B-\frac{A}{2}\right)(\Delta T-C)+AC,$$
$$M=\left(\frac{B}{A}+\frac{1}{2}\right)^2\exp\left[\left(B+\frac{A}{2}\right)\delta\right]-$$
$$-\left(\frac{B}{A}-\frac{1}{2}\right)^2\exp\left[-\left(B-\frac{A}{2}\right)\delta\right]-2\frac{B}{A}\exp A\delta.$$

The temperature t_1 of the heat-transfer agent at the outlet from the capillaries on the side of cold junctions can be determined from equation (22) when $y=0$. Disregarding the terms of the second order of smallness in this equation, we obtain the expression for the temperature of the heat-transfer agent on the surface of cold junctions:

$$t_1=T_1-C+\frac{\left(\frac{B}{A}-\frac{1}{2}\right)\exp\left(B+\frac{A}{2}\right)\delta}{M}\times$$
$$\times\left\{(t_2-T_1)\left(\frac{B}{A}-\frac{1}{2}\right)+\right.$$
$$\left.+\Delta T+C\left[(1+A\delta)\left(\frac{B}{A}+\frac{1}{2}\right)-1\right]\right\}. \tag{52}$$

It is most expedient to maintain a high temperature of hot junctions in the generators (see Fig. 29), which will permit a maximum use of the heat of hot heat-transfer agent, creating a highest temperature differential on the thermojunctions. Maximum value can be attained by the temperature of hot junctions with the equality of its temperature in the heat-transfer agent on the surface of hot junctions (when $y=\rho$), i.e., when $T_2=\Delta T+T_1=t_2$, or taking into account expression (51)

$$T_1=\frac{k_4-k_2\Delta T}{k_3-k_4}. \tag{53}$$

Substituting into this expression the values of coefficients k_2, k_3 and k_4, we obtain the equation for determining the temperature of cold junctions of this generator:

$$T_1=\frac{\begin{array}{c}\frac{\rho v_w c_{pr}\Pi}{\lambda}(t_3-\Delta T)-2B\frac{\left(\frac{B}{A}-\frac{1}{2}\right)\exp A\delta}{M}\times\\ \times\left\{2\Delta T+C\left[(1+A\delta)\left(\frac{B}{A}+\frac{1}{2}\right)-1\right]\right\}-\\ -\frac{z\Delta T^2}{\delta(m+1)}+C\left(B+\frac{A}{2}\right)\end{array}}{\frac{\rho v_w c_{pr}\Pi}{\lambda}-\frac{z\Delta T}{\delta(m+1)}}. \tag{54}$$

Thus, with given initial temperature of the heat-transfer agent t_3 and the temperature differential ΔT on the thermojunctions, using equations (51), (52) and (54) we can determine the temperatures of the hot T_2 and cold T_1 junctions of the battery and temperature of the heat-transfer agent at the entrance to the capillaries t_2 and at the exit from them t_1. After this it is necessary to calculate the cooling system, which will make it possible to maintain a given temperature differential on the thermojunctions in the calculated temperature interval.

From equation (49) it follows that from cold junctions of permeable thermal elements the external coolant flow removes the Peltier's heat liberated on their surface, the heat which reached the cold junctions as a result of thermal conductivity of the material of branches, and the physical heat of the heat-transfer agent which emerges on the surface of cold junctions at temperature t_1. These, then, are the three components of the heat flux from the surface of cold junctions of permeable thermal elements, which must be removed by the coolant flow which washes their surface from the outside.

The literature does not contain information on the heat-transfer coefficient from the hot permeable surface to the cold external flow in the presence of blowing through the permeable surface of a hot heat-transfer agent. However, if one assumes that the heat-transfer coefficient in this case is determined from the equations obtained for the case where a cold substance is blown into a hot heating flow, in particular from formula (7) then the amount of the removed heat in this case can be determined as follows:

$$\alpha_1 (T_1 - t_0) = \lambda k \left(\frac{dT}{dy} \right)_{y=0} + \frac{z \lambda \Delta T}{\delta (m+1)} T_1 k. \tag{55}$$

From equations (49) and (55) it is evident that in the determination of the heat-transfer coefficient using formula (7) it is necessary to consider also the increase in the temperature of the coolant due to the physical heat of the mixed-in heat-transfer agent which emerges on the surface of cold junctions of permeable thermal elements. If we know the initial expenditure of the coolant and its temperature as it enters thermal elements, then it is possible to write the following equation of heat balance:

$$\begin{aligned}(G_{\text{вх}} + \rho v_w k_1 bl) c_{p0} t_0^{\text{к}} - G_{\text{вх}} c_{p0} t_0^{\text{н}} = \rho v_w c_{pr} k_1 bl (t_1 - t_0) + \\ + \alpha_1 bl (T_1 - t_0),\end{aligned} \tag{56}$$

or taking into account expressions (44) and (55)

$$\begin{aligned}(G_{\text{вх}} + \rho v_w k_1 bl) c_{p0} t_0^{\text{к}} - G_{\text{вх}} c_{p0} t_0^{\text{н}} = \\ = \rho v_w c_{pr} k_1 bl (t_2 - t_0) - \frac{z \lambda m}{\delta (m+1)} \Delta T^2 kbl.\end{aligned}$$

Referring the initial expenditure of the coolant to the area of a generator and dividing both sides of equality (56) by $\rho v_w c_{pr} k_1 b l$, we obtain the expression for the final temperature of the coolant

$$t_0^{\kappa} = \frac{t_{\text{в}} + \left[\frac{\rho U_0}{\rho v_w k_1} \cdot \frac{c_{po}}{c_{pr}} - 0{,}5\right] t_0^{\text{н}} - \frac{\lambda}{\rho v_w c_{pr} \Pi} \cdot \frac{z \Delta T^2}{\delta (m+1)} \cdot \frac{m}{(m+1)}}{\left(\frac{\rho U_0}{\rho v_w k_1} + 1\right) \frac{c_{po}}{c_{pr}} + 0{,}5}. \qquad (57)$$

Thus, by knowing the final temperature of coolant, it is possible to determine its mean temperature along the length of the generator and, from equation (44), determine the heat-transfer coefficient for the calculation mode

$$\alpha_1 = \frac{\rho v_w c_{pr} k_1 (t_з - t_1) - \frac{z \Delta T}{\delta (m+1)} \cdot \frac{\lambda m}{(m+1)} k}{T_1 - t_0}. \qquad (58)$$

The heat-transfer coefficient value α_1 calculated from this expression can be actually ensured by selecting the necessary velocity of the main flow of the coolant, which can be determined from equation (7) or from other equations used for determining the heat-transfer coefficient in the case where the substance is blown through a permeable wall. In such a generator similar conditions of heat exchange on the surface of cold junctions can be realized by changing the cross section of the channel through which the coolant passes at its specified initial flow rate.

Consequently, the system of equations obtained makes it possible to accomplish a complete calculation of a generator with permeable thermal elements, blown by a heat-transfer agent in the direction from hot junctions to the cold.

The efficiency of such a generator is defined as the ratio of useful electrical power to the total heat expended:

$$\eta = \frac{N}{\rho v_w c_{pr} k_1 b l (t_з - t_0^{\text{н}})} = \frac{z \Delta T^2}{\delta (m+1)} \cdot \frac{\lambda}{\rho v_w c_{pr} \Pi} \cdot \frac{m}{(m+1)(t_з - t_0^{\kappa})}. \qquad (59)$$

This expression takes into account also the heat dissipation with stack gases.

To evaluate the efficiency of the conversion process of thermal energy to electrical power, we can use the expression which does not take into account the effect of heat dissipation with stack gases:

$$\eta' = \frac{N}{\rho v_w c_{pr} k_1 \delta l (t_2 - t_1)} = \frac{z\Delta T^2}{\delta (m+1)} \cdot \frac{\lambda}{\rho v_w c_{pr} \Pi} \cdot \frac{m}{(m+1)(t_2 - t_1)}. \qquad (60)$$

At present the comparison of the operating efficiency of generators with monolithic and permeable thermal elements is rather difficult due to the lack of sufficiently substantiated comparison methods.

If we attempt to compare these devices with the assumption that the batteries with identical geometric dimensions operate with an identical temperature differential on the junctions and, also, that the coolant flow rate and its temperature before batteries are identical, then the efficiency of the impermeable battery under the same conventional designations will have the following form:

$$\eta'_{\text{нп}} = \frac{z\Delta T^2}{\delta (m+1)} \cdot \frac{\lambda}{\rho v_w c_{pr} k_2} \cdot \frac{m}{(m+1)(t_3 - T_{2\text{нп}})}. \qquad (61)$$

This expression, just as in (60), does not take into account the heat dissipation with stack gases. Here $T_{2\text{нп}}$ - minimum possible temperature of the outgoing combustion products for the impermeable battery, which is equal to the temperature of hot junctions.

For the comparison of the efficiency of these devices, dividing expression (60) by (61) we obtain

$$\frac{\eta'}{\eta'_{\text{нп}}} = k \frac{t_3 - T_{2\text{нп}}}{t_3 - t_1}. \qquad (62)$$

The temperature of heating gases beyond the impermeable battery at the given condition of equality of temperatures of the hot junctions and heat-transfer agent at the output can be easily determined from the balance of heat on the hot side:

$$\rho v_w k_1 c_{pr} bl\,(t_3 - T_{2\text{кп}}) = \frac{z\lambda\Delta T}{\delta(m+1)} T_{2\text{кп}} bl + \\ + \frac{\lambda}{\delta} bl\Delta T - 0{,}5\frac{z\lambda\Delta T^2}{\delta(m+1)^2} bl. \tag{63}$$

Hence

$$T_{2\text{кп}} = \frac{t_3 - \frac{\lambda}{\rho v_w c_{pr} k_1}\cdot\frac{\Delta T}{\delta}\left[1 - 0.5\frac{z\Delta T}{(m+1)^2}\right]}{1 + \frac{\lambda}{\rho v_w k_1 c_{pr}}\cdot\frac{z\Delta T}{\delta(m+1)}}. \tag{64}$$

From equations (59) and (60) we can see that the efficiency of a permeable generator, taking into account the heat dissipation with stack gases, can be expressed in a following manner:

$$\eta = \frac{t_3 - t_1}{t_3 - t_0^{\text{н}}}\eta'. \tag{65}$$

For a generator with monolithic thermal elements the efficiency, by analogy with a generator with permeable thermal elements, it can be expressed as

$$\eta_{\text{кп}} = \frac{t_3 - T_{2\text{кп}}}{t_3 - t_0^{\text{н}}}\eta_{\text{кп}}' \tag{66}$$

(here the conventional designations are the same as for permeable thermal elements).

The efficiency ratio of generators with monolithic thermal elements and with permeable thermal elements, in accordance with equations (62), (65) and (66), has the form

$$\frac{\eta}{\eta_{\text{кп}}} = k \quad (k < 1). \tag{67}$$

Consequently, under these conditions of comparison, the battery with permeable thermal elements will be less effective, than that with monolithic thermal elements.

This position is valid only for these rather rigorous conditions, when identical temperature differential on thermojunctions is ensured in both cases by the hot and cold flows which have the identical initial and final parameters.

However, this is observed only to a certain value of ΔT. Actually, in batteries, both with permeable and impermeable thermal elements that operate in an identical temperature range of the heat-transfer agent ($t_3 - t_0$), the temperature differential on the junctions can be increased to a certain limit. An increase in the temperature differential on the thermojunctions leads to an increase in the Peltier's heat which is absorbed on the hot junctions, and an increase in the heat which is removed from them as a result of thermal conductivity of the material of the branches of thermal elements. In this case, if Peltier's heat which is absorbed on hot junctions is almost equal in both cases (it depends on the number of voids in the material of branches, i.e., on the porosity or capillarity), the quantities of heat removed as a result of thermal conductivity of the material of branches in monolithic and permeable thermal elements are sharply different.

In a battery with impermeable thermal elements the temperature profile differs insignificantly from the linear (see Chapter III), i.e., the value of the heat flux will increase proportionally to an increase in the temperature differential on the thermojunctions. In a battery with permeable thermal elements the temperature profile along the height of the thermal element is sharply curved and the temperature gradient on the surface of hot junctions with an identical with the impermeable thermal element temperature differential on the junctions will be considerably less. This difference in heat fluxes which enter the thermal element through the surface of hot junctions increases with an increase in the flow rate of the coolant blown through the permeable thermal element. As shown by the analysis of the expressions for the temperature distribution in the permeable wall, given in Chapter III, an increase in the temperature differential on the junctions result in a slower increase in the temperature gradient on the surface of junctions of the permeable thermal element in comparison with the monolithic thermal element. This determines, in accordance with the Fourier law, lesser heat fluxes from hot junctions in the permeable thermal elements.

The analysis of equations (50) and (63) makes it possible to reveal the redistribution of the effect of the absolute values of the Peltier's heat quantity and quantity $\lambda\left(\frac{dT}{dy}\right)_{y=\delta}$ for the reduction in temperature of the heat-transfer agent. Consequently, in impermeable thermal elements the heat removed as a result of thermal conductivity of the material of the branches of thermal elements is greater with respect to the absolute value than in the battery with permeable thermal elements, and the effect of increase in the temperature differential on junctions on the increase in the heat flux which leaves the surface of hot junctions, is also greater in the impermeable thermal elements than in permeable. This means that the maximum temperature, that can be achieved with all other conditions on the hot junctions of impermeable thermal elements with an increase ΔT being equal, will drop considerably faster as compared with permeable thermal elements.

Therefore, a battery with permeable thermal elements is capable of operating at considerably higher temperature differential on the junctions than that with monolithic thermal elements.

Thus, expression (67) is valid only up to a certain value of ΔT, attainable on monolithic thermal elements (the value of ΔT, maximally possible in this range, is $t_3-t_0^H$). With a further increase in the temperature differential on the junctions of blown thermal elements the efficiency of this device will be greater than the maximally possible efficiency of a battery with monolithic thermal elements.

Consequently, this procedure makes it possible to calculate all the thermal characteristics of a battery blown by a heat-transfer agent in the direction from hot junctions to the cold.

We shall analyze the battery which operates in this mode using the example of a specific device miscalculated in the various variants of change of many parameters. During the variance calculations it was assumed that the physical properties of the heat-transfer agent and the coolant flow were identical and equalled to $\mu=2.4\times10^{-7}$ kg/cm•s; $c_p=10^3$ W×s/kg•deg; $\lambda=3.8\cdot10^{-4}$ W/cm•deg, respectively. The average characteristics of the material of the branches of thermal elements - $z=0.5\times10^{-3}$ deg^{-1} and $\lambda=0.01$ W/cm•deg. The diameter of the capillaries was assumed to be equal to 0.1 cm and coefficients $k_1=F_{nop}/F=0.196$ and $k_2=F-F_{nop}/F=0.804$, which corresponds 25 capillaries for every 1 cm^2 of the battery surface. The ratio of load resistance to the internal resistance of the battery is assumed to be m=1, the temperature of the hot heat-transfer agent before battery $t_3=1500°K$ and the initial temperature of the coolant $t_0^H=300°K$.

Figure 30 shows the dependence of the temperatures of junctions, temperatures of the heat-transfer agent upon entering the capillaries and upon leaving them on the temperature differential on the junctions of permeable thermal elements calculated for the flow rate of the coolant $\rho v_w=10^{-4}$ kg/cm^2•s blown through the capillaries.

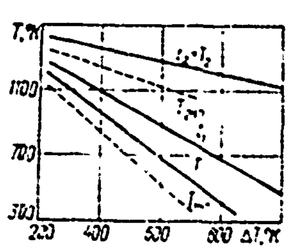

Figure 30. Temperatures of thermo-junctions and the blown heat-transfer agent as a function of the temperature differential maintained on the junctions.

Also shown here for comparison are the dependence of temperatures of the junctions of an impermeable thermal element (broken lines) which, at the same flow rate of the heating gas will be of the same initial temperature as for the permeable thermal element. From this figure one can see that with a permeable thermal element it is possible to achieve considerably higher temperature differential on the junctions than with a monolithic thermal element, with all other conditions being equal. In this instance the maximum possible useful electrical power removed from a generator with permeable thermal elements will be 1.2 times higher, as compared to a generator with monolithic thermal elements. In this case the weight of such a battery because of the capillaries will be almost 20% less than that of a battery with monolithic thermal elements.

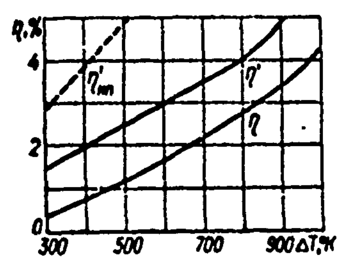

Figure 31. The effect of temperature differential on junctions as a function of the efficiency of the generator.

Figure 31 shows the dependence the efficiency of this battery on the temperature gradient on the thermojunctions. As seen from this figure and formula (62), the efficiency of a battery with impermeable thermal elements, without considering the heat dissipation with stack gases, at any temperature gradient on the junctions is higher than that of a battery with permeable thermal elements under similar operating conditions. This is easily explained by the fact that with identical ΔT on the thermojunctions (almost identical electrical power) the temperature gradient on the cold side of a permeable battery, and this means also the heat losses from the surface of cold junctions, is considerably higher than for a battery with monolithic thermal elements.

The interrelationship of the efficiency of batteries with permeable and monolithic thermal elements, calculated taking into account the heat dissipation with stack gases, as it follows from (67), is determined only by porosity with a selected comparison method. However, if we take thermopiles with different geometric dimensions, as in this case, but of the same weight, i.e., select a permeable thermopile with a greater number of voids, then the efficiency of both batteries, taking into account the heat loss with stack gases, will be identical at any flow rate of the coolant and the temperature differential on the junctions provided that the heat is totally regenerated in these circuits.

However, the battery with permeable thermal elements is capable of producing a greater amount of useful electrical power, than the battery of the same geometric dimensions with monolithic thermal elements.

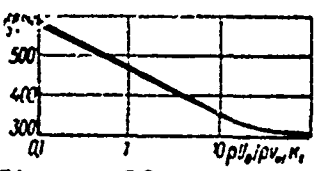

Figure 32

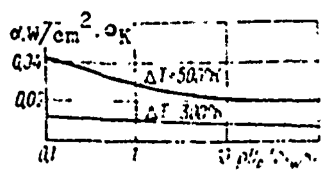

Figure 33

Figure 32. Heat-transfer coefficient α on the side of cold junctions as a function of $\rho U_0/\rho v_w k_1$.

Figure 33. The effect of the ratio $\rho U_0/\rho v_w k_1$ on the mean temperature of the coolant flow t_0^{cp}.

Figures 32 and 33 show the heat-transfer coefficient α on the side of cold junctions and the mean temperature of t_0^{cp} of the coolant flow along the length of the battery as a function of the ratio of the specific consumption of the coolant $\rho U_0/\rho v_w k_1$, supplied to the surface of cold junctions of a generator, and the heat-transfer agent leaving the permeable thermal elements on the surface of cold junctions. From these figures it is evident that with an increase in the ratio $\rho U_0/\rho v_w k_1$ at a fixed flow rate of the coolant the value of the necessary heat-transfer coefficient α decreases slightly, but in this case the mean temperature t_0^{cp} of the coolant flow decreases noticeably. This is significant, since the generator can operate at a greater temperature differential on the thermojunctions, which means greater output of useful electrical power.

3. CALCULATION OF A TWO-STAGE GENERATOR

The devices considered above are capable of operating both as independent generators as well as in the systems together with generators with monolithic thermal elements and with one another. As was already mentioned, such combined systems, in many instances, make it possible to more effectively convert the thermal and nuclear energy to electrical power. As an example let us examine two combined systems.

The operation of a two-stage generator with a thermal circuit shown in Fig. 5f, makes it possible, to a considerable degree, to eliminate the deficiencies inherent in each of the stages in their independent operation. In such a system each generator can operate in its own, optimum for it, mode. In this case both generators operate only with one hot and one cold heat source. The efficiency of any combined cycle, including that examined in [27], can be computed by the formula

$$\eta_\kappa = \eta_1 + \psi\eta_2(1 - \eta_1), \tag{68}$$

where η_1 and η_2 - efficiency of the first and second generators, respectively.

In this system one of the generators can be considered as an electro-generating device of the basic cycle, and another - as high- or low-temperature superstructure. Since values η_1 and

η_2 are low, the effect due to the combination of cycles of both generators is close to that due to the summation of these values, i.e., the efficiency of the generators working together, calculated by formula (68), will be almost twice as high as that of each of them operating independently. In this case the value of the coefficient ψ can be assumed to be equal to unity, since with a complete regeneration of heat in first stage, the same amount of heat will be delivered to the second stage of the generator as to the first.

In such a combined thermal cycle, not only the combustion products of an organic fuel, but also nuclear reactors and capsules with radioisotopes can be used as a hot source for the heating of hot junctions of thermal elements of the first stage. In these cases the coolant blown through permeable thermal elements of the first stage is heated in them, creating a temperature differential on the junctions, and then it is heated to a higher temperature in the clearance between the heating surface and hot junctions. Thermojunctions in this case are heated by the radiation of the heating surface and by convection of the substance which cools thermal elements in the clearance between the heating surface and hot junctions. Hot gas obtained thus is used as the heat-transfer agent in the second stage of a two-stage generator.

When a two-stage generator operates on the combustion products of an organic fuel, the thermal calculation of both stages can be accomplished using the methods presented in the chapter.

To evaluate the efficiency of the generator operating according to such thermal circuit, we will use the results of variance calculations for single-stage generators with permeable thermal elements given above. Let the useful electrical power of a two-stage generator be equal to 500 W, when using materials with identical properties in both stages, whose characteristics are given above. The optimum operational version of the two stages in this case will be the operating mode of a two-stage generator in which the first stage produces 200 W of useful electric power and the second - 300 W. The efficiencies of these generators, defined as the ratio of useful developed power to the product of the fuel expenditure for its calorific value, comprised 1.03% and 1.55% respectively. The total efficiency of a two-stage generator, computed thus, composed 2.58%. The main loss is due to heat dissipation with the physical heat of the combustion products which leave the device. Consequently, the efficiency of this device proved to be equal to the sum efficiency of both generators. The absolute value efficiency of a two-stage generator can be considerably greater when using a more qualitative material for the production of the branches of thermal elements with the figure of merit value $z>0.5\cdot10^{-3}$ deg^{-1}.

The joint operation of generators built according to the thermal circuits shown in Figs. 5c and b (diagram 5e) can be examined as another system of a compound generator. This generator can be used, for example, for cathode protection of main-line gas pipes.

Such a device consists of two stages and can be constructed in the form of walls of a duct or a rectangular channel. The first stage of the generator with monolithic thermal elements utilizes the heat of the combustion products which left the generator with permeable thermal elements.

The temperature differential on thermojunctions of the first stage is created by blowing of the ribbed or unribbed cold junctions by the gas from a gas line.

After cooling the cold junctions of monolithic thermal elements of the first stage the natural gas proceeds toward the cold junctions of permeable thermal elements of the second stage along channel 3 and, by being blown through them in the direction from cold junctions to the hot, creates on the junctions of permeable thermal elements the necessary temperature differential (Fig. 34). After this, the gas burns in the air upon reaching the surface of hot junctions of the second stage. The air necessary for combustion is sucked in by natural draft due to the difference in the specific gravity of the products of combustion and ambient medium. For this, the device is elevated above the ground by means of legs 4. Thus, hot junctions of the second stage located in the combustion zone can be heated to a high temperature. The sufficiently intense heat exchange, which occurs during the blowing of the gas through the pores (capillaries) of thermal elements, facilitates the obtaining of a large temperature differential on the junctions.

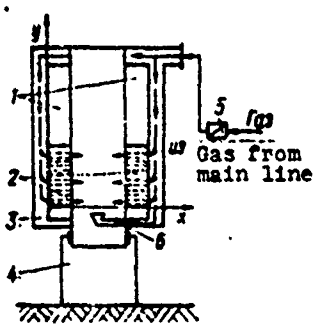

Figure 34. A diagram of a compound generator with permeable and monolithic thermal elements.

Combustion products give up part of their heat to the hot junctions of the second stage 2 of the generator, then enter the first stage 1 where they are cooled, releasing heat to the hot junctions of monolithic thermal elements. Convective cooling of the surface of cold junctions of the first stage makes it impossible to obtain the same high temperature differential on them as with internal cooling of the permeable thermal elements of the second stage. However, the first stage increases the energy efficiency of the generator by cooling the combustion products to lower temperatures, i.e., decreases the heat dissipation with stack gases.

The necessary flow rate and gas pressure which can be supplied directly from the main line are regulated by means of automatic adjuster 5. The flow rate of gas can vary depending on the value of the electric current or the temperature of hot junctions of the second stage, or on any other parameter.

It is expedient to realize the combustion process not just on the surface of hot junctions in the second stage. In view of the fact that the amount of heat for heating the hot junctions of first thermal elements can be insufficient, it is necessary to install burner 6 before the second stage for obtaining a sufficiently high temperature at the entrance.

In this device the heat dissipation from cold junctions can be reduced completely. The entire heat removed from the surfaces of cold junctions returns with the heated gas to the combustion zone.

The first and repetitive starts of this generator can be realized by means of an electric ignition device combined with burner 6. The igniter can be activated automatically by a temperature drop in the combustion products.

After deciding on the materials for the branches of thermal elements in the first and second stages, it is possible to select a temperature differential on the junctions, based on the fact of the most effective operating conditions of these materials. Optimum operating of the generator will be provided by the operation of all thermal elements at an invariable temperature differential on the junctions in each stage.

The coolant (natural gas) proceeds from the first stage to the second, which is blown partially $(\rho \dot{v}_{w} k_{1} \pi D l_{II})$ through the permeable thermal elements and burns up on the surface of hot junctions, and also partially ($G_г$) it bypasses the permeable thermal elements, entering the burner located at the entrance to the generator, i.e.,

$$G_0 = G_г + \rho v_w k_1 \pi D l_{II},$$

where l - length of the battery of the second stage; D - inside diameter of the generator.

For the burn-up of this amount of natural gas, G_B of air is supplied by natural draft in kg/s. The air excess ratio at the entrance to the generator will be $\beta = 1 + \frac{\rho v_w k_1 \pi D l_{II}}{G_г}$, since $\beta L_0 = \frac{G_B}{G_г}$. Consequently, with known flow rates of gas through the burner and permeable thermal elements the air excess ratio at the entrance is known. Therefore, one can easily determine the temperature of combustion products at the entrance, and further calculation of the stage with permeable thermal elements can be made according to the procedure given in paragraph 1 of this chapter.

The first stage with monolithic thermal elements is calculated with the known temperature differential on the junctions $\Delta T'$, flow rate of the coolant (natural gas) G_0 and its initial temperature $t_0^н$. Furthermore, from the calculation of the second stage we know the temperature of the coolant at the exit from the first stage $t_0^к$ (when $y = l_{II}$), the temperature of combustion products at their entrance to the first stage t_3 (when $y = l_{II}$) and its gradient at this point.

The balance of heat on the side of cold junctions of monolithic thermal elements can be written in the following form:

$$\alpha_0'(T_1' - t_0')\pi D dy = G_0 c_p dt_0'.$$

From this one can determine the temperature distribution of the cold junctions along the length of the first stage:

$$T_1' = \frac{G_0 c_p}{\alpha_0' \pi D} \cdot \frac{dt_0'}{dy} + t_0'. \qquad (69)$$

The balance of heat can be written otherwise.

$$G_0 c_p dt_0' = \lambda'\pi D dy \left(\frac{dT'}{dx}\right)_{x=0} + \frac{z'\lambda'}{\delta'(m+1)} \Delta T' T_1' \pi D dy. \tag{70}$$

Determining the temperature gradient on the surface of cold junctions by expression (10), from (70) we determine

$$T_1' = \frac{G_0 c_p \delta'(m+1)}{\pi D \lambda' z' \Delta T'} \cdot \frac{dt_0'}{dy} - \frac{1}{2} \cdot \frac{\Delta T'}{m+1} - \frac{m+1}{z'}. \tag{71}$$

After equating expressions (69) and (71) to each other and after the integration we obtain the temperature distribution of the coolant along the height of the first stage with monolithic thermal elements:

$$t_0' = \left(\frac{1}{2} \cdot \frac{\Delta T'}{m+1} + \frac{m+1}{z'} + t_0''\right) \exp \times$$
$$\times \left\{\frac{l_{II} - y}{\left[\frac{\alpha_0' \delta'(m+1)}{z'\lambda'\Delta T'} - 1\right] \frac{G_0 c_p}{\alpha_0' \pi D}}\right\} - \left(\frac{1}{2} \cdot \frac{\Delta T'}{m+1} + \frac{m+1}{z'}\right). \tag{72}$$

Knowing the distribution of t_0', it is possible to determine from (69) or (71) the temperature distribution of cold junctions T_1' along the length of the generator, and also the temperature distribution of the hot junctions T_2', since $T_2' = T_1' + \Delta T'$.

After this, we calculate the change in the temperature of combustion products along the length of the stage with monolithic thermal elements. Since the consumption of the products of combustion and the coolant in the impermeable stage are fixed and the temperature of the heating and coolant flows in section $y = l_{II}$ are known, the temperature differential which can be obtained on the junctions of monolithic thermal elements will be also completely defined. This gradient can be determined from the equation of heat balance of the stage with monolithic thermal elements:

$$\alpha_1' (t_3' - T_2') = \alpha_0' (T_1' - t_0') + \frac{z'\lambda' m}{\delta'(m+1)^2} (\Delta T')^2. \tag{73}$$

After substituting the temperature values of junctions, from equation (69) we obtain

$$(\Delta T')^2 + \frac{\alpha_1'\delta'}{\lambda'} \cdot \frac{(m+1)^2}{z'm} \Delta T' +$$
$$+ \frac{\alpha_1'\delta'}{\lambda'} \left[\frac{G_0 c_p}{\alpha_0'} \cdot \frac{(m+1)^2}{z'm\pi D} \left(\frac{\alpha_0'}{\alpha_1'} + 1 \right) \frac{dt_0'}{dy} + \right.$$
$$\left. + \frac{(m+1)^2}{z'm} (t_0' - t_3') \right] = 0. \qquad (74)$$

In this equation of the temperature values t_0' and t_3', and also the temperature gradient $\frac{dt_0'}{dy}$, are assumed to be the following for section $y = l_{II}$ after we calculate the stage with permeable thermal elements:

$$t_3' = t_3|_{y=l_{II}}; \quad t_0' = t_0|_{y=l_{II}}; \quad \frac{dt_0'}{dy} = \left(\frac{dt_0}{dy}\right)_{y=l_{II}}.$$

The temperature distribution of the heating gas can be determined from the equation of heat balance on the side of hot junctions of the monolithic thermal elements:

$$G_0(L_0 + 1) c_p dt_3' = \alpha_1' \pi D dy (t_3' - T_2').$$

Hence

$$\int_{l_{II}}^{l} dy = \int_{t_3''}^{t_3'} \frac{dt_3'}{\frac{\alpha_1' \pi D}{G_0 (L_0 + 1) c_p} (t_3' - T_2')}. \qquad (75)$$

From the calculations according to the given equations it follows that the instantaneous temperature value of the hot junctions is used in expression (75), which is determined from expression (6?) taking into account that $T_2' = T_1' + \Delta T'$.

Thus, using this method it is possible to carry out a complete thermal calculation of a compound generator in which both the permeable and monolithic thermal elements are used.

The total efficiency of the compound device can be determined by the relation

$$\eta = \frac{N_{I} + N_{II}}{G_{\text{т}} Q_{\text{р}}^{\text{н}}}, \qquad (76)$$

This notation of the efficiency takes into account also the heat dissipation with stack gases.

CHAPTER V

THERMOELECTRIC COOLERS WITH PERMEABLE THERMAL ELEMENTS

1. A METHOD FOR CALCULATING A COOLER

As was already mentioned, permeable thermal elements used for the construction of thermoelectric devices differ from the monolithic. Due to the developed internal heat-transfer surface in permeable thermal elements the heat transfer between the liquid (gas) blown through them and the solid material of branches occurs with small temperature differences, i.e., it is almost thermodynamically reversible.

Such a system of heat exchange establishes fundamental and essential peculiarities in the operation of the thermal element, offering a possibility to affect the cooling coefficient of the cooler or conditioner.

Let us analyze the operation of these thermal elements based on the examination of a temperature curve in the material of electrodes [12].

The nature of such a curve will vary depending on the given operation conditions of the thermal element. Figure 35 shows two temperature curves T=f(y) constructed for a cooling thermal

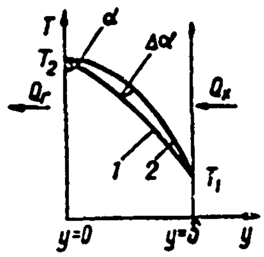

Figure 35. The nature of temperature curves in the non-blown and blown thermal elements.

element with the assumption that, both for the permeable thermal element and monolithic element, the temperatures of hot and cold junctions are equal to one another and, therefore, thermoelectric effects are also identical. The thermal element is blown by the cooling substance which is cooled by moving from hot junction to the cold.

Curve 1 which pertains to the monolithic thermal element is convex upwards due to the fact that the liberation of Joules heat causes an increase in the amount of heat along the y axis, which moves towards the cold junction, and also in the temperature gradient. The cooling coefficient ε and the refrigerating capacity Q_x are expressed in the following form:

$$\varepsilon = \frac{Q_x}{W}, \quad Q_x = Q_{п.г} - \frac{\lambda}{\mathrm{tg}\,\alpha} - W, \qquad (77)$$

where $Q_{п.г}$ - Peltier's heat released per unit surface of the hot junction: $\mathrm{tg}\alpha = \frac{1}{\left(\frac{dt}{dy}\right)_{y=0}}$, W - power expended.

Proceeding to thermal elements with internal heat exchange, curve 2 is obtained whose location relative to curve 1 can be characterized by an increase in the angle α by the value Δα. The additional deflection of curve 2 is due to the same reason, as the convexity of curve 1, i.e., by an increase in the amount of heat transferred to the cold junction along the y axis due to thermal conductivity. This, in turn, is connected with the internal heat supply from the cooled coolant. Thus, for the permeable thermal element $\varepsilon'=\frac{Q}{W}$, where Q is the total refrigerating capacity of the thermal element, which consists of the heat supplied due to the internal heat transfer from the cooled coolant $\rho v_g c_p (t_r - t_x)$, and the heat Q_x supplied from without to the cold junction:

$$Q=\rho v_g c_p (t_r - t_x) + Q'_x = Q_{a.r} - \frac{1}{\operatorname{tg}(\alpha+\Delta\alpha)} - W. \qquad (78)$$

Value Q'_x can be as small as desired or equal to zero. Expression (78) differs from (77) only by an increase in the angle α. Consequently, under identical conditions $\varepsilon'>\varepsilon$ always.

Here, besides the identity of the material properties, the equality of the passed currents and temperatures of the hot and cold junctions of monolithic and permeable thermal elements is implied under the equality of conditions. We also keep the sight of the fact that the difference in temperatures of the coolant being filtered and the material from which the permeable thermal element is made is negligible. The nonfulfillment of this condition cannot lead to other results qualitatively, since the value $\Delta\alpha$ is always positive.

The value of the angle $\Delta\alpha$ yields to control by changing the amount of the blown coolant, and it also depends on the temperature conditions and the design characteristics of the thermal element.

From formula (77) it is evident that with very large differences between the temperatures on the junctions, in the case of very low thermal elements when the angle α becomes very small, conditions in which Q_x=0 are possible. In other words, with large temperature differentials on the junctions (with very low branches), the cooling thermal element does not work. When using the finely porous or perforated thermal elements such conditions can be removed considerably, since, even with low values of the angle α, it is possible to retain a considerable magnitude $\alpha+\Delta\alpha$, having provided for this such a flow rate of the cooled substance which is necessary for obtaining the necessary value of $\Delta\alpha$.

The qualitative analysis given above for the operation of the monolithic and permeable cooling thermal elements shows the advantages of the latter based on energy characteristics. Therefore, it is expedient to analyze the operation of the permeable thermal element quantitatively and in detail.

The thermal element of the cooling thermoelectric device is shown schematically in Fig. 36. The thermal element branches are furnished with capillaries for the passage of the cooled substance which is blown in the direction from hot junctions to the cold, releasing heat to the material of the branches of thermal elements. Depending on the conditions under which the cooling battery with permeable thermal elements is used the supplying of the cooled substance to the capillaries can be realized in two ways. First, when the cooled substance circulates in a closed loop, its supply to the capillaries of the branches of thermal elements can be achieved through the channels made in the commutation plates of hot junctions (Fig. 36a). In this case the heat from the surface of hot junctions can be removed by any heat-transfer agent or by natural convection with sufficiently developed ribbing.

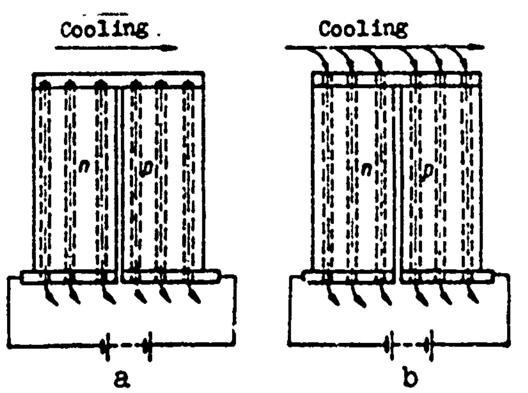

Figure 36. Circuits of permeable thermal elements of coolers.

In the case when the cooling battery with permeable thermal elements operates in an open circuit, the removal of heat from the hot junctions can be realized by the flow of the substance a portion of which is sucked through the capillaries of thermal elements after being cooled in them (Fig. 36b).

When the electric current is supplied, a difference in temperatures is created as a result of the Peltier effects:

$$W = I^2R + (T_2 - T_1)eI, \qquad (79)$$

where I - current passing through the thermal element; e - thermal electromotive force coefficient.

In this case the temperature profile along the height of the permeable electrode will be curved (see Fig. 35) and the temperature gradient on the hot side will be lower in comparison with the cold. Consequently, the overflow of heat due to thermal conductivity from the hot junctions, with all other conditions being equal, will be less than the overflow of heat in the impermeable thermal element.

If we assume that $Q'_x=0$, i.e., there is no heat supply to the surface of cold junctions, then, as a result of the Peltier effect on the cold junctions, only the heat which reached the cold junctions as a result of thermal conductivity of the material of the branches of thermal elements and the cooled substance is absorbed:

$$eIT_1 = \lambda(F - F_{\text{пор}})\left(\frac{dT}{dy}\right)_{y=0} + \lambda_v F_{\text{пор}}\left(\frac{dt}{dy}\right)_{y=0}. \qquad (80)$$

With the cooling of gases the heat arrived to the cold junctions by means of thermal conductivity of a gas, is negligibly small and the augend in (80) is disregarded in the future.

In the impermeable thermal element, as a result of the Peltier effect, absorbed on cold junctions is not only the heat which arrived there as a result of thermal conductivity, but also the heat removed from the cooled substance. Consequently, despite the fact that the temperature gradient on the cold side of the impermeable thermal element is less, the current strength necessary to maintain the same difference in the temperatures on the junctions will be greater with the same refrigerating capacity, which means that the applied power will also be greater.

The heat supplied to the thermal elements from the cooled substance and the applied power is removed from the side of hot junctions (value Q'_X is disregarded):

$$Q = W + \rho v_{\text{ш}} c_p F_{\text{пор}} (t_2 - t_1). \tag{81}$$

The area of application of the permeable thermal elements is limited by the need to operate at very small differences between the temperature of hot junctions, which must be slightly higher than the ambient temperature, and the temperature of the coolant which enters through this junction for cooling. Actually, if the coolant has a higher temperature in comparison with the hot junction, then it can be precooled to this temperature by a thermal contact with the ambient medium, without wasting the electric power. If the temperature of the coolant which returned for cooling is lower than the temperature of hot junctions and is equal to t_0, then upon entering onto the thermal element the coolant must be heated to the temperature close to that of the hot junctions, and then be cooled to a specific temperature. In this case, naturally, it would be necessary to develop an excess of the refrigerating capacity and the ratio of useful refrigerating capacity to total capacity in this case is equal to the coefficient $\frac{\rho v_{\text{ш}} c_p (t_0 - t_1)}{\rho v_{\text{ш}} c_p (t_2 - t_1) + Q'_x}$, which with $Q'_X=0$ transforms to $\frac{t_0 - t_1}{t_2 - t_1}$. In contrast to total refrigerating capacity and the cooling coefficient calculated according to it, the coefficient value decreases with an increase of t_1.

The value of the current necessary to maintain a specific difference in temperatures on the junctions, with the blowing of a certain amount of the cooled substance, can be determined from (80):

$$I = \frac{zT_1 \alpha \pi d z_1}{2Ke} - \sqrt{\left(\frac{zT_1 \alpha \pi d z_1}{2Ke}\right)^2 - \frac{\left(B - \frac{A}{2}\right)\left(\frac{B}{A} + \frac{1}{2}\right)^2 \alpha \pi d z_1 (F - F_{\text{пор}}) \exp\left[\left(B - \frac{A}{2}\right)\delta\right]}{NK\rho}(t_2 - T_1)}, \qquad (82)$$

where

$$A = \frac{4\alpha}{\rho v_w c_p d}; \quad B = \sqrt{\left(\frac{A}{2}\right)^2 + \frac{\alpha \pi d z_1}{\lambda (F - F_{\text{пор}})}},$$

$$K = \frac{\left(B - \frac{A}{2}\right)\left(\frac{B}{A} + \frac{1}{2}\right)\exp\left(B + \frac{A}{2}\right)\delta}{N} \times$$

$$\times \left[(1 + A\delta)\left(\frac{B}{A} + \frac{1}{2}\right) - 1\right] - A,$$

$$N = \left(\frac{B}{A} + \frac{1}{2}\right)^2 \exp\left(B + \frac{A}{2}\right)\delta -$$

$$-\left(\frac{B}{A} - \frac{1}{2}\right)^2 \exp\left(\frac{A}{2} - B\right)\delta - 2\frac{B}{A}\exp A\delta.$$

The temperature profile necessary for the calculation of the temperature gradient in the material of branches of the thermal element is taken from expression (21) with $t_2 = T_2$.

If the temperature of the cooled substance before the battery and the temperature differential on the junctions are given, then, after determining the current strength, it is possible to determine the temperature of the cooled gas (22) when $y=0$:

$$t_1 = T_1 - C + \frac{\left(\frac{B}{A} - \frac{1}{2}\right)\exp\left(B + \frac{A}{2}\right)\delta}{N} \times$$

$$\times \left\{\left(\frac{B}{A} + \frac{1}{2}\right)[t_2 - T_1 + C(1 + A\delta)] - C\right\}, \qquad (83)$$

where

$$C = \frac{I^2R}{\lambda a \pi d z_1}.$$

However, if in the calculation we know the initial and final temperatures of the cooled substance, then the necessary current and the temperature of the cold junctions can be determined by a joint solution of equations (82) and (83).

Knowing the current strength, from equation (79) it is possible to determine the applied electrical power, and from (81) - the amount of heat which must be removed from the hot junctions.

The effectiveness of such a battery (cooling coefficient) is defined as the ratio of the amount of heat removed from the cooled substance to the applied electrical power: $\varepsilon = \frac{\rho v_w c_p F_{\text{пор}} (t_2 - t_1)}{W}$.

2. ANALYSIS OF THE COOLER OPERATION

Let us examine a thermal element with a surface of 1 cm^2 whose branches are made from solid solutions Bi_2Te_3-Bi_2Se_3 (n-type) and Bi_2Te_3-Sb_2Te_3 (p-type). For both branches the average characteristic of the thermal-element material are the following: λ=0.013 W/cm·°K; e=350 μV/°K; σ=1150 $\Omega^{-1}\cdot cm^{-1}$. The figure of merit of the material, taking into account the imperfection of commutation is $z = 2.25\cdot 10^{-3}\,{}^{\circ}K^{-1}$. Air whose temperature before the thermal element was assumed to be equal to 323°K was examined as the cooled substance.

Figure 37 shows the dependence of the cooling coefficient ε on the velocity ρv_W of the blown air for the thermal elements of different height which have 25 capillaries 0.1 cm in diameter at the temperature of the cold junctions of 290°K. As seen from the figure the value of the cooling coefficient ε changes with a change in the flow rate according to the curve with a maximum, where with an increase in the height of thermal elements this maximum shifts to the side of lower flow rates of air. The position of maximums of the efficiency of thermal elements with varied height on a single straight line makes it possible to select the height of the thermal element with a given flow rate of the cooled air without any difficulty.

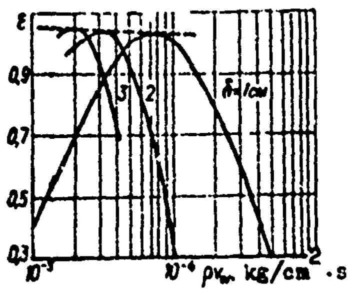

Figure 37

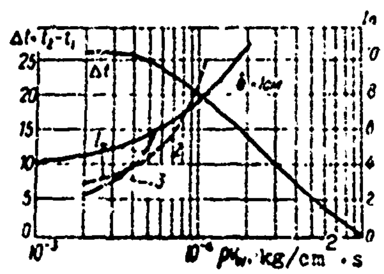

Figure 38

Figure 37. The efficiency ε of a permeable cooler as a function the specific consumption ρv_w of the cooled substance.

Figure 38. The depth of cooling of the air and current I as a function the consumption ρv_w of the cooled substance.

The degree of air cooling in the thermal element as a function of the flow rate is shown in Fig. 38. It turns out that with an identical flow rate the air blown through the capillaries is cooled to the same level in thermal elements of any height (with a spread that is within the accuracy limits of calculations). This means that in the thermal element of any height the same amount of heat will be removed from the cooled air provided all other conditions are equal. However, the efficiency of this heat removal will differ considerably with different δ (see Fig. 37).

The nature of a change in the feeding current strength I with a change in the flow rate of the cooled air in thermal elements of different height is also shown in Fig. 38. An increase in the current with a decrease in the height is explained by the increase in the temperature gradient on the cold junctions of the thermal element (Fig. 39) and, therefore, also by the increased influx of heat for the removal of which an increase in the Peltier effect is required.

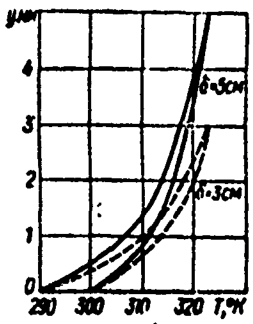

Figure 39. The effect of height of the thermal element y on the nature of temperature profiles when $\rho v_w = 10^{-4}$ kg/cm^2·s.

In the known circuits of thermoelectric coolers with monolithic thermal elements the necessary heat removal from the cooled medium is realized due to the Peltier effect on the cold junctions $\Pi = eIT_1$. With high values of heat removal the necessary current is rather considerable which, in turn, causes a large internal heat release as a result of the Joule effect. An increase in the Joule heat with an increase in current strength (when it is necessary to increase the heat removal from the cooled medium) contributes to the decrease in the cooling effect. Finally, the Joule heat can exceed Peltier's heat and the cooling of the junction will pass into its heating [6]. In the case of a circuit with permeable thermal elements the heat removal from the cooled medium is realized not only on the surface of the junctions, but also within the thermal element - along its entire height. The cold junctions receive only the heat due to thermal conductivity of the materials of the branches. Therefore, the current necessary to maintain the specific difference between the temperatures on the junctions, calculated from (82), will be smaller than in the usually considered devices with the same difference between the temperatures on the junctions. Consequently, Joule heat release has a different effect on the cooling capacity of the cooler (Fig. 40).

As can be seen from this figure, in the range of high flow rates of the cooled medium, an increase in height of the thermal

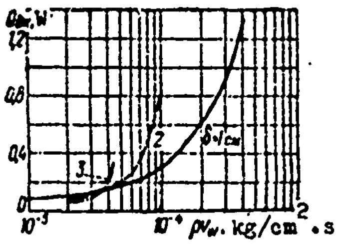

Figure 40. The value of Joule heat $Q_{дж}$ as a function of consumption of the cooled substance ρv_w.

element causes a rather considerable increase in the Joule heat release. However, in the range of low flow rates a completely opposite picture is observed - the greater the δ, the smaller the $Q_{дж}$. In the range of low flow rates the current strength increases less sharply with an increase in the flow rate of the cooled substance, and a lesser amount of Joule heat is released in the thermal elements of greater height. This parasitic heat release has a considerable effect also on the efficiency of the cooler (see Figs. 37 and 40). Thus, of the three thermal elements considered having different height, thermal element 3 cm in height is most effective in the consumption range up to $2.5 \cdot 10^{-5}$, 2 cm in height - in the range of $2.5 \cdot 10^{-5}$–$6 \cdot 10^{-5}$, for higher expenditures - thermal element 1 cm in height.

A comparison of this thermal element with a monolithic, carried out with identical cooling capacities, shows that the cooling coefficient of the permeable thermal element is 1.3-1.5 times higher than that of the impermeable. It is necessary to note that these advantages were obtained for a thermal element that was considerably lighter, since there were no ribs on the cold side and the weight of the thermal element was decreased because of the capillaries.

The effect of change in the temperature differential on the junctions on the characteristics of blown and nonblown thermal elements was examined during their parallel calculation. The calculation results for thermal elements 1 cm in height with an area of 1 cm^2 are given in Table 3.

The temperature of hot junctions in both cases is equal to 323°K. The blown thermal element had 25 capillaries 0.1 cm in diameter.

Table 3

ΔT, deg	Parameter	Thermal element not blown		Blown thermal element	
		ε_{max}	Q_{max}	ε_{max}	Q_{max}
33	ε	0,718	0,292	1,01	0,4
	Q, W.	0,33	0,8	0,31	0,6
	I, A	9,2	24,2	6,2	15,8
	W, W.	0,46	2,74	0,298	1,5
53	ε	0,216	0,148	0,3	—
	Q, W	0,269	0,379	0,31	—
	I, A	15,18	22,6	12,33	—
	W, W	1,245	2,56	1,032	—

As follows from the table, the blown thermal element has advantages over the nonblown thermal element with respect to energy characteristics.

These advantages make it possible to believe that permeable thermal elements will be utilized effectively for the air conditioning, deep cooling of gases and liquids in various devices, etc. However, a complex analysis of the perforated batteries with permeable thermal elements, which operate in specific circuits, requires further investigations.

CONCLUSION

The thermoelectric devices with permeable thermal elements examined in this work, in certain cases, can be more profitable than the similar devices with monolithic thermal elements. Much depends on the operating conditions and the purpose for which these devices are designed. Therefore, the comparison of the efficiency and the determination of the advisability of using the monolithic or permeable thermal elements should be accomplished separately for each specific circuit of the device used under certain specific conditions.

The book presents the bases of theoretical calculations for the different thermoelectric devices with permeable thermal elements and the analysis of their operation in order to draw the attention of specialists to the advisability of their further more detailed study.

For a practical realization of such thermoelectric units it is necessary to solve another series of questions. Specifically, the problems of electrical commutation of permeable thermal elements and also the technology of their manufacture from the various semiconductor thermoelectric materials, made both as finely porous and perforated, have not been resolved sufficiently.

Works undertaken in these directions at the Institute of Problems of the Study of Materials, AN USSR, which are encouraging, but which are rather narrowly directed toward the creation of thermoelectric modules for a high-temperature thermoelectric generator.

The selection of the material in the manufacture of permeable thermopiles is complicated by the fact that, virtually in all the circuits considered, they operate in an oxidizing medium. In view of this fact it is advisable at present to examine only the silicides of certain metals that are stable under the operating conditions of an oxidizing medium.

The experiments carried out at the Institute of Technical Thermophysics of the AN USSR showed the promise in the use of metallic thermocouples for the manufacture of permeable thermopiles. The technical-economical indices of the thermoelectric generator with permeable thermopiles, developed and tested here, are not inferior to the same indices of the devices used for the cathode protection of gas lines and the supply of low-power radio equipment.

A more thorough comprehensive thermodynamic analysis of thermoelectric devices with permeable thermal elements will unavoidably lead to the creation of even more economical and more ideal thermal circuits. Specifically, it is advisable to develop multistage thermoelectric coolers with permeable thermal elements for cooling of the flow of a substance. As shown by the rough estimates, the efficiency of such flow coolers will be considerably higher than that of the devices designed for a similar purpose with monolithic thermal elements, realized at present.

Thus, along with perfecting of devices with monolithic thermal elements and seeking new higher-quality thermoelectric materials, it is necessary to expand the works in further investigation and creation of thermoelectric devices with permeable thermal elements.

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